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Yaw moment control using an active differential and Electronic Stability Control system (ESC)

Master's Thesis in Automotive Engineering

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Division of Vehicle Engineering and Autonomous Systems
Vehicle Dynamics
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
Saab 9-3 Turbo X equipped with an eLSD and ESC.

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ABSTRACT

This thesis aims to compare the potential in handling and performance between a brake based Electronic Stability Control system (ESC) with a system that integrates an active differential with ESC. Two control strategies that integrate an active differential and ESC are presented for two types of active differentials. The differentials are the Electronic Limited Slip Differential (eLSD) and the Direction Sensitive Locking Differential (DSLD).

The control strategies are developed and the results are evaluated using a simulation model that is implemented in Simulink. They tested for one driving case, the open loop sine with dwell maneuver. The different solutions are evaluated mainly with regard to three criteria that are specified in the American National Highway Traffic Safety Administration FMVSS 126 law requirement (NTHSA, 2007). They are also evaluated with respect to the yaw rate vs. steering wheel angle response, vehicle speed during the maneuver and vehicle trajectory.

All tests were performed without any time delays simulated for any actuator. The results show only a small difference between a normal ESC and the integrated solutions. The integrated system with an eLSD however does perform better than ESC overall but by a small margin.

Because of the small difference in results from the first simulations time delays are implemented for a second round of simulations that only compare the best integrated solution (ESC + eLSD) to only ESC. This is done to evaluate how much of an effect the time delays have on the end results. After the second round of simulations the results do indeed differ more in favor of the integrated system with the eLSD compared to only ESC. The yaw rate respond better to steering wheel inputs with smaller overshoots and more speed is conserved through the maneuver. This shows that the simulation model might not be complete enough to evaluate the performance difference between those systems and implementing time delays more extensively for all actuators is a recommendation for future work.

Key words: Active differentials, Electronic Stability Control system (ESC), Active vehicle system integration, Vehicle dynamics, Vehicle stability, Yaw moment control

Reglering av girmoment med aktiv differential och
elektroniskt stabilitetssystem (ESC)
Examensarbete inom Automotive Engineering

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SAMMANFATTNING

Det här examensarbetet jämför den potentiella vinsten i väghållning och prestanda mellan ett bromsbaserat elektroniskt stabilitetssystem (ESC) och ett system som integrerar ESC med en aktiv differential. Två typer av aktiva differentialer undersöks; Electronic Limited Slip Differential (eLSD) och Direction Sensitive Locking Differential (DSLD) och två reglerstrategier som integrerar ESC med eLSD respektive DSLD.

Reglerstrategierna utvärderas med hjälp av en simuleringsmodell av en bil som är implementerad i Simulink. De är endast testade för ett körfall, det så kallade Sine With Dwell-provet. De olika lösningarna är utvärderade främst med avseende på tre kriterier som specificeras för det amerikanska lagkravet National Highway Traffic Safety Administration FMVSS 126 (NHTSA, 2007). De utvärderas också med avseende på girhastighetens respons mot styrvinklen, fordonshastigheten genom manövern och fordonets färdbana.

Alla simuleringar är utförda utan modellerade tidsfördröjningar för aktuatorerna. Resultaten visar bara små skillnader mellan ett normalt ESC-system och de integrerade lösningarna. Den integrerade lösningen med en eLSD presterar dock bättre jämfört med ESC, dock endast marginellt.

På grund av de små skillnaderna i resultaten från de första simuleringarna så implementeras tidsfördröjningar för en andra runda av simuleringar som endast jämför den bästa integrerade lösningen (ESC + eLSD) med ESC. Det görs för att undersöka hur mycket tidsfördröjningar påverkar slutresutaten. Efter den andra rundan av simuleringar så är skillnaderna större till fördel för den integrerade strategin med eLSD jämfört med bara ESC. Girhastigheten svarar bättre mot styrvinkeln med mindre översläng och hastigheten bibehålls genom manövern. Det här visar att simuleringsmodellen kanske inte är tillräckligt komplett för att utvärdera prestandaskillnaderna mellan dessa system och att implementera tidsfördröjningar mer utförligt för alla aktuatorer är en rekommendation för framtida arbete.

Nyckelord: Aktiv differential, Elektroniskt stabilitetssystem (ESC), Integration av aktiva fordonssystem, Fordonsdynamik, Fordonsstabilitet, Reglering av girmoment

Contents

Abbreviations

ABS	Anti-lock Braking System
AD	Active Differential
BoS	Beginning of Steer
CoS	Completion of Steer
DSLD	Direction Sensing Locking Differential
eLSD	Electronic Limited Slip Differential
ESC	Electronic Stability Control
NTHSA	National Highway Traffic Safety Administration
SWA	Steering Wheel Angle
YMC	Yaw Moment Control

1 Introduction

1.1 Background

Today most new cars in the market have an Electronic Stability Control (ESC) system which is an active safety system that stabilizes the yaw rotation of the car, using the brake system. It has proven to be an efficient safety system.

The active differential is another active system that has not had the same impact on the car market. One of the reasons is the need for additional hardware compared to the ESC which shares the same hardware as the ABS system, except for an additional pump, yaw rate sensor and steering wheel angle sensor. Furthermore the active differential is more common in cars with a higher ambition for performance as the purpose of the system is mainly to ensure the car can utilize all the potential traction that is available on the driven axle. It can however also be used for stabilizing the yaw rotation to a certain degree.

The main purposes of the two systems are different but they don’t necessarily need to be in conflict with each other if an integration of the two systems could be done. Having the two systems in a car could possibly even improve the performance in terms of traction and stability through integration.

For a front wheel driven cars the trend is that the engine power is increasing not only for performance cars but also for regular cars. The increased power requires better performance from the front axle and this could increase the possibility of front wheel drive cars being equipped with an active differential. For this reason it would be of interest to see if an integration of these two systems on a front wheel drive car could increase the stability performance. Raising the performance limit in terms of when the car loses speed and stability does not only make the car safer due to increased performance capabilities but it also makes the car more pleasant and confident to drive.

1.2 Objectives

The aim of this master thesis is to integrate an active differential (AD) with a brake based Electronic Stability Control system (ESC) and evaluate its combined performance compared to conventional solutions in a transient manoeuvre test. The integration consists of merging the two control strategies for the two systems to raise the performance of the car. This is done for two different active differential solutions; the Electronic Limited Slip Differential (eLSD) and the Direction Sensing Locking Differential (DSLD).

1.3 Problem description

The ESC system stabilizes the car by actuating the brakes on individual wheels. The AD improves the traction of the car by partially or fully locking the differential. Consequently because of the locked differential some yaw damping will be added as the outer wheel in a corner will be dragging behind the inner wheel.

Each of these systems individually improves the performance of a car with regards to safety. For a front wheel drive car the systems may produce conflicting actions when used together due to the ESC not being able to operate when the differential is locked. This would require an integration of the control strategies in terms of when and how they should be activated.

The task is then to develop a working control strategy for the integration of an AD and ESC. The control systems will be constructed and evaluated in Matlab/Simulink using an existing simulation model in Simulink. The model will be used as a base and may be extended or modified if necessary for the simulations.

1.4 Scope

This master thesis will be limited to developing control models in Matlab and Simulink.

- The work done will focus on over-steer control meaning the integrated control strategy will only intervene in situations when the yaw rate reaches unacceptably high levels.
- The evaluation of the integrated control strategy will be done through simulation in Simulink.
- Two types of active differentials are integrated with the ESC and they are the DSLD and eLSD.
- The driving case used in this thesis work for evaluating the integrated control strategy is the National Highway Traffic Safety Administration (NTHSA) Sine with Dwell maneuver test.
- The AD and ESC are modeled without any time delays.
- The friction acting between the road and the tires is always equal for all wheels.
- The friction coefficient is always assumed to be 1 unless stated otherwise.

2 Relevant theory

The theory presented in this chapter will be the foundation for the development of the control strategies. This chapter will be referred to when motivations are presented for the development of the control strategies. Furthermore the ISO coordinate system will be used as defined in Figure 1.

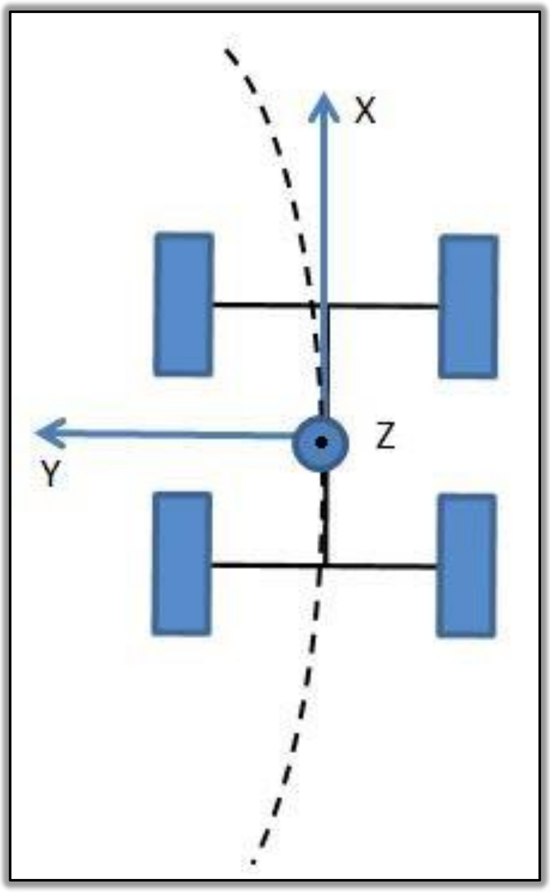

Figure 1 - ISO Coordinates. Vehicle in a left hand turn.

2.1 Friction circle

As each tire can generate a maximum amount of force, it is important to consider the tire force distribution in the longitudinal and lateral direction.

The introduction of the friction circle will make it easier to understand some of the control strategies presented in this report and the behavior of the vehicle in certain driving conditions.

The friction circle can be seen on Figure 2, where the total of lateral and longitudinal forces can never exceed the maximum tire force on a wheel according to Equation 2.1. Pushing the tires beyond the limits of the friction circle saturates the tire and the forces do not increase. This results in the vehicle not being able to maintain an increase in longitudinal and/or lateral acceleration.

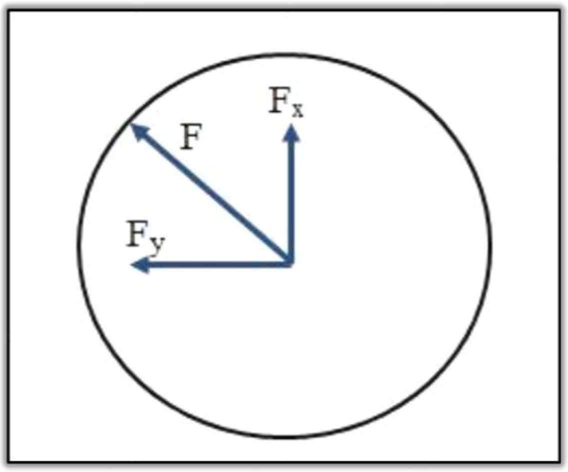

Figure 2 - Friction circle

$$F_{max} \geq F = \sqrt{F_x^2 + F_y^2} \qquad 2.1$$

2.2 Tire force saturation and stability

The effects of tire force saturation are important for developing the control strategies and the system may end up doing more harm than good if this is not understood properly.

The tire forces saturate on a wheel when the slip rate is excessive. When one or two wheels on an axle are saturated before the other axle, two things can happened depending on which axle is saturated first.

If the front axle is saturated first it means the front axle is not able to produce the same amount of lateral grip as the rear axle. This is because the potential for lateral forces from the front axle has been reached while it has not for the rear axle. This means the vehicle will have a negative yaw acceleration which results in a yaw rate decrease.

The yaw rate decrease makes the car deviate from its intended trajectory as the lateral displacement from the curvature center increases and the yaw-rate decreases. The vehicle will skid forward as it cannot maintain its cornering path without decreasing its velocity. The latter means the yaw acceleration of the vehicle would be negative. If the rear axle saturates first, the yaw rate will instead increase, meaning the vehicle most likely will end up spinning around its own axis and control of the vehicle will be lost.

Table 1 shows the change in vehicle motion due to front or rear axle saturation.

Table 1 - Effects of axle saturation

First axle to saturate	$\frac{dv_y}{dt}$	$\frac{dr}{dt}$
Front axle	> 0	< 0
Rear axle	< 0	> 0

Figure 24 in Appendix B shows the vehicle trajectory for front and rear axle saturation respectively for the same initial velocity.

2.3 Load transfer

Load transfer occurs when the vehicle body is accelerated in lateral or longitudinal direction and this result in a transfer of normal load in the opposite direction to the lateral acceleration, meaning the load transfer occurs from the inner wheels to the outer wheels. This has a large impact on how the control strategies are developed as the normal load directly influences the size of the friction circle on the corresponding wheel. Figure 3 shows how the radius of the friction circle becomes larger as the normal load increases.

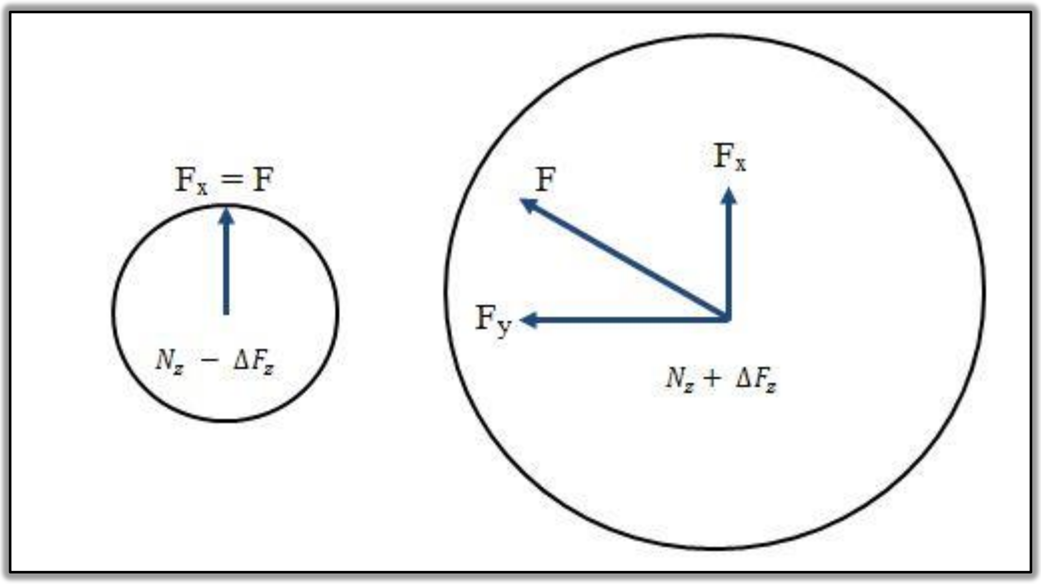

Figure 3 - The effect of lateral load transfer on the friction circle.

The normal load on each wheel is determined by Equations 2.2, 2.3, 2.4 and 2.5.

$$N_{z,FL} = N_{z0,FrontLeft} - \Delta F_{zy,Front} - \Delta F_{zx} \qquad 2.2$$

$$N_{z,FR} = N_{z0,FrontRight} + \Delta F_{zy,Front} - \Delta F_{zx} \qquad 2.3$$

$$N_{z,RL} = N_{z0,RL} - \Delta F_{zy,R} + \Delta F_{zx} \tag{2.4}$$

$$N_{z,RR} = N_{z0,RR} + \Delta F_{zy,R} + \Delta F_{zx} \tag{2.5}$$

The equations give the normal load for the corresponding wheel with regard to both lateral and longitudinal acceleration. The reference condition is when the vehicle is undertaking a left hand corner while accelerating which subsequently will result in a lateral load transfer from the left wheels to the right wheels and a longitudinal load transfer from the front wheels to the rear wheels.

2.4 Yaw moment control

Yaw moment control (YMC) is of central importance in this report. The concept is based on affecting the total vehicle yaw moment acting around the vertical axis of the vehicle and thereby affecting the yaw motion. This makes it possible to affect the vehicle behaviour with regard to both stability and manoeuvrability.

The yaw moment is controlled through individual wheel braking. This can be done with either negative braking torque or positive engine torque through the specific wheel. The very common brake based ESC system controls the yaw moment by using the brakes and torque vectoring differentials control the yaw moment by distributing the engine torque. Both of these systems are shown in Figure 4 for a left hand corner. Equation A.3 in Appendix A shows how the wheel forces affect the yaw acceleration

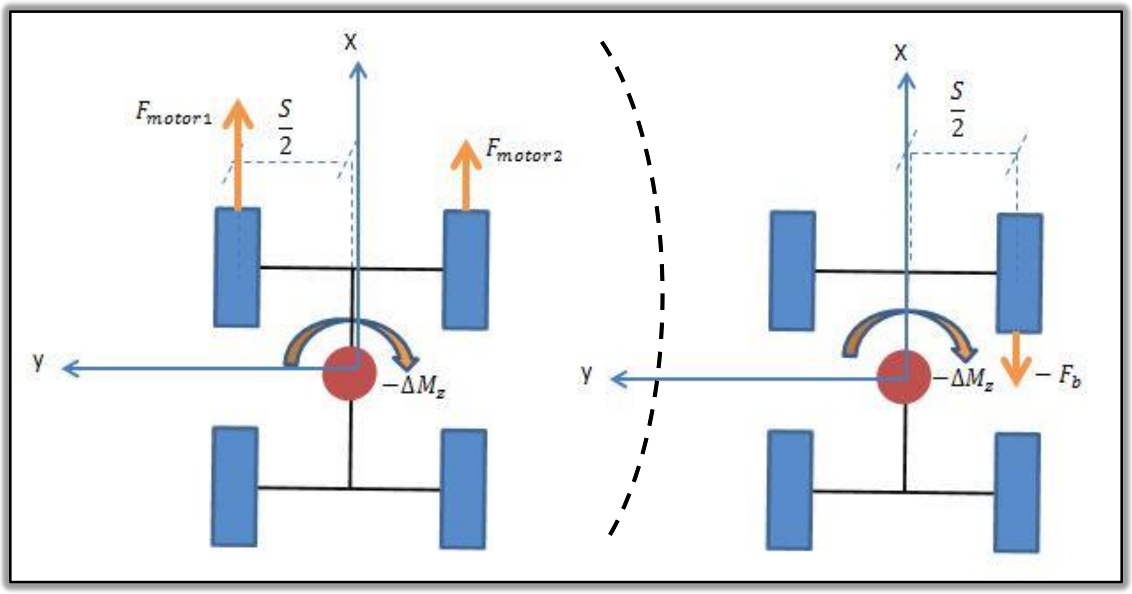

Figure 4 - Left: YMC by engine torque split. Right: YMC by brakes. Left hand turn.

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3 Traction and Braking Theory

3.1 Moment balance

When a two track vehicle travels through a corner its outer wheels will ideally rotate faster than the inner wheel. This is because the curve radius, and thereby the path, that the outer wheel travels is longer than the inner wheel, as shown in Figure 5. A differential allows the engine to transmit the same torque to two wheels spinning at different speeds. The relation between wheels and engine speed follows from Equations 3.1 and 3.2.

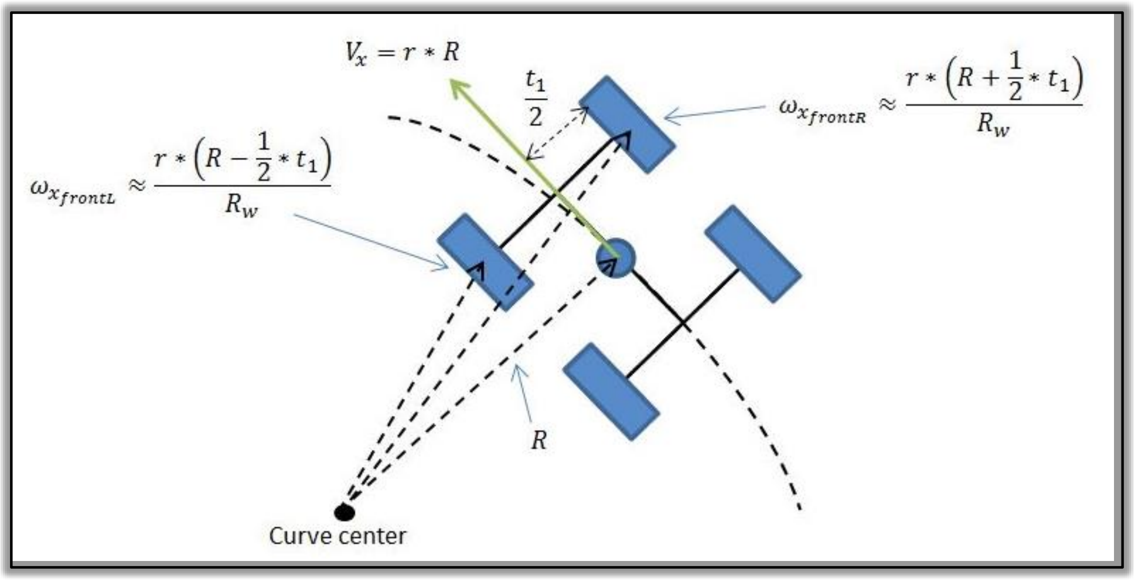

Figure 5 - Difference in rotational velocity between inner and outer wheel.

$$\omega_{\text{final drive}} = \frac{\omega_{\text{left}} + \omega_{\text{right}}}{2} \tag{3.1}$$

$$T_1 = T_2 = \frac{T_{\text{gearbox}}}{2} \tag{3.2}$$

The first equation describes the wheel speed relation and the second equation shows the torque on one wheel is always equal to the torque on the other wheel. This means the wheel with the lowest traction always limits the maximum torque for the other wheel on the axle. If the load transfer is so large that one wheel has no normal load neither of the wheels will be able to have any torque and the outer wheel will stand still while the inner wheel will spin at double the gearbox output speed according to Equation 3.1. It is also possible if one wheel is on low friction surface such as ice the other wheel will not be able to deliver a higher torque than the wheel on low friction surface. Both these cases would limit how large the total tractive force of the wheel could be and could lead to the car not being able to move at all.

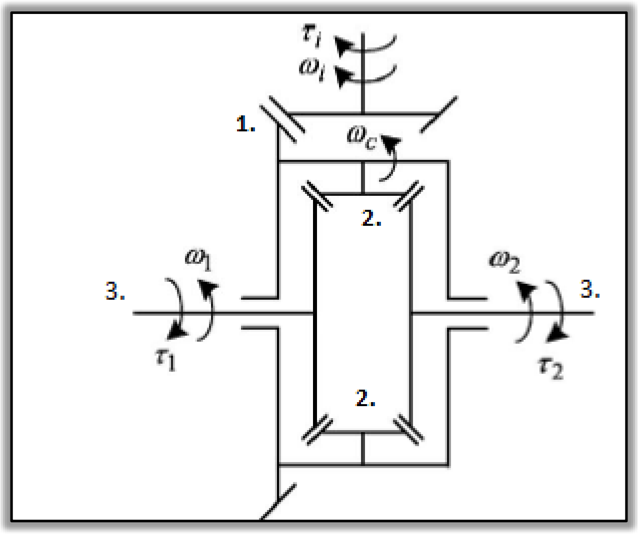

Figure 6 - Open differential

Figure 6 shows the mechanical layout of a differential. The torque is transferred from the gearbox to the differential housing through connection 1. It is then transferred from the housing to each driveshaft (3) through connection 2.

3.2 Locked differential

If both wheels are connected by a rigid axle or if the differential is locked, both wheels will always spin at the same rotational speed. With a locked differential it is possible for the torque to split unequally between the two wheels. The locked axle can be described by Equations 3.3 and 3.4.

$$\omega_{left} = \omega_{right} = \omega_{gearbox} \tag{3.3}$$

$$T_{gearbox} = T_{left} + T_{right} \tag{3.4}$$

With a locked differential the two previously mentioned situations of ice under one wheel and one airborne wheel will be different. In both those cases the wheel with a higher potential traction will be able to transmit a higher torque than the low traction wheel, as can be seen in Equation 3.4. This is an advantage for the locked differential compared to the open one as the vehicle won't lose its total tractive force if one wheel loses traction.

There is however disadvantages to the locked differential. As mentioned for the open differential the outer wheel will ideally always spin faster than the inner wheel when the vehicle is cornering. A locked differential will not allow this as it always forces both wheels to spin at the same speed. This will lead to a counteracting yaw moment when cornering, as long as the normal load on both wheels is large enough, which resists the yaw movement of the car due to the outer wheel dragging behind the inner wheel. With enough throttle input while cornering during high lateral acceleration the outer wheel could also produce a larger longitudinal forward force than the inner which actually instead creates a yaw moment that helps the yaw rotation.

Being able to control the lock will allow the differential to operate as either an open or a locked differential and combine the advantages of both. There are also differential designs that can be partially locked. The partial lock puts a limit on the level of torque transfer that is possible. If the car is running with a relatively high normal load and grip on both wheels a higher level of locking will increase the under steer resisting moment. If the vehicle is running with high grip on one wheel and low grip on the other a higher level of lock will increase the torque transfer from the low grip wheel to the high grip wheel.

3.3 Active differential solutions

3.3.1 Direction Sensitive Locking Differential (DSLD)

The DSLD is a differential that can be either fully locked or fully open. When it for example is set for performance it mechanically locks when the inner wheel rotates faster than the outer wheel. To know which wheel is the inner wheel the differential needs a signal which prepares it for which of four modes it should be set in; left turn lock, right turn lock, full open or full locked.

So if the differential is set to left turn mode it will mechanically lock when the left wheel starts rotating faster than the right wheel. The increase in rotational speed on the left wheel is a result of when the normal load on the left wheel is reduced during acceleration as explained in Chapter 2.3 when the differential is open. Locking the axle in this situation will allow torque to be transferred from the engine to the right wheel, as explained in Chapter 3.2.

3.3.2 Electronic Limited Slip Differential (eLSD)

The eLSD can besides being fully locked or open also be partially locked. A common design consists of an open differential with a clutch that connects the differential housing and one of the drive shafts, see Figure 7. Other designs exist as well. When the clutch is engaged it forces them to rotate in the same speed or reduce the speed differentiation depending on the clutch engagement force. This also controls how large the torque difference between the left and right wheel can be. Unlike the DSLD the eLSD needs to be constantly powered to maintain its lock as the clutch needs to be engaged.

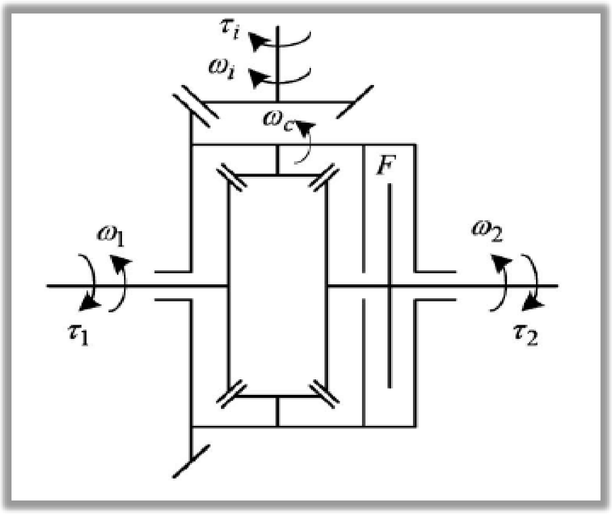

Figure 7 - Electronic Limited Slip Differential

4 Control Strategies

The control strategies describe when and how the actuators for the active systems should be used to control the car behaviour in the way that is desired.

4.1.1 Reference car model

To know when and how to operate the actuation of the subsystems a reference is needed that correlates with the desired vehicle behaviour. The reference then tells whether the current vehicle behaviour needs to be modified. This is based on an idealised vehicle model that gives the yaw rate which the driver expects for the current inputs. The idealised vehicle model is based on equations 4.2 and 4.3 and is called the reference vehicle model.

The simplified reference yaw rate is based on the steering wheel angle and longitudinal velocity of the car while driving through a corner with a large curve radius while the front and rear tire slip is assumed to be zero (Lotus, 2011).

$$\delta = \frac{L}{R} + \alpha_{front} - \alpha_{rear} \;\rightarrow\; \delta = \frac{L}{R} \tag{4.1}$$

$$R = \frac{L}{\delta} \tag{4.2}$$

$$r_{reference} = \frac{v_x}{R} \tag{4.3}$$

The actual vehicle yaw rate is also measured and then compared to the reference yaw rate. This is called the yaw rate error (see Equation 4.4) and it describes how much the actual vehicle deviates with regards to the ideal vehicle performance. The latter represents how the driver expects the vehicle to perform (Carlsson & Tunlid, 2011). When the yaw rate error is positive the real vehicle has a higher yaw rate than the ideal vehicle and the opposite when the yaw rate error is negative. The error is constantly monitored and determines when, how much and in what direction the control strategies should intervene.

$$r_{error} = |r_{actual}| - |r_{reference}| \tag{4.4}$$

The reference yaw rate is limited according to Equation 4.6 based on the real maximum friction coefficient. This is done to prevent the reference model to achieve unrealistically high levels of lateral acceleration and yaw rate. It could otherwise lead to small yaw rate errors even when the real car is unstable. The expression of the maximum reference yaw rate is derived by assuming the vehicle is steady state cornering and its cornering capacity strictly depends on the friction between the road and the tire.

$$a_{y\_ss} = r_{max\_ss} * u = \mu * g \quad 4.5$$

which can be rearranged to:

$$r_{max} = \frac{\mu * g}{u} \quad 4.6$$

4.1.2 ESC (Electronic Stability Control)

The ESC is a brake based system that operates based on the feedback of the yaw rate error described above. It actuates the brakes on individual wheels according to Figure 4 in Chapter 2.4 to control the yaw rate of the vehicle. The amount of braking is determined by Equation 4.7.

$$F_B = -r_{error} * k \quad 4.7$$

The coefficient k in Equation 4.7 is a parameter that is tuned until the ESC system performs adequately.

4.1.3 DSLD

The control strategy for the DSLD is a feed-forward and feedback based control system that uses the input parameters; steering wheel angle (δ_{swa}), vehicle yaw rate, r, and wheel speeds, ω (Carlsson & Tunlid, 2011). Which mode the DSLD will operate in depends on the input parameters and what conditions they fulfill. There are seven conditions and they are:

a) $\delta_{swa} < \delta_{crit}$. This condition determines whether the driver wants to turn in a corner. The critical steering wheel angle, δ_{crit}, should be set so that any disturbances while driving straight, such as road irregularities or wind, do not trigger this criterion. This condition applies for a left hand turn.

b) $r < r_{crit}$. This condition determines whether the vehicle is turning. r_{crit} needs to be set sufficiently high in order to make sure the vehicle is actually cornering and not making small yaw movements as a result of outside disturbances. This condition applies for a left hand turn.

c) $\frac{\omega_l}{\omega_r} > \omega_{crit}$ This condition determines whether the left wheel rotates faster than the right wheel in percentage specified by ω_{crit}. The ratio between the rotational velocity of the left and right wheel is compared against a ratio criteria to determine whether the car is driving over a mu split situation.

d) $r_{error} > r_{crit}$, this condition determines whether the vehicle has a unacceptably high yaw rate.

Mode conditions:

1) $\neg(2 \vee 3 \vee 4)$

In the first mode the car is running straight and the differential is open. Furthermore the first mode is in effect only when the car is not operating in second, third and fourth mode.

2) $a \wedge c \neg \wedge g$

The second and third mode is in effect when conditions a & c and b & d are true (see 4.1.3). In the second and third mode, the differential is conditionally locked, meaning it locks when the inner wheel reaches the same rotational velocity as the outer wheel. The second mode is in effect in a left hand turn while the third mode is in effect in a right hand turn.

3) $b \wedge d \neg \wedge g$

See mode condition 2)

4) $g \vee ((e \vee f)\neg \wedge (2 \vee 3))$

The car is operating in the fourth mode whenever there is a mu split situation or the car is over steering. Mu split situations is only considered when driving straight ahead.

4.2 eLSD Control Strategy

As no complete control strategies for the eLSD was found it had to be designed. It is based on the control strategy of the DSLD but was adapted to utilize the eLSD's additional feature which is the ability to partially lock the differential. It was decided to use the existing control strategy for the DSLD in a modified form. Modes one and four were kept from the DSLD control strategy and one additional was added.

The operating modes for the eLSD are presented as follows:

Mode conditions:

1) $r_{error} \wedge (a * c \vee b * d) \neg \wedge h \neg \wedge g$

$$L_{mag} = \frac{|\mathrm{r_{error}}|}{\mathrm{r_{crit}}}, L_{mag} = locking\ magnitude\ for\ differential\ (0 - 1)$$

Mode one locks the differential with a magnitude corresponding to the yaw rate error in order to negate any excessive yaw rate when full lock is not required, the reason being it is not desirable to reach negative levels of yaw rate error if it can be avoided.

$2)\quad g \vee (e \vee f) \neg \wedge (1 \vee 2)$

$$L_{mag} = 1$$

Mode two locks the differential completely whenever the vehicle has surpassed the critical yaw rate error level. It also locks when one of the driven wheels has a rotational velocity higher than the other wheel for a given amount and the vehicle is not cornering.

$3)\quad \neg \wedge (1 \vee 2)$

$$L_{mag} = 0$$

Mode three comes in effect when none of the other modes applies meaning the differential is open.

5 Integrated braking and active differential control

The integrated control strategy basically uses the AD and ESC in intervals of yaw rate error to stabilize the car. For the eLSD and DSLD two proposals have been done for their integration with the ESC system of the car. Both strategies are divided in three modes which operate in different intervals in terms of yaw rate error. Two yaw rate error limits divides the integrated control strategy into three intervals of yaw rate error.

5.1 Combined Action Strategy

The combined action strategy is presented in Figure 8.

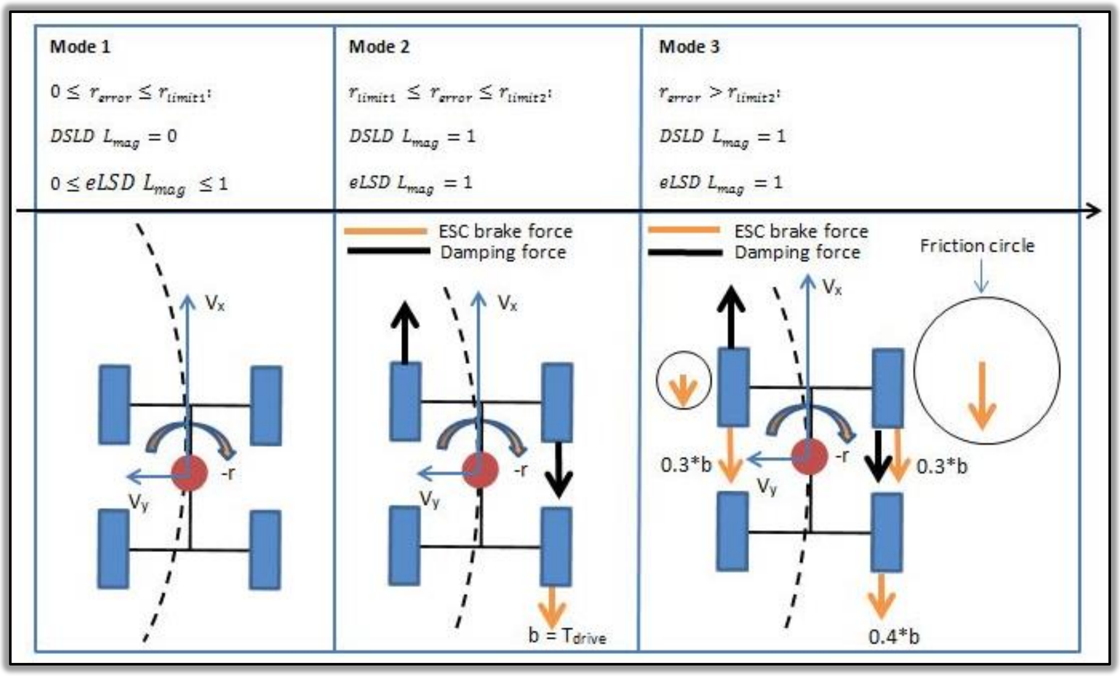

Figure 8 - The combined action strategy operates over a range of yaw rate error.

The combined action strategy applies for both the eLSD and DSLD.

In mode one of the combined action strategy, the previously described control strategies of the DSLD and eLSD are operating as normal. When mode two is in operation the control strategy locks the differential fully, utilizing the yaw damping provided by the locked differential to dampen any excessive yaw movement of the vehicle. Finally in mode three where the yaw rate error has reached critical levels the differential is kept fully locked and the ESC applies to both front wheels a brake force that corresponds to the yaw rate error multiplied with the coefficient 0.3 for the front wheels and 0.4 for the rear outer wheel. High levels of yaw rate error might see the inner front wheel saturated with brake force because of the lateral load transfer due to high lateral acceleration. Any brake force that can't be taken up by the inner wheel will be transferred to the outer wheel which has a higher normal load. The brake torque transfer will result in a yaw moment that counters the yaw movement of the vehicle in addition to the yaw damping effect due to the locked differential.

The main idea of the chosen strategy in mode 3 is to make use of the damping from the locked axle. The yaw damping is instantaneous and does not have any time delay compared to running the ESC alone. However in terms of efficiency the yaw moment control in mode 3 is not as good as using the ESC system only due to the necessity to saturate the inner wheel with ESC brake force. Furthermore as the time delay is not within the scope of this report, the main advantage of control strategy 1 might not be evident in the results acquired from the simulations. To increase the counteracting yaw moment, the rear outer wheel brake is actuated with a magnitude less than the total brake force on the front axle. The magnitude with which the rear brake is actuated with needs to be iterated to make sure the rear axle won't saturate before the front axle.

5.2 Seperate action strategy

The separate action strategy is presented in Figure 9.

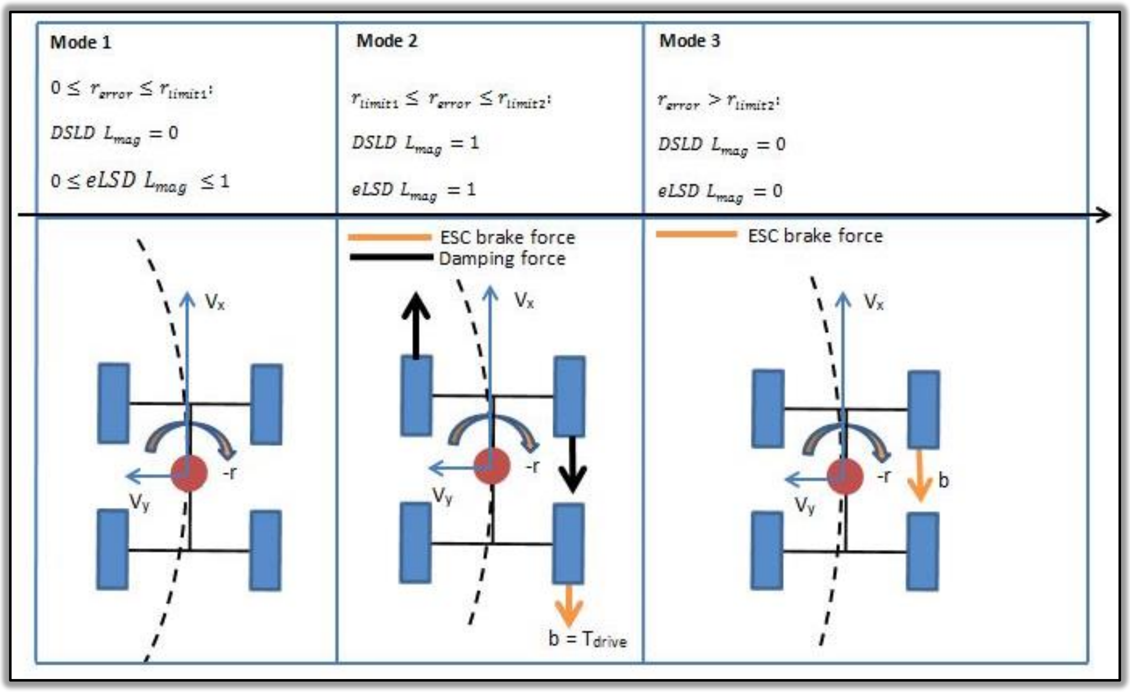

Figure 9 - The separate action strategy. It operates identically to the combined action strategy except for mode three.

The separate action strategy applies for both the eLSD and DSLD. It operates in the same way as the combined action strategy in mode one and mode two. However in mode three where the yaw rate error has reached critical levels, the differential is fully opened and the ESC intervenes and brakes the outer wheel on the front axle with a magnitude corresponding to the yaw rate error to reduce the critical levels of yaw rate.

6 Driving maneuvers

The vehicle is subjected to a large number of different situations in real life use and this puts a lot of demands on for example the electronic systems to be robust enough. There are a large number of standardized driving cases which are made to simulate different situations which the car may be subjected to during regular use. These three cases have been considered for the development of the integrated system:

- Sine with dwell transient test
- Braking in turn
- Start on split friction surface/ acceleration on a split friction surface

The first two driving cases both test the stability of the vehicle with the difference that braking in a turn also adds a significant forward load transfer. They are relevant because the integration of an active differential and ESC have a large potential to affect the stability of the vehicle. The third case tests how well torque can be transmitted between the front wheels and this is largely in favor of locking differentials. The ESC can however also help with this by braking the wheel on a low friction surface but this type of ESC function is not within the scope of this thesis.

6.1 Sine with Dwell

The Sine with Dwell (SWD) is an open loop maneuver used in the National Highway Traffic Safety Association FMVSS-126 (NHTSA, 2007) regulation which evaluates the stability of a car during transient cornering. The main purpose of the test is to evaluate the stabilizing performance of the ESC when the car reaches undesirably high levels of yaw rate.

The SWD maneuver is performed with a steering input which is a sinus input with a dwell part of half a second after reaching the second peak in steering wheel amplitude. The maneuver is performed with a steering frequency of 0.7 Hz with an initial longitudinal velocity of 80 km/h.

The evaluation of the car performance during the SWD maneuver is done by measuring three criteria which will determine whether the car is stable enough while still retaining acceptable level of maneuverability. Two of the criteria determine the stability of the car by comparing the second yaw rate peak of the car which follows the steering wheel angle, with the yaw rate of the car at one second after completion of steer (CoS) for the first stability criteria and 1.75 seconds after CoS for the second stability criteria. CoS is in effect when the steering wheel angle reaches less than five degrees. The stability criteria are determined through Equations 6.1 and 6.2 which show the yaw rate measurements for the stability criteria that are taken at after CoS.

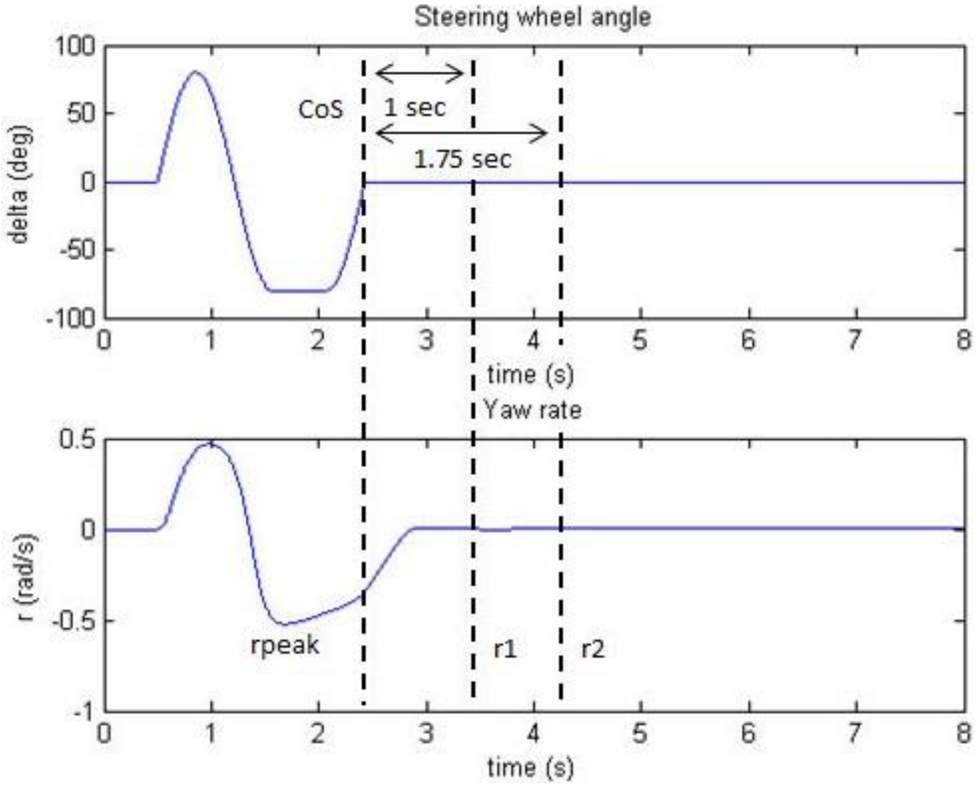

Figure 10 - The yaw rates for the stability criteria are taken well after CoS.

$$r_{ratio,1} = 100 \frac{r_1}{r_{peak}} \tag{6.1}$$

$$r_{ratio,2} = 100 * \frac{r_2}{r_{peak}} \tag{6.2}$$

For a car to be safe in critical driving situations, it needs to be maneuverable as well as stable. The maneuverability criteria make sure that vehicle does not become to sluggish due to the intervention of the active systems as it needs to be able to avoid an obstacle. It determines the cars maneuverability by measuring the lateral displacement from the obstacle 1.07 seconds after beginning of steer (BoS). Figure 11 shows the measurement is taken when the evasive maneuver is almost completed.

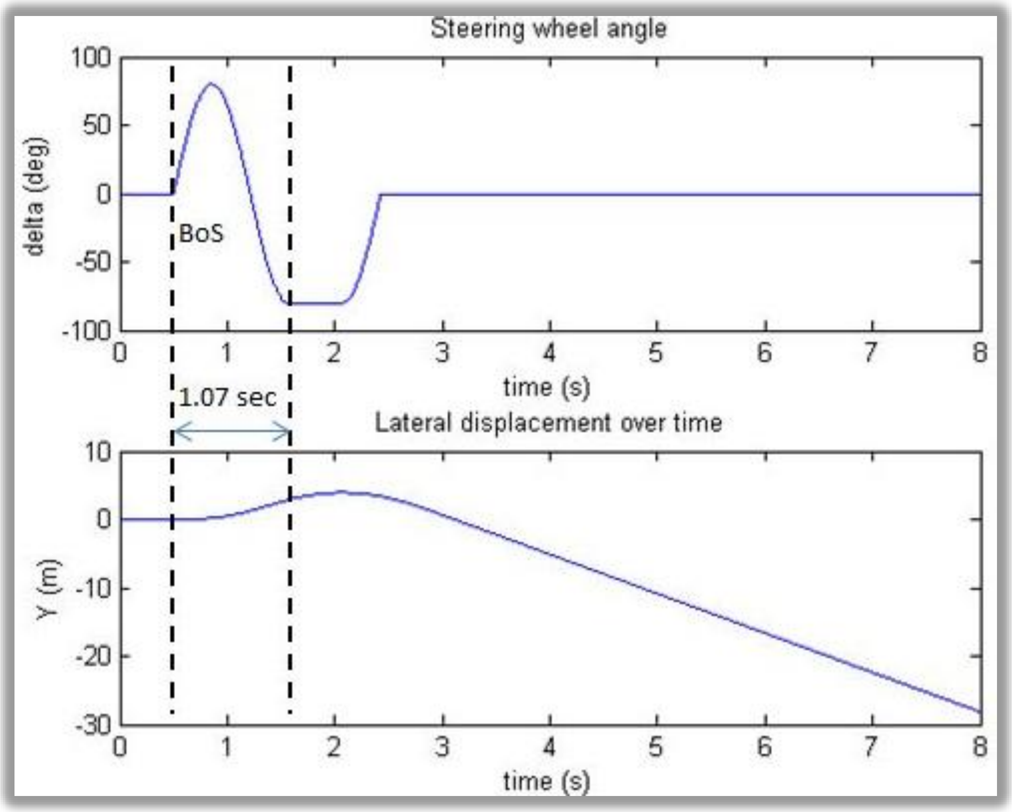

Figure 11 - Measurement of the lateral displacement is taken 1.07 seconds after BoS

BoS is in effect when the steering wheel angle is more than 5 degrees. Table 2 shows what the yaw rate ratios and lateral displacement of the car needs to be to pass the SWD test.

Table 2 - The criteria's for passing the SWD test.

First stability criteria	$r_{ratio,1} < 35\%$
Second stability criteria	$r_{ratio,2} < 20\%$
Maneuverability criteria (Lateral displacement)	$y_1 > 1.83$ m

7 Vehicle systems modelling

The control systems are designed with the help of a simulation model that uses the vehicle parameters of a Saab 9-3 (see Appendix F). The model consists of a number of subsystems that feed into each other and represent different parts of the car such as; driver model, powertrain model etc.

The model is for this thesis simulated in Simulink and the implementation can be seen in Figure 27 in Appendix D.

7.1 The driver model subsystem

The driver model is a simple open loop model. It outputs time depending values for the steering wheel angle, throttle pedal position and brake pedal position. The model also outputs a variable which contains the setting for which control systems should be active. The Simulink implementation is shown in Figure 29 Appendix D.

7.2 The active systems subsystem

This subsystem implements all the control strategies specified in Chapter 4 for the ESC and the active differential. It uses the driver inputs and outputs the individual brake torques as well as the level of locking for the eLSD or the mode for the DSLD. Simulink implementation is shown in Figure 30 in Appendix D.

7.3 Vehicle dynamics model subsystems

This collection of subsystems describes the complete behaviour of the vehicle based on the inputs from the driver and the active systems. An overview of the Simulink implementation is seen in Figure 28 in Appendix D.

7.3.1 Equations of motion

This subsystem describes the motion of the vehicle based on all forces acting on the vehicle. Figure 12 shows the lateral and longitudinal forces acting on the vehicle.

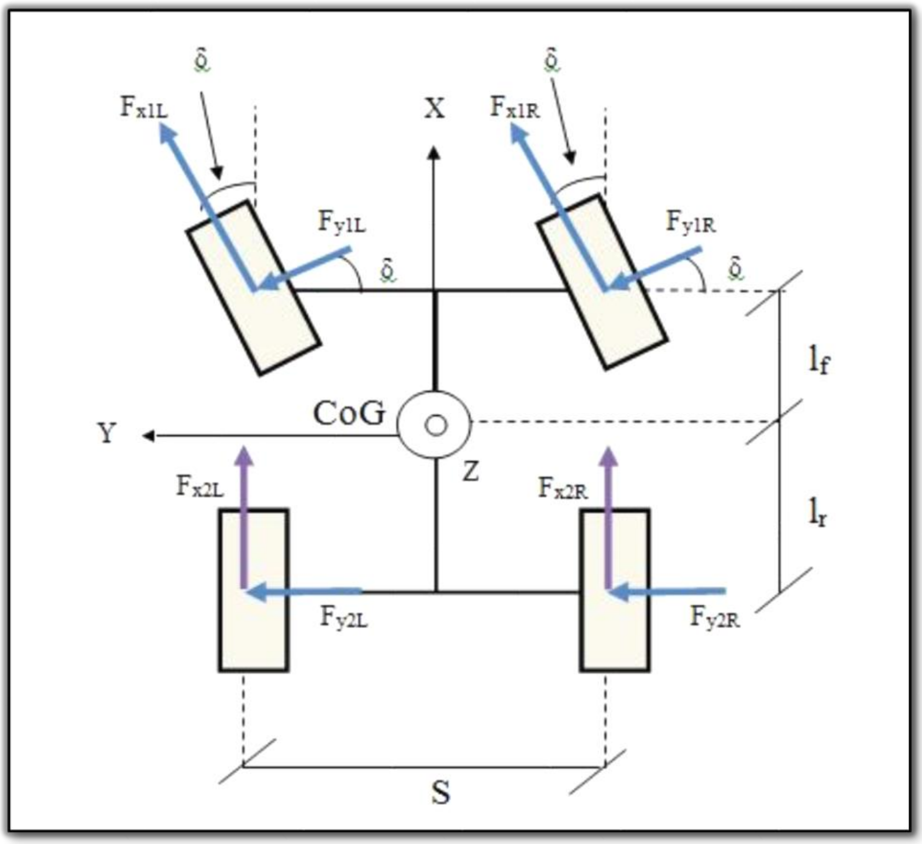

Figure 12 - Free body diagram of a two track vehicle

Based on this free body diagram equations 7.1, 7.2 and 7.3 describe the acceleration in the X- and Y direction and about the Z-axis in the vehicle coordinate frame.

$$\uparrow X: \quad m * a_x = m * \left(\frac{dv_x}{dt} - v_y * r\right) = \sum F_x \Rightarrow \frac{dv_x}{dt} = \frac{\sum F_x}{m} + v_y * r \qquad 7.1$$

$$\leftarrow \mathrm{Y}: \quad \mathrm{m} * \mathrm{a_y} = \mathrm{m} * \left(\frac{\mathrm{dv_y}}{\mathrm{dt}} + \mathrm{v_x} * \mathrm{r}\right) = \sum \mathrm{F_y} \Rightarrow \frac{\mathrm{dv_y}}{\mathrm{dt}} = \frac{\sum \mathrm{F_y}}{\mathrm{m}} - \mathrm{v_x} * \mathrm{r} \qquad 7.2$$

$$\circlearrowleft \mathrm{Z}: \quad \mathrm{I_{zz}} * \dot{\mathrm{r}} = \sum \mathrm{M_z} \Rightarrow \dot{\mathrm{r}} = \frac{\sum \mathrm{M_z}}{\mathrm{I_{zz}}} \qquad 7.3$$

The exact expressions used in the simulation model can be seen in Equation A.1-A.3 in Appendix A.

Integrating the accelerations will give the planar and rotational velocities of the vehicle in its coordinate frame. The trajectory of the vehicle is found by integrating the global velocities of the vehicle, shown in equations 7.4 and 7.5. The expressions of the vehicle global velocities are derived from the relation between the vehicle and global coordinate system shown in Figure 22 in Appendix A.

$$\mathrm{V_{X\,Global}} = \mathrm{V_x} * \cos(\mathrm{yaw}) - \mathrm{V_y} * \sin(\psi) \qquad 7.4$$

$$\mathrm{V_{Y\,Global}} = \mathrm{V_y} * \cos(\mathrm{yaw}) + \mathrm{V_x} * \sin(\psi) \qquad 7.5$$

The wheel speed is calculated based on Figure 13 using the difference between applied torque and tractive force from the tire as shown in Equation 7.6.

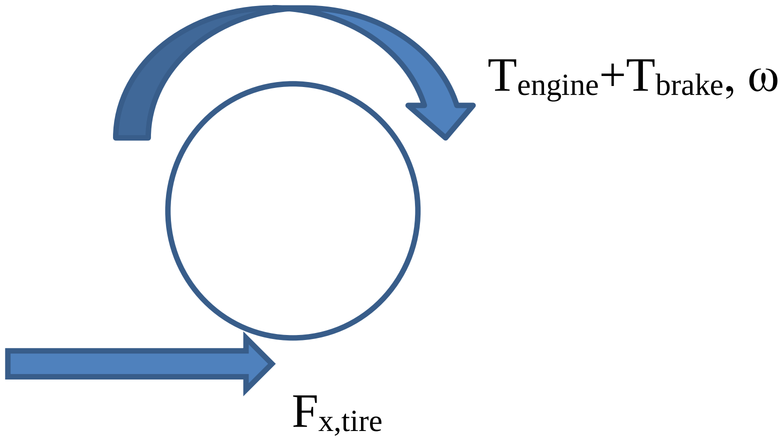

Figure 13 - Torques and forces acting on the wheel

$$I\dot{\omega} = T_{engine} + T_{brake} - F_{x,tire} * R_{wheel} \tag{7.6}$$

As Equation 7.6 bases the rotational acceleration on the difference between the traction torque and the applied torque on the wheel from the final drive will accelerate when they are not equal. This happens when the tire is saturated and the engine torque is higher than the tire can transmit to the ground. This happens when the tire is saturated and the applied torque on the on wheel from the final drive is larger than the tractive torque in that instance.

Figure 31 in Appendix D shows the subsystem in Simulink with the corresponding inputs and outputs.

7.3.2 Tire model

The forces produced by the tires are calculated using an empirical combined slip model (Pacejka & Besselink, 2012). This uses the slip angles and the longitudinal slips of the vehicles to calculate both the lateral and longitudinal forces of the tire.

The longitudinal slip rate (κ) for a wheel is calculated using Equation 7.7 and it is positive when the wheel moves faster than the ground, such as during heavy throttle input, and it is negative when the wheel moves slower than the ground, for example during braking.

$$\kappa = \frac{R_{wheel} * \omega - V_{road}}{\max(R_{wheel} * \omega,\ \ v_{road})} \tag{7.7}$$

Using Figure 23 in Appendix D Equations 7.8, 7.9, 7.10 and 7.11 are derived and they are used to calculate the slip angles (α) for each tire.

$$\alpha_{1L} = \delta - \arctan\left(\frac{v_y + l_f * r}{v_x - \frac{S}{2} * r}\right) \tag{7.8}$$

$$\alpha_{1R} = \delta - \arctan\left(\frac{v_y + l_f * r}{v_x + \frac{S}{2} * r}\right) \tag{7.9}$$

$$\alpha_{2L} = - \arctan\left(\frac{v_y - l_r * r}{v_x - \frac{S}{2} * r}\right) \tag{7.10}$$

$$\alpha_{2R} = - \arctan\left(\frac{v_y - l_r * r}{v_x + \frac{S}{2} * r}\right) \tag{7.11}$$

The combined slip tire formula 7.15 together with equations 7.12, 7.13 and 7.14 are then used to find the total force produced by the tire. The tire coefficients μ and c in tire model are dependent on the type of road surface, and they are given by a subsystem in Simulink called “road properties”.

$$\sigma_x = \frac{\kappa}{1 + |\kappa|} \tag{7.12}$$

$$\sigma_y = \frac{\tan(\alpha)}{1 + |\kappa|} \tag{7.13}$$

$$\sigma = \sqrt{\sigma_x^2 + \sigma_y^2} \tag{7.14}$$

$$F = \mu * F_z * \tanh\left(c * \frac{\sigma}{\mu}\right) \tag{7.15}$$

The total tire force is then split in longitudinal and lateral direction according to how big the slip is in either direction, see Equation 7.16 and 7.17.

$$F_x = F\frac{\sigma_x}{\sigma} \tag{7.16}$$

$$F_y = F\frac{\sigma_y}{\sigma} \tag{7.17}$$

If for example there is no lateral slip in equation 7.16, the total force, *F*, would be equal to the longitudinal force produced by the tire.

Figure 25 and Figure 26 in Appendix C show plots of the split in longitudinal and lateral tire forces and how it varies with increase in κ/α and constant α/κ. Figure 32 and Figure 33 in Appendix D shows the subsystem in Simulink with its inputs and outputs.

7.3.3 Load transfer

The load transfer uses the longitudinal and lateral accelerations and drag force as inputs to calculate the current change in wheel normal loads, dFzx and dFzy.

The roll inertia and damping for the vehicle is not known. The roll rate damping is therefore replaced by a first order filter on the undamped signal to approximate how a damped system would behave. The results of filtering can be seen in Figure 14 and Figure 15. Simulink implementation is also shown Figure 34 in Appendix D.

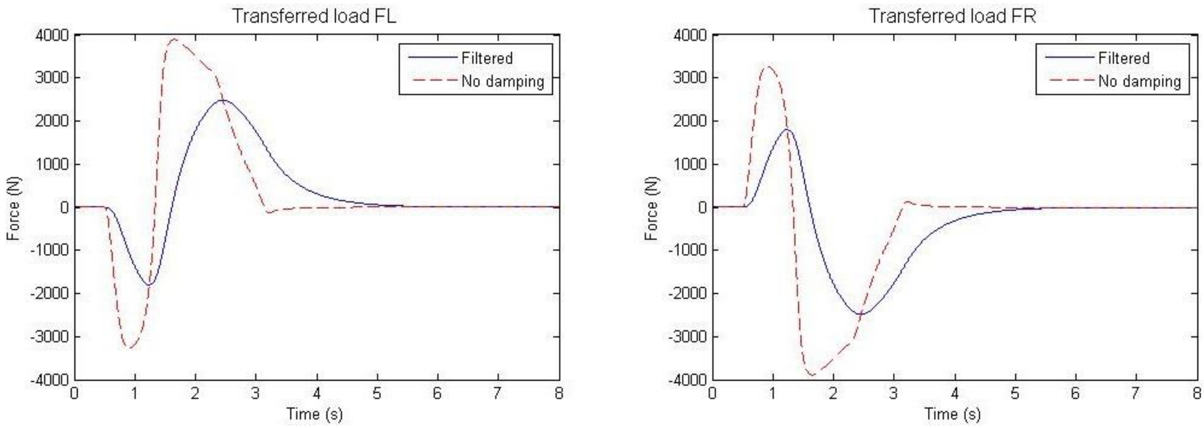

Figure 14 - Roll damping on front axle, transferred force, $\Delta F_{zy,F}, vs\ time$

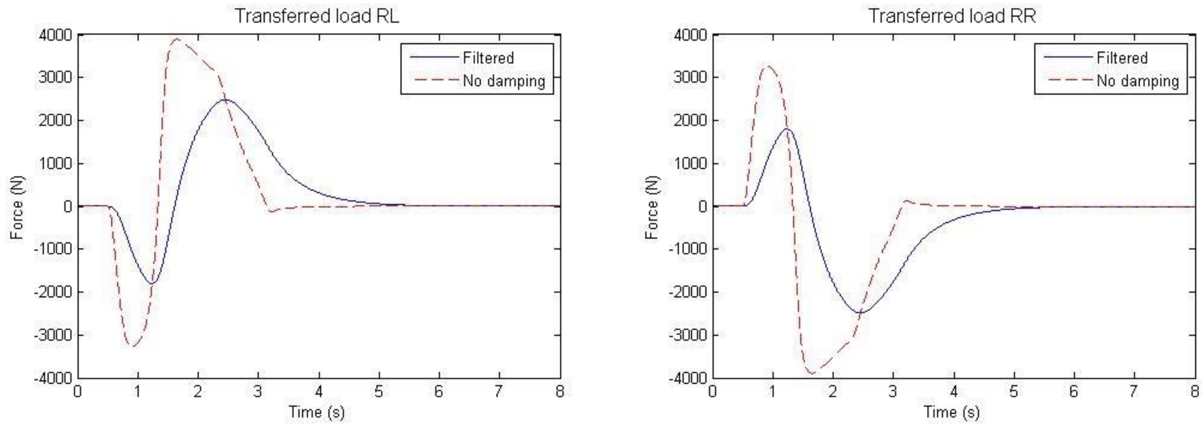

Figure 15 - Roll damping on rear axle, transferred force, $\Delta F_{zy,R}, vs\ time$

7.3.4 Powertrain model

The powertrain model incorporates both the power delivery from the engine as well as the torque split from the differential. The engine model use the throttle pedal position to determine how large the delivered torque should be. If the throttle pedal is at zero there will be a negative torque to simulate off throttle engine braking.

The behaviour of an open differential is simulated by calculating the potential torque for both driven wheels using the current normal load and friction coefficient, Equation 7.18.

$$T_{min} = \min(F_{z,1}\mu_1 , F_{z,2}\mu_2)\, R_w \qquad 7.18$$

The wheel with the lowest potential torque is then set as the torque output for both wheels. This will make both wheel torques identical and limited by the lowest traction wheel as described in chapter 3.1.

$$T_{high} = T_{min} + (T_{total} - 2T_{min})lock_{diff} \qquad 7.19$$

To simulate the effects of a locking differential on torque transfer the wheel with higher potential grip gets a higher torque according to Equation 7.19. For the eLSD partial lock limits the level of transferred torque while the DSLD can only be either one or zero.

The yaw damping effect of a locked differential is simulated using the difference ideal wheel speed. This will always vary according to the corner radius and vehicle track width and wheelbase. The difference between the right and left ideal wheel speeds is then multiplied with a coefficient and added to each wheel torque to create the damping effect as can be seen in Equations 7.20, 7.21 and 7.22. This will always reduce the torque on the inner wheel and always increase the torque on the other wheel as the yaw damping can never help the yaw motion but only counteract it.

$$T_{differential} = k(v_{x,R} - v_{x,L}) \qquad 7.20$$

$$T_{L,tot} = T_L + T_{differential} \qquad 7.21$$

$$T_{R,tot} = T_R - T_{differential} \qquad 7.22$$

The yaw damping should also decrease with load transfer since the effect would be gone if only one wheel has traction. The ideal wheel speed difference is therefore also multiplied with a load transfer coefficient, *k*, that varies from zero to one and is zero when one wheel has zero normal load. This is shown in equation 7.23.

$$T_{differential} = k(v_{x,R} - v_{x,L})n_{l.trans} \qquad 7.23$$

$$n_{l.trans} = \frac{\min(F_{z,1} + F_{z,2})}{m * g * \frac{b}{2l}} \qquad 7.24$$

See Figure 35 and Figure 36 in Appendix D for the Simulink implementation.

7.3.5 Brake subsystem

There is no speed sensing anti-lock brake feature in the model. It is instead replaced by a system that calculates the highest potential brake torque for each wheel, see Equation 7.25.

$$T_{i,max} = F_{z,i}\mu_i R_w \qquad 7.25$$

The incoming brake torque is then limited to not exceed what the tire can take to avoid excessive slip.

If the differential is locked when both front wheels are braked the subsystem will also split the total brake torque between the two wheels, see Equation 7.26.

$$T_i = \left(T_{1,brakes} + T_{2,brakes}\right)\frac{T_{i,max}}{T_{1,max} + T_{2,max}} \qquad 7.26$$

Simulink implementation is shown in Figure 37 in Appendix D.

8 Results

The results of the performance criteria for the vehicles with different solutions are presented in Table 3.

Table 3 - Sine with dwell criteria

Criteria	Stability 1 (%)	Stability 2 (%)	Maneuverability (m)
	Steering wheel amplitude = 100 deg		
Open differential	5,97	0,00	3,22
ESC	0,00	0,00	3,02
eLSD+ESC, Strategy 1	0,00	0,00	3,06
DSLD+ESC, Strategy 1	-0,02	0,00	3,07
eLSD+ESC, Strategy 2	0,00	0,00	2,99
DSLD+ESC, Strategy 2	0,00	0,00	3,00
	Steering wheel amplitude = 120 deg		
Open differential	48,58	17,28	3,38
ESC	0,00	0,00	3,17
eLSD+ESC, Strategy 1	-0,07	0,00	3,20
DSLD+ESC, Strategy 1	-0,29	0,00	3,21
eLSD+ESC, Strategy 2	0,00	0,00	3,15
DSLD+ESC, Strategy 2	0,01	0,00	3,16

The results show that up to 180 degrees steering wheel amplitude all vehicle configurations presented except for the open differential pass the stability and maneuverability criteria. Furthermore the ESC and the integrated control strategies for both types of active differentials manage to pass the criteria with a good margin.

To further distinguish the performance between the solutions the phase between the yaw rate and steering wheel input is examined. Figure 16 shows six different plots each with the phasing between the SWA and the yaw rate of the vehicle for 180 degrees of steering wheel amplitude. It's clear that the ESC and control strategy 2 for both types of active differentials are the best in terms of phasing compared to control strategy 1 and the open differential which is quite bad at 180 degrees of steering wheel amplitude.

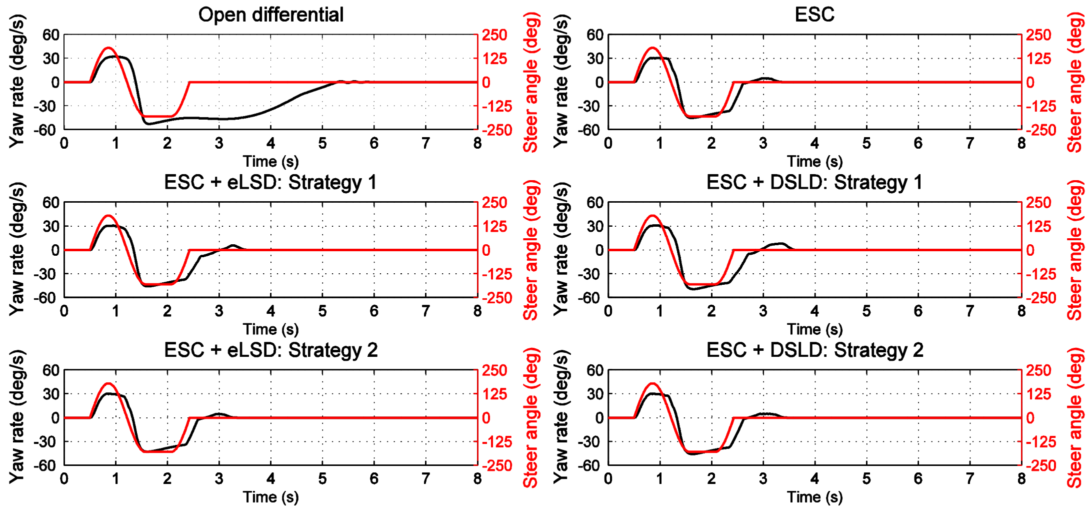

Figure 16 - Phasing between steering wheel angle and yaw rate.

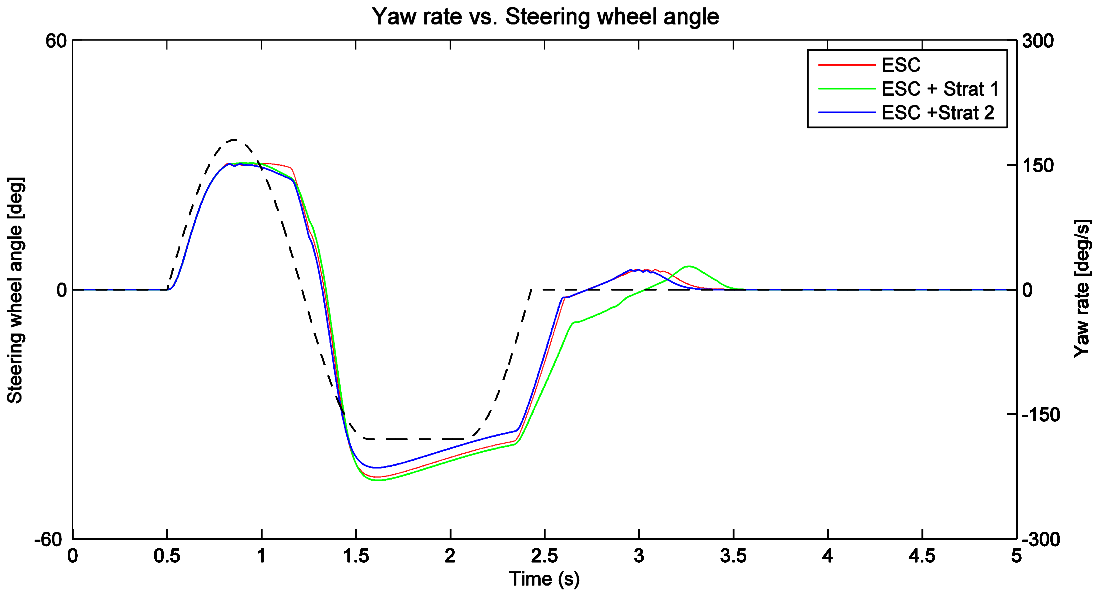

Figure 17 - Yaw rate vs. Steering wheel angle comparison

For increasing steering wheel amplitudes the difference between the ESC and control strategy 2 increases with the latter being more responsive in terms of yaw rate phasing, see Figure 17. The reason for this can be explained with the difference in wheel speed between the front inner and outer wheel increases with the steering wheel amplitude which consequently increases the damping forces as can be seen in equation 7.20. Figure 18 shows how the damping forces differ between cases with different steering wheel amplitude in mode 2 in control strategy 2.

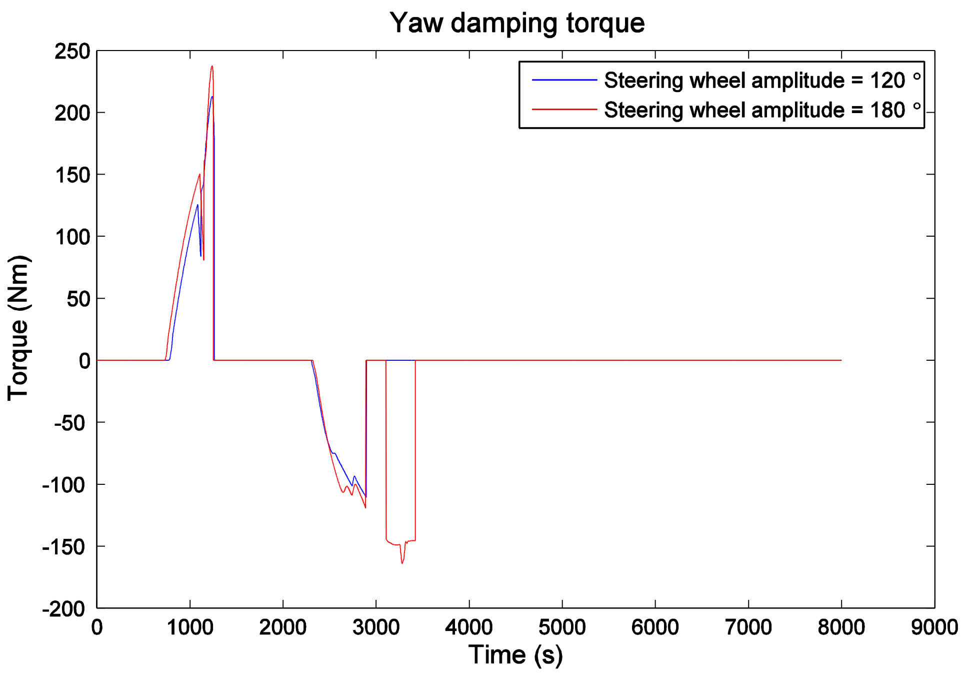

Figure 18 - Yaw damping comparison between the cases 120 and 180 degrees.

To show how much braking effort that was needed from the ESC, Control strategy 1 and 2 to stabilize the vehicle the final longitudinal velocity was checked. The longitudinal velocities can be seen on Figure 19 for the case of 180 degrees of steering wheel amplitude. It is evident by studying the plots that among the three solutions, control strategy 1 has the highest braking effort. The reason being that any negative yaw moment created is done by saturating the inner front wheel to such an extent that any surplus braking torque is transferred to the outer front wheel thus creating a negative yaw moment. Compared to other strategies more total braking force is required to achieve the same level of yaw damping.

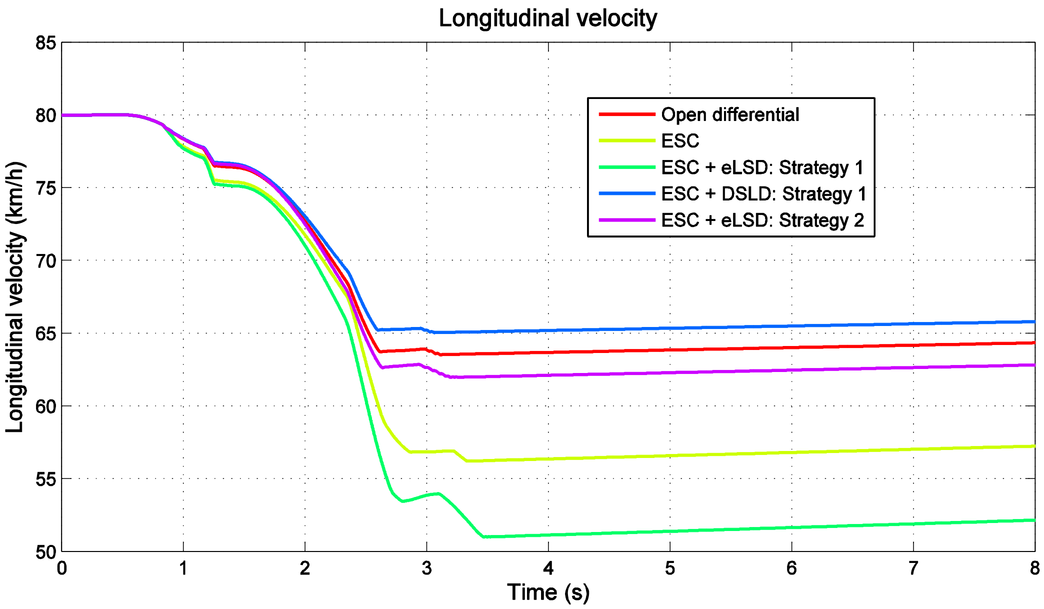

Figure 19 - Longitudinal velocity for the different solutions at 180 degrees of SWA.

To see whether there is a benefit to increase the second error limit the amount of times the vehicle enters yaw rate error mode 2 can be seen on Figure 38 and Figure 39 in Appendix E before increasing the yaw rate error limit.

While it can be argued that the operating range in mode 2 for control strategies 1 and 2 is too narrow in terms of making more use of the yaw rate damping of the locked differential, increasing the operating range of mode 2 partially reduces the number of times the ESC intervenes for the integrated control strategies and also increases the time spacing between the interventions. After the increase of the error limit, it can be seen on Figure 40 and Figure 41 in Appendix E that control strategy 2 with the eLSD intervenes the least amount of times. It is suggested that raising the second error limit would then be beneficial in terms of reducing the number of times the integrated control strategies and the ESC systems enters mode 3

Due to the small differences in results between the different solutions a simple time delay of 200ms is added to the ESC brake actuation to evaluate how much it affects the end results. This is done to give a hint of how much of an impact the time delay has on the results. Due to time constraints this is only implemented for the eLSD.

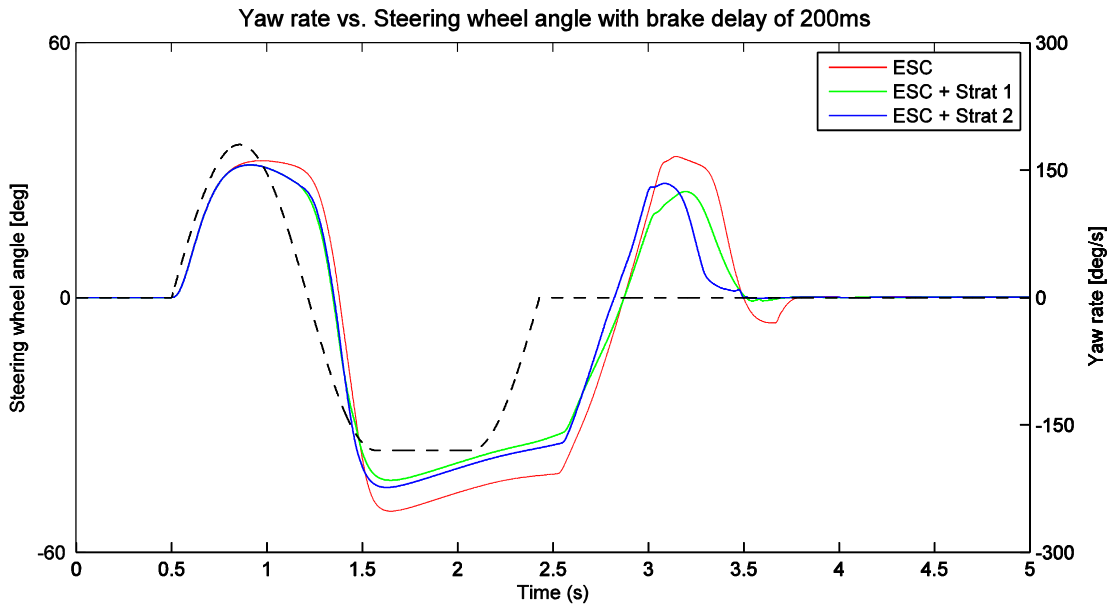

Figure 20 - Yaw rate comparison with brake delay of 200ms

Figure 20 (compared to Figure 17) shows that the time delay does indeed affect the results in favour of the combined systems. The response around 1s and 1.5s is faster for integrated strategies compared to the ESC. The overshoot around 3s is also larger for the ESC and it overshoots to a negative yaw rate value again.

Figure 21 also shows that the speed loss difference between strategy 2 and the ESC slightly larger, 2.8km/h speed loss at 4 seconds compared to 1.5km/h loss without delay. This means that strategy 2 can perform the same manoeuvre with a smaller speed loss.

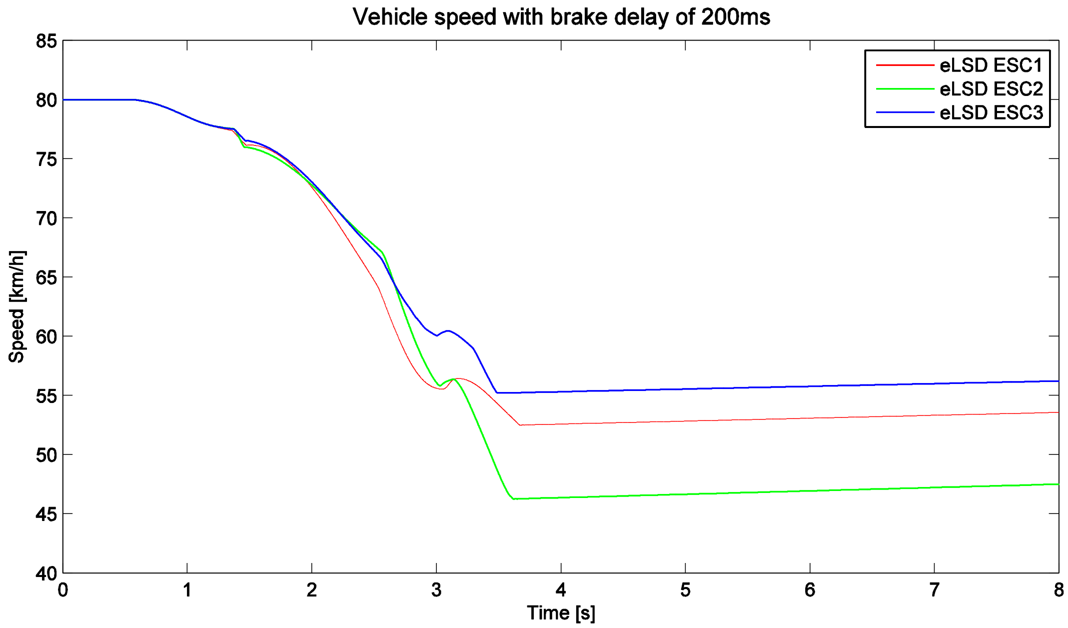

Figure 21 - Speed comparison with brake delay of 200ms

9 Discussion

The results of the three solutions pass the SWD test criteria. The trajectories of the solutions also seem to be in line with what is expected and that is to take the right hand turn in the SWD maneuver while having enough lateral displacement without becoming unstable. It has been hard to identify what else is expected of the vehicle trajectory except for any extreme deviation compared to the reference trajectory.

To get a better idea of which of the solutions are ahead in terms of performance the braking effort needed to stabilize the vehicle was investigated as well as the amount of times the ESC intervenes, also the time span between the interventions and finally the phase between the steering wheel input and the yaw rate of the different solutions

Control strategy 1 needs a higher braking effort to stabilize the vehicle compared to running control strategy 2 for both differentials and the ESC system. This is reflected in the longitudinal velocity of the vehicle when it drives in a straight path after completing the maneuver and this is due to the fact that any contribution to the yaw moment from the front axle that is needed to stabilize the vehicle requires a higher braking effort. Control strategy 2 with the eLSD requires the least braking effort and in direct comparison to running the ESC system this is due to the damping effect of the locked differential which reduces the necessary total braking force from the ESC to stabilize the vehicle. For control strategy 2 with the DSLD the longitudinal velocity is slightly lower compared to the ESC. This result didn’t turn out as expected for the same reason why control strategy 2 with the eLSD solution has the least braking effort.

While the braking effort tells how much the ESC intervenes in terms of total braking magnitude it was worth checking the number of times the ESC intervenes. Control strategy 1 for both differentials intervenes least frequently with bigger time spacing compared to the other solutions. Control strategy 2 with the eLSD had fewer interventions in comparison with the ESC system and control strategy 2 for the DSLD. The time spacing between some interventions is also quite small with around 2 hundredths of a second between the interventions which is unrealistic due to the reaction time of the ESC system being between 200-300 ms.

The time spacing and the number of interventions seems to relate to some extent with how high the second yaw error limit is set. The higher the second yaw error limit is for all solutions, the higher the brake forces it will intervene with according to equation 4.7. This was found when raising the second yaw rate error limit for all of the solutions. This would simply suggest that the solutions are braking harder with ESC interventions that lasts longer but are fewer in number. A lower yaw error limit then means the yaw rate error is dwelling around the second error limit more often due to the lower brake forces. Having a lot of interventions coming in at too early yaw rate error especially with a narrow time space in between might be too much of a discomfort for the driver and passenger. It might also lower the average driver’s confidence in the car if the system isn’t decisive enough and intervenes too frequently. The increase in error limit previously mentioned favored control strategy 2 with the eLSD the most and this may be explained in the damping effect of the locked differential having a bigger impact due to it working over a bigger range in mode 2.

It has been hard to distinguish which solution has the smallest phasing between the steering wheel input and the yaw rate. It is quite close between control strategy 2 with the eLSD and the ESC. While the ESC has a slightly smaller phasing for 120 steering wheel amplitude, the same is true for control strategy 2 with the eLSD at 180 steering wheel amplitude. Control strategy 1 for both differentials is noticeably worse compared to the other solutions. The reason for this may be that the yaw moment contribution due to the damping and the difference of the brake forces on the front axle changes with the lateral load transfer to a higher extent compared to control strategy 2 and the ESC system.

Based then on the overall results acquired, control strategy 2 with the eLSD is the best solution meaning an integration of the strategies will see a small gain in performance. Though it has to be said that the performance of control strategy 2 with the DSLD not being on par with the ESC system questions whether the yaw damping model of the locked DSLD as well for the eLSD is valid enough for simulations after the adaptations done to it to fit it into the Simulink model. The addition of the damping from the locked axle in mode 2 in the integrated control strategies should be beneficiary if the error limits are the same for all solutions. What also can be questioned is the realism of having ESC brake interventions with 2 hundredths of a second in between. The latter together with the fact that some ESC brake actuators not being able to apply small brake forces the wheels means that the yaw rate error limits needs to be increased in order to have a properly working integrated control strategy and ESC system. The modeling of time delays, proper ESC actuators along with use of a modeled ABS system in the model would probably give more accurate results which could further distinguish the results of the solutions.

A minor study was then done to give a hint on the impact time delay has on the results. The integrated control strategies shows that they are not as sensitive to the ESC brake actuation time delay as the ESC which is as expected due to the integrated control strategies are not solely relying on YMC through brake actuation. This minor study then augments the recommendation for future work which would be to make the ESC brake actuation more detailed in terms of gaining more accurate hardware and software performance.

Lastly, after raising the second yaw rate error limit it could be seen that the amount of times the ESC intervenes for all solutions was heavily reduced along with an increase in time spacing between the interventions. This would suggest that some tuning of the error limits and other parameters like the brake constant might or might not result in performance gains. It is however important to decide for what purpose the parameters need to be tuned for. As it has been seen, besides having all the solutions pass the stability and maneuverability criteria of the Sine with dwell what kind of comparison is needed to distinguish the solutions from each other in terms of performance?

10 Conclusion

Each solution has it strength and weakness. Their results differ with a small margin for most tests. The integrated control strategy 2 with an eLSD is however the solution with the most advantages when combining all results and is therefore also the best solution. Whether its small advantages would merit the extra costs for adding the system to a vehicle can however be debated.

Any front wheel drive car attempting to actuate the ESC and eLSD or DSLD simultaneously for over-steer control would have to abandon the standard concept of actuating the brakes on the front outer wheel. This could prove unwise as the risk for rear axle saturation with combined tire forces might end up with the control of the car becoming harder to retain compared to if the front axle would saturate first. With this in mind, it can be said that a rear wheel drive car could make better use of the integrated control strategies for over-steer control as mode 3 could let the ESC actuate the front outer brake and lock the differential at the rear axle thus reducing the risk of the rear axle saturating first. This means the rear wheel drive car could see a lower braking effort while achieving the same stabilization of the car with a more effective design of mode 3.

11 Recommendations

The results presented in this thesis are based on simplified models that could be developed further if more accurate results are desired. One of the major simplifications of the model is the disregard of reaction times for the controllers, sensors and actuators in the simulation model. Including them has shown to alter the results in favor of the integrated systems. Delays were however only implemented for the brake actuation. It could be implemented in more detail for the brakes and also be introduced for the differentials.

Real life brakes cannot produce braking torque as smooth and linearly as the model does. A more accurate model would model this brake behavior and limit how low brake torques they can produce. This means that a realistic model would put limits on how soft the brakes can brake if the system is tuned to actuate early when the yaw rate error is still small. There is also no ABS feature modeled into the simulations. This is a requirement in all cars today and could affect the results whenever brakes are used.

The differential locking model used is just an approximation. A less approximate model would force a kinematic constraint between the wheel speeds and set a limit to how much torque could be transferred as a result of slip rate change. While the peak levels of torque can easily be adjusted to be reasonable with the current model the buildup might look different if a kinematic constraint was to be used instead.

The parameters affecting brake/differential actuation are in this thesis equal for all systems due to time constraints and lack of tuning and testing. Tuning them separately for all systems and running more tests could improve the individual performance of all systems and draw a more fair comparison as they will use more of their full potential than they do now.

12 References

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Appendix A Equations of motion

$$\frac{dv_x}{dt} = \frac{\left((F_{x1L} + F_{x1R}) * \cos(\delta) + F_{x2L} + F_{x2R} - \left(F_{y1L} + F_{y1R}\right) * \sin(\delta)\right)}{m} + v_y * r \tag{A.1}$$

$$\frac{dv_y}{dt} = \frac{\left(F_{y1L} + F_{y1R}\right) * \cos(\delta) + F_{y2L} + F_{y2R} + (F_{x1L} + F_{x1R}) * \sin(\delta)}{m} - v_x * r \tag{A.2}$$

$$\begin{aligned} \dot{r} = (\left(F_{y1L} + F_{y1R}\right) * \cos(\delta) * l_f \ldots\ldots\ldots + (F_{x1R} - F_{x1L}) * \cos(\delta) * \frac{S}{2} \ldots\ldots\ldots \\ - \left(F_{y2L} + F_{y2R}\right) * l_r \ldots\ldots\ldots\ldots\ldots\ldots\ldots. + (F_{x2R} - F_{x2L}) * \frac{S}{2} \ldots\ldots\ldots.. \\ + \left(F_{y1L} - F_{y1R}\right) * \sin(\delta) * \frac{S}{2} \ldots\ldots\ldots.. + (F_{x1L} + F_{x1R}) * \sin(\delta) * l_f)/I_{zz} \end{aligned} \tag{A.3}$$

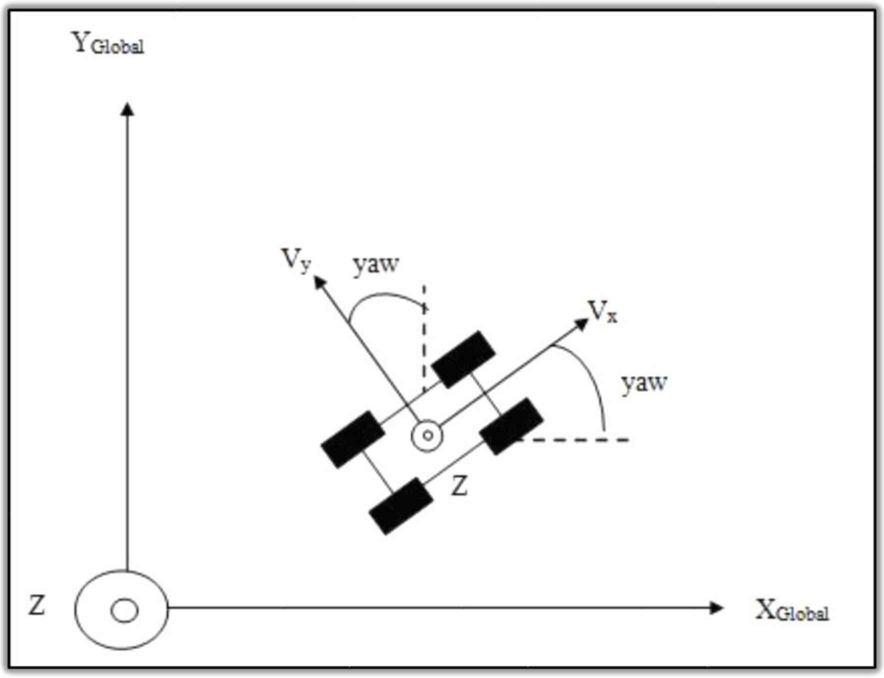

Figure 22 - Global and vehicle local coordinate system

Appendix B Free body diagrams

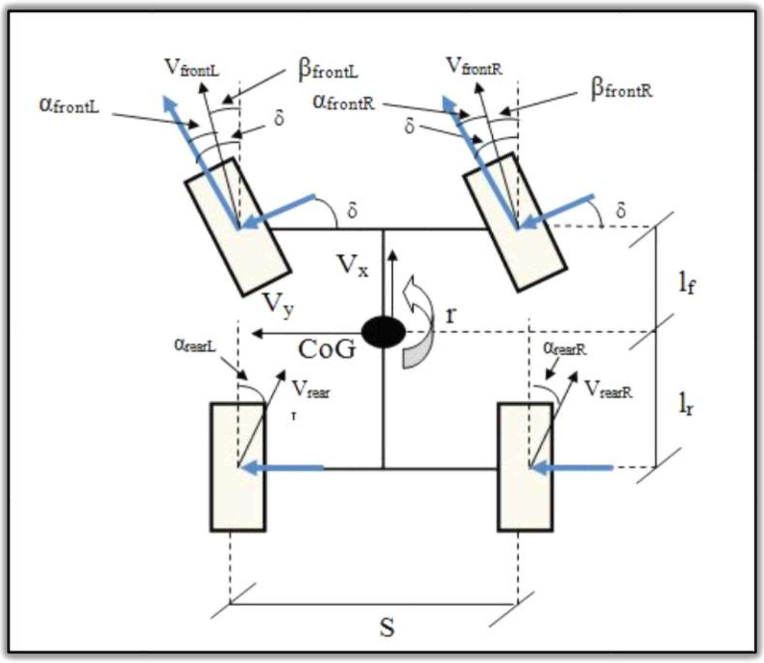

Figure 23 - Relation of lateral slip angles

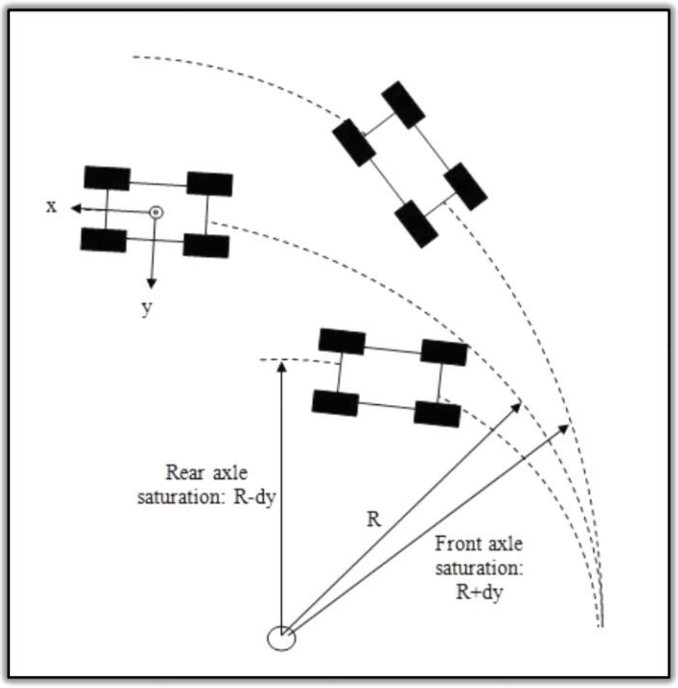

Figure 24 - Vehicle trajectory after axle saturation

Appendix C Tire model

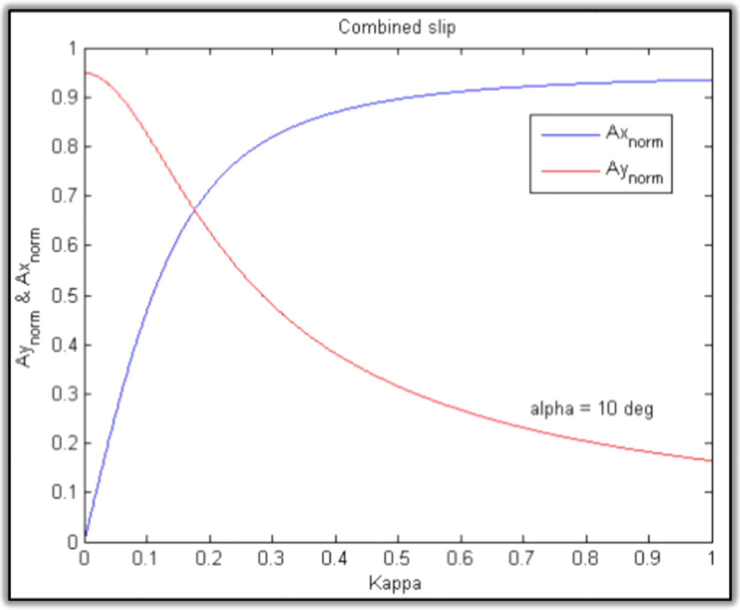

Figure 25 - Combined slip with varying kappa and constant alpha

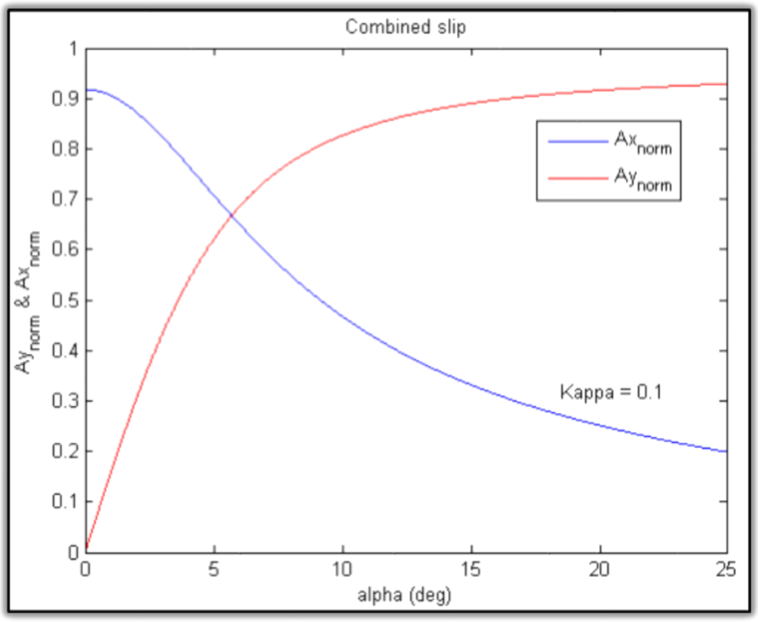

Figure 26 - Combined slip with varying alpha and constant kappa

Appendix D Simulink model blocks

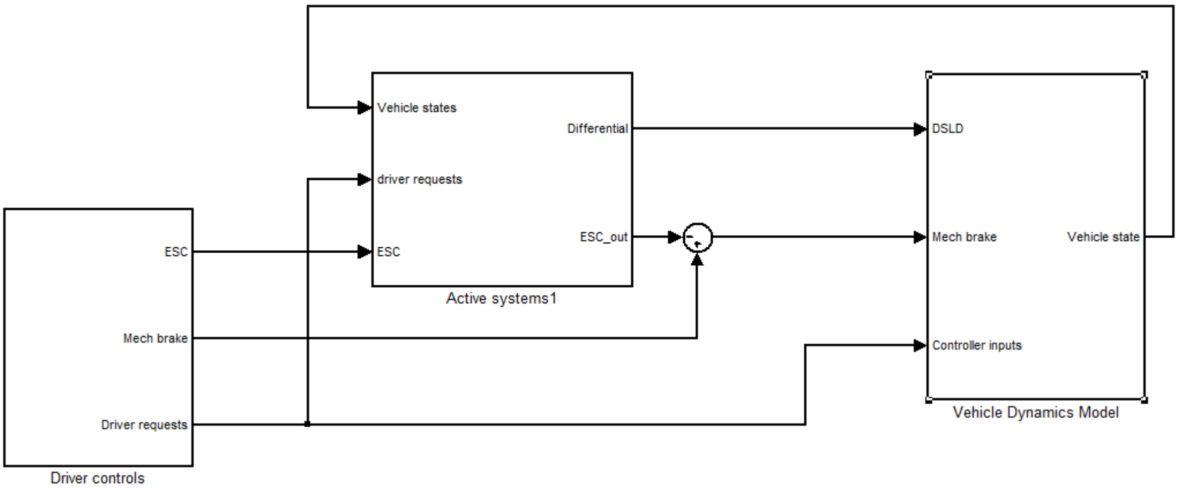

Figure 27 - Modified simulation model

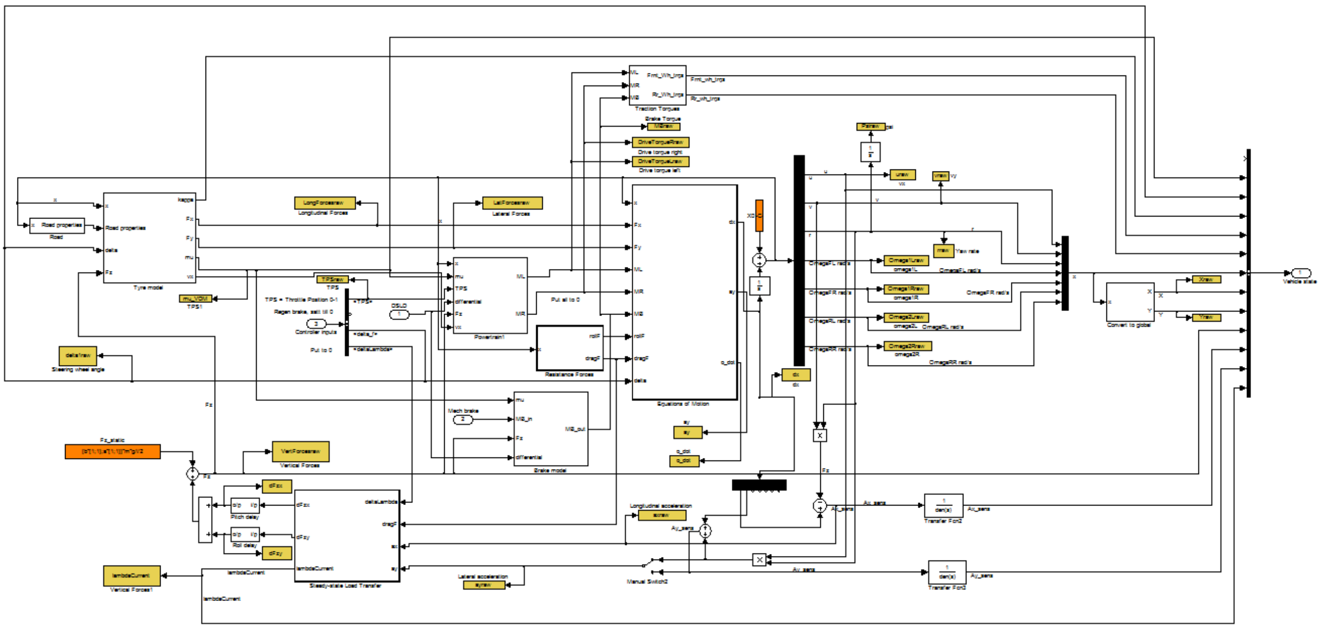

Figure 28 - Inside vehicle dynamics subsystem

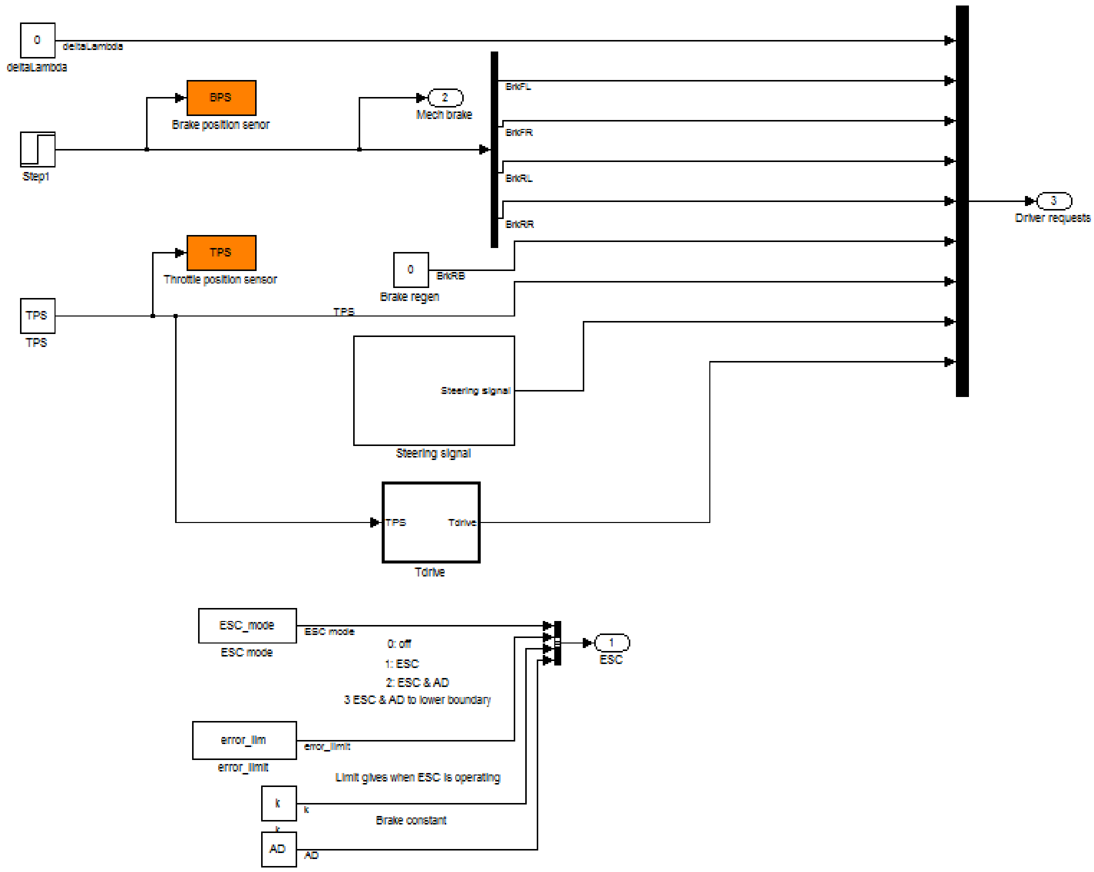

Figure 29 - Inside driver subsystem

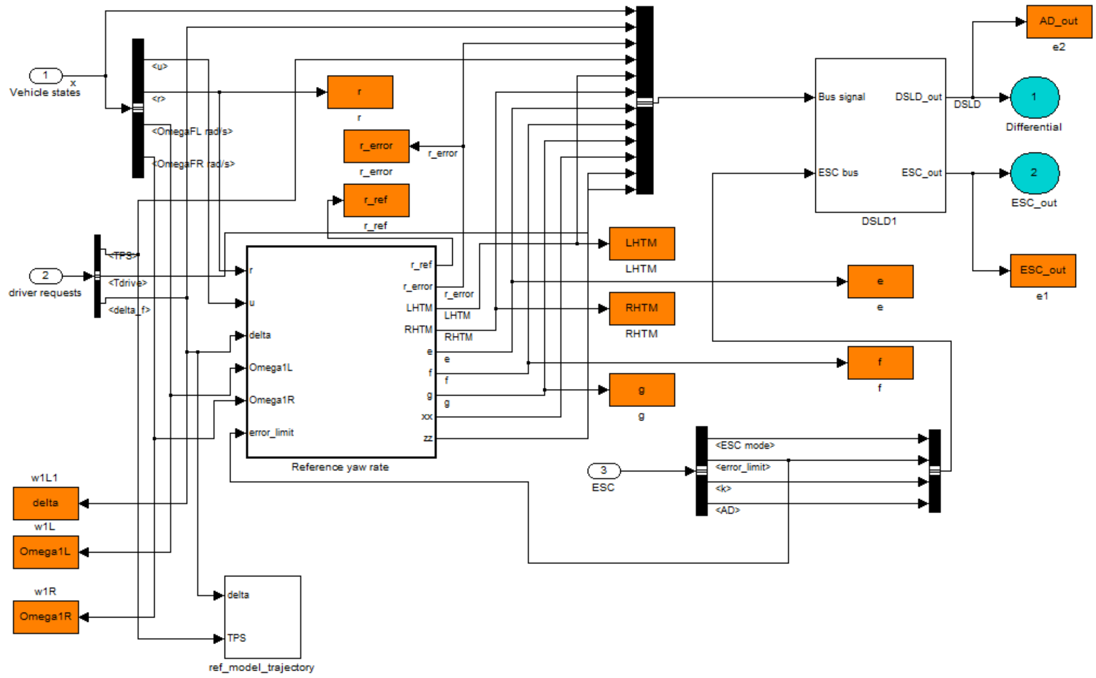

Figure 30 - Inside active systems subsystem

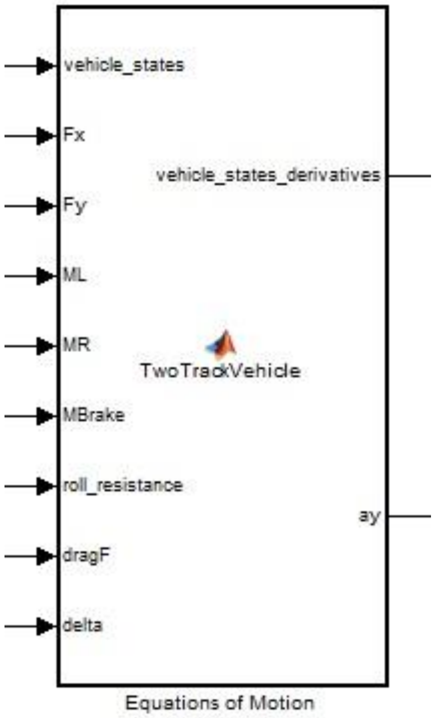

Figure 31 - Equations of motion subsystem

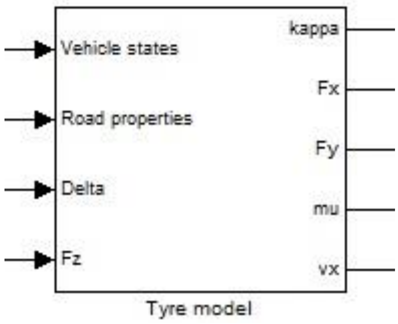

Figure 32 - Tire model subsystem

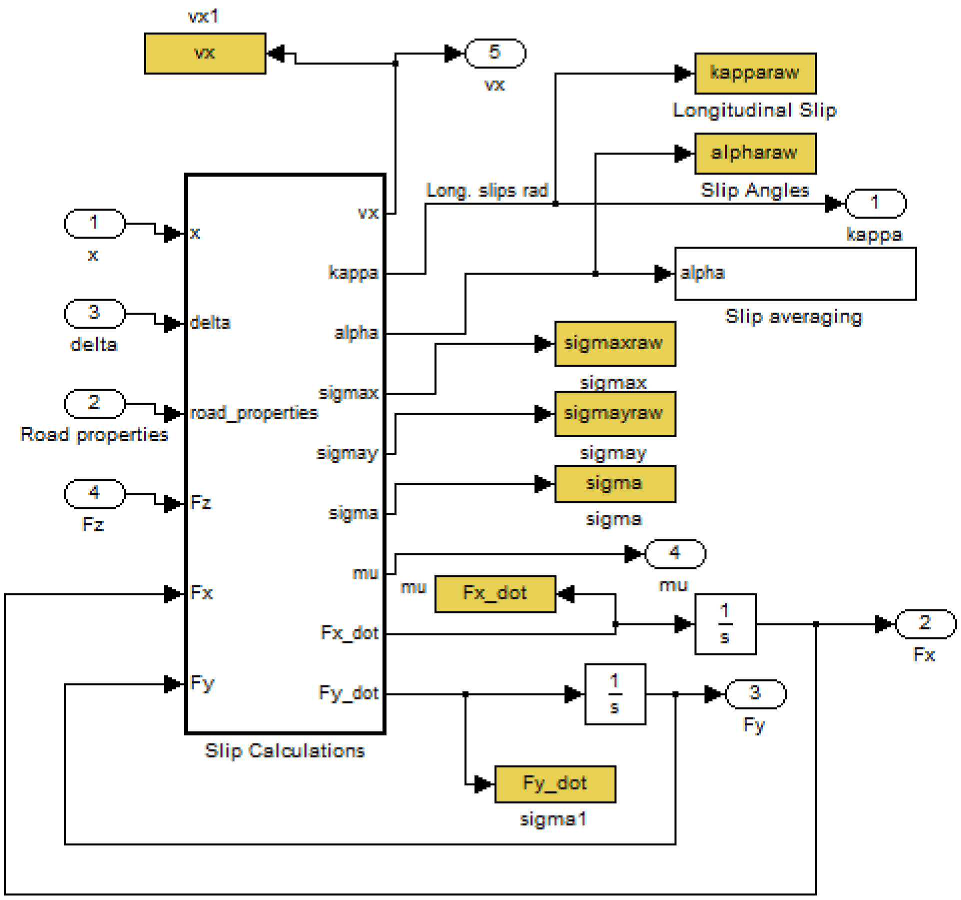

Figure 33 - Inside tire model subsystem

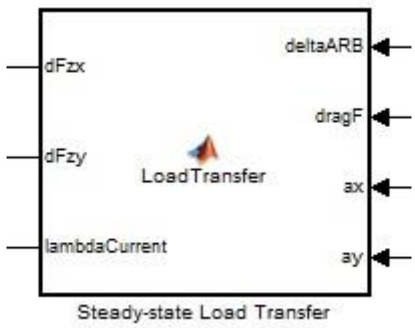

Figure 34 - Load transfer subsystem

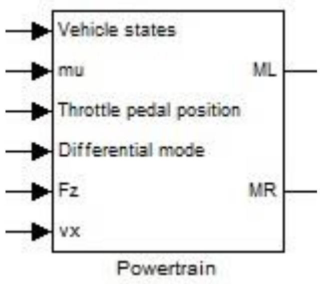

Figure 35 - Powertrain subsystem

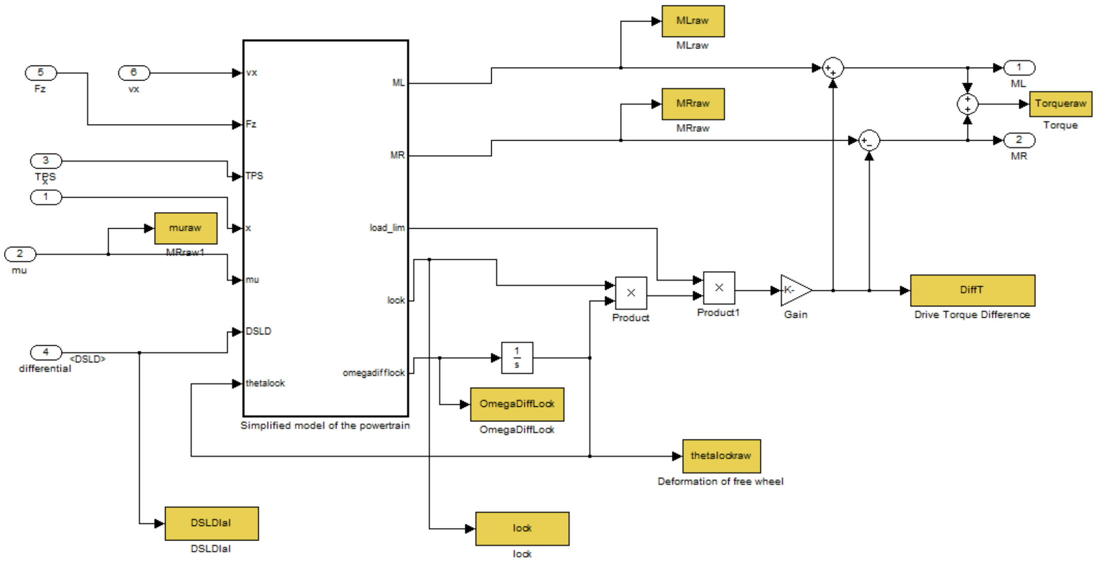

Figure 36 - Inside powertrain subsystem

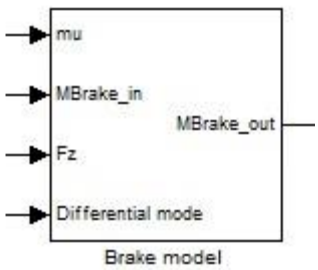

Figure 37 - Brake subsystem

Appendix E Brake actuation

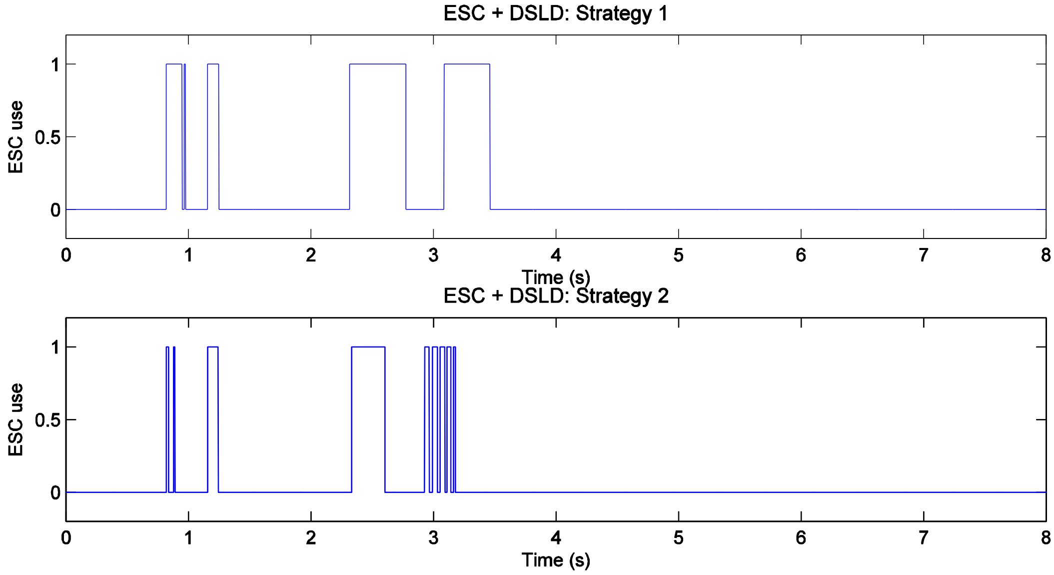

Figure 38 - ESC intervention for yaw error limit = 0.08, DSLD

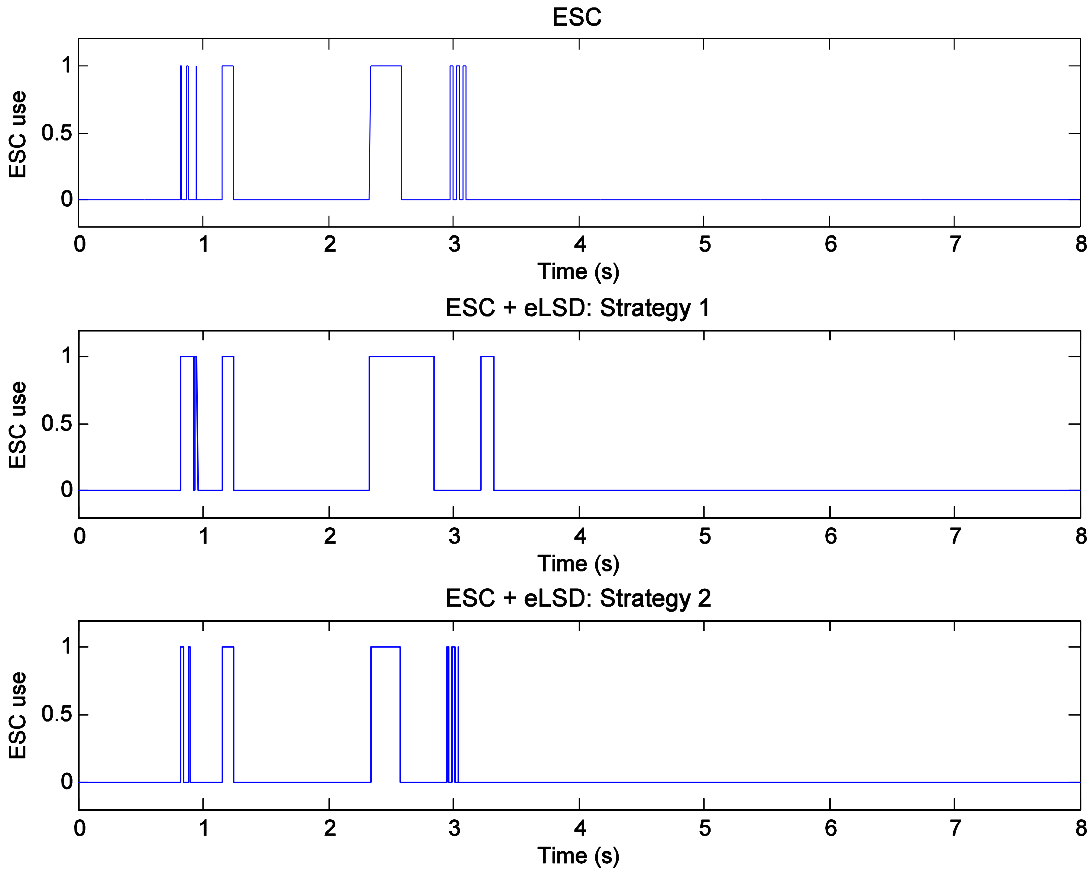

Figure 39 - ESC intervention for yaw error limit = 0.08, eLSD.

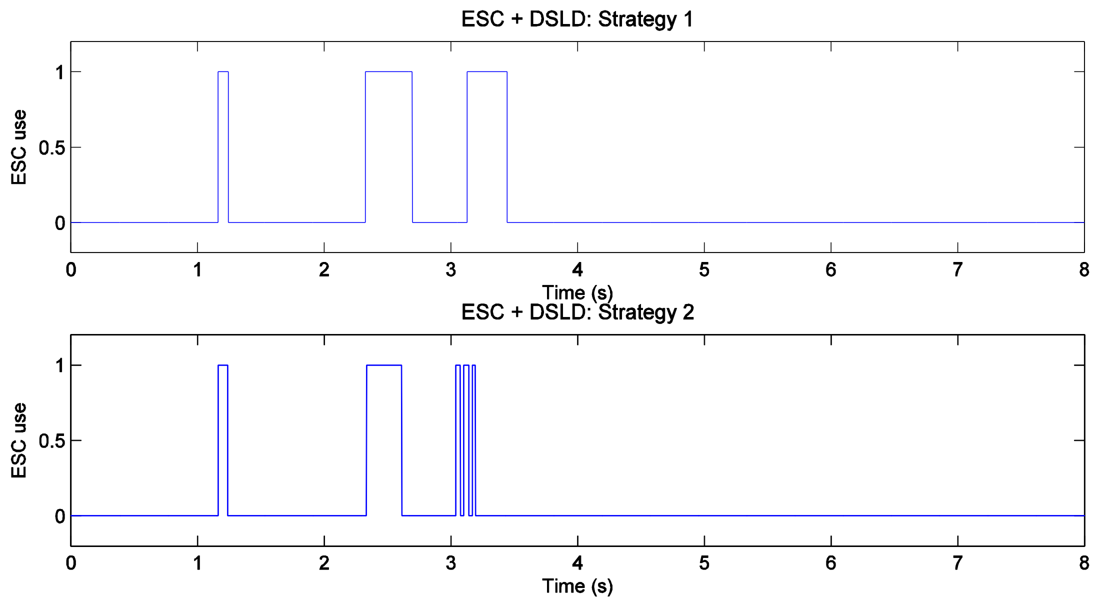

Figure 40 - ESC intervention for yaw error limit = 0.12, DSLD.

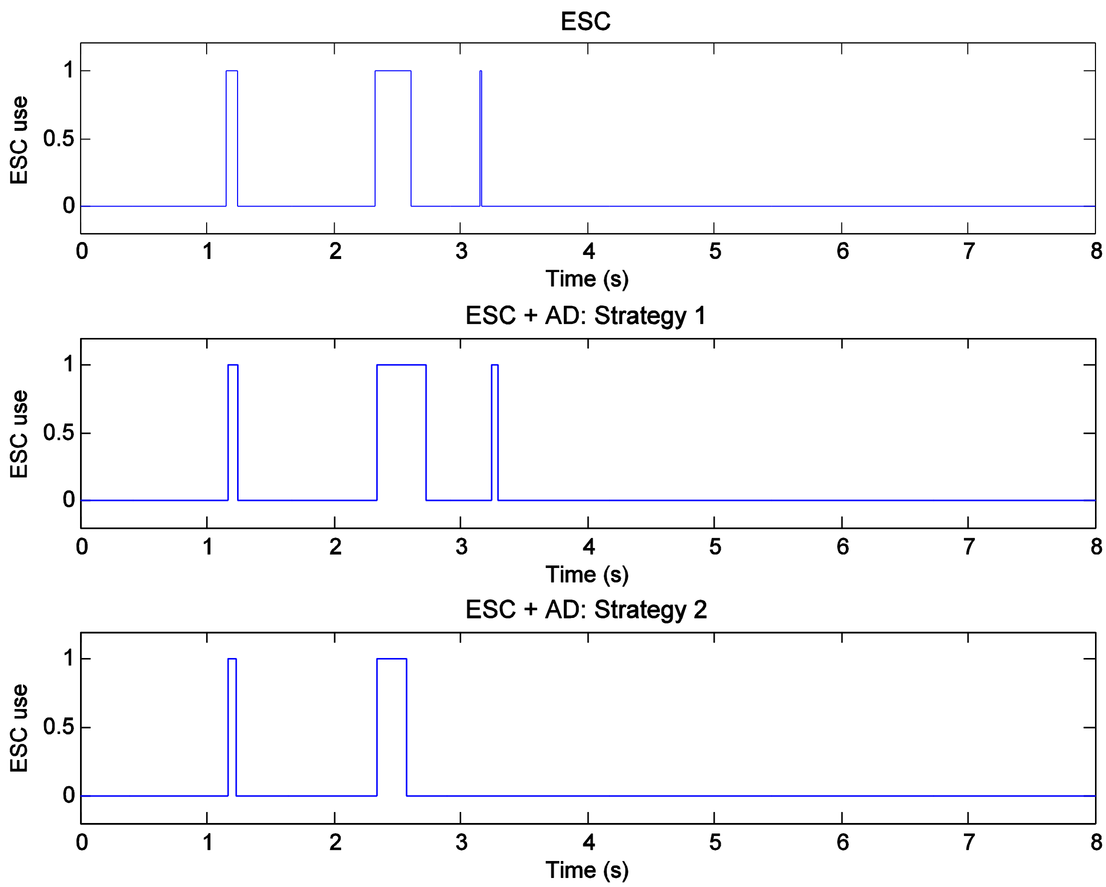

Figure 41 - ESC intervention for yaw error limit = 0.12, eLSD.

Appendix F Vehicle data

Table A – Vehicle Data (Klomp, 2007)

Property	Variable	Unit	Value
Vehicle Parameters			
Vehicle Mass	*a*	[kg]	**1675**
Yaw Moment of Inertia	*Izz*	[kgm^2]	**2617**
Wheel Moment of Inertia	*Iw*	[kgm^2]	**1**
Wheel Base [m]	*l*	[m]	**2,675**
Distance from CoG to Front Axle	*a*	[m]	**1,07**
Distance from C0G to Rear Axle	*b*	[m]	**1,605**
Height of CoG above Roll Axis	*h0*	[m]	**0,43**
Front Roll Center Height	*h1*	[m]	**0,045**
Rear Roll Center Height	*h2*	[m]	**0,1**
Track Width Front	*t1*	[m]	**1,517**
Track Width Rear	*t2*	[m]	**1,505**
Tyre Radius	*Rw*	[m]	**0,316**
Steering Ratio	*nst*	[-]	**15,9**
COG Height	*h*	[m]	**0,5025**
Frontal Area	*AF*	[m^2]	**2,17**
Drag Coefficient	*CD*	[-]	**0,3**
Coefficient of rolling resistance	*fr*	[-]	**0,01**
Cornering stiffness for reference model front	*Cf*	[N/deg]	**1500**
Cornering stiffness for reference model rear	*Cr*	[N/deg]	**1500**
Suspension Data			
Total Roll Stiffness	*c_phi*	[N/rad]	**70000**
Front/Total Roll Stiffness Coefficient	*lambda*	[-]	**0,51**
Front Roll Stiffness	*cphi1*	[N/rad]	**35700**
Rear Roll Stiffness	*cphi2*	[N/rad]	**34300**
Tyre data			
Friction	*mu1*	[N^-1]	**0,00006**
Lateral Stiffnes Parameter 1	*c0*	[1/rad]	**21,3**
Lateral Stiffnes Parameter 2	*c1*	[N^-1]	**0,000111**
	mu0		**0.95**
	C		**1.5**
	E		**0.7**
Rated Load	*Fz0*	[N]	**4000**
Vertical Load at one front Wheel	*Fzf*	[N]	**4930**
Vertical Load at one rear Wheel	*Fzr*	[N]	**3286**
Engine data			
Maximum Engine Torque	*Temax*	[Nm]	270
Minimum Engine Torque	*Temin*	[Nm]	-70
Third Gear Ratio	*i3*	[-]	1,179
Fourth Gear Ratio	*i4*	[-]	0,894
Final Gear Ratio	*ifinal*	[-]	4,059