Development of an active braking controller for brake systems on electric motor driven vehicles

BENNY TRUONG
Utveckling av en aktiv broms-regulator för bromssystem på elmotor-drivna fordon

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Braking a vehicle can be difficult and the largest braking force is not always the most efficient braking. A lot of problems occur in case the wheels start to lock up and slide on the road surface. This is more likely to happen on slippery roads but can happen on high traction roads as well. When the wheels lose the traction, lock up and start to slide on the road surface the braking forces between the tyres and the ground are reduced. This results in a longer brake distance, loss of steering ability and less stability since the tyres lose the major lateral forces.

A new controlled brake system is developed by first creating a system structure with several subsystems which each solves their tasks to achieve different objectives. The subsystems developed are the active braking control system, ABC activation system, brake blending system and dynamical brake torque distribution system. The objective of the controlled brake system is to reduce the brake distance, achieve regenerated energy and keep the vehicle’s steering ability.

The master thesis is proposing a controlled brake system for a heavy construction equipment vehicle. The work is done in cooperation with Volvo Construction Equipment and the developed system is implemented and tested in a simulation-model for one of company’s prototype wheel loader. The vehicle used in the thesis is a four-wheel-driven wheel loader with electric motors in each wheel hub and it has the ability for independent torque actuation for all individual wheels. The electric motors have the potential to be used as regenerative brakes where they produce a braking torque and power which can be used to charge a battery. The braking torque from an electric motor is however limited and not always sufficient, that is why it needs to be supplemented by friction brakes. The friction brakes are available at each wheel and are used when the requested braking torque exceeds the torque provided by the electric motor.

The brake blending strategy distributes the braking torque between electric motors and friction brakes to achieve regenerative braking. To reduce the brake distance the wheels are prevented from being locked up and slide. The active braking control system controls the wheel slip at each wheel to maintain a high friction between the tyre and the ground and thereby keeping the brake force of the vehicle as high as possible. The vehicle can maintain the steering ability even during an emergency braking by preventing the wheels from locking up and thereby keeping the major lateral forces. The wheel slip controller is a PID controller customized by velocity scaling and output tracking.

The result shows improvements with the new controlled brake system compared to the conventional brake system used at Volvo CE. The improvements are achieved in aspects of reduced brake distance, energy regeneration and maintained vehicle steering ability. The controlled brake system is a novel concept which can be implemented in a prototype vehicle for development and research purpose. However the system needs further development and more extensive testing before it can be implemented in a vehicle for the end user since the brake system is a crucial part of the vehicle and is strongly connected to the safety.
Abstrakt


Ett nytt reglerat bromssystem utvecklas genom att först skapa en struktur för systemet med flera delsystem som var och en löser sina uppgifter för att uppnå olika mål. De delsystem som utvecklas är active braking control systemet, ABC activation systemet, brake blending systemet och dynamical brake torque distribution systemet. Målet för det reglerade bromssystemet är att minska bromssträckan, kunna regenerera energi och behålla fordonets styrbart.


Resultaten visar förbättringar med det nya reglerade bromssystemet jämfört med det konventionella bromssystemet som används på Volvo CE. Förbättringarna består av minskad bromssträcka, regenerering av energi och bibehållna styrbartigheten på fordonet. Det reglerade bromssystemet är ett nytt koncept som kan implementeras i ett prototypfordon för utvecklingssyftet och forskningssyftet. Systemet behöver vidareutvecklas och kräver mer omfattande tester innan den kan implementeras i ett fordon för slutanvändaren eftersom bromssystemet är en viktig del av fordonet och är starkt relaterat till säkerheten.
# Nomenclature

## Notations

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<th>Description</th>
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<tr>
<td>$r$</td>
<td>Wheel’s radius (m)</td>
</tr>
<tr>
<td>$L$</td>
<td>Wheelbase, longitudinal distance from front wheel to rear wheel (m)</td>
</tr>
<tr>
<td>$L_f$</td>
<td>Longitudinal distance from front wheel to the center of mass</td>
</tr>
<tr>
<td>$L_r$</td>
<td>Longitudinal distance from rear wheel to the center of mass</td>
</tr>
<tr>
<td>$h_g$</td>
<td>Height of the center of mass from ground (m)</td>
</tr>
<tr>
<td>$m$</td>
<td>Vehicle mass (kg)</td>
</tr>
<tr>
<td>$J$</td>
<td>Wheel’s moment of inertia (kgm$^2$)</td>
</tr>
<tr>
<td>$v$</td>
<td>Longitudinal speed of the vehicle (m/s)</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Wheel’s angular velocity (rad/s)</td>
</tr>
<tr>
<td>$T_b$</td>
<td>Braking torque (Nm)</td>
</tr>
<tr>
<td>$F_zf$</td>
<td>Vertical force on the front wheel (N)</td>
</tr>
<tr>
<td>$F_zr$</td>
<td>Vertical force on the rear wheel (N)</td>
</tr>
<tr>
<td>$F_xf$</td>
<td>Longitudinal force at contact point between front tyre and road (N)</td>
</tr>
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<td>$N$</td>
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<tr>
<td>$g$</td>
<td>Gravitational acceleration (m/s$^2$)</td>
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<tr>
<td>$\lambda$</td>
<td>Longitudinal wheel slip</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Longitudinal friction coefficient</td>
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<tr>
<td>$a$</td>
<td>Vehicle’s acceleration</td>
</tr>
<tr>
<td>$\beta_f$</td>
<td>Braking force ratio for front wheel</td>
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<tr>
<td>$\beta_r$</td>
<td>Braking force ratio for rear wheel</td>
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## Abbreviations

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<td>ABS</td>
<td>Anti-lock Braking System</td>
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<tr>
<td>BBW</td>
<td>Brake-By-Wire</td>
</tr>
<tr>
<td>IMU</td>
<td>Inertial Measurement Unit</td>
</tr>
<tr>
<td>ECU</td>
<td>Electronic Control Unit</td>
</tr>
<tr>
<td>HAB</td>
<td>Hydraulic Actuated Brakes</td>
</tr>
<tr>
<td>EHB</td>
<td>Electro-Hydraulic Brakes</td>
</tr>
<tr>
<td>EMB</td>
<td>Electro-Mechanical Brakes</td>
</tr>
<tr>
<td>SOC</td>
<td>State Of Charge</td>
</tr>
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<td>BB</td>
<td>Brake Blending</td>
</tr>
<tr>
<td>ABC</td>
<td>Active Braking Control</td>
</tr>
<tr>
<td>DBTD</td>
<td>Dynamical Brake Torque Distribution</td>
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1 Introduction

1.1 Background

The brake systems on heavy construction vehicles at Volvo CE have not reached the same development as on passenger cars. This is probably since heavy construction vehicles do not reach as high speeds as cars, but instead the construction vehicles are multiple times heavier than a car and therefore the brake system needs more consideration. It is of great importance to ensure the safety of heavy construction vehicle and protect the driver, construction workers and also civilians since these vehicles are often driven at maximum speed when being transported from one working site to another. These vehicles operate under severe conditions where the road can be inclined or slippery by water and mud. When the vehicle makes a hard braking the wheels can lock up and start to slide and this will cause a longer brake distance and loss of control since the traction is lost. This can occur on the best road conditions but the consequences are even worse on bad road conditions. Losing the control of these machines is dangerous considering the massive weight and therefore it is crucial to have top performing brake systems.

The current construction vehicles manufactured by Volvo CE are only using friction brakes which are not controlled. The structure is a hydraulic linkage system where the pressure from the brake pedal propagates to the brakes. The thesis work involves development of a wheel loader which utilizes electric motors in each wheel to drive the vehicle. This allows for the opportunity to use the electric motors as generators to brake the vehicle. This additional brake system contributes to several advantages which improve the system, but the complexity of the system is increased as well since the motors need to be carefully controlled and constrained to make sure they are safe and perform as intended. The electric motors are combined with electro-hydraulic friction brakes which allow for the opportunity to implement a Brake-By-Wire system which can use controllers and algorithms to achieve a better braking performance.

1.2 Description of problems

The thesis research question is “Development of an active braking controller for brake systems on electric motor driven vehicles”.

The thesis is investigating solutions to control the brakes for a heavy construction vehicle. The vehicle used in the thesis is a four-wheel-driven wheel loader where each wheel is equipped with an electro-hydraulic friction brake and an electric motor. Both the friction brakes and the electric motors are producing a braking torque when the vehicle is braking. The kinetic energy of the vehicle converts into heat and dissipates in the cooling system during braking with the friction brakes. But the kinetic energy converts into regenerated electric energy when an electric motor is used to decelerate the vehicle. This energy can be used to charge an electric storage unit like a battery and later used to drive the vehicle. Regenerative braking system is an important element in fuel saving of a hybrid electric vehicle (Tehrani, et al., 2011).

When a vehicle is braking there is something called “weight transfer” which is caused by the inertial forces of the vehicle. It causes the normal forces to increase on the front wheels and to
decrease on the rear wheels. With a static brake torque distribution the wheel loader is equally distributing the braking force which causes the rear wheels to lock up and slide long before the front wheels during a hard brake, this happens because the normal load changes. When the rear wheels lock up first the vehicle gets an unstable motion where it could spin and turn around. The total braking force is also less since the front wheels can provide with more braking force when it increases the normal force. Usually it is preferred to have the front wheels lock first rather than having the rear wheels lock first. It is however best to distribute the braking force to the wheels according to the weight transfer.

A structure of the brake system is developed where the brakes are controlled to reduce the brake distance and also achieve regenerated energy during braking. The brake blending strategy integrates the electric motors with the friction and the system primarily uses the electric motors to decelerate the vehicle, the friction brakes will be used when the brake force of the electric motor is insufficient or to hold the vehicle at standstill. Energy regeneration is achieved by using the electric motors to produce a braking torque.

The benefits of having a brake blending strategy which uses electric motors to brake and reduces the use of friction brakes are:

- Braking energy can be recuperated
- The wear of the friction brakes are reduced
- The temperature of the friction brakes are reduced which has two advantages
  - The risk of brake fading is reduced, which occurs when the friction brakes are overheated
  - The cooling system for the friction brakes can be reduced

The active braking control and the dynamical brake torque distribution system are developed to improve the braking performance by reducing the brake distance, maintaining traction and steering ability. To achieve a reduced brake distance for the vehicle an active braking control based on a wheel-slip controller is developed. It applies an individual braking torque dependent on the traction available at each specific wheel. The benefit with this system is a shorter brake distance, which means it can stop much faster in case of an emergency situation. By keeping the traction on all wheels the vehicle will also maintain the lateral forces and hence it maintains the steering ability during the emergency braking which makes it easier to avoid obstacles.

The dynamical brake torque distribution system has the objective to counteract the weight shift dynamic of the vehicle during braking. The benefit of the dynamic torque distribution is that it makes all the wheels reach the preferred slip value approximately simultaneously. A more stable braking motion is achieved where the risk of getting a spin on the vehicle is reduced.

### 1.3 Requirement specifications

The requirements for the new brake system are:

- The brake distance shall be reduced compared to the old brake system
- The electric motors shall provide regenerated energy during braking
The brakes shall be able to provide the braking torque requested from the new brake system independently of the electric motor’s limits, e.g., provide enough braking torque to maintain the reference slip value during the active braking control.

1.4 Methodology

The method to achieve the requirements in chapter “1.3 Requirement specifications” are:

- Actively controlling the brake torque on each wheel to increase the brake forces on all wheels individually
- Use the electric motors to brake the vehicle by implementing regenerative braking
- Through brake blending the friction brakes will be added when the electric motor brake torques are insufficient

A literature study is done on the area of the current state of art and knowledge within the relevant areas for this research and its technologies. A structure of the brake system is proposed on how the different components and other logics should be arranged allowing for a feasible Brake-By-Wire solution. Subsystems are then developed containing strategies for brake blending, dynamical brake torque distribution and active braking control.

There is no hardware prototype available for a real implementation or hardware in the loop tests since the wheel loader is a concept under development. Instead the strategies are implemented in a Simulink-model of the vehicle to perform simulations and test-cases, which are then evaluated.

1.5 Delimitations

The delimitations of the project are:

- Only straight braking is considered and tested, therefore braking in curves is not included or tested in this thesis.
- The physical plant models of the vehicle and brakes are developed and provided by Volvo CE. It is the structure, strategy and all the functions which are developed in this thesis.
- There is no hardware prototype available for a real implementation or hardware in the loop tests since the wheel loader is a concept under development. The system is instead implemented in a Simulink-model of the vehicle to perform simulations and test-cases.
- It is assumed that the battery never reaches full charge during the tests since the regenerated energy is stored in the batteries. In case the battery is fully charged the additional energy can be dissipated in brake resistors (Tehrani, et al., 2011). This area is however not further investigated in this thesis.
- The regeneration of energy is the maximum generated energy from the electric motor and does not consider the ability of charging a specific battery. It proves the potential of the system’s ability to supply regenerated energy. To get the end result of how much of the regenerated energy can be stored in a battery one has to make further investigations of different batteries and strategies to charge them.
2 Frame-of-reference

2.1 Vehicle dynamics during a brake

The wheel loader has four wheels and the physical structure of the vehicle is very similar to a car but with dimensions several times larger and heavier than a passenger car. The only difference of importance related to the intended area is that the wheel loader is an articulated vehicle. The body of the wheel loader is divided into two parts, one front unit and one rear unit. The rear wheels are attached to the rear unit and the front wheels are attached to the front unit. These two parts are connected with an articulated joint and the vehicle pivots at the joint whenever the wheel loader is turning. The wheels are never turning sideways in relation to their respective unit.

![Figure 1. A turning wheel loader from Volvo CE where the articulated joint is visible](image)

This might induce a slightly different turning behavior but since only straight braking is considered and simulated the articulating joint does not affect the results and therefore most of the vehicle dynamics that apply to cars are also applicable to the wheel loader in this thesis.

The vehicle dynamics can be described by Newton’s second law and hence the deceleration of a vehicle during braking can be described as following. Figure 5 shows an illustration of the forces.

\[ m \frac{dv}{dt} = -(F_{xf} + F_{xr}) \]  \hspace{2cm} (2.1)

where \( m \) is the vehicle’s total mass, \( \frac{dv}{dt} \) is the acceleration and \( F_{xf}, F_{xr} \) are the longitudinal brake forces at the contact point between the tyre and the ground for the front wheel respectively the rear wheel. The angular dynamics for a single wheel motion can be described as following. Figure 2 shows an illustration of the equation.

\[ j \frac{d\omega}{dt} = F_{xr} - T_b \]  \hspace{2cm} (2.2)
where \( J \) is the wheel’s inertia, \( \frac{d\omega}{dt} \) is the wheel’s angular acceleration, \( F_x \) is the braking force from the ground acting on the contact point to the wheel, \( r \) is the wheel’s effective radius and \( T_b \) is the braking torque from the brakes, (Ehsani, et al., 2005).

![Figure 2. Dynamics for a single wheel motion during braking.](image)

When the vehicle is braking there are brake pads pressed against the brake plates on the wheels or axles. The different setups differ dependent on different kinds of brakes but the common result is that the force between brake pad and brake plate will produce a right-angled torque counteracting the current wheel speed, this counteracting torque is called the braking torque. The braking torque from the brakes is causing a longitudinal braking force, \( F_x \), with an opposite direction to the vehicles velocity and acts at the contact point between the tyre and the road. This braking force is inversely proportional to the wheel’s radius, \( r \). The equation below presents the relation between the braking force, braking torque and the wheel’s radius when the wheel is in equilibrium.

\[
F_x = \frac{T_b}{r}
\]  

(2.3)

The maximal longitudinal braking force, \( F_x \), that can be acted on the tyre is dependent on the vertical force, \( F_z \), on the tyre and the longitudinal friction coefficient, \( \mu \), between the tyre and the road surface.

\[
F_x = \mu F_z
\]  

(2.4)

The friction coefficient is varying dependent on the road type and road condition. Asphalt is usually considered as a high friction road which allows for high braking forces but if the asphalt gets wet the friction coefficient will be decreased. Ice and snow on the road will make the road very slippery because of the very low friction coefficient, which will cause a low limit on the maximal braking force.

The friction coefficient can be unpredictable and varying even when the vehicle is driving on a road with a homogeneous surface. The reason is because the friction coefficient is dependent on the wheel’s slip value and the slip is always present as long as the vehicle is
braking. The wheel’s longitudinal slip value is often calculated as a ratio which is the normalized relative velocity between the tyre and the road (Savaresi & Tanelli, 2010). Since only the straight forward braking is considered in this thesis the tyre sideslip angle is zero and hence the longitudinal wheel slip is following.

\[
\lambda = \frac{v - r\omega}{v}
\]  

Figure 3 below shows how the friction coefficient is dependent on the road surface condition and varies with the longitudinal slip ratio. The friction coefficient for each road is also strongly dependent on the tyre’s rubber and tread. Therefore the magnitude of the data in Figure 3 can be seen as an approximate but the main purpose is to show the characteristics of how the friction coefficient varies with different roads and slips. An important observation is how the friction coefficient has a maximum peak value at a certain slip before it starts to drop again for increasing slip values.

![Figure 3](image-url)

Figure 3. The friction coefficient as a function of slip ratio and road surface condition (Savaresi & Tanelli, 2010).

A high friction coefficient allows for high braking forces for a specific load on the wheels and therefore a high friction coefficient is desirable. As pointed out there is a peak value of the friction coefficient for a certain slip point where the friction coefficient is steadily decreasing beyond that slip point. The goal is to aim for a certain slip value where the friction coefficient is at its maximum value, this allows for a high braking force. As seen in Figure 3, there is no single optimal longitudinal slip which suits all road surface conditions.

The ideal longitudinal slip is considered to be at 0.2 to get the maximum braking force on the wheels. 0.2 in slip value corresponds to a 20 % speed difference between the normalized wheel speed and the ground. A wheel slip of 0.2 is approximately close to the average of the different maximum friction values for the roads. Using 0.2 as the preferred longitudinal slip is
very common and has also been recognized and used by (Ehsani, et al., 2005), (Tehrani, et al., 2011) and (Savaresi & Tanelli, 2010) among others in their work and that is the reason for choosing the slip value of 0.2.

### 2.2 Weight transfer dynamics during braking

During braking the inertial force of the vehicle’s mass is contributing to a weight transfer (Mutoh, et al., April 2007) which affects the normal force on each wheel. The weight transfer increases the normal forces on the front wheels and decreases the normal force on the rear wheels by the same amount. Figure 4 demonstrates the weight transfer. The cause of this phenomenon is because the vehicle’s center of gravity has a certain height from the ground and the brake forces are acting on the contact points between the wheels and the ground. The height on the center of gravity point becomes the lever and thus the weight shift becomes stronger when the center of gravity is increasing in height.

![Figure 4: Demonstrating the weight transfer where the vehicle is driving on the upper picture and braking on the lower picture.](image)
If the distribution ratio of the brake torques to the front and rear wheels are static the front wheels will start to slip much earlier than the rear wheels because of the weight transfer. Whether it is the front or rear wheels that starts to slip depends on the actual distributed ratio. When the ratio is 50% the brake torque distribution to the front and rear wheels are equal and the consequence is that the rear wheels will start to slip earlier than expected because of the decreased normal force on the rear wheels. If there is no ABS system the rear wheels will lock entirely and the vehicle will lose the lateral stability.

A vehicle with the front wheels locked and the rear wheels still rolling will have a stable straight-ahead motion but without steering ability. If the rear wheels are locked and the front wheels are still rolling the vehicle will get an unstable motion where the vehicle spins and turns around. Usually it is preferred that the front wheels lock first rather than having the rear wheels lock first (Jacobson, 2011).

Figure 5 shows the forces during braking and the load for the front wheel can be found by calculating the moment equation on the rear wheel at the contact point to the ground according to (2.6) (Jacobson, 2011). The vehicle suspension is considered to be stiff in the mathematical equations for the vehicle dynamics.

![Figure 5. Illustration of the vehicle’s parameters and forces during braking.](image)

\[
F_{zf} L - mg L_r - F_{inertia} h_g = 0
\]  

where \( F_{zf} \) is the normal force on the front wheel, \( L \) is the wheelbase, \( m \) is the vehicle’s mass, \( g \) is the gravitational acceleration, \( L_r \) is the longitudinal distance between the center of gravity to the center of the rear wheel, \( F_{inertia} \) is the inertial force when braking and the \( h_g \) is the
center of gravity’s height. Since the vehicle is braking the acceleration is negative and thus the force from the inertia is following.

\[ F_{\text{inertia}} = m(-a) \]  \hspace{1cm} (2.7)

By combining (2.6) and (2.7) the normal forces can be calculated on the front and rear wheels, \( F_{zf} \) and \( F_{zr} \). They are varying dependent on the vehicle’s mass, deceleration of the vehicle and the location of the vehicle’s center of gravity point. The total normal force on the front axle can be expressed as following (Ehsani, et al., 2005).

\[ F_{zf} = \frac{mg}{L} \left( L_r - h_g \frac{a}{g} \right) \]  \hspace{1cm} (2.8)

To find the normal force on the rear wheel the same procedure can be done but the moment equation in (2.6) needs to be calculated at the contact point between the front wheel and the ground instead of the rear wheel.

\[ F_{zr} = \frac{mg}{L} \left( L_f + h_g \frac{a}{g} \right) \]  \hspace{1cm} (2.9)

With (2.4) the normal force can be translated to the longitudinal maximal braking force for the front and rear wheels, \( F_{xf} \) and \( F_{xr} \)

\[ F_{xf} = \mu \left( \frac{mg}{L} \left( L_r - h_g \frac{a}{g} \right) \right) \]  \hspace{1cm} (2.10)

and

\[ F_{xr} = \mu \left( \frac{mg}{L} \left( L_f + h_g \frac{a}{g} \right) \right). \]  \hspace{1cm} (2.11)

According to an investigation done on an electric vehicle with independently driven front and rear wheels, (Mutoh, et al., April 2007), the weight transfer compensation successfully prevents the rear wheels from locking up long before the front wheels. The study considers the weight transfer dynamic and counteracts it by reducing the amount of brake torques to the rear wheels and increases the braking torque to the front wheels. All the wheels will then reach the maximal braking force simultaneously and thereby the vehicle will brake in an efficient way and it keeps the stability as long as possible. If the braking forces exceed the maximal braking force the wheels would start to slip simultaneously.

The load transfer estimation is done with acceleration and friction coefficient as signal inputs. The acceleration can easily be measured by an inertial measurement unit which is mounted close to the center of the vehicle’s mass. The friction coefficient is however an unknown parameter which depends on the road surface condition and changes continuously. To be independent of the friction coefficient a ratio calculation is done, \( \beta_f \) and \( \beta_r \), for the distribution
of the total braking force to the front and rear wheels’ braking forces (Mutoh, et al., April 2007). The front axle’s normal force ratio varies as following.

\[
\beta_f = \frac{F_{xf}}{F_{xf} + F_{xr}} \quad (2.12)
\]

\[
\beta_f = \frac{L_r - h_g \frac{a}{g}}{L} \quad (2.13)
\]

and since \(\beta_f\) and \(\beta_i\) are ratios, the sum of them is 1. The rear wheels’ ratio can simply be calculated as following.

\[
\beta_r = 1 - \beta_f \quad (2.14)
\]

Of course the complete calculation for the rear axle’s ratio can be calculated in the same procedure as following.

\[
\beta_r = \frac{F_{xr}}{F_{xf} + F_{xr}} \quad (2.15)
\]

\[
\beta_r = \frac{L_f + h_g \frac{a}{g}}{L} \quad (2.16)
\]

Observing the equations for the ratios it can be noticed that the calculation only needs the acceleration signal as an input. The ratios are multiplied with the total requested braking force of vehicle to get the proper amount of braking force distributed to the front and rear wheels.

\[
F_{xf} = \beta_f F_{\text{brake demand}} \quad (2.17)
\]

and

\[
F_{xr} = \beta_r F_{\text{brake demand}} \quad (2.18)
\]

where \(F_x\) is the braking force acting on the wheel, \(\beta\) is the brake ratio and \(F_{\text{brake demand}}\) is the total requested braking force for the vehicle.

### 2.3 Friction brakes

There are three different systems of braking actuators when it comes to friction brakes, hydraulic, electro-hydraulic and electro-mechanical (Savaresi & Tanelli, 2010). The hydraulic
actuating brakes are the conventional brakes and most commonly used in passenger cars. The pressure on the brake pedal is transmitted to the hydraulic system directly through valves, pumps and accumulators. Since the force is transferred directly through the brake fluid, the control of brake force is strongly limited. Vehicles using hydraulic actuators combined with ABS are usually considered to have a discrete force modulation because of the large pressure gradient in the hydraulic circuit, which makes the brake to alternate between on and off in a fast pace causing hard vibrations. Because of the physical direct linkage between the brake pedal and the brake pads, the hydraulic pressure applied to the brakes cannot be bypassed.

Electro-hydraulic actuating brakes provide a force feedback similar to hydraulic actuators at the brake pedal but instead of having a direct connection all the way to the brake pads, the pedal position is measured and electronically transferred to a hydraulic unit with an electronic control unit (ECU). From there the ECU transfers the signal and actuates the brake through hydraulic fluid.

The electro-mechanical actuating brakes are similar to the electric-hydraulic brakes but it is entirely mechanical and thus it is a dry system instead of using hydraulic fluid to transfer the brake force.

The two electronically driven actuators don’t have direct connection between the brake pedal and the brake unit which allows for a continuous force modulation, all this contributes to a much better control of the brake forces. The hard vibrations experienced with the hydraulic actuating brakes no longer exists (Savaresi & Tanelli, 2010).

| Table 1. Comparison of braking systems actuators (Savaresi & Tanelli, 2010). |
|---------------------------------|-----------------|-----------------|-----------------|
| Technology                      | HAB             | EHB             | EMB             |
| Force Modulation                | Hydraulic       | Electro-hydraulic | Electro-mechanical |
| Ergonomics                      | Discrete (on/off) | Continuous     | Continuous      |
| Environmental Issues            | Pedal vibrations | No vibrations   | No vibrations   |
| Issues                          | Toxic oils      | Toxic oils      | No oil          |

2.4 Electrical motor

Electrical brakes are in this case referring to electrical motors used as generators to provide a negative torque. The available torque an electric motor can provide is dependent on the angular speed and the rated power of the motor. As the speed increases the generated power increases until it reaches the base speed. The specified rated power of an electric motor is the limiting factor. During braking the available torque will increase as the speed is decreased (Ehsani, et al., 2005) since the available braking torque is limited by a constant power generation. Figure 6 shows how the different properties of an electric motor depend on each other. An electric motor delivers a constant torque up to a certain speed where the torque hyperbolically decreases with an increasing speed (Guo, et al., 27-31 March 2009).
The braking torque from electric motors need to be controlled when used as regenerative brakes since the braking torque is a torque in the opposite direction of the wheel speed. If the magnitude of the negative torque exceeds the magnitude of the maximal braking force as explained in equation (2.4) the wheels would start to spin backwards while the vehicle is still sliding forward. Locked wheels does not only increase the brake distance but it also lowers the regenerated energy since the rotational speed is close to zero when the wheels are locked and the power regeneration is proportional to the rotational speed (Tehrani, et al., 2011).

The friction brake provides the remaining torques in case the available torque from the electric motor is lower than the total requested torque. During braking the available torque from the electric motors increases as the speed of the electric motor decreases and thus the torque from the friction brake should gradually decrease to maintain the correct total braking torque.

During braking the electric motors turns the kinetic energy into electrical energy, thus it regenerates power. This power can charge a battery according to its limitations and free capacity. Additional energy can be dissipated in brake resistors (Tehrani, et al., 2011). During standstill the electrical motors can’t provide with a braking torque, hence the friction brakes are used when holding the vehicle during standstill.

### 2.5 Brake blending strategies

Brake blending is the strategy to distribute the total brake torque between the electric motors and the friction brakes. The most common goals are to regenerate maximal amount of energy with electric motors but the vehicle dynamics and braking technologies capabilities are the main constraints. Most studies are done on cars with regenerative brakes only available on one axle and electric motors are usually insufficient at providing enough braking torque. This is the reason for many brake blending strategies where the friction brakes are always in use. Braking with only one axle induces unstable dynamics on the vehicle and is not the most optimal way of braking. Therefore several studies have been done on how to optimize the brake blending control with some optimization techniques with the objectives of maximizing the regenerated energy with the constraints of keeping the vehicle stability and maintain a safe braking.
(Falcone, et al., 15-18 December 2009) present two model predictive approaches for controlling the regenerative braking at the rear axle in cornering maneuvers where they consider yaw movements as well. The study is focusing on the cornering characteristics of the vehicle.

(Guo, et al., 27-31 March 2009) have done a study on braking force distribution by optimizing it through genetic algorithms. The optimization process is a probabilistic global search that mimics the metaphor of natural biological evolution.

(Cao, et al., 2-5 July 2008) have combined the technique of sliding mode control and neural network identification. It utilizes the sliding mode control to control the brakes and to optimize the process the neural network technique is used to do on-line parameter adjustment and system identification to achieve self-learning, self-adapting and self-organization.

Different fuzzy controllers have been developed and they have the ability to convert the linguistic expressions into an automated fuzzy rules based control strategy. They are however experiencing difficulties to guarantee stability and robustness of the system (Kim & Lee, 12 May 1995). This is why different combinations have been implemented such as fuzzy sliding-mode control (Kim & Lee, 12 May 1995) which is more robust against parameter variation. The major drawbacks about fuzzy rules are that they require previously tuned rules by time-consuming trial-and-error procedures (Lin & Hsu, March 2003).

Some vehicles are found with individual torque distribution on each wheel which has regenerative brakes and friction brakes on each wheel. These vehicles have the advantage of having regenerative brakes available at all wheels and the potential to regenerate energy is bigger. The regenerative brakes don’t have to be blended with friction brakes at all times in order to get an optimal torque distribution and achieve a stable vehicle dynamic. Instead the brake blending’s most obvious purpose is to make sure the total braking torque is correct by blending with the friction brakes when the electric motors no longer can provide enough braking torque. The most common way to achieve this is to use a method called control allocation.

(Shyrokau, et al., 2013) use a multi-layer vehicle controller with an optimization-based control allocation method to achieve vehicle dynamics control and energy recuperation for a car. The brake blending algorithm is a weighting matrix which depends on normal forces, vehicle motion and recuperation algorithms based on thresholds of the battery’s and electric motor’s physical conditions.

(Shyrokau, et al., June 2013) is a work from the same authors above with one additional author and this work has implemented the same technique introduced in (Shyrokau, et al., 2013) . A multiple level control system is used with a vehicle dynamics control and an optimized control allocation which is based on a weighting strategy that considers actuators and tyre friction constraints. They are using multi-objective formulation of independent cost functions to reach optimal characteristics of vehicle motion and energy consumptions. The brake blending is focused on recuperation of energy and tyre energy dissipation.

Both these work are based on the same technique overall but the latter one is an offline optimization method which means they are calculating the optimal brake blending ratio based on the optimization algorithm on some simulations and this brake blending ratio will then be used as a static value. They are covering the energy efficiency but are more focused on
keeping the steering and vehicle stability, they are creating a solution for an over-actuated system for commercial cars. The brake distance and energy regeneration is not the main sole issue. Looking at the optimization constraints in the control allocation technique used in (Shyrokau, et al., 2013) it can be noticed that the use of regenerative brakes are optimized against the electric motor’s angular speed, temperature, SOC, vehicle velocity, battery’s voltage and fault indication.

The total regenerated energies are not comparable since the studies referred above are not using the same test-cases. The first five references mentioned in this chapter are using more complex brake blending strategies which are focused on the procedure of optimizing the energy recuperation and vehicle stability. Mostly the setups only have regenerative braking on one axle which means the friction brakes are always used during braking since all wheels are preferred to generate a braking torque. This is a challenging factor since the regenerative braking is less efficient compared to only using regenerative braking on all wheels. The two last mentioned references are based on a more hardware focused viewpoint where regenerative braking exists on all wheels. The optimization is not as complex as the rest. The structure is simple yet effective and the work is implementing a brake blending strategy in a more complete braking solution which considers most of the vehicle dynamics.

2.6 Active braking control

The active braking control (ABC) system controls the amount of total requested braking torque on each individual wheel to achieve the highest braking force as possible. The braking force between the tyre and the ground increases as the braking torque increases until the wheel is locking up and the tyre will then start to slide on the ground. It is well known that the tractive force is decreased when the wheels lock up and slide. Not only is the braking distance increased when the wheels are locking up but the steering ability becomes bad as well since the lateral forces are decreased. To achieve the maximal braking force the braking torque needs to be at its maximum without exceeding the value where the wheel is locking up and the tyre starts to slide.

This has also been known as an anti-lock braking system (ABS) which has the same purpose as ABC. The difference between the traditional ABS system and the more modern ABC system is that the standard ABS system is usually equipped with traditional hydraulic actuators which mainly uses rule-based control logics with discrete modulation of the braking torque while modern ABC systems uses electro-hydraulic or electro-mechanical actuators which allows for a continuous modulation of the braking torque (Savaresi & Tanelli, 2010). The result is that the ABC systems can be treated as a classical regulation problem where the wheel slip can be controlled and maintained at a certain slip value with higher accuracy and better result.

In braking control systems, two output variables are usually considered for regulation purposes; wheel deceleration and wheel longitudinal slip. The traditional controlled variable, which is still used in some ABS systems, is the wheel deceleration. This is due to the fact that it can be easily measured with a simple wheel encoder. However the dynamics of a classical regulation loop on the wheel deceleration critically depends on the road conditions and can cause problems if road surface changes rapidly. Henceforth deceleration-based control strategies inherently require the online estimation of the road characteristics. Furthermore the deceleration control is usually not implemented as a classical regulation loop but instead heuristic threshold-based rules are used (Savaresi & Tanelli, 2010).
On the other hand, the regulation of the wheel slip is very robust from the dynamical point of view, which means it can handle road surface variations better. But the slip measurement is critical since it requires the estimation of the speed of the vehicle. Noise sensitivity of slip control is a critical issue. The major flaw of slip control is that the measurement of the wheel slip is comparatively difficult and unreliable, especially at low speed. The sensitivity of wheel slip control to measurement errors is a key issue. (Savaresi & Tanelli, 2010)

(Lin, et al., 24 November 1993) develops a sliding mode control which is a robust nonlinear control design. The goal is to contain the system in a sliding surface plane where a predefined function of error is zero. Neural network control has been developed and is a method for nonlinear mapping between the input-output. The neural network consists of interconnected neurons which does a nonlinear transformation to the received signals. However it needs learning and training processes. Fuzzy logic control has also been researched but tuning a fuzzy logic controller requires experience and consists of trial and error procedure. It is rather time-consuming and specifically adapted to one type of a vehicle.

The challenge when controlling the slip is that the relation between the slip and the friction coefficient is nonlinear. A typical PID-controller is linear and therefore the ability to control the slip with a linear PID controller is limited. However there are methods to customize the PID-controller for it to become nonlinear, three alternatives are considered.

- PID-controller with a nonlinear slip function
- Friction-scheduled PID-controller
- Velocity-scaling PID-controller

(Solyom, June 2002) uses a velocity-scaling PI controller with gain scheduling to control the wheel slip. The scheduling is based on the slip and an estimated friction coefficient. The solution is simple with limitations but powerful and less resource demanding on hardware. The estimations for the friction coefficient and the vehicle speed are done by a Multi-Model Observer integrated in the simulator.

(Jiang, 2000) proposes a cascaded control structure for ABS where he implements three different controllers; a linear PID controller, robust controllers using loop-shaping method and a nonlinear slip function PID controller. The PID controller is simple but has sensitivity problems although it achieves satisfactory results. The robust controller is more effective but is hard and time-consuming to tune. The nonlinear slip function PID controller is getting better results than a linear PID controller but still maintains the easy tuning advantage.

**2.7 ABC activation**

The first step towards implementing an active braking control system is to be able to detect when it should be active and not active. The braking system has two modes; emergency braking and normal braking. The driver has to be able to control the braking torque during a normal, non-critical braking but if the driver is making an emergency braking the active braking control system needs to detect the situation in order to activate and automatically control the brakes. There has to be logics which sense the switch between manual and automatic braking mode. (Savaresi & Tanelli, 2010)
2.8 Characteristic of the longitudinal wheel slip

By simplifying the vehicle model into a one-wheel model the behavior of the longitudinal wheel slip can be analyzed in a more easy way.

\[
\begin{align*}
\{ & J\ddot{\omega} = rF_x - T_b \\
& m\dot{v} = -F_x 
\end{align*}
\]  

(2.19)

The relation between the vertical force and the longitudinal brake force on the wheel is according to below.

\[F_x = F_z\mu\]  

(2.20)

(2.20) can be substituted in (2.19) which will result in following.

\[
\begin{align*}
\{ & J\ddot{\omega} = rF_z\mu - T_b \\
& m\dot{v} = -F_z\mu 
\end{align*}
\]  

(2.21)

Recall below the equation for wheel slip, \(\lambda\), which is defined as the difference of the longitudinal speed between the ground and the wheel at their contact point.

\[\lambda = \frac{v - r\omega}{v}\]  

(2.22)

where \(v\) is the vehicle speed, \(r\) is the wheel’s effective rolling radius and \(\omega\) is the wheel’s angular speed. (2.22) can also be written in two further ways.

\[\omega = \frac{v}{r} (1 - \lambda)\]  

(2.23)

\[\lambda = 1 - \frac{\omega r}{v}\]  

(2.24)

by deriving (2.24) with time one will get.
\[
\dot{\lambda} = -\frac{r}{v} \dot{\omega} + \frac{r\omega}{v^2} \dot{v}
\] (2.25)

The first equation in (2.21) can change the state variable from \( \omega \) to \( \lambda \) since they are linked with the relation in (2.22), (Savaresi & Tanelli, 2010). This is done by substituting (2.23) and (2.25) into the first equation in (2.21). The dynamic equations will then become.

\[
\begin{cases}
\dot{\lambda} = -\frac{1}{v} \left( \frac{(1 - \lambda)}{m} + \frac{r^2}{J} \right) F_z \mu + \frac{r}{Jv} T_b \\
m \ddot{v} = -F_z \mu
\end{cases}
\] (2.26)

The longitudinal dynamics of the vehicle is much slower than the rotational dynamics because of the difference of inertia in the wheel compared to the whole vehicle. This makes it reasonable to neglect the second equation in (2.26) and thereby the equation of the system becomes a first order model of the wheel slip dynamics. Thus rewriting the first equation of (2.26) one gets.

\[
\dot{\lambda} v = -\frac{F_z \mu}{J} r^2 - \frac{F_z \mu}{m} (1 - \lambda) + \frac{r}{J} T_b
\] (2.27)

Equation (2.27) will be used later in the implementation part to motivate the choice of controller.
3 Implementation

3.1 Brake system strategy

The requested brake torque is decided by the driver’s pressure on the brake pedal during normal braking. The brake system dynamically distributes the brake torques to match the individual normal forces on each wheel based on the vehicle's deceleration to compensate for the weight transfer. When an emergency braking is detected by the ABC activation system the ABC system will take over the control of the brakes and provides an automatic braking torque. It increases the braking torque but still maintains the wheel’s traction and thereby the vehicle gets a shorter braking distance without losing the vehicle stability and steering ability.

The purpose of the brake blending system is to provide with regenerated energy from the electric motors and it is done based on the wheel speed. Thereby the brake blending system is active on both normal braking and emergency braking. The electric motors themselves are insufficient as the only brake system and thus the friction brakes provides with additional braking torque to guarantee that the total braking torque is corresponding to the requested brake torque.

The brake system strategies when the vehicle is braking are:

- To dynamically distribute the total brake demand from the pedal based on the acceleration to compensate for the weight transfer.
- Apply the required brake torque at each wheel with electric motor brakes.
- Blend in the friction brake to satisfy the requested brake torque if the electrical motor brake torque is less than the requested brake torque.
- If emergency braking is detected, switch from manual braking to active braking control.

3.2 Brake system structure

The brake system is structured into sub-systems consisting of ABC activation system, active braking control system, dynamical brake torque distribution system, brake blending system and the individual brakes. See Appendix A for a flowchart illustration of the structure. Every subsystem has a specific task and purpose with simple inputs and outputs.

When the brake pedal is pressed by the driver there are two ways the signal can propagate through the brake system. During normal braking the driver controls the brakes manually, this means the brake pedal is acting as a feed forward system where the vehicle is braking according to the driver’s manual request i.e. the brake pedal actuation. In manual braking the dynamical brake torque distribution makes sure to provide the proper amount of braking torque to the front and rear wheels based on the vehicle’s deceleration to compensate for the weight transfer. The second path is the active braking control system which applies an automatically controlled brake torque on each individual wheel to increase the brake forces without locking up the wheels. The ABC activation system decides if the actual braking is an emergency braking or normal braking and will then decide if the manual braking torque is to be used or the active braking control system should override the driver’s braking request. The brake blending system contains the strategy of mixing the regenerative braking with the
friction braking. The individual brakes are the electric motor brake and the controlled friction brake.

### 3.3 Dynamical brake torque distribution system

The braking is manually controlled by the driver through the brake pedal during a non-emergency braking. The driver controls the total braking torque but the distribution of the brake torque to the front and rear wheels will be dynamic based on the vehicle’s deceleration according to the chapter “2.2 Weight transfer dynamics during braking”. By dynamically distributing the brake torque to the front and rear wheels the vehicle can increase the total braking torque and prevent the rear wheels from locking up before the front wheels, thereby the risk of getting an unstable motion where the vehicle spins and turns around is reduced. The rear wheels saturate sooner than the front wheels when a static ratio is used for distributing the brake torque. The rear wheels will start to slide while the front wheels are still rolling and could provide additional braking torques. Thus the risk of getting an unstable motion where the vehicle spins around is increased when using a static brake torque distribution.

The system gets a brake pedal signal and a predefined brake torque value to identify the requested manual brake torque for the vehicle. The brake pedal decides how much of the predefined brake torque to request. The brake pedal value ranges from 0 to 1 which is multiplied with the predefined brake torque. A fully pressed brake pedal corresponds to 1 and a fully released brake pedal corresponds to 0.

The system receives the total torque request from the driver and calculates the brake torque ratio to the front and rear wheels with the vehicle acceleration signal from the inertial measurement unit on the vehicle. The system then outputs the brake torque request to the front and rear wheels which counteract the weight transfer during braking.

![Figure 8. The dynamical brake torque distribution system with inputs to the left and outputs to the right of the box.](image)

The system only needs the acceleration as a real-time input signal. The remaining parameters used in the calculation are static geometrical values like the wheelbase, the center of gravity’s height and the longitudinal distance from the center of gravity to the rear wheel axle. The center of gravity’s height is the most difficult parameter to measure and is therefore tuned through simulations to find the appropriate value. The center of gravity’s height is found where both the front and rear wheels are approximately locking simultaneously without the ABC system. The important thing is to make sure the wheels either lock simultaneously or having the front wheels lock slightly before the rear wheels since this is preferred (Jacobson, 2011).
Figure 9. The real-time calculation of the brake torque ratio for the front and rear wheels.

Equation (2.13) and (2.14) is implemented in Figure 9 and that is how the dynamical brake torque distribution system distributes the brake torques to the front and rear wheels.

### 3.4 ABC activation system

The ABC activation recognizes whether the braking is an emergency braking or normal braking and it will switch between manual braking and automatic braking according to different conditions. The ABC activation system is developed as a state machine in Simulink. Figure 10 shows the system’s input and output, an illustration of the state machine can be found in Appendix B: ABC activation system.

The ABC activation system has two modes; ABC ON and ABC OFF. The active braking control system is automatically controlling the brake torque during ABC ON and the driver is manually braking during ABC OFF. Within each of the two modes there are two states which are separated by the speed of the vehicle since the controlled braking gets unreliable and less necessary when the vehicle is moving in a slow speed.

When the ABC is off the torque is manually controlled through the brake pedal no matter if the vehicle speed is high or low. But the ABC system can only be turned on when the vehicle is moving in high speed. The ABC system uses the last manual braking torque value when it activates and starts to compensate from that value. The last braking torque is maintained as a static value when the vehicle reaches a very low speed during ABC ON mode until the ABC
activation system turns off the automatic braking. (3.1) shows how the braking torque is defined by the ABC activation system.

$$
f(x) = \begin{cases} 
    \text{Torque\textsubscript{manual}}, & \text{ABC} = \text{OFF} \text{ and } 0 < v < \infty \\
    \text{Torque\textsubscript{last} - Torque\textsubscript{auto}}, & \text{ABC} = \text{ON} \text{ and } v > v\textsubscript{set} \\
    \text{Torque\textsubscript{last}}, & \text{ABC} = \text{ON} \text{ and } v < v\textsubscript{set}
\end{cases}
$$

(3.1)

where $f(x)$ is the output of the ABC activation system which is the required braking torque, $\text{Torque\textsubscript{manual}}$ is the manual braking torque from the pedal, $\text{Torque\textsubscript{auto}}$ is the automatic braking torque controlled by the ABC system, $\text{Torque\textsubscript{last}}$ is the last value of the output torque, $v$ is the actual speed of the vehicle and $v\textsubscript{set}$ is a parameter threshold set for which speed the ABC is allowed to be activated and for when to transition the ABC brake torque to be static according to the third equation in (3.1).

The ABC system is not able to activate when the vehicle starts moving since the vehicle speed is too low. Once the vehicle increases the speed above the speed threshold value the state transitions to the high speed mode within the ABC OFF mode. A hysteresis, $v\textsubscript{hysteresis}$, is added to prevent the system from chattering between the high speed state and the low speed state in the ABC OFF mode. The condition to switch from the low speed to the high speed within the ABC OFF mode is following.

$$
v > v\textsubscript{set} + v\textsubscript{hysteresis}
$$

(3.2)

To get back to the low speed mode the similar but reversed condition is applied.

$$
v < v\textsubscript{set} - v\textsubscript{hysteresis}
$$

(3.3)

In the high speed state within the ABC OFF mode there are two possible transitions, either going back to low speed or going to ABC ON since the vehicle is considered to have enough speed for a reliable control with the ABC system. To activate the ABC in high speed the following condition needs to be fulfilled.

$$
\text{Torque\textsubscript{manual}} > \text{(Torque\textsubscript{manual} - Torque\textsubscript{auto}) \&\& slip} \\
> s\text{lip}_\text{set} \&\& v\text{vehicle} > v\text{set}
$$

(3.4)

where $-\infty < \text{Torque\textsubscript{auto}} < \infty$. The first condition in (3.4) detects that the driver’s braking demand exceeds the controlled braking torque according to the ABC system, the second condition detects if the vehicle starts to slip more than allowed and the third condition makes sure the vehicle is still going in a speed fast enough to activate the ABC system.

The preferred slip value to keep is 0.2 and this is the reference slip value the ABC system will maintain. However the $s\text{lip}_\text{set}$ value in (3.4) is the slip value on when to activate the controller in the ABC system, the choice of this value is critical and makes a large impact on the controller’s performance. Instead of activating the controller at 0.2 when the slip value already exceeds the reference value the controller needs be activated slightly before it reaches the reference value. During simulations the $s\text{lip}_\text{set}$ value was set to 0.18 and the performance of the controller was greatly improved by almost eliminating the overshoot compared to activating the controller at 0.2 slip. Having the $s\text{lip}_\text{set}$ value at 0.18 is close enough to 0.2 to not cause any unnecessary activation. This simply means that the definition of an emergency braking is when the slip exceeds 0.18 in slip instead of 0.2, this among with the other
conditions in (3.4). The results of a test case, as described in chapter “4.1 Test procedure and setup”, when the $\text{slip}_\text{set}$ value is set to 0.2 and 0.18 are presented in “Appendix C: Comparison of $\text{slip}_\text{set}$ activation”. The choice of activating the ABC at 0.18 slip is reasonable and have been implemented by others. (Tehrani, et al., 2011) uses 0.18 slip as a threshold where the manual braking is applied when the slip value is 0.18 or less. When the slip value exceeds 0.18 slip the controller decreases the braking torque.

Once the ABC activates in high speed mode the output brake torque is according to the second equation in (3.1), it uses the last manual brake torque value and controls it by subtracting the last manual brake torque with the controlled brake torque. The output brake torque is the sum of both values. The controlled brake torque is a customized PID controller which will be more detailed described in the chapter about the ABC system.

It is possible to make the output brake torque just a value of the controlled brake torque without including the last manual brake torque value as following.

$$ f(x) = -\text{Torque}_{\text{auto}} $$ \hspace{1cm} (3.5)

This approach works with almost the same result, the difference is that the output value might start being controlled from a starting value further away from the last used manual braking torque output. This causes the transition to be larger than necessary when going from the manual brake torque to the controlled brake torque. Instead the output controlled brake torque used is in (3.1) but is repeated for the purpose of easier understanding.

$$ f(x) = \text{Torque}_{\text{last}} - \text{Torque}_{\text{auto}} $$ \hspace{1cm} (3.6)

A comparison was made which confirmed that (3.6) gave better results than (3.5) where the step response was more stable with a faster settling time and with less overshoot. (3.6) is the chosen to be implemented since it gave the best performance. The comparison made between (3.5) and (3.6) were using a test case described in “4.1 Test procedure and setup” and the results are presented in “Appendix D: Comparison of different $f(x)$”.

The ABC system requires the driver to keep the brake pedal pushed down to maintain active. The system will end the braking if the driver releases the brake pedal and thus the ABC activation system will check for this action. When the following condition is fulfilled the ABC system turns off and transitions back to the high speed state in ABC OFF mode.

$$ \text{Torque}_{\text{manual}} == 0 $$ \hspace{1cm} (3.7)

An important issue in this area is to make sure that the system does not chatter between two different states. Therefore the conditions need to be carefully designed with decent margins to reduce the risk of unintended switching between two different states.

The state transitions to the low speed state within the ABC ON mode when the ABC system has effectively reduced the vehicle speed to a certain value. The vehicle speed is slow enough to switch the controlled braking torque off and replace the output brake torque with the last brake torque value as a static output. An alternative here is to take the last output brake torque and gradually reduce the braking torque output to get a smoother end of the braking.
At the low speed state in the ABC ON mode the system checks for a released brake pedal or a vehicle speed below the threshold, $v_{stop}$, in which case it will return the control of the brakes back to the driver by transitioning to the low speed state in ABC OFF mode as following.

$$v < v_{stop} \mid Torque_{manual} = 0$$ (3.8)

### 3.5 Active braking control system

The ABC system controls the amount of total requested braking torque on each individual wheel to achieve the highest braking force as possible. The controller’s input is the wheel slip and output is the torque. There is one controller for each wheel making it possible for the system to control each wheel individually.

A simple controller with the ability for easy tuning is preferable and a PID controller is considered to fulfill these requirements, furthermore Volvo CE desires an investigation of such a controller. A linear PID controller is studied and implemented but unsatisfactory results are obtained due to the braking characteristics being a nonlinear behavior. The vehicle’s velocity, wheels’ slip and road friction coefficient are all nonlinear and this is causing the braking characteristic to be nonlinear. The best controller would counteract all three nonlinearities but such a controller would be too complex and therefore three different gain-scheduled PID controllers are considered where each controller counteracts one of the mentioned nonlinearity. The preferable controller is the one that achieves the best overall performance despite only counteracting one of the mentioned nonlinearity. The three nonlinear gain-scheduled PID controllers are:

- Velocity-scaling PID controller
- PID controller with a nonlinear slip function
- Friction-scheduled PID controller

There has to be an appropriate scheduling variable in order to implement a gain-scheduled PID controller (Lingman, 2005). The friction coefficient is a suitable scheduling variable. (Solyom, June 2002) uses the maximum friction coefficient as one of the scheduling variables which is being estimated. The friction coefficient needs to be estimated to achieve the surface identification and since no such estimation is included in the thesis the friction-scheduled PID controller is not further investigated or implemented.

The PID controller with a nonlinear slip function is defined as (Jiang, 2000).

$$u(t) = K_p f(e(t), \alpha, \beta) + K_i \int e(t) \, dt, \alpha, \beta) + K_d \frac{d}{dt} e(t), \alpha, \beta)$$ (3.9)

$f(x, \alpha, \beta)$ is a nonlinear function where $x$ is $e(t)$ and $e(t)$ is the wheel’s slip. The nonlinear slip function is defined as.

$$f(x, \alpha, \beta) = \begin{cases} \text{sign}(x) |x|^\alpha, & \text{if } |x| > \beta \\ x^{\beta-1}, & \text{if } |x| \leq \beta \end{cases}$$ (3.10)
where $\alpha$ and $\beta$ are two tunable parameters, $\alpha$ is a value from 0 to 1 and $\beta$ is a small positive number to create a linear part when $x$ is close to zero to avoid oscillations when $x$ is small.

During an emergency braking the ABC activation system switches to the active braking control system. The ABC system implemented is based on a wheel slip controller which controls the total brake torque on the wheel to keep the longitudinal wheel slip at 0.2. The wheel slip is controlled by a velocity-scaling PID controller with output tracking. It is based on a conventional linear PID controller where the error is scaled by the vehicle’s velocity and the output of the controller is compared to the output of the brake system. A linear PID controller is defined as following.

$$u(t) = K_pe(t) + K_i \int e(t) \, dt + K_d \frac{d}{dt} e(t)$$  \hspace{1cm} (3.11)

where $e(t)$ is the wheel slip. Equation (2.27) shows how the dynamics of the wheel’s slip, $\dot{\lambda}$, is scaled with the inverse of the longitudinal velocity, $v$. By scaling the error in the PID controller with the velocity the system becomes independent of the vehicles velocity. This is a similar approach as done in (Solyom, June 2002). Thereby the error, $e(t)$, becomes as following.

$$e(t) = v(t)(\lambda(t) - \lambda_{ref})$$  \hspace{1cm} (3.12)

where $v(t)$ is the vehicle’s velocity, $\lambda(t)$ is the measured wheel slip and $\lambda_{ref}$ is the reference slip value. The longitudinal wheel slip value is calculated accordingly.

$$\lambda = \frac{v - r \omega}{v}$$  \hspace{1cm} (3.13)

where $\lambda$ is the wheel’s longitudinal slip, $v$ is the vehicle’s speed calculated from the measured acceleration measured by the IMU on the vehicle, $r$ is the wheel’s effective radius and $\omega$ is the wheel’s angular velocity.

Since the wheel slip controller is calculating the control signal continuously even when the ABC control is not active the integral part of the PID controller will get a stationary offset. Therefore an output tracking signal that feeds back the output $f(x)$ to the controller needs to be included to eliminate this integral offset. It can be explained as a way for the PID controller to know the actual output of the system, $f(x)$. Thereby a closed loop system is maintained accordingly.

$$e_i(t) = v(t)(\lambda(t) - \lambda_{ref}) + (y_{tr}(t) - u(t))$$  \hspace{1cm} (3.14)

where $u(t)$ is the output control signal from the PID controller i.e. the $Torque_{auto}$ in (3.1) and $y_{tr}(t)$ is the output brake torque from the ABC activation system i.e. $f(x)$ in (3.1). Equation (3.14) will compensate for when there is a difference in the output brake torque signal and the output signal from the PID controller. The resulting velocity-scaling PID controller with output tracking becomes.

$$u(t) = K_pe(t)v(t) + K_i \int e(t)v(t) + (y_{tr}(t) - u(t))dt + K_d \frac{d}{dt} e(t)v(t)$$  \hspace{1cm} (3.15)
The PID parameters are tuned with the Ziegler–Nichols method and then iteratively adjusted through simulations and the best performance is achieved with the following parameters.

- \(K_p = 350,000\)
- \(K_i = 4,000,000\)
- \(K_d = 10,000\)

The linear PID controller, the velocity-scaling PID controller presented in equation (3.15) and the PID controller with a nonlinear slip function from (Jiang, 2000) are implemented. The comparison is using a test case described in “4.1 Test procedure and setup” and the results of the linear PID controller, the velocity-scaling PID controller and the PID controller with a nonlinear slip function are presented in “Appendix E: Comparison of different wheel slip controllers”.

The nonlinear slip function PID controller and the velocity-scaling PID controller have a better braking response than the linear PID controller. The disadvantage with the nonlinear slip function PID controller is that two additional parameters require tuning whereas the velocity-scaling PID controller doesn’t have additional parameters that require tuning.

The comparison between the different controllers, in Table 7, shows that the brake distance is insignificantly small between each controller. However the velocity-scaling PID controller achieves the most stable response with the least overshoot during the controlled braking. It is also simple and most effective without additional tuning parameters and that is the reason for choosing the velocity-scaling PID controller for the final implementation.

### 3.6 Brake blending system

Electric motors need to always be combined with friction brakes when regenerative brakes are available on only one axle since all wheels need to brake to maintain the vehicle stability. The brake blending system is the strategy on how the total brake torque is distributed between the electric motors and the friction brakes. By having electric motors on all wheels there is an opportunity of only using the electric motors as regenerative brakes until the electric motors reach their own capacity of delivering the brake torque.

There are some brake blending optimizations developed for cars but they are complex and hard to implement in a real-time application because of the heavy computational processing using resource-restricted hardware. It is desired to achieve a simple brake blending strategy with the ability to be tuned and calibrated easily, which is also preferred by Volvo CE. The above mentioned optimized brake blending algorithms are therefore not the best suitable strategies since a novel brake blending strategy is desired which is easily applicable without requiring excessive computational power.

The proposed brake blending system requires all the brake torque from the electric motors since the objective is to regenerate as much energy as possible. The total required brake torque is sent to the lookup table of the motor’s maximal brake torque available depending on its angular speed. The lookup table for the electric motor is calculated with the torque and speed acted on the wheel which means the gear ratio is included. Figure 11 shows the relation between the electric motor’s speed and torque acted on the wheel.
The lookup table represents the maximum available brake torque from the electric motor based on the angular speed and is the limiting factor when requesting an electric brake torque.

The brake blending system receives the measured output brake torque from the electric motor which is compared to the requested total brake torque. If the delivered brake torque from the electric motor is insufficient, the friction brake will be requested to deliver the rest of the brake torque as following:

\[ T_{friction} = T_{total} - T_{electric} \]  \hspace{1cm} (3.16)

The brake system architecture is a series coupled system where the total braking torque, \( T_{total} \), is the combined resulting torque by both the friction brakes, \( T_{friction} \), and the electric motor, \( T_{electric} \). The friction brakes and electric motors are not physically dependent of each other and therefore the total braking torque can either be only friction brakes, only electric motors brake or a combination of both.

The regenerative brakes are turned off and the friction brakes completely deliver all the brake torque when the vehicle is at standstill since the electric motors have difficulties of braking when the electric motor has no angular speed. The brake blending is not dependent on the ABC activation and thus the brake blending is active as long as the wheel speed is above a certain speed threshold. The electric motors shutoff and the friction brakes will take over when the wheel speed decreases below the threshold since they are more ideal for very low speed and most importantly holding the vehicle at standstill.

### 3.7 Friction brakes

Volvo CE wants a simple controller with the ability of easy tuning. A PID controller is explicitly requested by the company to be investigated. It is detected that a PI-controller gives satisfactory results and a PID-controller does not make a noticeable improvement. Since a PI
controller is simpler and also works satisfyingly it is chosen to be implemented. The friction brakes are closed-loop controlled with PI controllers to make sure the total braking torque requested is satisfied. The PI-controller is expressed as following.

\[ u_{\text{pressure}}(t) = K_p e(t) + K_i \int e(t) \, dt \]  

(3.17)

where \( u_{\text{pressure}}(t) \) is the input pressure to the friction brakes and \( e(t) \) is the error accordingly.

\[ e(t) = T_{\text{req}} - T_{\text{out}} \]  

(3.18)

The PI-controller is saturated at 5 MPa brake pressure with back-calculation as anti-windup method. The parameters for the P- and I-gains are tuned with the Ziegler–Nichols method for the highest friction road and from simulations the parameters of the controller are:

- \( K_p = 200 \)
- \( K_i = 200 \)
4 Result

4.1 Test procedure and setup

The physical models of the wheel loader, friction brakes and electric motors have been developed by Volvo CE. The vehicle model is based on a Volvo GSP standardized model developed for the L350 wheel loader. The vehicle parameters are modified to match the wheel loader used in this thesis, the L500 wheel loader. The vehicle model is using SimMechanics blockset in Simulink with Matlab R2012b. The tyre models are developed by TNO Delft-Tyre which are using the Pacejka Tyre model with the Magic Formula.

The simulations are done with a test-case constituting of an emergency braking so the shortest possible brake distance is measured and the ABC system can prevent the wheels from locking up. The vehicle is accelerating to the maximum vehicle speed of 30 km/h. The driver is then applying the brakes at simulation time 11 seconds, the brake pedal travels for 1 second before it reaches full brake demand at simulation time 12 seconds. The brakes are then held at full brake demand until the vehicle stops to a standstill i.e. 0 km/h.

It is assumed that the tyres are the same for all simulations so the friction coefficient is dependent on the road condition. Low friction coefficient corresponds to slippery roads like mud, snow and ice. High friction coefficient corresponds to asphalt and dry roads. The friction coefficients do not change during a simulation, the road condition is the same for the entire simulation and when the friction coefficients are changed the simulation is redone.

First a quantitative collection of data is made with the same simulation test case for eight different friction coefficients ranging from 0.23 to 0.78 in an evenly spread interval. All the simulations are done with different configurations to recognize the improvement contributed by each system. There are three different configurations where the active braking control system and the brake blending system is turned on and off. The dynamical brake torque distribution system is always on during this quantitative collection of data since it is not affecting the recorded data in a significant magnitude. The different configurations of the systems during the quantitative test are seen in Table 2.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Active Braking Control</th>
<th>Brake Blending</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Off</td>
<td>Off</td>
</tr>
<tr>
<td>B</td>
<td>On</td>
<td>Off</td>
</tr>
<tr>
<td>C</td>
<td>On</td>
<td>On</td>
</tr>
</tbody>
</table>

The first configuration, A, is the conventional configuration corresponding to the vehicle without the developed systems. This means the wheels are not prevented from being locked and there is no energy regenerated, the electric motors are not used when there is no energy regenerated.

The purpose of the second configuration, B, is to confirm how the individual systems are contributing and affecting the results. Therefor the ABC is turned on and the BB is turned off which means the wheels are prevented from being locked but there is no energy regenerated.
The third configuration, C, is corresponding to the complete developed system where both the ABC and BB is turned on and thus the wheels are prevented from being locked while the electric motors will regenerate energy.

Additional four qualitative test cases are made for a qualitative analysis on the results. These are all made with the friction coefficient of 0.78 and can be seen in Table 3.

Table 3. The different system configurations for the four qualitative test cases.

<table>
<thead>
<tr>
<th>Test case</th>
<th>Active Braking Control</th>
<th>Brake Blending</th>
<th>Dynamical Brake Torque Distribution</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test case 1</td>
<td>Off</td>
<td>Off</td>
<td>Off</td>
</tr>
<tr>
<td>Test case 2</td>
<td>Off</td>
<td>Off</td>
<td>On</td>
</tr>
<tr>
<td>Test case 3</td>
<td>On</td>
<td>On</td>
<td>Off</td>
</tr>
<tr>
<td>Test case 4</td>
<td>On</td>
<td>On</td>
<td>On</td>
</tr>
</tbody>
</table>

In test case 1 to 4 the wheel slip is presented in figures. It can be recalled from equation (2.5) that the slip is calculated as.

\[ \lambda = \frac{v - r\omega}{v} \]  \hspace{1cm} (4.1)

But according to the standards used at Volvo CE the slip calculation is.

\[ \lambda = \frac{r\omega - v}{v} \]  \hspace{1cm} (4.2)

Both equation (4.1) and (4.2) can be used when calculating the slip since the slip is a normalized speed difference of the wheel and the ground. The difference is only opposite sign values and is a matter of definition. When the vehicle is accelerating the slip value is the same for both equations but the sign is negative in (4.1) and positive in (4.2). When the vehicle is braking the slip value is also the same for both equations but the sign is positive in (4.1) and negative in (4.2). Locked wheels that are sliding on the ground are therefore having a slip value of 1 in (4.1) and -1 in (4.2).

To be in line with the slip calculation used at Volvo CE the slip value presented in the results are using the equation (4.2) where the slip value is -1 for locked wheels which slide on the ground.
4.2 Quantitative test

The simulations are done with three different configurations where the DBTD system is always on and the friction coefficients between the tyre and the road is varying, the variation is in the range from 0.23 to 0.78 in an evenly spread spectrum.

Table 4 below presents the total brake distance measured for all the configuration cases and friction coefficients. Also the case when the ABC and BB are off is compared with the case when the ABC and BB are on. The measured value is the brake distance reduction in meters. Last in the table is the brake distance reduction presented in percentage which shows the improvement compared to the total brake distance on the specific friction coefficients.

Table 4. Comparison of brake distance on different system configurations with varying friction coefficients.

<table>
<thead>
<tr>
<th>Friction Coefficient</th>
<th>0,23</th>
<th>0,31</th>
<th>0,39</th>
<th>0,47</th>
<th>0,54</th>
<th>0,62</th>
<th>0,70</th>
<th>0,78</th>
</tr>
</thead>
<tbody>
<tr>
<td>Brake Distance [m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(ABC = off, BB = off)</td>
<td>34,41</td>
<td>25,96</td>
<td>20,91</td>
<td>17,56</td>
<td>15,20</td>
<td>13,51</td>
<td>12,28</td>
<td>11,33</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Distance [m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(ABC = on, BB = off)</td>
<td>29,43</td>
<td>21,64</td>
<td>17,27</td>
<td>14,59</td>
<td>12,87</td>
<td>11,73</td>
<td>10,96</td>
<td>10,42</td>
</tr>
<tr>
<td></td>
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<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Brake Distance [m]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(ABC = on, BB = on)</td>
<td>29,05</td>
<td>21,41</td>
<td>17,1</td>
<td>14,46</td>
<td>12,76</td>
<td>11,63</td>
<td>10,87</td>
<td>10,35</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Distance Reduced [m]</td>
<td>5.36</td>
<td>4.56</td>
<td>3.81</td>
<td>3.10</td>
<td>2.44</td>
<td>1.88</td>
<td>1.41</td>
<td>0.99</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake Distance Reduced [%]</td>
<td>15.6%</td>
<td>17.6%</td>
<td>18.2%</td>
<td>17.6%</td>
<td>16.1%</td>
<td>13.9%</td>
<td>11.4%</td>
<td>8.7%</td>
</tr>
</tbody>
</table>

The values in the table are presented in a graphical way for easier understanding. The result is translated into some graphs where the trend of the result is more noticeable. Figure 12 below represents the brake distance when comparing the different system configurations.
Table 4 and Figure 12 confirm the brake distance being greatly reduced by the active braking control system. The second configuration shows that the brake blending is slightly contributing to this improvement but the active braking control system is the main reason to the improved brake distance. The total brake distance increases when the friction is decreased. This occurs because the braking forces between the wheels and the ground decrease since the braking force are proportional to the friction coefficient as seen in equation (2.4). For the setup where the ABC is off the wheels are locking up and this results in a longer brake distance since the friction decreases when the wheels are locking up, as seen in Figure 3.

Figure 13 below shows the total reduced brake distance in meters when the ABC and BB are on compared to when the ABC and BB are off.
Figure 13. The brake distance reduced with the new brake system in meters.

Figure 13 shows how the amount of reduced brake distance increases with a decreased friction coefficient. This means the ABC system reduces more distance on low friction roads compared to high friction roads. On roads with a friction coefficient of 0.23 the ABC is reducing the most brake distance, which is more than 5 meters.

Figure 14 compares the reduced brake distance with the total brake distance for the conventional setup on each friction coefficient. It shows the improvement in brake distance with the new brake system.

Figure 14. Brake distance reduced with the new system in percent.
The efficiency of the active brake system at each friction coefficient value is calculated by comparing the total brake distance when the ABC and BB are on with the total brake distance when the ABC and BB are turned off. Figure 14 shows that the active brake system reduces the brake distance from 9% to 18%.

The amount of energy regenerated during the simulations is the total energy output from all the four electric motors and the value is measured from the moment the driver presses down the brake pedal through the entire simulation until the vehicle stops. Table 5 below presents how much energy the electric motors can regenerate during the braking. The equivalent amount of energy is also presented in Wh-unit for easier understanding of the amount of energy regenerated. The simulation below is done with both the ABC and BB on.

Table 5. Data of energy regeneration and dissipation with the new brake system.

<table>
<thead>
<tr>
<th>Friction Coefficient</th>
<th>0.23</th>
<th>0.31</th>
<th>0.39</th>
<th>0.47</th>
<th>0.54</th>
<th>0.62</th>
<th>0.70</th>
<th>0.78</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Regenerated [MJ]</td>
<td>3.05</td>
<td>2.85</td>
<td>2.49</td>
<td>2.17</td>
<td>1.92</td>
<td>1.73</td>
<td>1.59</td>
<td>1.48</td>
</tr>
<tr>
<td>Energy Regenerated [Wh]</td>
<td>846.2</td>
<td>792.9</td>
<td>691.9</td>
<td>602.9</td>
<td>533.6</td>
<td>480.5</td>
<td>440.4</td>
<td>409.9</td>
</tr>
<tr>
<td>Friction Energy [MJ]</td>
<td>0.37</td>
<td>0.60</td>
<td>1.01</td>
<td>1.38</td>
<td>1.68</td>
<td>1.92</td>
<td>2.12</td>
<td>2.28</td>
</tr>
<tr>
<td>Energy Recuperated [%]</td>
<td>89.3%</td>
<td>82.6%</td>
<td>71.2%</td>
<td>61.2%</td>
<td>53.4%</td>
<td>47.3%</td>
<td>42.8%</td>
<td>39.3%</td>
</tr>
</tbody>
</table>

The values in the table are presented in a graphical way for easier understanding. The result is translated into some graphs where the trend of the result is more noticeable.

When the vehicle is braking the kinetic energy from the moving vehicle will either be transformed into electric energy by the electric motors, i.e. regenerated energy, or the energy will be dissipated as heat through the friction brakes. Figure 15 below shows how much of the kinetic energy is regenerated by the electric motors and how much is dissipated through the friction brakes.
The share of how much of the kinetic energy is possible to recuperate through regenerative braking is presented in Figure 16 below.

Figure 15. Amount of energy regenerated and dissipated compared.

Figure 16. Percentage of energy recuperated during the brake tests.
Table 5 shows how the regenerated energy is increasing with a decreasing friction and this is because the braking forces are decreasing along with the friction and hence the friction brakes are less used and the electric motor usage is increased. Table 5 together with Figure 15 and Figure 16 confirm that the brake blending system is regenerating from 39 % to 89 % of the vehicles kinetic energy from braking.

The use of friction brakes are reduced when the brake blending system is on since the electric motors produce a certain amount of the braking torque. Table 6 below calculates the friction brake usage based on how much of the energy is regenerated and dissipated. The friction brake usage is compared with the ABC active on both test cases because without ABC the wheels will lock and most of the energy is dissipated at the contact point between the tyre and the ground instead of at the brakes.

Table 6. Data of the friction brake usage on different friction coefficients with the new brake system.

<table>
<thead>
<tr>
<th>Friction Coefficient</th>
<th>0.23</th>
<th>0.31</th>
<th>0.39</th>
<th>0.47</th>
<th>0.54</th>
<th>0.62</th>
<th>0.70</th>
<th>0.78</th>
</tr>
</thead>
</table>

Figure 17 shows how much the friction brakes are used when comparing the case of having the brake blending on with the case of having the brake blending off.
Table 6 shows how the friction brake usage is decreased for a decreasing friction coefficient since the lower friction only allows for lower braking forces. The electric motor only provides a limited braking torque so on low friction roads the total amount of braking torque is low and most of the braking torque comes from the electric motor. On high friction roads the total amount of braking torque is higher and since the braking torque from the electric motor is limited the friction brakes are used more. Table 6 and Figure 17 confirm that the friction brake usage is reduced with 51 % to 91 % when having the brake blending system on compared to when the brake blending system is off.
4.3 **Test case 1**

In test case 1 the friction coefficient is 0.78 and all the systems; ABC, BB and DBTD are turned off, which corresponds to the conventional brake system where only the friction brakes are used. Since the dynamical brake torque distribution is off the distribution ratio is set to a static value of 50 %, which means the distribution of the requested brake torque from the pedal will always be 50 % to the front wheels and 50 % to the rear wheels.

Figure 18 below shows the wheel slip of the front and rear wheel. 0 in wheel slip corresponds to the case where the wheel is not affected by a torque, i.e. the wheel is free-rolling or standing still. -1 in wheel slip corresponds to a fully locked wheel where it slides on the ground.

Figure 18 shows that the slip increases for both the front and the rear wheels until they are locked, they continue to slide until the vehicle has stopped. The rear wheels are locking up before the front wheels and therefore there is a risk of getting an unstable motion where the vehicle gets a spin as explained in chapter “2.2 Weight transfer dynamics during braking”.

![Figure 18. wheel slip without ABC, BB and DBTD.](image)

Figure 19 shows the vehicle speed along with the normalized wheel speed for them to be comparable. The wheel speed drops to zero while the vehicle speed still decreases which corresponds to the wheels being locked while the vehicle still moves forward. This confirms the wheels being locked while the vehicle continues to slide forward.
Test case 1 has all the systems; ABC, BB and DBTD turned off. It shows how the wheels quickly lock and start to slide even on a high friction road. This results in a longer brake distance, no traction, lack of lateral forces and therefore no steering ability. The rear wheels locked up before the front wheels which increases the risk of getting a spin on the vehicle since the rear wheels have no traction while the front wheels still have traction for a moment. Since only the friction brakes are used no energy is regenerated either.
4.4 Test case 2

In test case 2 the friction coefficient is 0.78 with ABC and BB turned off while DBTD is on to see how the system performs with only a dynamical brake torque distribution.

Figure 20 shows how the front wheels start to slip before the rear wheels since the DBTD system is compensating for the weight transfer. The front wheels lock first and therefore the risk of getting an unstable motion of the vehicle where it spins around is reduced. Getting the front and rear wheels to start sliding at the exact same time can be hard to tune but the main objective is to make sure that the rear wheels lock slightly after the front wheels, since this is preferred (Jacobson, 2011).

![Actual slip front wheel](image)

![Actual slip rear wheel](image)

Figure 20. Wheel slip with dynamical brake torque distribution.

Figure 21 shows the normalized wheel speed and the vehicle speed. The rear wheels start to decrease the speed before the front wheels but the DBTD system starts to compensate and prevents the rear wheels from slipping before the front wheels.
Test case 2 which only has the dynamical brake torque distribution system active shows how all the wheels are being locked again and start to slide but this time the DBTD system prevents the rear wheels from locking up before the front wheels. The front wheels start to slip 0.1 seconds before the rear wheels and the risk of getting a spin of the vehicle is reduced.
4.5 Test case 3

In test case 3 the friction coefficient is 0.78 with the ABC and BB turned on while DBTD is off. Figure 22 shows when the ABC system activates and deactivates itself according to the conditions in the ABC activation system. The driver is manually controlling the brakes when the brake mode is on 1 and 2, the difference between these two modes are the different vehicle speed. The ABC is activated when the brake mode reaches 3 and the ABC is transferring to an intermediate level at 0 before it returns the control of the brakes to the driver at mode 1. It can be seen that the front wheels does not activate the ABC system until the end of the braking.

Figure 22. The ABC Activation system shows when the ABC system activates.

Figure 23 shows how the ABC system successfully controls the rear wheel slip at 0.2 during the emergency braking without locking the wheel. But the front wheel is only maintained at 0.1 slip since it doesn’t get enough braking torque to increase the slip to 0.2.
Figure 23. The wheel slip when ABC and BB is on and DBTD is off.

It can be seen in Figure 24 that the wheels never lockup because of the ABC system but the front wheels are not braking as hard as the rear wheels.

Figure 24. The normalized wheelspeeds and the vehicle speed with ABC and BB but without DBTD.
Test case 3 which has the DBTD system turned off and ABC on shows how the DBTD system is needed in order to activate the ABC and provide enough braking torque to maintain a wheel slip of 0.2. Without the DBTD system the front wheels can only reach 0.1 in slip.
4.6 Test case 4

In test case 4 the friction coefficient is 0.78 with all the systems active; ABC, BB and DBTD are turned on.

Figure 25 shows when the ABC system activates and deactivates itself according to the conditions in the ABC activation system. The driver is manually controlling the brakes when the brake mode is on 1 and 2, the difference between these two modes are the different vehicle speed. The ABC is activated when the brake mode reaches 3 and the ABC is transferring to an intermediate level at 0 before it return the control of the brake to the driver at mode 1. It can be seen how both the front and rear wheels are activating the ABC system at level 3 almost simultaneously. The front wheels activates about 0.1 seconds before the rear wheels, just as preferred (Jacobson, 2011).

![ABC Activation, front wheel](image)

![ABC Activation, rear wheel](image)

Figure 25. Shows when the ABC controlled braking is active.

Figure 26 below presents how the wheel slip is successfully maintained at 0.2 with a controlled braking torque when the ABC is activated. As the speed decreases and reaches standstill the slip returns to 0 and the system will return the control of the brakes back to the driver. Since the slip never reaches -1 when the ABC is active the wheels never lock during the braking. It can also be seen how the rear wheels are reaching a slip value of 0.2 about 0.1 seconds after the front wheels reaches 0.2 slip.
Figure 26. Wheel slips maintained at 0.2 and never locks during active ABC.

Figure 27 shows the vehicle speed and the normalized wheel speeds and it confirms how the wheels never lock during the ABC controlled braking. Both the front and rear wheels are braking equally hard in relation to their respective normal forces and the total braking torque is higher.

Figure 27. The normalized wheelspeed never drops to zero during braking and thus never locks.
Test case 4 with all the systems; ABC, BB and DBTD turned on shows a significant improvement where the emergency braking is successfully detected before the wheels lock up. All the wheels are maintained at the reference slip value of 0.2 where the highest braking forces are achieved. The traction and lateral forces are kept since the wheels never lock up and thereby the steering ability is maintained.
5 Conclusion

Braking is a crucial part of the safety and reliability of a vehicle. Implementing a Brake-By-Wire system, which uses a combination of electric motors and friction brakes, can open up for major advantages. At the same time the complexity is increased as well. When using electric motors the braking controller is even more important.

The results from the test cases confirm how the dynamical brake torque distribution system prevents the rear wheels from locking up and slide before the front wheels because of the weight shift. Thereby it reduces the risk of getting a spin on the vehicle. The DBTD system is also important in order to make the active braking control system work as intended and maintain the reference slip value. Without the DBTD system the front wheels don’t get enough braking torque to maintain 0.2 slip. The brake blending system succeeds to regenerate energy during braking with the electric motors and can regenerate the braking energy with an increasing efficiency on slippery road conditions. The active braking control system is successfully preventing the wheels from locking up and individually controls the brakes on each wheel to maintain a reference slip value of 0.2. It provides an increased braking force compared to the old brake system. This results in a reduced braking distance on all road conditions. By preventing the wheels from locking up the ABC system maintains the ability to steer and control the vehicle even during an emergency braking.

Some significant improvements can be seen when comparing the performance of the new brake system developed in the thesis with the old conventional brake system. The total brake distance is reduced with 9 % to 18 % during an emergency braking. The shorter brake distance contributes to a safer braking since the vehicle can stop on a shorter distance in case any obstacles suddenly appear in front of the vehicle. 39 % to 89 % of the braking energy is regenerated through regenerative braking. The regenerated energy is stored in the battery and can be used to drive the vehicle. The friction brakes are 51 % to 91 % less used. Since the friction brakes are less used the cooling system can be reduced and the production cost can be decreased.

The new brake system is developed for a four-wheel-driven wheel loader which uses friction brakes and electric motors to brake. The results are satisfying and an improved performance is achieved in brake distance, energy regeneration, friction brake usage, vehicle stability and steering ability compared to the conventional brake system. The functionality works reasonable and the next step is to make fail-safe tests and analyze what happens if something fails. This should then be implemented in hardware in the loop tests.

The choice of controller is made considering the requirements of it being simple and easy to tune, furthermore Volvo CE desires an investigation of such a controller. There are more complex controllers that can be investigated in the future. The requirements of the thesis defined in the introduction are all considered to be fulfilled.
6 Discussion

If the slip dynamics in equation (2.27) is linearized in operating points around the peak in the friction curve it is noticed that the tyre slip dynamics is stable to the left of the peak where the slope of the curve is positive. Further the tyre slip dynamics is unstable to the right of the peak where the slope of the curve is negative (Solyom, June 2002). This is probably the main reason for the increased performance of the ABC controller when activating the automatic braking at 0.18 slip since the peak is not yet reached and the slip is still on the positive slope, hence the tyre slip dynamics is on the stable side.

The electric motors have a faster response time compared to the friction brakes (Hori, 2009). This is probably the reason to why the brake distance is slightly reduced when the BB is on, configuration C, compared to when the BB is off, configuration B, in Table 4 and Figure 12.

One combination of configuration is not performed during the quantitative test in Table 4. The combination not tested is when the ABC is turned off and BB is turned on, i.e., not controlling the brakes which means they are not prevented from locking up but still being able to regenerate energy by braking with the electric motors. The reason is because the electric motors have the ability to drive the wheels backwards. The conventional friction brakes will at all times counteract the current speed direction of the wheel whether it is forward or backward. This is the reason why friction brakes are ideal to use when holding the vehicle at standstill. If there is no ABC system active and the negative torque from the electric motor is greater than the braking force from the ground acted on the wheel the electric motors will make the wheels spin backwards while the vehicle is still sliding forward. Thus when using electric motors to brake a vehicle there has to be some kind of brake control system implemented. Therefore this fourth combination of configuration is not simulated since there is no realistic implementation of this setup without further development. Theoretically there is a possibility to simply shut the electric motors off once the wheel encoder is indicating that the wheel speed is close to zero. This would however not be a preferred braking solution since the wheels could still lock up and the regenerative braking would be less efficient.

Figure 18 in test case 1 contains a peak of the slip right before the wheels lock. This is most probably due to how the slip is being calculated and how the negative torque from the friction brakes are being calculated. The moment the wheel is being locked the wheel speed goes to zero and the brake force induced by the friction brakes might get slightly greater than the force induced on the wheel from the ground. This is probably only a phenomenon in the simulations and does not exist in a real implementation since a real friction brake is only countering the force induced by the ground and will never exceed it. The difference is that the simulations are discrete and works in iterations with a certain step time. The peak would disappear as the step time of the simulation would decrease to an infinitely small value of time step. Since the time duration of the peak is very short it is insignificant to the result and is therefore neglected. The peaks on the wheel speeds in Figure 19 are caused by the same phenomenon. The calculation of the slip is not reliable once the vehicle speed is close to reaching zero since the slip is calculated with the vehicle speed in the denominator which would cause the slip calculation to divide by zero and go to an infinite. So once the speed is close to zero the denominator in the slip calculation is substituted with the smallest value possible for Matlab to prevent the simulation from crashing. The slip value is not reliable and hence not used to control the brakes once the vehicle speed reaches zero. The ABC system turns off when the vehicle reaches a slow speed.
Figure 26 in test case 4 shows a peak of the slip in the end of the test case, right before the vehicle comes to total standstill. This peak in the slip occurs right when the vehicle speed reaches zero and the control of the brakes has been transferred back to the driver. It is caused by the driver holding the brakes until the vehicle entirely stops. The peak corresponds to a hard stop when the deceleration of the vehicle goes to zero very fast and thus the weight shift resets the normal forces back to the original proportions when the vehicle is at standstill. This can be described as the phenomenon when a vehicle is making a hard braking where the weight shift causes the chassis to tilt forward. Once the vehicle has stopped entirely the chassis is bouncing back to the original position since the deceleration decreased to zero. These peaks can be avoided by simulating the driver making a soft braking by gradually decreasing the brake demand when the vehicle reaches zero speed but this would result in a longer brake distance, which contradicts one of the requirements. Additionally the driver is not part of the automated brake system and therefore it is not of importance for the system’s functionality whether the driver decides to make a hard stop or soft stop at the end of the braking. The small peaks on the wheel speeds in the end of Figure 27 are caused by the same phenomenon since the wheel speeds are normalized based on the angular velocity of the wheels.
7 Future work

The ABC control system and the friction brakes have the same PID- and PI-parameters for both front and rear wheels in respectively controllers. In this thesis satisfactory results were achieved but improvement can be done by tuning the parameters individually for the front and rear wheels.

Since only straight braking is considered in the simulations a thorough study should be done on the behavior and the performance when the vehicle is turning at the same time as it is braking, so called braking in curves. Where yaw, pitch and roll motion is considered and braking on inclined roads.

The energy regenerated from the electric motors is the calculated possible energy output. The electric motors output a power which is used to charge a battery. The actual amount of charge that can be stored in the battery, the effective regenerated energy, is dependent on the battery and the charging strategy. It is useful to make a study on batteries and charging strategies to find the effective regenerated energy that can be stored. This is also important since the state of charge of the battery affects the available torque from the electric motor. The battery should not be fully charged for regenerative braking to be possible. It is suggested in the thesis to use brake resistors when the batteries are saturated but there might exist better solutions.


Appendix A: Brake system structure

Figure 28. Illustration of the brake system structure.
Appendix B: ABC activation system

Figure 29. The state machine in Simulink of the ABC Activation system.
Appendix C: Comparison of $\text{slip}_{\text{set}}$ activation

Figure 30. The wheel slip during the braking when the $\text{slip}_{\text{set}}$ value is 0.2

Figure 31. The wheel slip during the braking when the $\text{slip}_{\text{set}}$ value is 0.18
Appendix D: Comparison of different $f(x)$

Figure 32. The wheel slip during the braking when $f(x) = -\text{Torque}_{\text{auto}}$

Figure 33. The wheel slip during the braking when $f(x) = \text{Torque}_{\text{last}} - \text{Torque}_{\text{auto}}$
Appendix E: Comparison of different wheel slip controllers

Figure 34. The wheel slip during the braking when using a linear PID controller.

Figure 35. The wheel slip during the braking when using a PID controller with a nonlinear slip function.
Figure 36. The wheel slip during the braking when using a velocity-scaling PID-controller.

Table 7. Comparison of the brake distance between the different wheel slip controllers.

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<th>Friction Coefficient</th>
<th>0,23</th>
<th>0,31</th>
<th>0,39</th>
<th>0,47</th>
<th>0,54</th>
<th>0,62</th>
<th>0,70</th>
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<td>21,43</td>
<td>17,1</td>
<td>14,47</td>
<td>12,76</td>
<td>11,62</td>
<td>10,86</td>
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<td>Brake Distance [m] (velocity-scaling PID)</td>
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<td>17,1</td>
<td>14,46</td>
<td>12,76</td>
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