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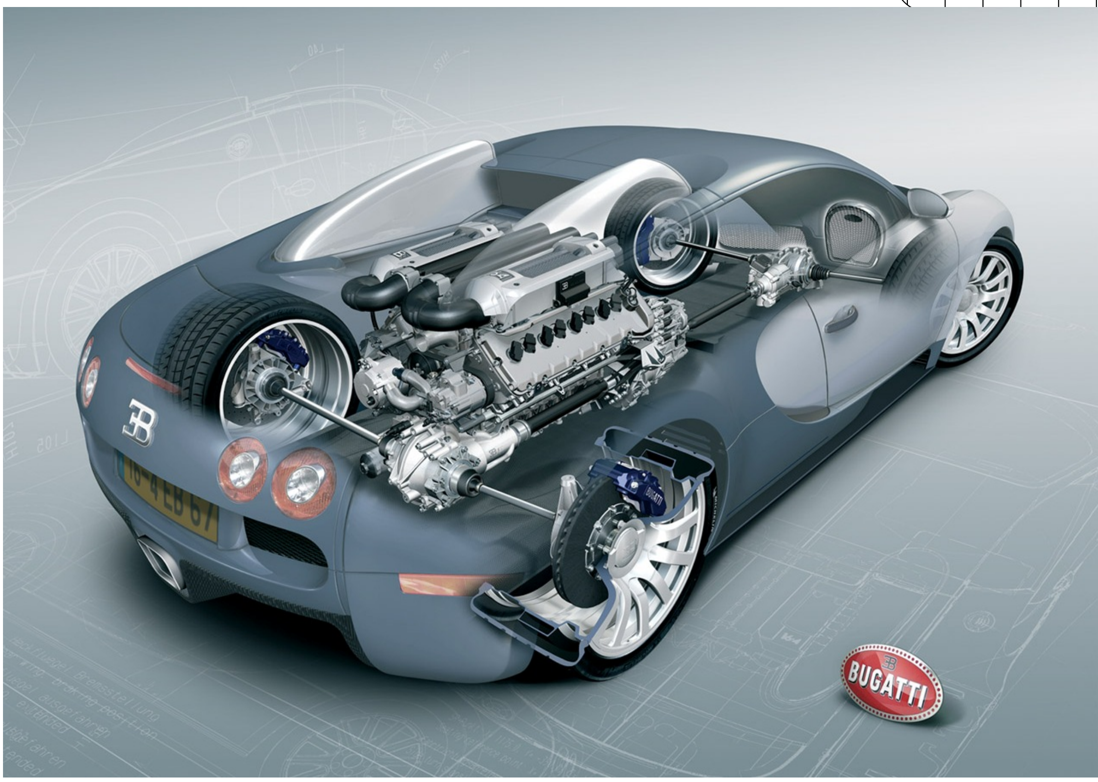

Analysis of vehicle acceleration and cornering performance with the Direction Sensitive Locking Differential (DSLD)

Master's Thesis in Automotive Engineering

MATTIAS CARLSSON
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CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
The 2006 Bugatti Veyron with an eLSD

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Abstract

The purpose of this Master's Thesis was to, through simulations, evaluate the advantages of the Direction Sensitive Locking Differential, DSLD, in various driving situations. The purpose was also to simulate situations that indicates possible risks of a mechanically locked differential. A vehicle model with chassis and drivetrain and a driver model that can follow a predefined path and given speed curves were developed in MATLAB/Simulink. Then a number of driving situations were simulated in MATLAB/Simulink showing different effects of the DSLD and how it affected the performance and maneuverability of the vehicle.

The results of the simulations show that the DSLD preferably is positioned on the front axle in a four wheel driven vehicle and the DSLD makes the largest difference in cornering ability and acceleration in a front wheel driven vehicle. The DSLD will both contribute to produce more power/grip/yaw when accelerating in corners as well as it will reduce yaw motion when in oversteered situations, such as the Sine with Dwell.

No problems has been shown with the DSLD remaining locked in the a prefered direction. It has followed the quickest reactions of the driver and the fastest movements of the vehicle without any problems. The worst case simulation showed that even when the DSLD was forced to lock in the wrong direction, it wasn't any notable loss in cornering ability when unlocking it with a short brake activation.

What needs to be developed further is the interactions and cooperation between the DSLD and the brake, chassis and powertrain/engine control and the other systems that already exists in a vehicle. By developing these interactions the maximum effect of the differential can be achieved and the risk that the systems will work against each other will be reduced. The main thing that is needed is a signal from the DSLD to the other systems with the information whether and how the differential is locked and then the other systems will have to take that into account.

Keywords: DSLD, Direction Sensitive Locking Differential, Vehicle Dynamics

Analys av accelerationsprestanda och kurvtagningsförmåga hos fordon med en riktningskänslig låsningsbar differential (DSLD)
Examensarbete inom mastersprogrammet Automotive Engineering
MATTIAS CARLSSON
MARKUS TUNLID
Institutionen för tillämpad mekanik
Avdelningen för fordonsteknik och autonoma system
Chalmers Tekniska Högskola

Sammandrag

Examensarbetets syfte var att genom simuleringar av olika körsituationer visa på fördelar i ett fordons prestanda med en riktningskänslig låsningsbar differential (DSLD) men också att simulera situationer som visar på eventuella risker i manövrerbarhet med en mekaniskt låst differential. En fordonsmodell med chassi och drivlina och en förarmodell som kan följa en given bana med en given hastighetsprofil togs fram i MATLAB/Simulink. Därefter simulerades ett antal körsituationer som visade på olika effekter av DSLD'n och hur den påverkade bilens prestanda och manövrerbarhet.

Resultaten av simuleringarna visar att DSLD'n med fördel placeras på framaxeln i ett fyrhjulsdrivet fordon och att den ger störst skillnad i acceleration och kurvtagningsförmåga i ett framhjulsdrivet fordon. Den bidrar både till att ge mer kraft/grepp/yaw vid gaspådrag i kurvor likväl som den fungerar till att dämpa rörelser vid överstyrning som i t.ex. Sine with Dwell.

DSLD'n har inte visat några problem med att den stannar kvar låst i fel riktning och motverkar en eventuell sväng, utan den följde även de snabbaste omställningar som föraren och fordonet gjorde. Simuleringar av den på förhand befarade sämsta körsituationen visade att även när DSLD'n tvingas till att låsa i fel riktning så är det inte några större förluster i kurvtagningsförmåga att låsa upp den med hjälp av en kort bromsaktivering.

Det som behöver utvecklas är interaktionen och samarbetet mellan DSLD'n och ABS, traction control, ESP, motor-styrning och de övriga systemen som redan finns i dagens fordon. Genom att utveckla den kopplingen kan maximal effekt av DSLD'n uppnås och risken för att systemen ska motverka varandra minskar.

Nyckelord: DSLD, Riktningskänslig låsningsbar differential, fordonsdynamik

Contents

Preface

This Master's Thesis was carried out at the AWD Control Software & Vehicle Dynamics group at Haldex Traction AB in Landskrona, Sweden and was supported by the department of Applied Mechanics at Chalmers University of Technology in Göteborg, Sweden. The work was performed from February to November 2009 and the report was completed in April 2011.

We would like to thank our supervisors at Haldex Traction, Tord Diswall and Niklas Westerlund, for giving us the opportunity to do our Master's Thesis work in their group and for their feedback during the project. We also would like to thank our supervisor at Chalmers, Dr. Mathias Lidberg, for his feedback on our work and for the discussions regarding vehicle dynamics.

Finally we would like to thank the inventor of the DSLD, Jonas Alfredsson, who has contributed with a lot of ideas and thoughts that have increased our understanding of the DSLD itself and the possible benefits of using it as well as our understanding of vehicle dynamics in practice.

Göteborg June 2011
Mattias Carlsson, Markus Tunlid

Notations

Uppercase Letters

A	state matrix
B	input matrix
C	output matrix
C_α	cornering stiffness
$C_{\alpha f}$	cornering stiffness front axle
$C_{\alpha r}$	cornering stiffness rear axle
C_λ	longitudinal stiffness
D	input to output coupling matrix
F_x	longitudinal force
F_y	lateral force
F_z	vertical load on each wheel
I_{zz}	yaw moment of inertia
K_{us}	understeer coefficient
N_f	static load on the front axle
N_r	static load on the rear axle
V_x	vehicle longitudinal velocity
V_y	vehicle lateral velocity

Lowercase Letters

a_x	longitudinal acceleration
a_y	lateral acceleration
$c_{\varphi f}$	roll stiffness front axle
$c_{\varphi r}$	roll stiffness rear axle
g	gravity
h'	distance between roll axis and CoG
h_f	roll center height front axle
h_r	roll center height rear axle
l	length between front and rear axle
l_f	length from CoG to front axle
l_r	length from CoG to rear axle
m	vehicle mass
r	yaw rate
$\dot{r}$	yaw acceleration
r_e	free rolling radius
s_f	width from CoG to a front wheel
s_r	width from CoG to a rear wheel
u	vehicle longitudinal velocity
v	vehicle lateral velocity

Greek Letters

α	slip angle
α_f	slip angle front axle
α_r	slip ange rear axle
δ	wheel steering angle
λ	longitudinal slip
ϕ	roll angle
ψ	yaw angle
ω	wheel rotational speed

Abbreviations

ABS	Anti-looking Brake System
BoS	Beginning of Steer
CoG	Center of Gravity
CoS	Completion of Steer
DSLD	Direction Sensitive Locking Differential
ESP	Electronic Stability Program
LSD	Limited Slip Differential
eLSD	Electronic Limited Slip Differential
SWA	Steering Wheel Angle
WSA	Wheel Steering Angle

1 Introduction

As an introduction to this thesis work a short background about the Direction Sensitive Locking Differential (DSLD) and what previously has been done in this field is presented. Also the purpose, approach and delimitations of the work are being presented.

1.1 Background

The DSLD was invented by Jonas Alfredsson who applied for the patent in 2005. The main reason for the invention was that the open differential normally used in road going vehicles has one functional problem and that the existing solutions for this problem are not that good. The open differential gives both driving wheels the same amount of torque at all time. The problem occurs when one of the driving wheels looses traction and starts to speed up. The wheel will then spin and since it's not possible to transfer torque to the other wheel, both will have a reduced possibility to generate torque [Alfredsson, 2006].

This is the main drawback of the open differential and it's the reason for the development of differentials that are possible to lock to some extent, some even completely. The problems with most of these solutions are that they use friction in some way. This reduces the efficiency of the vehicle and some differentials need to be controlled using micro processors at all time.

The DSLD is working in a slightly different way and it's either open or locked. Some control strategies have been developed and some basic simulations and comparisons to Limited Slip Differentials, LSD's, have been performed by Carlén and Yngve. Their work shows that the concept of the DSLD is working but their simulations are basic, just one maneuver, and they don't have a model of the differential [Carlén and Yngve, 2006].

A prototype of the differential was constructed by Brolin et al. following the ideas that Jonas Alfredsson presented in the patent. That resulted in the first prototype of a DSLD that was possible to fit into a vehicle. They fitted the DSLD in a formula student car and was able to drive the vehicle a couple of laps at a small race track [Brolin et al., 2008]. Furture test laps with the DSLD in the formula student car was performed by Palmenäs et al. and a control system for the DSLD using lateral acceleration was constructed. Even though the control system for the DSLD was fairly simple it was clear that the DSLD behaved in a good way and that it contributed to the vehicle performance as intended [Palmenäs, 2008].

All together this is an interesting field both regarding the possibilities for better performance of the vehicle and the possibility to use the engine power in a better way.

1.2 Purpose

The purpose of this work is to evaluate the performance of a vehicle with a DSLD compared to a vehicle with an ordinary open differential. The evaluation should be performed in more advanced driving situations compared to what has been done in previous works. Comparisons should be done for a front, a rear and a four wheel driven vehicle. It should be able to simulate the DSLD as the vehicle goes through multiple turns following each other in different ways, with varying throttle position.

The worst possible case should be simulated. Either if the interaction between the Electronic Stability Program (ESP) and the DSLD doesn't work or if the differential doesn't behave as intended. The risk is that the differential will be locked in the wrong direction and thereby increase understeer when it's not intended to.

1.3 Approach

To be able to simulate more advanced driving situations a complete vehicle model is needed. It will include a driveline together with the DSLD, a driver and the chassis characteristics. This model will be developed in MATLAB/Simulink based on a drive line model provided from Haldex. The driver should be able to follow a specific path at a given speed. The driver output will be the throttle, clutch, brake, gear and Steering Wheel Angle (SWA) and together with the friction from the road this will be the inputs that the vehicle should consider.

For evaluation of the model itself a test path is to be built. This simulation will force the model into extreme situations where it is easier to make sure that the DSLD locks and unlocks as intended. Testing the model at this path will also point out potential problem situations and situations where the differential needs to interact with the ESP of the vehicle. This simulation is used only for evaluating the model and no results will be presented from this test.

The differential will be evaluated using the performance of the complete vehicle model where the interesting is the difference between the one with and the one without the DSLD. The exact values of angles, accelerations and so on isn't so important as the comparison between the booth cases. To make the evaluation of the DSLD easier the simulations are divided into a couple of shorter driving situations that focuses on different criterions for the DSLD and it's influences on the vehicle.

1.4 Delimitations

To reduce the amount of work and to get the results from the simulations influenced only by what's most important, the DSLD, there has been some delimitations in this work. The delimitations are also done to reduce possible errors in functions and systems that are more complicated and complex than what can be handled within this kind of work.

As described the DSLD is active when approaching the handling limit of the vehicle. When reducing the power from the engine or braking a spinning wheel the effect of the differential will be reduced. This reduction makes it harder to decide if it's the differential that isn't contributing correct or if it's the system that's reducing the power or braking the wheel that isn't tuned properly. Therefor these systems are not simulated in this work:

- Traction Control – reduces the engine power (Engine Intervention) or brakes a spinning wheel. Engine Intervention is useful if both wheels on the driving shaft with a locked differential is spinning.

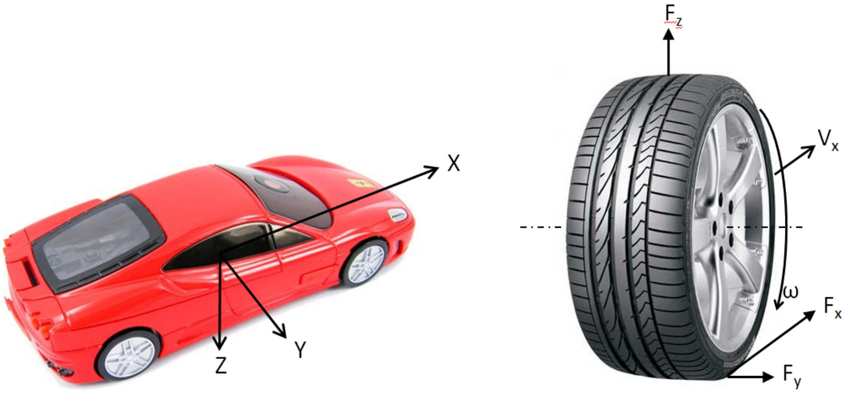
- ESP – brakes one specific wheel when the vehicle is under- or oversteering above a certain limit.

For the control system for the DSLD some signals are considered as known. This could be either from sensors or estimated values. The signals are:

- The ground to tire friction coefficient μ – estimated from algorithms according to, for example [Alvarez et al., 2005, Gustafsson, 1997].
- Wheel speeds ω – measured directly with wheel speed sensors.
- Vehicle velocity V_x – measured directly or estimated by the wheel speeds.

2 Vehicle and tire dynamics

In this section some basic theory regarding vehicle dynamics is presented that is needed for the following work. First the coordinates used for the vehicle and for the wheels are defined as in Figure 2.1.



(a) Vehicle coordinates (b) Tire coordinates

Figure 2.1: Vehicle and tire coordinates

The equations that follows will be presented as:

$$\begin{bmatrix} \text{front left} \\ \text{front right} \\ \text{rear right} \\ \text{rear left} \end{bmatrix}$$

2.1 Planar vehicle motion and load transfer

When a vehicle is moving a resistant force is generated that counteracts the motion of the vehicle. This force consists of two main parts, rolling resistance that comes from the tires and aerodynamic resistance that comes from the air flow around the vehicle. These forces are presented in equation 2.1. The air resistance increases with the square of the speed and it's therefore of great interest in vehicle dynamics. The rolling resistance is also affected by the speed since the coefficient of rolling resistance isn't constant. But the influence is so small that it is neglected.

$$\begin{aligned} F_{x\text{ roll}} &= fmg \\ F_{x\text{ air}} &= \frac{C_d \rho_{\text{air}} A_f V_x^2}{2} \end{aligned} \tag{2.1}$$

To be able to present the path that the vehicle travels or to feed the driver model with information about where it is supposed to drive, the local vehicle coordinates has to be transformed to earth fixed coordinates. The derivation can be read in e.g. [Carlén and Yngve, 2006] and the transformation from local to global velocities can be written as:

$$
\begin{aligned}
\frac{dX}{dt} &= u\cos(\psi) - v\sin(\psi) \\
\frac{dY}{dt} &= u\sin(\psi) + v\cos(\psi) \\
\frac{d\psi}{dt} &= r
\end{aligned}
\tag{2.2}
$$

Another important concept in vehicle dynamics is the understeer coefficient, K_{us}. For steady state turning K_{us} is dependent on the static load of each axle and the axle characteristics and can be written as:

$$
K_{us} = \frac{N_f}{C_{\alpha f}} - \frac{N_r}{C_{\alpha r}} \tag{2.3}
$$

The understeer coefficient can also be thought of as the gradient of steer angle with respect to the lateral acceleration and can be defined at high speed turning as:

$$
\frac{d\delta}{da_y} = K_{us} = \frac{d(\alpha_f - \alpha_r)}{da_y} \tag{2.4}
$$

If the coefficient has a value above zero the vehicle is referred to as understeered. If the value is zero the vehicle is neutral steer and if the value is below zero the vehicle is over steered.

When the vehicle accelerates, brakes or turns the normal load on each tire changes from the static value. For example during acceleration the normal load on the front wheels decreases and the normal load on the rear wheels increases. In the same way the normal load on the outer wheels increase during cornering and the normal load on the inner wheels decreases. For a rigid vehicle without any suspension the normal forces for each wheel are:

$$
\vec{F}_z = \underbrace{\frac{m}{2l}\begin{bmatrix} l_r \\ l_r \\ l_f \\ l_f \end{bmatrix} g}_{\text{static load}} + \underbrace{\frac{m}{2l}\begin{bmatrix} -h \\ -h \\ h \\ h \end{bmatrix} a_x}_{\text{longitudinal load transfer}} + \underbrace{\frac{mh}{2ls}\begin{bmatrix} l_r \\ -l_r \\ -l_f \\ l_f \end{bmatrix} a_y}_{\text{lateral load transfer}} \tag{2.5}
$$

If taking the roll dynamics and the suspension of the vehicle into account and assuming steady state cornering the lateral load transfer can be written as:

$$
\Delta F_{zi}\mid_{\dot{\phi}=0} = \frac{1}{2s_i}\left(\frac{c_{\phi i}}{c_{\phi f} + c_{\phi r} - mh'g}h' + \frac{l - l_i}{l}h_i\right) m a_y \quad , i = f, r \tag{2.6}
$$

Finally, by assuming constant acceleration or deceleration, pitch dynamics can be neglected [Klomp, 2008]. The normal forces on each tire then becomes:

$$\vec{F}_z = \begin{bmatrix} \frac{m(gl_r - a_x h)}{2l} + \Delta F_{zf} \\ \frac{m(gl_r - a_x h)}{2l} - \Delta F_{zf} \\ \frac{m(gl_f + a_x h)}{2l} - \Delta F_{zr} \\ \frac{m(gl_f + a_x h)}{2l} + \Delta F_{zr} \end{bmatrix} \tag{2.7}$$

The effect of the lateral load transfer of interest here is the fact that the tire force capacity increases with vertical load degressively. This means that the total capacity of an axle decreases when subjected to lateral load transfer. However, this means that the capacity of the outer wheel still exceeds the capacity of the inner wheel. This effect has large influence when choosing the type of differential to use.

2.2 Tire slip model

Because the only contact between the vehicle and the road is through the tires, the force that is needed to accelerate the vehicle in any direction must be generated in the contact patch between the tires and the road. The other part of the total force influencing the vehicle dynamics is the air resistance which doesn't influence the tires. The lift created from aerodynamic is rather small at low speeds and is therefor neglected.

2.2.1 Longitudinal

The force that a tire generates can be divided in two different parts, longitudinal and lateral. A longitudinal force is created when the wheel is rotating with a slightly different speed than the vehicle is traveling. The difference in speed is called slip and is a very important concept when dealing with vehicle and tire dynamics. The slip of a braking, free rolling and accelerating tire is showed in Figure 2.2.

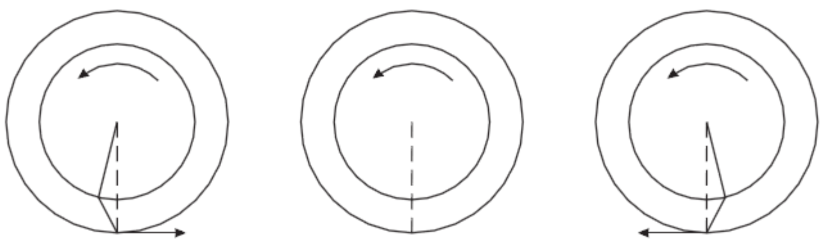


Figure 2.2: Longitudinal slip for braking, free rolling and accelerating

The longitudinal slip for an accelerating wheel is defined as:

$$\lambda = -\frac{V_x - r_e\omega}{r_e\omega} \tag{2.8}$$

and for a braking wheel as:

$$\lambda = -\frac{V_x - r_e\omega}{V_x} \tag{2.9}$$

Slip is positive when accelerating and negative when decelerating. The free rolling radius or effective radius, r_e, is not the same as the actual radius of the wheel. This is due to the deformation caused by the load of the vehicle. r_e is defined as:

$$r_e = \frac{V_x}{\omega} \tag{2.10}$$

The driving force generated from the longitudinal slip is defined, for small longitudinal slip, as:

$$F_x = C_\lambda \lambda \tag{2.11}$$

Where the longitudinal slip stiffness, C_λ, is defined as:

$$C_\lambda = \left.\frac{\partial F_x}{\partial \lambda}\right|_{\lambda=0} \tag{2.12}$$

When the slip increases the force also increases up to a maximum value. The maximum available friction of a tire is reached somewhere a bit below 10% depending on the type of tire and the road surface. This means that for a large slip the tire is more sliding on the surface than rolling and this gives a lower friction as can be seen in Figure 2.3

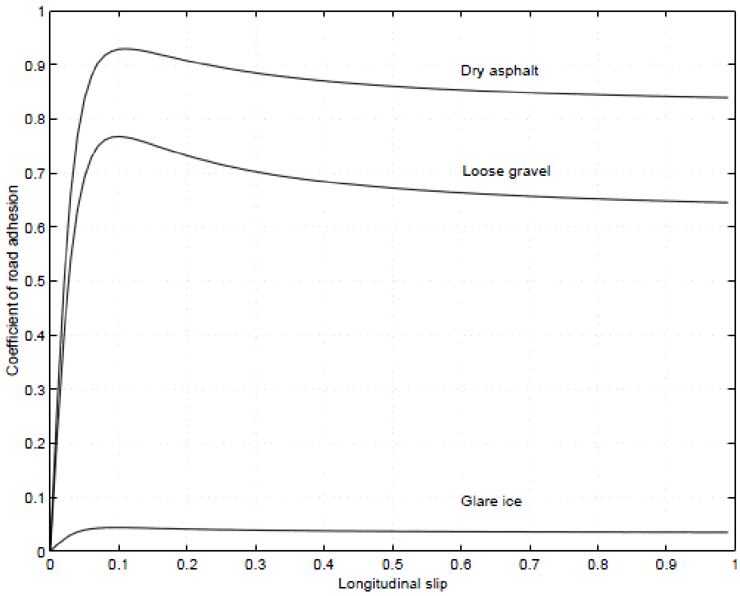


Figure 2.3: Longitudinal friction vs slip

2.2.2 Lateral

If a wheel is heading in a slightly different direction than it's traveling an angle between these two directions is created. This is the slip angle and it generates a lateral force perpendicular to the direction of the wheel, as shown in Figure 2.4.

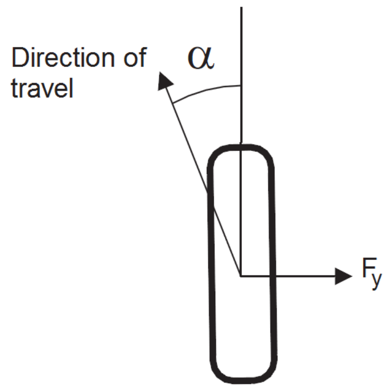


Figure 2.4: Slip angle

The slip angle or the lateral slip is defined as:

$$\alpha = \arctan\left(-\frac{V_y}{|V_x|}\right) \tag{2.13}$$

The four wheels of a vehicle has one specific slip angle each that can be written as:

$$\begin{aligned}
\alpha_{fl} &= \delta_f - \arctan\left(\frac{v + l_f r}{u + s_f r}\right) \\
\alpha_{fr} &= \delta_f - \arctan\left(\frac{v + l_f r}{u - s_f r}\right) \\
\alpha_{rr} &= \delta_r - \arctan\left(\frac{v - l_r r}{u - s_r r}\right) \\
\alpha_{rl} &= \delta_r - \arctan\left(\frac{v - l_r r}{u + s_r r}\right)
\end{aligned} \tag{2.14}$$

Where δ_f is the steering angle at the front wheels, δ_r the steering angle at the rear wheels, r the yaw rate of the vehicle, l_f the length from the Centre of Gravity, CoG, to the front axle, l_r the length to the rear axle, s_f the width from the CoG to a front wheel and s_r the width to a rear wheel. The cornering force generated from the slip angle is defined, for small slip angles, as:

$$F_y = C_\alpha \alpha \tag{2.15}$$

Where the cornering stiffness, C_α, is defined as:

$$C_\alpha = \left.\frac{\partial F_y}{\partial \alpha}\right|_{\alpha=0} \tag{2.16}$$

The lateral force, F_y, from the tire is, for a given load on the tire and for a fix longitudinal slip, depending on the slip angle as shown in Figure 2.5

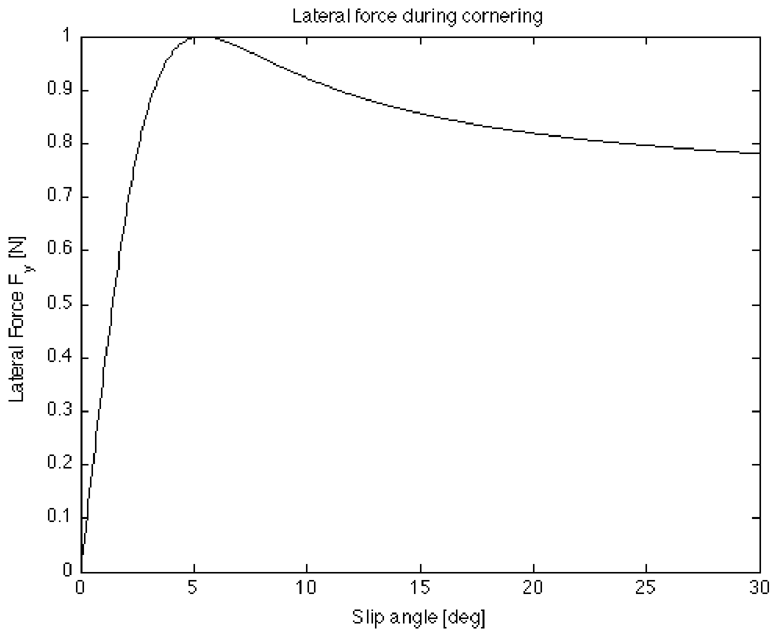


Figure 2.5: Lateral force vs lateral slip

2.2.3 Friction ellipse

As mentioned before a tire can generate forces in two different directions. But the situations where a tire only generates a longitudinal or a lateral force are almost only theoretical and for a vehicle in motion a tire needs to generate forces in both directions at all times. Introducing lateral slip tends to reduce the longitudinal force at a given longitudinal slip and vice versa. The maximum available friction can be described using a so called friction circle, or more correct a friction ellipse, seen i Figure 2.6.

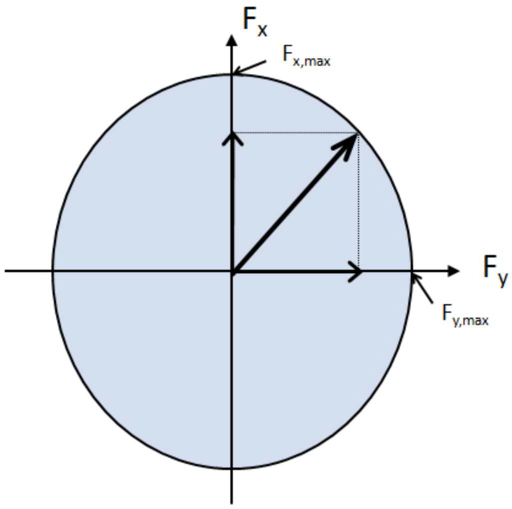


Figure 2.6: Friction ellipse

The friction ellipse can be described using the maximum longitudinal force and the maximum lateral force that the tire can generate. These forces are dependent on the friction, the vertical load and tire properties and combining them with the standard equation of an ellipse the friction ellipse can be described as:

$$\left(\frac{F_x}{F_{x,max}}\right)^2 + \left(\frac{F_y}{F_{y,max}}\right)^2 = 1 \tag{2.17}$$

Another way of describing the decrease in available lateral force is as in Figure 2.7 where it's easy to see that an increase in longitudinal slip for a given slip angle reduces the amount of available lateral force. The values in Figure 2.7 are the results of our slip model presented in Section 3.2.

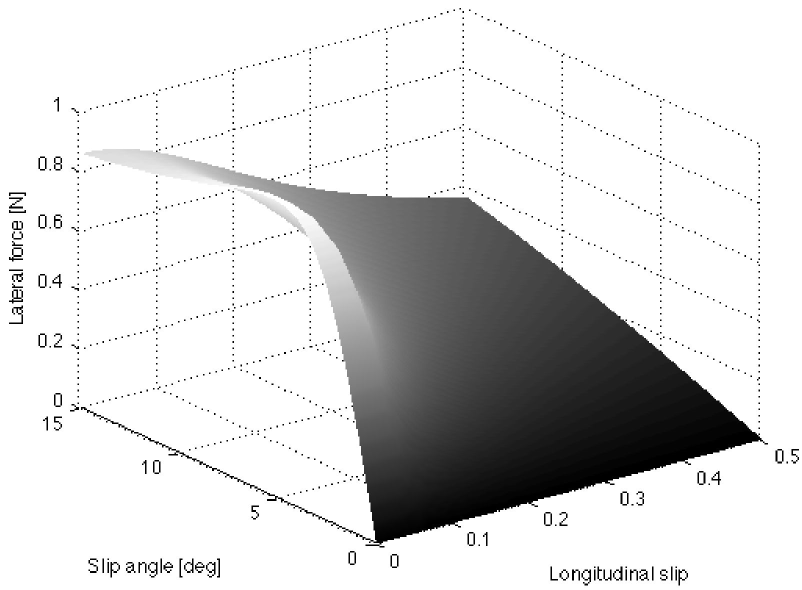


Figure 2.7: Available lateral force depending on longitudinal slip and slip angle

2.3 Differentials

The differential and the final drive in a vehicle have two main purposes. The differential should allow the outer wheel to rotate faster then the inner wheel during cornering and at the same time transfer the drive torque from the engine to the wheels. The final drive should gear down the rotating speed of the drive shaft and gear up the drive torque to the wheels.

There are two main possibilities in how the speeds and torque can be divided. The first is using an open differential that allows the speed between the inner and outer wheel to differentiate and at the same time divides the drive torque evenly between the wheels. The second is to use a rigid axle that keeps the rotating speeds of the wheels the same and divides the drive torque depending on the difference in resistance of each wheel. The torque distributions and speed relations for the open differential and the rigid axle can be written as:

$$\left.\begin{aligned} T_{outer} &= T_{inner} = \frac{T_{drive}}{2} \\ \omega_{in} &= \frac{\omega_{outer} + \omega_{inner}}{2} \end{aligned}\right\} \text{ open differential}$$
$$\left.\begin{aligned} T_{drive} &= T_{outer} + T_{inner} \\ \omega_{outer} &= \omega_{inner} = \omega_{in} \end{aligned}\right\} \text{ rigid axle} \tag{2.18}$$

The main problem with a rigid axle is that it has an understeering effect when cornering at low speeds and low level of lateral acceleration. This is because the inner and outer wheels has the same rotational speed at the same time as they have to travel different distances. This gives that the inner wheel will have a positive slip that is larger than for the outer wheel and at low levels of input torque the outer wheel may even have a negative slip. This will give a larger positive force on the inner wheel than on the outer wheel and this will counteract the turning motion of the vehicle.

The purpose of the open differential is to get rid of this drawback. The open differential is far superior the rigid axle for normal driving. The drawback of the open differential is during cornering and hard acceleration. Since the available longitudinal force on the inner wheel is reduced during cornering, as explained in Section 2.1, and that the drive torque is divided equally also a smaller force can be used on the outer wheel.

In this situation it is preferred to have a rigid axle which will transfer drive torque to the outer wheel and also keep the inner wheel from spinning. When transferring torque to the outer wheel the drawback of the understeering effect is reduced and could even become an oversteering effect. The limit between where it's more preferable with an open differential or a rigid axle is further explained in Section 2.3.

There are a lot of different differentials that tries to combine the advantages of the open differential at low speeds and low levels of acceleration and the rigid axle at high levels of acceleration. They all works as almost open differentials when the differences in rotating speeds between the inner and outer wheels are low. Then when the inner wheel looses it's grip and tends to spin they, in different ways, locks the axles to each other to transfer torque to the outer wheel. The main differences between the differentials are the way they lock the axles. For example limited slip differentials either they are electronically controlled or not uses friction and differences in rotating speed to produce torque.

Both LSD's and Torsen differentials are compromises between the open differential and a rigid axle and are not optimized solutions. The problem that might occur with a LSD is that it can not totaly reduce the differentiation at high levels of torque difference between the driven wheels. Also power losses occurs in form of heat when the differential is slipping. Torsen differentials looses it's effect when the force needed to rotate the inner wheel becomes low and therefore looses it's effect when it's as most needed. Another problem with the Torsen differential is that the complex mechanical function makes it large and rather heavy and it also has the same problem with heat production as the LSD's.

The interesting point where the inner wheel has started to speed up and has reached the same speed as the outer wheel is called the cross-over point and is central in the concept of lockable differentials. Figure 2.8 shows how the forces for the inner and outer wheels are combined when the differential is open and when it's locked. Also the cross-over point is marked at the point where the lines for the open and locked differential match. There are two lines for the open differential and two lines for the locked differential, rigid axle. These represents the outer and inner wheels and the lines between shows which points corresponds to each other.

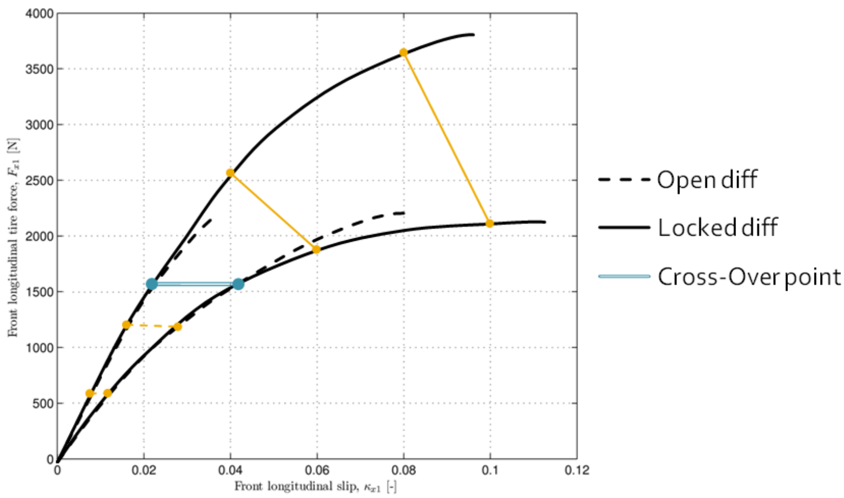


Figure 2.8: Outer wheel at the top left and inner wheel at the down right

Above the cross-over point it's preferable to have a rigid axle and below the cross-over point it's preferable to have an open differential. This is therefore the point when the differential should go from open to locked.

2.4 The DSLD

The DSLD works as either an open differential or a rigid axle, nothing in between. At low speeds and low levels of acceleration the DSLD works as an open differential and sufficient levels of acceleration when the inner wheel tries to speed up during cornering it works as a rigid axle. It doesn't have the limitation in torque

Table 2.1: Working modes of the DSLD

1	open
2	right turn
3	left turn
4	locked

transfer as the LSD's or Torsen differential mentioned above and it doesn't use dynamic friction to generate the torque and therefore it doesn't generate heat.

For an eLSD the control system has to decide how much the differential should be locked at all times whereas the control system for the DSLD only needs to decide if the differential should lock in any direction or be open. The DSLD could be locked in only one direction and it lockes itself at the cross-over point.

The modes that the differential can be set to are presented in Table 2.1. The first mode, open, makes the differential work as an ordinary open differential. Mode four, locked, works as a rigid axle where differentiation in any direction isn't allowed. The right turn or left turn modes allows differentiation in one direction but not in the other. In for example the right turn mode, the left (outer) wheel is allowed to rotate faster than the right (inner) wheel but the right wheel isn't allowed to rotate faster than the left.

In cornering this will make the DSLD work as an open differential until the inner wheel starts to speed up and overtakes the outer wheel. When this happens the DSLD locks and the differential works as a rigid axle. When the drive torque or the cornering decreases or the grip of the inner wheel increases the differential unlocks and acts as an open differential again. During this kind of maneuver the control system for the DSLD is set to only one mode the whole time and the differential locks and unlocks itself at the cross-over point.

The DSLD is an open differential that can be locked using a number of rollers placed between one of the inner shafts and the house of the differential, see Figure 2.9. A couple of magnets controlled by the control system allows the rollers to either rotate with the house or to follow the inner center. If the rollers rotates with the house the differential is open and differentiation is allowed to occur in any direction. If the rollers are allowed to rotate with the center and the center and the house rotates with different speeds the rollers will get wedge together with the inner shaft and the outher wall as the wall of the house is curved. The control system thereby through the magnets can control if any differentiation is allowed or in which direction it's not allowed.

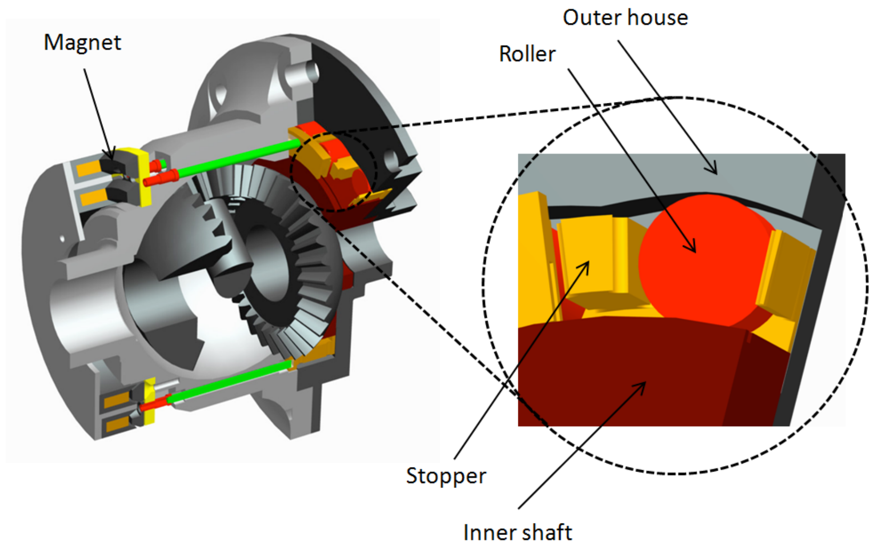


Figure 2.9: The DSLD

This means that when the DSLD is set to mode one the rollers are locked in the middle and differentiation in any direction is allowed. When the control system decides that mode two or three are preferred the rollers are allowed to rotate with the center in one direction. This gives that the DSLD isn't locked in one state or the other just because the mode isn't mode one, there has to be a differentiation in that direction too. When mode four is set by the control system the rollers are free to move in any direction, if there's a difference in rotation speed then the DSLD will lock as a rigid axle. The angle that the rollers will have to rotate is only a few degrees.

3 Vehicle system modeling

The basic vehicle model should include the chassis and the tires and it should take lateral/longitudinal dynamics and load transfer in to account. To be able to follow a path the driver model needs to look ahead and predict where the vehicle will be in a moment. Then the driver model should decide the throttle, brake, clutch, gear and steering wheel angle.

3.1 Driveline

The driveline model includes an engine that supplies torque depending on rotating speed and throttle position. A five-speed manual gearbox determines the gear ratio and a clutch engages or disengages the engine to the gearbox. If the vehicle is a front wheel drive vehicle or an all wheel drive vehicle the gearbox is connected directly to the front differential or, if the vehicle is a rear wheel drive vehicle, via a drive shaft to the rear differential. The differentials splits the torque between the left and right drive shafts that transfers the torque to the wheels. In the all wheel drive vehicle a drive shaft is connected to the front differential transferring torque backward to the rear differential through a clutch. This clutch locks the front and rear differentials to each other if there is any speed differences between the incoming drive shaft and the rear differential. This makes the all wheel driven vehicle front wheel driven as long as the front wheels doesn't spin.

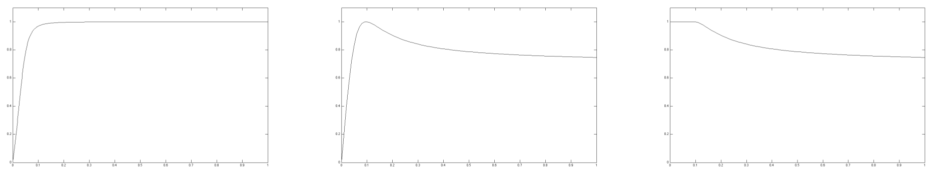
For the driveline models with the DSLD, the DSLD is placed on either the front or the rear axle. The differential torque generated from the DSLD is added at one side and subtracted at the other, after the open differential but before the drive shafts.

3.2 Tire model

The tire forces are calculated in two steps. The longitudinal and the lateral forces are calculated respectively according to the theory in Section 2.2.1 and 2.2.2. These calculated forces are valid if the longitudinal slip and the lateral slip angles are small. But though the DSLD contributes most to the performance of the vehicle on the handling limit of the vehicle the model has to be reasonable valid both at large longitudinal slip and large slip angles.

Pacejka's model for combined slip that is described in Section 2.2.3 works well if the force/slip function is constant from the maximum value towards higher slip values, as in Figure 3.1a. Then the lateral force reduces at high levels of longitudinal slip. But in the magic tire formula, as can be seen in Figure 3.1b, the longitudinal force reduces at high levels of slip and therefor the Pacejka model for combined slip isn't valid in this region. The error comes from that the model compares the total force, longitudinal and lateral, to the maximum force available, $\mu * F_z$, and reduces the lateral force to the level where the total force equals the maximum force. The problem is that when the longitudinal slip increases, at high levels of slip, and thereby the longitudinal force decreases it gives an increased lateral force. This means that the more the wheel spins the more lateral force can be generate and that is obviously not the way it should be.

The problem with Pacejka's model is solved using a maximum combined force/slip function, Figure 3.1c, that has $\mu * F_z$ as maximum force from zero slip up to the level of slip where the magic tire formula has it's maximum value. At higher slip levels the maximum combined function has the same shape as the longitudinal and lateral slip functions.



(a) Pacejka (b) Magic tire formula (c) Maximum combined

Figure 3.1: Force/slip functions

If the total force exceeds the available force from the maximum combined function the total force is reduced to that level. The new total force are then divided between longitudinal and lateral forces as:

$$F_x = \frac{\lambda_x}{\lambda_{tot}}, \quad F_y = \frac{\lambda_y}{\lambda_{tot}} \tag{3.1}$$

Where:

$$\lambda_{tot} = \sqrt{{\lambda_x}^2 * {\lambda_y}^2} \tag{3.2}$$

This may not be the optimal way of deciding the tire forces but it gives results with the right properties meaning that the longitudinal force decreases when a wheel spins and that the lateral force reduces at the same time. The avaliable longitudinal force depending on longitudinal slip and slip angle can be seen in Figure 3.2. As mentioned i Section 1.3 it's the properties of the model that are important so that a good comparison can be made and not the exact values.

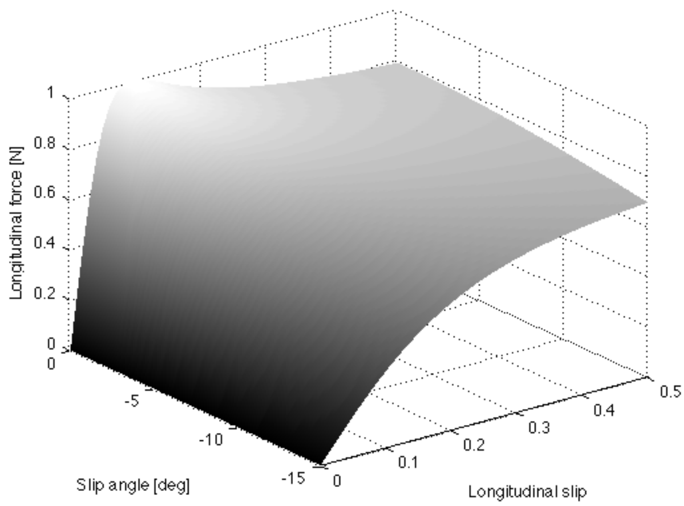


Figure 3.2: Avaliable longitudinal force depending on longitudinal slip and tire slip angle

Another source of error in the tire model is that the slip is calculated and the tire forces are applied instantly. Normally it takes some rotation of the wheel to build up the tire force. This simplification will give slightly faster changes and larger fluctuations in torques and speeds than in reality so the model should be slightly more damped.

3.3 Path

For the different simulations two different kinds of inputs to the vehicle are used. The first one is where all inputs are known before the simulation starts. This can be done for a test like the Sine with Dwell, described in Section 5.2.1, where the vehicle's behavior under a specific steering sequence is of interest. This is a open loop system where no feedbacck to the driver model is needed.

The other one is where the vehicle is supposed to follow a certain road or path. It could be a circle, a straight line or a combination of bends and straights as in the 90 degree turn, described in Section 5.1.3. This is a closed loop system where feedback to the driver model is needed continuously. For the vehicle to be able to follow such a path the X and Y global positions are defined together with the global direction of the path. It can also be of interest for the vehicle to go at different speeds at different sections of the path and on different surfaces, for example ice or asphalt. Therefor the desired velocity at all points at the path is defined together with the friction coefficient for the left/right and front/rear wheels. See Table 3.1 for what's included in the path.

Table 3.1: Path contents

global X position
global Y position
global direction
desired velocity
front left friction
front right friction
rear right friction
rear left friction

3.4 Driver model

For the vehicle to be able to follow the path, described in Section 3.3, a driver model is needed. The driver predicts where the vehicle will be in a certain time-step and compares this position to the path. The driver also compares the direction of travel to the direction of the path. To be able to do that some inputs are needed and they are listed in Table 3.2a. When the driver knows where the vehicle should be according to the path compared to where it will end up according to the present velocity and the present error in direction the driver has the possibility to compensate by adjusting the steering wheel angle. The other outputs that the driver can adjust are presented in Table 3.2b.

The SWA is mechanically linked to the wheel steering angle, WSA, of the front wheels. To prevent the driver from steering too much a limit on the WSA is

Table 3.2: Driver signals

(a) Inputs

global X position
global Y position
vehicle yaw rate, r
longitudinal velocity, V_x
lateral velocity, V_y

(b) Outputs

Steering wheel angle
Throttle
Brake
Clutch
Gear

introduced. The limit corresponds to the WSA that will give the largest steady-state lateral acceleration as a function of velocity. The limit is presented in Figure 3.3 where the shaded area is above the maximum.

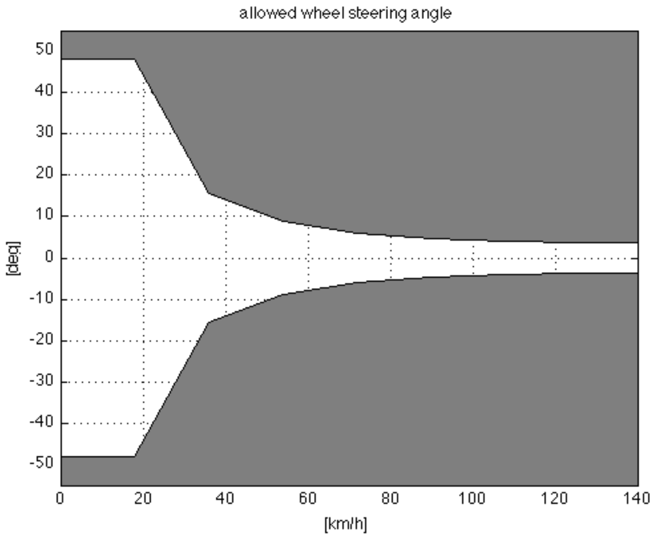


Figure 3.3: Maximum wheel steering angle that the driver is allowed to use as a function of velocity

The driver also compares the velocity of the vehicle with the velocity that is defined in the path. If it's too low the driver increases the throttle and if the velocity is too high the throttle output will be decreased. This adjustment is done using a PI-integrator. The driver also has the possibility to apply the brakes if the vehicle speed is too high compared to the speed given from the path.

For the vehicle to be able to drive at large varieties of speeds the gears of the car has to be changed. The driver knows at what speed ranges the vehicle can drive on the different gears and when it's time for a gear change. At these speeds the driver reduces the throttle, engages the clutch and changes the gear. Then the driver disengages the clutch and applies the throttle again depending on the desired speed.

4 Control system design

This control system determines the present driving situation and the therefore desired mode for the DSLD from SWA, yaw rate, reference yaw rate and wheel rotational speeds. The reference model used to calculate the reference yaw rate are presented and also a model of the physical DSLD that calculates the extra torque that the differential contributes with.

4.1 DSLD - Selecting working mode

According to the theory in Section 2.4 the DSLD can be set to four different working modes. The modes are presented in Table 2.1 and are fully open, fully locked or conditionally locked in either direction. When the vehicle is going straight ahead the differential is normally fully open (mode one). If the vehicle is turning the mode is set to conditionally locked (mode two or three) depending on the direction of the turn. Conditionally locked, in a turn, means that the outer wheel is allowed to rotate faster than the inner wheel but note vice versa. Mode four, fully locked, is used as traction control to prevent one wheel from spinning or when a yaw damping effect, due to oversteering, is wanted.

To decide what mode that should be used for the present driving situation some logic criterions are used. The criterions are based on SWA, vehicle yaw rate, r, and wheel speeds, ω, and are listed in Table 4.1a. The first two criterions (a-b) determines whether the SWA is in a turning mode or not and the two following criterions (c-d) that the vehicle actually is turning in the direction that the driver wants. Criterions e and f are used for traction control and determines if the wheels on the axle with the DSLD rotates with different speeds and which wheel that rotates faster. These six criterions are determined by Carlén and Yngve [Carlén and Yngve, 2006]. The last criteria (g) compares the yaw rate of the vehicle to the yaw rate from the reference model and is used to determine if the vehicle oversteers.

Table 4.1: Criterions to decide the mode for the DSLD

(a) Criterions

a	$\text{SWA} > \text{SWA}_{crit}$
b	$\text{SWA} < -\text{SWA}_{crit}$
c	$r > r_{crit}$
d	$r < -r_{crit}$
e	$\omega_l/\omega_r > \omega_{crit}$
f	$\omega_r/\omega_l > \omega_{crit}$
g	$(r - r_{ref})\,\text{sgn}(r) > r_{crit}$

(b) Modes

1	$\neg\ (2 \lor 3 \lor 4)$
2	$\text{a} \land \text{c} \neg\land \text{g}$
3	$\text{b} \land \text{d} \neg\land \text{g}$
4	$\text{g} \lor ((\text{e} \lor \text{f}) \neg\land (2 \lor 3))$

For a mode to be set a combination of the criterions has to be fulfilled. As can be seen in Table 4.1b where the combinations are presented the g criterion is the strongest criterion. If criterion g is fulfilled mode four is always set. This means that if the vehicle oversteers the DSLD will lock to reduce the yaw. Second strongest are the criterions for turning, i.e. a and c respective b and d. If criterion g isn't fulfilled and the turning criterions are fulfilled the mode will be two for a right turn respectively three for a left turn. Third strongest are the criterions for traction control, i.e. e and f. So if the vehicle isn't oversteering and isn't turning and still one of the wheels are rotating faster than the other these criterions will set the mode four for the DSLD. This will lock the axle and transfer torque to the wheel with grip as a traction control. Finally if the vehicle isn't oversteering or turning and the wheels aren't spinning the control system will set mode one, fully open, for the DSLD.

These calculations to determine the mode are done every tenth of a second to cancel out fluctuation in the signals and to speed up the simulations.

4.2 Reference model

A reference model is an estimation of what yaw dynamics that is expected given a specific steering wheel angle and vehicle velocity. In this case the reference model is used to determine the yaw rate that the driver expects in a turn. The model is dependent on fixed vehicle parameters such as length and weight along with two variables, vehicle speed and SWA. The DSLD control system set mode 4 if the difference between the reference yaw rate and the actual yaw rate exceeds a certain value. If the vehicle is oversteered the DSLD is used in yaw damping mode. Equation 4.1 defines the reference model that is used [Klomp, 2008].

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx + Du \end{aligned} \tag{4.1}$$

where:

$$\begin{aligned} A &= \begin{bmatrix} \frac{-C_{\alpha r}-C_{\alpha f}}{V_x m} & \frac{C_{\alpha r}l_r-C_{\alpha f}l_f}{V_x m} - V_x \\ \frac{C_{\alpha r}l_r-C_{\alpha f}l_f}{V_x I_{zz}} & \frac{-C_{\alpha r}l_r^2-C_{\alpha f}l_f^2}{V_x I_{zz}} \end{bmatrix} & B &= \begin{bmatrix} \frac{C_{\alpha f}}{m} \\ \frac{C_{\alpha f}l_f}{I_{zz}} \end{bmatrix} \\ C &= \begin{bmatrix} 0 & 1 \end{bmatrix} & D &= 0 \\ x &= \begin{bmatrix} V_y \\ r \end{bmatrix} \qquad y = r & u &= \delta \end{aligned} \tag{4.2}$$

4.3 Mechanical model

According to the the mechanical function described in Section 2.4 the ideal model of the DSLD should be stiff when the differential is locked. But to be able to simulate when the differential locks and to eliminate the possible singularity in the model the differential is modeled as a torsional spring and damper system. With a stiff spring and a damper coefficient that matches the stiffness this is a good estimation both for generating the desired torque and the rotating angle of the differential.

The problem with this kind of system is that it will take a bit longer time to stabilize at the right amount of torque and that the wheel speeds will oscillate some before stabilizing.

5 Simulation procedure

The simulations are divided into two parts, performance and stability. The performance part focuses on the gain in vehicle acceleration and cornering performance while the stability part focuses on the gain in stability in oversteering situations but also the stability and reliability of the DSLD. For example, what happens if the differential doesn't unlock as intended?

5.1 Performance

Three simulations are used to evaluate the increase in vehicle acceleration and cornering performance. In these simulations the driver tries to follow a specific path and corrects for any differences between the position and direction of path compared to the position and direction of the vehicle.

5.1.1 Split-μ acceleration

This test is done to see if the DSLD can improve the vehicle acceleration from standstill when the left and right wheels are running on different surfaces. The driver follows a straight line while accelerating at maximum throttle. The possible risk with the differential is a large yaw motion of the vehicle that would lead to a larger required SWA. Therefor a good result would be a low acceleration time along with a small SWA input. To get a clearer result the simulations are performed once with gearshifts and once in second gear.

5.1.2 Checkboard acceleration

This simulation is done to see if the DSLD is quick enough to lock and unlock between the different μ parts of the track. The driver accelerates at full throttle while following a straight line. A good result would be a low acceleration time, small SWA input from the driver and also that the differential locks and unlocks as it is intended to do.

5.1.3 90 degree turn

To simulate the increase in cornering performance due to the DSLD a ninety degree turn is used. Midway in the turn, the driver accelerates to a specific velocity so that the differential locks. A good result would be a quick acceleration time and that the path of the vehicle doesn't differs from the given path.

5.2 Stability

Two simulations are performed to evaluate the DSLD's influence on vehicle stability and what will happen if the differential locks in the wrong state. The simulations are the Sine with Dwell where the differential is supposed to reduce oversteer and Worst case where a brake input on the outer wheel is supposed to unlock a differential that is locked in the wrong direction.

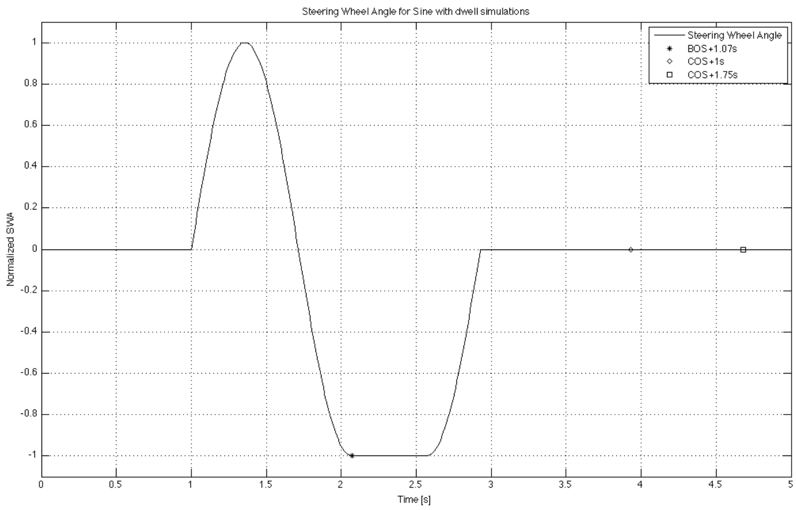


Figure 5.1: Steering Wheel Angle for Sine with Dwell

5.2.1 Sine with Dwell

As described in Section 2.3 a locked axle will give a more understeered behavior of the vehicle. This can be used when the vehicle tends to oversteer, a locked axle will contribute to a yaw damping moment. The Sine with Dwell simulation is done to evaluate the performance of the DSLD as a yaw damping device. This is a standard test and the criterions that the vehicle should be able to manage are clearly stated [NHTSA, 2006].

Figure 5.1 shows the SWA input to the vehicle, along with the measuring times for the different criterions. There are three criterions for the test, two yaw rate criterions and one maneuverability criterion. The first one, YYR_1, is the yaw rate one second after COS (Completion of Steer) divided with the second peak in yaw rate. The second criteria, YYR_2, is calculated in the same way 1.75 seconds after COS, these are shown in Equation 5.1. The maximum allowed values for these criterions are 35% and 20%, respectively.

$$\begin{aligned} \mathrm{YYR}_1 &= \tfrac{r_{COS+1}}{r_{peak}} \\ \mathrm{YYR}_2 &= \tfrac{r_{COS+1.75}}{r_{peak}} \end{aligned} \tag{5.1}$$

The last criteria for the Sine with Dwell test is the maneuverability criteria. This criteria states that 1.07 seconds after BOS (Beginning of Steer) the vehicles CoG must have traveled at least 1.83 meters sideways from the straight path before the steering maneuver starts.

The maximum steering wheel angle is determined by a standard procedure driving the vehicle in a circle. This procedure would give one maximum angle for each driveline and that is not preferred when comparing the different results to each other. Therefor three different maximum angles are used. The first maximum angle where all the vehicles succeeds during the test. The second angle at a point where the vehicles starts to show different results and the third angle at a point where some of the vehicles succeeds and some fails. This will give clear results of how the DSLD influence the oversteer of the different driveline configurations.

A reduction in oversteer is preferred and therefor the DSLD is engaged using the reference model described in Section 4.2.

5.2.2 Worst case

If, for some reason, the DSLD would be locked as in state 3, left turn, when the vehicle is driving through a right turn where correct mode is mode 2, the differential would contribute to understeer in the same way as a rigid axle. To be able to unlock the differential in that kind of situation it's not enough just to change the mode of the differential to mode 2 or mode 1. Then the DSLD would stay locked as long as the left outer wheel tries to rotate faster than the inner right wheel. To unlock the differential a braking force on the outer wheel is required to reduce the torque produced by the differential. When the speed of the outer wheel reduces below the speed of the inner wheel the differential will unlock and the braking force is no longer needed. Then the outer wheel can speed up and and the DSLD acts as an open differential again.

The worst case simulation is done to find out if it's possible to unlock the differential in this kind of situations and how this will affect the vehicles path. It is obvious that a braking force on the outer wheel will increase understeer. Therefor the brake input should be as short as possible. On the other hand the brake input has to be long enough so that the DSLD really unlocks. Therefor it is of interest to simulate different lengths of the brake input. It is also interesting to simulate how a too long brake input will affect the vehicle.

In the simulation the driver keeps a constant speed and a constant SWA. At the beginning the differential is set to mode four. This will make the differential lock in the wrong direction i.e the outer wheel isn't allowed to rotate faster than the inner wheel. In the turn a brake input is sent to the outer wheel to unlock the differential.

When the differential is locked in this state there are two alternativ modes that the differential can be set to. It can be it's normal turning mode, which in the simulation will be mode two or it can be mode one. The effect of this parameter and how it affects the path of the vehicle will also be studied to be able to reduce the understeering effect of the unlocking of the DSLD.

6 Results

The simulations are divided between simulations used to evaluate the performance of the vehicle and simulations used to evaluate the stability of the vehicle and the robustness of the DSLD. The results are presented specifically for each simulation.

6.1 Performance

The performance results will be focusing on vehicle acceleration and cornering performance but also the major disadvantages in each simulation will be presented.

6.1.1 Split-μ acceleration

The acceleration on split-μ shows an increase in vehicle acceleration for all vehicles with the DSLD, as can be seen in Figure 6.1. The high-μ surface is 1.0 and the low-μ is 0.3.

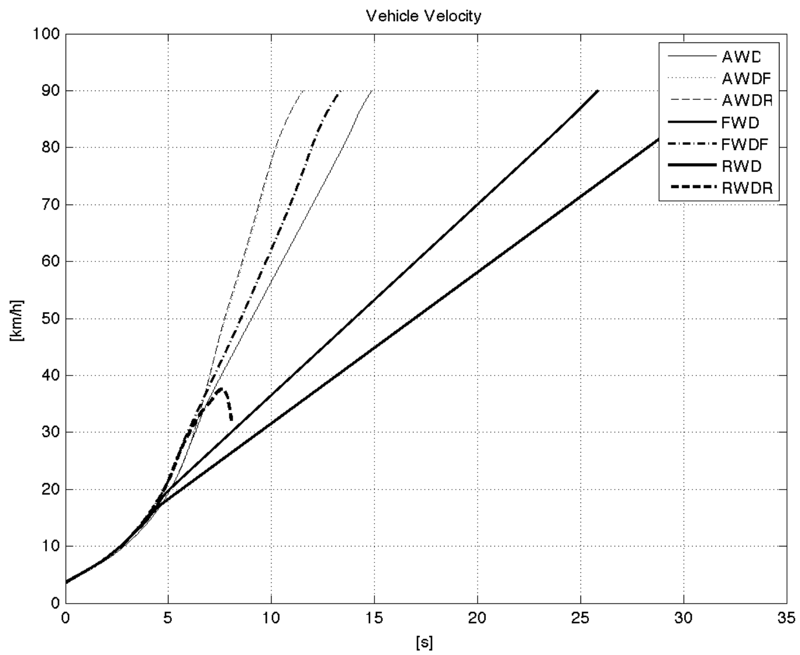


Figure 6.1: Vehicle velocity in second gear during the split-μ acceleration

Interesting to notice is that the front wheel driven vehicle with the DSLD is faster then the all wheel driven vehicle without the DSLD. This is possible due to the large difference in friction between the right and the left wheels. The front wheel driven vehicle can use the force from one wheel with high friction and one with low friction when the all wheel driven vehicle vith an open differential only can use the force from four wheels with low friction. That makes it possible to get a higher vehicle acceleration with the two wheel driven vehicles if there is a large enough difference in friction between the wheels.

The risks with the split-μ acceleration is that the yaw motion of the vehicle and the Wheel Steering Angle would be large and that the vehicle thereby would be hard to control. This happens for the rear wheel driven vehicle with the DSLD and can be seen as the acceleration starts to decrease when the vehicle starts to spin. The Wheel Steering Angle for the same simulation can be seen in Figure 6.2 where the shaded area represents the maximum Wheel Steering Angle that the driver model is allowed to use, as described in Section 3.4.

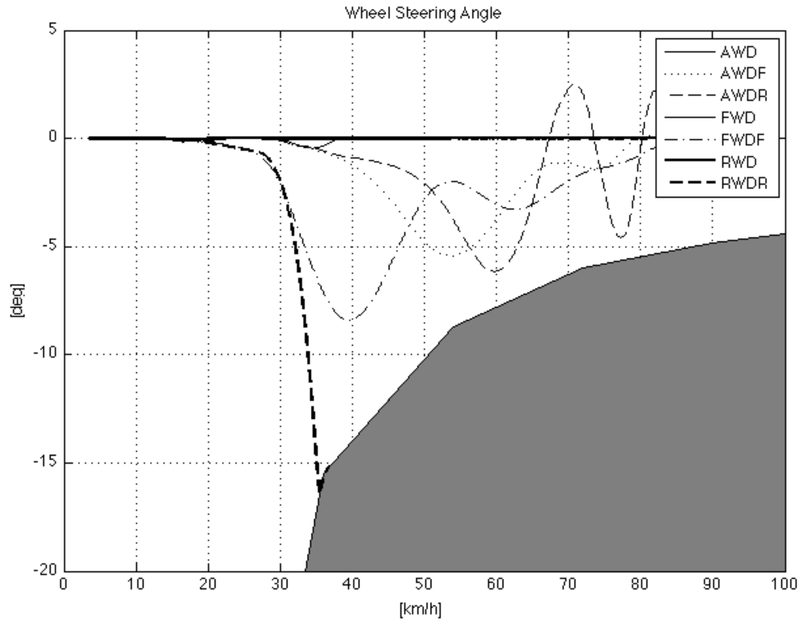


Figure 6.2: Wheel Steering Angle in second gear during the split-μ acceleration

In Figure 6.2 it's easy to see that the driver uses all the steering angle for the rear wheel driven vehicle with the DSLD when it starts to spin. Most interesting to compare is the Wheel Steering Angle for the front wheel driven vehicle with the DSLD and the all wheel driven vehicle with the DSLD in the rear. The maximum Wheel Steering Angle is larger for the front wheel driven vehicle but occurs at lower speed so the margin to the maximum allowed angle is larger than for the all wheel driven vehicle. The effect of the steering angles are shown in the vehicle yaw angle, Figure 6.3.

The smaller steering angle for the all wheel driven vehicle with the DSLD in the rear creates a larger vehicle yaw angle than for the front wheel driven one with the DSLD. The all wheel driven vehicle is more prone to oversteer and the vehicle yaw oscillates more before the driver succeeds to stabilize the vehicle.

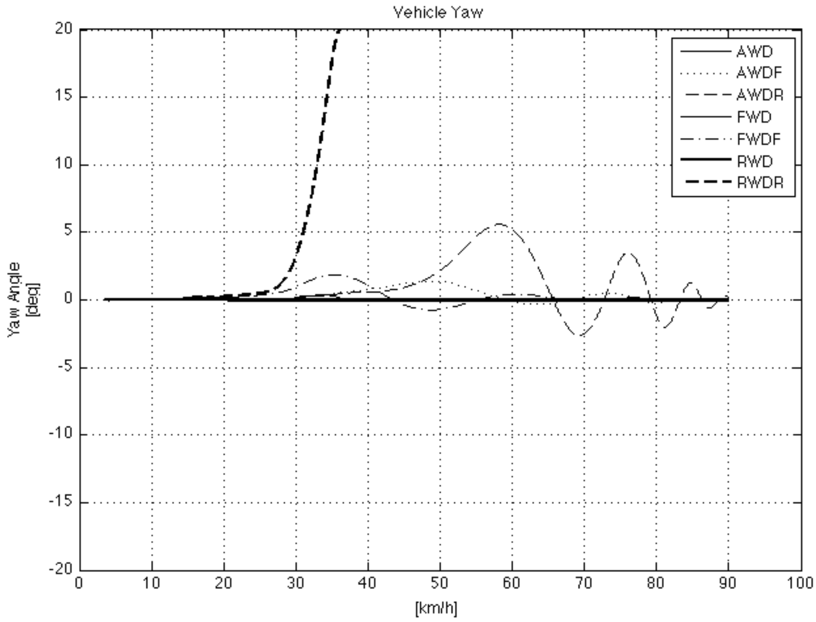


Figure 6.3: Vehicle yaw angle in second gear during the split-μ acceleration

6.1.2 Checkboard acceleration

The vehicle acceleration in second gear during the checkboard acceleration can be seen in Figure 6.4 where the results are similar to the ones for the split-μ acceleration.

Highest accelerations are reached for the all wheel driven vehicles with the DSLD followed by the normal all wheel driven vehicle and the front wheel driven vehicle with the DSLD. The largest improvement is reached for the front wheel driven vehicle with the DSLD.

The risks with the checkboard acceleration were that the DSLD would introduce a large vehicle yaw motion and that the differential wouldn't be able to lock and unlock fast enough. The vehicle yaw dynamics for the front wheel driven vehicle with and without the differential can be seen in Figure 6.5.

It can be seen that the DSLD introduces a larger yaw motion that in fact will demand a larger steering angle from the driver. But comparing the frequencies in yaw rate between the normal front wheel driven vehicle and the one with the DSLD it can also be seen that the frequency for the vehicle with the DSLD is half of the one without. This means that the driver will only have to correct the steering angle ones per change in friction compared to twice per change in friction for the normal front wheel driven vehicle.

To evaluate the ability of the DSLD to lock and unlock fast enough the wheel speeds for the front wheel driven vehicle with and without the DSLD are presented in Figure 6.6.

The differences in wheel rotational speeds shows that the DSLD unlocks and locks every time the friction changes. It can also be seen that the combined friction of the two wheels isn't enough for this engine. This makes both wheels spin at the same time as the differential locks the left and right wheels together. To further increase the vehicle acceleration engine torque should be limited when

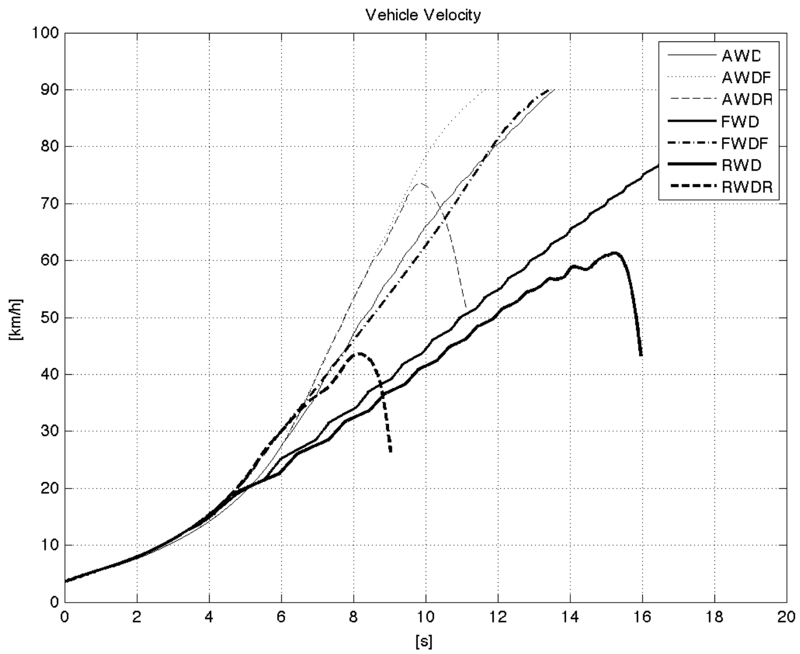


Figure 6.4: Vehicle velocity in second gear during the checkboard acceleration

both the driving wheels starts to spin.

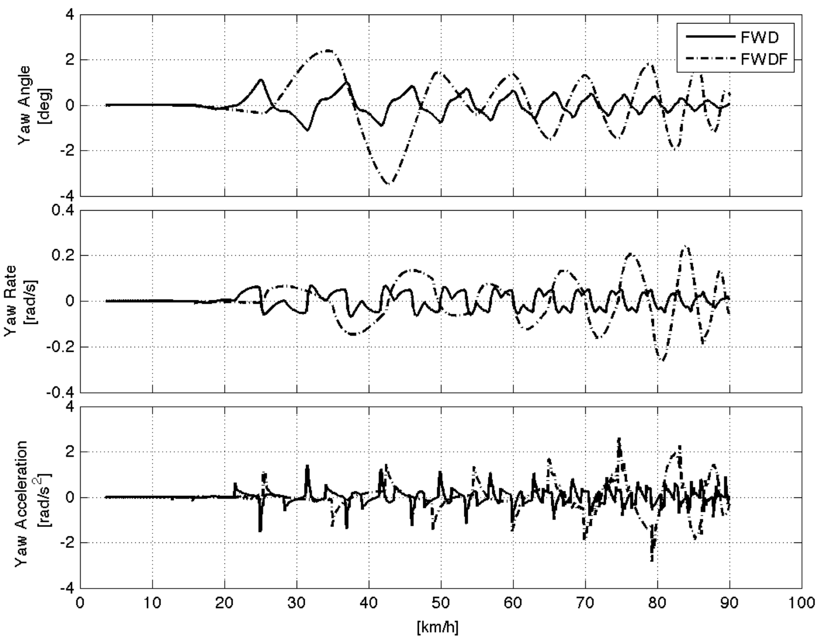
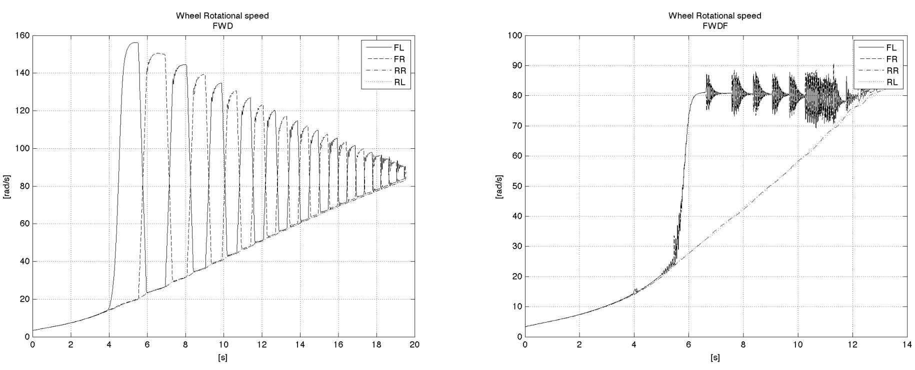


Figure 6.5: Vehicle yaw angle, yaw rate and yaw acceleration in second gear during the checkboard acceleration



(a) FWD without DSLD

(b) FWD with DSLD

Figure 6.6: Wheel rotational speeds in second gear during the checkboard acceleration

6.1.3 90 degree turn

Starting with the vehicle acceleration during the 90 degree turn that is presented in Figure 6.7 it can be seen that the two rear wheel driven vehicles are starting to spin out during the acceleration. It can also be seen that the all wheel driven vehicles doesn't improve in acceleration and that the only improvement is for the front wheel driven vehicle with the DSLD.

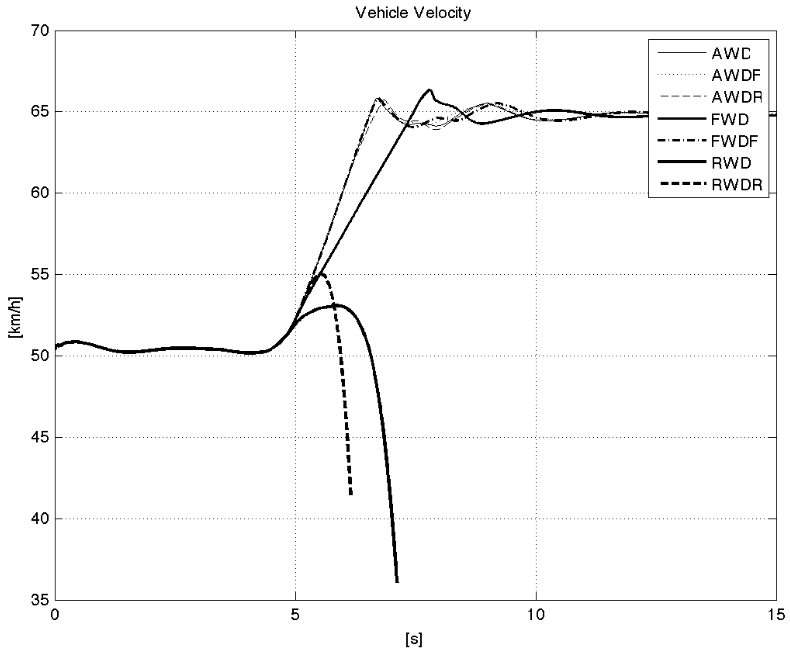


Figure 6.7: Vehicle velocity in second gear during the 90 degree turn

The results for the cornering performances are presented in form of global coordinates during the turn, Figure 6.8. The all wheel driven vehicles are all better than the front wheel driven vehicles and also in this simulation the largest improvement is achieved for the front wheel driven vehicle with the DSLD. It's also interesting that the all wheel driven vehicle with the DSLD in the rear tends to oversteer but that the driver succeeds to compensate for that and keep the vehicle on the path.

To be able to increase the cornering performance an increase in lateral acceleration is needed. Focusing on the front wheel driven vehicles the lateral accelerations are presented in Figure 6.9. It can be seen that the lateral acceleration is significantly higher in the end of the acceleration at about seven seconds for the vehicle with the DSLD than it is for the normal front wheel driven vehicle.

Looking at the global positions for the vehicles in the end of the simulation, Figure 6.10, when all vehicles have traveled the same time it can be seen that all vehicles with the DSLD improves their distances traveling to the right and once again that the largest improvement is reached for the front wheel driven vehicle with the DSLD.

A general problem that has been shown is that the vehicles with DSLD on the rear axle tend to spin out when accelerating in a bend. This is because there is to much power so that both the left and the right rear wheels starts to spin.

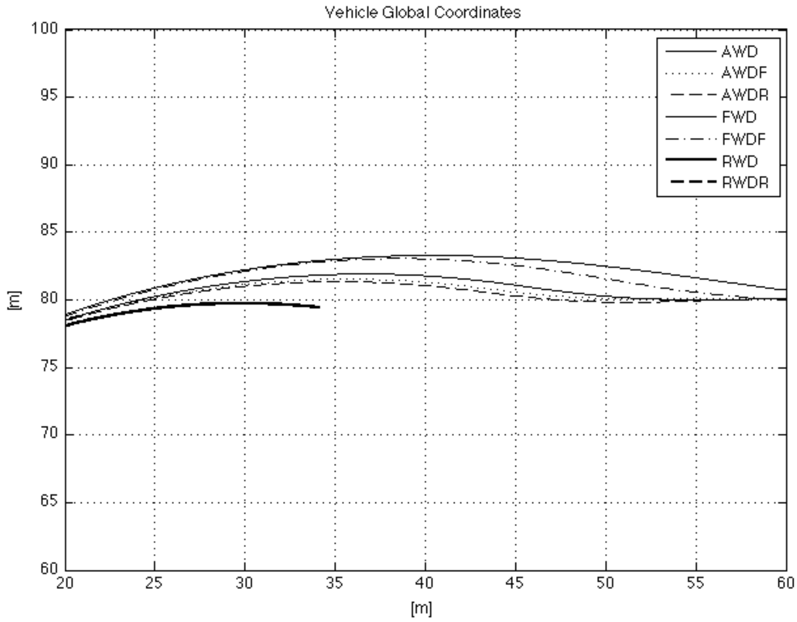


Figure 6.8: Global coordinates for the vehicle in the end of the 90 degree turn

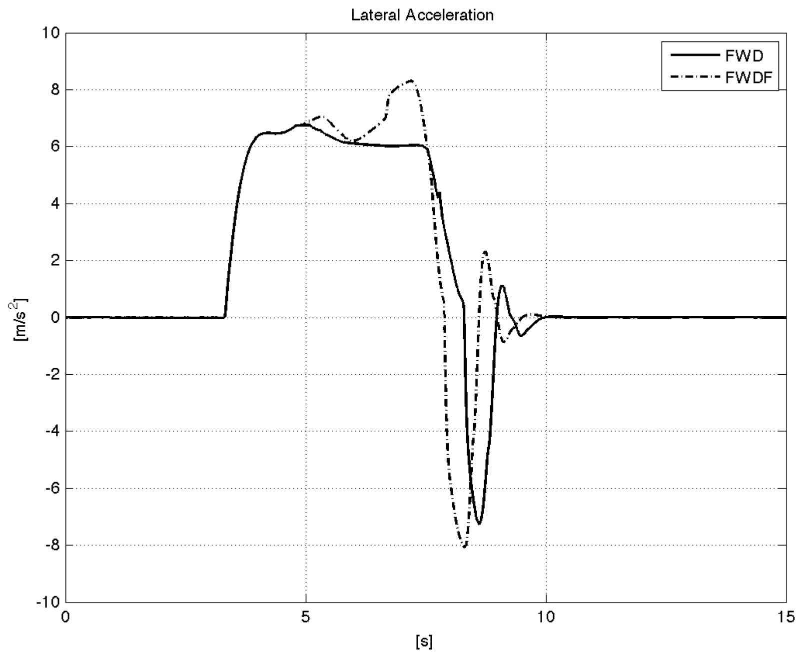


Figure 6.9: Lateral accelerations for the front wheel driven vehicles during the 90 degree turn

When this happens almost all the lateral force from the rear tires disappears and the vehicle becomes too oversteered.

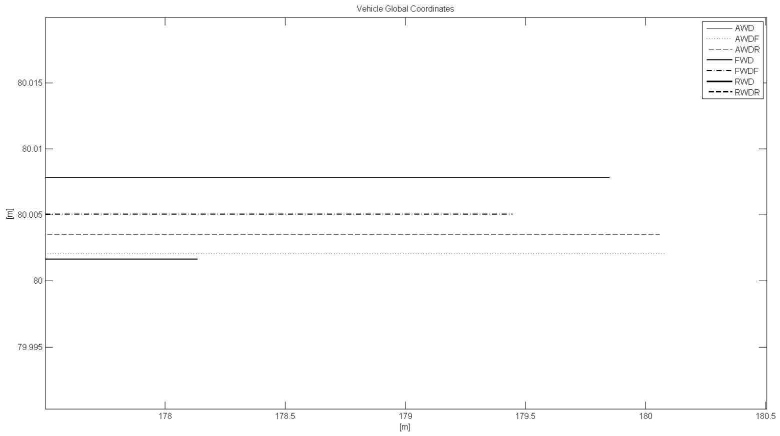


Figure 6.10: Global positions in the end of the 90 degree turn

6.2 Stability

The influence of the DSLD on the vehicle stability is evaluated in the Sine with Dwell simulation. The aim with the DSLD is to be able to reduce oversteer in an oversteered situation. The risk for increased understeer due to the DSLD is evaluated during the Worst case simulation.

6.2.1 Sine with Dwell

As described in Section 5.2.1 the test is performed with three different maximum Wheel Steering Angles. The results are presented in Tables 6.1, 6.2 and 6.3 for the different angles respectively.

Table 6.1: $\delta = 3.9$

	AWD	AWDF	AWDR	FWD	FWDF	RWD	RWDR
YYR_1	0.35%	0.02%	0.03%	0.30%	0.02%	0.35%	0.02%
YYR_2	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%	0.00%
Man	$2.70m$	$2.67m$	$2.69m$	$2.70m$	$2.67m$	$2.70m$	$2.68m$

Table 6.2: $\delta = 4.1$

	AWD	AWDF	AWDR	FWD	FWDF	RWD	RWDR
YYR_1	13.33%	0.05%	0.19%	7.24%	0.04%	15.06%	0.32%
YYR_2	0.02%	0.00%	0.00%	0.01%	0.00%	0.02%	0.00%
Man	$2.79m$	$2.76m$	$2.77m$	$2.79m$	$2.76m$	$2.80m$	$2.77m$

Interesting to notice is that the vehicles showing the best results are the ones with the DSLD. So even if the rear wheel driven vehicle with the DSLD has some problems at the highest Wheel Steering Angle, this results clearly shows an improvement in yaw damping using the DSLD.

Figure 6.11 shows the yaw angle, yaw rate and the yaw acceleration for the FWD vehicle at a maximum Wheel Steering Angle of 4.3 degrees. The best

Table 6.3: $\delta = 4.3$

	AWD	AWDF	AWDR	FWD	FWDF	RWD	RWDR
YYR_1	111.8%	0.25%	33.28%	108.6%	0.23%	111.5%	70.00%
YYR_2	126.2%	0.00%	0.01%	124.7%	0.00%	125.1%	2.01%
Man	$2.89m$	$2.85m$	$2.86m$	$2.88m$	$2.85m$	$2.89m$	$2.87m$

possible result would be if the yaw rate had the same shape as the steering wheel angle in Figure 5.1 and it can be seen that the vehicle with the DSLD follows that curve much better than the normal front wheel driven vehicle.

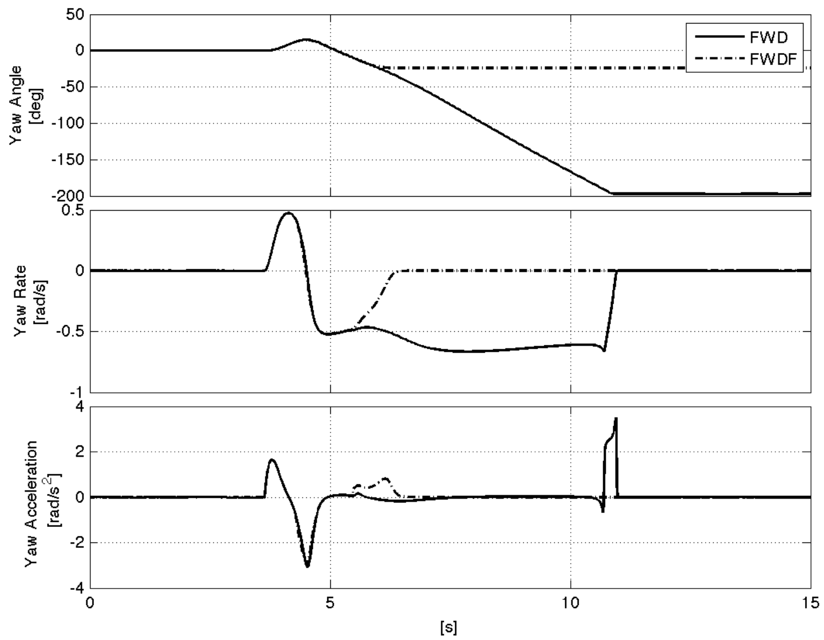


Figure 6.11: Yaw, yaw rate and yaw acceleration for the FWD vehicle with and without the DSLD at a maximum Wheel Steering Angle of 4.3 degrees

6.2.2 Worst case

In this simulation the aim is to see if it's possible to unlock the DSLD if the differential for some reason would be locked in the wrong direction and how much that will increase the understeer of the vehicle. The difference in influence on understeer between setting the DSLD in normal turning mode and open mode will also be presented but first the normal turning mode is used.

The global coordinates during the turn for different lengths of the brake input can be seen in Figure 6.12.

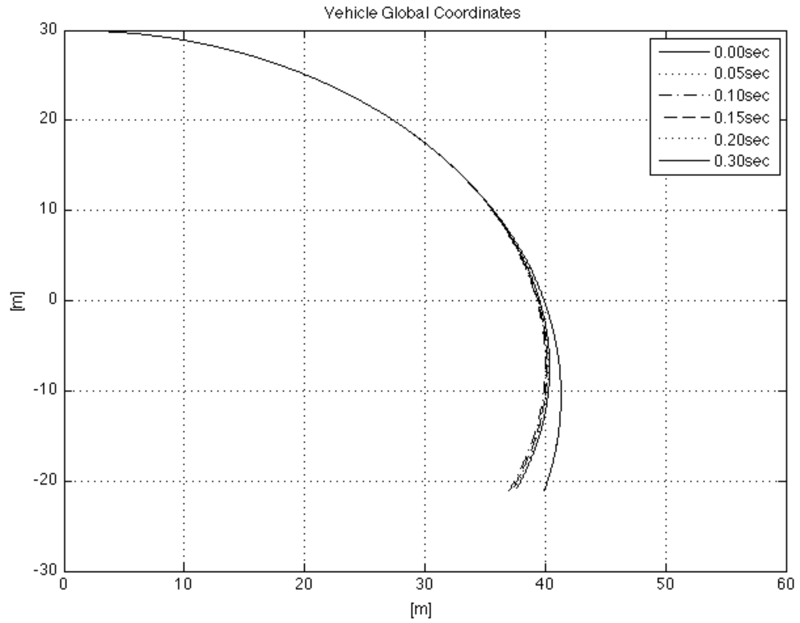


Figure 6.12: Vehicle global coordinated during the Worst case simulation with the DSLD in it's normal turning mode.

The vehicle without any brake input and the vehicle with a brake input of 0.05 seconds are to the right meaning that they are most understeered and that the 0.05 second brake input is to short to unlock the DSLD. The vehicle with a brake input of 0.1 second is the one that is the least understeered and the ones with longer brake inputs are a bit more understeered but still far better than the one with a to short brake input. In this case it means that a brake input of 0.1 second is enough to unlock the DSLD but not too long so that the understeer increases more than necessarily.

Looking at the torque through the DSLD during the same maneuver, Figure 6.13, it can be seen the differential produces a torque that increases the understeer of the vehicle when it's locked in the wrong direction.

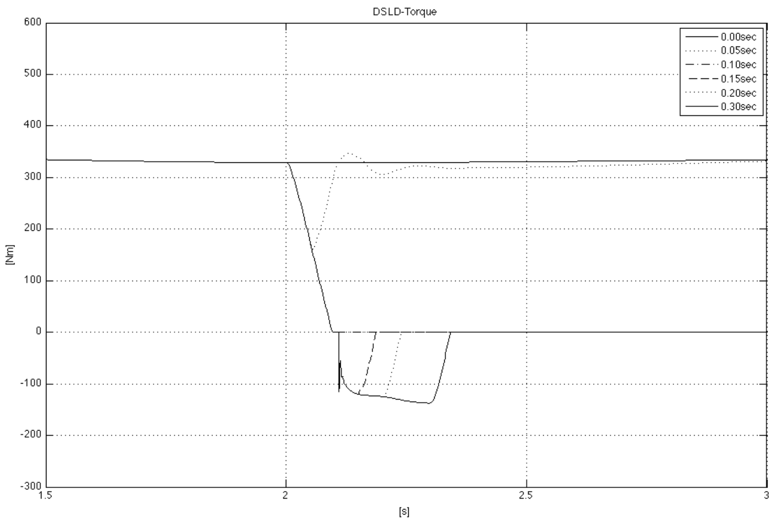


Figure 6.13: Torque through the DSLD in turning mode

It can be seen here as well that the 0.05 second brake input doesn't succeeds to reduce the torque completely and thereby not unlock the DSLD and that the brake inputs longer then 0.1 seconds are too long. They will, when the differential is set to turning mode, make the differential brake the inner wheel through the differential after it has been unlocked and locked again in the other direction. This is the difference between setting the differential in turning and open mode. The lateral acceleration for the same test, with the differential set to it's normal turning mode, is presented in Figure 6.14.

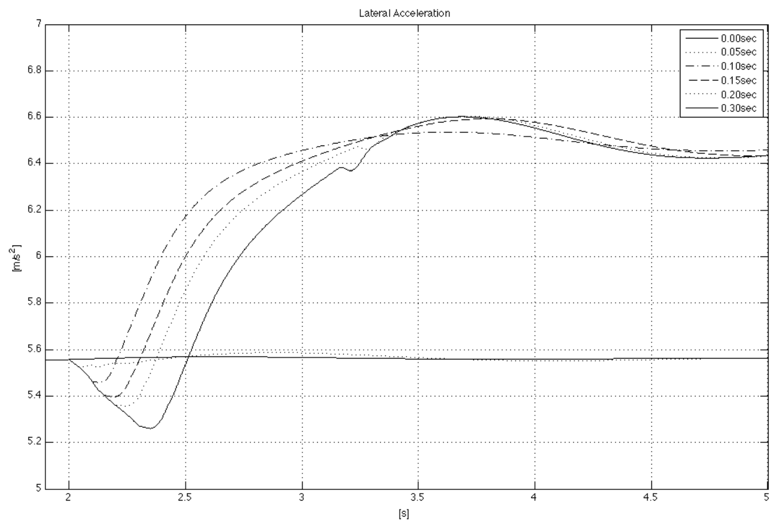


Figure 6.14: Lateral acceleration in turning mode

The difference if the DSLD is set to open mode instead of turning mode is that the inner wheel wouldn't be affected if the brake input is to long. The difference in lateral acceleration between the open and turning modes for a brake input of 0.4 seconds is presented in Figure 6.15.

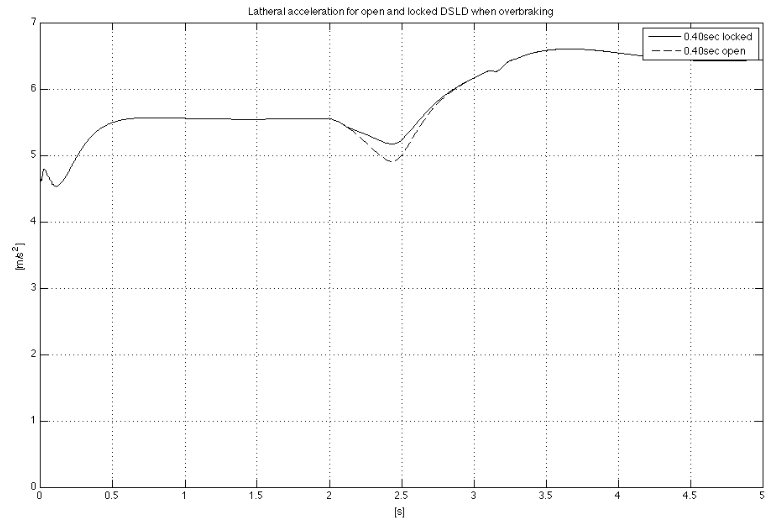


Figure 6.15: Difference in lateral acceleration between open and turning mode for a brake input of 0.4 seconds

Here it can be seen that the least decrease in lateral acceleration is reached by setting the DSLD in it's normal turning mode and that's the mode that should be preferred in this kind of situations.

7 Conclusions

The results of the simulations shows that the DSLD has the potential and possibility to increase the handling performance as well as the stability of any drive line simulated.

The performance simulations shows that the DSLD preferably is positioned on the front axle in a four wheel driven vehicle but that it also contributes to vehicle performance being positioned on the rear axle. Also in the front wheel driven vehicle and in the rear wheel driven one the differential increases the cornering performance if the engine torque is controlled in a good way. The largest differences in cornering ability and acceleration are reached by positioning the differential in a front wheel driven vehicle. The fact that the engine torque has to be controlled and reduced when both the wheels on the driven axle starts to spin is crucial if the implementation of the DSLD should be successful in a real vehicle.

The DSLD has shown no problems remaining locked in the wrong direction and thereby counteracting any turning motion of the vehicle. It has followed the quickest reactions of the driver and the fastest movements of the vehicle without problems. The worst-case simulation shows that when the DSLD is forced to lock in the wrong direction, it isn't any particular loss in cornering ability when unlocking it with a short brake activation. The state of the differential has to be known by the traction control system in the vehicle and that system has to interact with the control system for the differential.

Finally the results of the sine-with-dwell simulation shows that the DSLD can be used to reduce yaw-motions when the vehicle is in an oversteered situation. Here, as in the brake activation situation mentioned, the traction control system in the vehicle has to interact with the control system of the differential to reach a good result. This is the main part of the DSLD development that has to be done future on as well as evaluating the performance of the controll system for the DSLD.

8 Recommendations

The simulations have shown that there is one specific area that has to be developed more and that is the interaction and cooperation between the DSLD and the ABS, traction control, ESP, engine control and the other systems that already exists in a vehicle. To be able to achieve the maximum effect of the differential and to reduce the risk that the systems will counter act to each other the systems will have to communicate. The central information is a signal from the DSLD to the other systems with the information if and how the differential is locked. The other systems will have to take that into consideration when deciding e.g. engine torque and so on.

The other main part of the development of the DSLD that isn't covered at all in this work is the physical testing. In this moment of writing tests are being done in the Chalmers Formula student 2010 car. The CFS10 car has a DSLD on the rear axle and is being evaluated with the focus on performance. A lot of these tests are needed before the DSLD can be used in a standard passenger vehicle, but that's for someone else to carry out.

References

[Alfredsson, 2006] Alfredsson, J. (2006). *Locking differential.* Patent, WO/2006/041384.

[Alvarez et al., 2005] Alvarez, L., Yi, J., Horowitz, R., and Olmos, L. (2005). *Dynamic Friction Model-Based Tire-Road Friction Estimation and Emergency Braking Control. Journal of Dynamic Systems, Measurement and Control*, 127:22–32.

[Brolin et al., 2008] Brolin, M., Christensson, P., Framby, J., and Sandberg, C. (2008). *Konstruktion och tillverkning av riktningskänslig låsbar differentialväxel.* Kandidatarbete, Chalmers University of Technology.

[Carlén and Yngve, 2006] Carlén, M. and Yngve, S. (2006). *Control strategies for electronically controlled limited slip differentials.* Master's thesis, Chalmers University of Technology.

[Gustafsson, 1997] Gustafsson, F. (1997). *Slip-based Tire-Road Friction Estimation. Automatica*, 33(6):1087–1099.

[Klomp, 2008] Klomp, M. (2008). *Effective Axle Cornering Characteristics.* Lecture Notes - TME100, Chalmers University of Technology.

[NHTSA, 2006] NHTSA (2006). *PROPOSED FMVSS No.126, Electronic Stability Control Systems.*

[Palmenäs, 2008] Palmenäs, G. (2008). *Internal project.* Internal project at Vehicle Engineering and Autonomous Systems supervised by Mathias Lidberg, Chalmers University of Technology.

A Simulation model

The model in Simulink is built as a driver, the drive train, DSLD, chassis and tires.

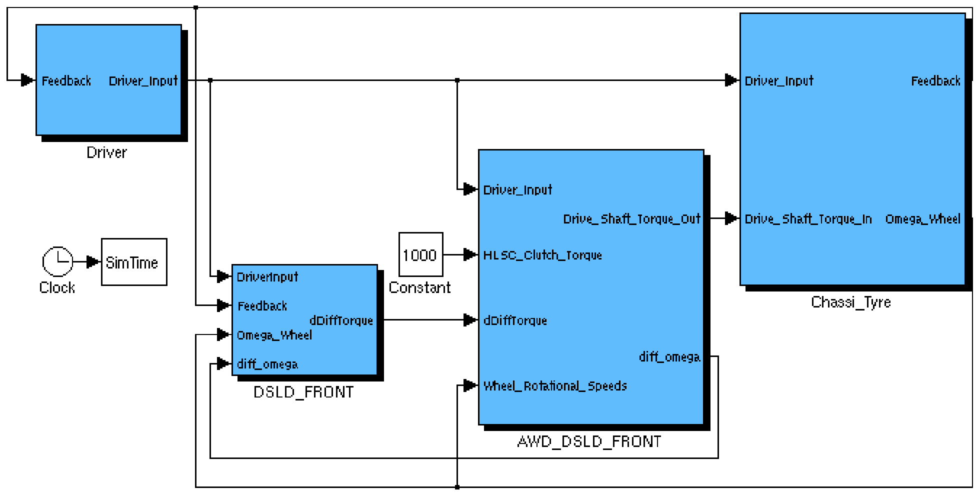


Figure A.1: Simulink model of the vehicle with driver, DSLD, drive train, chassis and tires

A.1 Driver

The driver needs to be able to use the steering wheel, throttle, brake, clutch and gear shifter. The path that the driver should follow is predefined and consists of points along a path. Every point consist of global x and y positions and the direction of travel. The velocity which the driver accelerates or decelerates to. The friction on each tire, that could be different between left and right as well as front and rear. This could be seen on the AWD wheel speeds in the checkboard acceleration simulation, Figure C.2.2.19. The path of the vehicle is further explained in Section 3.3.

To know which point that should be used the driver predicts where the vehicle will be after a certain time. This position is compared to the path and the closest point is used. The driver continually changes it's outputs to come closer to the path. To reduce the simulation time and to be able to drive in path that intersects itself like an eight shape, the driver only looks at a small part of the path.

The top right part of the Figure A.2 defines which part of the path that the driver looks at. This is done to reduce the simulation time and also prevents the driver from looking at the wrong part of the track if the track intersects itself.

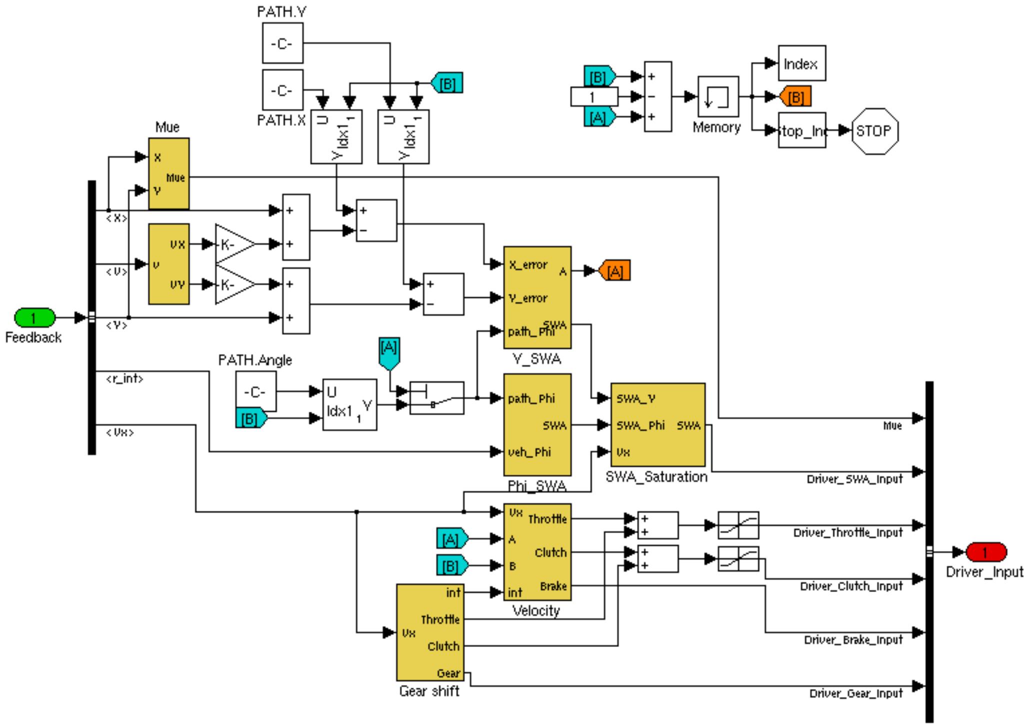


Figure A.2: The driver of the vehicle that generates throttle, clutch, brake and gear inputs to the vehicle. Friction is also determined.

A.1.1 Wheel angle

How the driver changes the wheel angle is dependent on two things. The first is the distance between where the vehicle will be and where it should be, Figure A.3. The driver estimates where the vehicle will be positioned after a certain time. The distance between this point and the closest point on the path gives the output. The driver considers if the vehicle is positioned to left, right top or bottom of the path. The reason why four sectors are used and not two is to have a smooth output. If only two sectors are used the output could oscillate between positive and negative sign when on the limit between them.

The second input is the difference between what yaw angle the vehicle has and what it should be, Figure A.4. The driver compares the yaw angle from the path with the yaw angle that the vehicle has, this difference gives the output. The driver considers a difference of 5 degrees to be the same as a difference of 365 degrees or -355 degrees.

These two are combined and also limited, the limit is tested and corresponds to the angle that produces the largest lateral acceleration as a function of the velocity, Figure A.5. The signal is also filtered since there is a distance between the points of the path, if the vehicle is exactly on the path the distance between points produces an error. To reduce this error either a filter are used or the distance between the points are reduced. The wheel angles are small but will give an oscillating behavior that shows in for example the lateral acceleration of the vehicle.

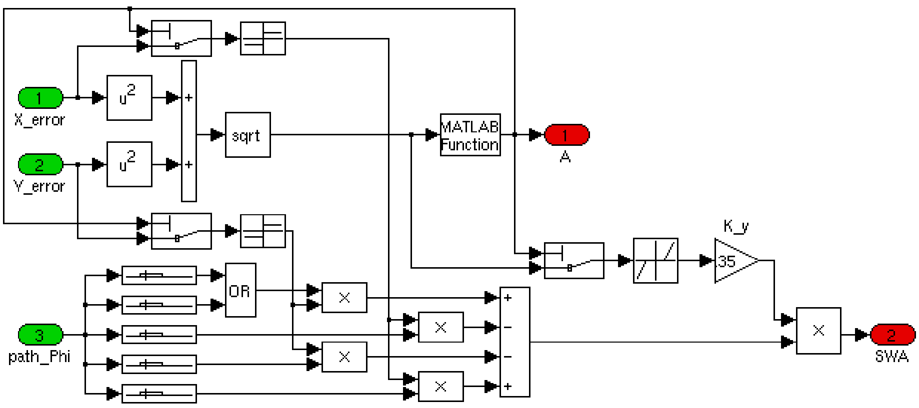


Figure A.3: Wheel steering angle depending on the distance between the vehicle and the path

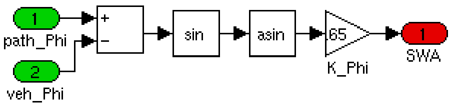


Figure A.4: Wheel steering angle depending on the difference in yaw between the path and the vehicle

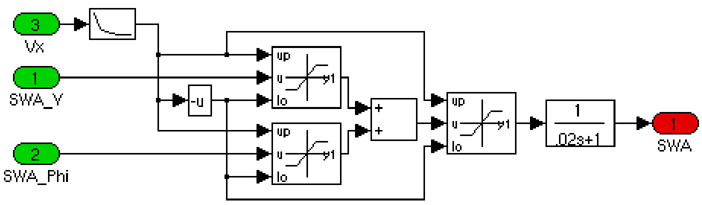


Figure A.5: The combined wheel steering angle with limitations

A.1.2 Velocity

The speed of the vehicle is dependent on which gear and throttle that the driver uses. To be able to change gear the driver also have to be able to control the clutch. In the simulations on Mantorp the driver also have to be able to brake. Therefor the brake signal is included in the model although it isn't used. The driver compares the velocity of the vehicle and what velocity is expected. If the velocity of the vehicle is too high the brakes are applied and if the velocity is too low the throttle is applied, Figure A.6. To be able to reach a target speed a PI-regulator is used, Figure A.7. This will also give a quicker response.

If the driver brakes or changes gear the integrator is set to zero, this is to prevent the output to be too large. If the output is large it will take some time to counteract this large output and the velocities will probably be wrong. If the target velocity is 80km/h and a gear change is initiated at 78km/h. The error of 2km/h will be integrated for as long as the gear change takes place which probably will give a maximum throttle at 81km/h before reducing.

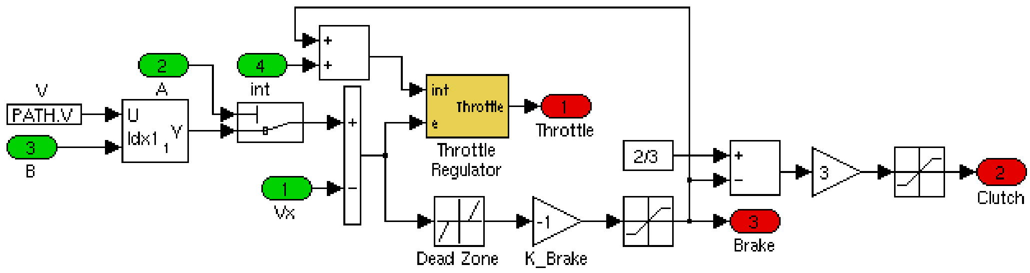


Figure A.6: The driver throttle and brake input to follow the velocity of the path

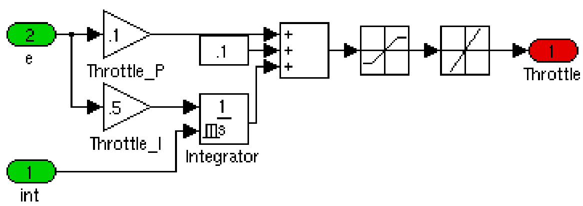


Figure A.7: PI-regulator for the throttle with the ability to put the integrator to zero

The gears are defined for different velocities and if a boundary is crossed a gear change is initiated. The driver reduces the throttle, engages the clutch and changes the gear. Then the clutch is disengaged and the throttle is increased again. During the gear change the integrator part of the throttle is set to zero. Every gear has a minimum and a maximum speed. These speeds are set to get the most power out of the engine.

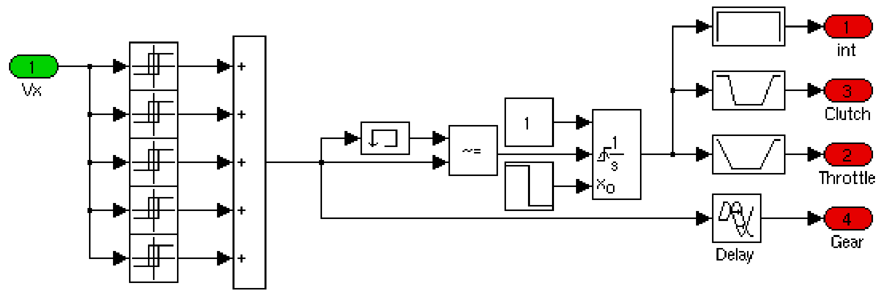


Figure A.8: Driver gear change with throttle and clutch

A.1.3 Friction

The friction is calculated in approximately the same way as the velocity and position of the path. The difference is that the friction is the actual friction of where the vehicle is located and not in the point where the driver looks. The friction is individual for each tire and not for the ground.

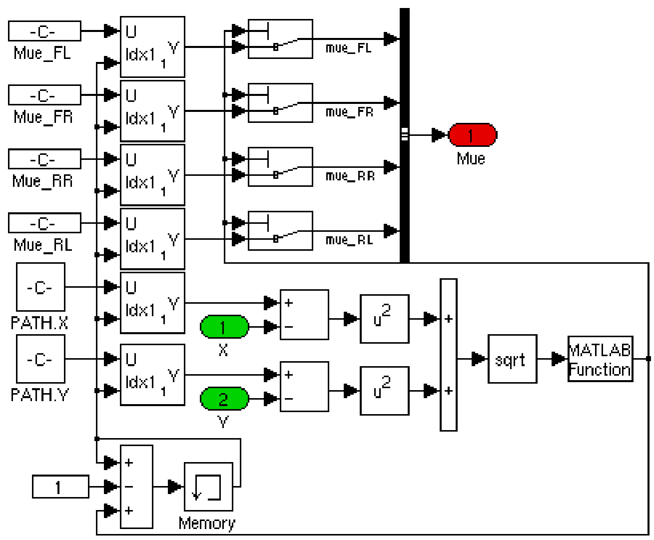


Figure A.9: Friction that acts on the tires

A.2 Chassis and tires

The chassis and tire model are described in Section 3.2. The model consists of the four wheels, the planar motion of the vehicle and the load transfer.

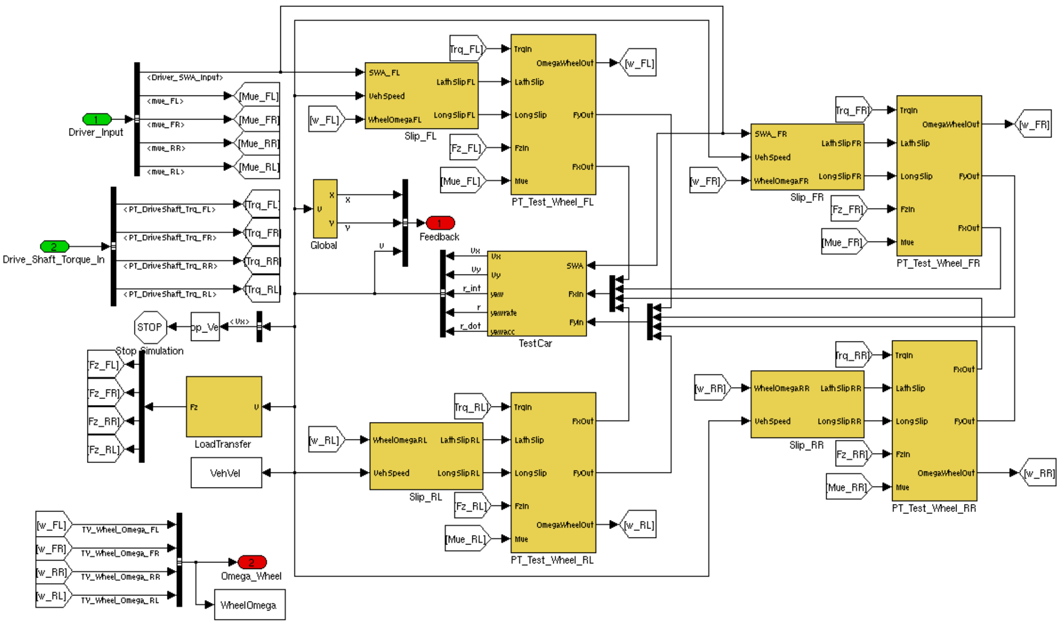


Figure A.10: The chassis and the tires of the vehicle

A.2.1 Wheel

To calculate the forces that are generated from the tires, the longitudinal and lateral slips first has to be calculated. Figure A.11 shows the slips for the front left wheel. How they are calculated are further described in Section 2.2. Figure A.12 shows how the slips are used to calculate the forces. The left part of the model is the tire model that is described in Section 3.2. The tire model gives the forces in the longitudinal and the lateral directions. The longitudinal force accelerates or decelerates the wheel rotational speed.

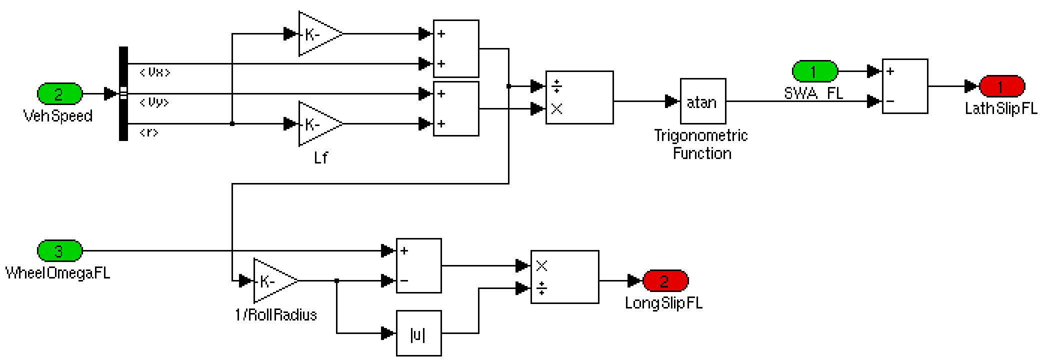


Figure A.11: Lateral and longitudinal slip of the front left tire

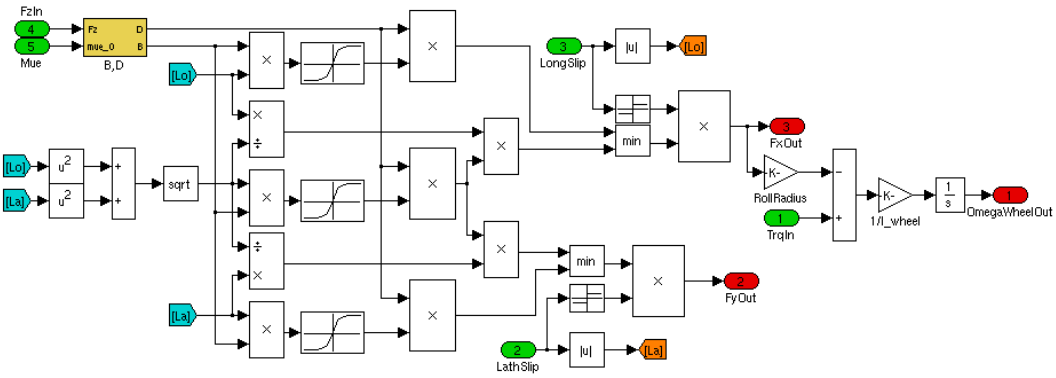


Figure A.12: Tire model and rotational speed of the front left tire

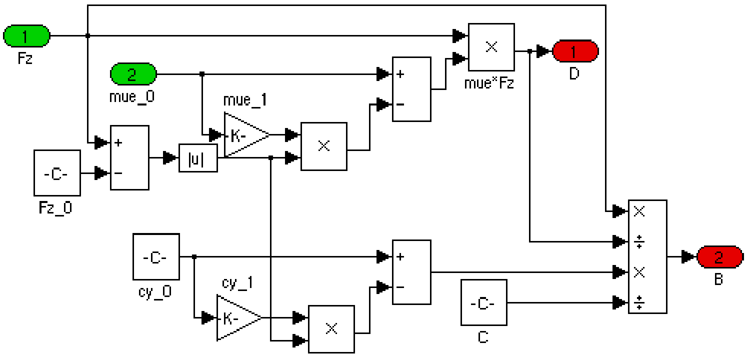


Figure A.13: Part of the tire model

A.2.2 Vehicle motion

The forces from the wheels are used to calculate the motion of the vehicle, Figure A.14. The model consists of the equations that are presented in Section 2.1. The Velocities of the vehicle are transformed into global positions, Figure A.15.

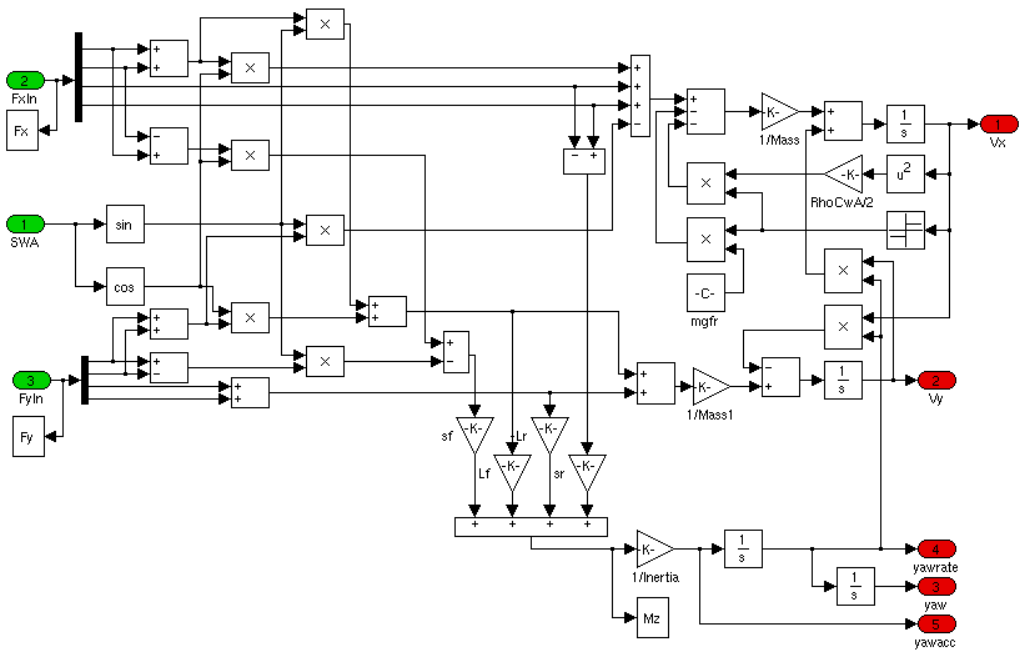


Figure A.14: Converting the forces to accelerations to velocities of the vehicle

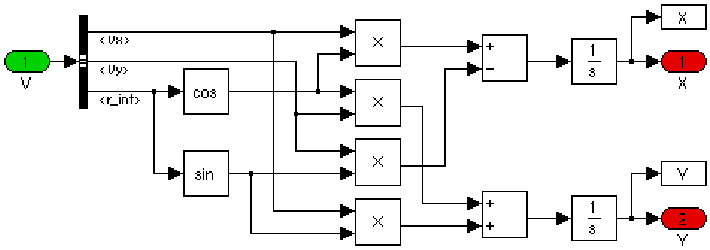


Figure A.15: Calculating the global position from the velocities of the vehicle

A.2.3 Load transfer

Because of the lateral and longitudinal acceleration of the vehicle the weight distribution continuously changes. How it changes is shown in Figure A.16.

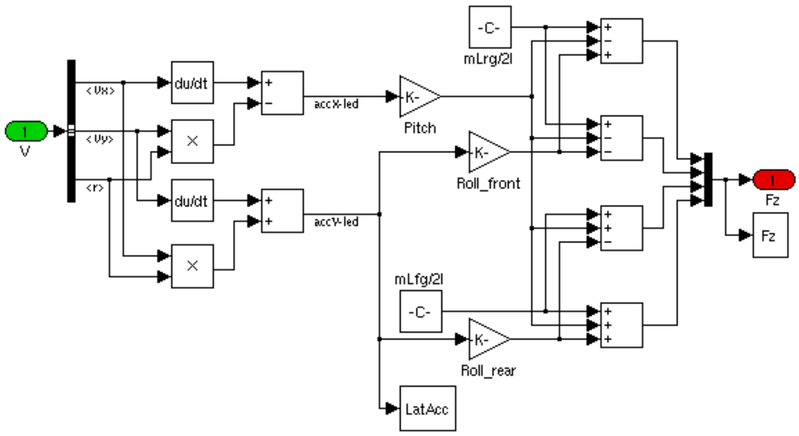


Figure A.16: Calculating the load transfer of the vehicle

A.3 DSLD

The model of the DSLD is divided into two parts, first the state is calculated and then the torque from the differential is calculated.

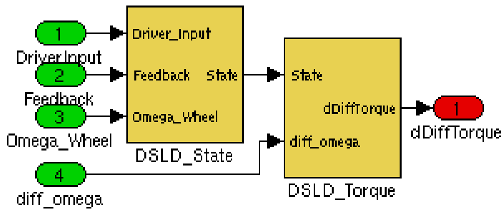


Figure A.17: DSLD

A.3.1 DSLD state

As described in Section 4.1 the states are chosen as in Table 4.1. In Figure A.18 it's shown that the states in the simulation also are dependent on throttle and brake inputs from the driver.

Since the driver uses zero throttle input when the brakes are applied, the throttle criteria is harder than the brake criteria. When the vehicle brakes the state should be open to prevent any braking through the differential which could interfere with the ABS.

The control system for the DSLD doesn't determine the state every simulation step. The state is chosen every tenth of a second using a clock that can be reset. The state is decided every time the clock resets and stays constant in between.

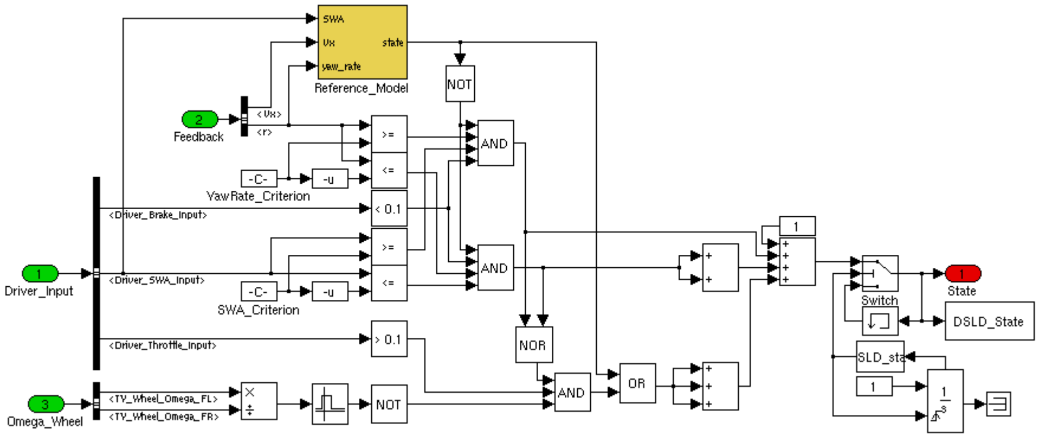


Figure A.18: Determining the states of the DSLD

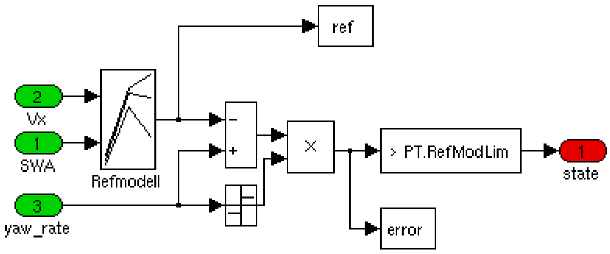


Figure A.19: The reference model that is used to determine the states of the DSLD

A.3.2 DSLD torque

To model the differential an integrator is used, this integrates the rotational velocity difference between the two outgoing shafts. This is the angular difference between the shafts. This is also how the actual differential works. When the angle is above a certain value it starts to produce torque. This can be compared to the real DSLD that looks when the rollers has travelled a certain angle.

The torque is calculated from a spring stiffness and damping stiffness, later this torque is added to the torque from the original differential on one side and subtracted from the other side. If the angle is positive the torque always have to be positive to correspond with the real differential. Since there is a damper it can happen that the torque becomes negative for a positive angle. The torque is therefor limited to either a maximum of zero for negative angles or a minimum of zero for positive angles.

When the differential is in an open mode the rollers stays at zero degrees angle. In the model this means that the integral is set to zero when the state is set to open. Although a locked differential stays locked even if the state is open. This means that the angle must return to zero before the integral is set to zero which is the same as the DSLD unlocks itself.

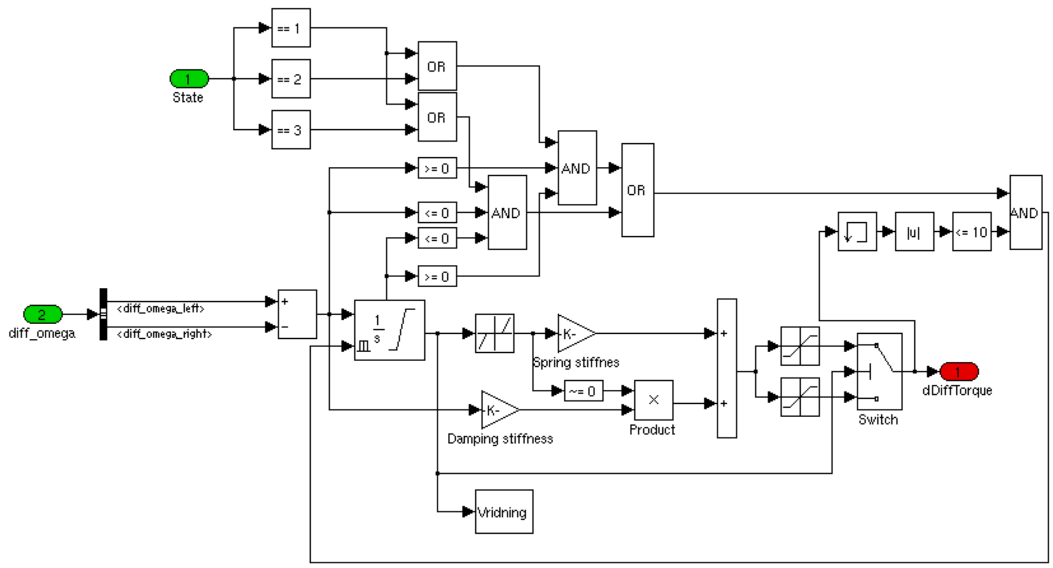


Figure A.20: Torque generated from the DSLD

B Simulation procedure

A few parameters are changed between all simulations. The main difference is the path that the driver should follow including for instance velocity and friction. These parameters are described in section 3.3 and 3.4. Also a part of the driver could be disconnected or replaced, for instance the vehicle velocity input to be able to drive in the same gear or simply replacing the driver steer output in the Sine with Dwell.

B.1 Split-μ and Checkboard acceleration

Two simulations are performed on the Split-μ acceleration and two on the Checkboard acceleration with the only difference in the driver. One simulation is with a normal driver and one is with a slightly modified driver. The modified driver has a constant velocity input to the gear change model. This leads to that the driver doesn't change gear while accelerating.

The important values of the simulations are presented in Table B.1.

Table B.1: Simulation values

	split-μ		checkboard	
Driver	normal	second gear	normal	second gear
Starting velocity [m/s]	1	1	1	1
Starting gear	1	2	1	2
End velocity [km/h]	100	90	100	90
High-μ	1.0	1.0	1.0	1.0
Low-μ	0.3	0.3	0.3	0.3
length of friction surfaces [m]	inf	inf	10	10

The lower end velocity on the modified driver simulation is due to the rpm of the engine. It's impossible for the vehicle to travel at 100 km/h on second gear. Full throttle is always given by the driver since the target velocity is considerable higher than the end velocity. In the split-μ simulation the driver follows a straight line with high-μ on the left side and low-μ on the right side. In the Checkboard simulation the friction changes between left and right side every ten meters.

The path is defined in such way that the friction is specific for a certain tire. The friction is given from where the CoG is located on the path. The left and right wheels never has the same friction even if the vehicle should drift sideways so that the left wheels would be on the right side of the path. The length between the axles determine the friction for the front and rear wheels. If the vehicle has a low yaw the friction on each tire are very similar to real life but if the vehicle has a large yaw the path is not as good. There are also sharp edges between the high and low-μ areas.

B.2 90 degree turn

This is to simulate an acceleration in a corner. In the beginning the driver follows a straight line at a constant speed, the path makes a ninety degree turn and the driver accelerate. Table B.2 shows the important information for the simulation.

Table B.2: Simulation values

Starting velocity	14	m/s
Turn radius	30	m
Acceleration point	27	deg
Target velocity	18	m/s

B.3 Sine with Dwell

In the sine with dwell test the driver doesn't follow any path but uses the steering wheel as described in section 5.2.1. The starting velocity is set slightly higher than 80km/h, the driver disengages the clutch which will make the car roll. Rolling and air resistance will make the vehicle slow down and when the velocity is equal to 80km/h the turning sequence is started. The clutch is disengaged during the whole simulation.

B.4 Worst case

During this simulation the driver keeps a constant steering angle, 0.08 radians at the wheels, and a constant velocity, 15 m/s. At the beginning of the simulation the differential is set to state four, locked, during a small time. This state isn't normal for these driving conditions. Since the driver keeps a constant velocity this leads to that the differential looks when the outer wheel rotates faster than the inner wheel. This is done to make the differential look in the wrong direction.

Since the differential is looked in a unwanted way a brake input signal is sent to the outer wheel to make the differential self unlock. Two different simulations are done, one is when the differential is set to its normal turning mode, and one where the differential is open. When in its normal mode the inner wheel isn't allowed to rotate faster than the outer wheel and in the open mode it is.

After two seconds brake input signals are sent to the outer wheel. Eight different lengths of signals are used, from 0.05 seconds to 0.40 seconds.

C Simulation results

The results from the simulations are presented as figures

C.1 Split-μ acceleration

The split friction simulations are done with and without gear shifts. The simulation without gear shifts uses second gear all the way.

C.1.1 Gear shift

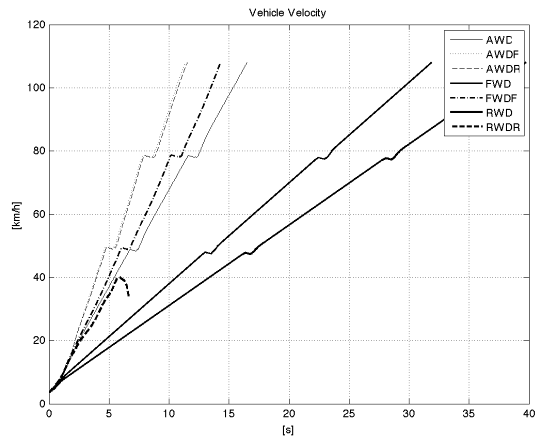


Figure C.1.1.1: Vehicle velocity vs. time

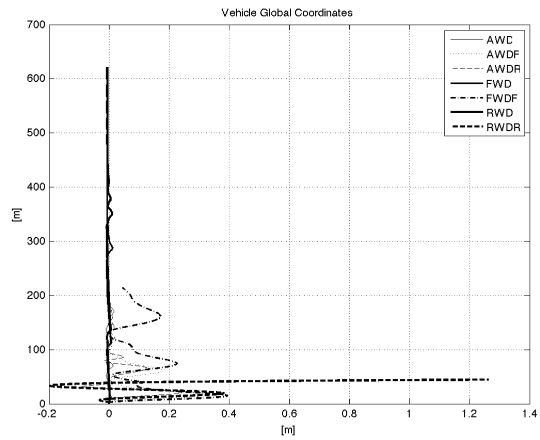


Figure C.1.1.2: Vehicle global coordinates

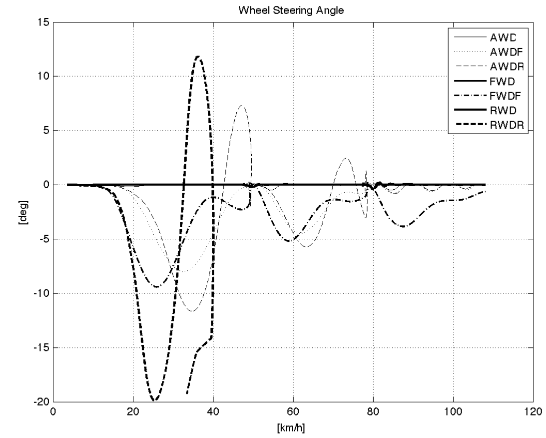


Figure C.1.1.3: Steering angle at the wheel vs. velocity

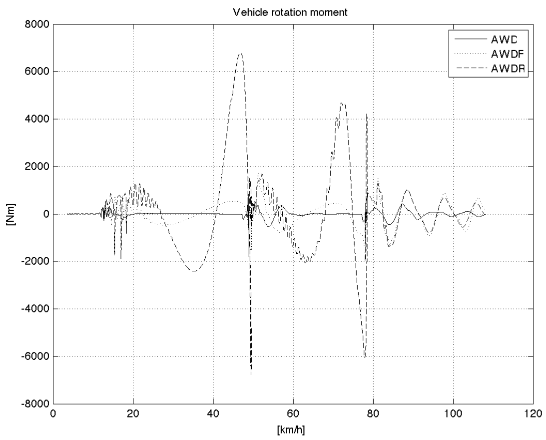


Figure C.1.1.4: Vehicle rotation moment vs. velocity for AWD vehicles

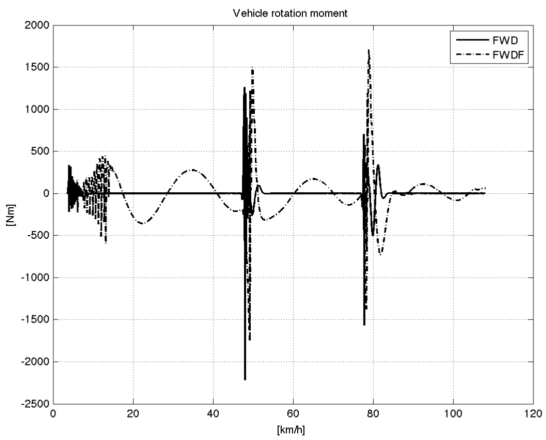


Figure C.1.1.5: Vehicle rotation moment vs. velocity for FWD vehicles

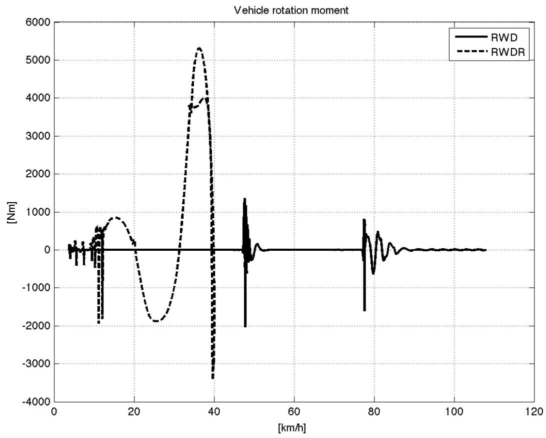


Figure C.1.1.6: Vehicle rotation moment vs. velocity for RWD vehicles

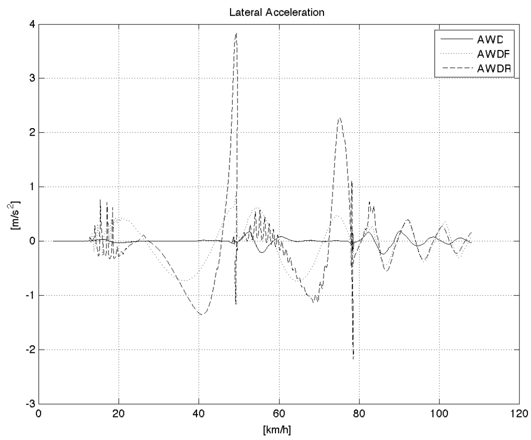


Figure C.1.1.7: Vehicle lateral acceleration vs. velocity for AWD vehicles

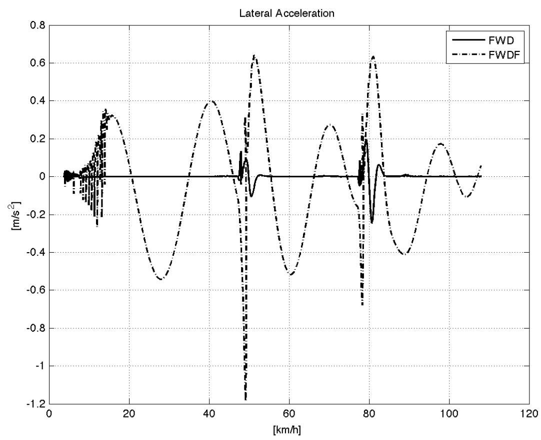


Figure C.1.1.8: Vehicle lateral acceleration vs. velocity for FWD vehicles

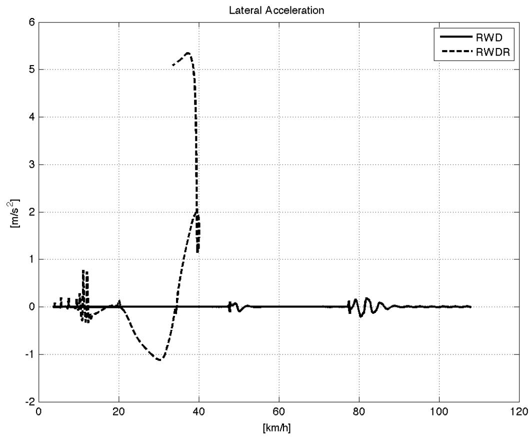


Figure C.1.1.9: Vehicle lateral acceleration vs. velocity for RWD vehicles

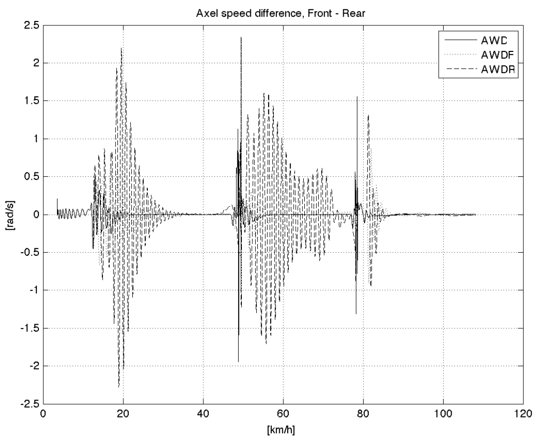


Figure C.1.1.10: Difference in axle rotational speed vs. velocity for AWD vehicles

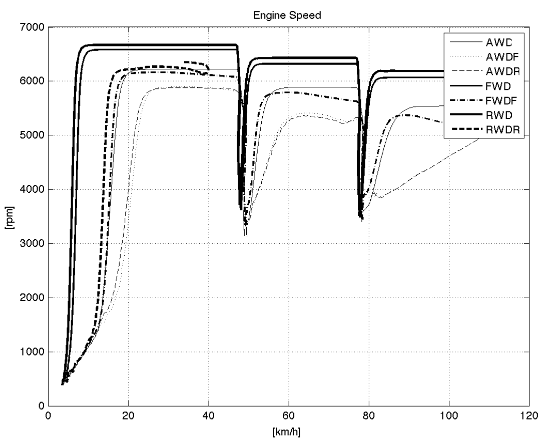


Figure C.1.1.11: Engine rotational speed vs. velocity

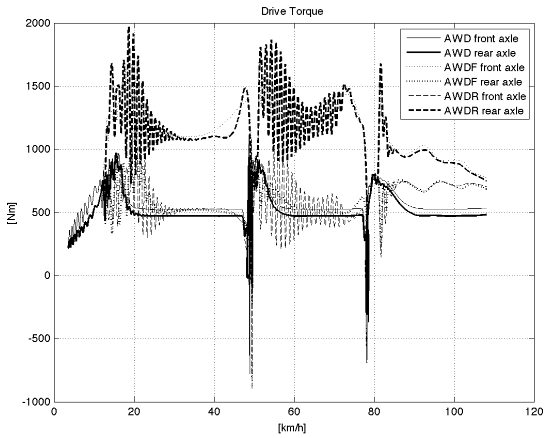


Figure C.1.1.12: Drive torque on front and rear axle vs. velocity for AWD vehicles

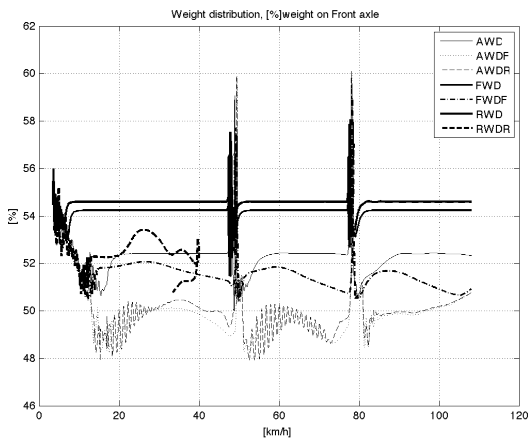


Figure C.1.1.13: Vehicle weight distribution vs. velocity for AWD vehicles

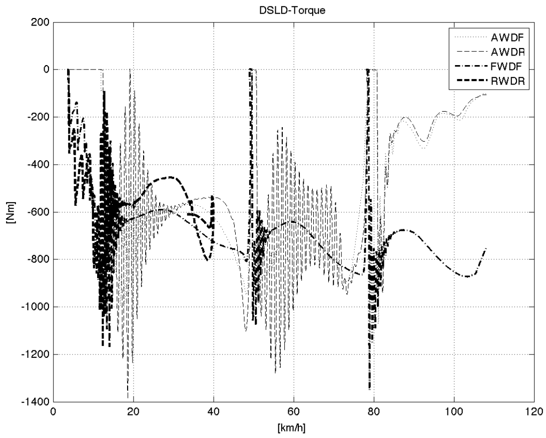


Figure C.1.1.14: DSLD torque vs. velocity

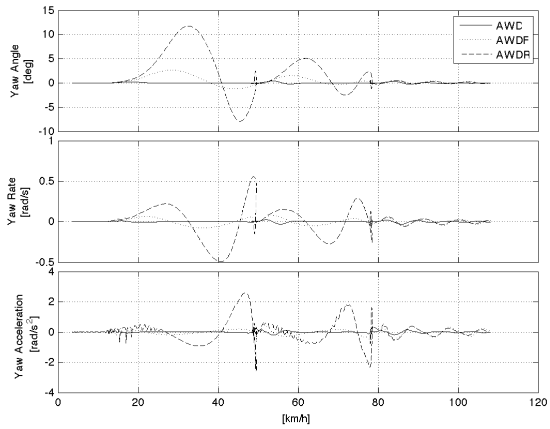


Figure C.1.1.15: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for AWD vehicles

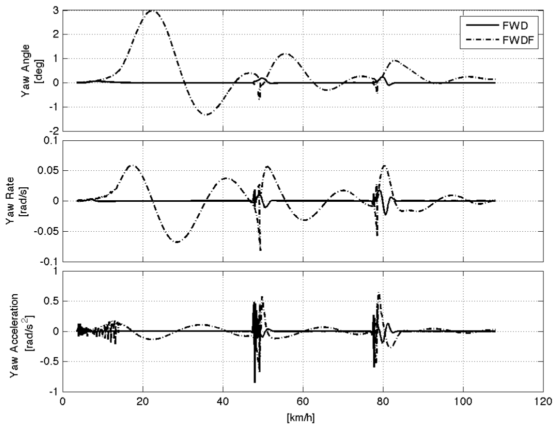


Figure C.1.1.16: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for FWD vehicles

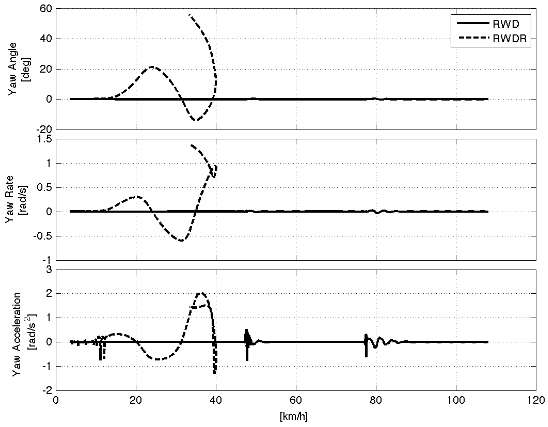


Figure C.1.1.17: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for RWD vehicles

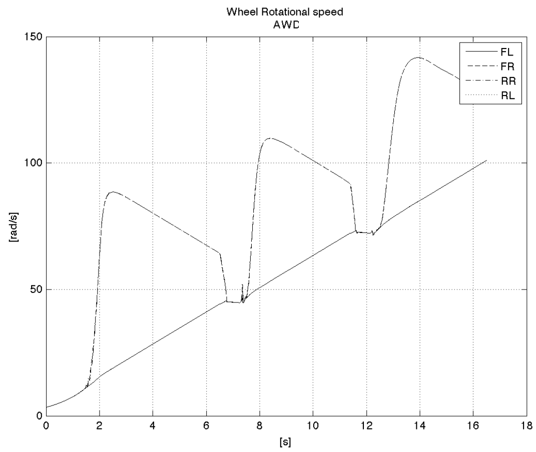


Figure C.1.1.18: Wheel rotational speeds vs time for AWD without DSLD

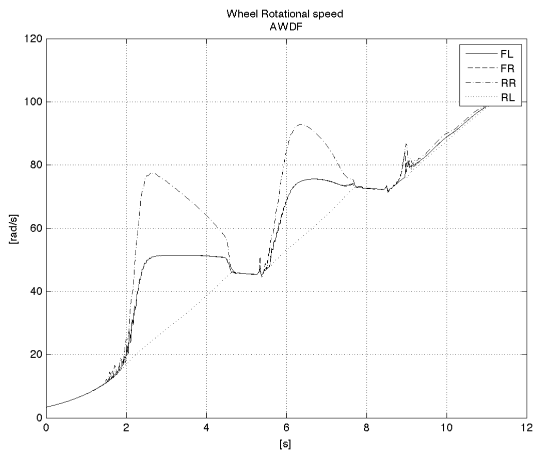


Figure C.1.1.19: Wheel rotational speeds vs time for AWD with DSLD on front axle

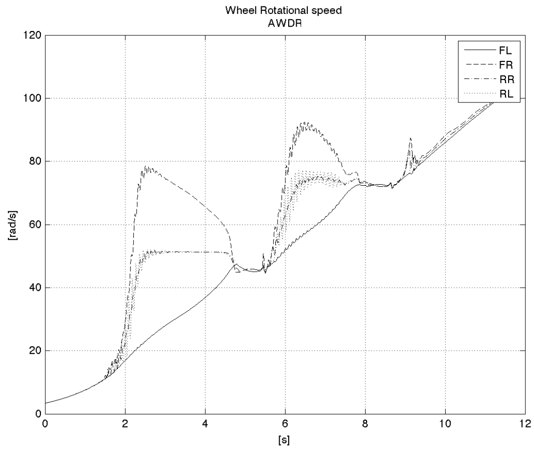


Figure C.1.1.20: Wheel rotational speeds vs time for AWD with DSLD on rear axle

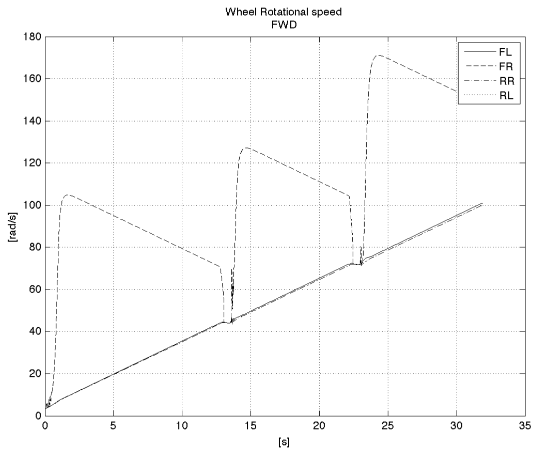


Figure C.1.1.21: Wheel rotational speeds vs time for FWD without DSLD

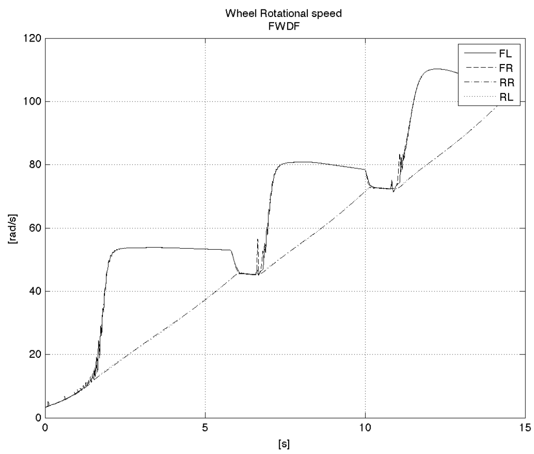


Figure C.1.1.22: Wheel rotational speeds vs time for FWD with DSLD

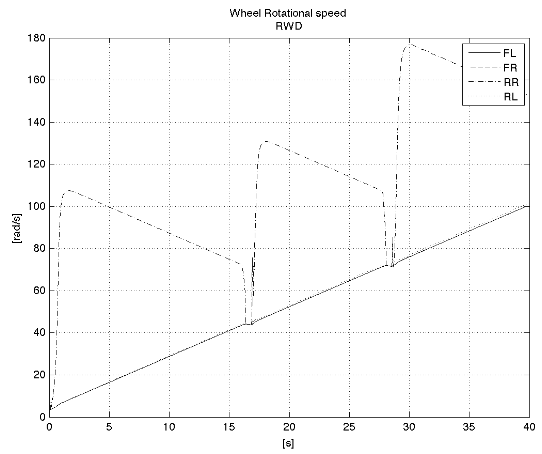


Figure C.1.1.23: Wheel rotational speeds vs time for RWD without DSLD

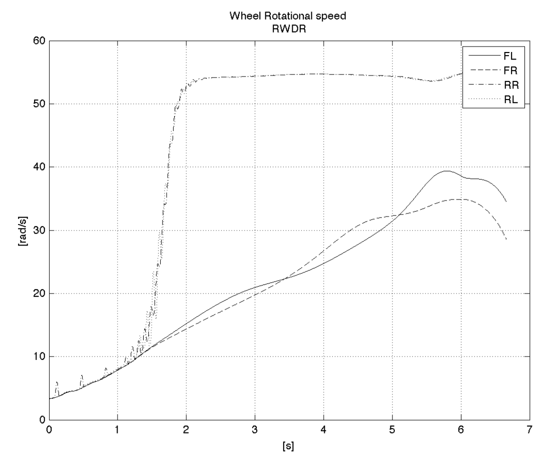


Figure C.1.1.24: Wheel rotational speeds vs time for RWD with DSLD

C.1.2 Second gear

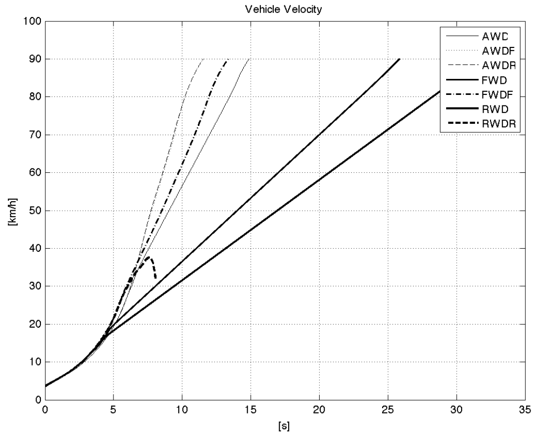


Figure C.1.2.1: Vehicle velocity vs. time

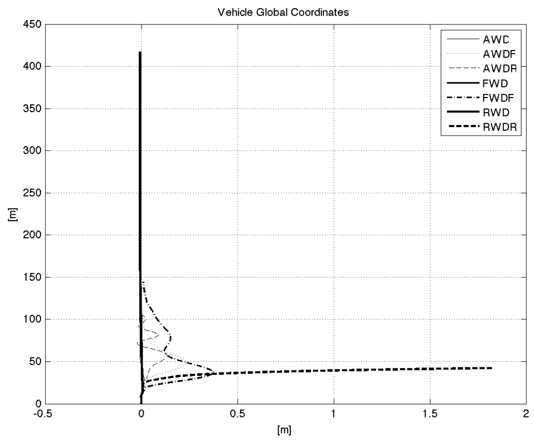


Figure C.1.2.2: Vehicle global coordinates

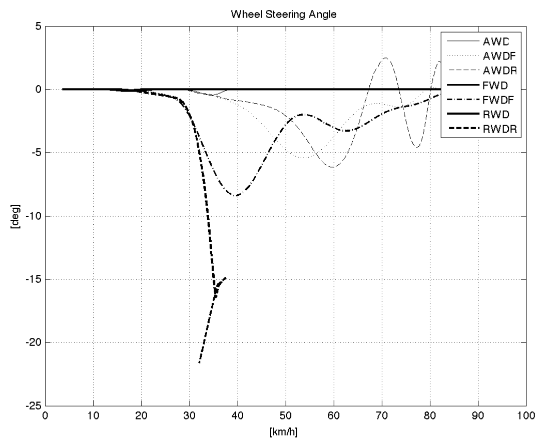


Figure C.1.2.3: Steering angle at the wheel vs. velocity

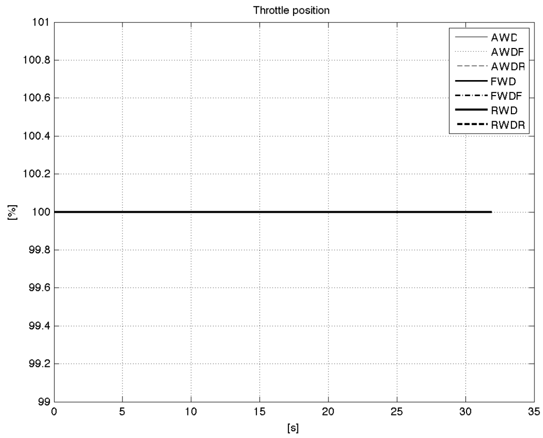


Figure C.1.2.4: Throttle position vs. time

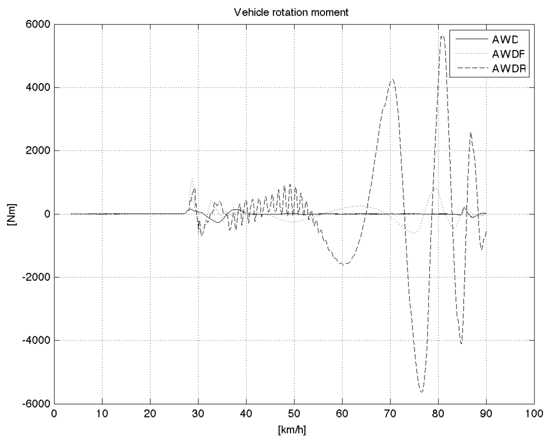


Figure C.1.2.5: Vehicle rotation moment vs. velocity for AWD vehicles

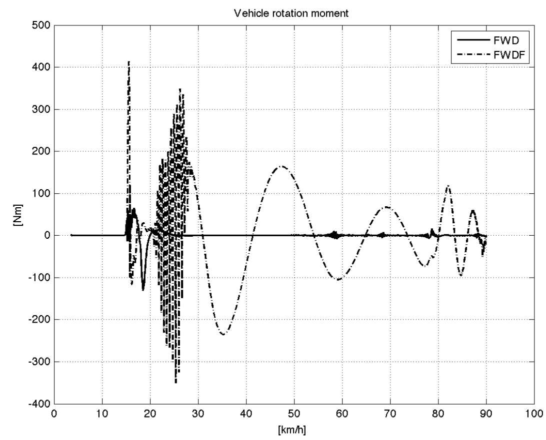


Figure C.1.2.6: Vehicle rotation moment vs. velocity for FWD vehicles

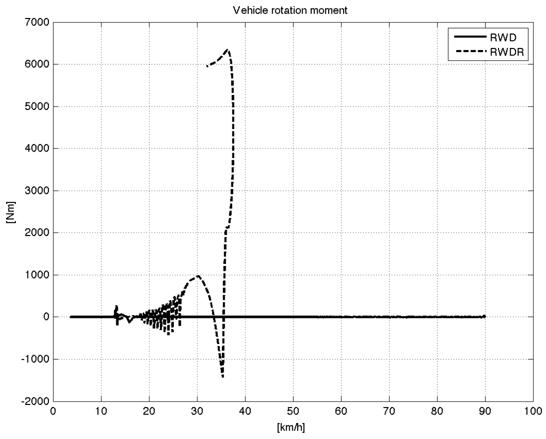


Figure C.1.2.7: Vehicle rotation moment vs. velocity for RWD vehicles

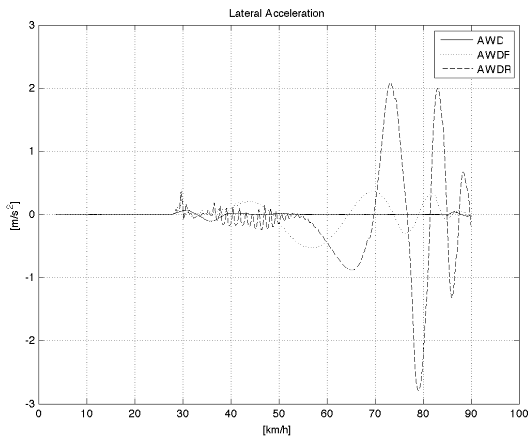


Figure C.1.2.8: Vehicle lateral acceleration vs. velocity for AWD vehicles

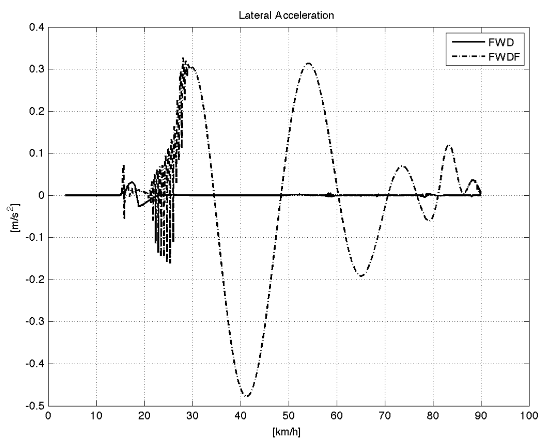


Figure C.1.2.9: Vehicle lateral acceleration vs. velocity for FWD vehicles

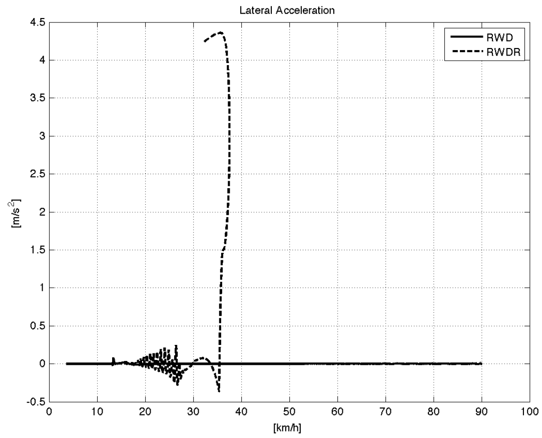


Figure C.1.2.10: Vehicle lateral acceleration vs. velocity for RWD vehicles

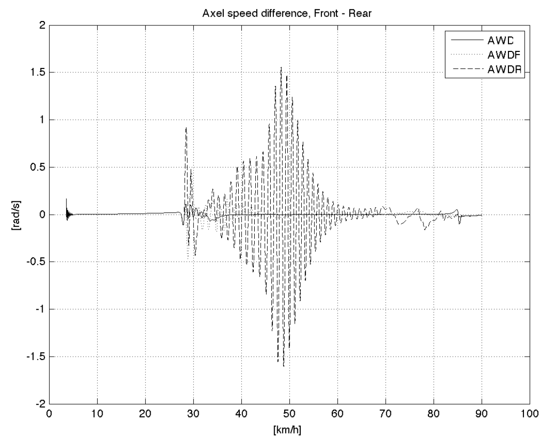


Figure C.1.2.11: Difference in axle rotational speed vs. velocity for AWD vehicles

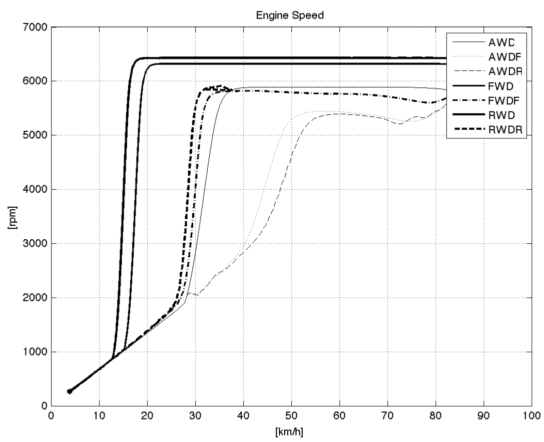


Figure C.1.2.12: Engine rotational speed vs. velocity

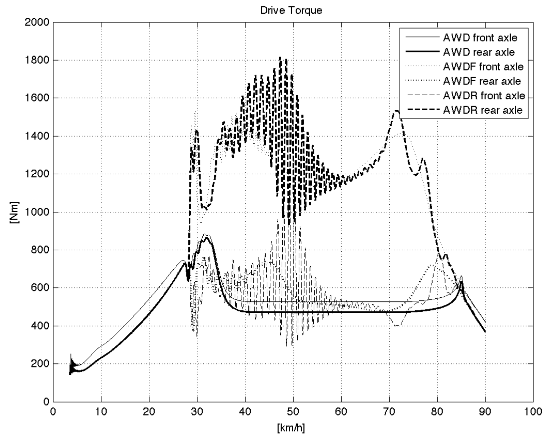


Figure C.1.2.13: Drive torque on front and rear axle vs. velocity for AWD vehicles

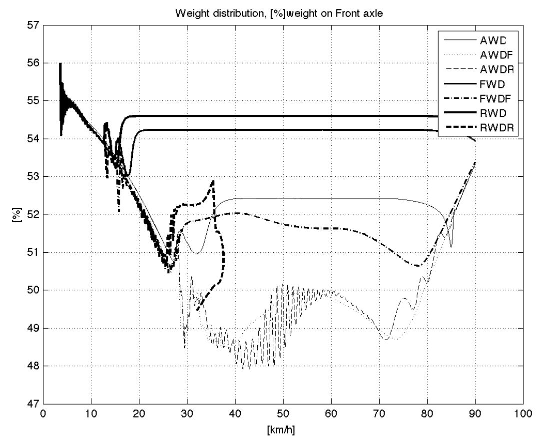


Figure C.1.2.14: Vehicle weight distribution vs. velocity for AWD vehicles

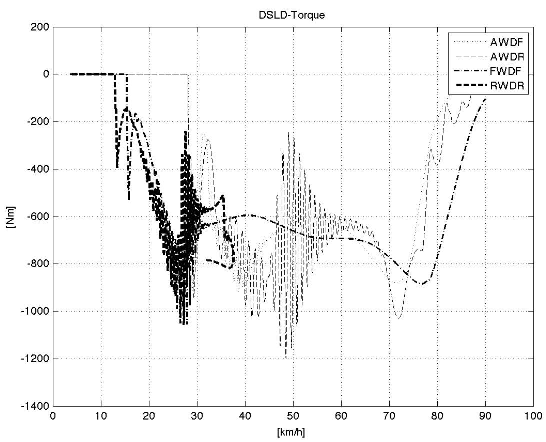


Figure C.1.2.15: DSLD torque vs. velocity

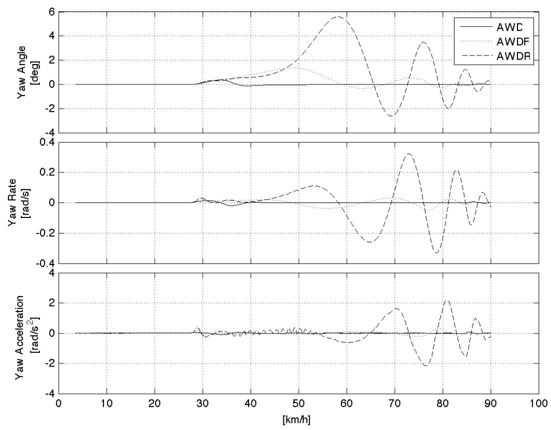


Figure C.1.2.16: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for AWD vehicles

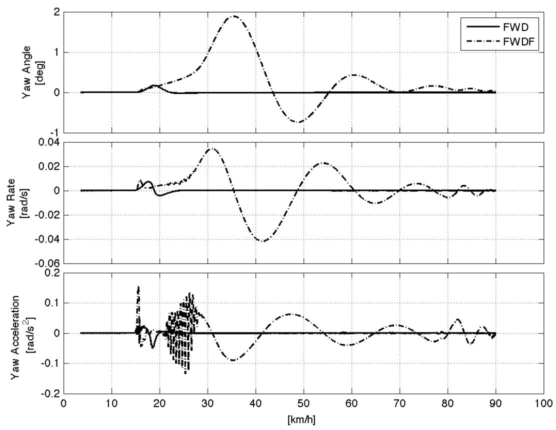


Figure C.1.2.17: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for FWD vehicles

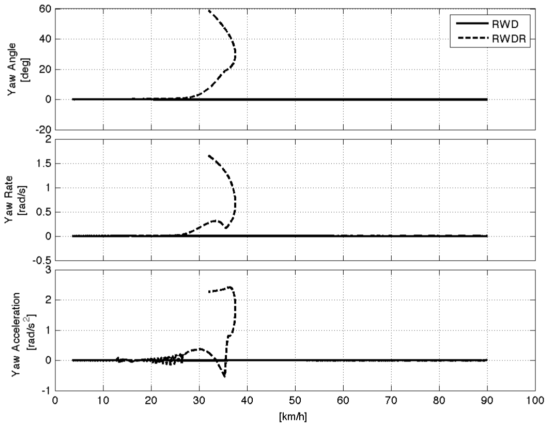


Figure C.1.2.18: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for RWD vehicles

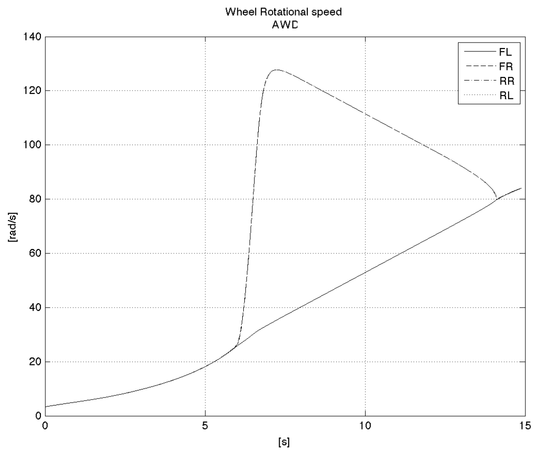


Figure C.1.2.19: Wheel rotational speeds vs time for AWD without DSLD

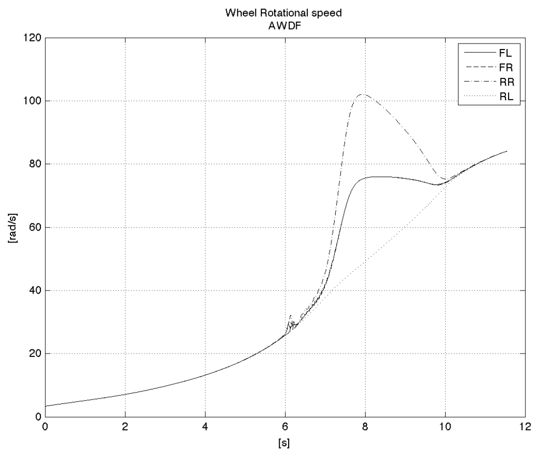


Figure C.1.2.20: Wheel rotational speeds vs time for AWD with DSLD on front axle

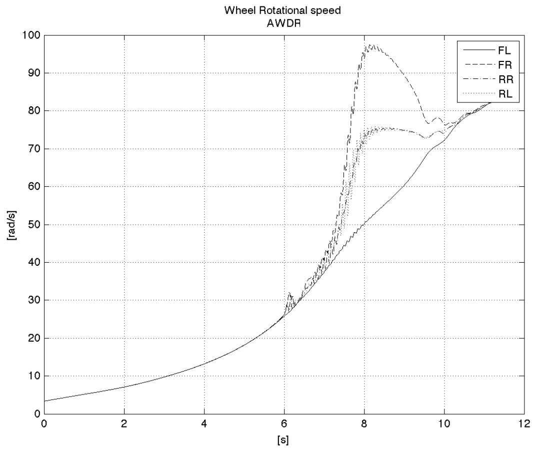


Figure C.1.2.21: Wheel rotational speeds vs time for AWD with DSLD on rear axle

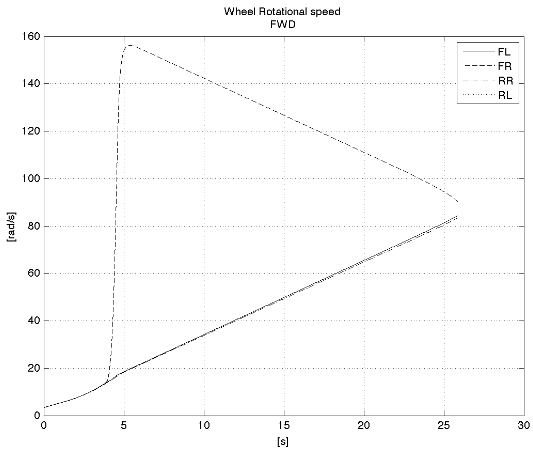


Figure C.1.2.22: Wheel rotational speeds vs time for FWD without DSLD

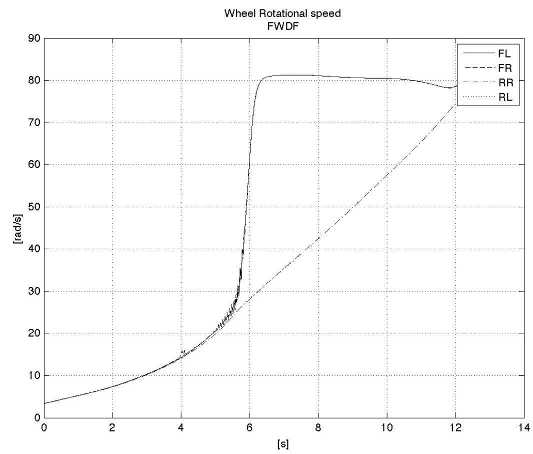


Figure C.1.2.23: Wheel rotational speeds vs time for FWD with DSLD

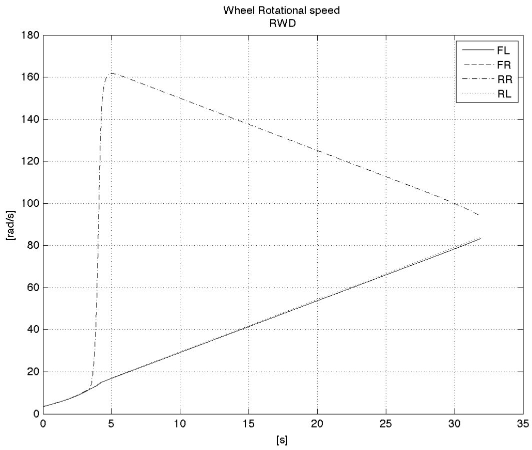


Figure C.1.2.24: Wheel rotational speeds vs time for RWD without DSLD

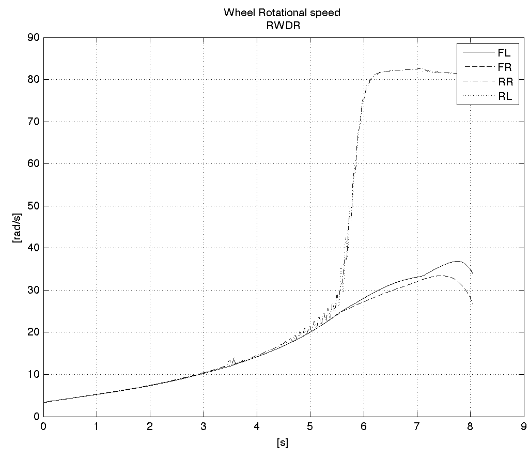


Figure C.1.2.25: Wheel rotational speeds vs time for RWD with DSLD

C.2 Checkboard acceleration

The Checkboard simulations are done with and without gear shifts. The simulation without gear shifts uses second gear all the way.

C.2.1 Gear shift

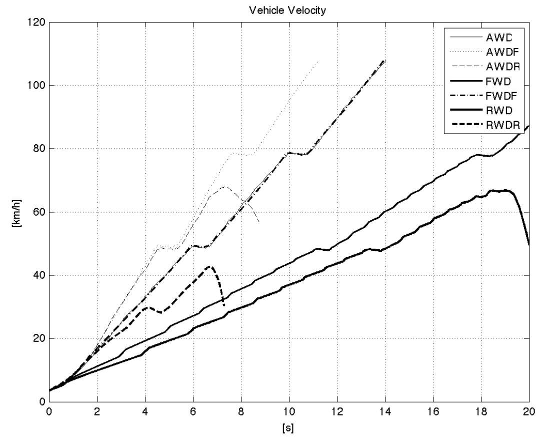


Figure C.2.1.1: Vehicle velocity vs. time

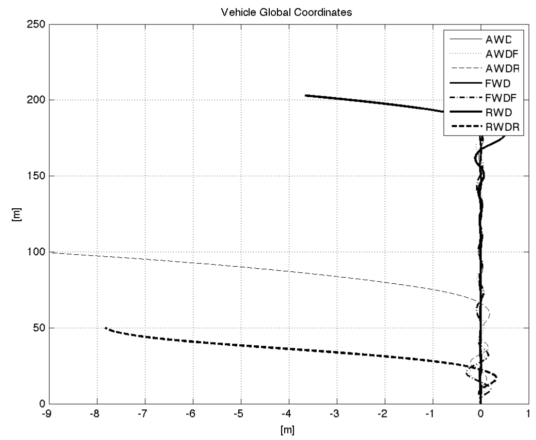


Figure C.2.1.2: Vehicle global coordinates

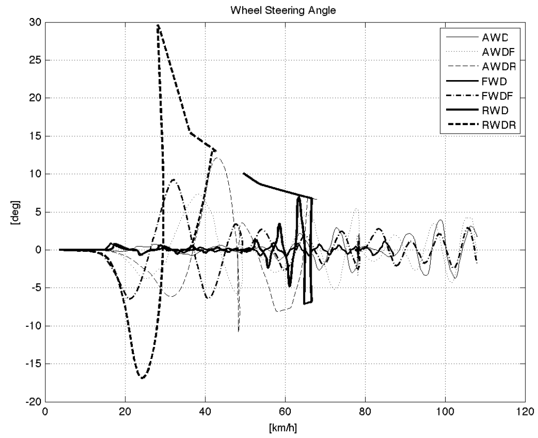


Figure C.2.1.3: Steering angle at the wheel vs. velocity

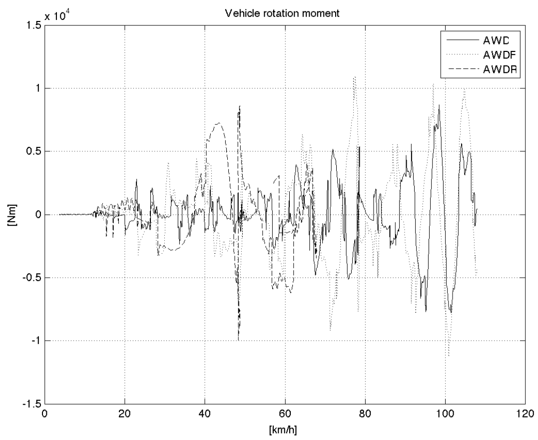


Figure C.2.1.4: Vehicle rotation moment vs. velocity for AWD vehicles

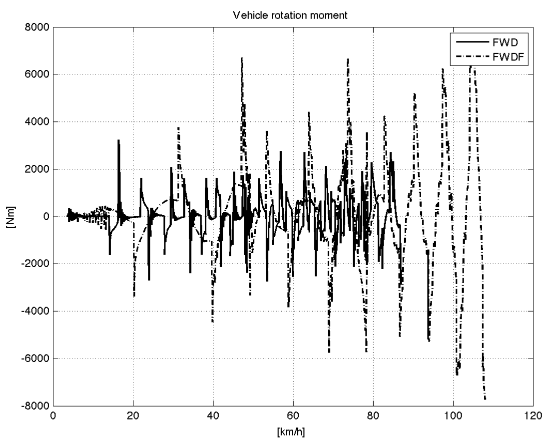


Figure C.2.1.5: Vehicle rotation moment vs. velocity for FWD vehicles

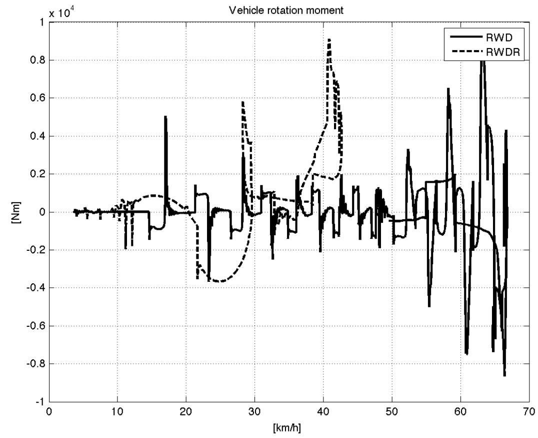


Figure C.2.1.6: Vehicle rotation moment vs. velocity for RWD vehicles

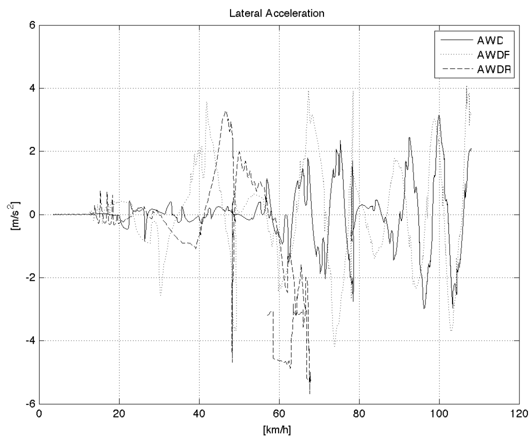


Figure C.2.1.7: Vehicle lateral acceleration vs. velocity for AWD vehicles

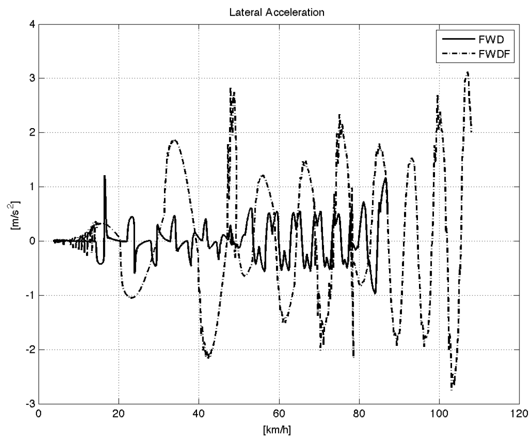


Figure C.2.1.8: Vehicle lateral acceleration vs. velocity for FWD vehicles

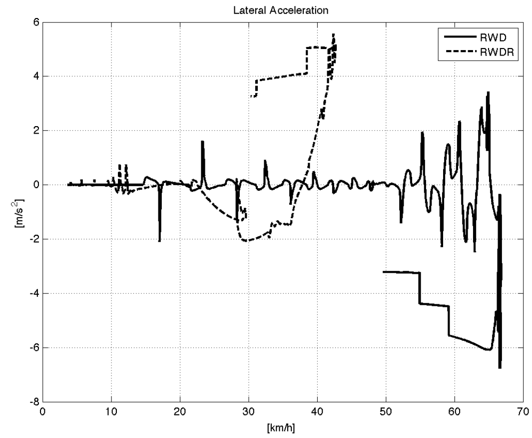


Figure C.2.1.9: Vehicle lateral acceleration vs. velocity for RWD vehicles

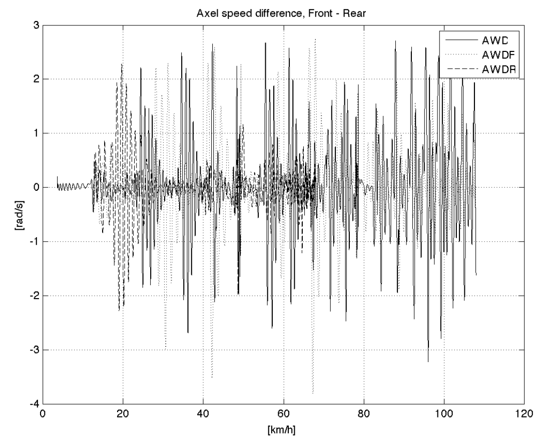


Figure C.2.1.10: Difference in axle rotational speed vs. velocity for AWD vehicles

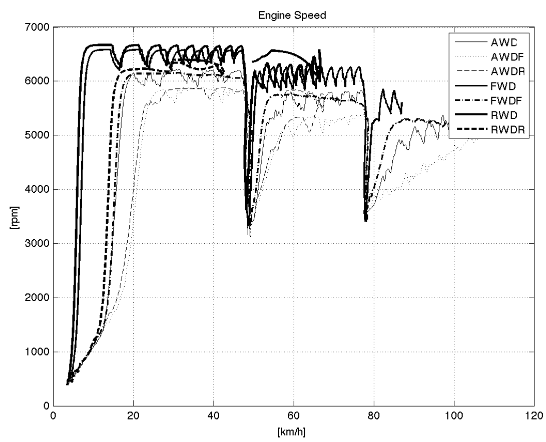


Figure C.2.1.11: Engine rotational speed vs. velocity

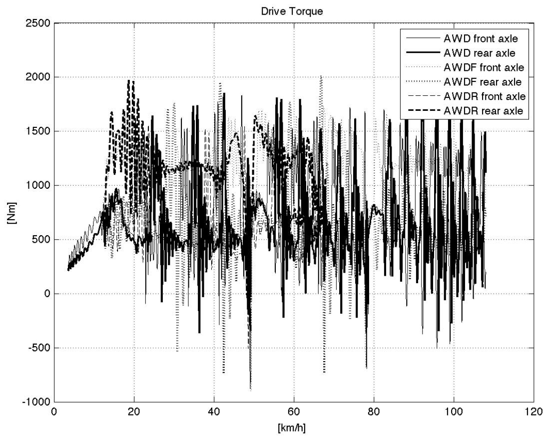


Figure C.2.1.12: Drive torque on front and rear axle vs. velocity for AWD vehicles

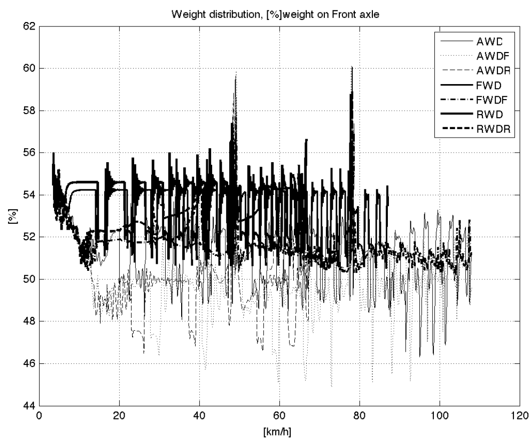


Figure C.2.1.13: Vehicle weight distribution vs. velocity for AWD vehicles

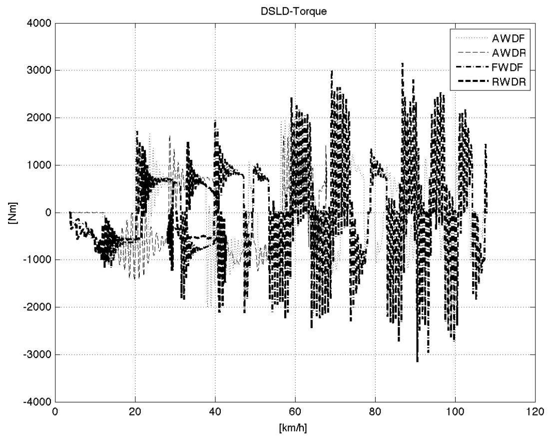


Figure C.2.1.14: DSLD torque vs. velocity

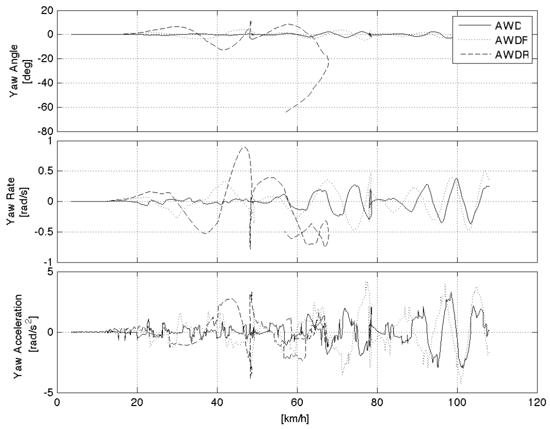


Figure C.2.1.15: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for AWD vehicles

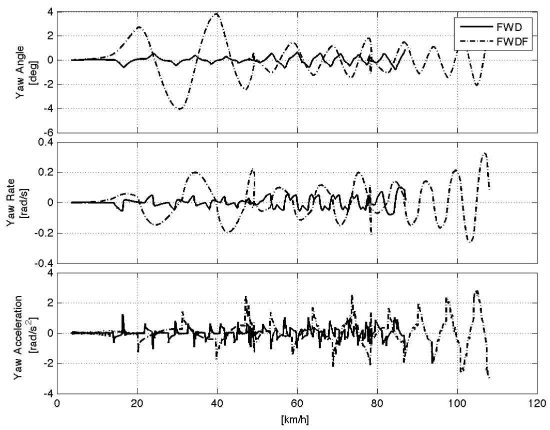


Figure C.2.1.16: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for FWD vehicles

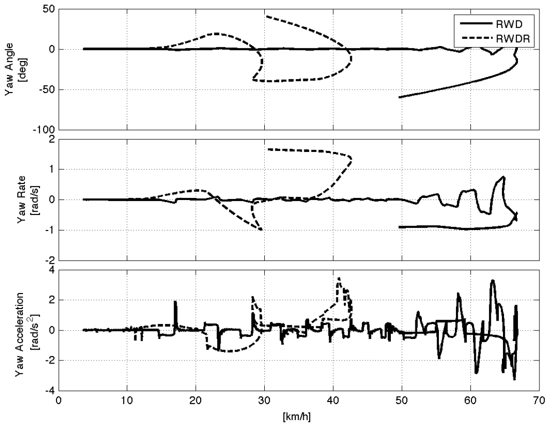


Figure C.2.1.17: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for RWD vehicles

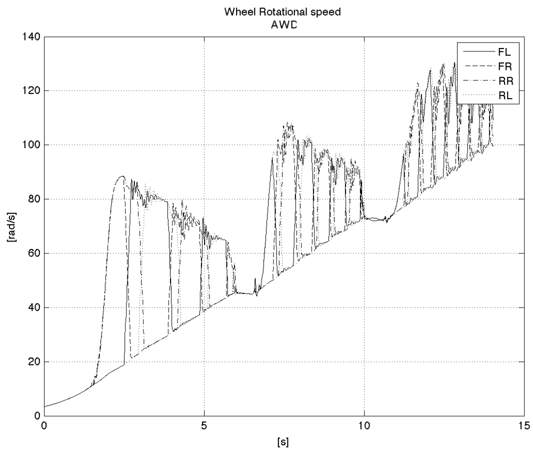


Figure C.2.1.18: Wheel rotational speeds vs time for AWD without DSLD

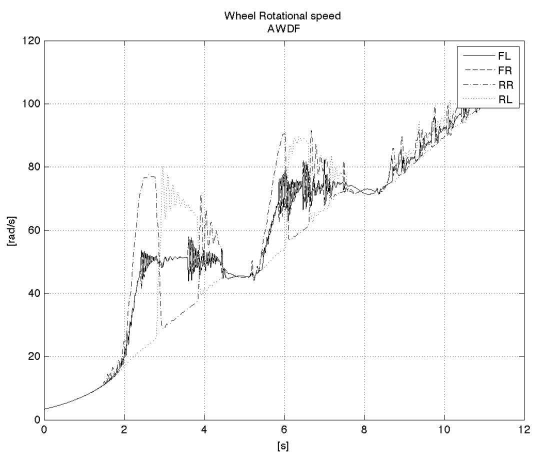


Figure C.2.1.19: Wheel rotational speeds vs time for AWD with DSLD on front axle

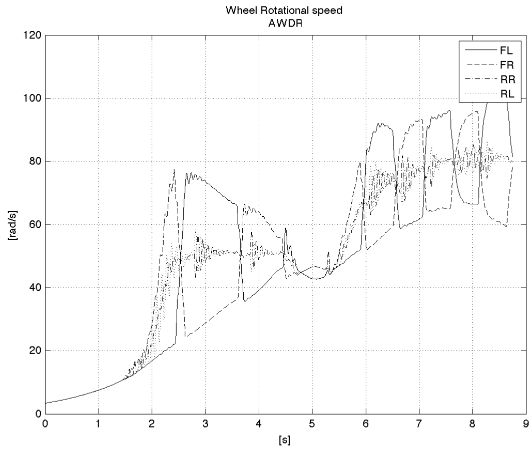


Figure C.2.1.20: Wheel rotational speeds vs time for AWD with DSLD on rear axle

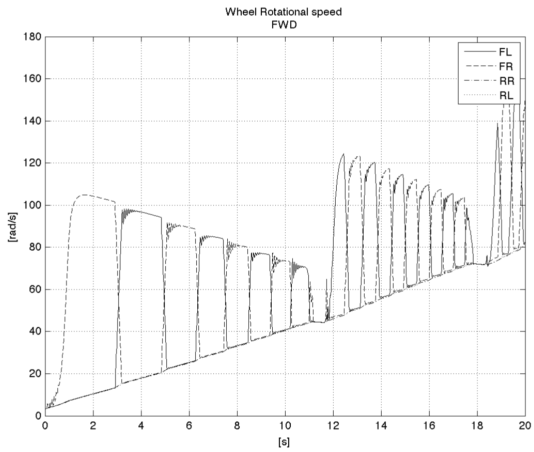


Figure C.2.1.21: Wheel rotational speeds vs time for FWD without DSLD

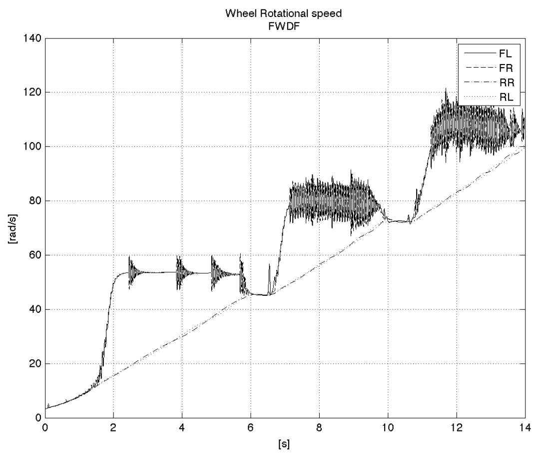


Figure C.2.1.22: Wheel rotational speeds vs time for FWD with DSLD

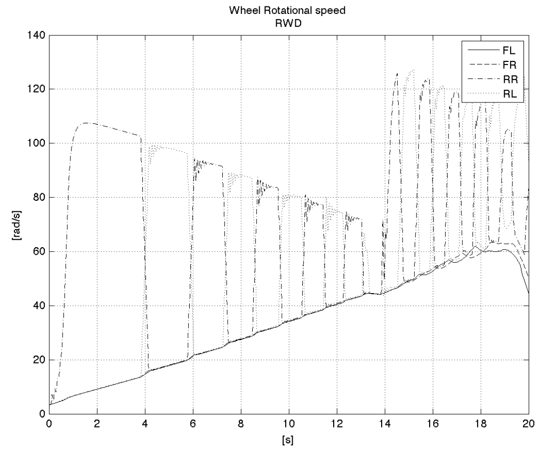


Figure C.2.1.23: Wheel rotational speeds vs time for RWD without DSLD

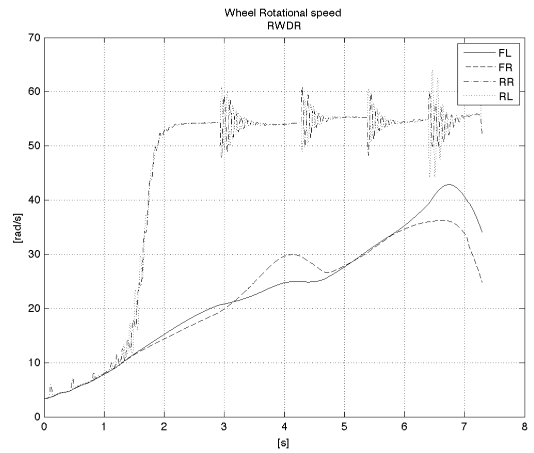


Figure C.2.1.24: Wheel rotational speeds vs time for RWD with DSLD

C.2.2 Second gear

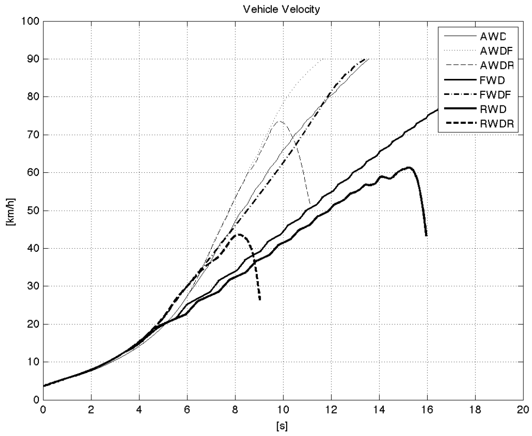


Figure C.2.2.1: Vehicle velocity vs. time

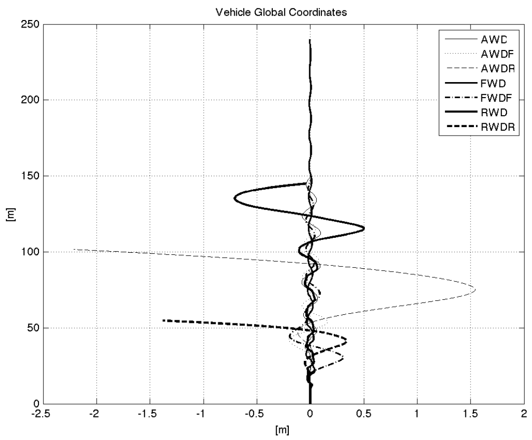


Figure C.2.2.2: Vehicle global coordinates

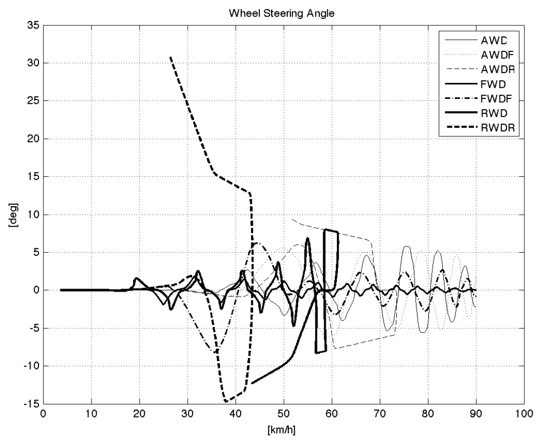


Figure C.2.2.3: Steering angle at the wheel vs. velocity

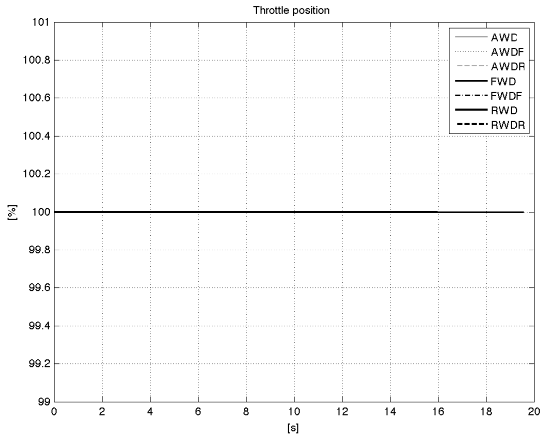


Figure C.2.2.4: Throttle position vs. time

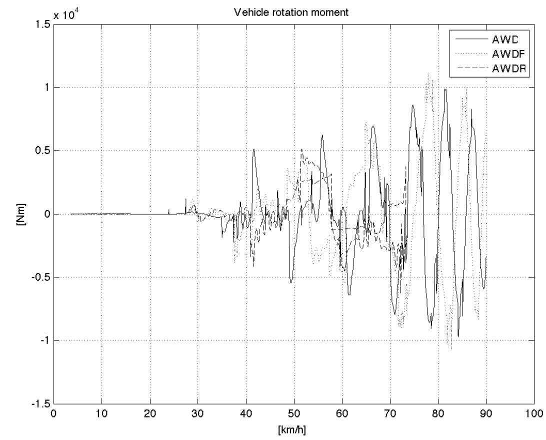


Figure C.2.2.5: Vehicle rotation moment vs. velocity for AWD vehicles

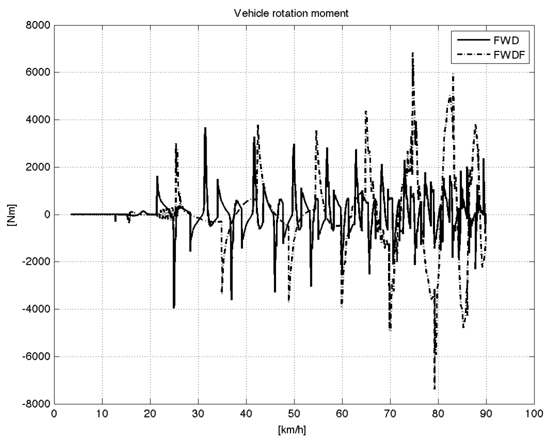


Figure C.2.2.6: Vehicle rotation moment vs. velocity for FWD vehicles

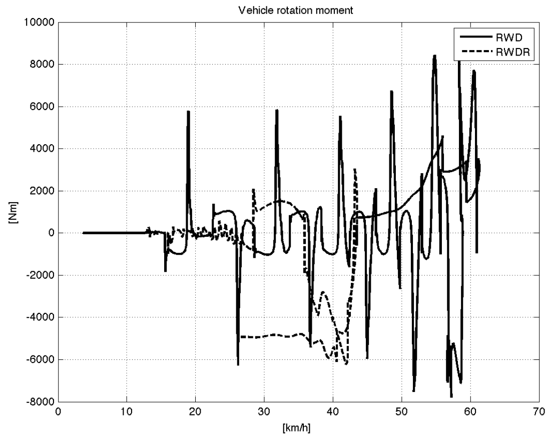


Figure C.2.2.7: Vehicle rotation moment vs. velocity for RWD vehicles

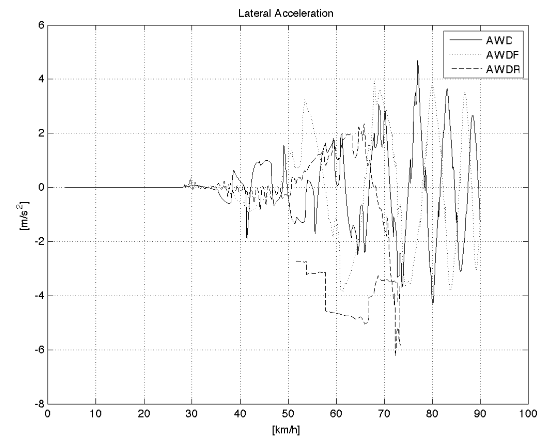


Figure C.2.2.8: Vehicle lateral acceleration vs. velocity for AWD vehicles

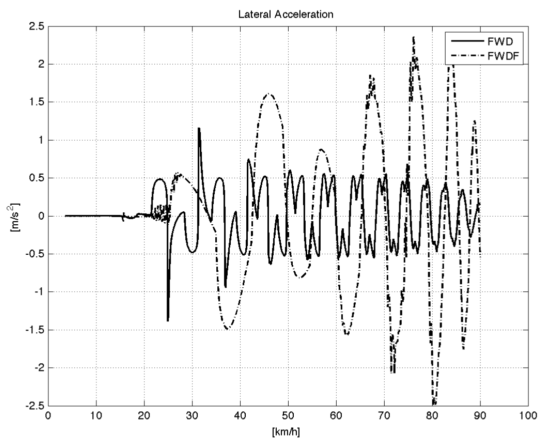


Figure C.2.2.9: Vehicle lateral acceleration vs. velocity for FWD vehicles

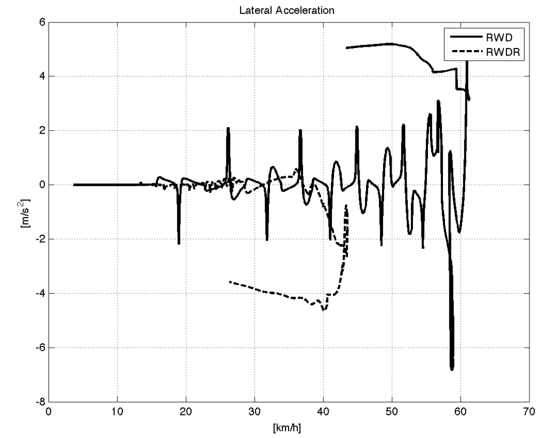


Figure C.2.2.10: Vehicle lateral acceleration vs. velocity for RWD vehicles

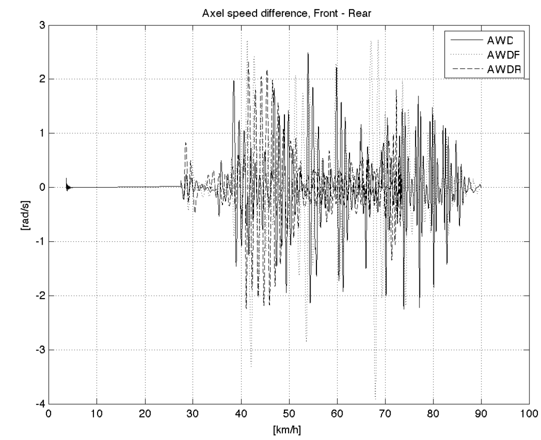


Figure C.2.2.11: Difference in axle rotational speed vs. velocity for AWD vehicles

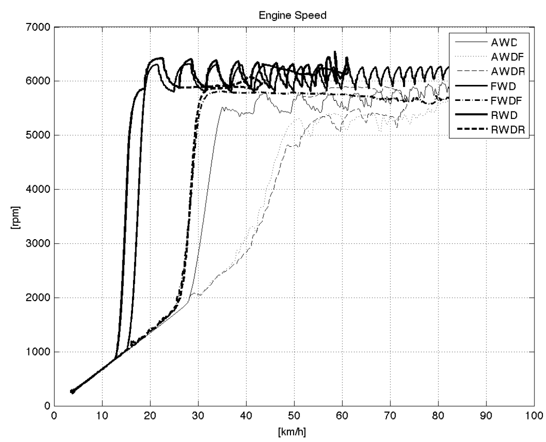


Figure C.2.2.12: Engine rotational speed vs. velocity

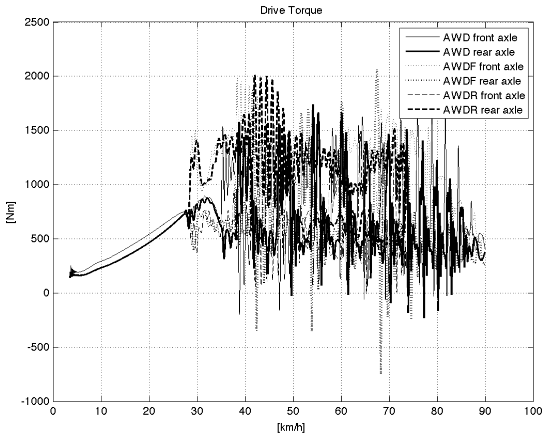


Figure C.2.2.13: Drive torque on front and rear axle vs. velocity for AWD vehicles

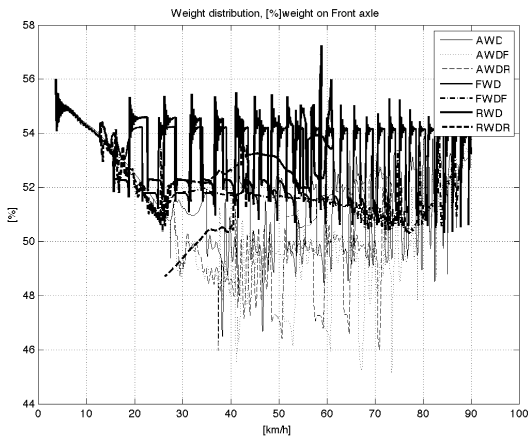


Figure C.2.2.14: Vehicle weight distribution vs. velocity for AWD vehicles

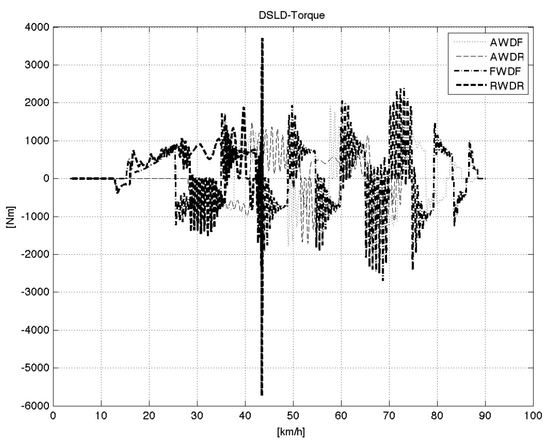


Figure C.2.2.15: DSLD torque vs. velocity

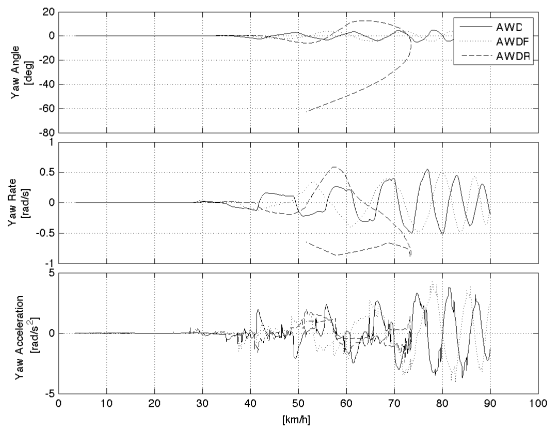


Figure C.2.2.16: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for AWD vehicles

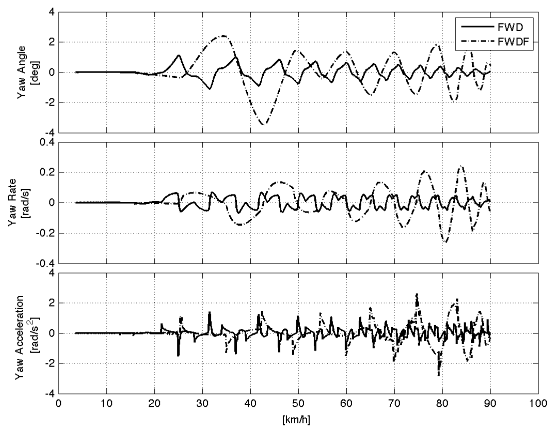


Figure C.2.2.17: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for FWD vehicles

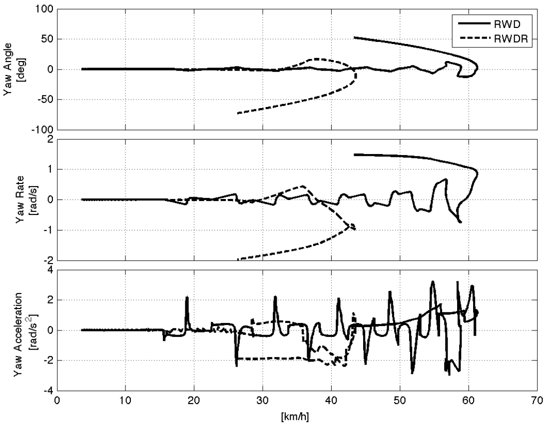


Figure C.2.2.18: Vehicle yaw, yaw rate and yaw acceleration vs. velocity for RWD vehicles

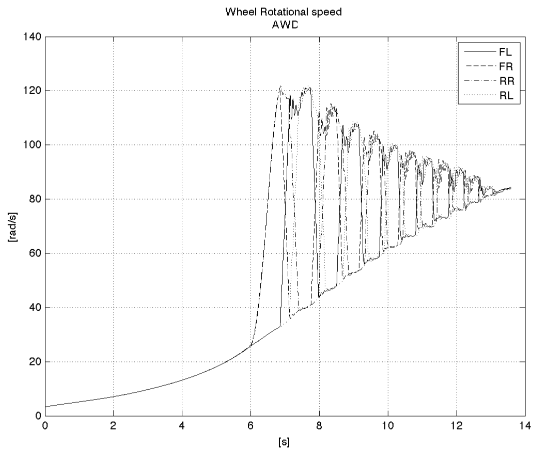


Figure C.2.2.19: Wheel rotational speeds vs time for AWD without DSLD

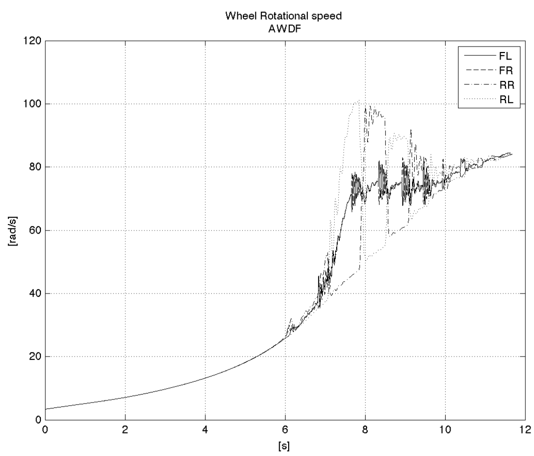


Figure C.2.2.20: Wheel rotational speeds vs time for AWD with DSLD on front axle

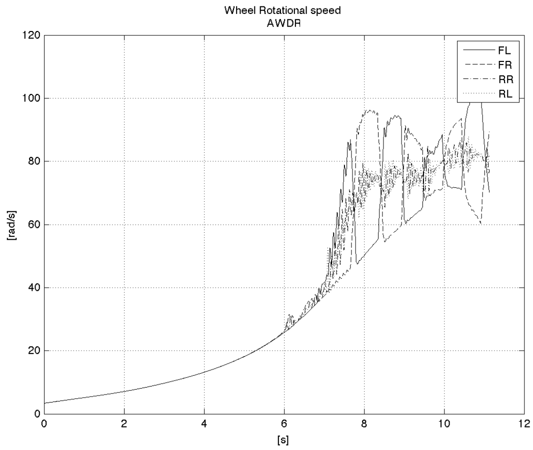


Figure C.2.2.21: Wheel rotational speeds vs time for AWD with DSLD on rear axle

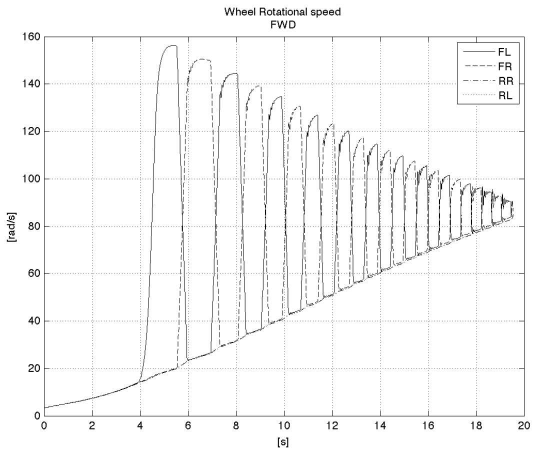


Figure C.2.2.22: Wheel rotational speeds vs time for FWD without DSLD

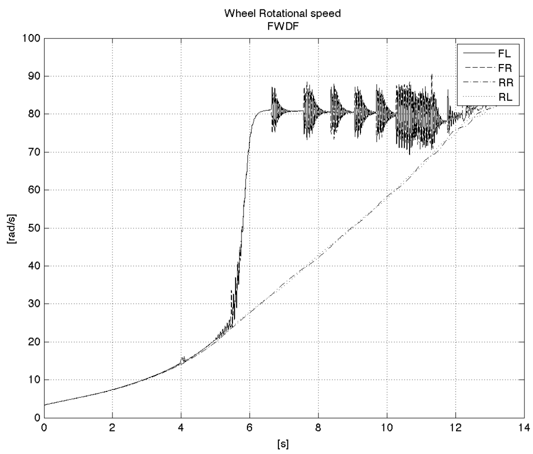


Figure C.2.2.23: Wheel rotational speeds vs time for FWD with DSLD

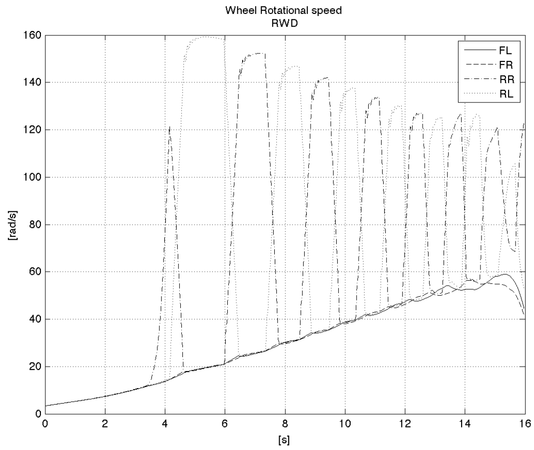


Figure C.2.2.24: Wheel rotational speeds vs time for RWD without DSLD

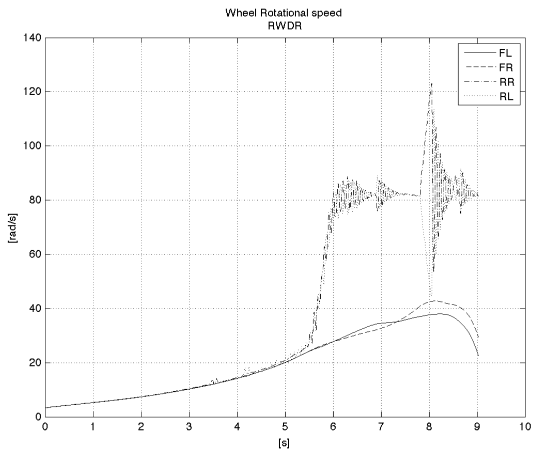


Figure C.2.2.25: Wheel rotational speeds vs time for RWD with DSLD

C.3 90 degree turn

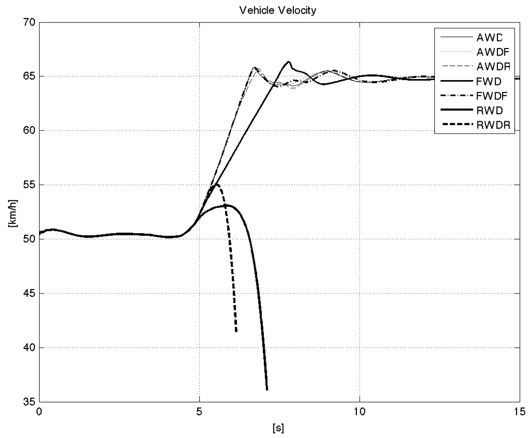


Figure C.3.0.1: Vehicle velocity vs. time

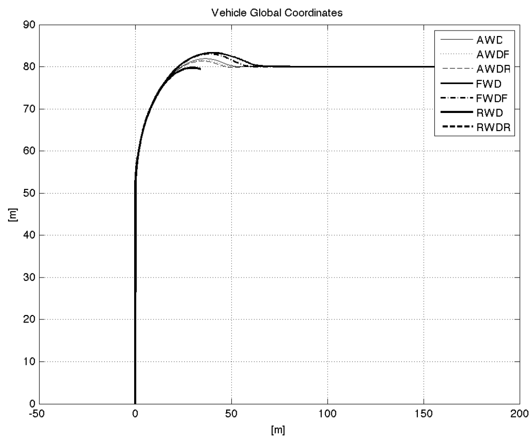


Figure C.3.0.2: Vehicle global coordinates

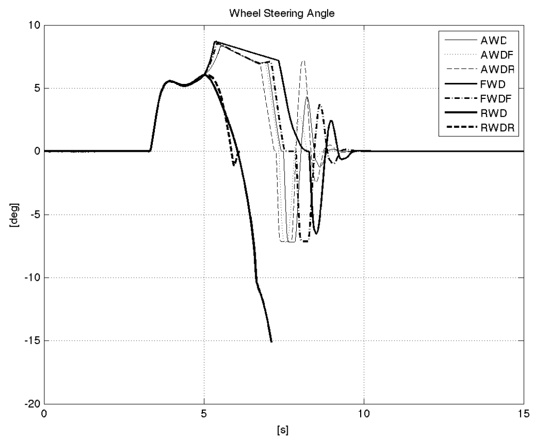


Figure C.3.0.3: Steering angle at the wheel vs. time

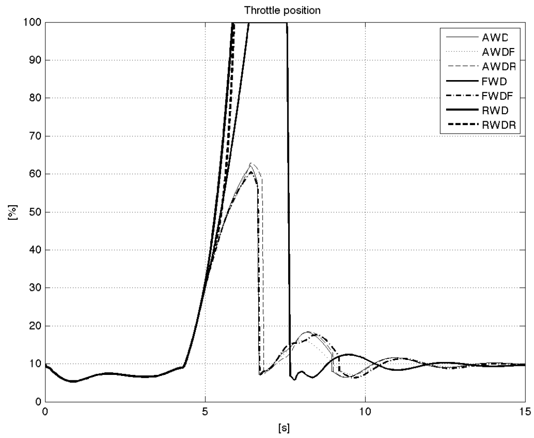


Figure C.3.0.4: Throttle position vs. time

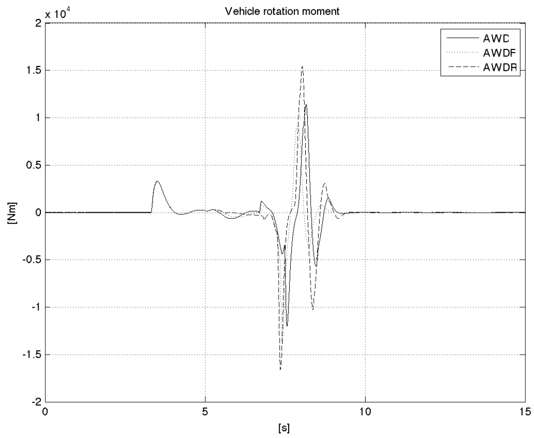


Figure C.3.0.5: Vehicle rotation moment vs. time for AWD vehicles

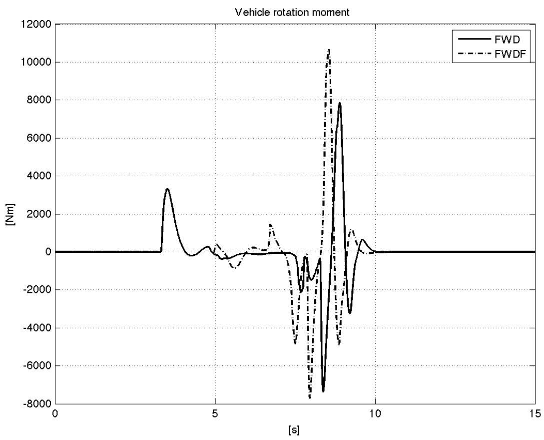


Figure C.3.0.6: Vehicle rotation moment vs. time for FWD vehicles

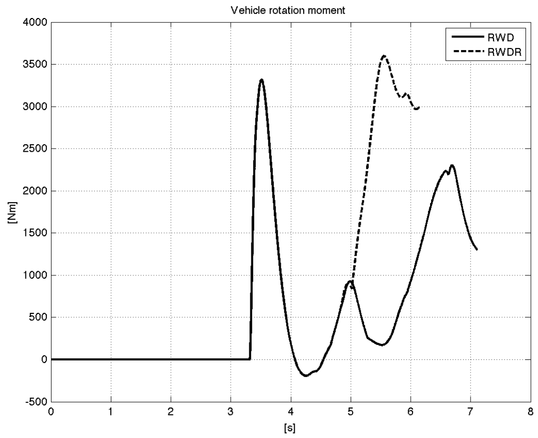


Figure C.3.0.7: Vehicle rotation moment vs. time for RWD vehicles

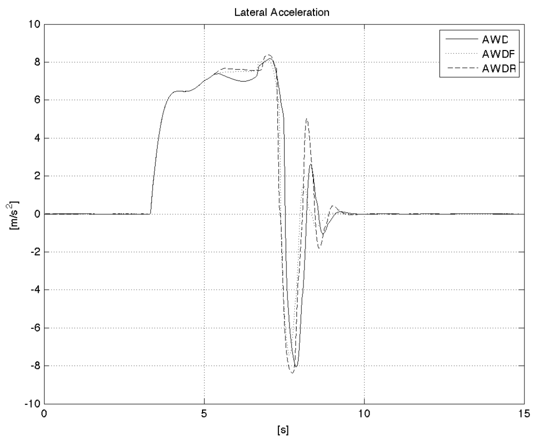


Figure C.3.0.8: Vehicle lateral acceleration vs. time for AWD vehicles

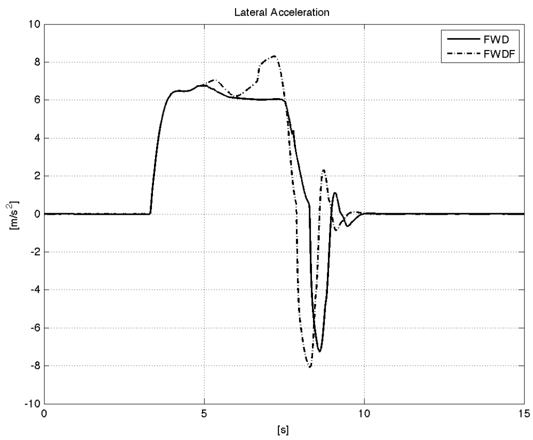


Figure C.3.0.9: Vehicle lateral acceleration vs. time for FWD vehicles

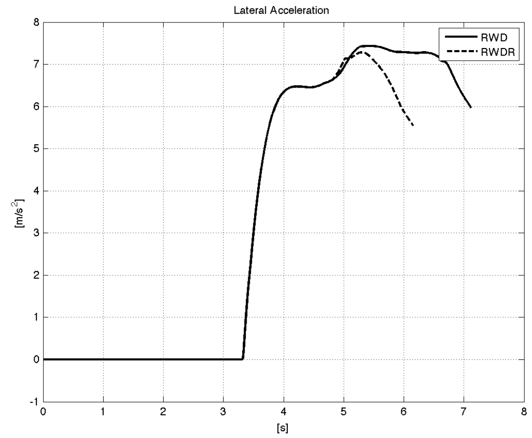


Figure C.3.0.10: Vehicle lateral acceleration vs. time for RWD vehicles

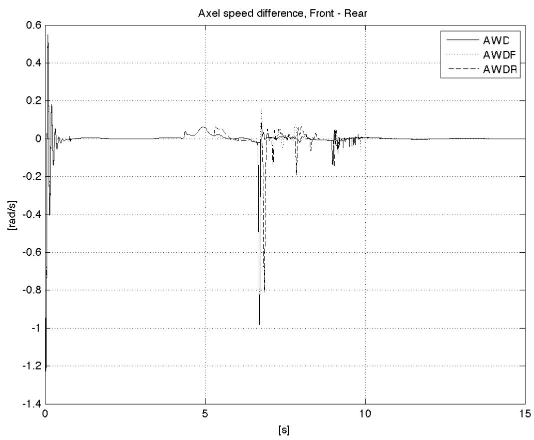


Figure C.3.0.11: Difference in axle rotational speed vs. time for AWD vehicles

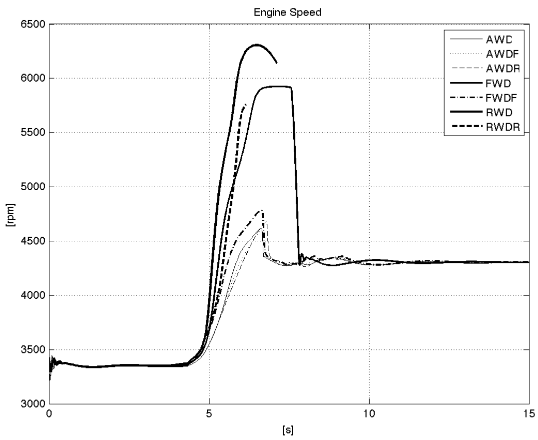


Figure C.3.0.12: Engine rotational speed vs. time

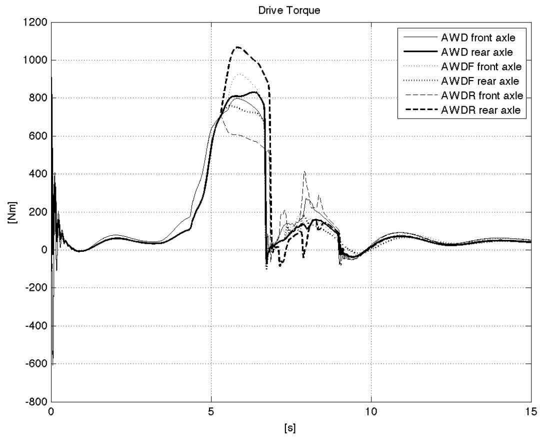


Figure C.3.0.13: Drive torque on front and rear axle vs. time for AWD vehicles

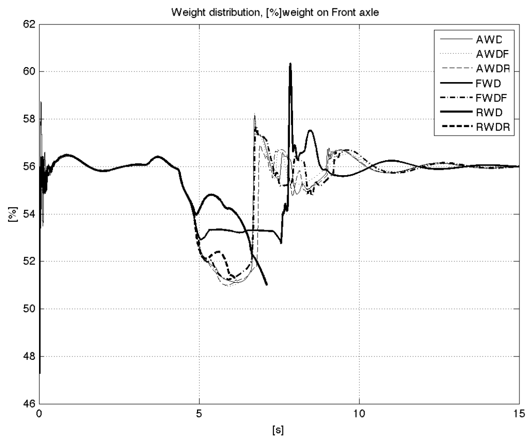


Figure C.3.0.14: Vehicle weight distribution vs. time for AWD vehicles

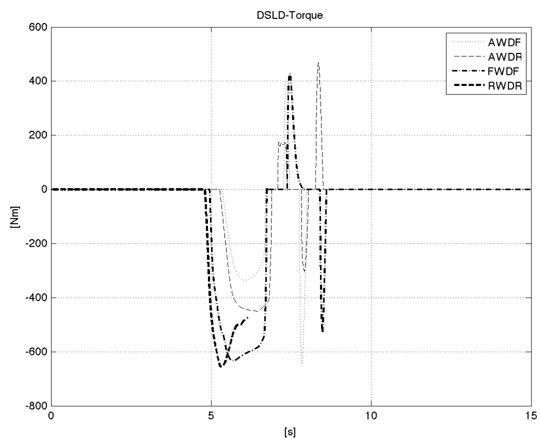


Figure C.3.0.15: DSLD torque vs. time

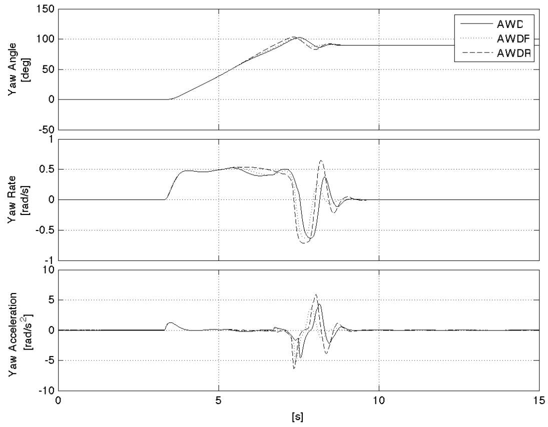


Figure C.3.0.16: Vehicle yaw, yaw rate and yaw acceleration vs. time for AWD vehicles

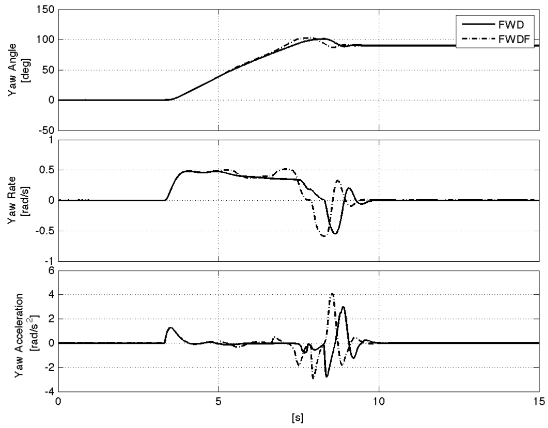


Figure C.3.0.17: Vehicle yaw, yaw rate and yaw acceleration vs. time for FWD vehicles

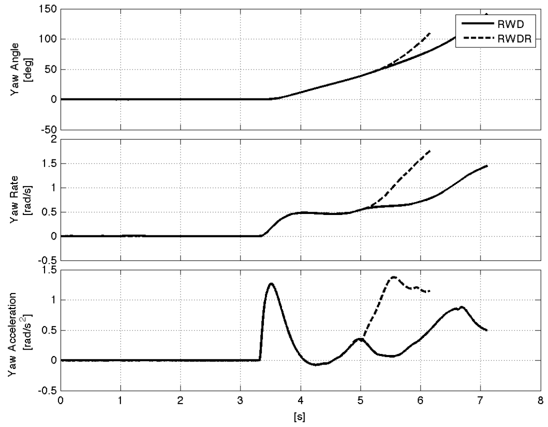


Figure C.3.0.18: Vehicle yaw, yaw rate and yaw acceleration vs. time for RWD vehicles

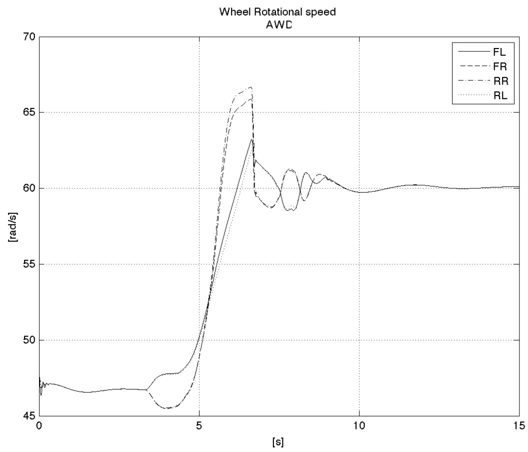


Figure C.3.0.19: Wheel rotational speeds vs time for AWD without DSLD

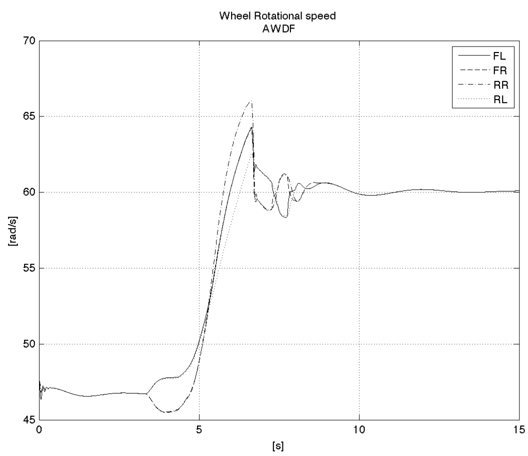


Figure C.3.0.20: Wheel rotational speeds vs time for AWD with DSLD on front axle

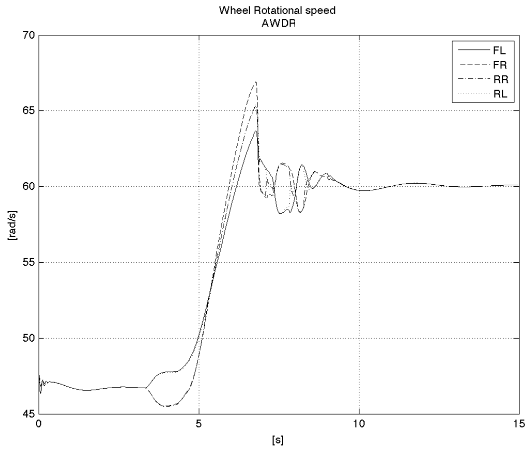


Figure C.3.0.21: Wheel rotational speeds vs time for AWD with DSLD on rear axle

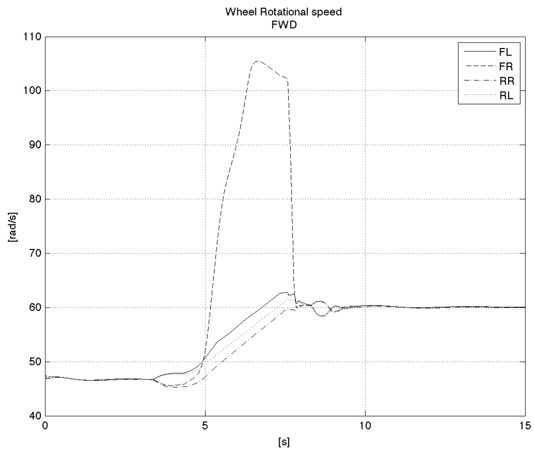


Figure C.3.0.22: Wheel rotational speeds vs time for FWD without DSLD

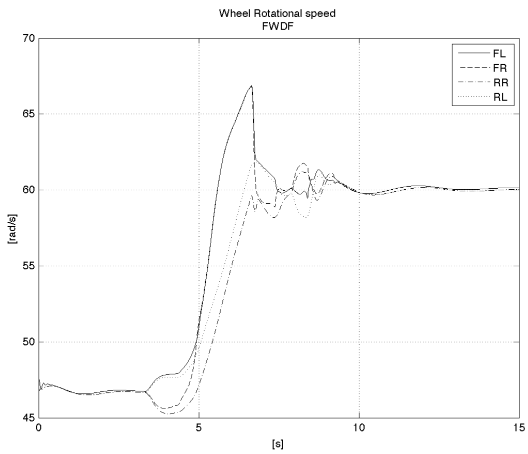


Figure C.3.0.23: Wheel rotational speeds vs time for FWD with DSLD

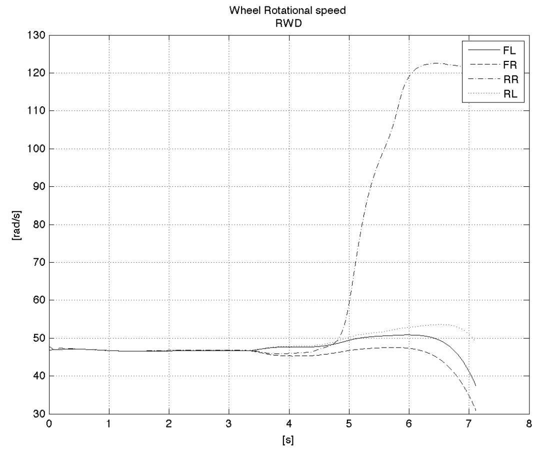


Figure C.3.0.24: Wheel rotational speeds vs time for RWD without DSLD

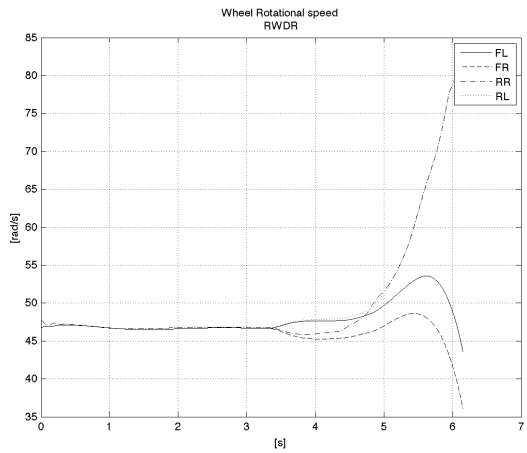


Figure C.3.0.25: Wheel rotational speeds vs time for RWD with DSLD

C.4 Sine with Dwell

The sine with dwell simulation are divided into three different angles to be able to see the difference with and without the DSLD

C.4.1 3.9deg

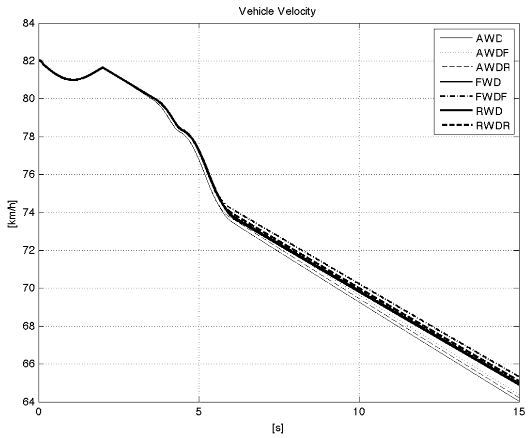


Figure C.4.1.1: Vehicle velocity vs. time

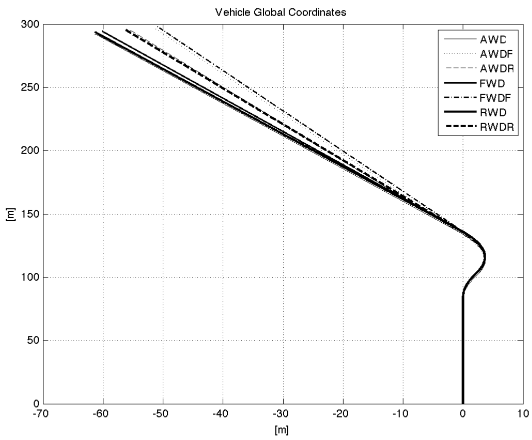


Figure C.4.1.2: Vehicle global coordinates

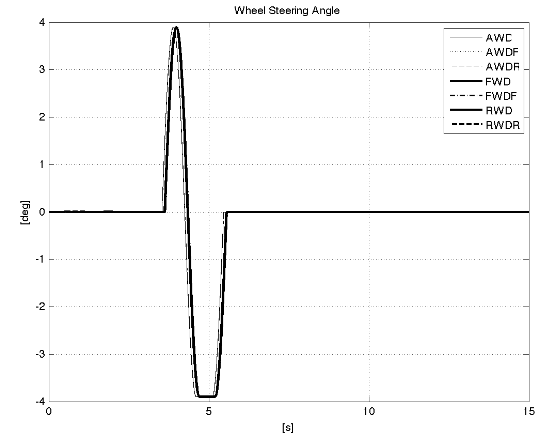


Figure C.4.1.3: Steering angle at the wheel vs. time

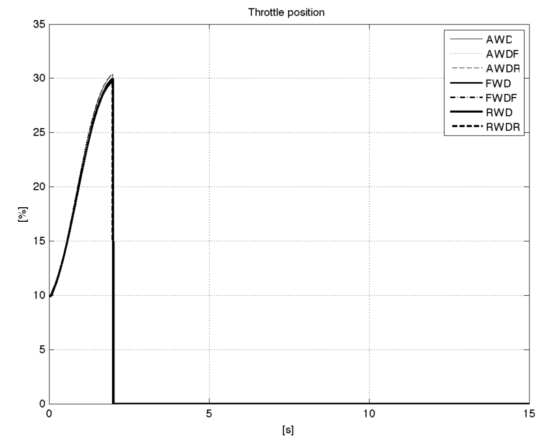


Figure C.4.1.4: Throttle position vs. time

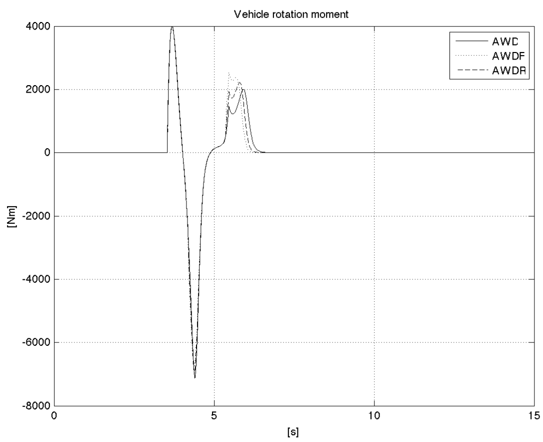


Figure C.4.1.5: Vehicle rotation moment vs. time for AWD vehicles

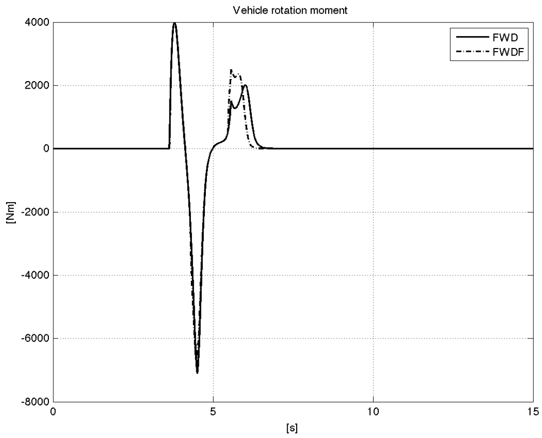


Figure C.4.1.6: Vehicle rotation moment vs. time for FWD vehicles

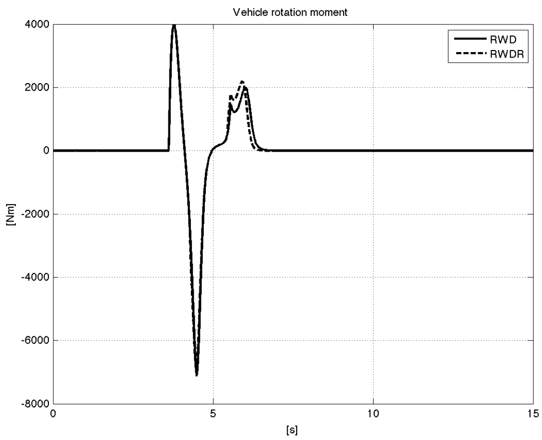


Figure C.4.1.7: Vehicle rotation moment vs. time for RWD vehicles

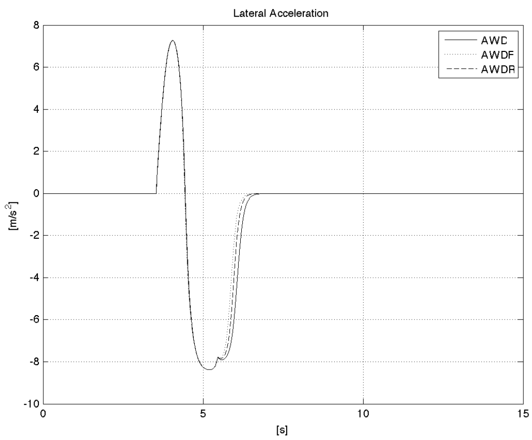


Figure C.4.1.8: Vehicle lateral acceleration vs. time for AWD vehicles

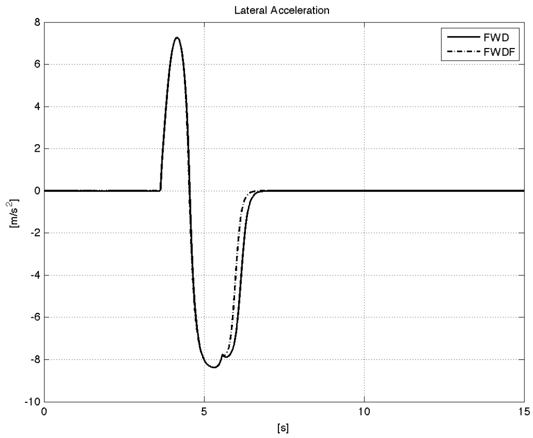


Figure C.4.1.9: Vehicle lateral acceleration vs. time for FWD vehicles

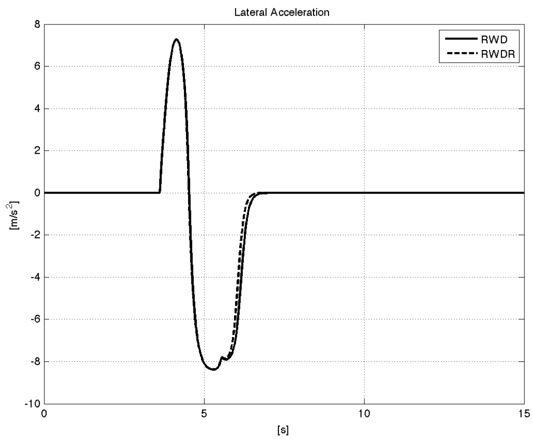


Figure C.4.1.10: Vehicle lateral acceleration vs. time for RWD vehicles

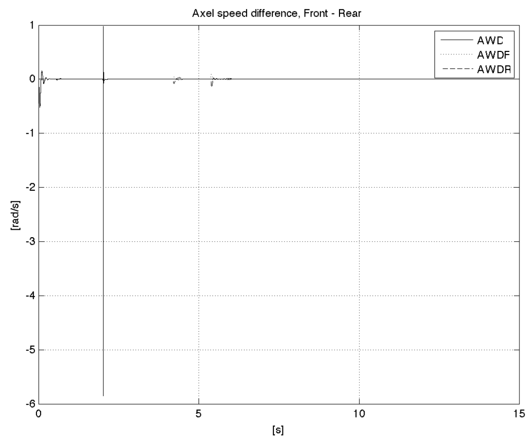


Figure C.4.1.11: Difference in axle rotational speed vs. time for AWD vehicles

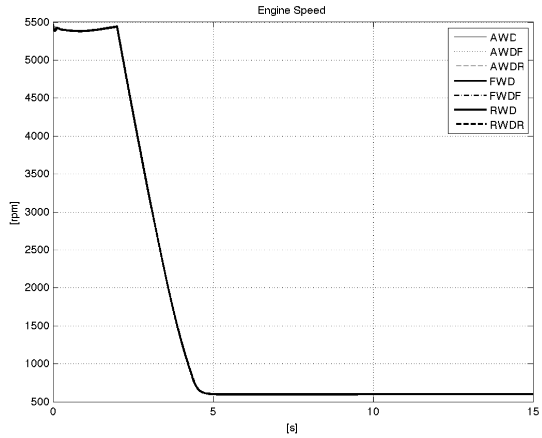


Figure C.4.1.12: Engine rotational speed vs. time

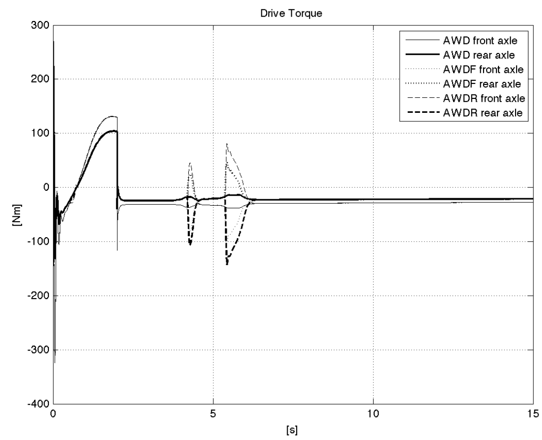


Figure C.4.1.13: Drive torque on front and rear axle vs. time for AWD vehicles

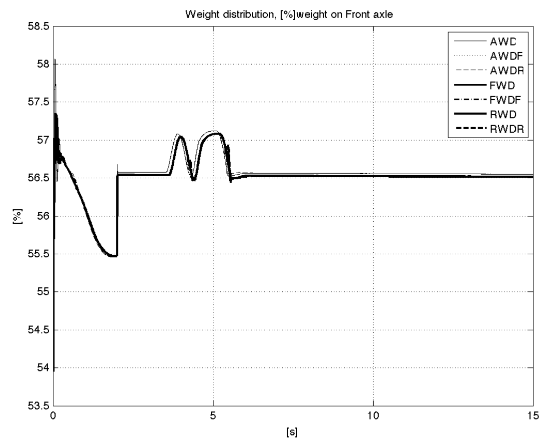


Figure C.4.1.14: Vehicle weight distribution vs. time for AWD vehicles

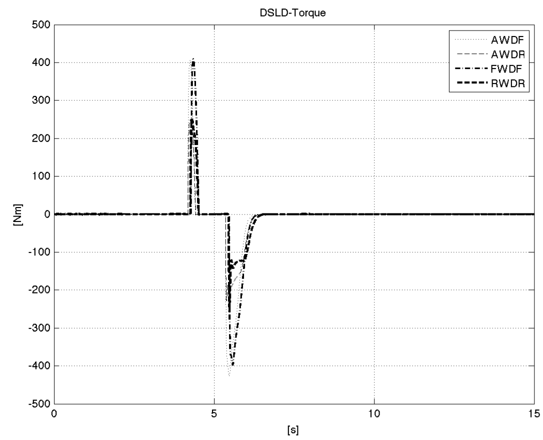


Figure C.4.1.15: DSLD torque vs. time

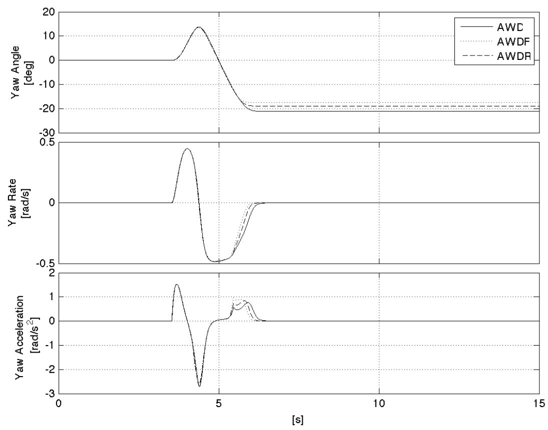


Figure C.4.1.16: Vehicle yaw, yaw rate and yaw acceleration vs. time for AWD vehicles

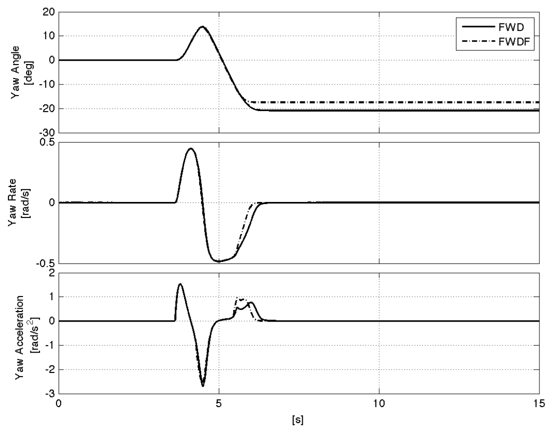


Figure C.4.1.17: Vehicle yaw, yaw rate and yaw acceleration vs. time for FWD vehicles

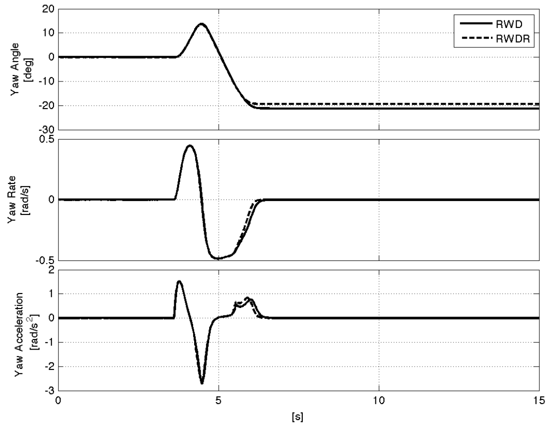


Figure C.4.1.18: Vehicle yaw, yaw rate and yaw acceleration vs. time for RWD vehicles

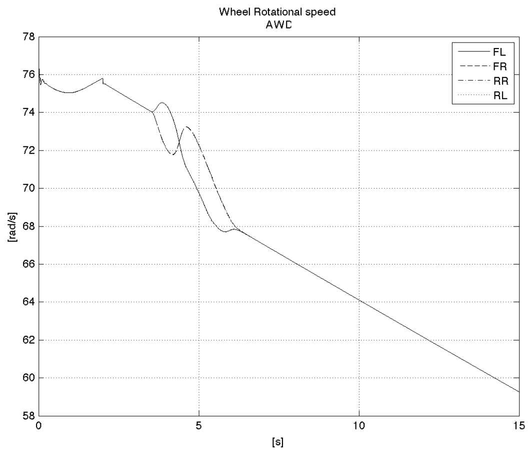


Figure C.4.1.19: Wheel rotational speeds vs time for AWD without DSLD

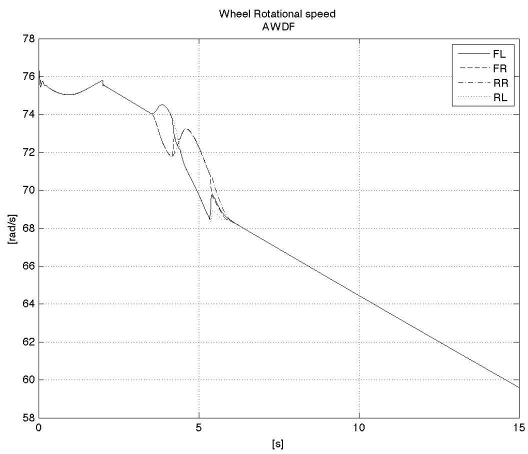


Figure C.4.1.20: Wheel rotational speeds vs time for AWD with DSLD on front axle

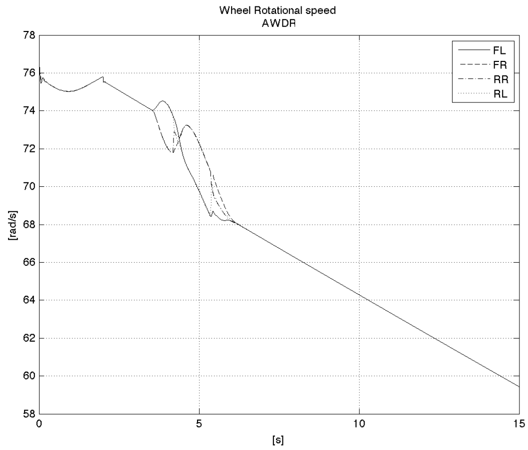


Figure C.4.1.21: Wheel rotational speeds vs time for AWD with DSLD on rear axle

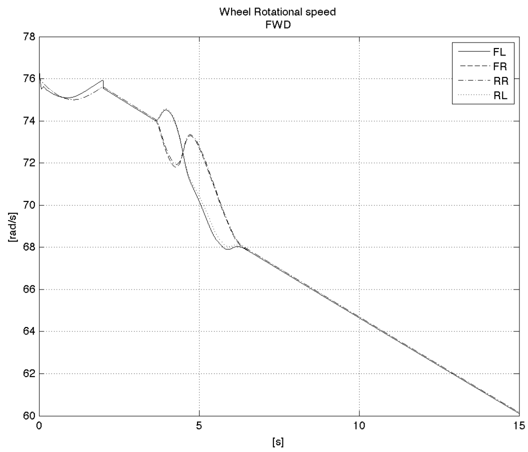


Figure C.4.1.22: Wheel rotational speeds vs time for FWD without DSLD

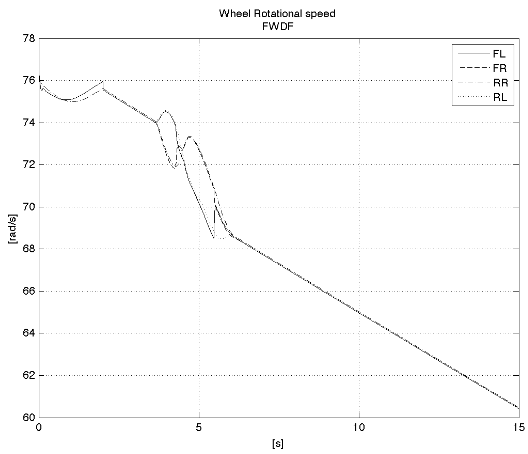


Figure C.4.1.23: Wheel rotational speeds vs time for FWD with DSLD

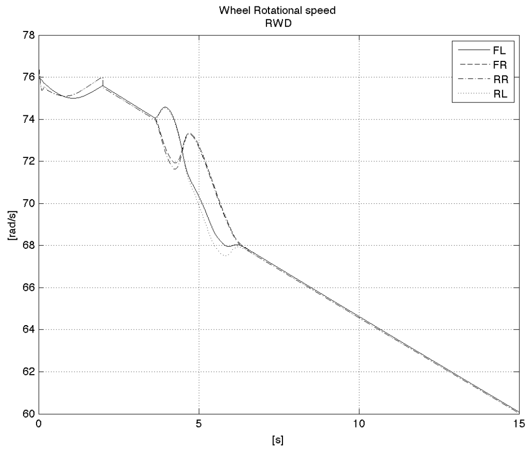


Figure C.4.1.24: Wheel rotational speeds vs time for RWD without DSLD

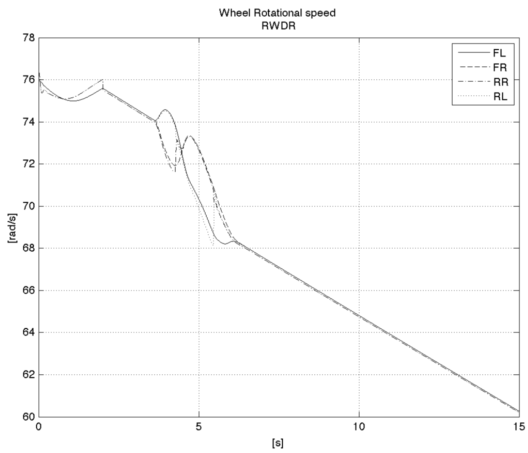


Figure C.4.1.25: Wheel rotational speeds vs time for RWD with DSLD

C.4.2 4.1deg

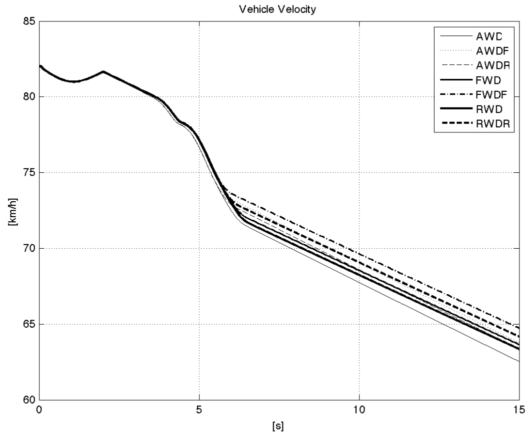


Figure C.4.2.1: Vehicle velocity vs. time

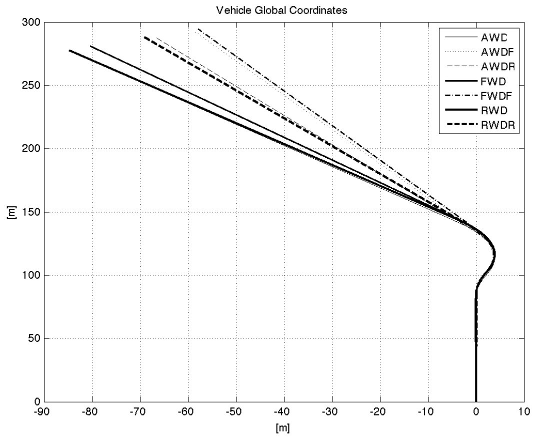


Figure C.4.2.2: Vehicle global coordinates

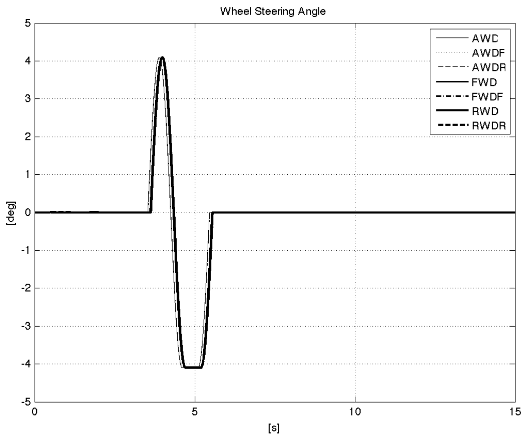


Figure C.4.2.3: Steering angle at the wheel vs. time

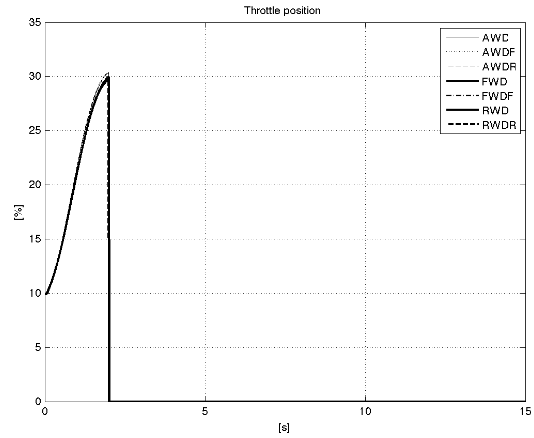


Figure C.4.2.4: Throttle position vs. time

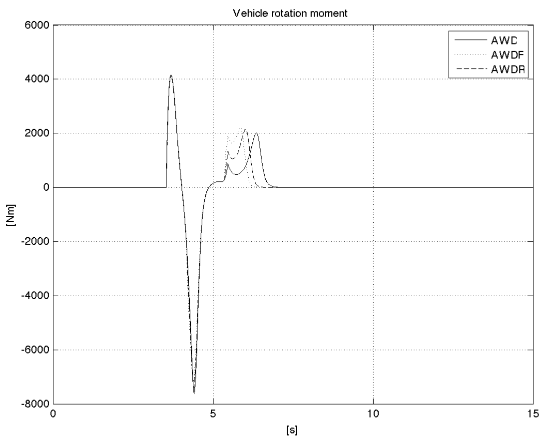


Figure C.4.2.5: Vehicle rotation moment vs. time for AWD vehicles

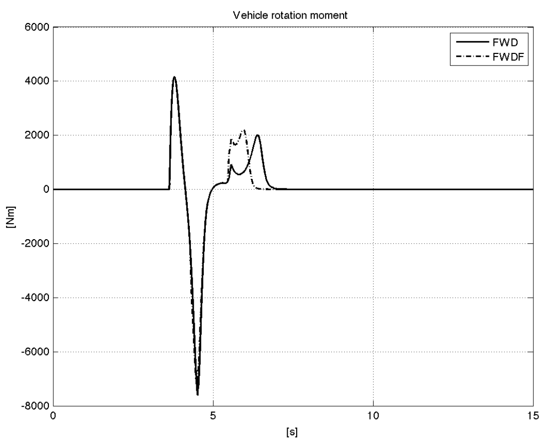


Figure C.4.2.6: Vehicle rotation moment vs. time for FWD vehicles

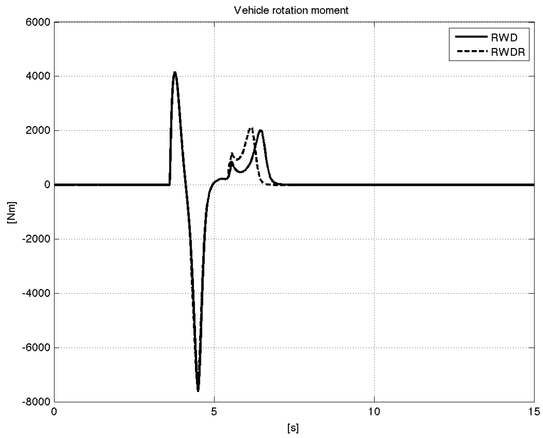


Figure C.4.2.7: Vehicle rotation moment vs. time for RWD vehicles

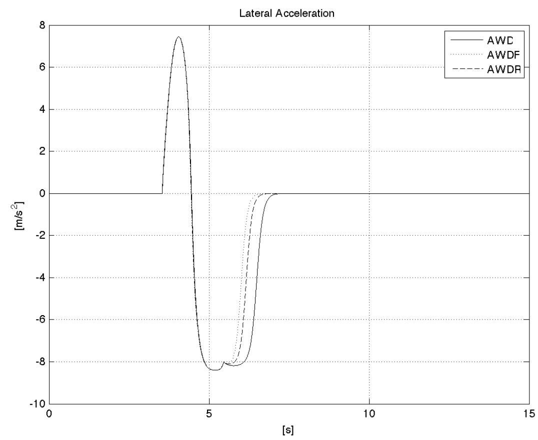


Figure C.4.2.8: Vehicle lateral acceleration vs. time for AWD vehicles

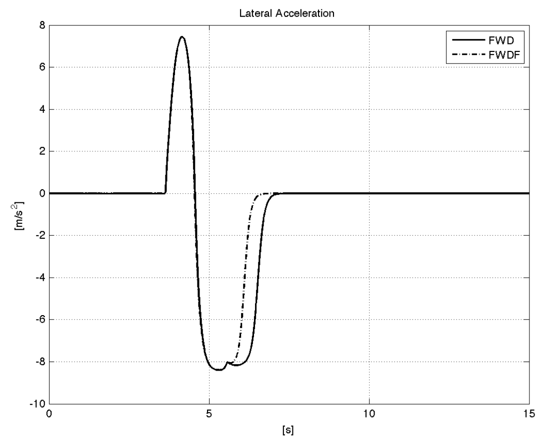


Figure C.4.2.9: Vehicle lateral acceleration vs. time for FWD vehicles

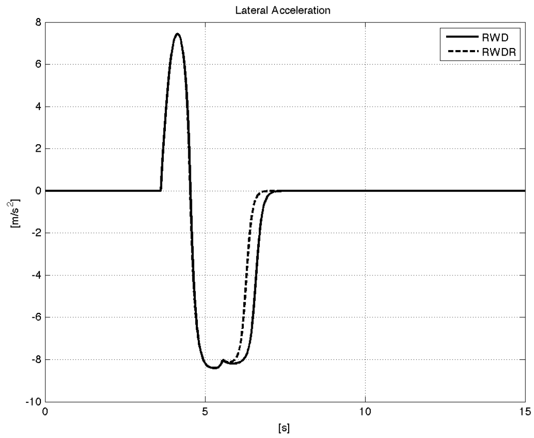


Figure C.4.2.10: Vehicle lateral acceleration vs. time for RWD vehicles

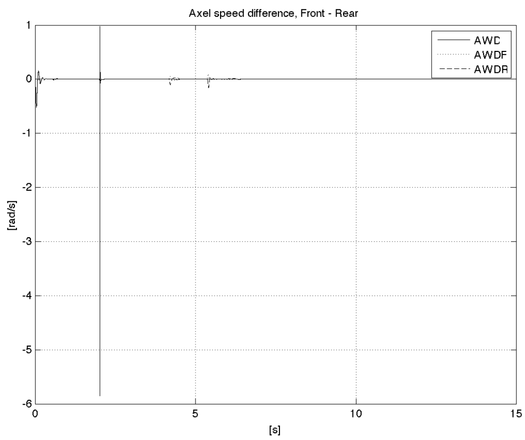


Figure C.4.2.11: Difference in axle rotational speed vs. time for AWD vehicles

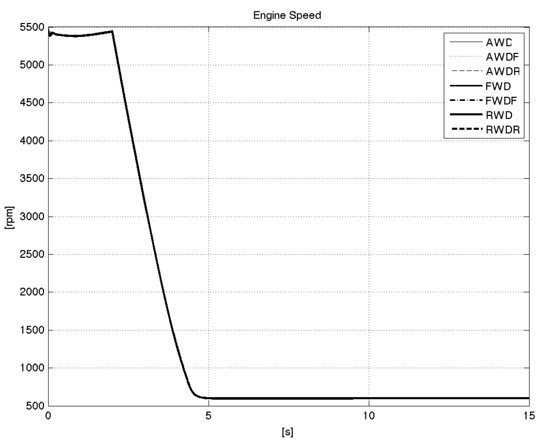


Figure C.4.2.12: Engine rotational speed vs. time

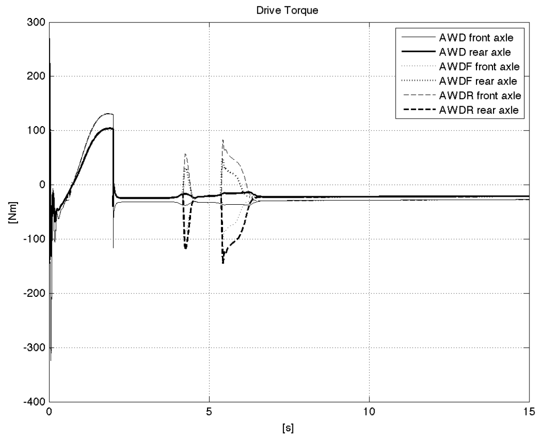


Figure C.4.2.13: Drive torque on front and rear axle vs. time for AWD vehicles

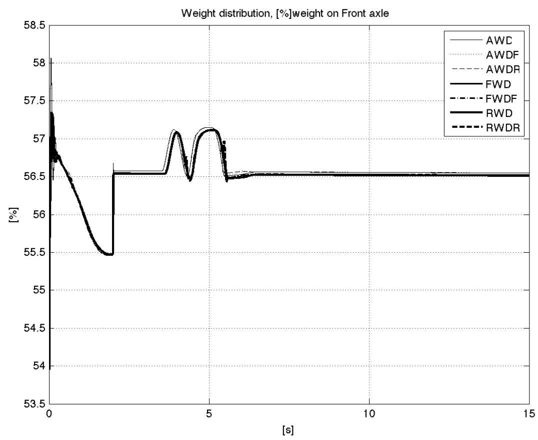


Figure C.4.2.14: Vehicle weight distribution vs. time for AWD vehicles

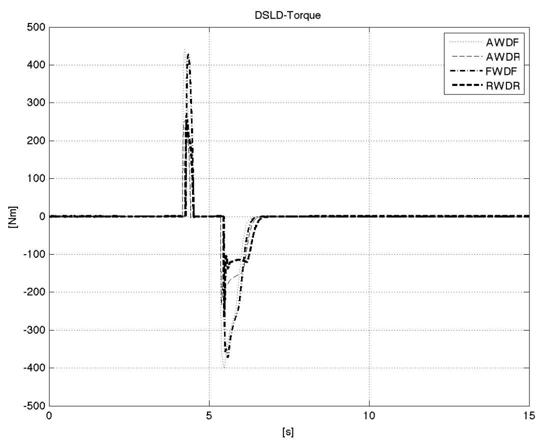


Figure C.4.2.15: DSLD torque vs. time

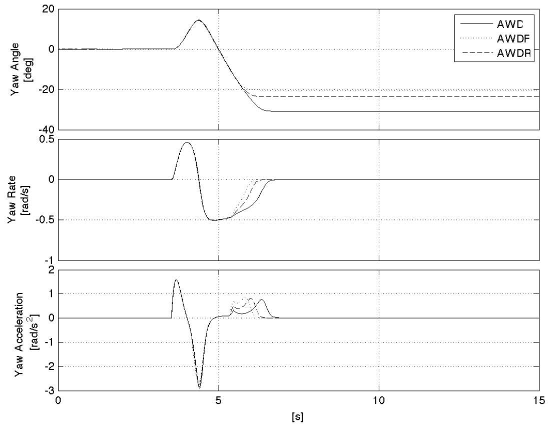


Figure C.4.2.16: Vehicle yaw, yaw rate and yaw acceleration vs. time for AWD vehicles

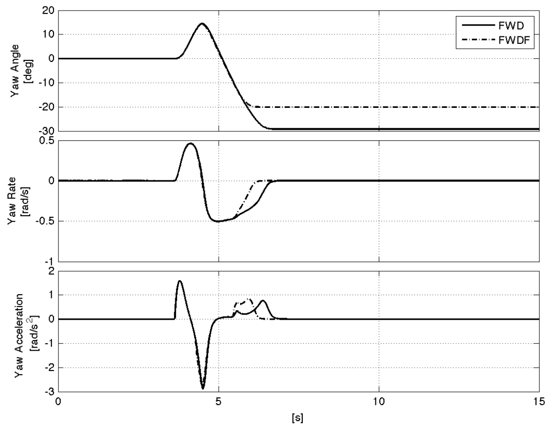


Figure C.4.2.17: Vehicle yaw, yaw rate and yaw acceleration vs. time for FWD vehicles

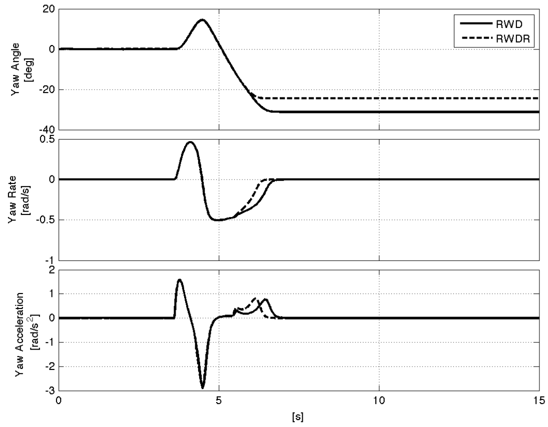


Figure C.4.2.18: Vehicle yaw, yaw rate and yaw acceleration vs. time for RWD vehicles

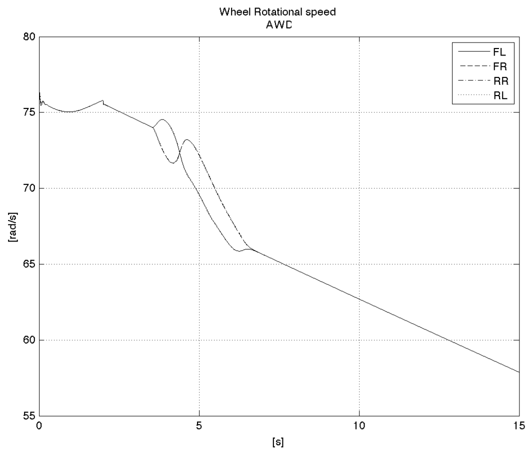


Figure C.4.2.19: Wheel rotational speeds vs time for AWD without DSLD

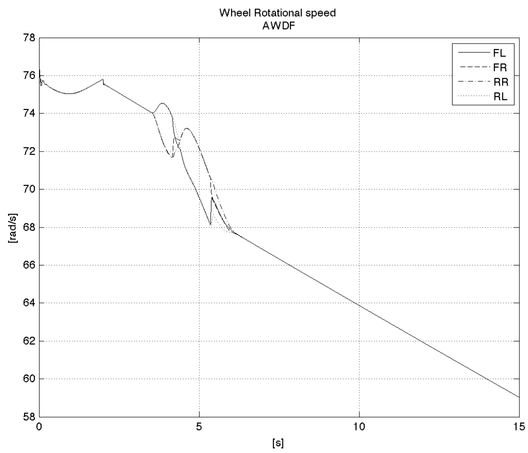


Figure C.4.2.20: Wheel rotational speeds vs time for AWD with DSLD on front axle

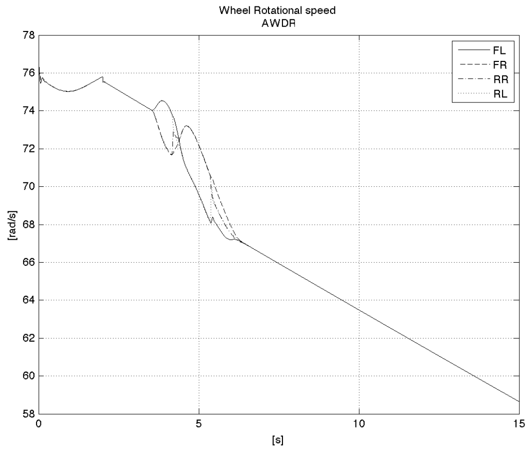


Figure C.4.2.21: Wheel rotational speeds vs time for AWD with DSLD on rear axle

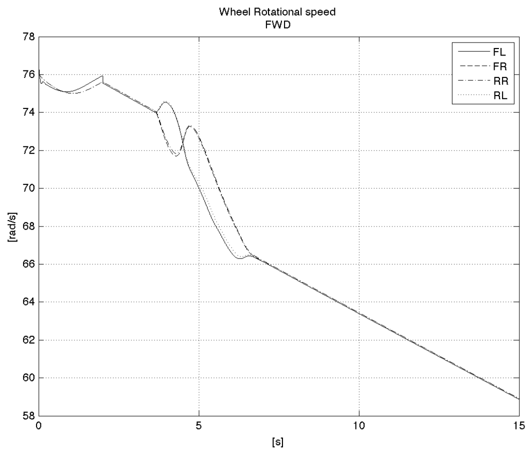


Figure C.4.2.22: Wheel rotational speeds vs time for FWD without DSLD

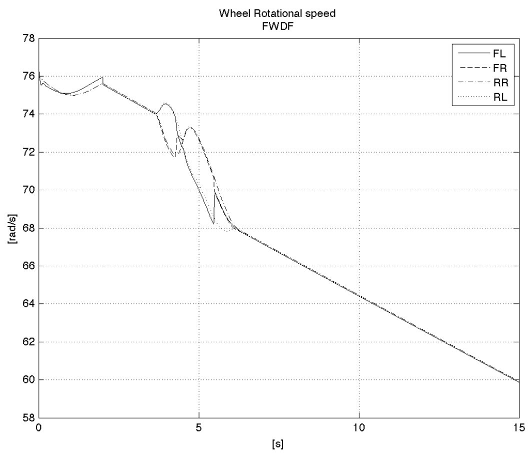


Figure C.4.2.23: Wheel rotational speeds vs time for FWD with DSLD

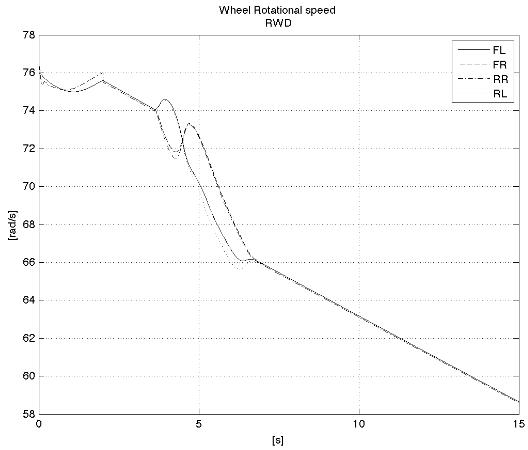


Figure C.4.2.24: Wheel rotational speeds vs time for RWD without DSLD

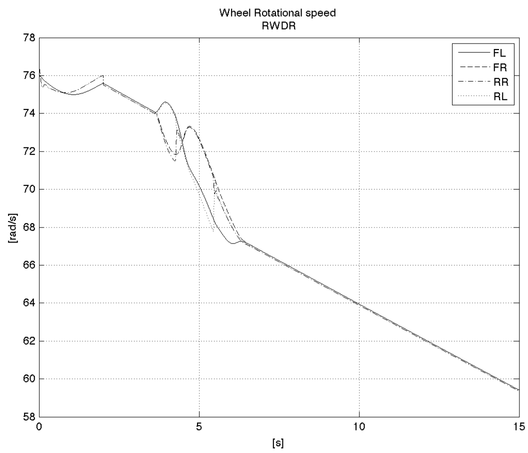


Figure C.4.2.25: Wheel rotational speeds vs time for RWD with DSLD

C.4.3 4.3deg

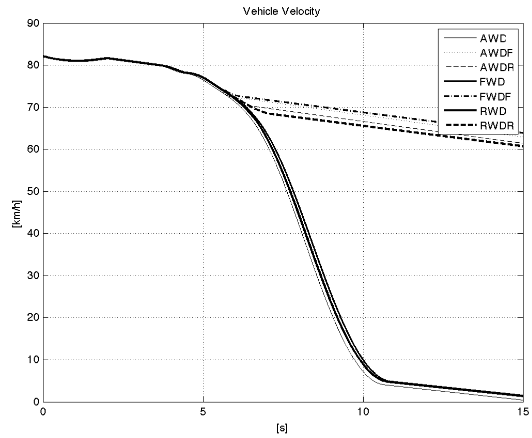


Figure C.4.3.1: Vehicle velocity vs. time

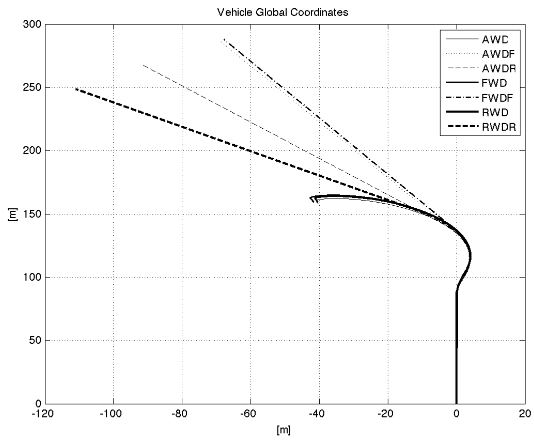


Figure C.4.3.2: Vehicle global coordinates

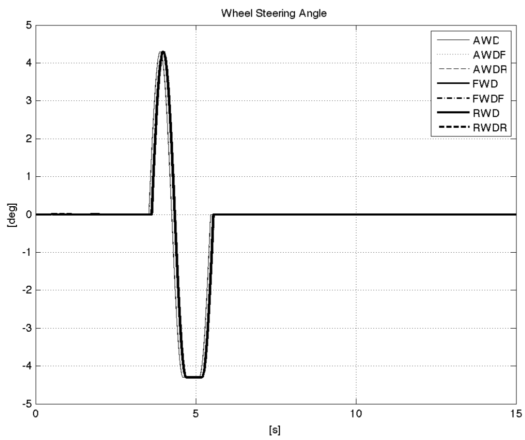


Figure C.4.3.3: Steering angle at the wheel vs. time

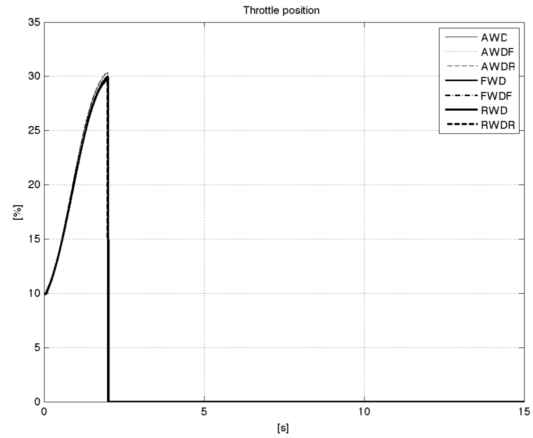


Figure C.4.3.4: Throttle position vs. time

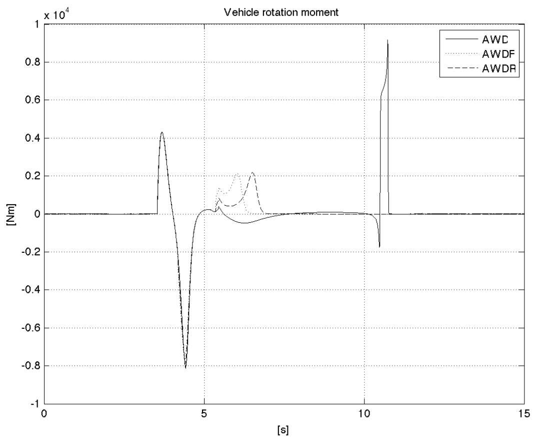


Figure C.4.3.5: Vehicle rotation moment vs. time for AWD vehicles

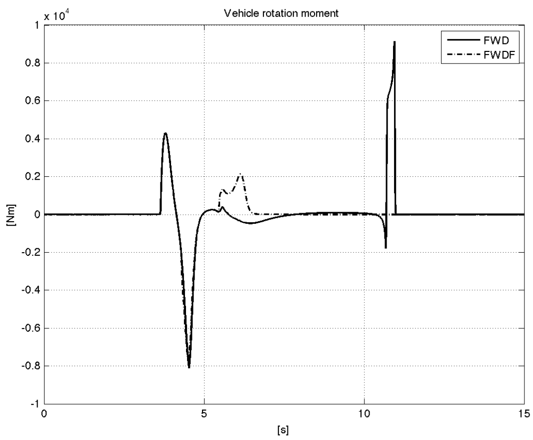


Figure C.4.3.6: Vehicle rotation moment vs. time for FWD vehicles

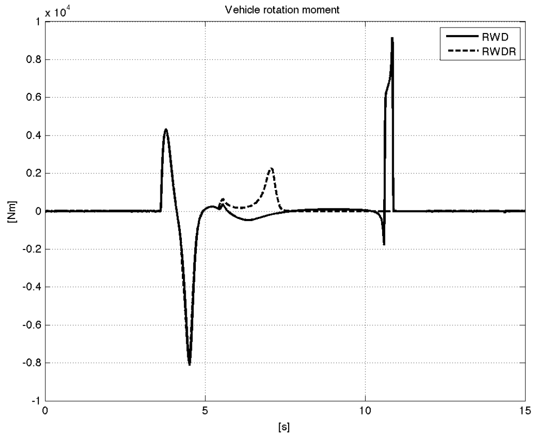


Figure C.4.3.7: Vehicle rotation moment vs. time for RWD vehicles

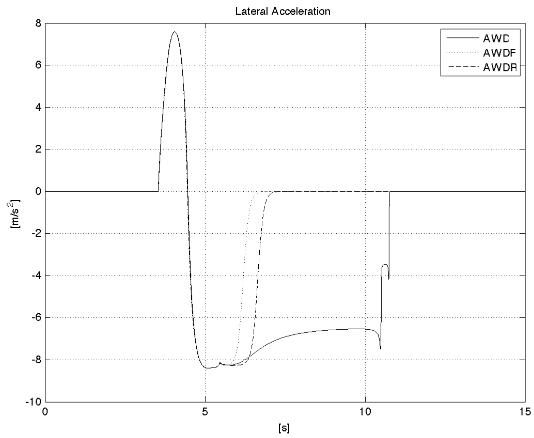


Figure C.4.3.8: Vehicle lateral acceleration vs. time for AWD vehicles

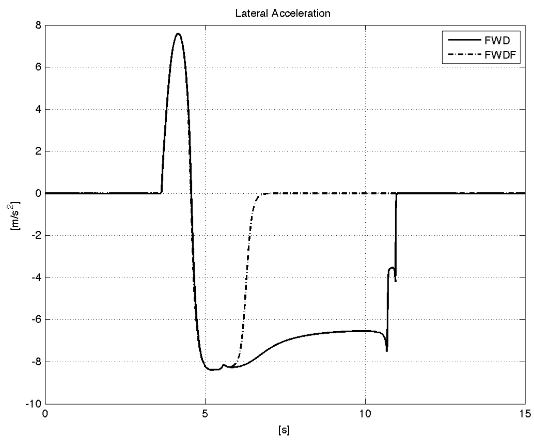


Figure C.4.3.9: Vehicle lateral acceleration vs. time for FWD vehicles

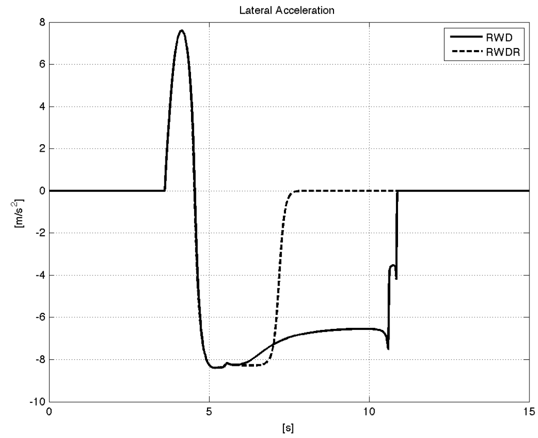


Figure C.4.3.10: Vehicle lateral acceleration vs. time for RWD vehicles

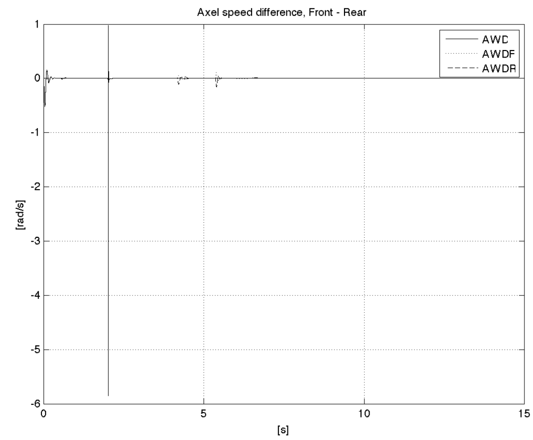


Figure C.4.3.11: Difference in axle rotational speed vs. time for AWD vehicles

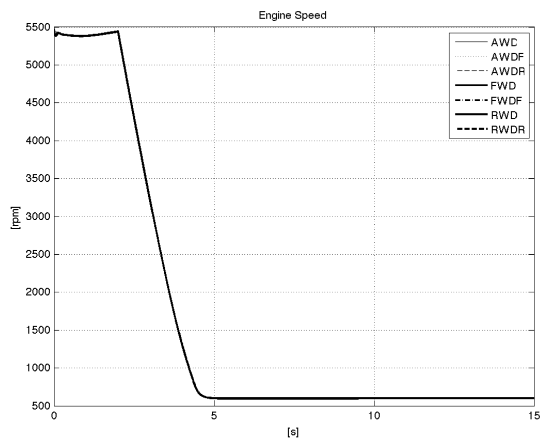


Figure C.4.3.12: Engine rotational speed vs. time

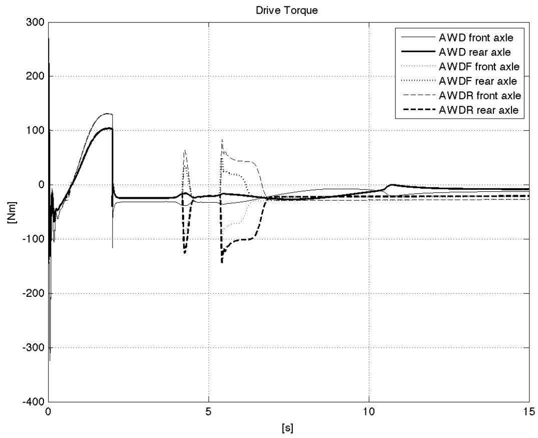


Figure C.4.3.13: Drive torque on front and rear axle vs. time for AWD vehicles

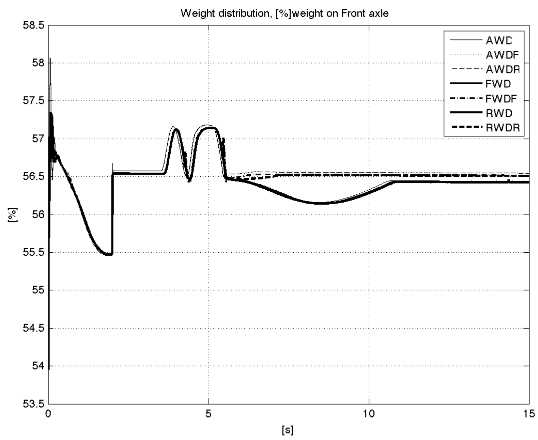


Figure C.4.3.14: Vehicle weight distribution vs. time for AWD vehicles

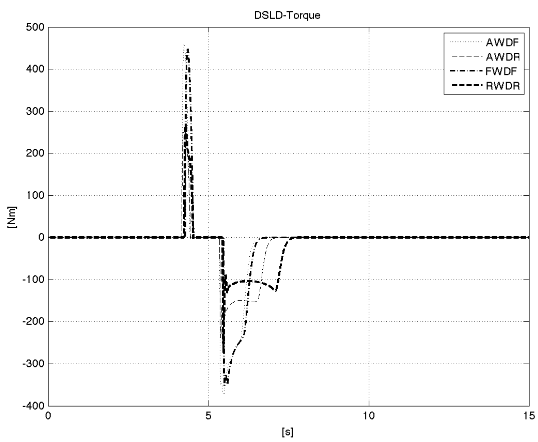


Figure C.4.3.15: DSLD torque vs. time

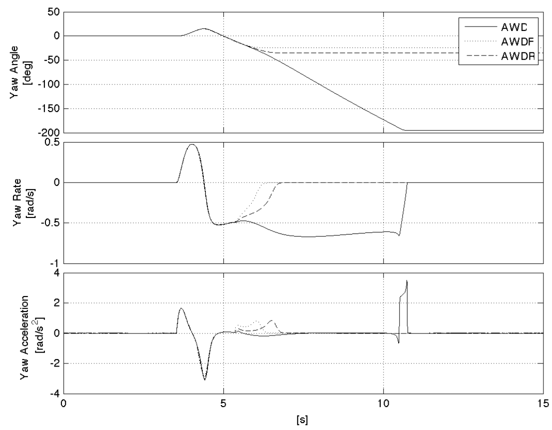


Figure C.4.3.16: Vehicle yaw, yaw rate and yaw acceleration vs. time for AWD vehicles

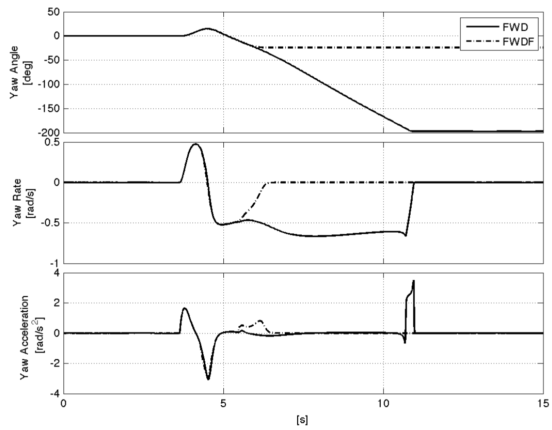


Figure C.4.3.17: Vehicle yaw, yaw rate and yaw acceleration vs. time for FWD vehicles

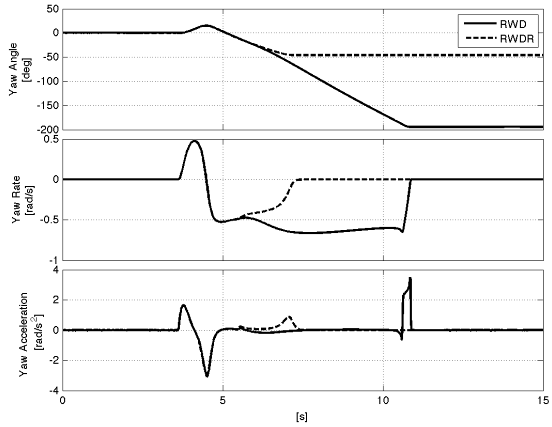


Figure C.4.3.18: Vehicle yaw, yaw rate and yaw acceleration vs. time for RWD vehicles

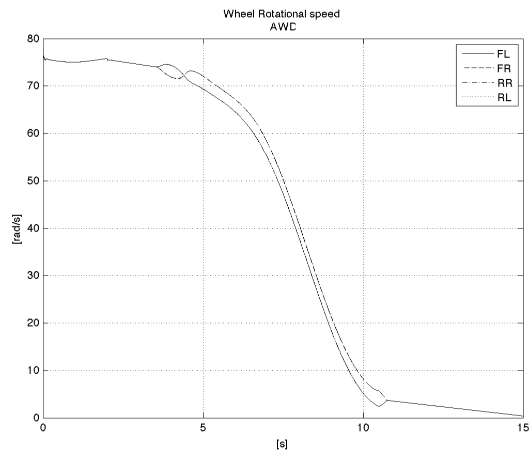


Figure C.4.3.19: Wheel rotational speeds vs time for AWD without DSLD

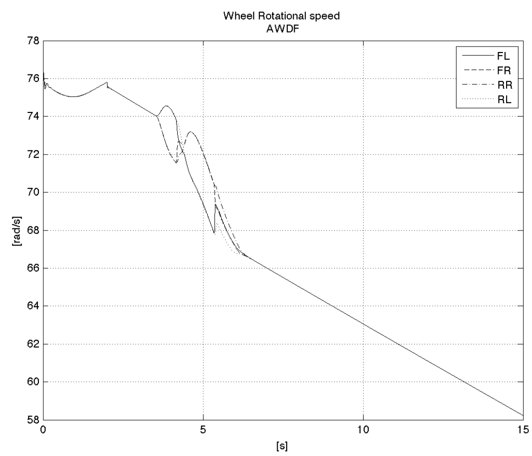


Figure C.4.3.20: Wheel rotational speeds vs time for AWD with DSLD on front axle

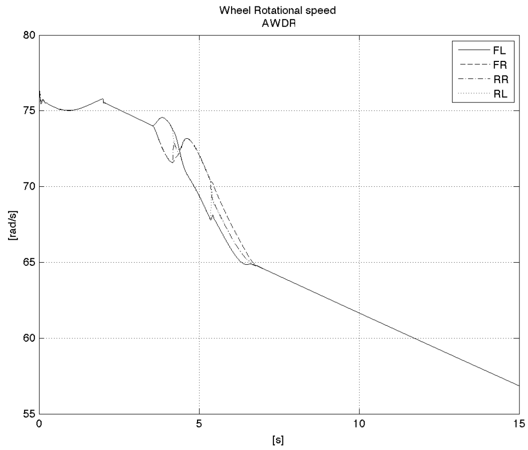


Figure C.4.3.21: Wheel rotational speeds vs time for AWD with DSLD on rear axle

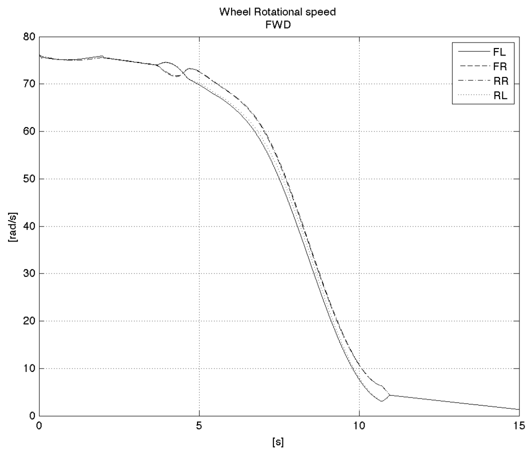


Figure C.4.3.22: Wheel rotational speeds vs time for FWD without DSLD

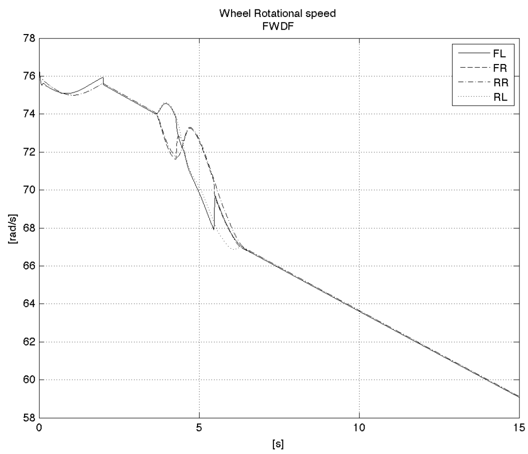


Figure C.4.3.23: Wheel rotational speeds vs time for FWD with DSLD

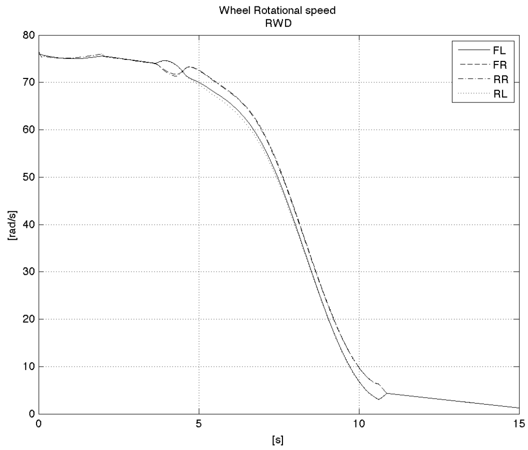


Figure C.4.3.24: Wheel rotational speeds vs time for RWD without DSLD

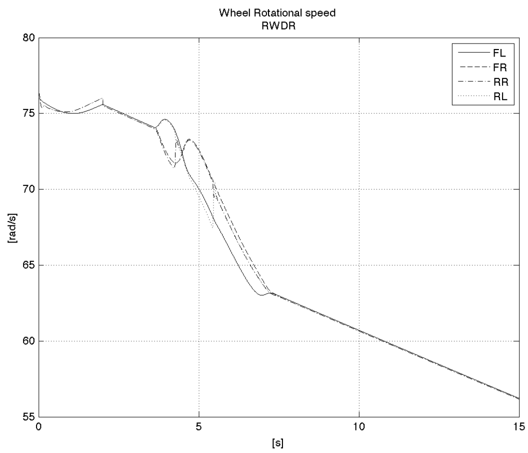


Figure C.4.3.25: Wheel rotational speeds vs time for RWD with DSLD

C.5 Worst case

The Worst case simulation are performed setting the differential to either its normal mode. Which in a turn will give a turning mode. The other way is to set the DSLD into an open mode. This simulation is performed only on the FWD vehicle with DSLD.

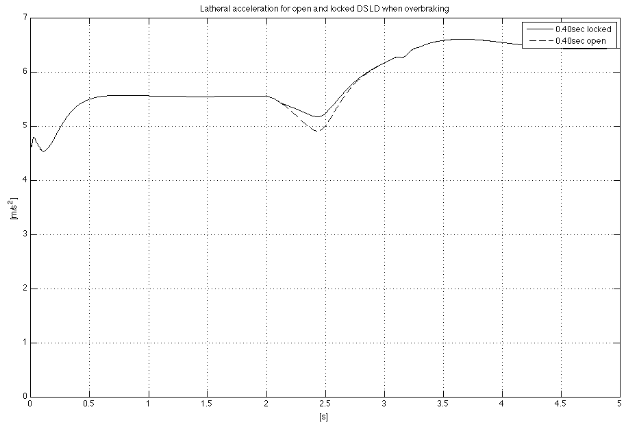


Figure C.5.0.26: Vehicle global coordinates for open and normal mode. At a to long brake input

C.5.1 Turning mode

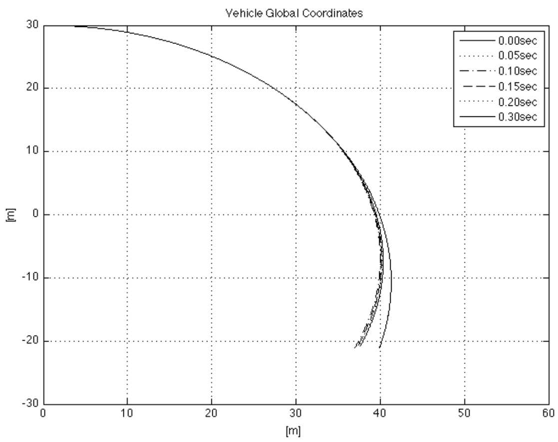


Figure C.5.1.1: Vehicle global coordinates

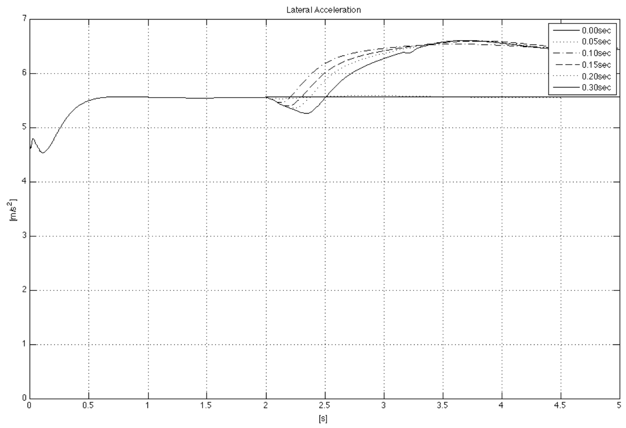


Figure C.5.1.2: Vehicle lateral acceleration vs. time

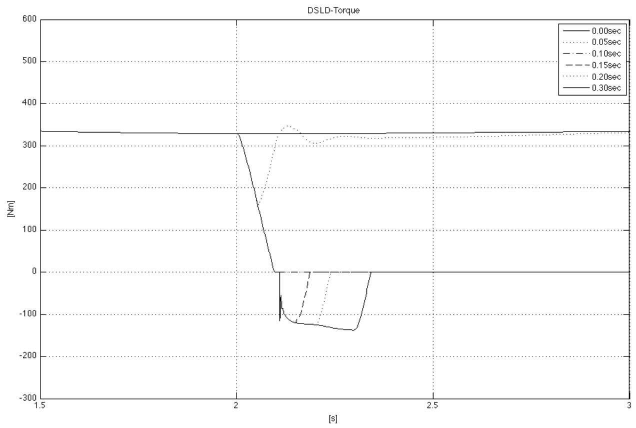


Figure C.5.1.3: DSLD torque vs. time

C.5.2 Open mode

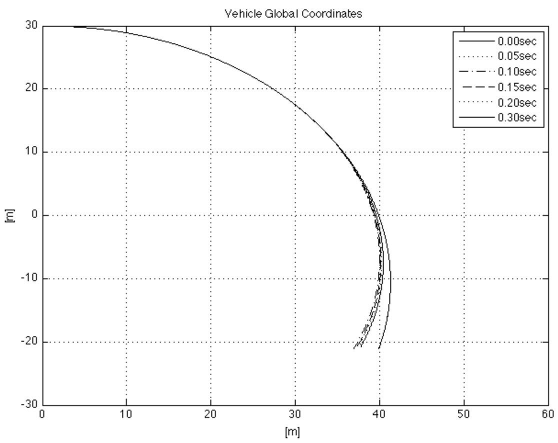


Figure C.5.2.1: Vehicle global coordinates

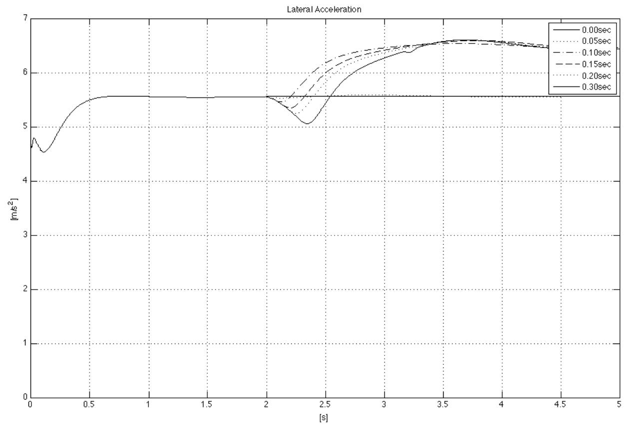


Figure C.5.2.2: Vehicle lateral acceleration vs. time

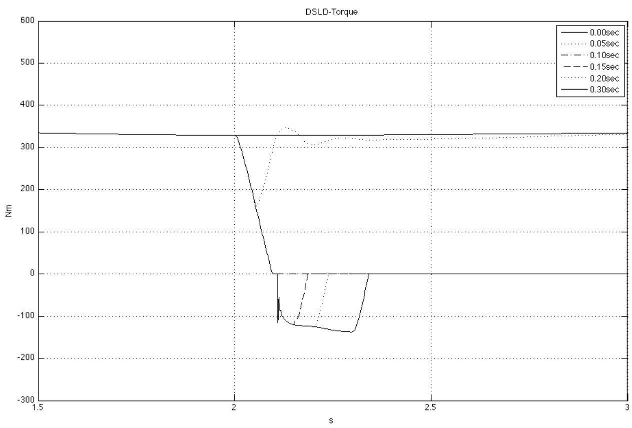


Figure C.5.2.3: DSLD torque vs. time