

The test and simulation of ABS on rough, **non-deformable** terrains

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Abstract

It is well known that the performance of many antilock braking systems (ABS) deteriorate on rough, **non-deformable surfaces** due to a number of factors such as axle oscillations, wheel speed fluctuations and deficiencies in the algorithms. Rough terrain excitation further contribute to tyre problems such as loss of vertical contact and poor contact patch generation that leads to reduced longitudinal force generation. In this study, a slightly modified version of the Bosch ABS algorithm is implemented in Matlab/Simulink using co-simulation with a validated full vehicle ADAMS model that incorporate a valid high-fidelity FTire model. A non-ABS test vehicle is fitted with a commercial ABS modulator controlled by an embedded computer. The co-simulation model is validated with vehicle test data on both flat and rough terrains. Initial results show that wheel speed fluctuations on rough terrain cause inaccuracies in the estimation of vehicle velocity and excessive noise on the derived rotational acceleration values. This leads to inaccurate longitudinal slip calculation and poor control state decisions respectively. It is concluded that, although the correlation is not yet as desired, the combined use of a simulation model and test vehicle can be a useful tool in the research of ABS on rough terrains.

Keywords: Anti-lock Braking System, rough terrain, Bosch algorithm, simulating ABS, ADAMS, FTire.

1) Introduction

The safety of modern vehicles can be attributed to a wide range of complex systems that aim to prevent loss of life, whether it assists the driver in an active sense or simply provides passive protection. However, the effectiveness of the braking system of the vehicle has always been one of the primary factors that will save occupant's lives.

With the increased popularity of multi-purpose vehicles such as Sport Utility Vehicles (SUVs), that is often used under off-road conditions, a new challenge appeared namely that ABS brake performance is

negatively affected by rough road conditions. Road input excitation deteriorates braking performance which result in longer stopping distances due to a number of suspension and tyre contributing factors. ABS on such terrains is faced with a particularly challenging task as rapid pressure changes add further complications to the already noisy wheel speeds. Solving the braking problem on rough terrains will provide increased safety to off-road drivers and motorists who travel on roads with poor condition.

This study aims to develop a simulation model and a test platform to conduct further research into the rough road ABS braking problem.

1.1) Deterioration of performance

Road input excitation from rough terrain leads to a number of contributing factors to the rough road ABS problem which include, amongst others, normal load variations due to axle oscillations and results in increased stopping distances and reduced lateral control of the vehicle. Naturally this becomes especially clear at the resonance region of the suspension system (Van der Jagt et al., 1989). Vehicle body motions such as roll, pitch and yaw that are excited by rough road input also disturb the slip control of the wheel (Reul and Winner, 2009). Sudden change in wheel speed, as the wheel encounters an obstacle for example, will also result in fluctuations in adhesion, since peak friction of the tyre is dependent on wheel speed (Satoh and Shiraishi, 1983). Several other issues are also listed in Koçlu & Cinar (2011) with specific focus on the effect of worn dampers on ABS. Controller malfunction can occur because of noisy wheel speed sensors (Blundell and Harty, 2004), and faults in the algorithm that leads to poor control decisions (Watanabe and Noguchi, 1990).

1.2) Possible solutions

Since longitudinal braking force generation is heavily dependent on vertical load, the rough terrain braking problem can essentially be improved with an adequate suspension control. Niemz & Winner (2006), and Reul & Winner, (2009) explain the use of a semi-active suspension to switch between soft and hard damping during the braking manoeuvre to minimise vertical load variation on the braking wheels. Results indicate a 3.5% reduction in braking distance when braking on rough terrains with Continuous Damping Control (CDC) vs. fixed hard damping during the same manoeuvre.

The inaccurate calculation of longitudinal slip arising from poor vehicle velocity estimation has a severe effect on the ABS algorithm used in this study. In the ABS algorithm, a check is performed at the start of each control cycle for excessive slip and brake pressure is consequently reduced. Jiang and Gao, (2000)

proposed an adaptive nonlinear filter that uses wheel speed information alone to estimate vehicle velocity. Daiss and Kiencke, (1995) discuss the use of additional sensors such as a longitudinal accelerometer and a gyroscope for use with a Kalman filter and fuzzy logic to yield an estimation of vehicle velocity to within 1% of the true value.

1.3) Tyre models

As for most vehicle dynamic simulations, tyres contribute extensively to the complexity of the problem. This is due to the highly non-linear visco-elastic road contact mechanism, which leads to challenging mathematical modelling. Furthermore, rough terrain simulation adds various other complexities such as vibrational excitation of the carcass and complicated contact patch mechanics to the problem. Thus, the choice of tyre model will ultimately determine the success of any ABS simulation. A high fidelity tyre model with good transient capabilities and accurate contact patch dynamics, which is valid up to high frequencies, is needed. The tyre model must be capable of describing lateral, longitudinal and vertical tyre forces accurately.

Some of the tyre models that have been used in previous ABS studies includes the Fiala model, (Ozdalyan and Blundell, 1998), but is restricted for use on flat road simulation. The Pacejka 2002 tyre model is the best of the semi-empirical model class and some modified versions of this, such as a stretched string or contact mass models, are specially developed models for ABS simulations. These models incorporate a first order differential equation to account for the important relaxation length phenomenon that is dependent on vertical load and slip angle (Jaiswal et al., 2010). The Short Wavelength Intermediate Frequency Tyre (SWIFT) model is often recommended for ABS simulation (MSC Software web page, 2012). SWIFT is essentially a semi-empirical Pacejka 2002 model combined with a rigid ring model to better describe modal response.

The Flexible Ring Tire (FTire) model, as the name suggests, is an advanced tyre model that uses a flexible ring connected to a rigid hub. Gipser, (1999) developed the model over the past 15 years. The tyre model is based on a structural dynamics approach unlike the empirical or semi-empirical models explained before. Radial, tangential and lateral stiffness's define a flexible ring and are made up of some 100 to 200 "belt elements" that are used to numerically approximate the generated tyre forces. This model takes modal responses into account and is accurate up to 120 Hz (Oosten, 2011). FTire requires many input parameters and tyre data determined by laboratory tests to set the model up correctly (Stallmann, 2013). The FTire model can accurately simulate driving on a rough road, and is accurate to high frequency excitations

(Stallmann and Els, 2014). Apart from the extensive parameterization effort required, FTire is expected to provide the best results from all the tyre models considered under rough terrain braking conditions.

2) Control algorithm implementation

Several ABS strategies and control methodologies are available today. State-of-the-art control strategies, such as sliding mode, gain scheduling and fuzzy logic control, finds its use in ABS research as software implemented simulation models (Aly et al., 2011). The original Bosch algorithm follows simple bang-bang control and was chosen for the use in this study as it is well documented and easily implementable in both simulation and on an embedded system. It is published by Bauer and Bosch (1999) and used in simulation by Day and Roberts (2002). A hydraulic modulator uses an arrangement of solenoid valves to switch between three states, namely pump, dump and hold to control the angular acceleration and longitudinal slip of each of the four wheels of the vehicle. As with conventional ABS, the system used in this study only accepts four wheel speeds as input, and derives the required angular acceleration and velocity estimation to perform its decision making logic.

2.1) Bosch algorithm

The ABS algorithm uses a bang-bang strategy to control brake pressure and ultimately longitudinal force generation. Eight phases are used in the algorithm that dictates the control state of each wheel, namely pump, dump or hold, or an alternating combination of these states, such as pump-hold or dump-hold. A typical control cycle, together with the eight Bosch phases and their respective states for braking on high friction surfaces is depicted in Figure 1.

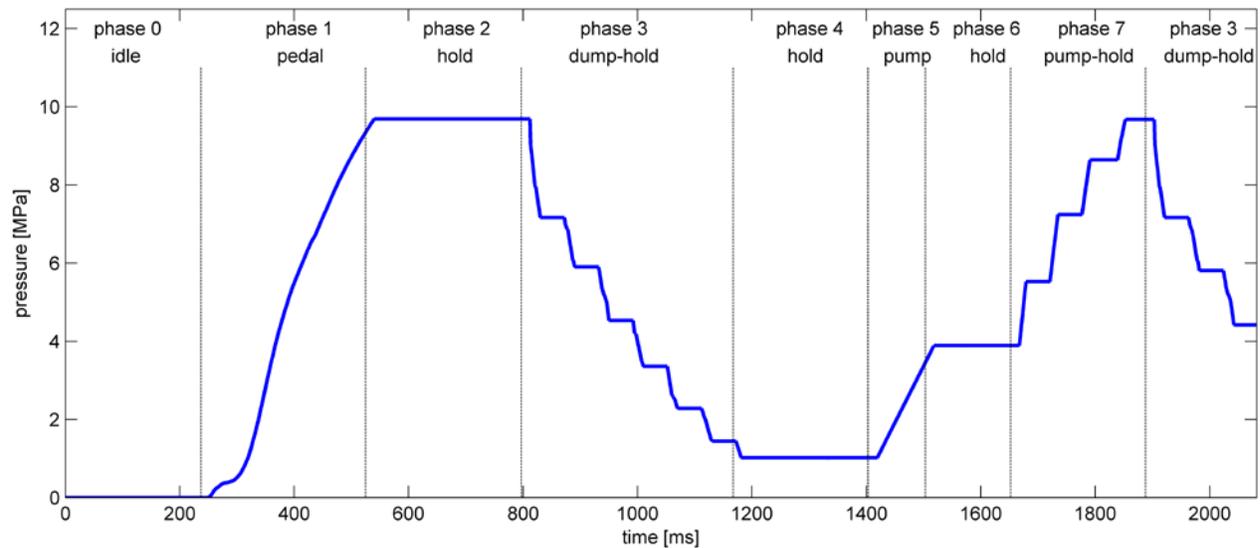


Figure 1: Brake pressure, phases and states for a typical control cycle on a high traction flat surface

The two variables that are used in the algorithm for phase decision and ultimately brake pressure is wheel angular acceleration and longitudinal slip. These variables are bounded by three thresholds, minimum and maximum angular acceleration and maximum allowable longitudinal slip, and is vehicle and tyre specific. The choice of thresholds is important to ABS performance and was tuned in simulation with the starting point taken as recommended by Day and Roberts, (2002). The final thresholds for both simulation and test are set as -50 and 50 rad/s^2 for the minimum and maximum angular acceleration and 0.15 for slip.

The classical Bosch algorithm does not decrease the pressure in a step-wise manner in phase 3 as indicated in Figure 1, but rather dumps the pressure as quickly as possible to keep longitudinal slip of the tyre below the set threshold in the least amount of time. However, the response delays associated with the mechanical relays used on the test vehicle rendered the classic algorithm ineffective, as the controller senses too late that lateral stability has been regained, and the pressure has been lost. Thus, a slightly modified algorithm was used in an attempt to address the response delays of the electro-hydraulic hardware. Furthermore, the classical Bosch algorithm makes use of a “yaw-moment build up delay system” (GMA) to minimise the common mu-split braking problem. GMA senses which side of the vehicle experiences more traction, and compensates by introducing a delay between control cycles, and hence pressure build up, at the high tractive wheels. Although beneficial in a mu-split scenario, the GMA system responds to a brake-in-the-turn scenario by increasing the dynamic loads at the front wheels, and hence increased lateral force, which results in an unwanted oversteering effect. An additional accelerometer is

employed to deactivate the GMA at high lateral acceleration (Bauer and Bosch, 1999). Since this study is only concerned with straight line braking on high tractive surfaces, the GMA system will be omitted.

2.2) Reference velocity

The angular velocity of each wheel on the test vehicle, ω , is measured by means of a proximity sensor that detects sixty passing serrations per rotation. Four frequency to voltage converters convert the pickup signals to analogue voltages. An exponential averaging filter is used on the embedded computer to minimise noise amplification during the time derivative. The accurate estimation of vehicle velocity is of ultimate importance to the success of ABS since the calculation of longitudinal slip depends on the velocity of the vehicle and is also used to determine when the ABS should be switched off to prevent numerical instabilities. During normal driving conditions, little longitudinal slip is present and the no-slip assumption is valid to calculate the vehicle velocity as;

$$v = \omega r \quad \text{Eq.1}$$

However, during severe braking manoeuvres slip is significant and the no-slip assumption does not hold. Also, the effective rolling radius on rough terrain may deviate significantly from that on a smooth terrain. This leads to the problem that vehicle velocity can no longer be calculated by direct measurement of the wheel speed sensors as in Eq. 1. At best, the velocity can be estimated with certain assumptions, such as a constant deceleration value, and is now called the reference velocity. Equation 2 is an iterative equation that is to be used when any measureable longitudinal slip is present;

$$v_{i+1} = v_i - R\Delta t \quad \text{Eq. 2}$$

R in Eq.2 is chosen to be 9.81, corresponding to the maximum acceleration that the vehicle will experience. The wheel angular acceleration is not only used as the variable that controls the Bosch phase, but is also used to determine whether or not the wheel is slipping. Consider the linear acceleration for a free rolling tyre with no slippage;

$$a_0 = \alpha r \quad \text{Eq.3}$$

If a_0 is calculated to be larger than μg , **significant** longitudinal slip is present and Eq. 2 should be used to calculate the reference velocity. **The above method is used individually for all four wheels, and four reference velocity values are thus calculated in each iteration. From these four values, the largest is selected as the overall reference velocity, as it is assumed that the algorithm will only operate during**

braking manoeuvres and as such, the ωr -term can never be larger than the true velocity of the vehicle. It should be noted that this statement may not be true on rough terrain braking or braking-in-the-turn manoeuvres, since wheel suspension and different corner radii, respectively, will cause pronounced differences between the overall reference velocity and the individual wheel centre velocities.

3). Test platform & ADAMS model

The vehicle that is used as the test platform for this study is a 1997 Land Rover Defender 110 TDi. It is retrofitted with several systems and sensors that facilitate effective testing and accurate measurement. The standard spring and damper suspension setup has been replaced with a hydro-pneumatic semi-active suspension system (Els et al, 2007). Since this vehicle did not come standard with ABS at the time, it was retrofitted with a four-channel Wabco modulator that has since become standard on the newer Land Rover Defender Puma models. The hydraulic modulator houses two solenoid valves per wheel and a shuttle valve that registers pedal actuation. All control and data acquisition on the vehicle is done by means of an embedded computer running the Linux operating system, and digital outputs are used to switch relays to actuate the solenoids in the hydraulic modulator. An Inertial Navigation System (INS) provides true vehicle speed. The embedded computer was set to measure and calculate all necessary inputs to the ABS algorithm at a 1000Hz, whereas the algorithm controlled the hydraulic modulator at a much lower rate of 100Hz, which was predominantly restricted by the switching time of the relay-solenoid combination. In order to measure all the relevant forces that act on the hub of the vehicle, a Wheel Force Transducer (WFT) was used during the braking tests. The WFT measures all the forces and moments through an arrangement of six load cells and a telemetry system and allows for easy validation and performance evaluation. Four pressure transducers are also installed in the brake lines as close to the callipers as possible that allow for accurate measurement of the brake pressure that is supplied to the pad. The WFT is used on the front right wheel, as shown in Figure 2. Note the use of outriggers as a safety precaution.



Figure 2: Test vehicle showing Wheel Force Transducer and outriggers

A 15 degree of freedom full car model of the test platform, including driver, passenger and outriggers, is modelled in ADAMS by Thoresson et al (2009) and is used with co-simulation in Matlab/Simulink for braking on rough terrain (Hamersma and Els, 2014). A validated FTire model (Bosch et al., 2014) is used for simulating ABS brake manoeuvres on the Belgian Paving at Gerotek Test Facilities (Gerotek Test Facilities, n.d.). Becker and Els, (2014) measured the three dimensional Belgian Paving profile with great accuracy to be used as a road profile in simulation, and was found to approximate a class D road as per ISO 8608, (1995). The simulation model is set to run at the same discrete rate as the embedded computer at 1000Hz. A driver model is implemented to ensure that all simulations are done in a straight as possible line (Botha, 2011).

4). Modelling of ABS

Accurate modelling of ABS hydraulics is extremely challenging, therefore many researchers use Hardware-in-the-Loop (HiL) to circumvent this (Heidrich et al., 2013). In this study, a data driven modelling approach is used to simulate the ABS system. It follows that several lookup tables and measured delays form part of

the Simulink model that is used in co-simulation with ADAMS. The Simulink solver was set to solve at discrete time steps, in this case 1 millisecond, to have the simulation model follow the same operation as the embedded controller. The ABS control logic is implemented in Simulink as an interpreted function that determines the Bosch phase to be converted to pressure. Look-up tables provide the pressure value for the next iteration depending on the current value and Bosch phase. Finally, pressure is converted to brake torque by using a linear relationship that is also drawn from measurements. Figure 3 depicts the layout of the Simulink model used in co-simulation.

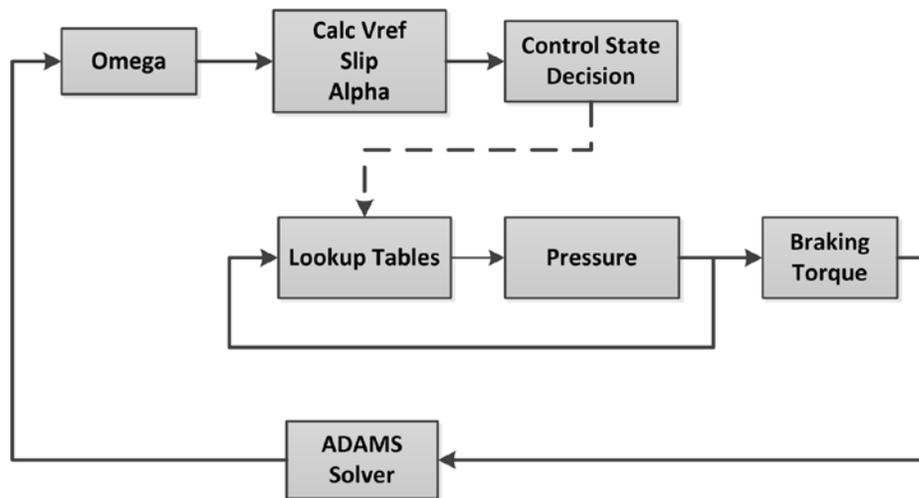


Figure 3: Simulink model layout

4.1) Pressure look-up

A test program coded on the embedded computer in C is used to cycle the hydraulic modulator through the three states (pump, dump and hold) and their alternating combinations (dump-hold and pump-hold) to produce look-up tables that are used to model the pressure in the Matlab/Simulink environment. The three individual states are fairly simplistic to characterise, however the quick alternation between states results in transients that are not fully captured by the single state look-up and thus the alternating states have separate look-up tables. Figure 4 indicates the measured system pressure for the different states of the ABS cycle.

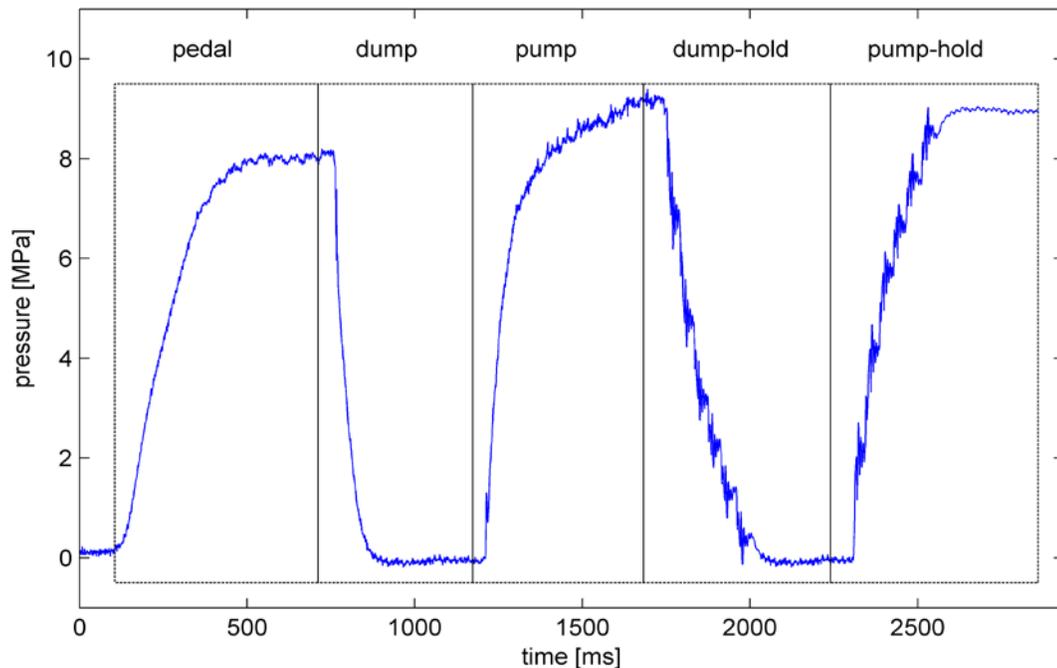


Figure 4: Pressure produced by test program to construct lookup table

In Figure 4, a section of the data is selected, lightly filtered, and all double values eliminated to produce a monotonic function. Note the response delays that are present in Figure 4, albeit exaggerated here. Using the pressure from the monotonic function as the output of the lookup table, and the same output shifted one iteration prior as the input of the lookup table, a pressure dependent lookup table is constructed. Having pressure dependent lookup tables forces the simulation model to sample at the same rate at which the lookup tables were constructed.

The Bosch phases 3, 7 and 8 that alternate between pump and hold, or dump and hold states, require separate look-up tables. These phases are set to alternate between states at double the controller's frequency and transients, such as response delays, play a major role and thus the pressure characteristics are different from the normal pump, dump or hold states. The same technique as explained before is used, however a more aggressive filter is needed to process the data. Since it is known at which time intervals the states alternate, simple averaging and linear spacing functions are employed to filter the data. Figure 5 shows the measured and filtered data.

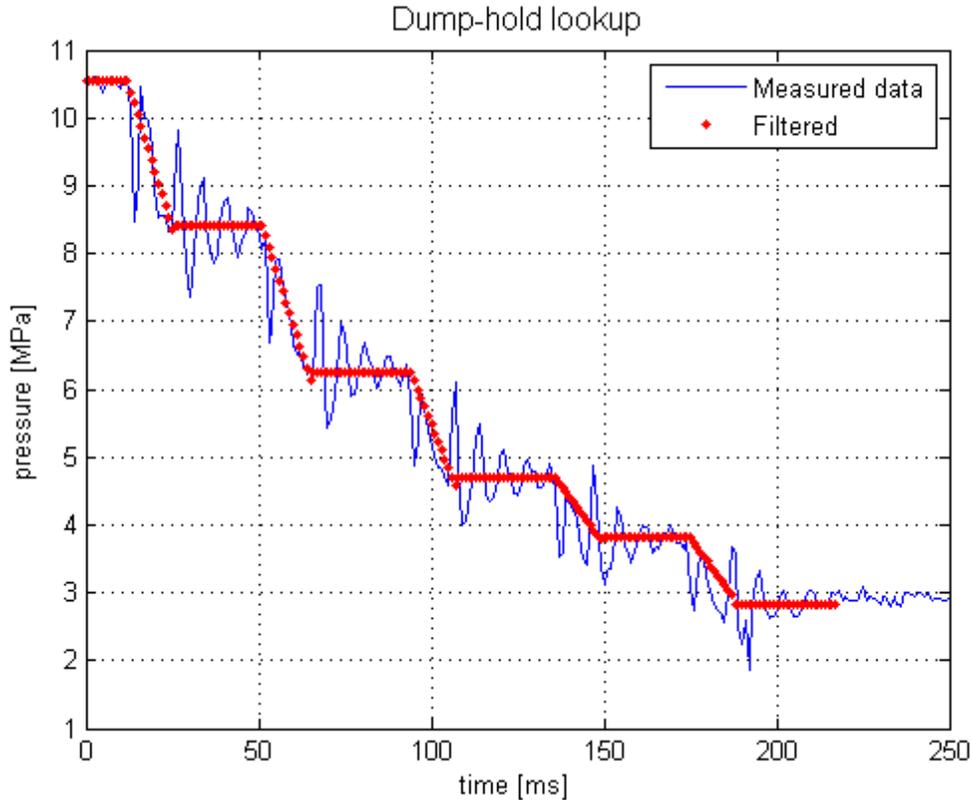


Figure 5: Pressure data used for dump-hold look-up table

The dump-hold states are set to alternate at double the controller frequency, making each cycle 10ms long since the controller operates at 50Hz. It is thus easy to identify which state is active during which time interval and subsequently the pressure data is averaged during the hold intervals, while a linear spacing function is used to produce a gradient for the dump intervals, as seen in Figure 5. After the data has been properly filtered, it is used in the lookup table for the specific Bosch phases. Severe aliasing may be present in the measured pressure data, but is of little concern since the trend of the data is known and is filtered accordingly. For the pump-hold and dump-hold states, a duty cycle is defined to allow for steadier increase or decrease of pressure. In this study, a pump-hold duty cycle of 50% (5ms pump + 5ms hold = 10ms cycle) is used. The dump-hold duty cycle was defined at 33% (3.3ms dump + 6.7ms hold = 10ms cycle).

The test program also provided the opportunity to measure the response delays associated with each state. These delays are measured such to capture all the underlying delays associated with the electro-mechanical relays, solenoids, pressure build up and torque development in a single response delay value. Several

additional delays were also noticed in initial test data, and Table 1 summarizes the delays for each state, as well as several other delays that were observed in test data.

Table 1: Response delays for the different phases

State	Bosch phase	Response delay [ms]
Pedal	0, 1	8
Pump	5,7	6
Dump	3, 8	14
Hold	2,3,4,6,7	8
Overall delay	0-8	40
Slip delay	0-8	80
Alpha delay	0-8	30

The response delays were measured as the time between the command signal and an observable pressure change. Note that the pedal state delay is measured from the instant at which the shuttle valve registers that the pedal is being actuated by the driver and the instant at which a change in pressure can be observed. The “overall”, “slip” and “alpha” delays were measured from test data as an initial guess and tuned in simulation until the respective quantities from the simulation results, such as slip and angular acceleration, showed good correlation with test data. As mentioned before, the use of the “dump-hold” alternating states during phase 3 is different than the classical Bosch algorithm, and is used in this study because of a slow acting controller. The Simulink model shown previously in Figure 3 is now expanded to include the delays as measured from the test program to produce Figure 6.

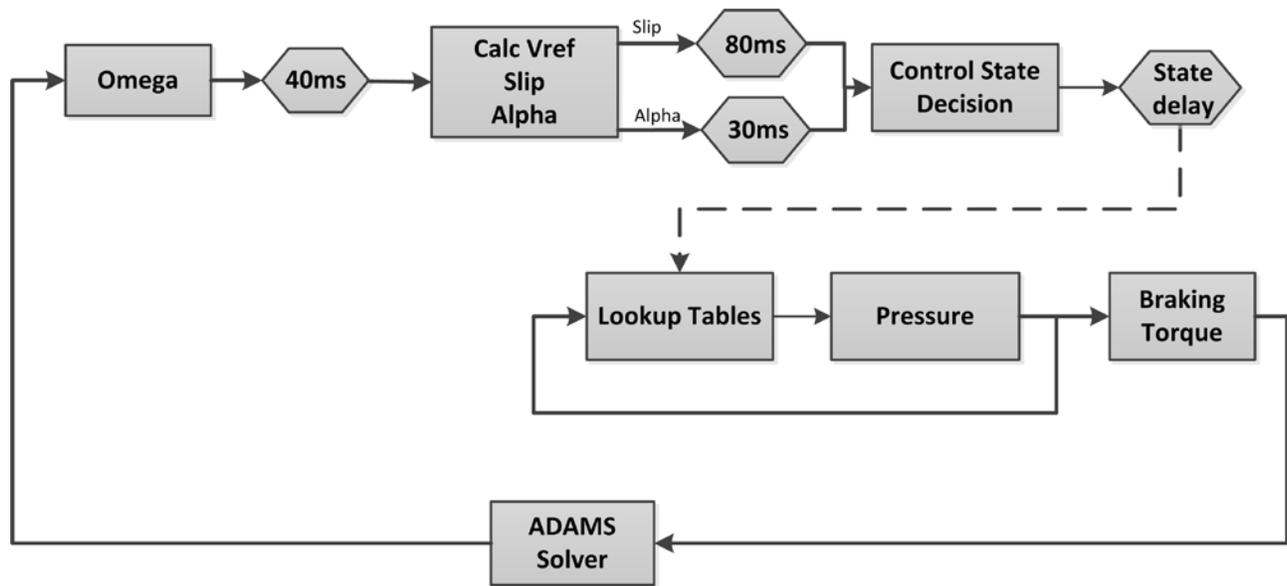


Figure 6: Simulink model expanded to include response delays

Considering Figure 6, it can be seen that the combined delays of 40ms and 80ms for the slip quantity will result in the controller reacting a total of 120ms too slow during the 3rd Bosch phase and the controller will decrease the pressure far below what is necessary to bring the wheel to acceptable slip levels. Thus, the use of the “dump” state in phase 3 resulted in very long stopping distances and the implementation of the “dump-hold” alternating states was used as a countermeasure.

4.2) Pressure to torque

The linear relationship between braking pressure and torque requires geometry and an accurate estimation of the friction coefficient between the brake pad and the disk. Budynas & Nisbett, (2008) motivates the use of the uniform wear friction model, and is often used in braking simulations. However since both the braking torque and pressure can be accurately measured on the test vehicle, the linear relationship is experimentally determined for use in the simulation model, thereby eliminating the need for accurate friction coefficient estimation. Figure 7 shows brake torque, measured with the WFT, as a function of pressure that was recorded during an ABS braking manoeuvre on a flat road.

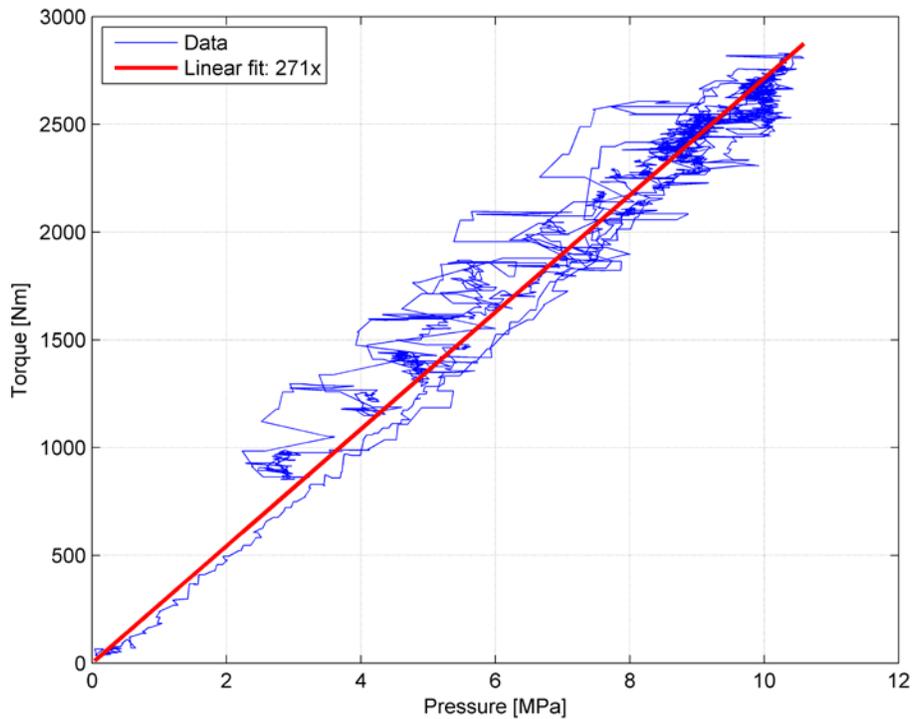


Figure 7: Torque vs. pressure and linear fit as recorded during ABS braking manoeuvre

A least squares linear fit through the origin provides a gain that is used in the Simulink model. The scatter of the data around the trend line may be attributed to several factors, such as non-ABS callipers that are used on the test vehicle that may retain latent pressure, and an increase in temperature that introduces non-linearities in the system. Since the wheel will lock up at a minimum pressure value, any increase beyond this value will skew the trend, as the linear relationship pressure vs. braking torque is no longer valid. The horizontal lines lying above the trend line in Figure 7 are evidence of this.

5). Validation

Perhaps the most important validation step of this study is the pressure lookup modelling, since this portion of the model will influence all other correlation. The method used here is to use data from a test run performed on flat terrain and replay the same pressure control commands to the simulation model. Figure 8 shows the pressure from the test data and the returned simulation pressure.

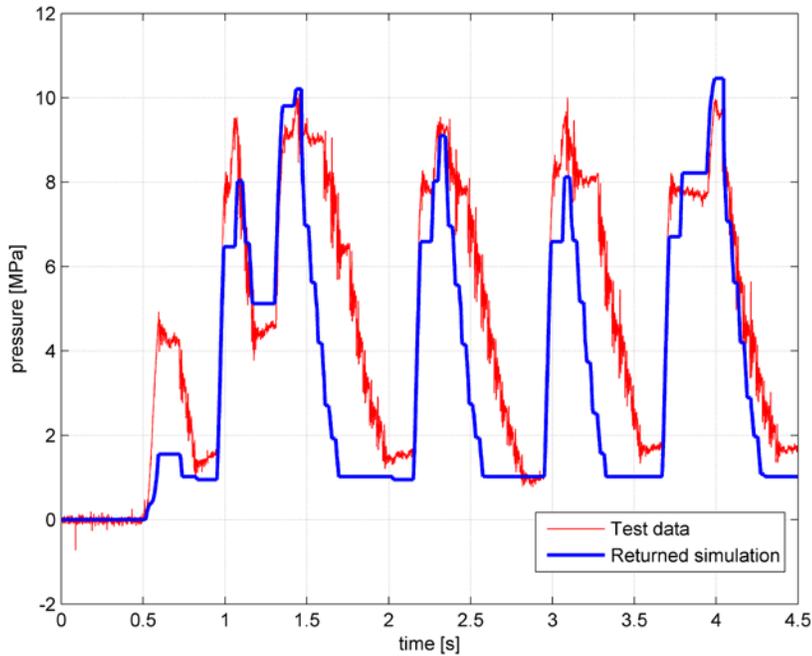


Figure 8: Validation of the pressure look-up portion of the simulation model

As can be seen, the test data deviates from the returned simulation results during the dump-hold states, but correlates well with the pump states. It seems as if the modulator lags somewhat during the start of the dump-hold states for the three middle control cycles, but correlates well with the last control cycle. Thus, it may be concluded that the pressure lookup tables are accurate, but the response delays seem to be inconsistent with test data. This inconsistency can be attributed to the hardware used, such as the old hydraulic modulator or the electro-mechanical relays that switches inconsistently.

The INS provided true vehicle velocity to be used for the validation of the reference velocity calculated by the ABS algorithm. To ensure the algorithm remains accurate while braking on rough terrain, a brake test with ABS deactivated was done on the Belgian paving, since this will provide the worst case scenario for reference velocity calculation. The deactivation of ABS involved disconnecting the relay box that switches the hydraulic solenoids to prevent pressure modulation, but leaving the controller active to calculate all the required quantities for its control making logic. This allowed the reference velocity to be calculated by the ABS algorithm, and is plotted together with the measured INS velocity in Figure 9.

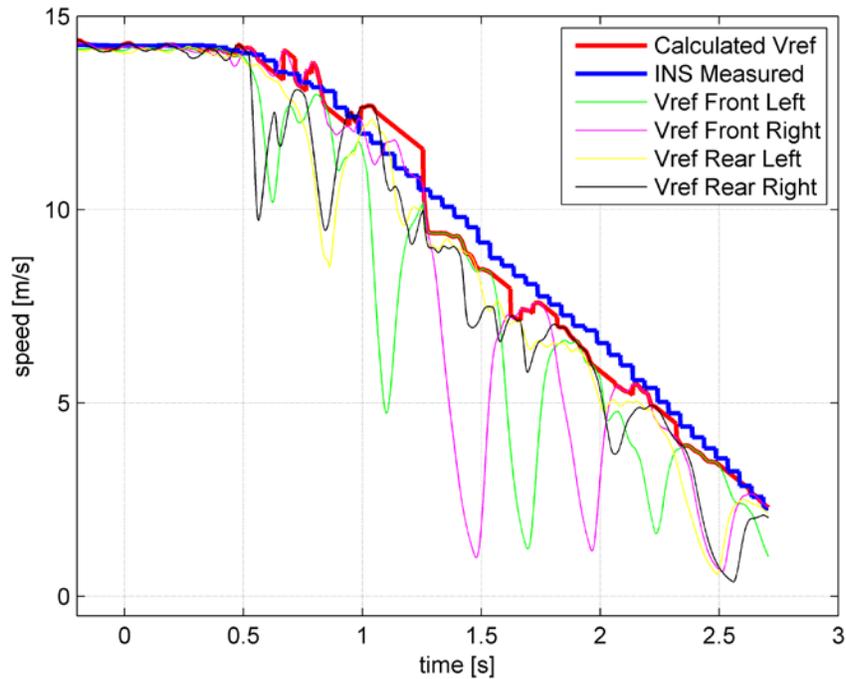


Figure 9: Validation of the reference velocity portion of the simulation model

The recorded angular velocities, multiplied with the constant rolling radius, are also included in Figure 9. From the figure, it may seem as if the ABS controller performed the necessary actions to prevent the wheels from locking. However, since pressure control was deactivated, the rapid variation in wheel speed is purely due to road excitation, which leads to severe roll and pitch motions with varying load transfer as an effect, and may even lead to loss of wheel contact. These phenomena explain exactly why ABS performance deteriorates on rough terrains.

Figure 9 provides a clear illustration of how the largest V_{ref} quantity is taken as the reference velocity of the vehicle by the ABS algorithm while the wheels that are close to lockup are rejected. The flaws in the estimation can also be seen however, as in the case of the two small bumps between 0.5s and 1.0s, where higher than actual wheel speeds, perhaps from sensor noise or vibration, are chosen for V_{ref} calculation. The straight V_{ref} line segment between 1.0s and 1.5s shows how the algorithm detects that no speed sensor provides accurate information and uses the linear deceleration formula to calculate V_{ref} . In conclusion, although somewhat crude and simple, it can thus be seen that the ABS algorithm calculates an accurate reference velocity when compared to the measured INS velocity, with a maximum absolute relative error of 18.4% and a mean of 5.6%.

6). Results

For the purpose of this study, the hydraulic modulator was set to not allow any pressure to the rear wheels, thereby ensuring that the reference velocity of the vehicle could be accurately calculated from the free rolling wheels. Thus inaccuracies arising from the longitudinal slip calculation will not play a role and the ABS algorithm can be evaluated in isolation.

6.1) Flat road

The simulation model is first validated on a high friction flat road. Minimal road disturbances are present to provide a benchmark from which the performance can be evaluated. The driver of the test vehicle performed a panic stop at a speed of 60km/h and kept the vehicle as straight as possible during the manoeuvre. Figure 10 shows the test data vs. the simulation results for the front left wheel;

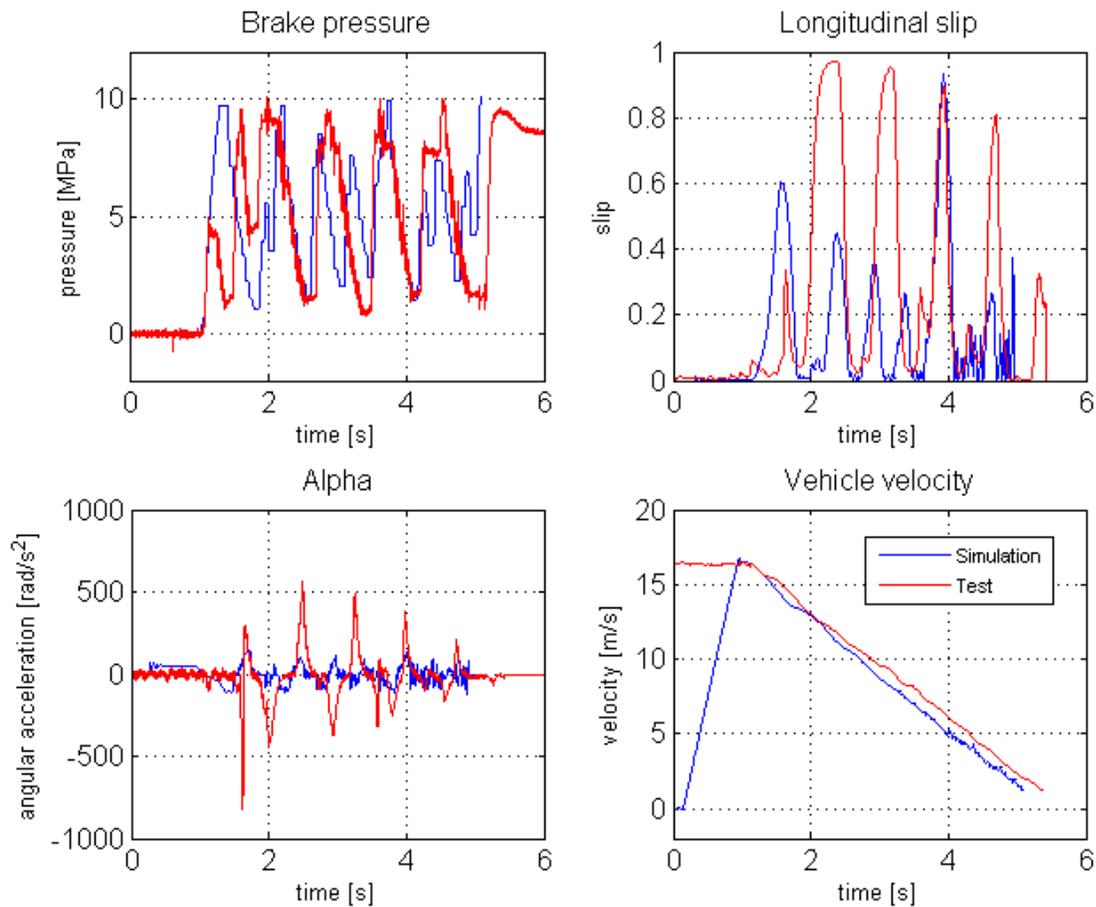


Figure 10: Simulation results and test data for flat road ABS

The stopping distance as determined by simulation is 33.29m while the test resulted in a slightly longer distance of 34.04m. Pressure, slip and vehicle velocity show good correlation, while angular acceleration shows a much higher magnitude in test data than with simulation. This may be attributed to uncertainties in the frequency to voltage converter that cannot be measured and hence not accurately modelled.

For both test data and simulation, slip and angular acceleration are far beyond the set thresholds. This is due to a lagged response from the controller which leads to the severe overshoot. Reducing the thresholds will not remedy the retarded response, as angular acceleration noise fluctuates close to the current thresholds as it is. The dump-hold state of phase 3 is by far the state in which the controller spends the most time during the ABS cycle and it may seem that a steeper dump-hold gradient might solve the overshoot issue. The use of the dump state alone, as opposed to the stepwise dump-hold, for phase 3 solved the overshoot issue to some extent, but proved to yield even worse stopping distances. This is once again because the controller failed to hold pressure at phase 4 due to the slow response and the pressure was dumped to zero. It is particularly challenging for the pump to restore pressure from zero which lead to other complications in the control cycle which eventually resulted in the large stopping distances.

6.2) Belgian paving

Correlation of ABS on rough terrain is the aim of this study and is discussed from this point forward. A panic stop is performed at a speed of 60km/h on the Belgian paving and the vehicle was brought to a complete stop. Figure 11 shows the recorded data for the front left wheel;

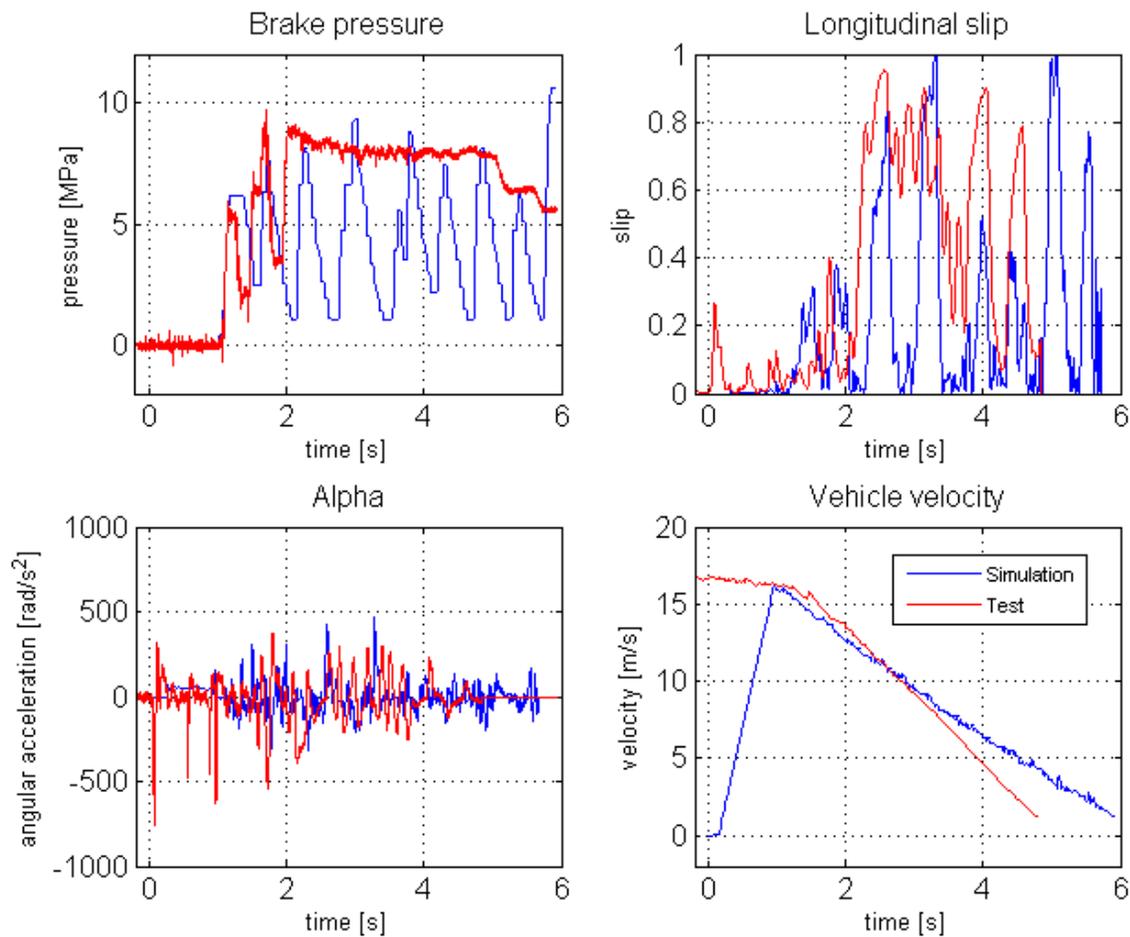


Figure 11: Simulation results and test data for Belgian Paving ABS

Simulation results produced a stopping distance of 39.91m while test results yielded a far shorter distance of 31.57m. This large discrepancy in stopping distance is attributed to the difference in brake pressure, as seen in Figure 11. Multiple tests confirmed that the controller remains in the 3rd Bosch phase to counter excessive slip, but no pressure is seen to be released. Although advantageous for shortest braking distance, it defeats the purpose of ABS and is likely attributed to angular acceleration noise, arising from noisy wheel speed sensors, interfering with the controller's logic.

7.) Conclusion

A test vehicle was prepared with self-installed Anti-lock Braking System to research the deterioration of ABS performance on rough terrain. The Bosch ABS algorithm was implemented in C-language on an on-board computer to control an off-the-shelf hydraulic modulator by measuring wheel speed and calculating the

reference velocity of the vehicle. Pressure sensors and an INS provided the means to evaluate braking performance and validate with the simulation model.

A data-driven simulation model using pressure lookup tables and response delays was built from test data. The detailed mechanics of the Bosch algorithm and the implementation thereof in the model was explained. The validation of the pressure look-up portion of the simulation model showed poor correlation for the alternating states, but adequate correlation otherwise. Using the INS velocity as the true vehicle speed, it was seen that the reference velocity calculation is accurate even on rough terrain with ABS deactivated.

Since the scope of this study is concerned with rough terrain braking performance, an ADAMS model is used to compare simulation results and test data for both flat terrain and the Belgian paving. The result of this comparison was not as successful as anticipated and poor correlation was observed for pressure and ultimately braking distance. However, insight was gained to why the performance of ABS deteriorates on such terrains and the proposed solutions from literature can be evaluated on the test platform with the aid of the simulation model.

8.) Future work and recommendations

The developed test platform and simulation model provides new opportunities for optimising anti-lock braking algorithms on various terrains. Combined suspension and ABS control strategies can also be developed and evaluated on the vehicle.

The correlation between simulation results and test data is not yet as desired and more attention should be given to the accuracy of the tyre model and the method of pressure lookup. A more advanced method of estimating the velocity of the vehicle, **such as the use of individual wheel centre velocities as separate reference velocities together with a Kalman filter for velocity estimation**, should be considered to increase the accuracy of the calculated longitudinal slip. Hardware upgrades, such as Original Equipment Manufacturer (OEM) wheel speed sensors and solid state relays, might aid with the malfunctioning of the controller due to noisy wheel speed signals and slow controller response respectively. Since control in both simulation and test vehicle is still performed with classic logic, a more advanced supervisory control method should be considered, such as Matlab's Stateflow toolbox.

9.) Nomenclature

Symbol	Description	Unit
a_0	Linear deceleration	[m/s ²]
r	Rolling radius of tyre	[m]
R	Assumed deceleration constant (9.81)	[m/s ²]
v	Reference velocity	[m/s]
Δt	Time step	[s]
α	Angular acceleration	[rad/s ²]
μ	Friction coefficient	[]
ω	Angular velocity	[rad/s]

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