POLITECNICO DI TORINO

Master’s Degree in Automotive Engineering

Master’s Degree Thesis

OPTIMIZATION OF BRAKING DISTANCES BY MEANS OF INTERVENTIONS ON THE TIRE AND ON THE ABS CONTROL

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April 2019

Dedication

I dedicate this work to my parents Beatriz Helena de Resende Barbosa Cordeiro and José Francisco Cordeiro that always believed in my potential and always supported my dreams.

They taught me the importance of hard work and respect to others.

Preface

Thesis presented at Politecnico di Torino to conclude the Automotive Engineering Master of Science course as part of a Double Degree Program – with the Universidade Federal De Pernambuco – UFPE.

The course was held between the Academic Years of 2017, 2018 and 2019. The thesis presentation was carried out in April 2019.

This work was oriented by Professor Massimiliana Carello from the Department of Mechanic and Aerospace Engineering (DIMEAS) with the co-orientation of the engineer Paolo Massai.

Turin, 2019-04-04

(Raquel Barbosa Cordeiro)

Acknowledgements

I would like to thank my Maserati tutor Alessandro Capobianco for all the knowledge shared with me from the first until the last day of internship.

To Luca Dusini, Fabio Gerbino, Mariano de Rosa, for making the Maserati team of 'Vehicle Dynamics and Calculations' the ideal team for the development of this thesis.

To the Breaks team of Maserati, for sharing experimental data and technical information.

To Maserati Chassis department, for the coworking between teams.

To Maserati s.p.a – Modena to be the ideal environment to intellectual and personal development.

To my Politecnico di Torino supervisor Massimiliana Carello for believing in my work and helping me during all the thesis.

To my Politecnico di Torino co-supervisor Paolo Massai for helping me making important career and life decisions.

To my mother Beatriz Helena de Resende Barbosa Cordeiro, a strong woman as I dream to be.

To my father José Francisco Cordeiro, that taught me the importance of family.

To my brother Guilherme Barbosa Cordeiro that always have been my best friend.

To my grandmother Beatriz Resende Barbosa for always putting me in her prayers.

To my long-standing friends, that always showed support even with the long distance.

To the friends I met while living abroad (Torino and Modena) for being my family during those years.

Index

1. Introduction

The following thesis was developed by an automotive engineering student of Politecnico di Torino in Maserati-Alfa Romeo in Modena – Italy during an internship of six months.

Since the brake system is like a black box to car makers that buy it from suppliers, the aim of this thesis is to create a model of Integrated Brake System on Matlab-Simulink that connects to the multibody model of CarRealTime, in order to simulate in a virtual environment various brake maneuvers in various surfaces.

Then compare the simulation results with real data generated in experimental hard brake maneuvers provided by Maserati – Alfa Romeo in different road friction coefficients, for two car models that utilize the Integrated Brake System - IBS - technology (figure 1.1):

- AR949 – Stelvio
- AR952 – Giulia

The comparison done between simulation results and experimental data is done by means of

- Vehicle longitudinal acceleration;
- Estimated wheel speeds;
- Estimated vehicle speed;
- Brake pressure at the 4 corners;
- Main system pressure.

And the braking distance is presented.

Figure 1.1 - MK C1 for FIAT

Then performed sensitivity analysis on the simulations parameters (brake system and tires) in order to understand the influence of those variables in the braking behavior and be able to predict which subsystem modify in order to improve the performance of braking distance.

Structure of the Thesis:

- Chapter 2 presents the CarRealTime model description used in the simulations, and how systems and subsystems are set;
- Chapter 3 describes theoretical characteristics of rim and tires;
- Chapter 4 presents the functions of braking system and braking dynamic performance (in ideal and actual braking);
- Chapter 5 presents the Bosch ABS braking system;
- Chapter 6 presents the Continental's IBS system, that is used in the cars that generated the experimental data.
- Chapter 7 describes the maneuvers realized in the experimental maneuvers and the ones reproduced in the simulations.
- Chapter 8 presents the cosimulation logic realized on the Matlab-Simulink environment. Describes the procedures of obtaining the brake pressure on the four corners calipers based on the brake pedal stroke input;
- Chapter 9 presents the validation of the model by means of the correlation of simulation results with experimental data, and the tuned simulation parameters in order to obtain this correlation;
- Chapter 10 presents the sensitivity analysis results, with percentual variation of parameters and its consequence on braking distance.
- Chapter 11 presents the study conclusions;
- Chapter 11 shows recommendations to further work;
- Chapter 12 presents the list of references used in the construction of the study.

2. Vehicle Model Description

The vehicle model used in this thesis is the one of VI-CarRealTime that will be described below.

2.1CarRealTime model

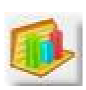
The VI-CarRealTime model and simulation environment targeted to a simplified four wheels vehicle model. Its functionalities include the ability to assemble the vehicle system by collecting its fundamental subsystems, specifying dynamic maneuver schedules, launching standalone or Matlab-Simulink embedded simulations, post-processing the obtained results [10].

The environment based on underlying solver consists of:

- Symbolically derived parameterized equations of motion;
- Pacejka tire model;
- Virtual driver model.

In CarRealTime environment the user can track over 900 outputs variables during the simulation, and can plot graphically animate the simulation results.

The CarRealTime framework is divided into three work models:

- Build 
  The Build mode allows you to edit model data and change system configuration. The Build mode is the default for starting VI- CarRealTime.
- Test 
  The Test mode allows you to run the model.
- Review 
  The Review mode allows you to visualize analysis results using either VI-Animator or Adams/Postprocessor. You can postprocess the output of standard VI-CarRealTime events. Postprocessing has two formats: animation and plots.

2.1.1Build Mode

VI-CarRealTime Build mode lets you edit model data and change system configuration (figure 2.1).

The Build button in the menu toolbar allows to:

- Load model:
- Load Example Full Vehicle Model
- Register example Databases

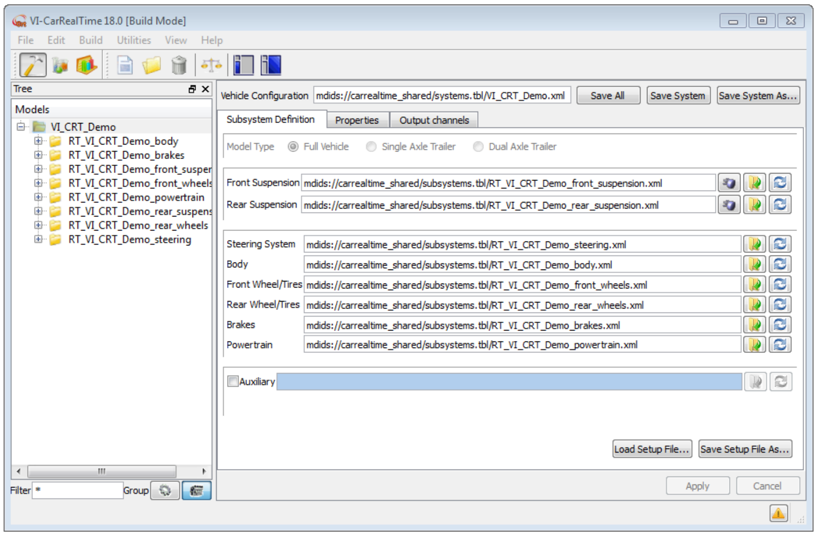


Figure 2.1 – CarRealTime build mode

2.1.1.1Vehicle Model

VI-CarRealTime contains a simplified model of a four-wheeled vehicle. The vehicle is a reduced degree-of-freedom (DOF) model. It includes only 14 degrees of freedom distributed as follows.

The model includes 5 rigid parts (figure 2.2):

- Vehicle chassis (sprung mass)
- 4 wheel parts(unsprung masses)

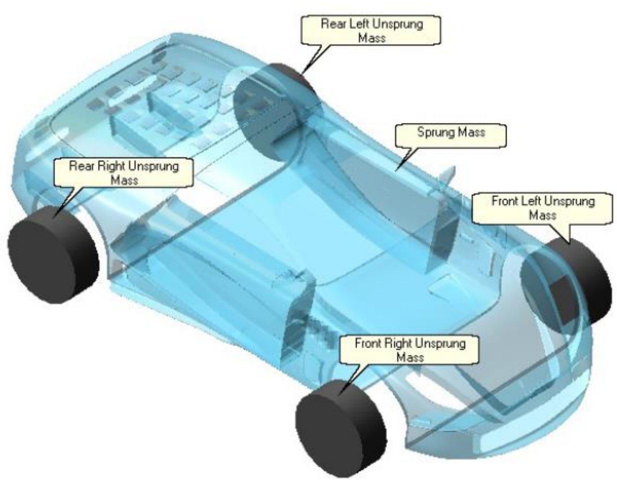


Figure 2.2 - CarRealTime vehicle simplified model

The vehicle has 6 DOFs while parts have 2 DOFs each (one for the motion with respect to the vehicle body and the other wheel spin).

The VI-CarRealTime suspension does not have linkages or bushings, and the steering system does not have parts for the steering wheel or rack.

Suspension and steering system properties (kinematic, compliance and component data) are described by lookup tables using a conceptual approach.

For each suspension the possibility of considering an additional stiffness in series to the main spring is supplied (further 4 DOFs when the feature is activated for all the suspension) – but this was not considered in this thesis.

Body chassis torsional compliances may also be included (up to 6DOFs) so the total vehicle DOFs increases to 20 when all compliances DOFs are active – but this was not considered in this thesis.

For each wheel is possible to enable an additional longitudinal degree of freedom (further 4DOFs when the feature is activated for all the wheels) – but this was also not considered in this thesis.

Other vehicle subsystems (brakes and powertrain) are described using differential and algebraic equations, so no extra part is present in the model.

The vehicle model's intent is to accurately predict overall vehicle behavior for cornering, braking, and acceleration-performance studies for four-wheeled vehicles with independent-front and independent-rear suspensions. The simplified model is described in terms of commands and functions that use an internal development environment working as a symbolic manipulator tailored for deriving multibody equations and a code generator. Given the model description, it outputs C code for a simulation program for the particular model that is described. Parameters for the model (for example, wheelbase, spring stiffness, etc.) can be changed at run time and are passed in by input files.

Convention and references

Coordinate systems

The reference frames and coordinate systems used in VI-CarRealTime are shown in the figure 2.3. These coordinates systems are consistent with the following standards:

- ISO 8855, Road vehicles – Vehicle dynamics and road-holding ability – Vocabulary. 1991
- SAE Recommended Practice J670f, Vehicle Dynamics Terminology.

Global Reference Frame

The root body of the tree representing the rigid-body system is the global frame. The body is a Newtonian or inertial frame, which means that it does not accelerate in translation and does not rotate.

The global inertial frame of reference, N (N denotes Newtonian), for VI-CarRealTime models has unit vectors n_x, n_y, and n_z, and origin point N_0, as shown in the figure below.

Vehicle Reference Frame

The vehicle model is positioned with respect to the global frame.

The origin of the Vehicle Reference System S_0 is located at Z=0 of Global Reference Frame and at half front of the vehicle track (figure 2.3).

The triad is oriented, at design time, as the Global Reference Frame with:

- X+ axis pointing forward in the direction of motion
- Y+ axis pointing leftward
- Z+ axis pointing upward

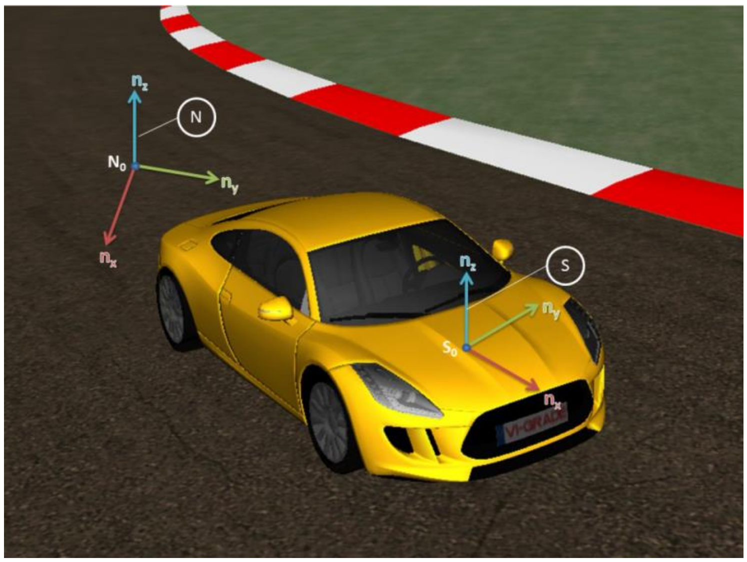


Figure 2.3 - CarRealTime reference frames

Wheel Location Reference System

It is defined by a triad positioned at the wheel center (left and right, at design time); the orientation is the same as VI-CarRealTime global references system with X+ forward, Y+ left and Z+ vertical up.

Driver Location Reference System

Is defined by a triad positioned on the rear axle midpoint; the orientation is the same of the vehicle reference system.

Road Reference System

It is defined by a triad having the Z+ axis oriented as the road normal vector, the X+ axis as the vehicle forward direction and the Y+ axis accordingly.

Gyro Reference System

The gyro reference system (gyro maker) is located on the body (by default it is located coincident to the sensor point) and is oriented with the Z axis coincident with the global Z during the simulation with the X axis along the vehicle movement direction, so it follows the vehicle displacement, follows the yaw but not the pitch of the vehicle.

Suspension Reference System

It is defined by a triad positioned in the hub center (same as wheel center); the orientation is defined by camber/toe/caster angles.

Model Configuration

The definition of the vehicle model in VI-CarRealTime is realized using the system files (figure 2.4), having xml format and stored in vehicle database under system.tbl table.

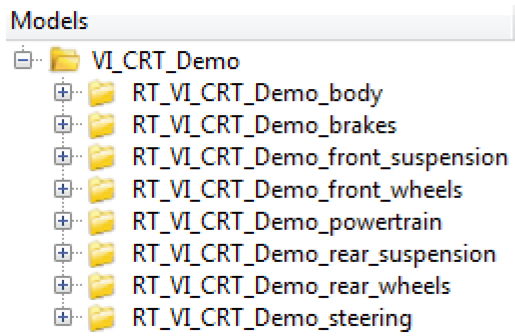


Figure 2.4 -System file

The system file contains the following set of information:

- Units
- List of vehicle subsystem files:
  - Front Suspension
  - Rear Suspension
  - Steering
  - Body
  - Powertrain
  - Front wheel and tires
  - Rear wheel and tires
  - Brakes
  - Auxiliary Subsystem

The property file editor can be used to select appropriate property file for each of the subsystems (figure 2.5).

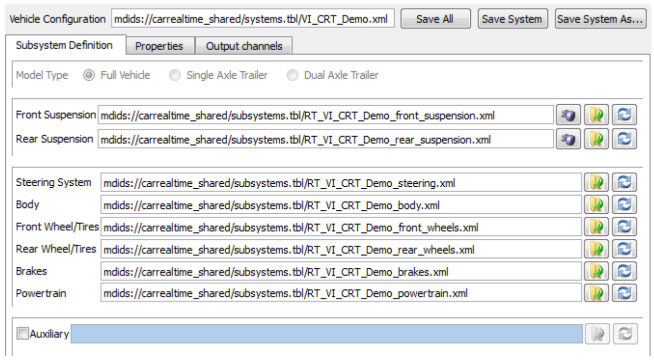


Figure 2.5 - Subsystem definition

It is possible to exchange subsystems among different vehicle models by selecting them in the system property editor. It is important to notice that the those subsystems can be substituted to external logics or subsystem created in other software, as illustrated on the figure 2.6.

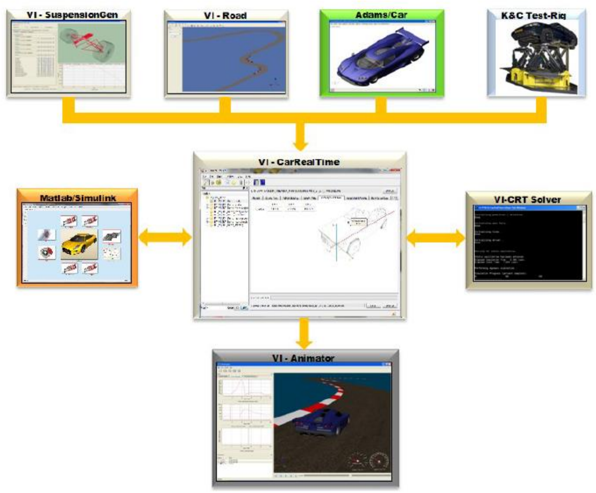


Figure 2.6 - CarRealTime associated with other software

Vehicle Properties

Vehicle system files contain a set of information beyond the list of the subsystems. The auxiliary information stored in the system files can be edited using the property editor (figure 2.7).

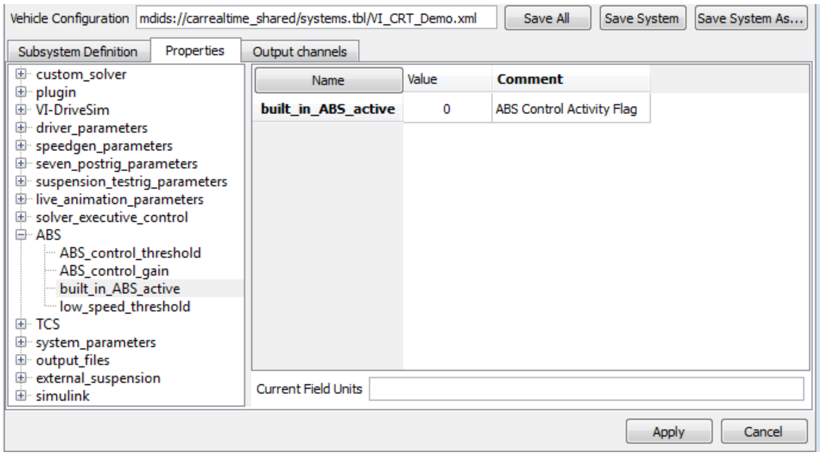


Figure 2.7 - CarRealTime vehicle properties

As in this study the ABS is reproduced on Matlab-Simulink, the ABS from CarRealTime is deactivated so that its behavior can be calculated and simulated on Matlab-Simulink. And all braking functions are deactivated from CarRealTime by means of CarRealTime block input on Matlab-Simulink so they can be recreated.

<u>Body Subsystem</u>

The body subsystem properties collect mass, inertia, setup and accessory information about the sprung mass part of VI-CarRealTime.

<u>Sprung mass</u>

The sprung mass Property Editor (figure 2.8) access the sprung mass inertia, property of vehicle chassis. It also contains the reference to the Adams command file for vehicle model animation in Adams/PPT, the vehicle wheelbase and chassis compliance data.

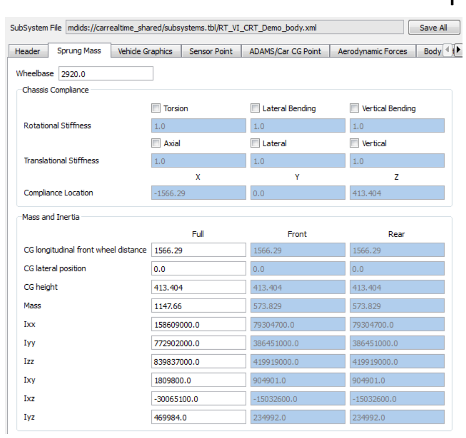


Figure 2.8 – Sprung mass property editor

<u>Wheels Subsystem</u>

Wheels subsystem properties collect mass, inertia, tire property file information of unsprung mass pairs of VI-CarRealTime vehicle model.

Wheel subsystem Property Editor (figure 2.9) includes the following data:

- Tire Property File

property file for tire forces computation.

- Spin Inertia (one wheel)

it sets the spin inertia of the wheel.

- Ixx (hub carrier + wheel)

Ixx moment of inertia of unsprung mass. It is measured in the Wheel Location reference frame.

- Iyy (hub carrier + wheel)

Iyy moment of inertia of unsprung mass. It is measured in the Wheel Location reference frame.

- Izz (hub carrier + wheel)

Izz moment of inertia of unsprung mass. It is measured in the Wheel Location reference frame.

- Unsprung Mass: it includes the mass of a single wheel and the portion of suspension masses not belonging to sprung part.
- Wheel Center Height: it is the vertical position of wheel center expressed in Global Reference Frame.

Note: Cross moment of inertia of unsprung mass are neglected.

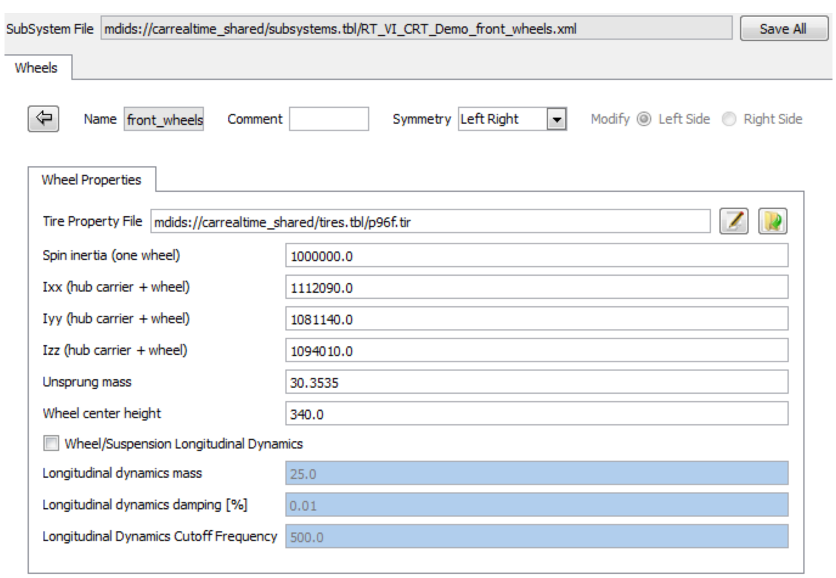


Figure 2.9 - Wheel Subsystem Property Editor

Brakes Subsystem

The VI-CarRealTime vehicle model allows open or closed-loop brake torques to be specified at the wheels. The brake model in the vehicle model represents a four-wheel disk brake configuration.

All brake parameters associated with the brake on one wheel can be specified independently. The low-speed brake model is quite robust neat zero speed. This allows the vehicle to reasonably approximate coming to a dead stop.

The brake model includes the following parameters:

- master_cylinder_pressure_gain: is the scalar relating the driver brake demand to master cylinder pressure;
- mu (front/rear): friction coefficient;
- effective_piston_radius (front/rear): radius for applying friction force;
- piston_area (front/rear);
- lockup_natural_frequency (front/rear): the brake model includes a 1DOF spring-damper system to model the wheel lockup; the parameter sets system natural frequency;
- lockup_damping_ratio (front/rear): the parameter sets lockup model damping.
- lockup_speed (front/rear): the parameter sets the lockup speed.

2.1.1 Test Mode

In the VI-CarRealTime test mode simulation evens can be created and run.

For this thesis simulations, DrivingMachine events were used. For the events some parameters have to be chosen, as:

- Vehicle model;
- Driving machine file;
- Road data File;
- Integration time;
- Mode of Simulation.

The vehicle model is the one defined at the build mode.

The driving machine file (figure 2.10.a) is composed by mini maneuvers (figure 2.10.b) that form a more complex maneuver. Inside each mini maneuver are set the drive's decisions as:

- Steering;
- Throttle;
- Braking;
- Gear shift;
- Clutch.

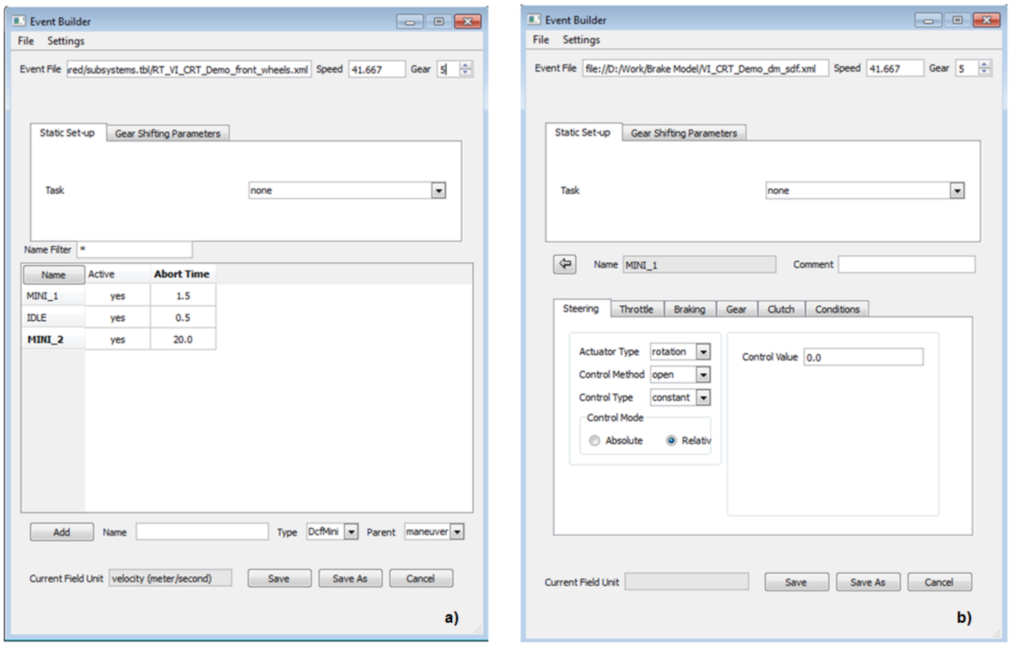
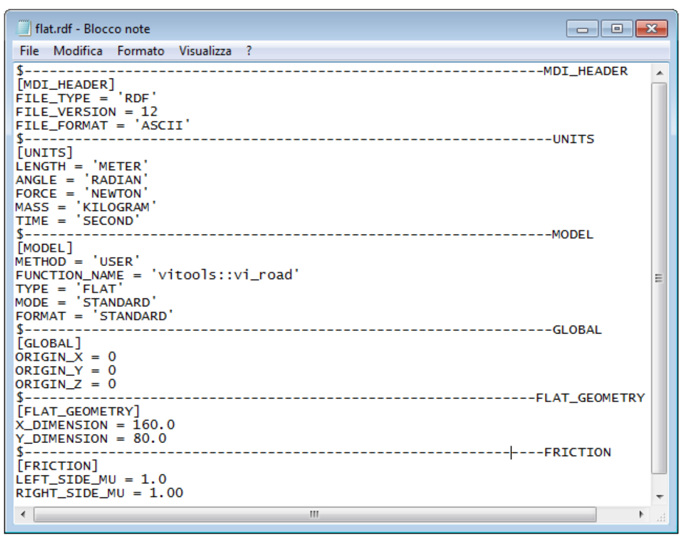


Figure 2.10- (a) Driving Machine file; (b) Mini maneuver

The road data file (figure 20.11) describes the road conditions, where in this thesis is useful to set the road friction coefficient of left and right side.




```
$--------------------------------------------------------------MDI_HEADER
[MDI_HEADER]
FILE_TYPE = 'RDF'
FILE_VERSION = 12
FILE_FORMAT = 'ASCII'
$-------------------------------------------------------------------UNITS
[UNITS]
LENGTH = 'METER'
ANGLE = 'RADIAN'
FORCE = 'NEWTON'
MASS = 'KILOGRAM'
TIME = 'SECOND'
$-------------------------------------------------------------------MODEL
[MODEL]
METHOD = 'USER'
FUNCTION_NAME = 'vitools::vi_road'
TYPE = 'FLAT'
MODE = 'STANDARD'
FORMAT = 'STANDARD'
$------------------------------------------------------------------GLOBAL
[GLOBAL]
ORIGIN_X = 0
ORIGIN_Y = 0
ORIGIN_Z = 0
$-----------------------------------------------------------FLAT_GEOMETRY
[FLAT_GEOMETRY]
X_DIMENSION = 160.0
Y_DIMENSION = 80.0
$----------------------------------------------------------------FRICTION
[FRICTION]
LEFT_SIDE_MU = 1.0
RIGHT_SIDE_MU = 1.00
```

Figure 2.11 - Road data file

3. Wheels And Tires

Since wheels characteristics have an enormous influence on the braking distance, in this chapter some rim and tires definitions and characteristics are shown.

The functions of the wheels are:

- To support the vehicle weight by exchanging vertical forces with the road surface;
- To exchange with the road surface longitudinal and side forces, allowing the vehicle to follow the desired path [4].

The importance of the capacity of exert longitudinal and side forces becomes even more important with increasing speed.

Wheels are composed by two elements: rim and tire.

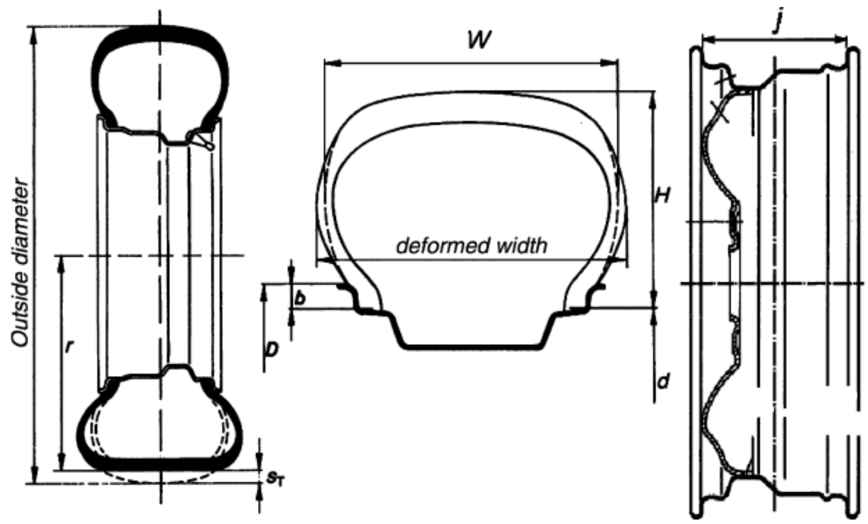


Figure 3.1 - Section of a complete wheel including rim and tire.

3.1 Rim

The main characteristic of the rim is to allow an easy and fast substitution of the tires. It has standardized dimensions in order to allow to be exchanged with tires of other manufacturers.

The wheel is made of a disc and flange that are usually integral, being the wheel fixed to its hub with bolts.

Rims can be done with stamped and welded steel or cast aluminum or magnesium. The rim contains some windows that allows an improvement in brake cooling and by a central hole able to sustain the wheel before the bolts are tightened.

The rim presents a bell shape in order to make possible the correct positioning in relation to the symmetry plane of the tire with respect to the car and to create a space that is necessary for hub and brake installation.

3.2 Tire

Around the rim there is a flexible structure and its tube that are used to maintain inflation pressure. Nowadays, due to safety reasons, the tubeless tires are preferred. So the tire is hermetically fitted to the wheel.

The tire is a composite structure made out of many layers of rubberized fabric (plies) with reinforcements cords.

Each tire is designated by a group of numbers and letters, as in the example:

175/65 R 14 82 T

The figure 3.1 is used in order to explain the meaning of this designation:

- The first figure (175) shows the dimension W, measured in mm. Since the tire is a deformable structure, this measure must be referred to an undeformed condition with no load applied and correct inflation.
- The second figure (65) is the aspect ratio of the tire, given by the ratio H/W between radial height and the width. Aspect radios are expressed as percentage. If this number is omitted, it can be considered H as being 80% of W.
- The following letter shows the type of tire plies. R stands for radial. Otherwise the designation is omitted.
- The third figure represents the rim diameter, measured in inches.
- The fourth figure is the load factor [9], determines the admissible vertical load at a certain inflation pressure (Appendix A).
- The last letter indicates the maximum allowed speed for the tire, as shown in the table 3.1.

Speed(km/h)	80	130	150	160	170	180	190	210	240	270	300
Letter	F	M	P	Q	R	S	T	H	V	W	Y

Table 3.1 – Letters showing the maximum allowed speed of a tire

The tires used on experimental data and on simulation were:

- AR 949 – Stelvio: 235/60 R 18;
- AR952 – Giulia: 225/50 R 17.

3.2.1 Tire operation

Taking in consideration that the tire can be in prepared ground (paved or concrete surfaces) or on unprepared ground (natural surfaces or dirty ground), the only thing that changes in the analysis is the deformation of the ground.

Two aspects on on-road driving are considered:

- The elasticity on the tire structure that gives to the tire absorption capabilities.

- The adhesion between rubber and ground, that allows ground forces. The adhesion is caused by physical adhesion and local deformation.

The fiction coefficient μ is defined as the ration between the lateral force and the vertical force; It can be broken down into a longitudinal friction coefficient :

$$\mu_x = \frac{F_x}{F_z} \tag{3.1}$$

and the transversal friction coefficient:

$$\mu_y = \frac{F_y}{F_z} \tag{3.2}$$

3.2.1.1 Longitudinal Force

Considering a pneumatic wheel rolling on level road. If a braking moment M_b is applied to it, the resulting distributions of normal pressure and longitudinal forces are represented in the figure 3.2.a. The thread band is stretched circumferentially in the zone that precedes the ground contact, while in free rolling the same part of the tire is compressed.

The peripheral velocity of the tread band in the leading zone of the contact $\Omega R_e'$ is by consequence higher than that (ΩR) of the undeformed wheel. The effective rolling radius R_e', whose value R_e in free rolling was between R_l and R, grows towards R and, if M_b is large enough becomes bigger than R.

The instantaneous center of rotation in consequently located below the surface of the road. The angular velocity Ω, also represented as w, of the wheel is lower than that characterizing free rolling in the same conditions ($\Omega_0 = {}^{V}\!/_{R_e}$). In such conditions it is possible to define longitudinal slip as:

$$\sigma = \frac{\Omega}{\Omega_o} - 1 = \frac{\Omega R_e - V}{V} \tag{3.3}$$

If the wheel is driving, instead of braking, the leading part of the contact zone is compressed instead of being stretched (figure 3.2.b). The value of the effective rolling radius R_e' is smaller than that characterizing free rolling and it is usually smaller than R_l; the angular velocity of the wheel is greater than Ω_0.

The slip defined in the Eq. (3.3) is positive for driving and negative for braking.

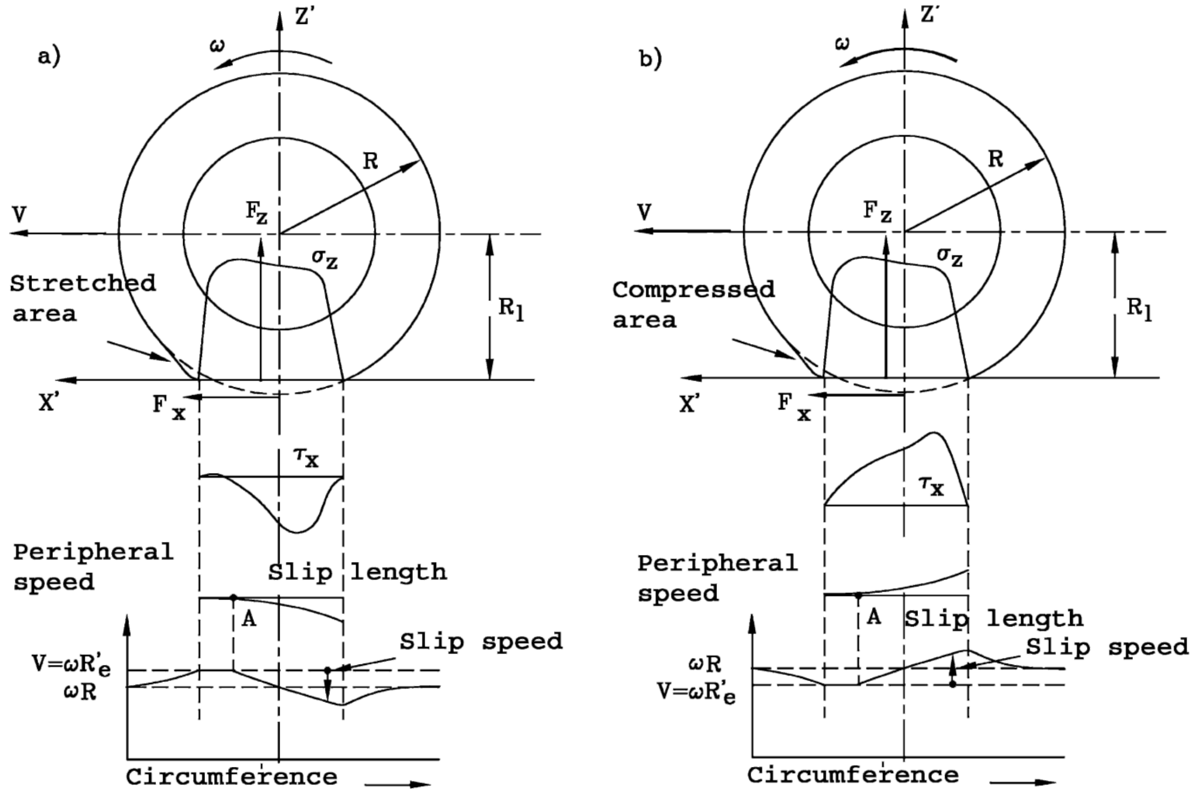


Figure 3.2 –Contact pressure distribution for braking (a) and driving (b) wheel.

The VI-CarRealTime model has it's tires based on the Pacejka's empirical model, also known as magical formula, is an excellent approximation of longitudinal force F_x as function of the slip σ. This mathematical expression allows to express forces F_x and F_y and the aligning torque M_z as functions of the normal force F_z, of the camber (ϒ) and the sideslip (α) angles.

The equation for the longitudinal force F_x as function of the slip σ is:

$$F_x = D\, sin(C\, arctan\{B(1-E)(\sigma+S_\mathrm{h}) + E\,\mathrm{arctan}[B(\sigma+S_\mathrm{h})]\}) + S_v \qquad (3.4)$$

Where B, C, D, E, S_v and S_h are coefficients that depend on the load F_z and on the camber angle. These obtained from experimental testing and have no direct physical meaning. Actually S_v and S_h were introduced to allow non-vanishing values of F_x when σ=0.

The D Coefficient yields directly the maximum value of F_x, apart from the effect of S_v. The product BCD gives the slope of the curve for σ+S_h=0. The coefficient values are expressed as functions of a number of coefficients b_i, which can be considered as characteristic of any specific tire, but depend also on the speed and road conditions:

$$C = b_0 \qquad\qquad D = \mu_p F_z \quad ,$$

Where a value of 1.65 is suggested and:

$$\mu_p = b_1 F_z + b_2, \qquad BCD = (b_3 F_z^2 + b_4 F_z)e^{-b_5 F_z},$$

$$E = b_6 F_z^2 + b_7 F_z + b_8, \qquad S_h = b_9 F_z + b_{10}, \qquad S_v = 0.$$

Is important to note that the product BCD is the slip stiffness of the tire.

If a symmetrical behavior for negative and positive values of force F_x is accepted, this model can be used for both braking and driving. The results obtained from it are expressed in the following units: F_z is in kN, longitudinal slip as percentage and F_x in N.

A group of curves $F_x(\sigma)$ obtained for vertical loads F_z = 1, 2, 3 kN for a radial tire 175/70 R 13 is shown in the figure 3.3. It can be noticed that the tire's longitudinal force is bigger on its modulus for bigger values of normal force.

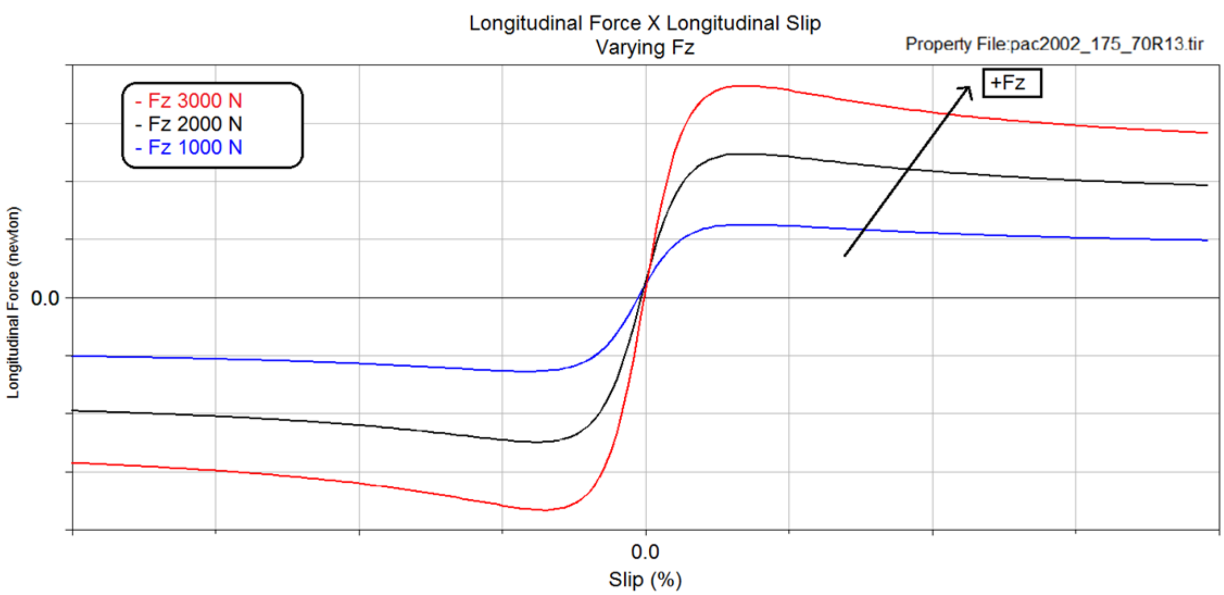


Figure 3.3 - Longitudinal force vs Longitudinal slip

4. Braking

Since the aim of this thesis is the study of the braking distance, in this chapter are presented the braking system tasks and the characteristics of dynamic braking, both on ideal conditions and on actual conditions, and in order to understand it better also the load distribution behavior is introduced.

4.1 Braking System

The braking system must accomplish the following tasks:

- To stop the vehicle completely, this function implies braking moments that are as strong as possible on the wheel [4];
- To control the speed, when the natural deceleration of the vehicle due to mechanical friction and motion resistance is not enough; this function implies braking moments on the wheels are moderate, but applied for a long time;
- To keep the vehicle still on a slope.

Because of these tasks the braking system is one of the safety systems of the vehicle. By consequence the State Authority introduce regulations that describe design conditions and operational requirements for this system.

Vehicle manufacturers and their components suppliers are responsible for regulation compliance of their products, including correct fabrication and system reliability for a period of time. Users, also, must play their role on the maintenance since many parts of this system are subjected to wear and the safety functions cannot be assured without the necessary maintenance and parts replacement.

4.2 Braking Dynamic Performance

4.2.1 Load distribution

Before talking about braking in ideal conditions is useful to talk about load distribution on the ground. And this will be done regarding driving in dynamic performance:

For vehicles with two axles, consider a vehicle as a rigid body symmetrical with respect to xy-plane, and neglecting the compliance of the suspensions and of the body, it can be modeled as a beam on two supports, and normal forces F_{z_1} and F_{z_2} acting on the axles can be computed easily.

With the vehicle at a standstill on level road the normal forces follow:

$$F_{z_1} = \frac{mgb}{l} \tag{4.1}$$

$$F_{z_2} = \frac{mga}{l} \tag{4.2}$$

The forces acting on a two-axle vehicle moving on road with longitudinal angle α (positive when moving uphill) are shown in figure 4.1.

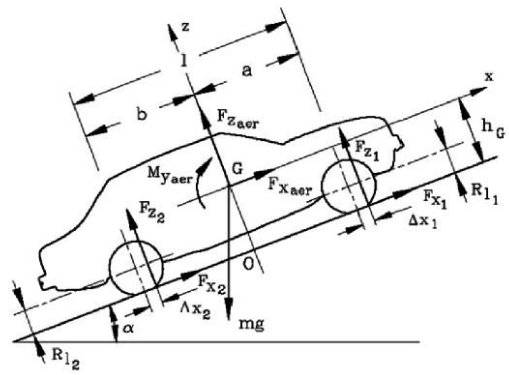


Figure 4.1 - Forces acting on a vehicle moving on an inclined road

Considering the inertia force acting in x direction on the center of mass, the dynamic equilibrium equations for translations in in the x and z direction and rotations about the point O are [5]:

$$F_{x_1} + F_{x_2} + F_{x_{aer}} - mgsin(\alpha) = m\dot{V} \tag{4.3}$$

$$F_{z_1} + F_{z_2} + F_{z_{aer}} - mgcos(\alpha) = 0 \tag{4.4}$$

$$F_{z_1}(a + \Delta x_1) - F_{z_2}(b - \Delta x_2) + mgh_G \sin(\alpha) - M_{aer} + |F_{x_{aer}}|h_G = mh_G\dot{V} \tag{4.5}$$

If the rolling resistance is atributed completely to the forward displacement of the resultant F_{z_i} of contact pressures σ_z, distances Δx_i can be easily computed as

$$\Delta x_i = R_{l_i} f = R_{l_i}(f_o + KV^2) \tag{4.6}$$

Apart of cases of vehicles with different wheels on the various axles, the values of Δx_i are all the same.

The Eq. (4.4) and Eq. (4.5) can be solved in the normal forces acting on the axles, yielding

$$F_{z_1} = mg\frac{(b - \Delta x_2)cos(\alpha) - h_G sin(\alpha) - K_1 V^2 - \frac{h_G}{g}\dot{V}}{l + \Delta x_1 - \Delta x_2} \tag{4.7}$$

$$F_{z_2} = mg\frac{(a+\Delta x_1)cos(\alpha) + h_G sin(\alpha) - K_2V^2 - \frac{h_G}{g}\dot{V}}{l + \Delta x_1 - \Delta x_2} \tag{4.8}$$

Where:

$$K_1 = \frac{\rho S}{2mg}\left[C_x h_G - lC_{M_y} + (b - \Delta x_2)C_z\right]$$

$$K_2 = \frac{\rho S}{2mg}\left[-C_x h_G + lC_{M_y} + (a + \Delta x_1)C_z\right]$$

Δx_i numbers are usually small and can be neglected. If considered, they introduce a further weak dependence of the vertical loads of the square of the speed, owing to the term KV^2 in the rolling resistance.

4.2.2 Braking In Ideal Conditions

Braking in ideal conditions means that all the wheels brake with the same longitudinal force coefficient μ_x [5].

In the Ideal braking, the scheme of forces showed on the previous session (figure4.1) is the same but now the braking forces are negative.

The total braking force F_x can be defined as:

$$F_x = \Sigma\mu_{x_i}F_{z_i} \tag{4.9}$$

The longitudinal equation of motion of the vehicle is:

$$\frac{dV}{dt} = \frac{\Sigma\mu_{x_i}F_{z_i} - \frac{1}{2}\rho V^2 SC_x - f\Sigma F_{z_i} - mgsin(\alpha)}{m}, \tag{4.10}$$

Being m the actual vehicle mass, and α the positive uphill grades. The rotating parts of the vehicle do not enter the calculations because they are slowed by the brakes.

Aerodynamic drag and rolling resistance can be neglected since they are smaller than braking forces.

Since in ideal braking all the force coefficients μ_{x_i} are assumed to be equal, the acceleration is:

$$\frac{dV}{dt} = \mu_{x_i}\left[gcos(\alpha) - \frac{1}{2m}\rho V^2 SC_z\right] - gsin(\alpha) \tag{4.11}$$

On road level, for a vehicle without aerodynamic lift, Eq. (4.11) reduces to:

$$\frac{dV}{dt} = \mu_x g \tag{4.12}$$

In ideal conditions, the maximum deceleration can be found introducing the maximum negative value of μ_x on the Eq. (4.12).

The assumption of ideal braking infer that the braking torques applied on all wheels are proportional to the forces F_z, if the radii of the wheels are all equal.

If, during braking, μ_x can be assumed to remain constant, the decelerations of the vehicle is constant, and the usual formulae hold for computing the space and time needed to slow from speed V_1 to speed V_2:

$$t_{V_1 \to V_2} = \frac{V_1 - V_2}{|\mu_x| g}, \qquad s_{V_1 \to V_2} = \frac{V_1^2 - V_2^2}{2|\mu_x| g} \tag{4.13}$$

Then the time and the space to stop the vehicle from speed V are

$$t_{arr} = \frac{V}{|\mu_x| g}, \qquad s_{arr} = \frac{V^2}{2|\mu_x| g} \tag{4.14}$$

The time needed to stop the vehicle increases linearly with the speed while the space increases quadratically.

To calculate the forces F_x that the wheels must exert to perform an ideal braking maneuver, forces F_z on the wheels must be computed first. This is done using the formulae calculated on the load distribution section. But, for vehicles with low aerodynamic vertical loading, such as all commercial and passenger vehicles, aerodynamic loads can be neglected. Drag forces can also be neglected, and then the equations reduce to:

$$F_{z_1} = \frac{m}{l}\left[gbcos(\alpha) - gh_G sin(\alpha) - h_G \frac{dV}{dt}\right] \tag{4.15}$$

$$F_{z_2} = \frac{m}{l}\left[gacos(\alpha) + gh_G sin(\alpha) + h_G \frac{dV}{dt}\right] \tag{4.16}$$

Since the values of μ_x are all identical in ideal braking, the values of longitudinal forces F_x can be computed by introducing the Eq. (4.11)

$$\frac{dV}{dt} = \mu_x gcos(\alpha) - gsin(\alpha)$$

In the equations (4.15) and (4.16)

$$F_{x_1} = \mu_x F_{z_1} = \mu_x \frac{mg}{l} cos(\alpha)(b - h_G \mu_x); \tag{4.17}$$

$$F_{x_2} = \mu_x F_{z_2} = \mu_x \frac{mg}{l} cos(\alpha)(a + h_G \mu_x). \tag{4.18}$$

By adding Eq. (4.17) to Eq. (4.18), it follows that:

$$F_{x_1} + F_{x_2} = \mu_x mgcos(\alpha), \tag{4.19}$$

So:

$$\mu_x = \frac{F_{x_1} + F_{x_2}}{mgcos(\alpha)} \tag{4.20}$$

By introducing the value of μ_x into equations of (4.17) and (4.18) and subtracting the second equation from the first, it follows that

$$\left(F_{x_1} + F_{x_2}\right)^2 + mgcos(\alpha)\left(F_{x_1}\frac{a}{h_G} - F_{x_2}\frac{b}{h_G}\right) = 0. \tag{4.21}$$

This equation plot is shown in the figure 4.2.

The parabola is the place of all pairs of values of F_{x_1} and F_{x_2} leading to ideal braking.

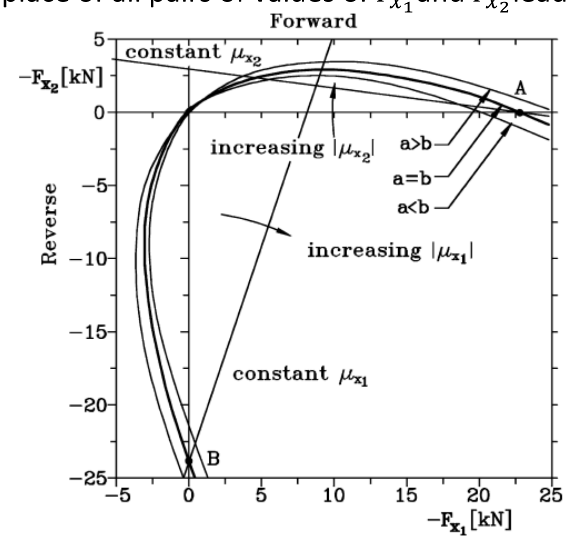


Figure 4.2 – Braking in ideal conditions. Relationship between F_{x_1} and F_{x_2}

The only a part of this plot is actually of interest: That with negative values of the forces (braking in forward motion) and with braking forces actually achievable, ex: μx with reasonable values (Fig. 4.3).

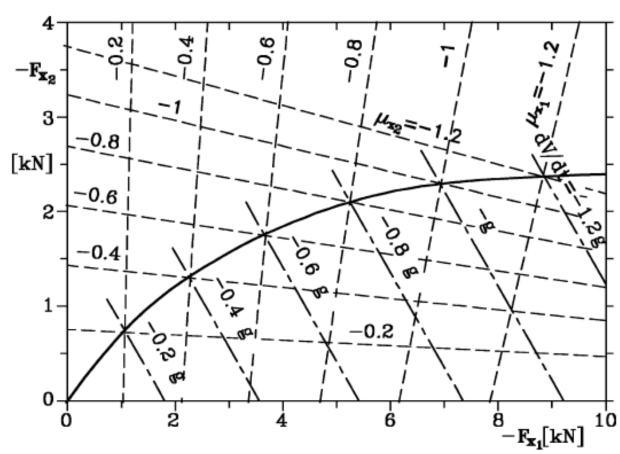


Figure 4.3 - Enlargement of the useful zone of the plot of fig 4.2

4.2.3 Braking In Actual Conditions

The relationship between front and rear wheels braking moments is different from that stated in order to comply with the conditions needed to obtain ideal braking, and is imposed by the parameters of the actual braking system of the vehicle.

A ratio can be defined between the braking moments at the front and the rear wheels as following:

$$K_b = \frac{M_{b_1}}{M_{b_1}}$$

If all wheels have the same radius, its value coincides with the ratio between the braking forces.

For each value of deceleration, a value of K_b can be found in the plot of figure 4.4, in order to ensure ideal braking. It depends on the actual layout of the braking system.

In hydraulic braking systems, the brake torque is related to the pressure in the hydraulic system by means of relation of the type:

$$M_b = \epsilon_b(Ap - Q_s), \tag{4.22}$$

Where ϵ_b, usually referred to as the efficiency of the brake, is the ratio between the braking torque and the force exerted on the braking elements and by consequence has the dimensions of a length. A is the area of the pistons, p is the pressure and Q_s is the restoring force due to the springs, when present.

So K_b can be represented as:

$$K_b = \frac{\epsilon_{b_1}(A_1 p_1 - Q_{s_1})}{\epsilon_{b_2}(A_2 p_2 - Q_{s_2})} \tag{4.23}$$

If no spring is present as in the case of disk brakes, it can be represented as:

$$K_b = \frac{\epsilon_{b_1} A_1 p_1}{\epsilon_{b_2} A_2 p_2} \tag{4.24}$$

In disk brakes, ϵ_b is almost constant and, as an approximation, is the product of the average radius of the brake, the friction coefficient and the number of braking elements acting on the axle, since braking torques again refer to the whole axle. If the pressure acting on front and rear axle is the same, K_b depends only on geometrical parameters and it is constant.

The efficiency of the brakes is a complex function of velocity and temperature and, during braking, if can change due to the combined effect of this two parameters. Usually when the brake heats up there is a reduction of the braking torque, at least in the beginning. Then with the reduction of the speed, can restore the initial values.

If K_b is constant, the characteristic line on the plane M_{b_1}, M_{b_2} is a straight line passing through the origin (Fig. 4.4).

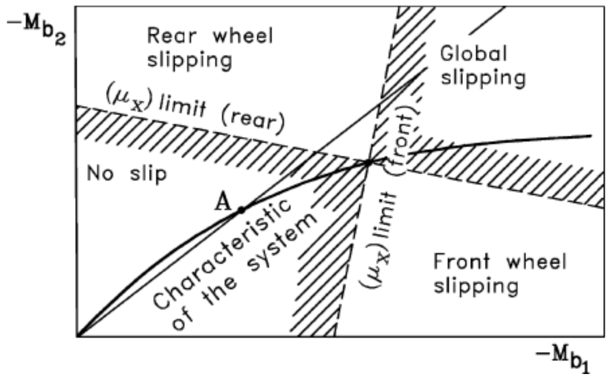


Figure 4.4 - Conditions for ideal braking, characteristic line for a system with constant K_b

The intersection of the characteristics of the braking system with the curve yielding ideal braking defines the situation in which the system performs in ideal conditions. On the left of point A, i.e. for low values of deceleration, the rear wheels brake less than required and the value of μ_{x_2} is smaller than μ_{x_1}. If the limit condition occur in this zone, for example poor traction roads, the front wheels lock first [5].

Instead, all working conditions beyond point A are characterized by

$$\mu_{x_2} > \mu_{x_1}$$

And the rear wheels brake more than needed, that means that the braking capacity of the front wheels is underexploited. In such case, when the limit conditions are reached, the rear wheels lock first (figure 4.4).

On the viewpoint of handling, is recommended that

$$\mu_{x_2} < \mu_{x_1},$$

since this increases the vehicle's stability. Characteristics of the braking system should lie completely below the line for ideal braking. Locking of the rear wheels is a condition that must be avoided since it causes directional instability.

In A the ideal conditions are reached: If the limit value of the longitudinal force coefficient happens at that point, simultaneous locking of all wheels occurs.

The ratio $K_b$values for which the ideal conditions occur at a given value of the longitudinal force coefficient μ_x^* can immediately calculated,

$$K_b^* = \frac{b + h_G|\mu_x^*|}{a - h_G|\mu_x^*|}. \tag{4.25}$$

5. Antilock Braking System (ABS) Model – Bosch Model

Although the braking system used in the cars for experimental data is the Continental's IBS, the ABS Bosch model is shown in this chapter for a comparison reason since it is one of the most known braking system.

In some braking conditions is possible that the wheels block, that could be due to road conditions like slippery or wet road surfaces, or due to really strong activation of the brakes in case of abrupt need to brake. The ABS evaluates constantly the wheel speed and the calculated wheel slip and detects when at least one wheel is about to lock up, and then controls the pressure on the four wheels maintaining the brake pressure constant or reducing it. Doing so, besides preventing the wheels to lock up it also guarantees better steerability and smaller braking distance, improving safety [7].

5.1 System Overview

The ABS braking system is based on the components of the conventional brake system as shown in the figure 5.1.

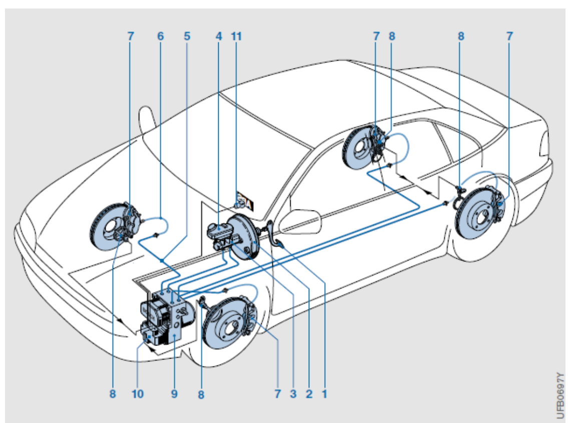


Figure 5.1 - Braking system with ABS

1. Brake pedal
2. Brake booster
3. Master Cylinder
4. Reservoir
5. Brake line
6. Brake hose
7. Wheel brake with wheel – brake cylinder
8. Wheel-speed sensor
9. Hydraulic modulator
10. ABS control unit (in this case, attached unit fixed onto hydraulic modulator)
11. ABS warning lamp

5.1.1. Wheel-speed sensors

The rotation speed of the wheels is a very important input variable for the ABS control system. The wheel-speed sensors detect the speed of rotation of the wheels and pass this information to the control unit by means of electrical signals.

A car can cave three or four wheel-speed sensors depending on the fitted version of the ABS system. The signals of the wheel-speed are used to calculate the degree of slip between the wheels and the road surface, and therefore allowing the system to detect if one or more wheels is about to lock up.

5.1.2. Electronic control unit (ECU)

The ECU processes the information received from the sensors according to pre-defined control algorithms. The results of those calculations form the basis for the control signal that will be then sent to the hydraulic modulator.

5.1.3. Hydraulic modulator

The hydraulic modulator includes a series of solenoid valves that can close or open the hydraulic circuits between the master cylinder (Fig. 5.2,1) and the brakes (4). In addition it can connect the brakes to the return pump (6). Solenoid valves with two hydraulic connections and two valve positions are used (2/2) solenoid valves. The inlet valve (7) between the master cylinder and the brake controls the application of pressure, while the outlet valve (8) between the brake and the return pump controls release of the pressure. There is one of such pair of solenoid valves for each brake.

In normal conditions, the solenoid valves in the hydraulic modulator are at the 'application of pressure' setting. This means that the inlet valve is open. The hydraulic modulator forms a direct connection between the master cylinder and the brakes. Consequently, the brake pressure built in the master cylinder when the brakes are applied is directly transmitted to the brakes as each wheel.

As the slip increases due to heavy braking or due to braking on slippery surfaces, the risk of wheels locking up increases. The solenoid are then switch to 'maintain pressure' mode. The connection between the master cylinder and the brakes is shut off, it means that the inlet valve is closed, any further increase on the master cylinder does not change the pressure on the brakes.

If the slip of at least one wheels increases more than a certain 'safe threshold' despite this action, the pressure in the brakes must be reduced. To do so the solenoid valves are set to a 'release pressure' mode, this means that the inlet pressure is closed and the outlet valve is open, allowing the return pump to draw brake fluid from the brakes in the a controlled manner. The pressure on the concerned brake reduces and the wheel-lock does not occur.

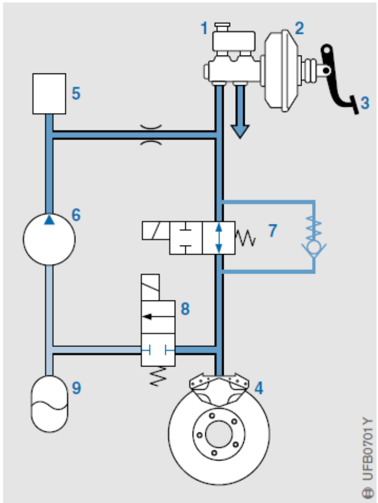


Figure 5.2 - Principle of hydraulic modulator with 2/2 solenoid valves (schematic)

Master Cylinder with reservoir

1. Brake booster
2. Brake pedal
3. Wheel brake with wheel-brake cylinder

Hydraulic modulator with

4. Damping chamber
5. Return Pump
6. Inlet valve (shown in open setting)
7. Outlet valve (shown in closed setting)
8. Brake-fluid accumulator

5.2. Requirements placed on ABS

The ABS system must accomplish many requirements of safety associated with dynamic braking response and braking-system technology.

5.2.1. Handling stability and steerability

The ABS should ensure handling stability and steerability on all kinds of surface (dry roads, wet roads, black ice).

The ABS should utilize as much as possible the adhesion between tires and road surface, giving preference to handling stability and steerability over braking distance. It should not make any difference to the system and to the system actuation if the driver applies the brakes violently or gradually to the point that the wheels would lock.

The control should be able to adapt fast to changes in road surface grip.

In case of braking under different grip on left and right (μ −split), the ABS should control the brake pressure in order to control the unavoidable yaw moment under a certain thresholds that allows average driver to counteract this yaw moment easily.

When cornering and braking the ABS system should ensure handling stability and steerability, braking to standstill as quick as possible if its speed is below the corner's limit speed (that is the absolute maximum speed at which a vehicle can successfully negotiate a bend of a given radius with the drive disengaged).

The system should ensure handling stability and steerability maintaining the best possible braking on bumpy or uneven roads, independent on the force that the driver applied to the brakes.

The system should be able to detect aquaplaning and respond appropriately to it, ensuring controllability and course.

5.2.2. Effective range

The Antilock Braking System must be effective to all the possible range of a vehicle speed down to the crawling speed (minimum speed limit approximately 2.5km/h). If below this speed the wheels lock up, the vehicle travelled distance before it is completely still is not critical.

5.2.3 Timing characteristics

Adjustments in order to overcome the braking system hysteresis (delayed reaction to release of brakes) and the effects of the engine (when braking with the drive engaged) must take as little time as possible.

Vehicle pitching due to suspension vibration must be prevented.

5.2.4 Reliability

A monitoring circuit that continuously check that the ABS is working properly is necessary. If a fault is detected, the ABS should be switched off and a warning lamp should alert the driver.

6. Integrated Brake System (IBS) Model – Continental Model

The brake system used in the Alfa Romeo Stelvio and Giulia, that generated the experimental data is the MK C1 for FIAT [1] that is the Continental's integrated brake system (figure 6.1), and its features are described in this chapter.

Figure 6.1 - Continental's MK C1 for FIAT

MK C1 is a brake system for passenger vehicles which integrates generation and wheel-individual modulation of brake pressures in one module. The energy to provide these pressures is supplied by the on-board power supply of the vehicle in what is called a 'power on demand' energy concept. No vacuum is required. For normal operation, MK C1 employs by-wire brake technology (electro-hydraulic brake) with pedal feedback provided by a simulator. In an 'unboosted' fall-back operational mode, which is automatically active if power should be lost, the driver gets direct hydraulic access to the wheel brakes via the brake pedal. MK C1 supports wheel brakes with higher-than-standard air gaps.

For the conversion of electrical energy into hydraulic actuation energy, the MK C1 compact module uses an electromechanical-hydraulic linear actuator. According to the MK C1 power-on-demand energy supply concept, the required power is taken from the on-board power supply of the vehicle.

6.1. Hydraulic Layout (diagonal-split brake circuits)

The MK C1 can work in two different operation states, or normal operation or hydraulic fallback mode.

In **normal operation**, the driver applies a certain force in the brake pedal (figure 6.3) that generates a brake rod stroke with a consequent pressure rise on the Tandem Master Cylinder (TMC), this pressure arrives in the pedal feel simulator that pushes a small piston against an spring and/or elastic element that its characteristic is defined by means of the car braking style. The spring (or elastic) member is stiffer for more sporty modes (figure 6.2) and softer for more comfortable modes (figure 6.2). The stiffness of the elastic member characterizes the resistance that the brake pressure faces inside the TMC circuit. The stiffer it is, the more resistance it faces. This will generate what is called 'pedal feel' that is the response characteristic that the driver feels while applying a force in the brake pedal.

Since the 'stroke to pressure' behavior in the TMC changes according to the car braking stile characterized by the elastic element in the pedal feel simulator, the measured pressure inside the tandem master cylinder is also influenced by it. The electric signal carrying the information of TMC's pressure will activate the linear actuator (electric motor), the last will generate the main system pressure depending on the pressure on the TMC and on the vehicle estimated speed. Then this pressure will be administrated by the IBS valves in order to deliver to the four wheels brakes the best pressure in order to avoid blocking the wheels. Normal operation mode includes stand-by, regular braking and controlled braking (ABS, TCS, ESC).

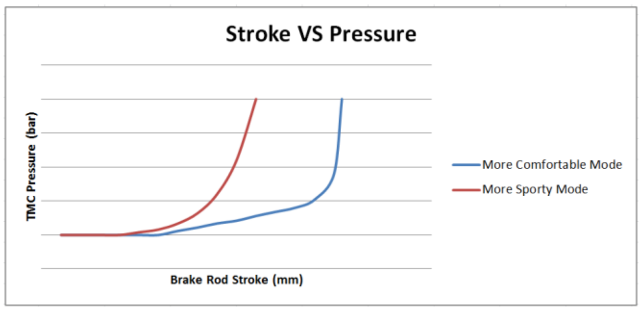


Figure 6.2 – Continental's Pushrod travel vs Pushrod force and Pressure. Comfort and Sport mode

In **Hydraulic fallback mode** there is not the step that and electrical signal is generated by the pressure sensor and activates the electrical motor, but the main system pressure is the same pressure as the tandem master cylinder one, so in this case, there is not the influence of the estimated speed on the main system brake pressure. The model is just working in this mode when you have some problem in the model, like for example the electrical motor is not working properly then a closed simulator valve prevents brake fluid from flowing into the simulator.

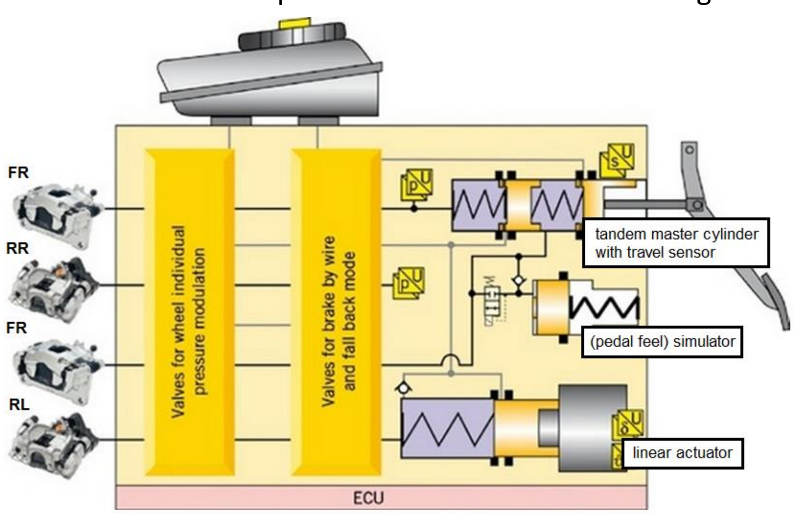


Figure 6.3 – Integrated Brake System (IBS) model [3]

6.2.Hydraulic Control Unit

The hydraulic components of the MK C1 module consists in two subunits: A linear actuator to provide pressurized brake fluid electrically and in controlled fashion, and a hydraulic block with electromagnetic valves to control the actuation chains.

6.2.1 Pedal actuation unit

The pedal actuation unit contains the tandem master cylinder (TMC) and the pedal feel simulator. The primary piston of the TMC is connected to the brake pedal via a pushrod.

6.2.2 Tandem master cylinder

The MK C1 TMC is a ʻʻpedal TMCʼʼ, that means that the pedal is coupled directly to its primary piston by a pushrod. The cross section of the primary piston determines the pedal-force-to-brake-pressure ratio in unboosted fallback mode.

Compared to a conventional brake system, the TMC diameter is reduced, allowing for higher brake pressure in the unboosted fallback mode.

Each chamber stores a reserve of brake fluid to actuate the wheel brakes in unboosted fallback mode, with the volume capacity configured to be the sufficient for accommodating locking of the wheel brakes.

6.2.3 Simulator

The simulator consist in a housing with a piston and a spring and/or a rubber element, and originates the pedal feel during normal operation. In this operational mode, the simulator is hydraulically connected to the primary chamber of the TMC, which by virtue of its hydraulic resistance dampens fast pedal movements and also emulates familiar pedal feel. However, a feature is implemented in order to avoid ʻʻstickingʼʼ pedal feel during releasing the brake pedal.

The spring simulator has a strong progressive characteristic, in that a small preload force enables the simulator piston to move back to reset position when the simulator is not being actuated. And a strongly rising force limits the volume intake on the simulator when arriving to full extension.

6.2.3.1 Valves

<u>Pressure modulation valves</u>

The inlet and outlet valves perform the same functions as in a conventional brake system, allowing the pressure to increase or decrease according to brake request that takes into account both the driver (during normal braking) and the MK C1 automated brake functions such as ABS, ECS, TCS, etc.

<u>Actuator Valves</u>

The unit ʻMK C1 actuator valvesʼ contains the valves allowing MK C1 to switch between its operational modes.

7 Maneuvers

In this chapter is presented the real maneuvers performed in order to obtain the experimental data and the simulation maneuvers that reproduced, in a virtual way, the real ones.

7.1 Real Maneuvers

Real data generated in real brake test were provided in order to allow comparison between simulation and real numbers. Below is listed the real maneuvers (tables 7.1 and 7.2) done to obtain the data:

Vehicles:

AR 949 - Stelvio

Maneuver	Speed Before Braking	Final Speed	Mu
Hard Brake	100 km/h	0km/h	High Mu (1)

Table 7.1 – Experimental maneuver data for AR949 - Stelvio

AR 952 – Giulia

Maneuver	Speed Before Braking	Final Speed	Mu
Hard Brake	108 km/h	0km/h	Wet Road (0.75)

Table 7.2 – Experimental maneuvers data for AR952 – Giulia

Provided data for each maneuver:

- Real Brake pressure on the four corners;
- Estimated brake pressure on the four corners;
- Main system pressure;
- Four wheels slip;
- Four wheels linear speed;
- Vehicle estimated speed;
- Vehicle GPS speed (just for AR949);
- Brake rod travel;
- Yaw Rate;
- Longitudinal Acceleration;
- Engine Speed;
- Engine torque;
- Main clock;
- Clutch Position;
- Yaw Rate;

7.2 Simulation Maneuvers

As the model does not work well in low speed the simulation finishes at 20km/h.

The simulation maneuvers were set on VI-CarRealTime test mode, on the driving machine file as three mini maneuvers in order to replicate the maneuvers that generated the experimental data. The tree mini maneuvers are:

1. Constant Speed: 1.5 seconds of constant speed maneuver were implemented, where:
   - No steering is implemented;
   - Throttle, braking, gear and clutch were set in order to maintain the desired constant speed.
2. Idle: Were implemented 0.5 seconds of idle condition between constant speed and hard brake, where:
   - No steering is implemented;
   - Throttle is off;
   - Braking is off;
   - No gear shifting;
   - Clutch is off.
3. Hard Braking: It was implemented until the end of the simulation that will be when the car reached the speed of 20km/h, where:
   - No steering is implemented;
   - Throttle is off;
   - The brake pedal stroke map (in time) is implemented;
   - The gear shifting map is implemented;
   - Clutch is off.

8 Cosimulation Modeling

In this chapter are described the virtual model and all the steps done on the virtual environment in order to obtain the same outputs as the experimental data ones (ex: brake pressure in the corners), having a defined input equivalent to the experimental input (ex: brake pedal stroke).

In this study were used some different software in order to obtain the most accurate results. A CarRealTime block was introduced in the Matlab-Simulink interface (figure 8.1) in a closed loop in order to provide and receive signals allowing modification of Simulink logic, that interferes the car behavior. This simulation in more than one software simultaneously is called cosimulation.

In the cosimulation the CarRealTime block was providing the following signals:

- Wheel Speeds (FL- front left, FR- front right, RL- rear left, RR- rear right);
- Chassis Speed;
- Chassis Acceleration;
- Brake pedal Stroke;

And receiving the following signals:

- Disabling the CRT internal brake system ;
- Brake torque (FL, FR, RL, RR);

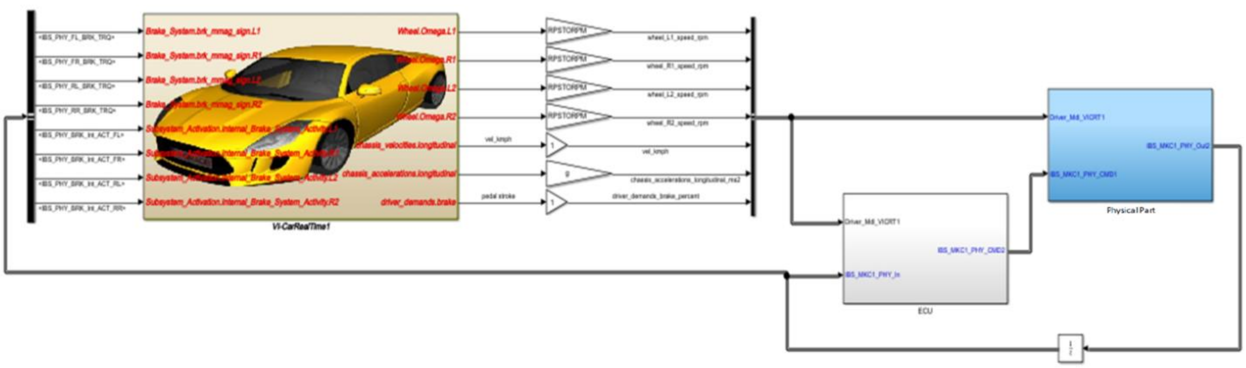


Figure 8.1 - Matlab-Simulink environment

The Simulink logic environment was divided in two blocks:

- ECU – Electronic Control Unit;
- Physical Part.

8.1 ECU – Electronic Control Unit

The Matlab-Simulink ECU block simulates the physical car's Electronic Control Unit. And its logic is described in this session.

8.1.1 Model Input

As channels (signals) were provided from experimental data from real maneuvers, in the virtual model was used as input the brake pedal pushrod stroke in time (figure 8.2) as the measured experimental data.

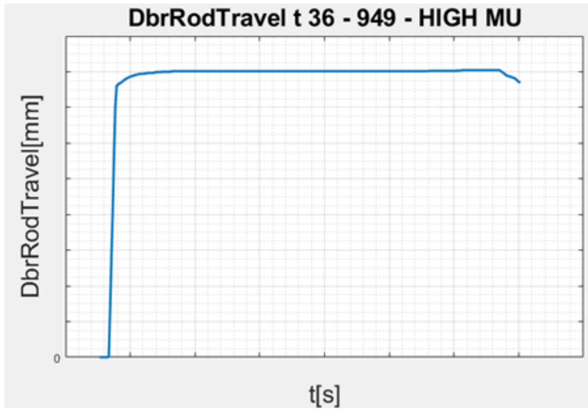


Figure 8.2 - Brake pedal pushrod travel – Experimental data

Once knowing the brake pedal pushrod stroke, it was possible to calculate the driver's requested deceleration. This could be done using maps provided by Continental and some other tabled relations between variables.

As the main system pressure does not depend only on the pressure of the tandem master cylinder that also actuates on the pedal feel simulator, but also depends on the vehicle estimated speed, in order to calculate the driver's requested deceleration, both the brake pedal pushrod stroke and the estimated vehicle speed were used, as shown in the schematic representation of figure 8.3.

The signal of the brake pedal rod stroke enters the subsystem and it is modulated by a value equivalent to que pedal deformation since the maximum value of stroke defined by Continental's tandem master cylinder is smaller than the maximum value of the rod stroke measured in the experimental data. So the brake pedal rod stroke was modulated using the both the maximum stroke values from the measured experimental data and from the Continental provided data.

Then the modulated value of the brake pedal rod stroke enters in 'Ibs_Min_Map' (figure 8.4.b) and on 'Ibs Max_Map' (figure 8.4.d) that are a direct relation between stroke and deceleration. Each of this blocks output are deceleration values [g], and those values are weighted depending on the vehicle estimated speed.

The 'IBS_Speed_weighter' (figure 8.4.c) gives an output from zero to one depending on the input that is the vehicle estimated speed. Then depending on its output, the values exiting the 'Ibs_Min_Map' and the 'Ibs_Max_Map' are combined. For example, if the 'IBS_Speed_weighter' output is 0.6, the output from 'IBS_Max_Map is multiplied by 0.6 and the 'Ibs_Min_Map' by 0.4 and then sum both of those values.

In parallel to this calculated pressure there is an increment on the deceleration coming from the following blocks: ‘Stroke to pressure – Continental – Comfort’ (figure 8.5) and ‘Pressure to deceleration’ (figure 8.4.a). The ‘Stroke to pressure – Continental – Comfort’ block comes from a graph provided by continental with the relation between stroke, pressure and force in the tandem master cylinder, so came from there the relation between stroke to pressure, being the input the brake rod stroke and the output the TMC pressure. Then the ‘Pressure to deceleration’ block gets the pressure calculated in the previous block and transforms it in deceleration that is added to the previous calculated ones.

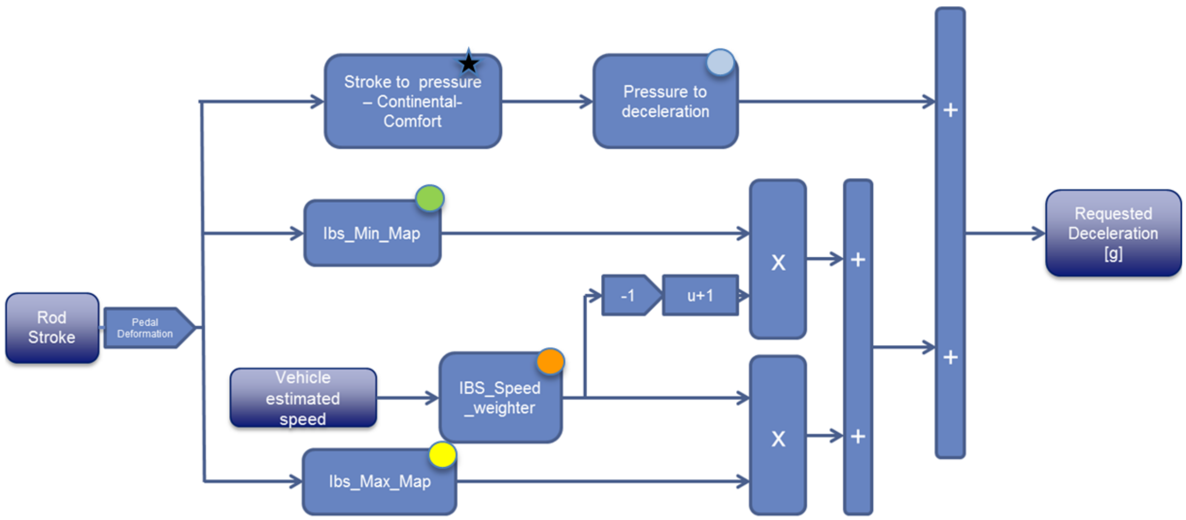


Figure 8.3 - Schematic representation of obtaining driver's desired deceleration

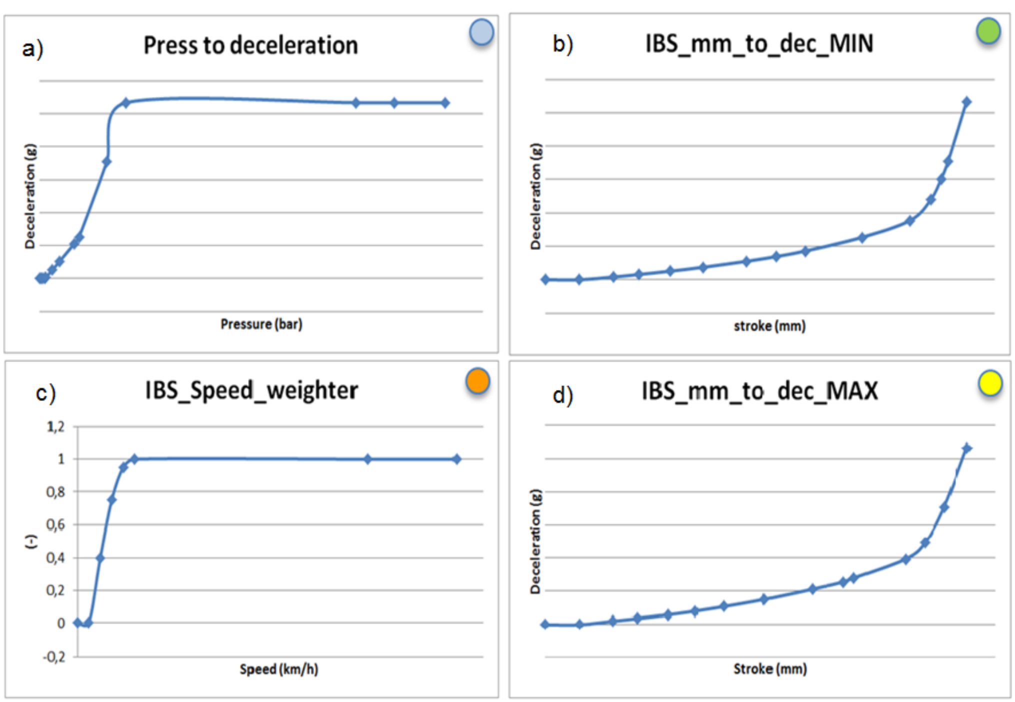


Figure 8.4 - Plots of the blocks used in the scheme for obtain the driver's desired deceleration

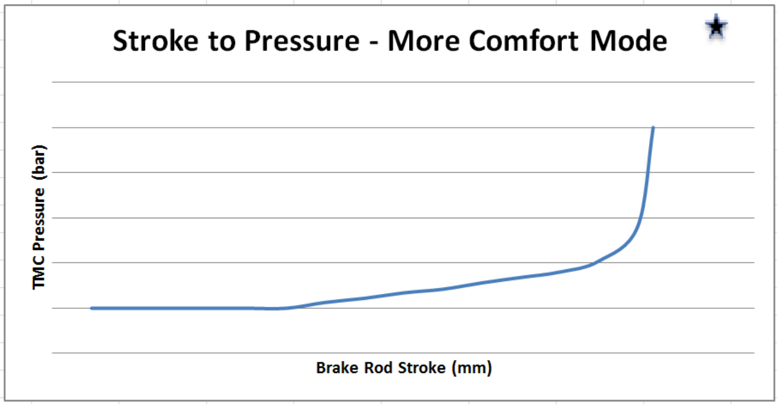


Figure 8.5 – Stroke to pressure – Continental – Comfort

8.1.2 Pedal Pushrod Deformation

As the Tandem Master Cylinder's maximum stroke defined by continental (figure 8.7) is smaller than the maximum stroke measured on experimental data (figure 8.6), this difference in stroke is considered to be the pedal pushrod deformation. So the pedal pushrod stroke is modulated in order to keep the pressure on the tandem master cylinder inside the map's range.

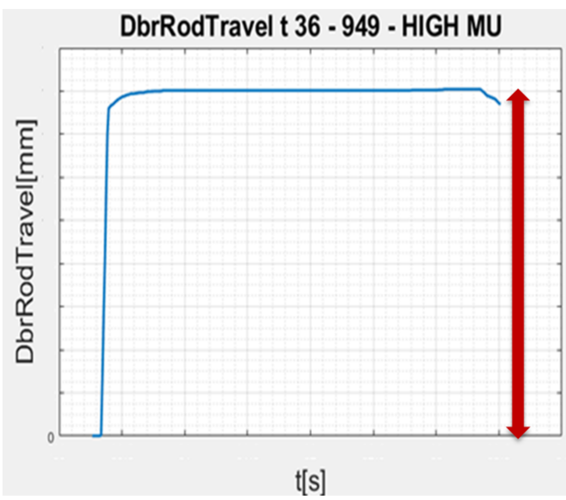


Figure 8.6 - Brake pedal pushrod travel- Experimental data

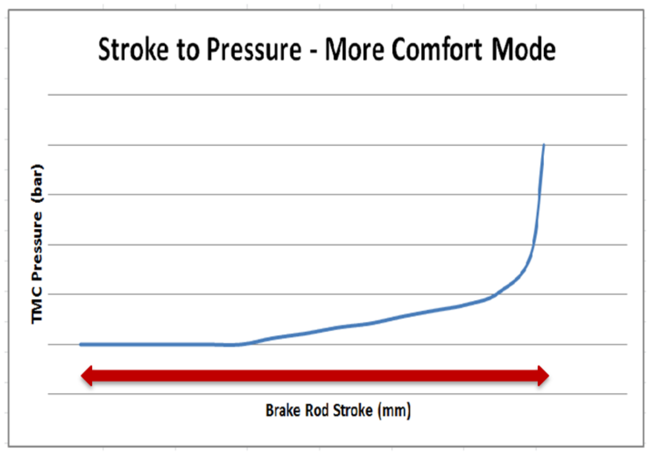


Figure 8.7 - Stroke to pressure – Continental – Comfort

8.1.3 Relation between Fx1 and Fx2

Once discovered the driver's desired acceleration, is possible to calculate the brake force needed on the front and rear axle, and with these brake forces is possible to calculate the needed braking pressure.

The figure 8.8 shows the relation between F_{x_1} and F_2 during ideal braking, evidencing also the lines at constant deceleration and constant friction coefficient on the front and on the rear axle.

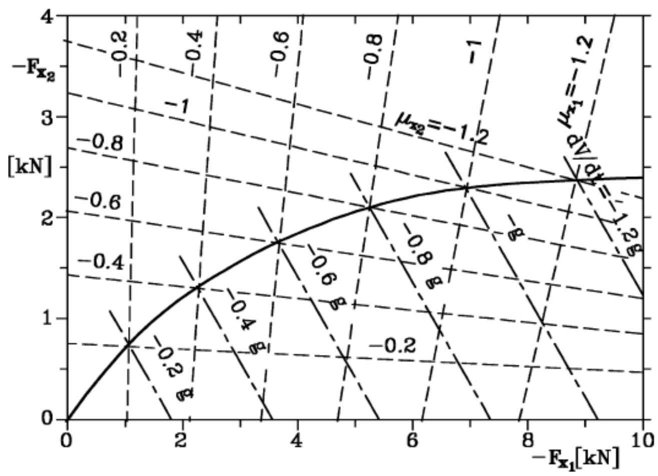


Figure 8.8 - Braking in ideal conditions. Relationship between F_{x_1} and F_{x_2}

In order to stablish the relation between the brake force on the front axle and on the rear axle, is set a curve of a line and a parabola in the calculations (figure 8.9).

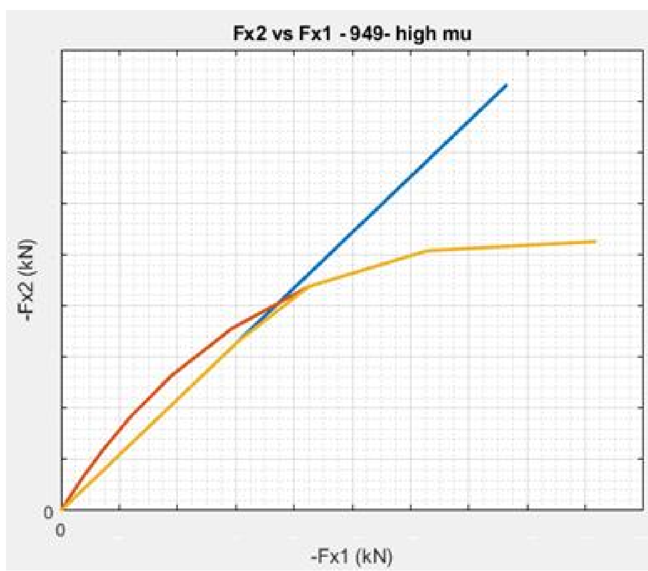


Figure 8.9 - Simulation's relation between F_{x_1} and F_{x_2}

The equation of the line (in blue) is defined when the same brake pressure is exerted on the front and rear axle, obtaining a fixed relation between brake forces due to the hydraulic bias. The equation of the line was defined by the following two relations:

The Eq. (8.1) being the relation between brake force and longitudinal acceleration, while the Eq. (8.2) being the consequence of the hydraulic relation between brake pressure and brake force.

$$F_{x_1} + F_{x_2} = m.acc_x; \tag{8.1}$$

$$\begin{cases} P.BiasF = F_{x_1} \\ P.BiasR = F_{x_2} \end{cases} \rightarrow \text{So: } F_{x_2} = \frac{BiasR}{BiasF} F_{x_1}; \tag{8.2}$$

The parabola (in red) is defined due to the load transfer to the front axle during brake maneuvers, and when it happens, a bigger brake force is allowed on the front axle, and by consequence, as the weight on the rear axle is reduced [6], the brake force has to be reduced in order not to block the rear axle's wheels – Electronic Brake-force distribution (EBD). The parabola was calculated with the following two equations.

The Eq. (8.3) being the longitudinal force on front axle, generated by the normal force acting on the front axle, and the Eq. (8.4) reports the relation between the forces at the front and rear axles that must hold to male ideal brake possible:

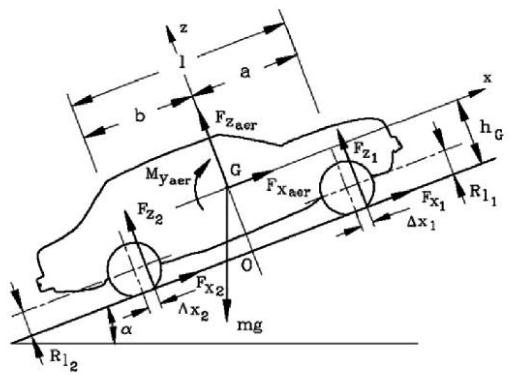


Figure 8.10 - Forces acting on a vehicle moving on an inclined road

$$F_{x_1} = \mu_x F_{z_1} = \mu_x \frac{mg}{l} cos(\alpha)(b - h_G \mu_x); \tag{8.3}$$

$$\left(F_{x_1} + F_{x_2}\right)^2 + mgcos(\alpha)\left(F_{x_1}\frac{a}{h_G} - F_{x_2}\frac{b}{h_G}\right) = 0; \tag{8.4}$$

Being the variables of the formulas also represented on the figure 8.10:

- $F_{x_1}, F_{x_2,}$: longitudinal force on front and rear axles;
- F_{z_1}: normal force acting on the front axle;
- m: vehicle mass;
- μ_x: longitudinal force coefficient;
- l: wheelbase;
- α: longitudinal grade angle (positive when moving uphill);
- a: distance between the vehicle mass center and the front axle;
- b: distance between the vehicle mass center and the rear axle;
- h_G: vehicle mass center height;
- $index_{1,2}$: 1 for front axle and 2 for rear axle;
- acc_x: longitudinal acceleration.

So a curve (in yellow) was constructed getting the minimum value of F_{x_2} between the values of the line and the parabola in order to guarantee better vehicle stability.

The previous graph is reported as shown in figure 8.9 has the deceleration limited to the tire's limit adherence with the road. If the acceleration is not limited by the tire's limit, as the requested deceleration is so high in modulus, the results are like the figure 8.11.b (in the example the requested deceleration is -4g).

In that case as the requested deceleration is so high in modulus, it brings the curve to a positive value of F_{x_2} that means traction and not braking on the rear axle, that makes no sense in a braking maneuver.

So the only part of the plot of figure 8.11.a that is of interest is the one with negative values of braking forces (braking in forward motion) and with achievable braking forces.

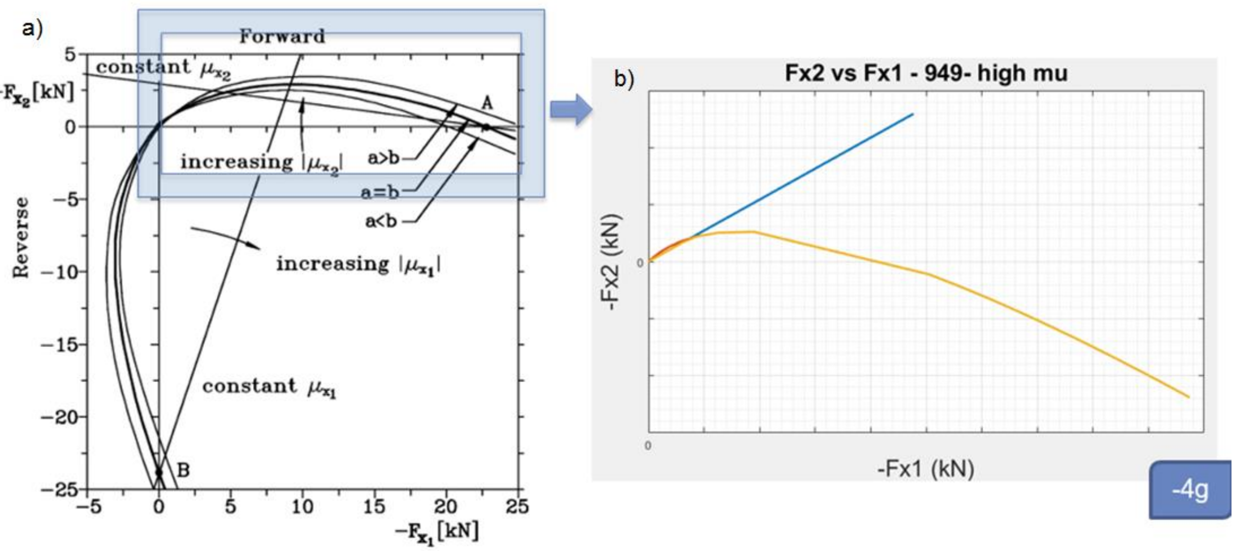


Figure 8.11- Braking in ideal conditions. Relationship between F_{x_1} and F_{x_2}. (a) Theoretical; (b) Simulation

8.1.4 System Pressure

The system pressure is defined as being the one needed in order to provide the maximum brake force between F_{x_1} and F_{x_2} for each point of the yellow curve (figure 8.9).

The front brake force is divided by the front Hydraulic Bias in order to find the pressure target on the front axle, and the same is done on the rear axle. So then the main system pressure is defined by the maximum between this two pressures as shown in the scheme of figure 8.12.

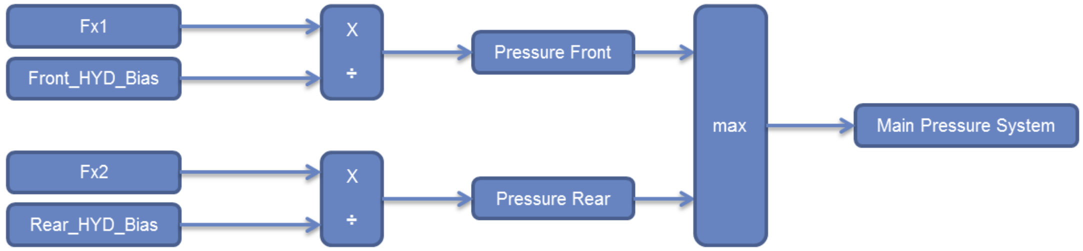


Figure 8.12 - Schematic representation of obtaining the main pressure of the system

The hydraulic bias is defined by:

$$FrontHydBIAS = \frac{2 \cdot \mu_{pad_{front}} \cdot A_{cyl_{front}} . n_{cyl_{front}} \cdot R_{eff_{Disk_{front}}} \cdot 2}{\cdot R_{eff_{Wheel_{front}}}} \tag{8.5}$$

$$RearHydBIAS = \frac{2 \cdot \mu_{pad_{rear}} \cdot A_{cyl_{rear}} . n_{cyl_{rear}} \cdot R_{eff_{Disk_{rear}}} \cdot 2}{\cdot R_{eff_{Wheel_{rear}}}} \tag{8.6}$$

Being:

2: Due to two pad surfaces in contact with the brake disc;

μ_{pad}: Pad friction coefficient;

A_{cyl}: Cylinder's area (represented in blue);

n_{cyl}: Number of cylinders;

$R_{eff_{Disk}}$: Disk's effective radius (represented in red);

$R_{eff_{Wheel}}$: Wheel's effective rolling radius (represented in green).

2: Since there are two wheels for each axle.

The figure 8.13 shows a schematic diagram of disc brake. In this type of brake, a force is applied to both sides of a rotor and braking action is achieved through the frictional action of inboard and outboard brake pads against the rotor [8]. It represents where $R_{eff_{Disk}}$, $R_{eff_{Wheel}}$ and A_{cyl} are measured:

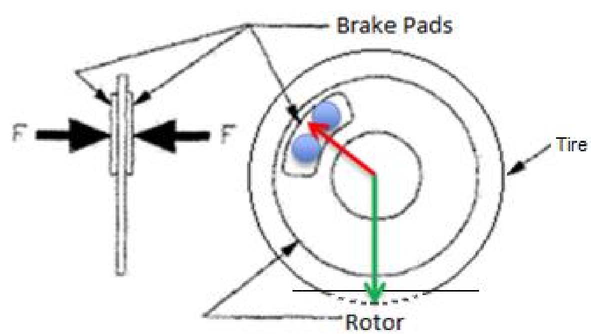


Figure 8.13 – Disk brake configuration example

Obs: For the hydraulic bias, a fixed value for the effective rolling radius is used.

8.1.5 Speed Estimator

One of the big problems of estimating the speed is that during hard braking or riding in slippery surfaces the wheels do not roll according with the vehicle speed, when at least one wheel locks, it does not provide a reliable vehicle speed, the vehicle speed has to be estimated taking into account the vehicle acceleration and all wheel speeds in order to estimate the real vehicle speed.

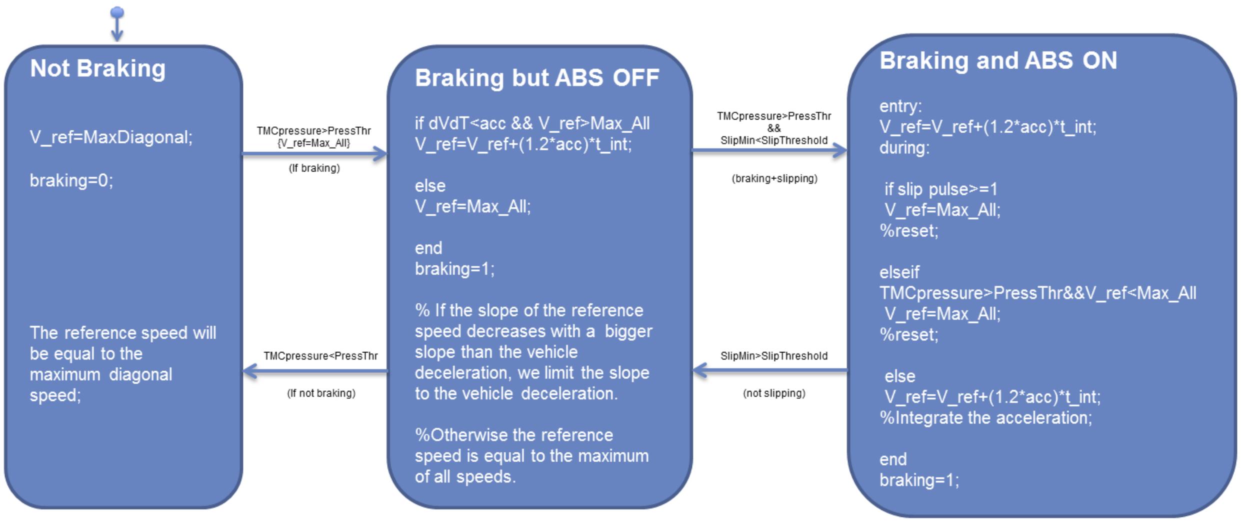


Figure 8.14 - Speed estimator logic

So a part of this project was to create a vehicle speed estimator, and it was done in the Matlab-Stateflow taking into account all estimated wheel speeds and the vehicle acceleration, composed by three states: 1) Not braking; 2) Braking with ABS off; 3) Braking with ABS on. It's logic is represented in the figure 8.14.

During:

1) <u>Not Braking:</u> Set the estimated vehicle speed to be the Maximum Diagonal speed, which would be, the maximum value between (Front Left linear wheel sped + Rear Right linear wheel speed)/2 and (Front Right linear wheel speed +Rear Left linear wheel speed)/2. It corresponds to a reliable value of vehicle speed since the wheels are not slipping/locked.
2) <u>Braking with ABS off:</u> The vehicle estimated speed is equal to the maximum linear speed between the four wheels. But in this case the rate of the vehicle speed is limited by 1.2 times vehicle's measured acceleration in order to avoid big drops in the vehicle speed estimation.
3) <u>Braking with ABS on:</u> The speed estimator integrates the vehicle acceleration in time. This integrator resets to the maximum linear speed of all wheels if the estimated speed is smaller than at least one of the estimated linear wheel speeds, and also if there is a big increase in the slip (modulus) – represented by 'pulse' in the logics.

Passages between blocks:

1) The logic passes from the first block - Not braking - to the second one – Braking with ABS off – if the system brake pressure is bigger than a certain threshold it means that the driver is pressing the brake pedal.
2) The logic passes from the second block - Braking with ABS off - to the third - Braking with ABS on – if the system pressure is bigger than a certain threshold and If the minimum slip of all wheels (remember that slip in braking is negative) is smaller than a slip threshold, it means that the driver is braking and that at least one of the wheels is slipping.
3) The logic passes from the third block - Braking with ABS on - to the second - Braking with ABS off - if the minimum slip of all wheels is bigger than a certain threshold, it means that the wheels are not slipping anymore.
4) The logic passes from the second block- Braking with ABS off - to the first one - Not braking - if the system brake pressure is smaller than a certain threshold, it means that the driver is not pressing the brake pedal.

The following image (figure 8.15) represents the results of the estimation of the speed in yellow, being the 4 curves oscillating under it, the four estimated linear speeds, and the pink curve the CarRealTime real speed:

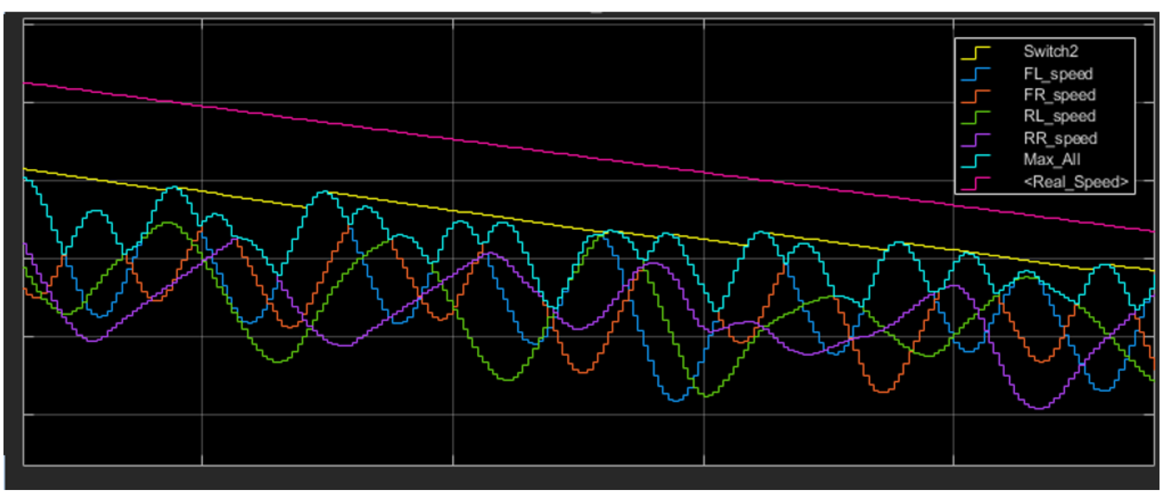


Figure 8.15 - Estimated speed (in yellow)

In order to avoid jumps in the estimation of the speed, a rate limiter was implemented. It limits the decrease of the estimated speed to 1.2*acceleration. When the driver is not braking, it does not limit it, it is set to be the maximum diagonal speed, as shown in the figure 8.16.

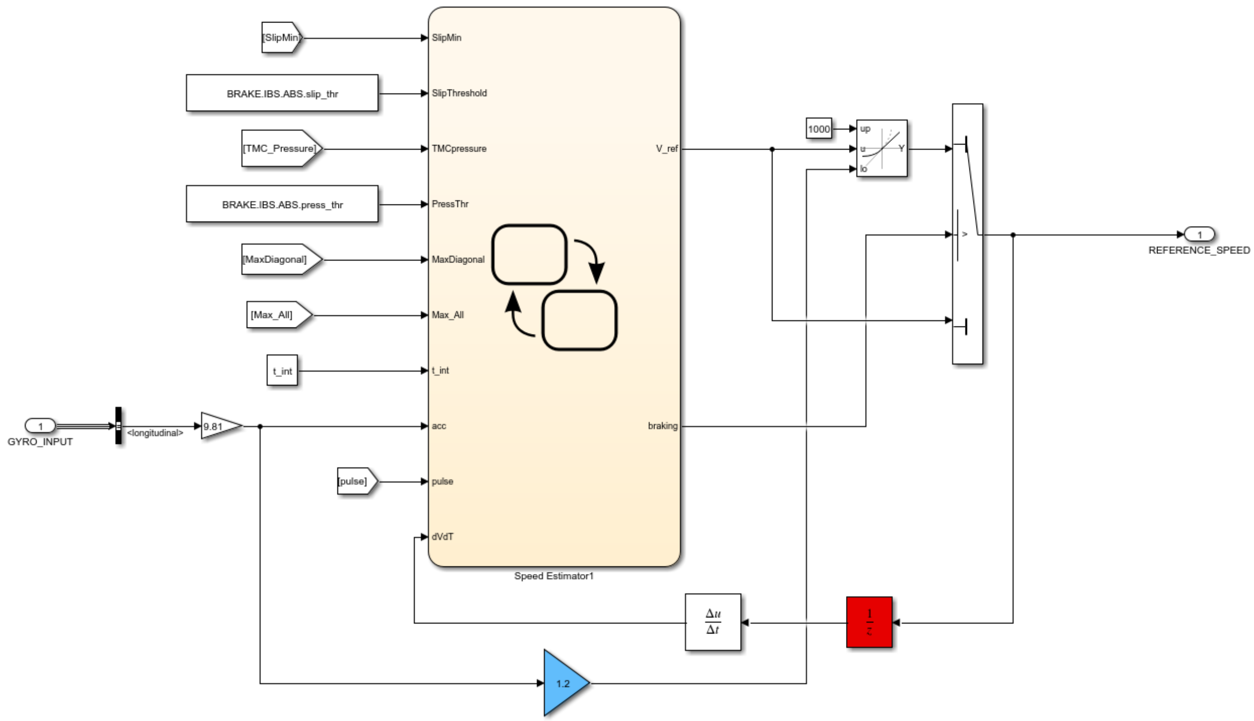


Figure 8.16 - Simulink speed estimator with anti-jump feature

The next image (figure 8.17) shows in yellow the estimated speed before the rate limiter and the blue curve the estimated speed after the rate limiter:

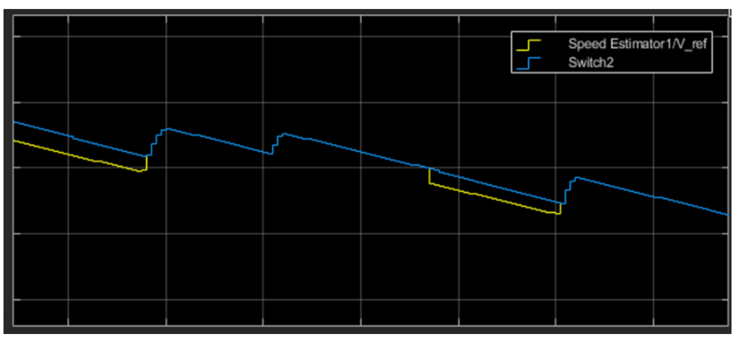


Figure 8.17 - Before (in yellow) and after (in blue) the implementation of the anti-jump feature

8.1.6 Effective Rolling radius estimation

In order to better estimate the speed and the slip, was observed the need of better estimating the effective rolling radius since it is not constant. Is known that the wheel's rotational instant center is normally located under the road level and that rolling radius decreases in traction conditions and increases in braking conditions [2] as shown in figure 8.18.

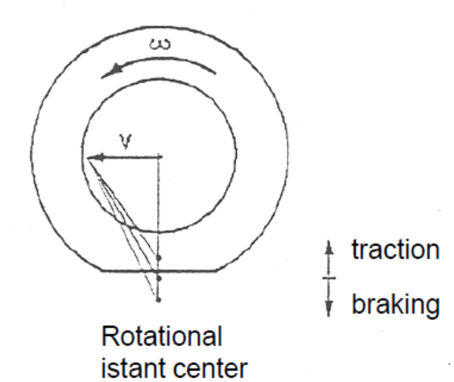


Figure 8.18 - Rotational instant center

A block was created in the Simulink logic in order to better estimate the effective rolling radius. To do so, the following three Equations were used:

- Relation between $\boldsymbol{F_x}$ and vehicle longitudinal acceleration:

$$\Sigma F_x = m \cdot acc \tag{8.7}$$

- Relation between F_x and Slip:

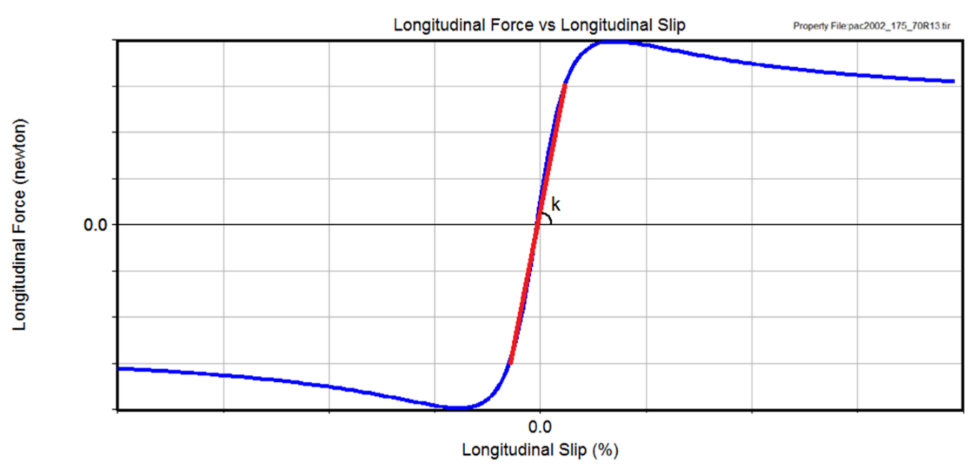


Figure 8.19 - Longitudinal force versus longitudinal slip, with the slope K for small slip values

Considering small values of slip, the relation is in the linear range, represented by the red line in the figure 8.19:

$$F_x = K.\sigma \tag{8.8}$$

- Equation of the slip:

$$\sigma = \frac{w \cdot R_e - V_{estim}}{V_{estim}} \tag{8.9}$$

Making a rough approximation saying that $V_{estim} = wR_e'$, being w the angular speed on one wheel: (Remember that R_e (undeformed radius) is a constant value and R_e' (rolling effective radius) isn't.)

$$\sigma = \frac{w \cdot R_e - wR_e'}{wR_e'} \tag{8.10}$$

Substituting the Equation (8.8) on the (8.7):

$$m \cdot acc = K.\sigma \tag{8.11}$$

Substituting the Equation (8.10) on the (8.11):

$$m \cdot acc = K\,\frac{w \cdot R_e - wR_e'}{wR_e'} \quad \therefore \quad m \cdot acc = K\left(\frac{R_e}{R_e'} - 1\right) \tag{8.12}$$

Isolating $\frac{R_e}{R_e'}$:

$$\frac{R_e}{R_e'} = \frac{m \cdot acc}{K} + 1 \quad \therefore \quad R_e'\left(\frac{m \cdot acc}{K} + 1\right) = R_e \tag{8.13}$$

The following equations of R_e' is obtained:

$$R_e' = \frac{R_e}{\left(\frac{m \cdot acc}{K} + 1\right)} \tag{8.14}$$

Calling $K_1 = \frac{m}{K}$, and remembering that during braking the acceleration is negative and that the effective rolling radius should increase. Then, finally the dynamic relation used in the effective rolling radius estimation is found:

$$R_e' = \frac{R_e}{(K_1 \cdot acc + 1)} \tag{8.15}$$

8.1.7 ABS Valves Logic

As described before, the valve logic is calculated on the ECU in the following format (figure 8.20):

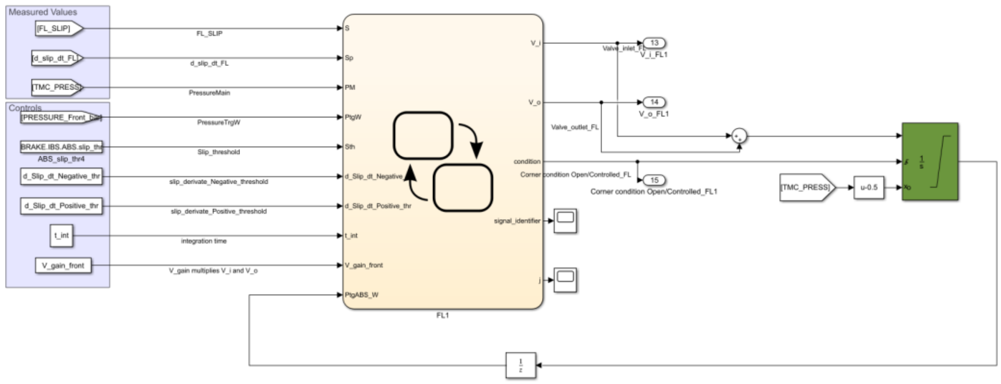


Figure 8.20 - Simulink ABS valve logic FL

The model have different approaches for front and rear valves, since the front valves work directly on OPEN condition or on ABS mode control. And the rear axle valves work on OPEN condition, EBD mode or on ABS mode control.

The valves logics use as input the following channels:

- Measured Values:
  - Brake pressure at the corner;
  - Wheel slip;
  - Derivate of the wheel slip;
  - Main System pressure;
  - Wheel acceleration;
- Controls:
  - Front and rear axle pressure;
  - EBD_gain – constant that multiplies the inlet and outlet pressure valve rate at EBD;
  - Slip threshold (slip target);
  - Positive threshold of the derivate of the slip;
  - Negative threshold of the derivate of the slip;
  - Integration time;
  - V_gain – multiplies the values of V_i and V_o;
  - Wheel acceleration threshold.

8.1.7.1 ABS Valves Logic – Front Valves

In order to control the brake pressure at the corners, were stablished slip target and thresholds for the derivate of the slip, a positive one (SpD) and a negative one (SpC), so that the pressure can be controlled based on the slope of the slip.

The control of the pressure is done by V_i (inlet pressure valve rate) and V_o (outlet pressure valve rate), being the value of V_i positive and V_o negative. The variable “Condition” is created to determine if the system is on ABS or EBD or in none of them. This logic was created in Stateflow from Matlab (figure 8.21), and represented schematically in the figure 8.22.

If the logic is in OPEN condition, the brake pressure at the four corners is the same as the system pressure, and variable “Condition” is set to zero. Then if the slip of at least one of the wheels is smaller than a certain threshold, the ABS is activated.

The main goal is to keep the slip close to a target, so analyzing the slip and its derivate, the brake pressure on the front wheels can be controlled.

Remembering that the slip during braking is negative:

1. If Slip is smaller than the Slip threshold, then its derivate should be analyzed:

- If the slip derivate is smaller than the negative slip derivate threshold(Sp<SpC), it means that the slip is going in the opposite direction of the slip target in a really fast way, so the brake pressure has to be decreased a lot in order to make the slip closer to the target value. To do so, were set the values of V_i and V_o, being V_i=0 and V_o equal to a negative value big in its modulus;
- If the slip derivate is smaller than zero and bigger than the negative slip threshold (SpC<Sp<0), it means that the slip is decreasing not in a really fast way, going in the opposite desired direction. There is the need to decrease a bit the brake pressure on that wheel’s brake caliper, to try to make it to go in the direction of the slip threshold. To do so, were set the values of V_i=0 and V_o equal to a negative value and small in its modulus;
- If the slip derivate is bigger than zero and smaller than the positive slip threshold (0<Sp<SpD), it means that the slip is smoothly going in the direction of the slip threshold, and this is exactly what is wanted, so the brake pressure is maintained constant. To do so, were set values of V_i=0 and V_o=0;
- If the slip derivate is bigger than the positive slip derivate threshold (SpD<Sp), it means that the slip is going in the direction of the slip threshold that is a good thing, but as it is going in a really fast way (big slope) there is the need to increase a bit the brake pressure, in order to avoid the slip passing through the slip threshold and getting far away from it. To do so, were set V_i equal to a small positive value and V_o=0;

2. If the slip is bigger than the slip threshold, then its derivate is analyzed:

- If the slip derivate is bigger than the positive slip derivate threshold (Sp>SpD), it means that the slip is going in the opposite direction of the slip threshold in a really fast way, and this is exactly the opposite thing that is desired. So in order to try to make the slip go in the direction of the slip threshold, there is the need to increase a lot the brake pressure on that brake caliper. To do so, were set V_i equal to a big positive value and V_o=0;
- If the slip derivate is bigger than zero and smaller than the positive threshold (SpD>Sp>0), it means that the slip is going in the opposite direction of the slip threshold but in a slow way. So there is the need to increase a bit the brake pressure in order to try to make the slip go in the direction of the slip threshold. To do so, were set V_i equal to a small positive value and V_o=0;
- If the slip derivate is smaller than zero and bigger than the negative threshold (0>Sp>Spc) it means that the slip is going in the direction of the slip threshold in a smooth way, and this is exactly the objective. So the pressure is maintained constant setting V_i and V_o equal to zero.
- If the slip derivate is smaller than the negative slip derivate threshold (Sp<SpC) it means that the slip is going in the direction of the slip threshold but in a really fast way, and in order to avoid that the slip crosses the slip threshold and goes far from it there is the need to decrease a bit the brake pressure. To do so, were set V_i=0 and V_o equal to a negative value small in its modulus;

The passages between blocks are:

- It passes from OPEN to ABS if the slip of at least one of the wheels is smaller than the slip threshold;
- It passes from ABS to OPEN if the pressure target on ABS is equal to the main system pressure and if the ABS did at least a minimum number of control cycles.

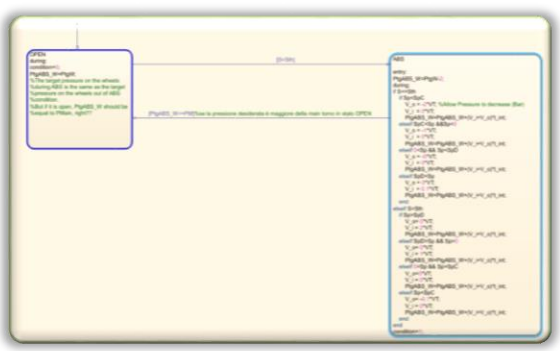


Figure 8.21 - Simulink – Matlab-Stateflow front valves logic

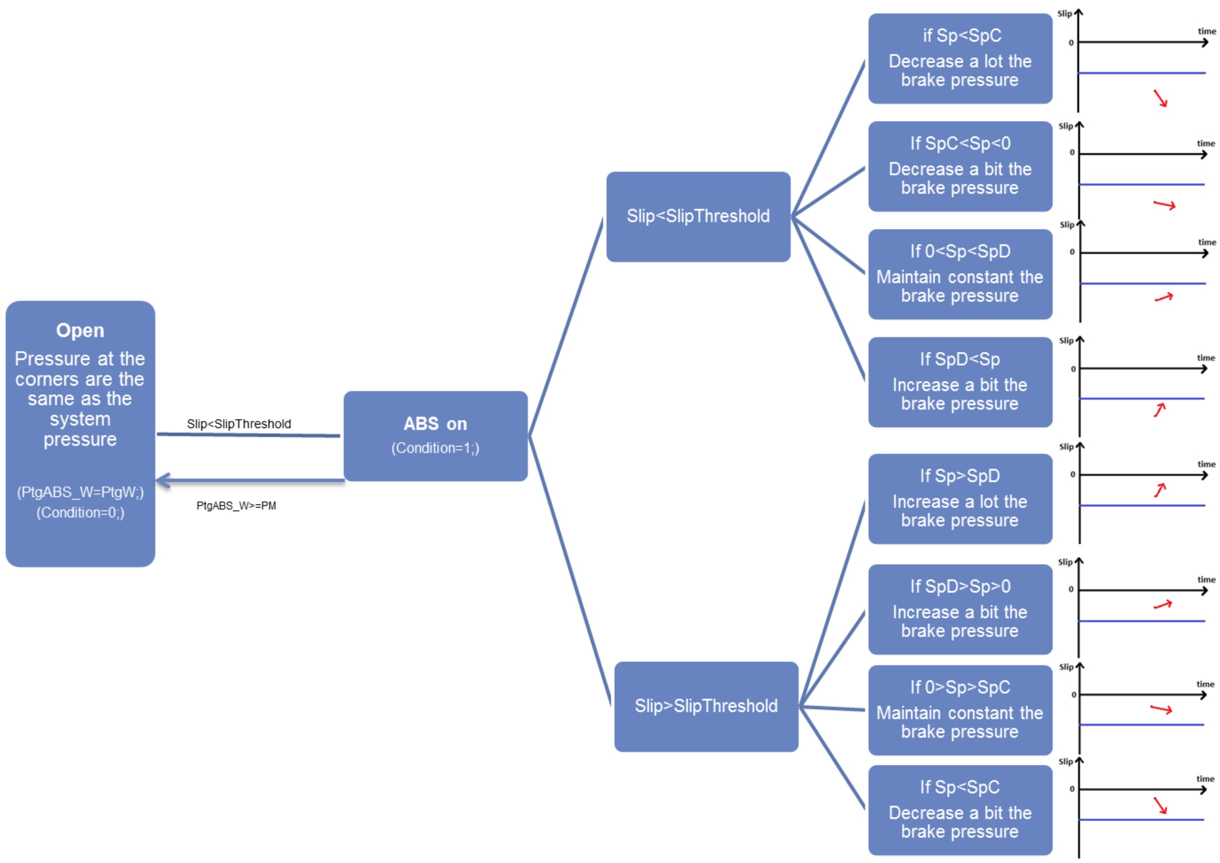


Figure 8.22 - Schematic representation of ABS front valves logic

8.1.7.2 ABS Valves Logic – Rear Valves

The logic created in Matlab-Stateflow (figure 8.24) of the rear valves is almost the same as the front ones. The only thing that changes is the introduction of the EBD block, represented schematically in the figure 8.23.

The EBD block controls the brake pressure on the rear axle. When the logics is in this block, it increases or decreases the pressure in order to keep it close to the target pressure on the rear axle, that was defined previously as being the pressure necessary in order to provide the ideal brake force on the rear axle (yellow curve including part of a line and part of a parabola).

The passages between blocks are:

- It passes from OPEN to EBD if the brake pressure target on the rear wheels is smaller than the pressure main system pressure;

- It passes from EBD to ABS if the slip is smaller than the slip threshold or if the wheel acceleration is smaller than the wheel acceleration threshold for more than a certain time interval.
- It passes from ABS to EBD if the pressure target on ABS is bigger or equal to the pressure target on the rear axle. (The pressure target on the rear axle is defined by the relation between F_{x_1} and F_{x_2} – line and parabola defined previously on figure 8.9;
- It passes from EBD to OPEN if the pressure target on the wheels is the same as the main system pressure.

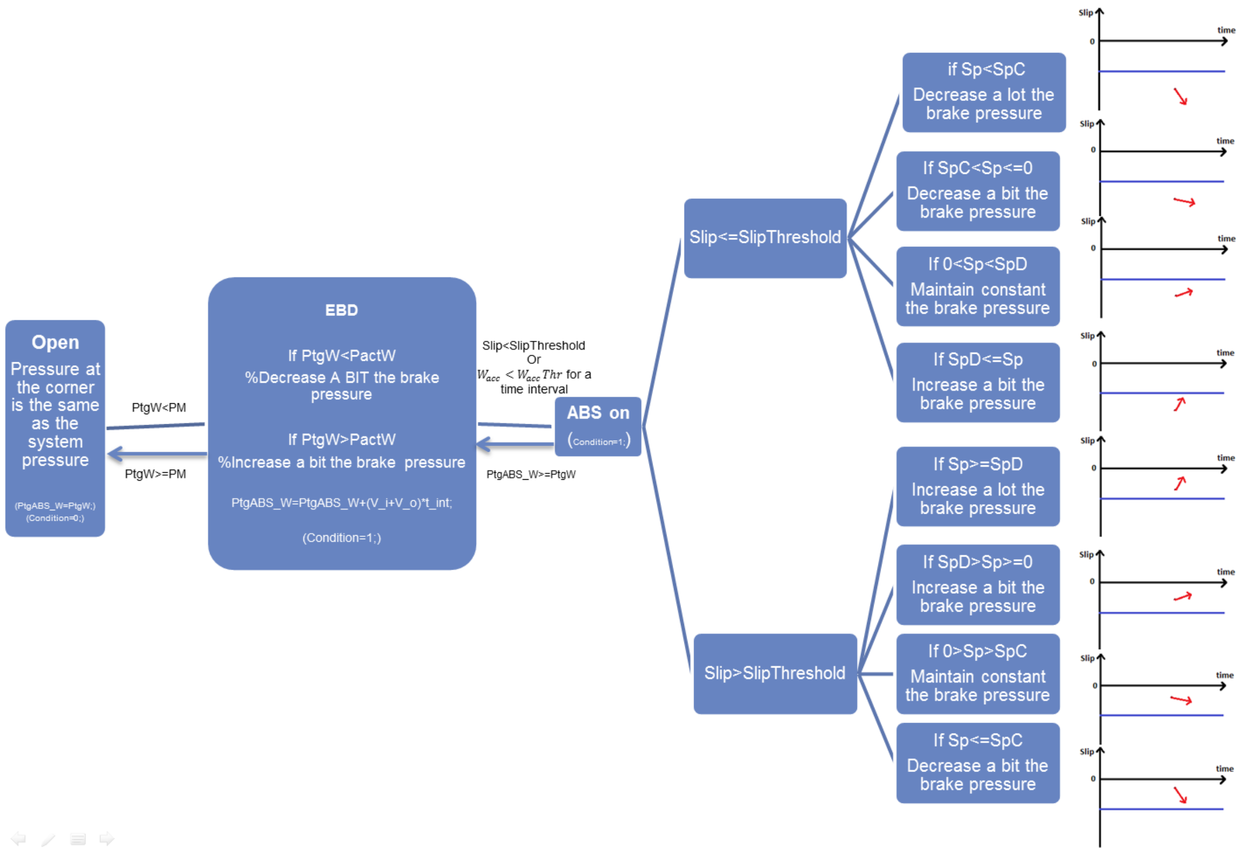


Figure 8.23 - Schematic representation of ABS rear valves logic

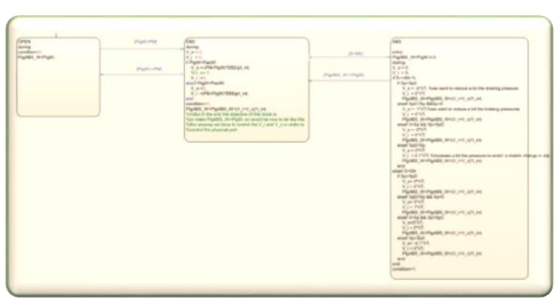


Figure 8.24 - Simulink - Stateflow rear valves logic

8.2Valve Physical Model

In the 'Physical Part' block was constructed the valve physical model, described below:

The Matlab-Simulink valve physical model was designed in a way that when the model is not on ABS or on in EBD (condition =0) the switch (figure 8.25) is the condition that makes the corner pressure equal to the main system pressure.

Instead, when the model is on ABS or EBD (condition =1) the switch is in the condition that makes the corner pressure equal to the pressure coming from the integrator, that is the one integrating the logics with V_i and V_o coming from the logical part. The integrator resets to main pressure when there is an increase of 'condition' in order to have continuity between when passing from not using the integrator to using it.

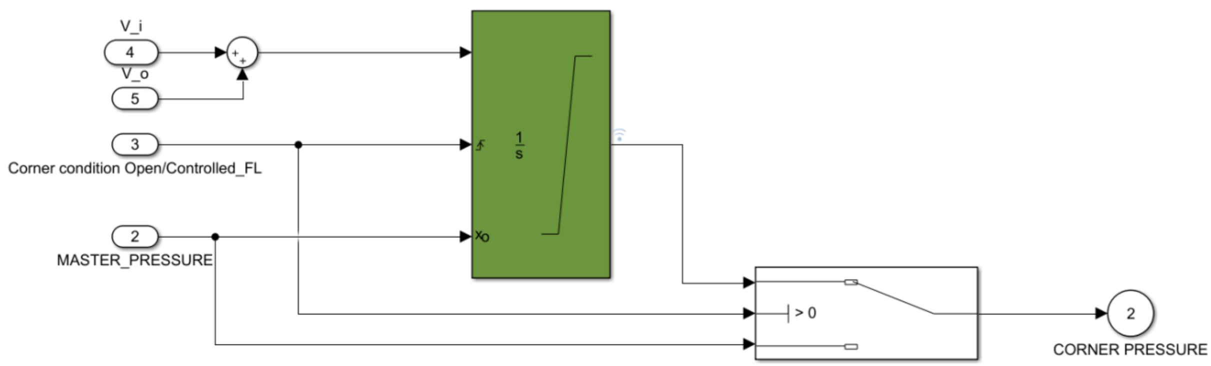


Figure 8.25 –Valve physical model blocks

9 Validation

One of the main objectives of this study is to validate the virtual model created, this means to input the same data as the experimental ones and obtain results of the vehicle dynamics similar to the real ones, also having the intermediate results close to the real values, like different pressures, wheel speed, vehicle acceleration, etc.

9.1 Tuned parameters

Some parameters had to be adjusted in order to correlate the results, the tuned parameters can be physical or logical.

Tuned **physical** parameters in the virtual model:

- Pedal deformation;
- Tires:
    - LMUX (grip) – Scale factor of F_x peak friction coefficient;
    - LKX – scale factor for F_x Slip stiffness;
- Road friction coefficient:
    - Adapting the wet mu coefficient;
- Change in center of gravity height;
- Gear shift;
- Brake pad's mu.

Tuned **logical** parameters in the virtual model:

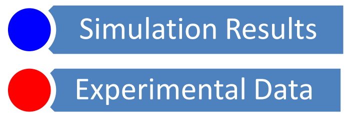
- Max_g = logical maximum deceleration (to line-parabola);
- Estimated wheel radius on logic;
- ABS logics:
    - V_i = inlet valve pressure rate;
    - V_o = outlet valve pressure rate;
    - V_gain_front = front multiplying factor in ABS logics (ex: V_o=-2. V_gain_front);
    - V_gain_rear = rear multiplying factor in ABS logics (ex: V_i=2. V_gain_rear);
    - SpC = negative threshold for the derivate of the slip;
    - SpD= positive threshold for the derivate of the slip;
    - Slip Threshold;

Maneuvers:

- The maneuvers were set as the experimental data in order to compare results, but as the model does not work well at low speeds, the simulation finishes at 20km/h.

9.2 Correlation Results

In this session the simulation results (in blue) are compared with the experimental data (in red).



Here is presented correlation of one maneuver for each car:

- AR 949 – Stelvio: Hard brake on dry surface with initial speed of 100 km/h;
- AR 952 – Giulia: Hard brake on wet surface with initial speed of 108km/h;

The following outputs were compared:

- Vehicle longitudinal acceleration;
- Estimated wheel speeds;
- Estimated vehicle speed;
- Brake pressure at the 4 corners;
- Main system pressure.

And the end the braking distance is presented.

<u>AR-949: Stelvio</u>

The results for the correlation of the hard brake with Stelvio on dry surface are the following:

- Estimated wheel speeds (figure 9.1.1):

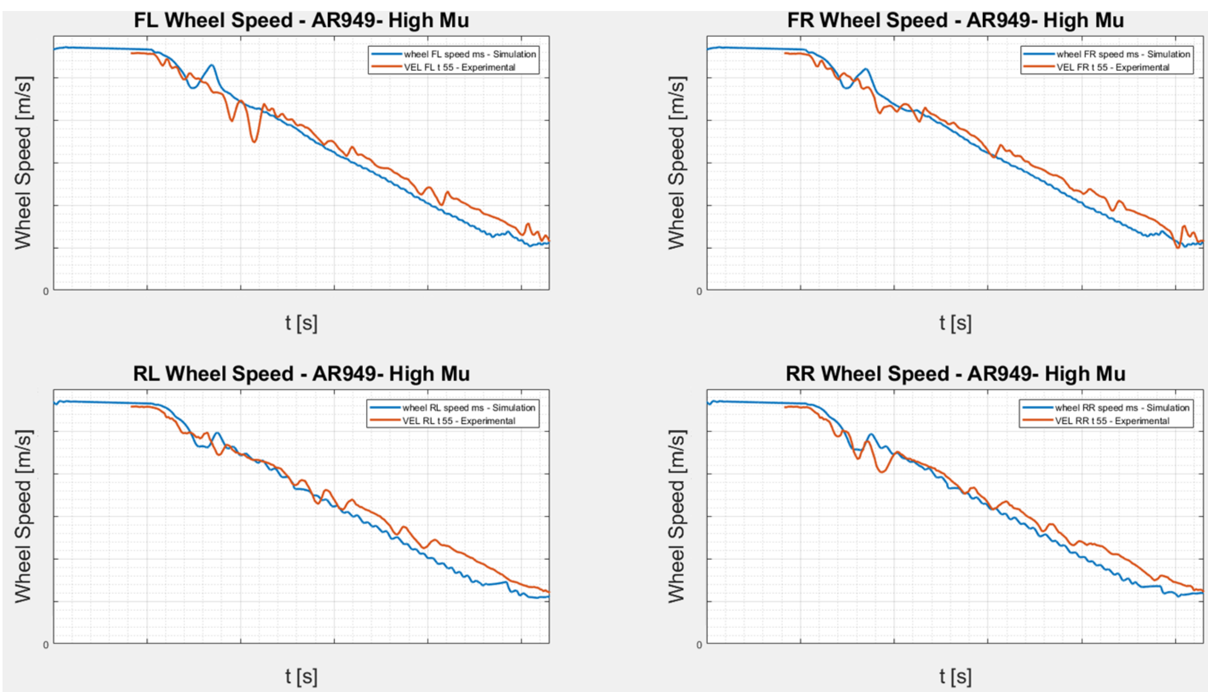


Figure 9.1.1 – Estimated wheel speed – AR949 on dry road

- Estimated speed (figure 9.1.2):

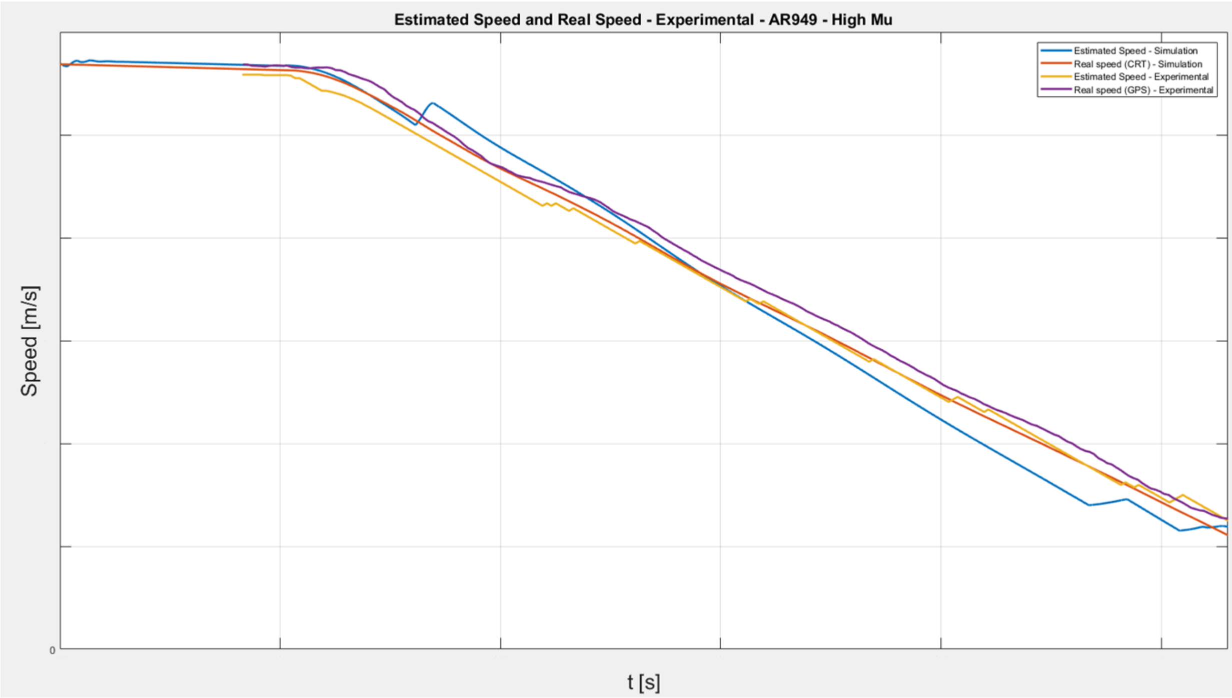


Figure 9.1.2 – Estimated vehicle speed – AR949 on dry road

In figure 9.1.3 the estimated speed is represented in yellow, the CarRealTime real speed represented in pink, and the four estimated wheel speed represented in purple, green, blue and red:

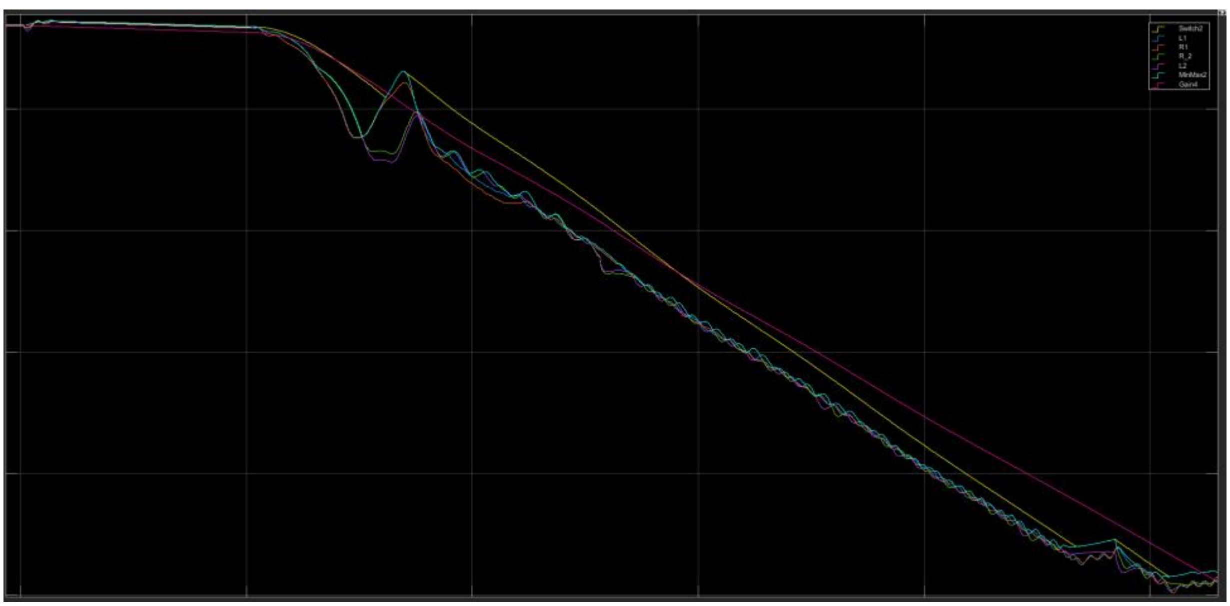


Figure 9.1.3 – Estimated vehicle speed (yellow) and estimated wheel speeds – AR949 on dry road

- Brake pressure at the four corners (figure 9.1.4):

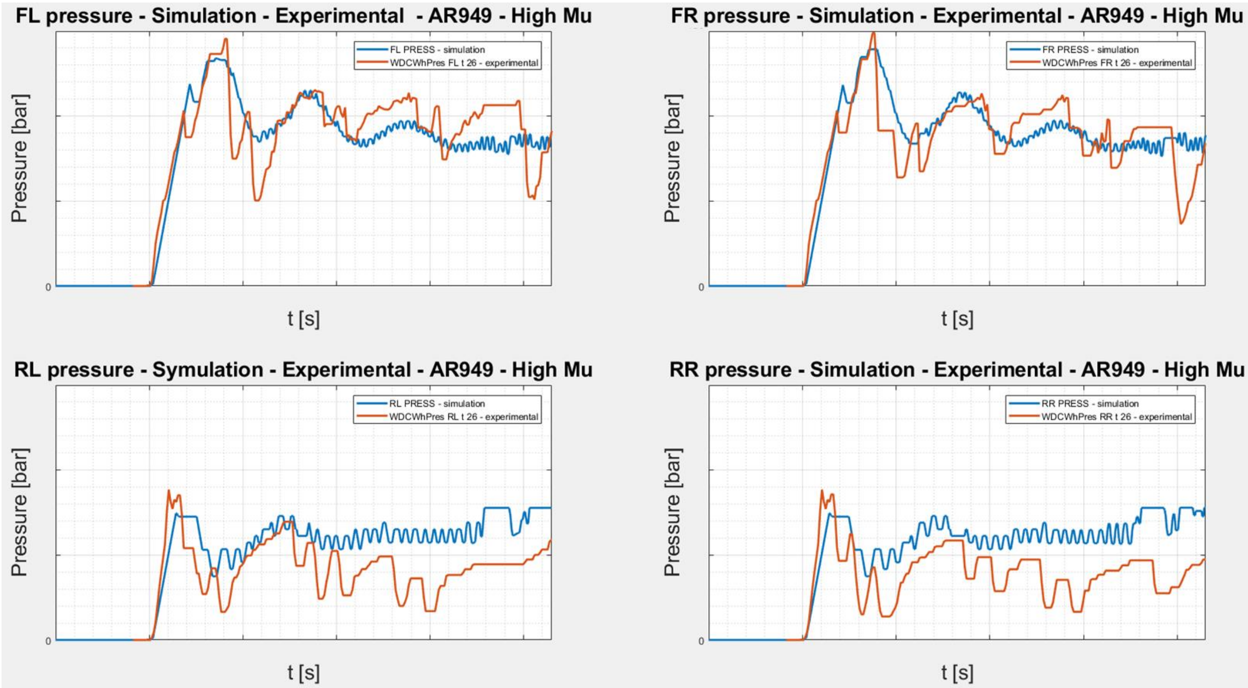


Figure 9.1.4 – Brake pressure on the four corners – AR949 on dry road

- Main system pressure (figure 9.1.5):

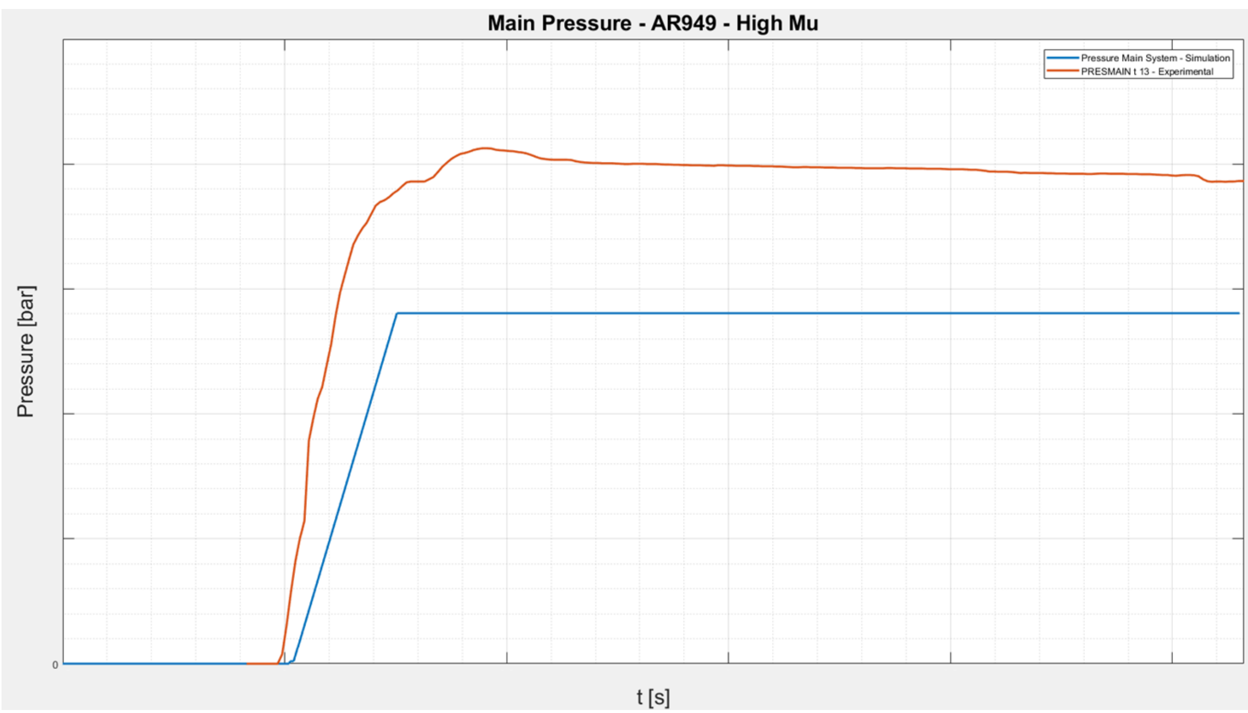


Figure 9.1.5 – Main System Pressure – AR949 on dry road

- Vehicle longitudinal acceleration (figure 9.1.6):

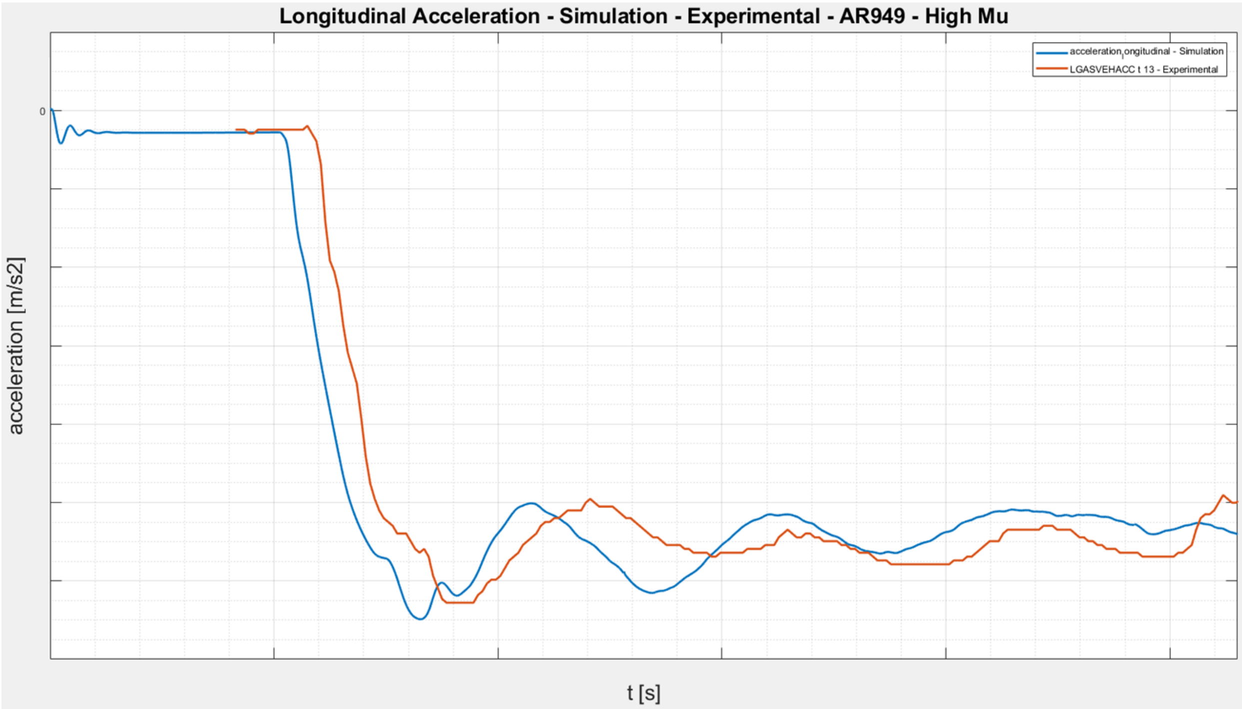


Figure 9.1.6 – Vehicle longitudinal acceleration – AR949 on dry road

- Braking distance (figure 9.1.7):

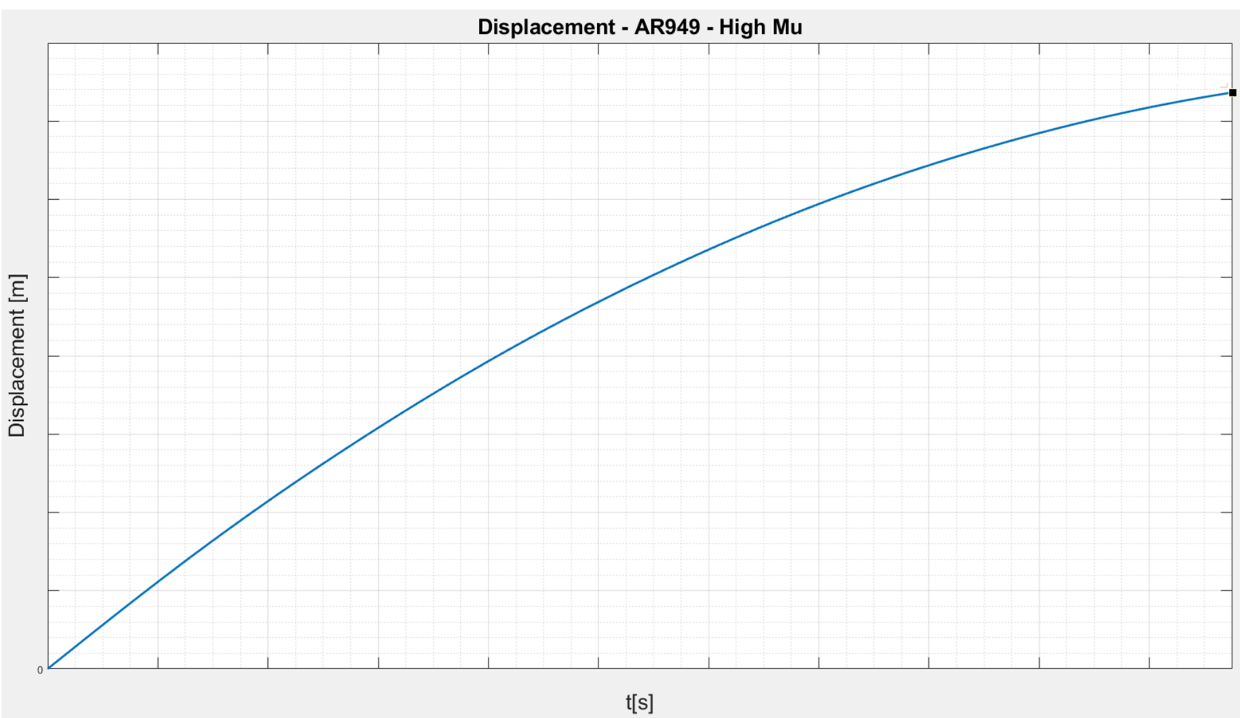


Figure 9.1.7 – Braking distance – AR949 on dry road

<u>AR-952: Giulia</u>

The results for the correlation of the hard brake with Giulia on wet surface are the following:

- Estimated wheel speeds (figure 9.2.1):

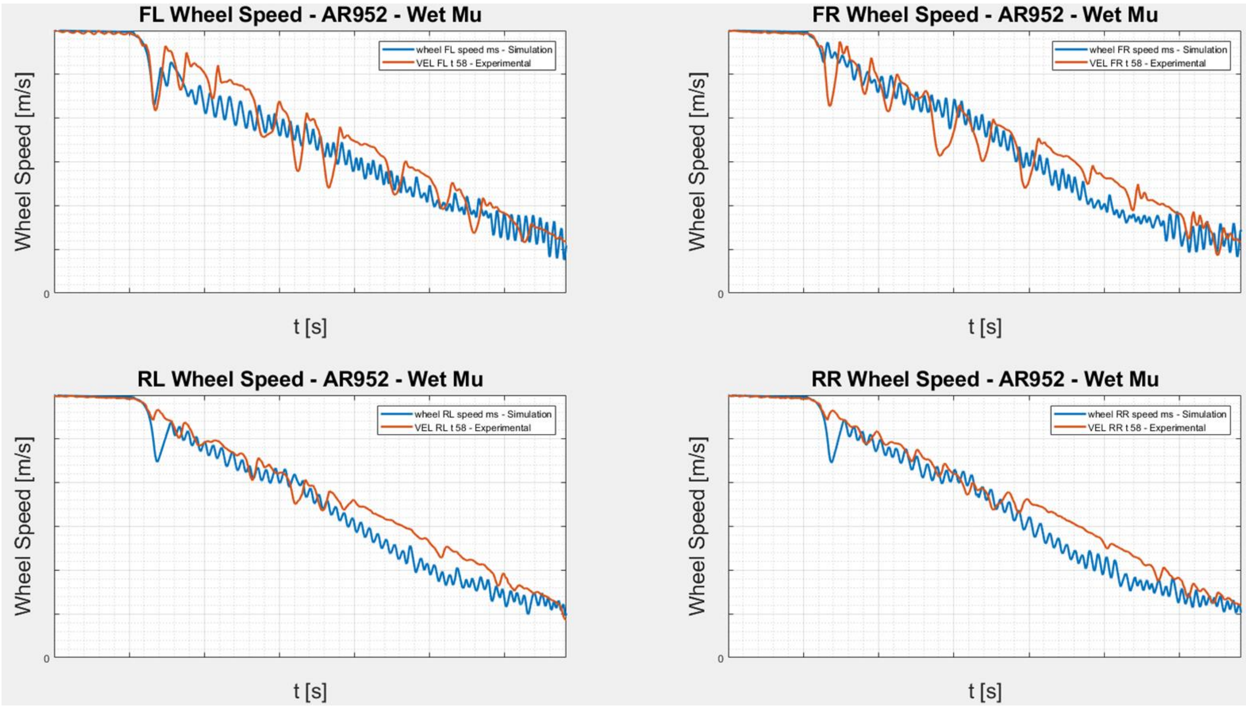


Figure 9.2.1 – Estimated wheel speed – AR952 on wet road

- Estimated speed (figure 9.2.2):

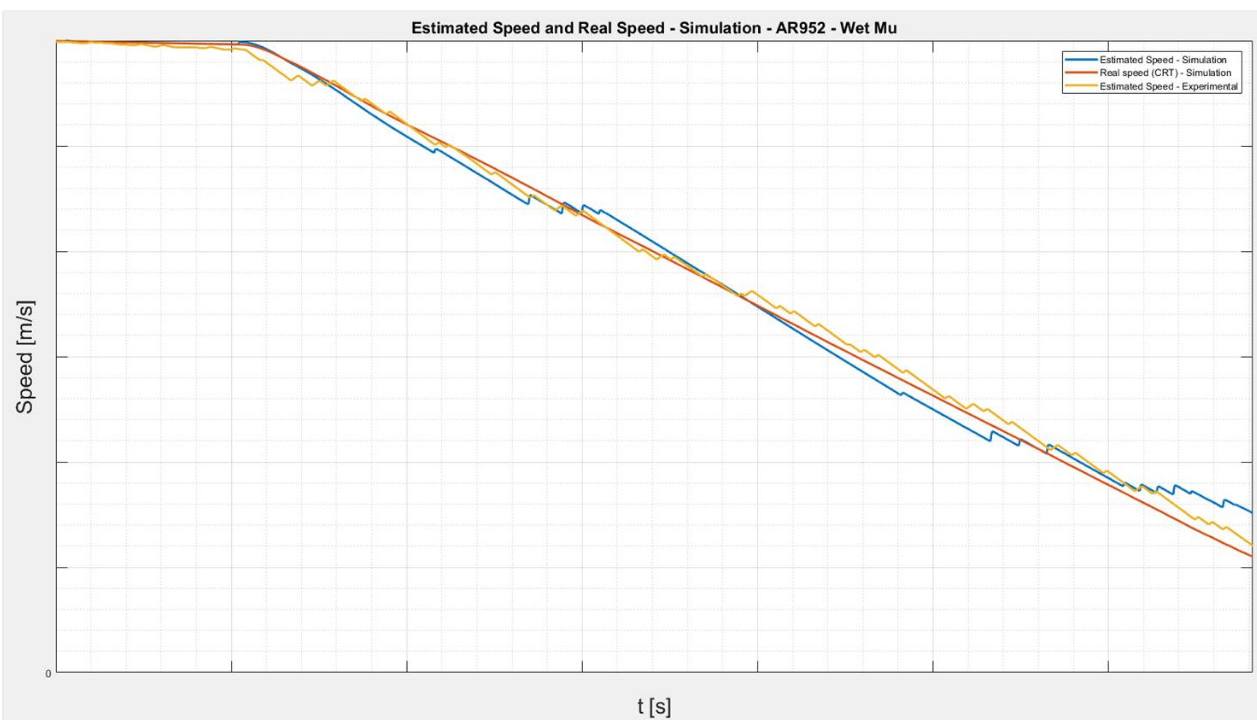


Figure 9.2.2 – Estimated vehicle speed – AR952 on wet road

In figure 9.2.3 the estimated speed is represented in yellow, the CarRealTime real speed represented in pink, and the four estimated wheel speed represented in purple, green, blue and red:

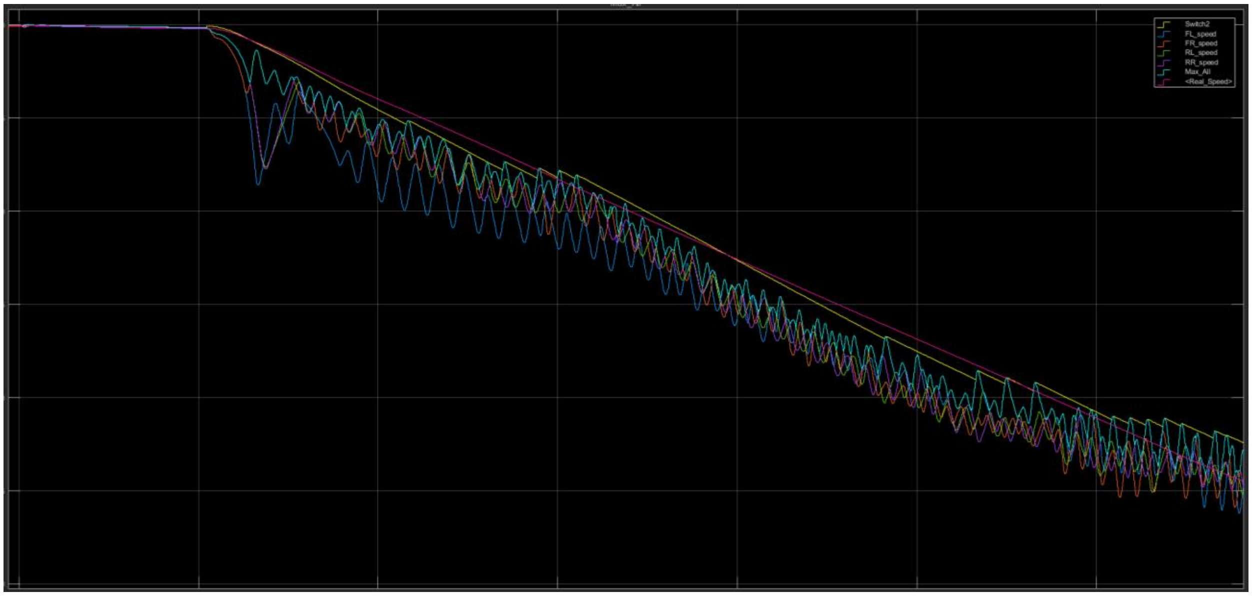


Figure 9.2.3 – Estimated vehicle speed (yellow) and estimated wheel speeds – AR952 on wet road

- Brake pressure at the four corners (figure 9.2.4):

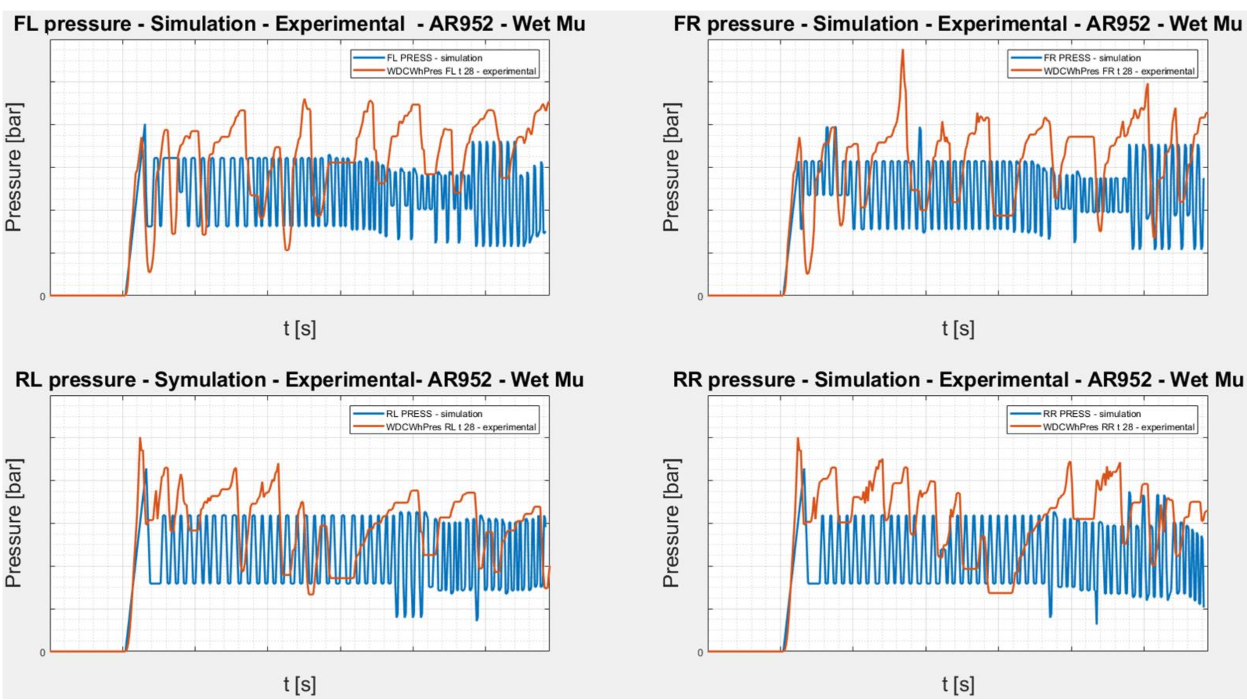


Figure 9.2.4 – Brake pressure on the four corners – AR952 on wet road

- Main system pressure (figure 9.2.5):

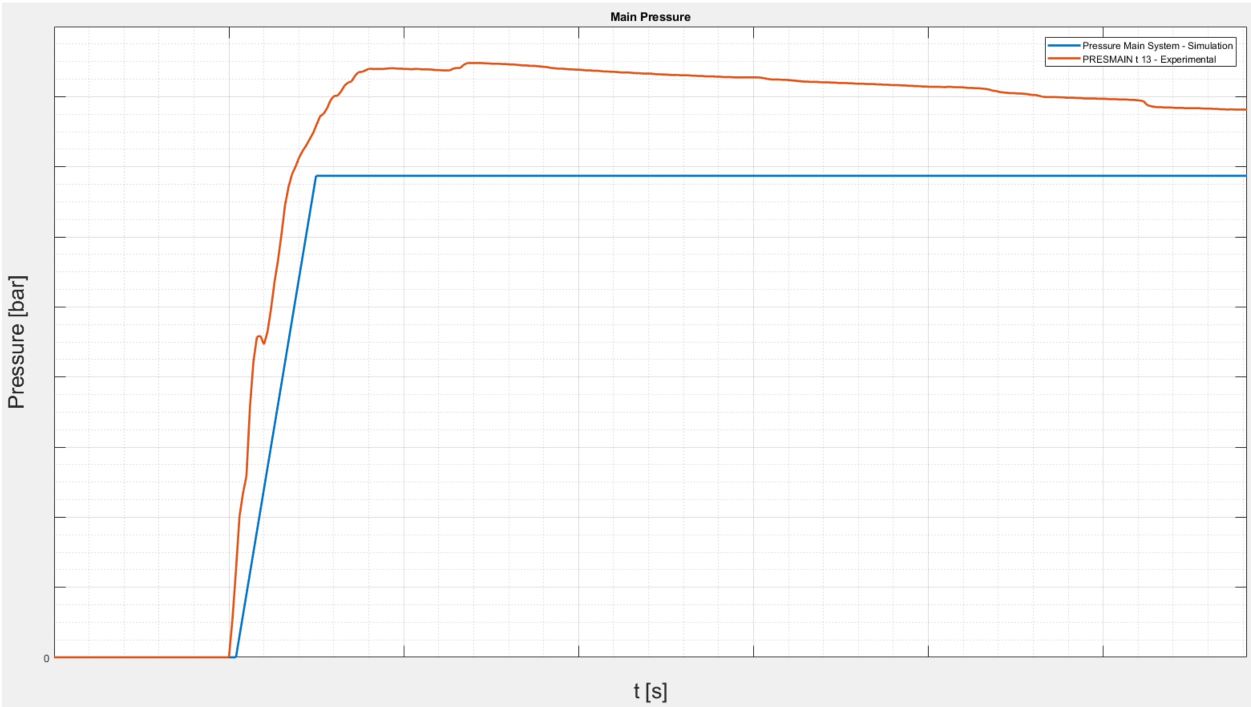


Figure 9.2.5 – Main System Pressure – AR952 on wet road

- Vehicle longitudinal acceleration (figure 9.2.6):

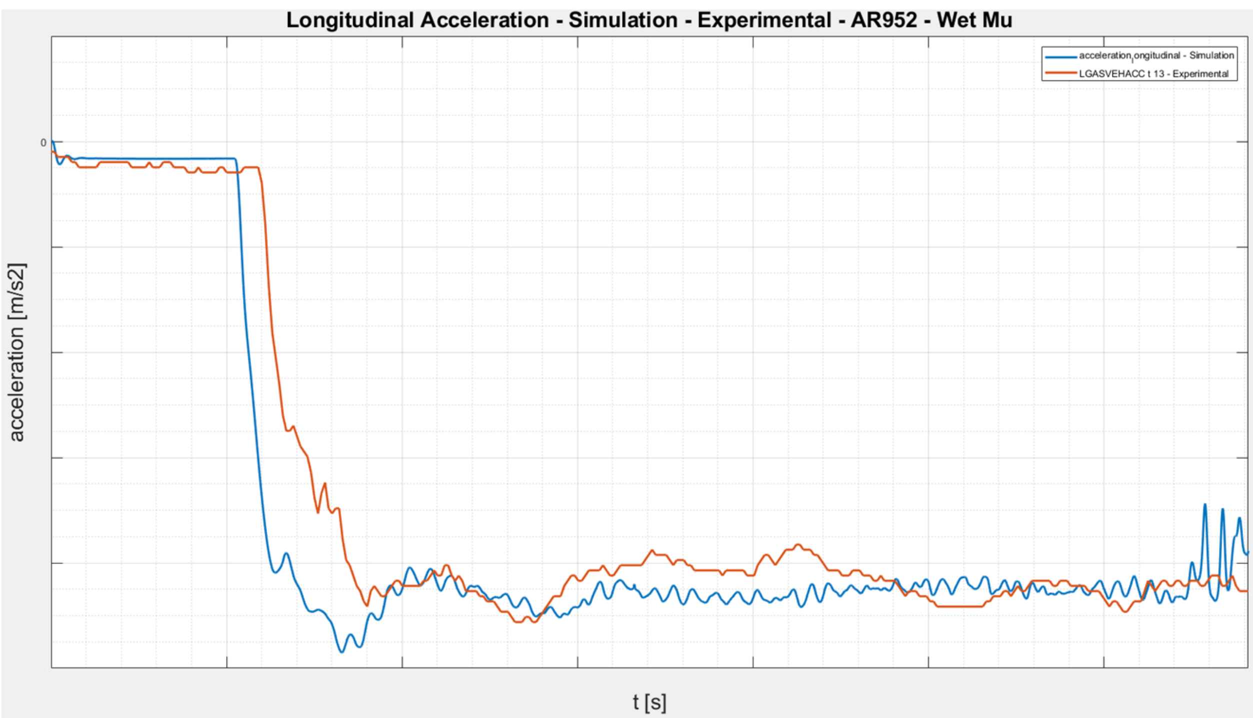


Figure 9.2.6 – Vehicle longitudinal acceleration – AR952 on wet road

- Braking distance (figure 9.2.7):

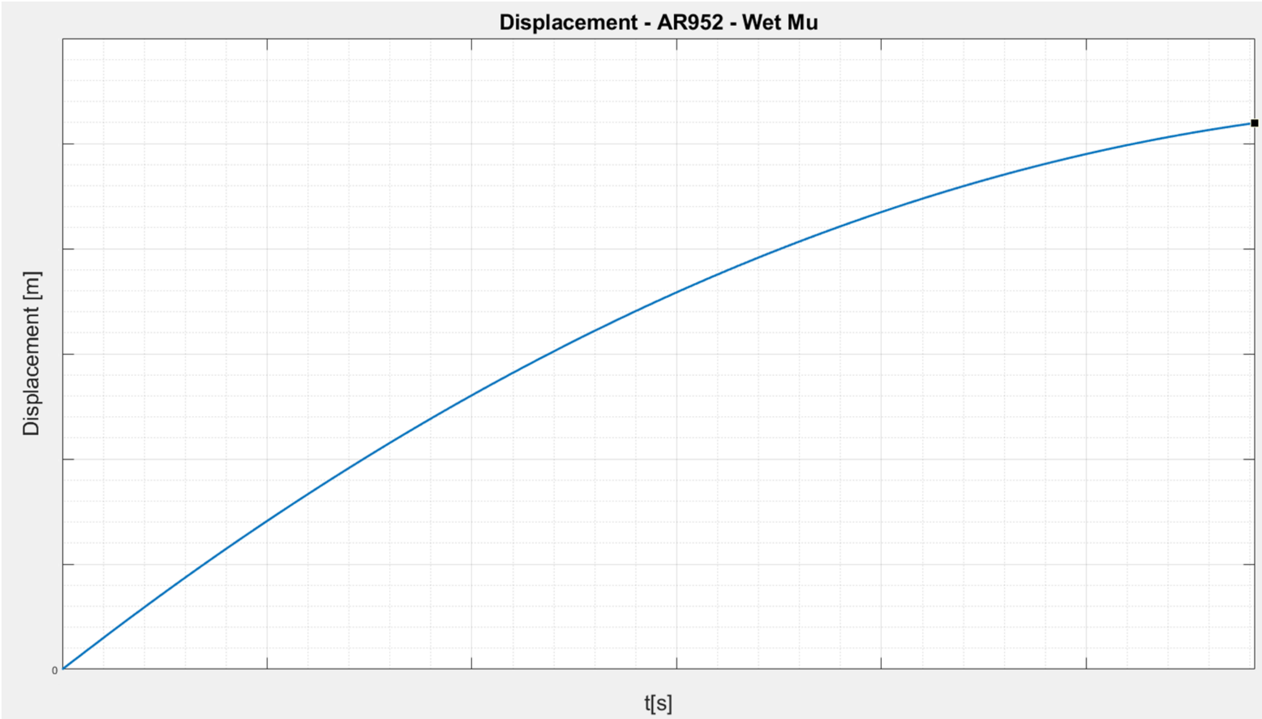


Figure 9.2.7 – Braking distance – AR952 on wet road

10 Sensitivity Analysis

The objective of this section is to study the influence of certain parameters on the braking behavior.

The sensitivity analysis was made on the AR 949 – Stelvio, on hard brake maneuver at high mu (initial speed Vo=100km/h), and varying the following parameters:

- LMUX - Scale factor of F_x peak friction coefficient: +/-10%;
- LKX - Scale factor of F_x slip stiffness: +/-10%;
- Pad's Mu - Brake pad's coefficient of friction: +/-20%;
- CarRealTime Real Speed
- Slip Threshold +/-10%;
- Tire's Structural Longitudinal Stiffness +/-10%;

So for each parameter variation, the results were compared with the baseline results (the validated result presented on the previous session) and with the experimental results as shown below. The results in black are the experimental ones, the blue are the baseline ones and the red are the ones changing the parameters of the sensitivity analysis.

Baseline

Parameter variation

Experimental data

For all the parameters variation were compared:

- Braking distance;
- Corner pressure;
- Longitudinal Acceleration;
- Wheel estimated speed;
- Vehicle estimated speed.

As the first example, it's shown the comparison between the results when substituting the vehicle speed estimator to the CarRealTime vehicle longitudinal speed and the standard simulation, which uses the vehicle speed estimator.

10.1 CarRealTime Real Speed

As in the model was used a speed estimator that estimates the vehicle speed based on the wheel speeds, it is probable that it brings errors to the final braking distance, so in this sensitivity analysis the estimated speed was substituted for the real CarRealTime simulation speed. And as expected the braking distance reduced, but not also in such a significant way.

The results of the comparisons are shown in a graphical way:

- Estimated wheel speed (figure 10.1.1):

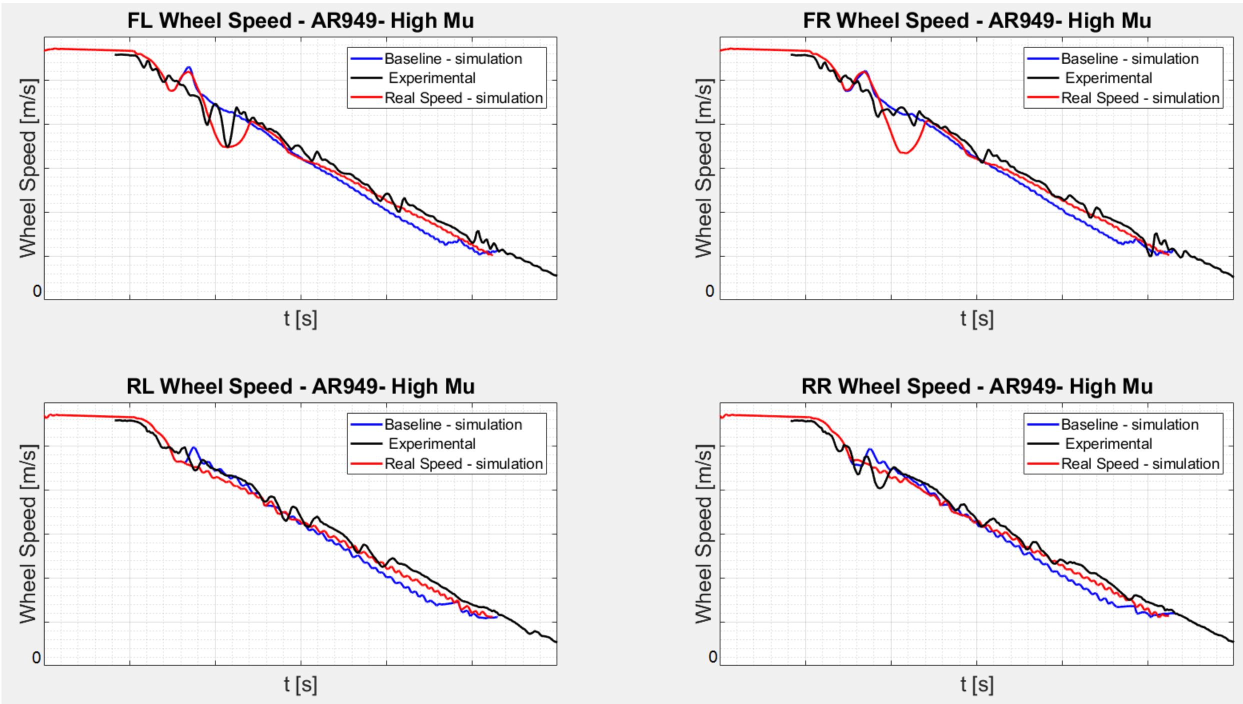


Figure 10.1.1 – Estimated wheel speed – Baseline vs CRT real speed

- Estimated vehicle speed (figure 10.1.2):

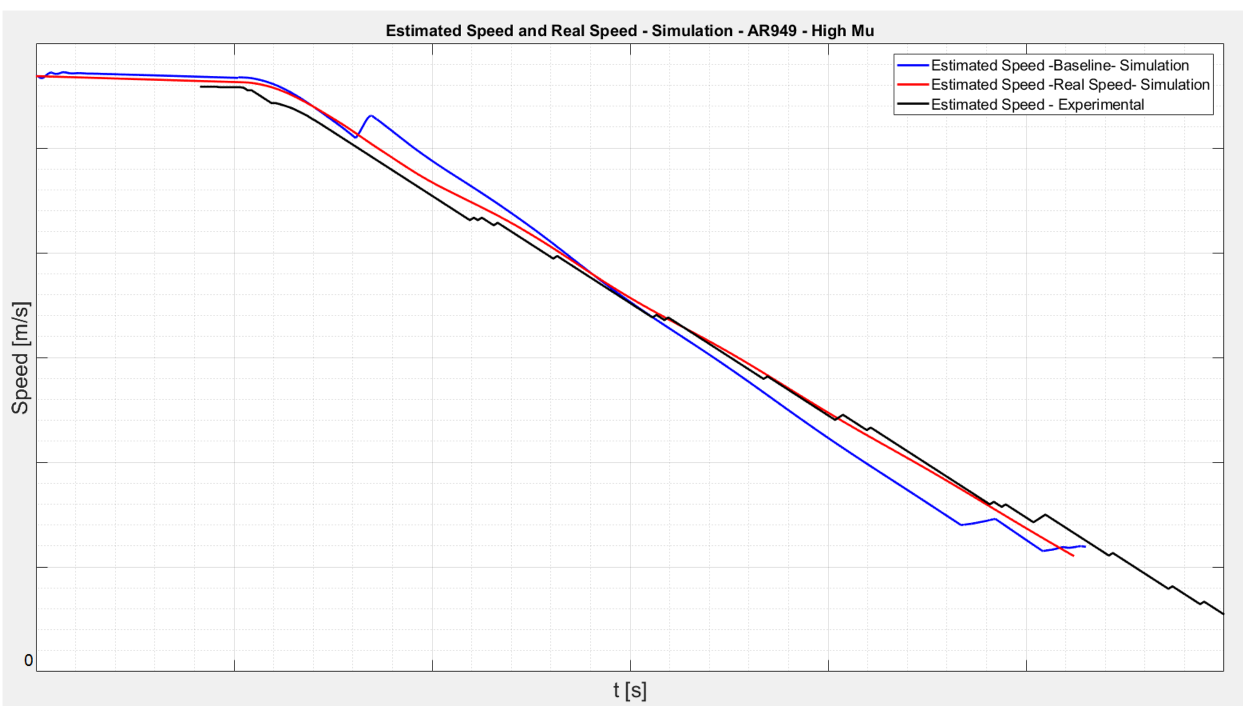


Figure 10.1.2 – Estimated vehicle speed – Baseline vs CRT real speed

- Pressure on the four corners (figure 10.1.3): (FL- front left; FR- front right; RL- rear left; RR – rear right)

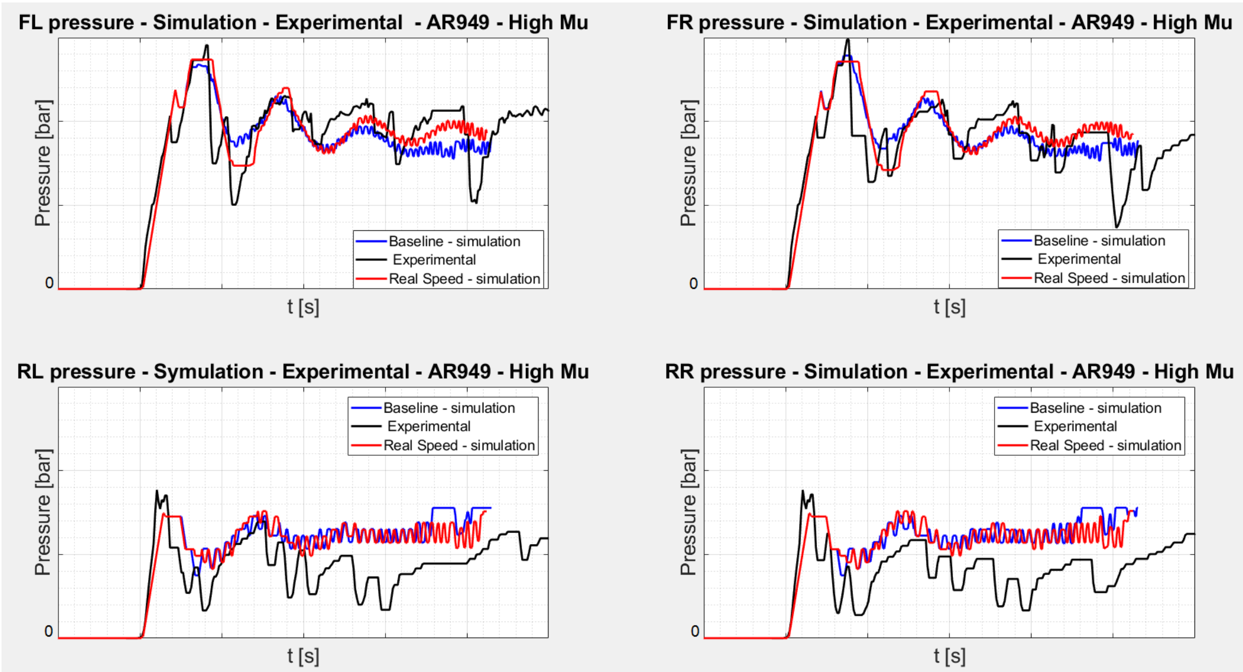


Figure 10.1.3 – Brake pressure on the four corners – Baseline vs CRT real speed

- Longitudinal acceleration (figure 10.1.4):

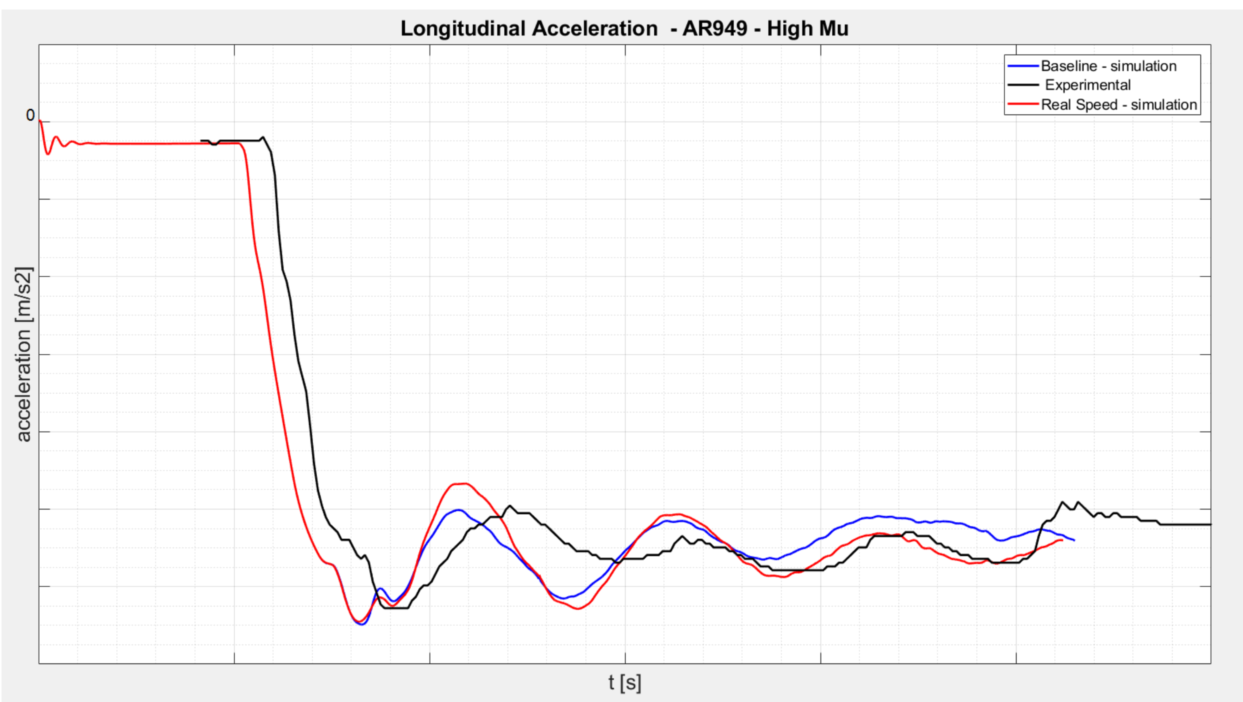


Figure 10.1.4 – Vehicle longitudinal acceleration – Baseline vs CRT real speed

- Braking distance (figure 10.1.5): It Decreased 0.30%.

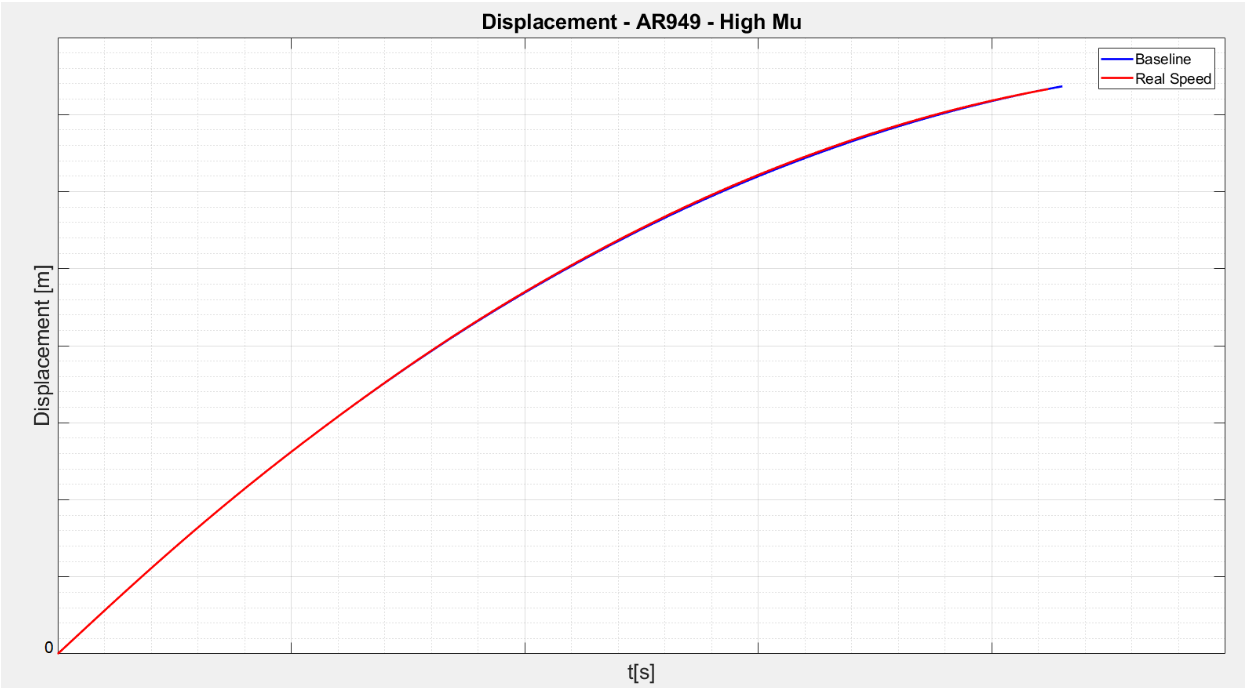


Figure 10.1.5 – Braking distance – Baseline vs CRT real speed

The substitution brings less errors to the calculation, so as expected the introduction of the real speed of CarRealTime made the estimated wheel speeds, estimated vehicle speed and the front brake pressures closer to the experimental results.

10.2 LMUX - Scale factor of F_x peak friction coefficient:

The scale factor of F_x friction coefficient (LMUX) was changed of +/-10% in the tire's file '.tir' inside VI-CarRealTime environment then run the in the Matlab-Simulink environment.

As expected, increasing the scale factor F_x peak friction coefficient (LMUX) the brake force on each tire is bigger (figure 10.2), allowing the vehicle to stop in a smaller distance. Instead, while decreasing the LMUX the brake force on each tire is smaller, increasing the braking distance.

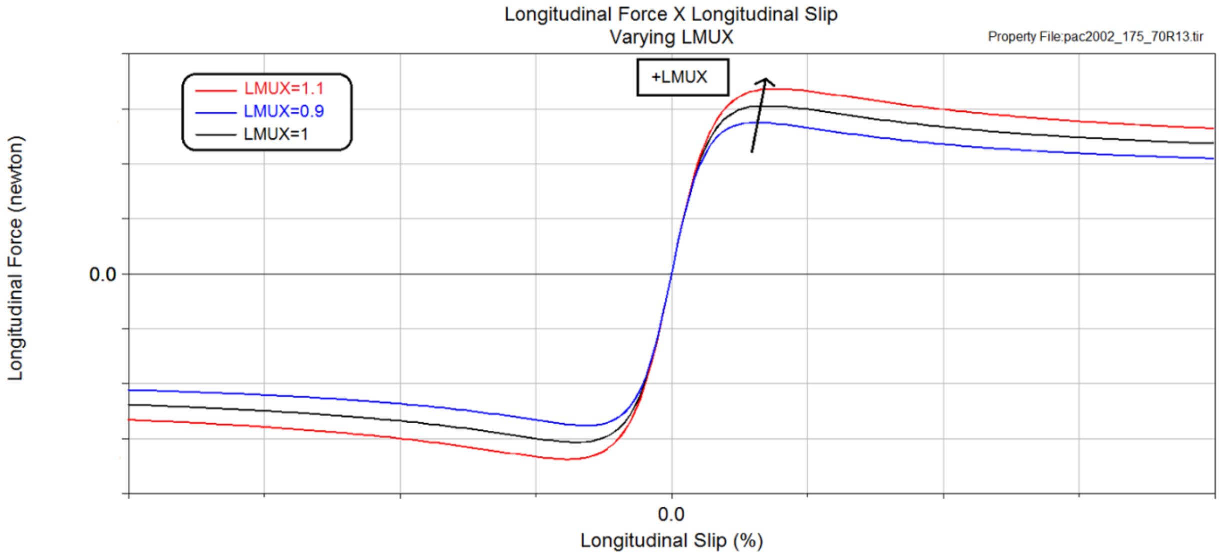


Figure 10.2 - Curves Fx(σ) for 175/70 R 15 tire obtained for different values of LMUX.

The results of the comparisons are shown in a graphical way for +/-10% LMUX:

<u>+10%LMUX:</u>

- Estimated wheel speeds (figure 10.3.1):

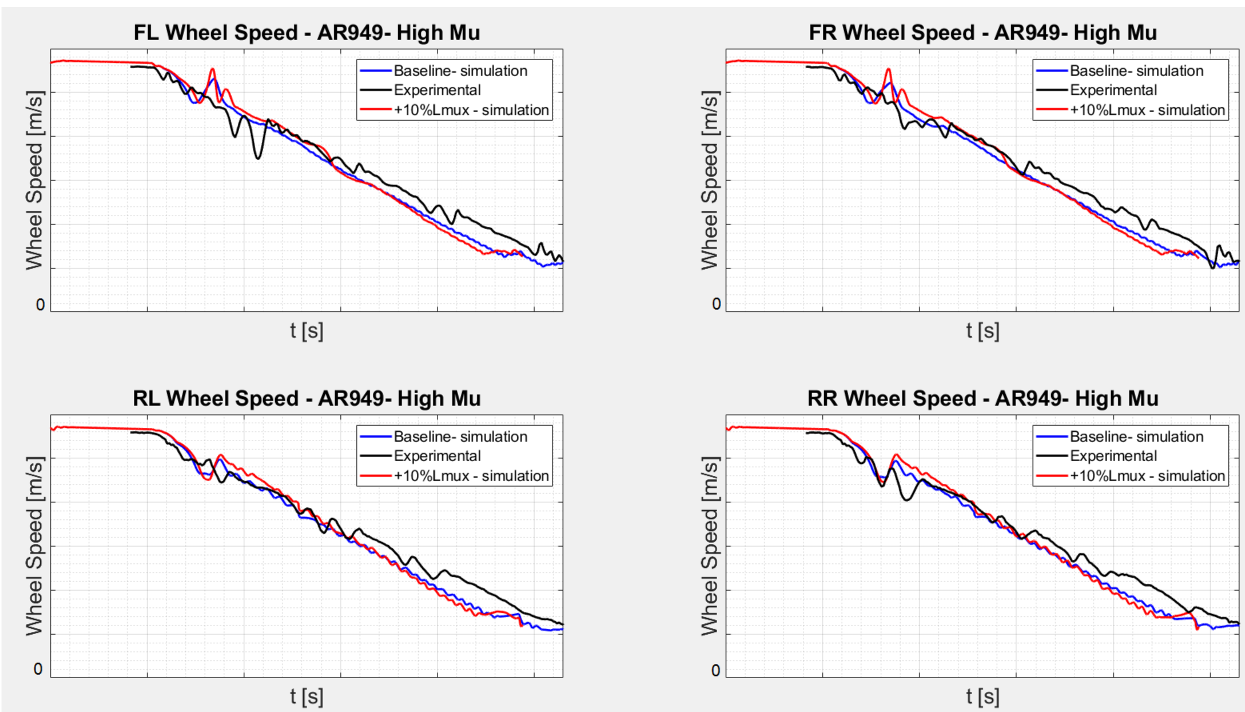


Figure 10.3.1 – Estimated wheel speed – Baseline vs +10%LMUX

- Brake pressure at the four corners (figure 10.3.2):

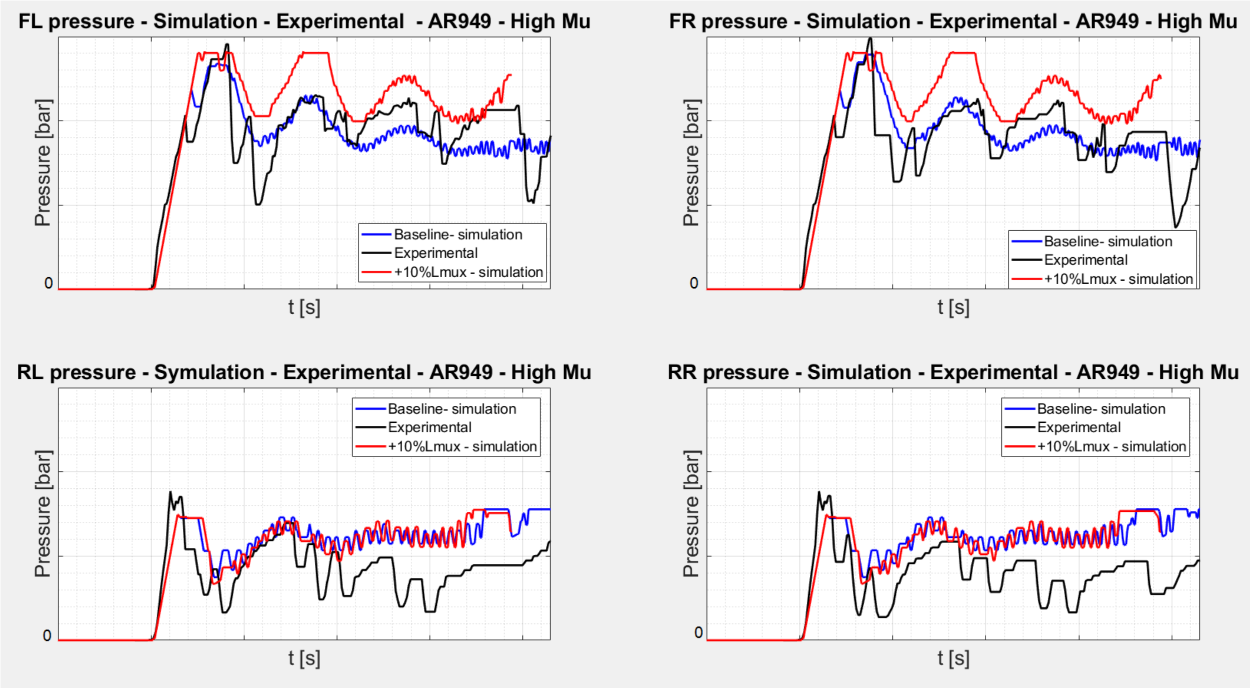


Figure 10.3.2 – Brake pressure on the four corners – Baseline vs +10%LMUX

- Vehicle longitudinal acceleration (figure 10.3.3):

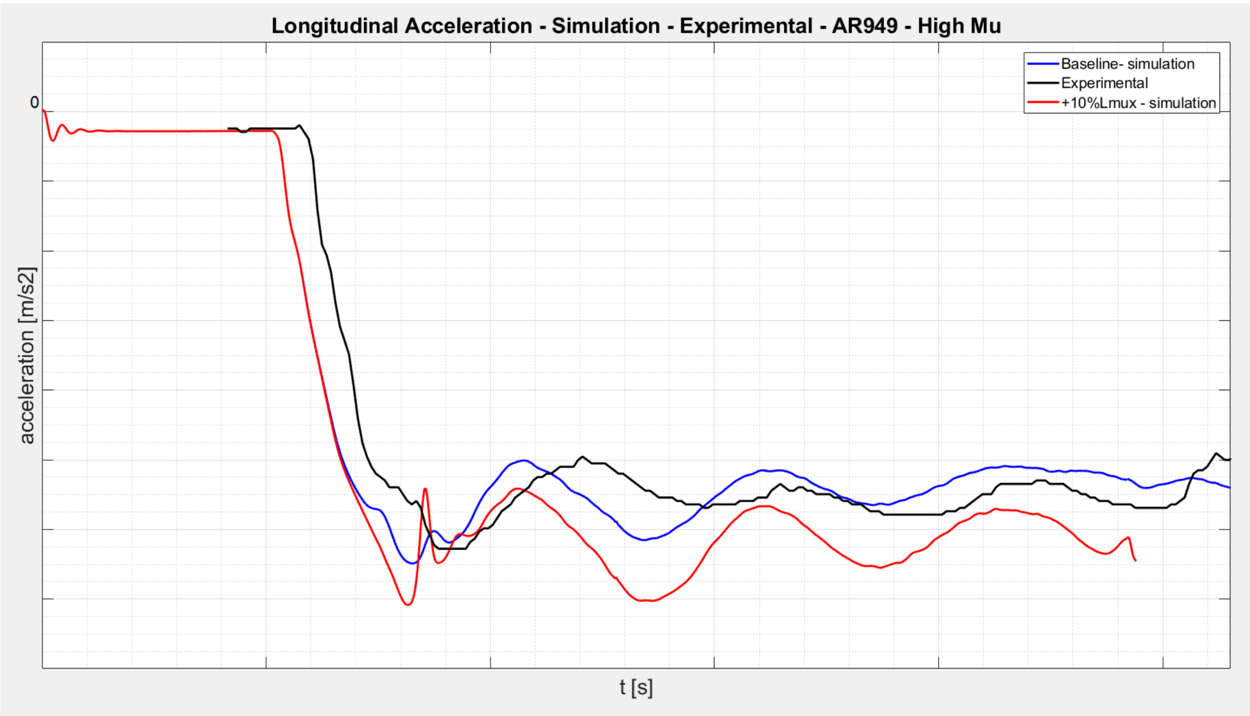


Figure 10.3.3 – Vehicle longitudinal acceleration – Baseline vs +10%LMUX

- Braking distance (figure 10.3.4): It decreased 8.28%.

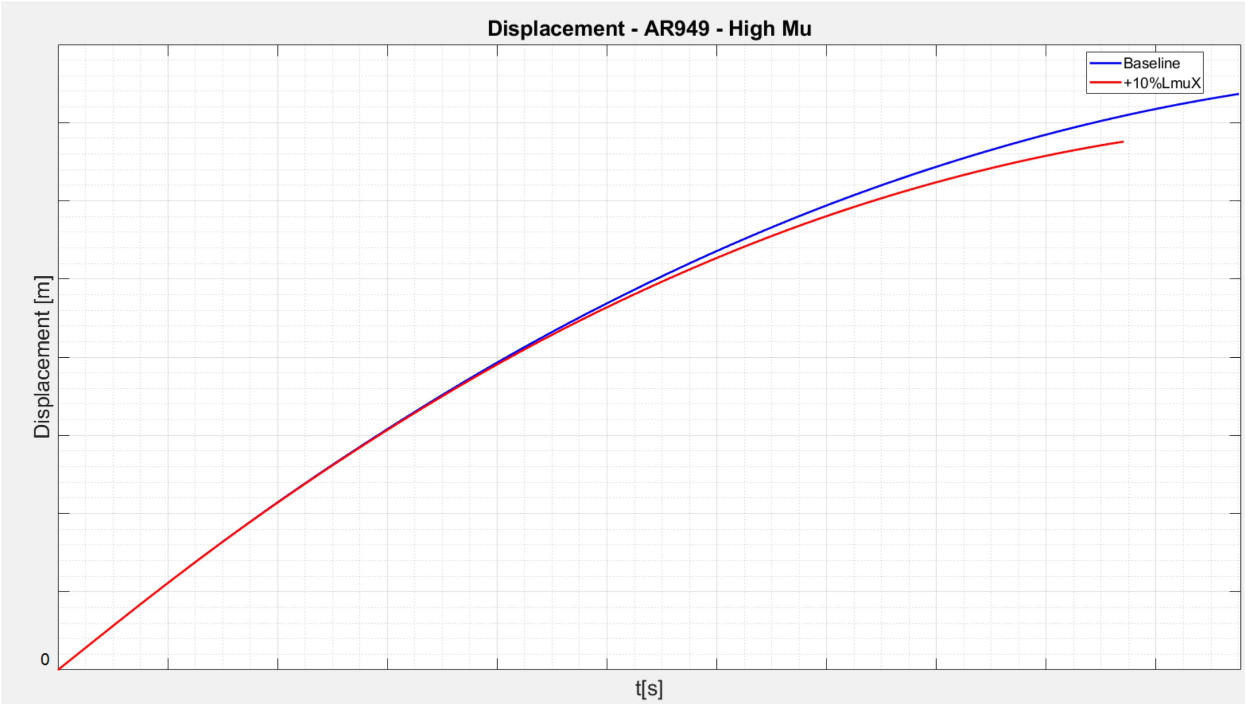


Figure 10.3.4 – Braking distance – Baseline vs +10%LMUX

<u>-10%LMUX:</u>

- Estimated wheel speeds (figure 10.4.1):

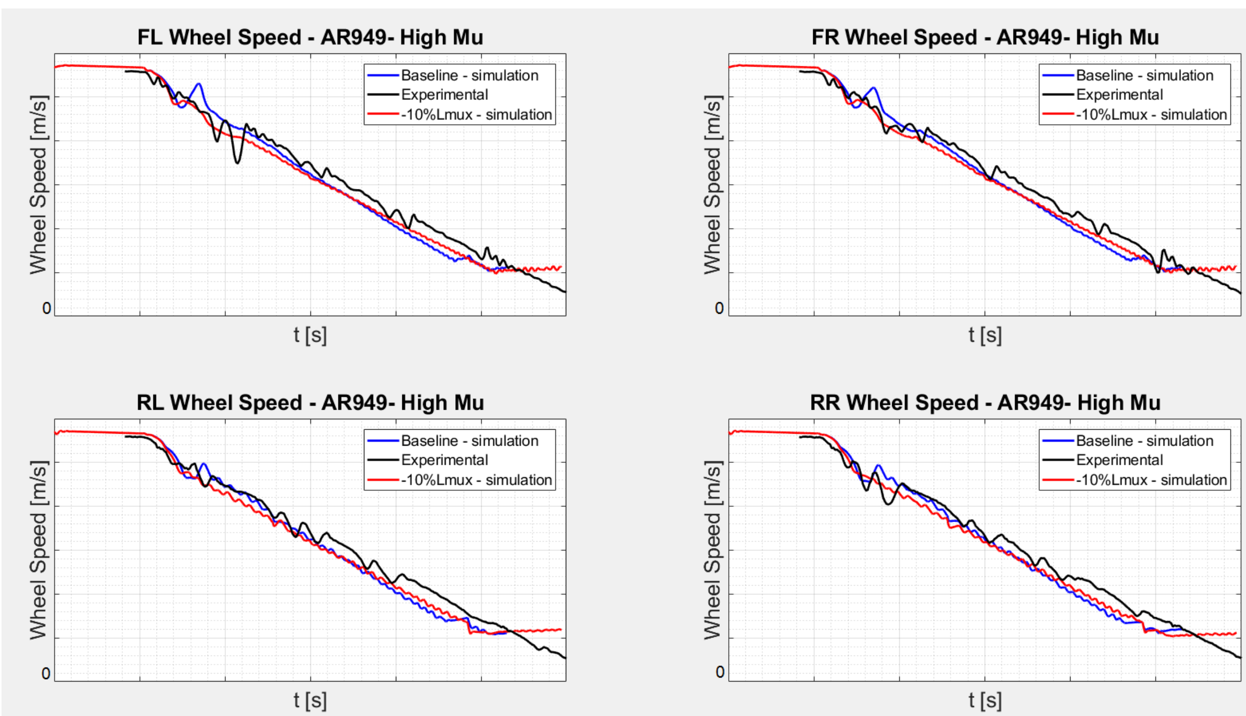


Figure 10.4.1 – Estimated wheel speed – Baseline vs -10%LMUX

- Brake pressure at the four corners (figure 10.4.2):

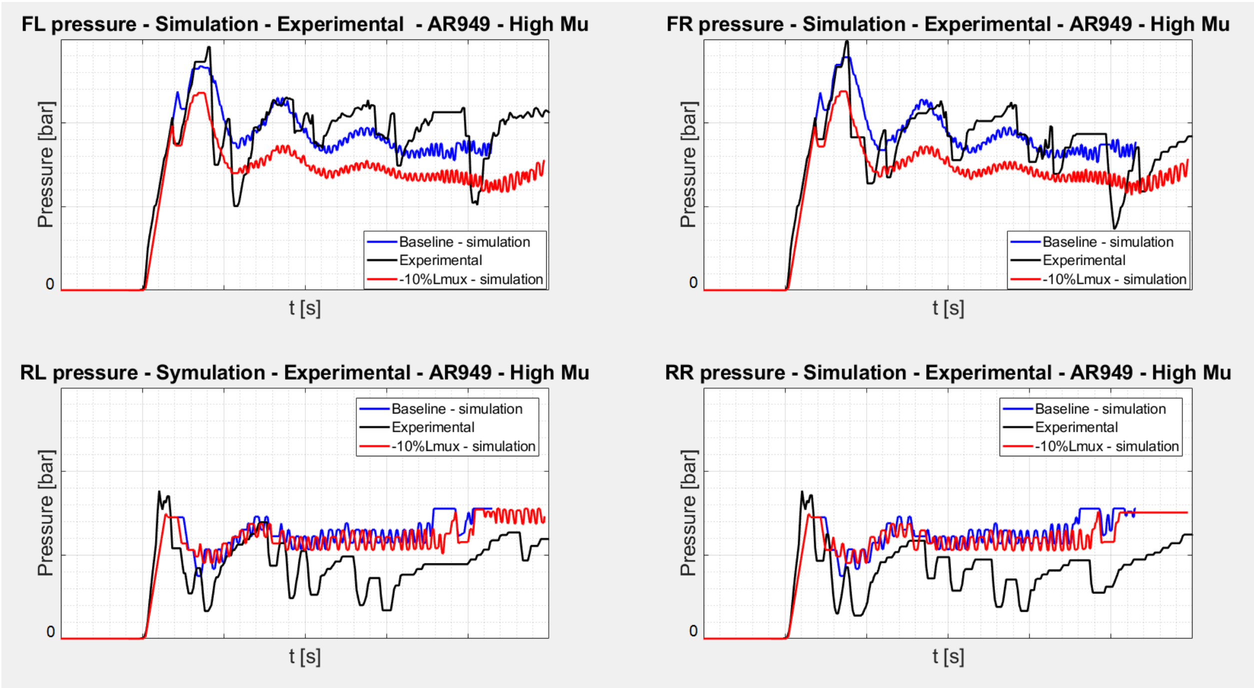


Figure 10.4.2 – Brake pressure on the four corners – Baseline vs -10%LMUX

- Vehicle longitudinal acceleration (figure 10.4.3):

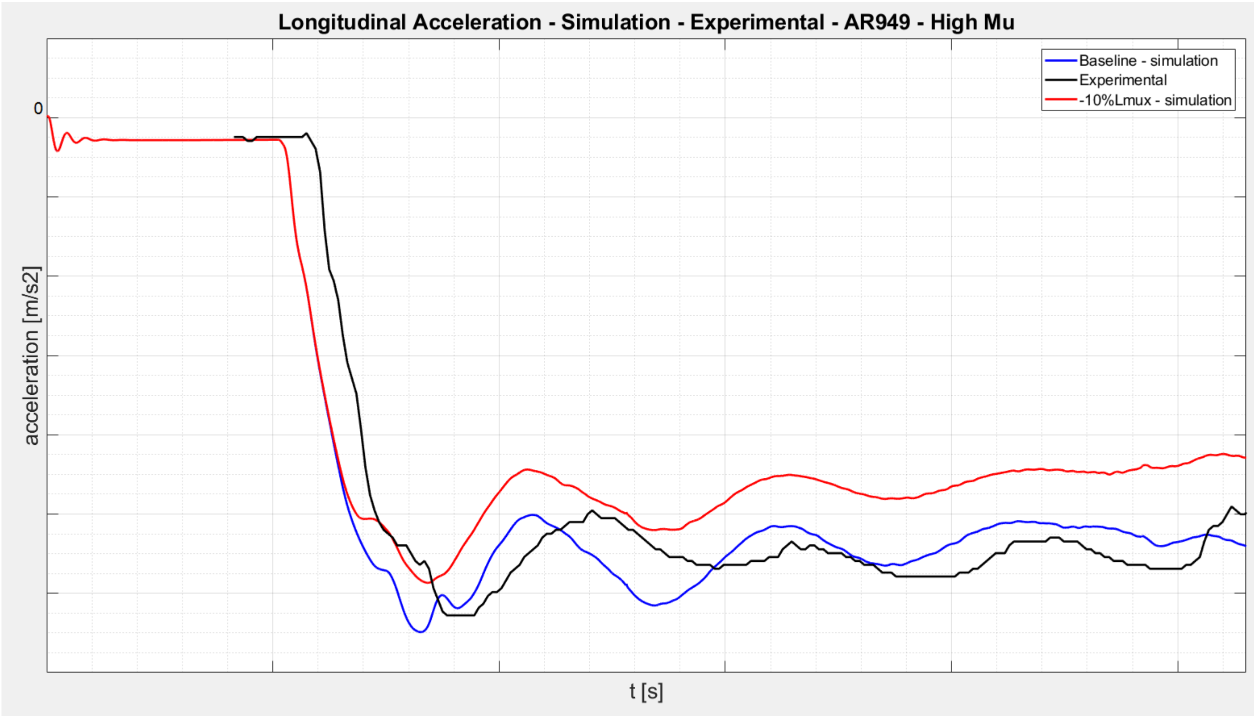


Figure 10.4.3 – Vehicle longitudinal acceleration – Baseline vs -10%LMUX

- Braking distance (figure 10.4.4): It increased 13.44%.

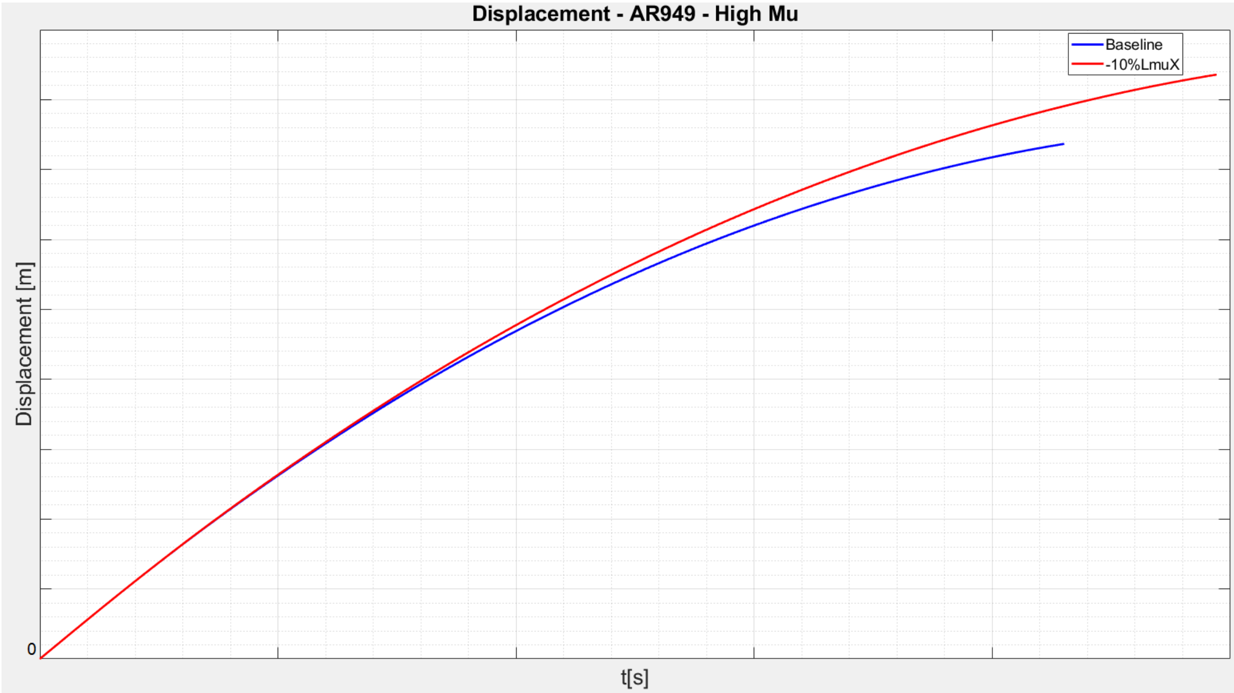


Figure 10.4.4 – Braking distance – Baseline vs -10%LMUX

10.3 LKX - Scale factor of F_x slip stiffness:

The longitudinal force F_x that the wheel exchanges with the road is a function of σ. It vanishes then $\sigma = 0$ (free rolling conditions) to increase almost linearly for small values of σ.

$$F_x = C_\sigma . \sigma$$

Where slip stiffness (or longitudinal stiffness) is a constant and can be defined as:

$$C_\sigma = \left(\frac{\delta F_x}{\delta \sigma}\right)_{\sigma=0}$$

Outside this range, its absolute value decreases in braking up to the value of σ=-1, which characterizes free sliding (locking of the wheel). In driving the force decreases above the stated range but σ can have any positive value, up to infinity when the wheel spins while the vehicle is not moving.

The figure 10.5 represents the influence of the scale factor of F_x slip stiffness (LKX) on the graph of tire's longitudinal force versus the longitudinal slip, the arrow indicates the consequence of increasing LKX, in this example increasing and decreasing LKX of 15%.

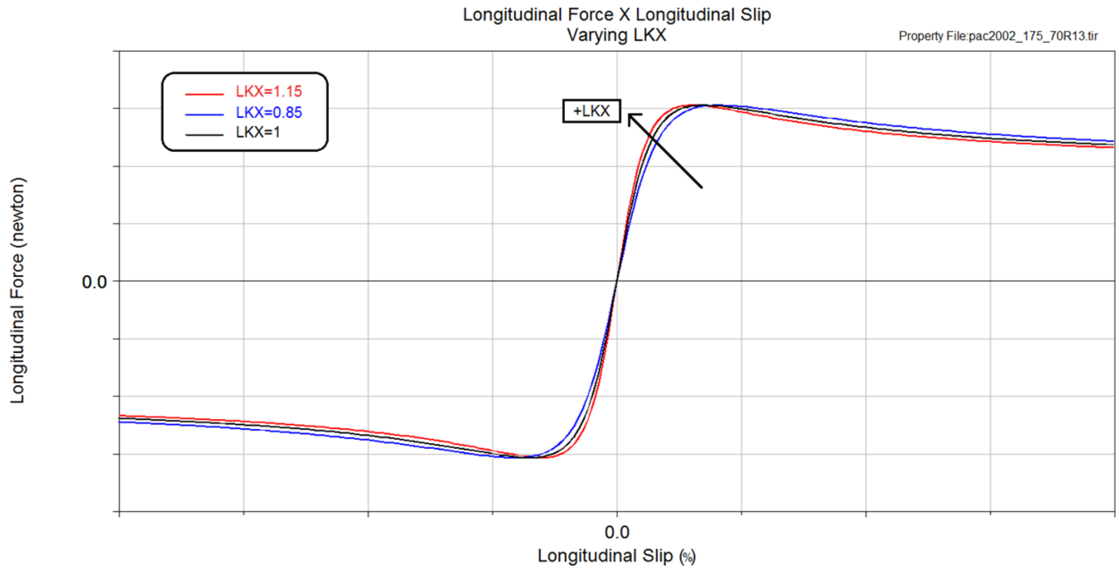


Figure 10.5 - Curves Fx(σ) for 175/70 R 15 tire obtained for different values of LKX.

The sensitivity analysis was made varying +/- 10% the LKX on the tire used in the AR 949 simulations.

Increasing and decreasing of 10% scale factor of F_x slip stiffness, the braking distance increased. This means that for the Slip threshold chosen, the baseline's LKX is a good value.

The results of the comparisons are shown in a graphical way for +/-10% LKX:

+10%LKX:

- Estimated wheel speeds (figure 10.6.1):

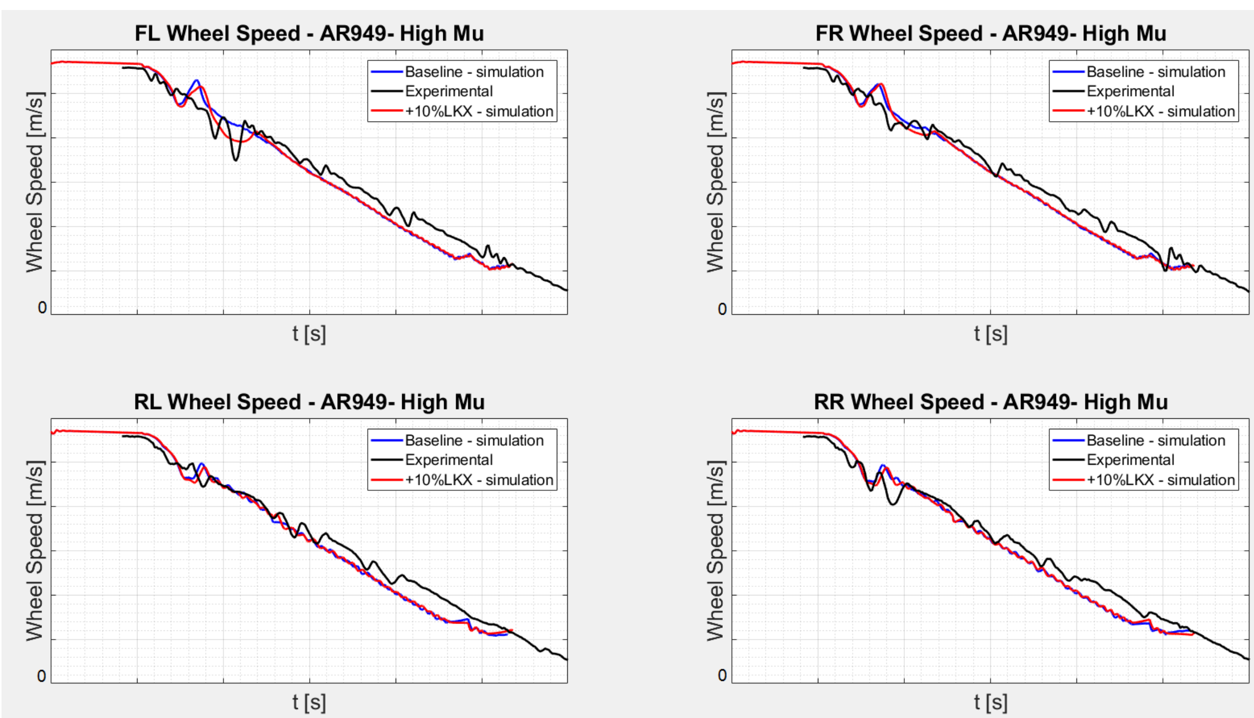


Figure 10.6.1 – Estimated wheel speed – Baseline vs +10%LKX

- Brake pressure at the four corners (figure 10.6.2):

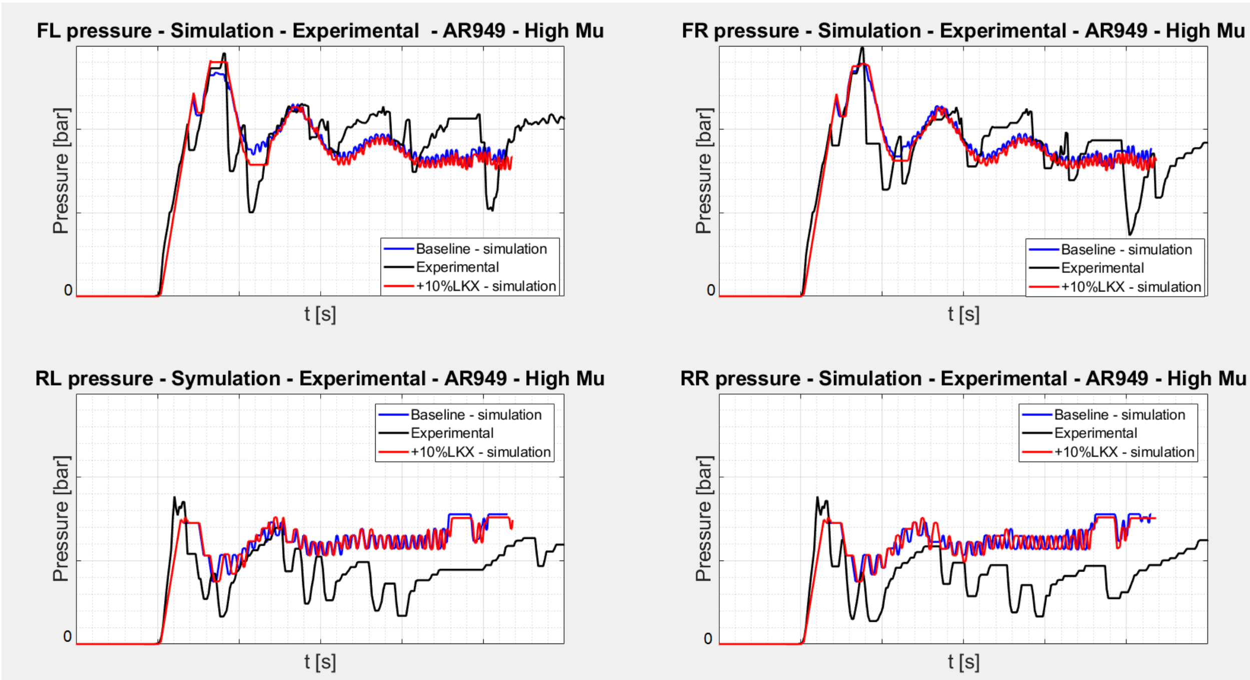


Figure 10.6.2 – Brake pressure on the four corners – Baseline vs +10%LKX

- Vehicle longitudinal acceleration (figure 10.6.3):

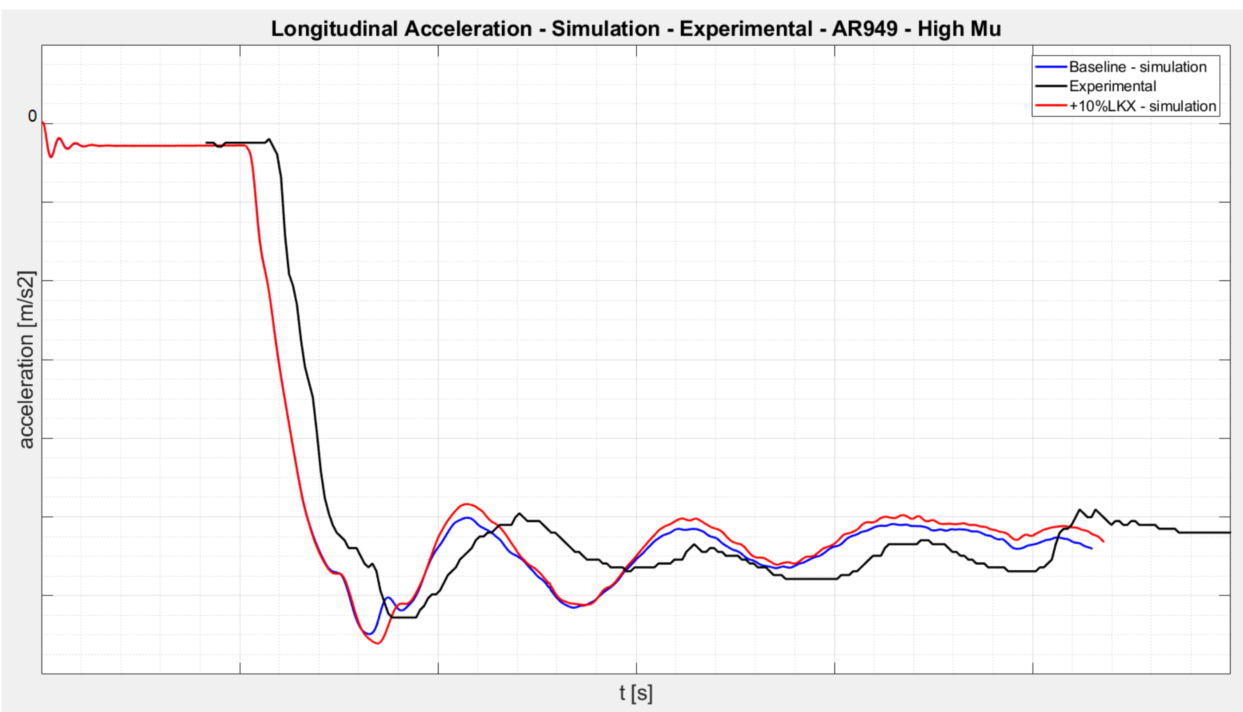


Figure 10.6.3 – Vehicle longitudinal acceleration – Baseline vs +10%LKX

- Braking distance (figure 10.6.4): It increased 0.95%.

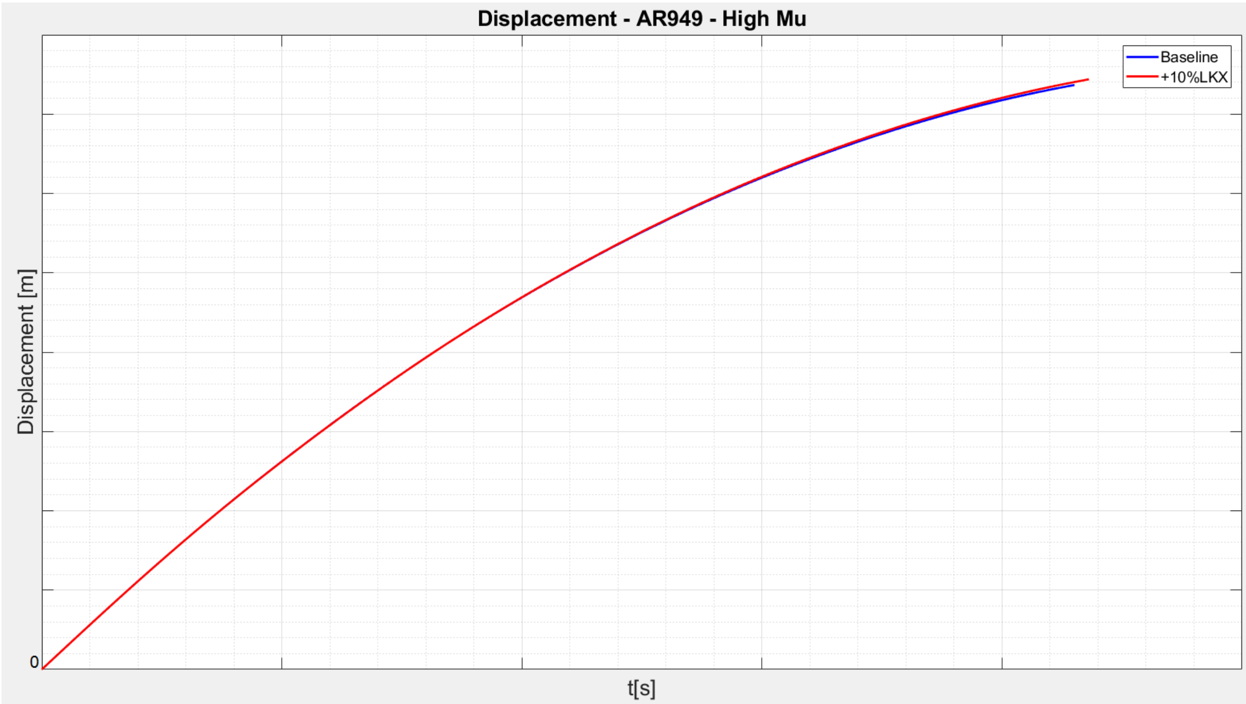


Figure 10.6.4 – Braking distance – Baseline vs +10%LKX

<u>-10%LKX:</u>

- Estimated wheel speeds (figure 10.7.1):

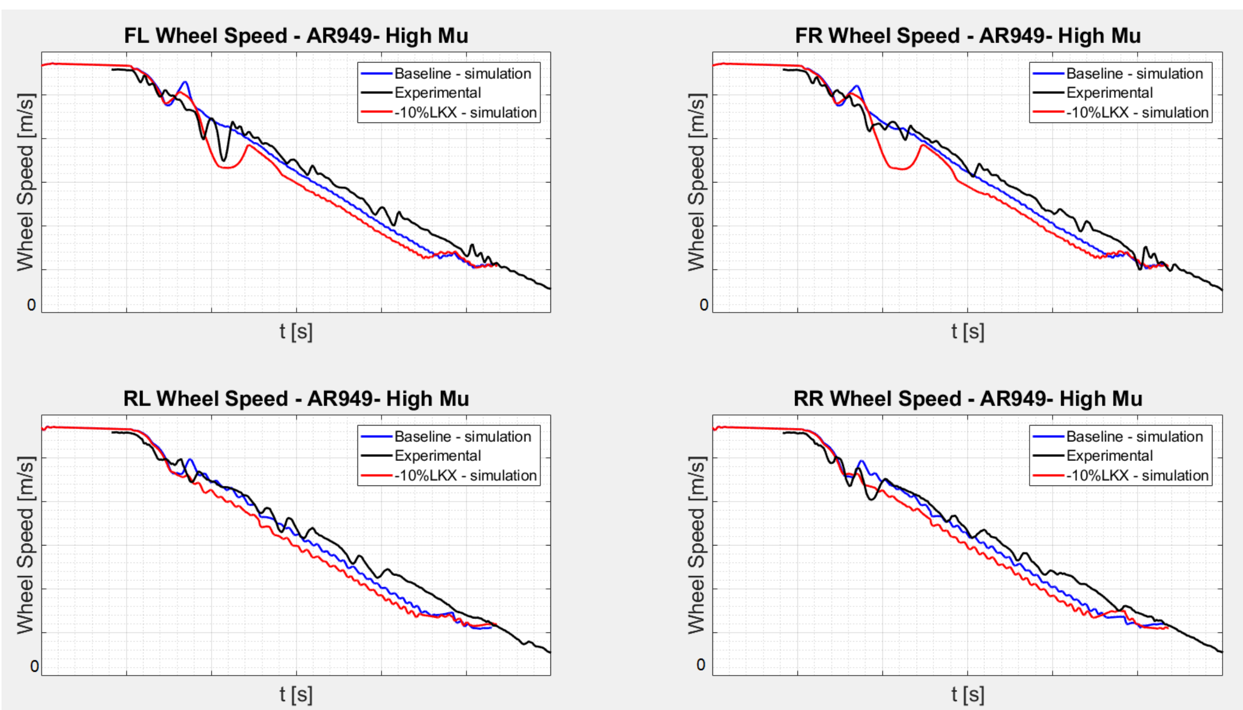


Figure 10.7.1 – Estimated wheel speed – Baseline vs -10%LKX

- Brake pressure at the four corners (figure 10.7.2):

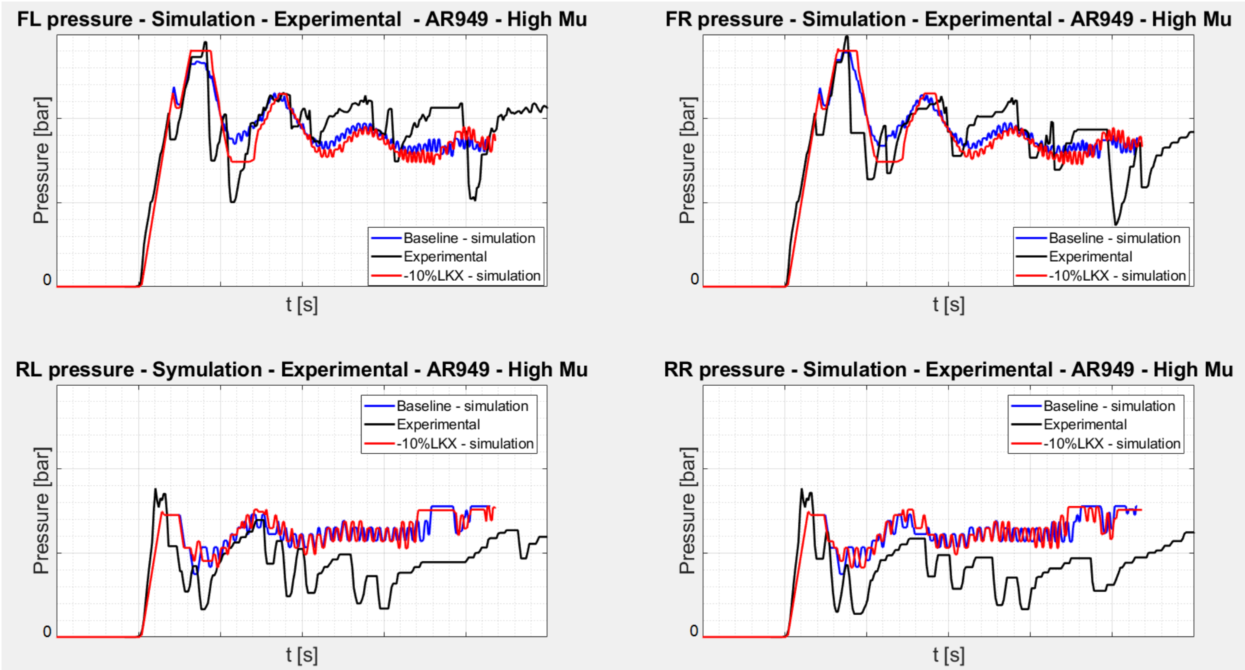


Figure 10.7.2 – Brake pressure on the four corners – Baseline vs -10%LKX

- Vehicle longitudinal acceleration (figure 10.7.3):

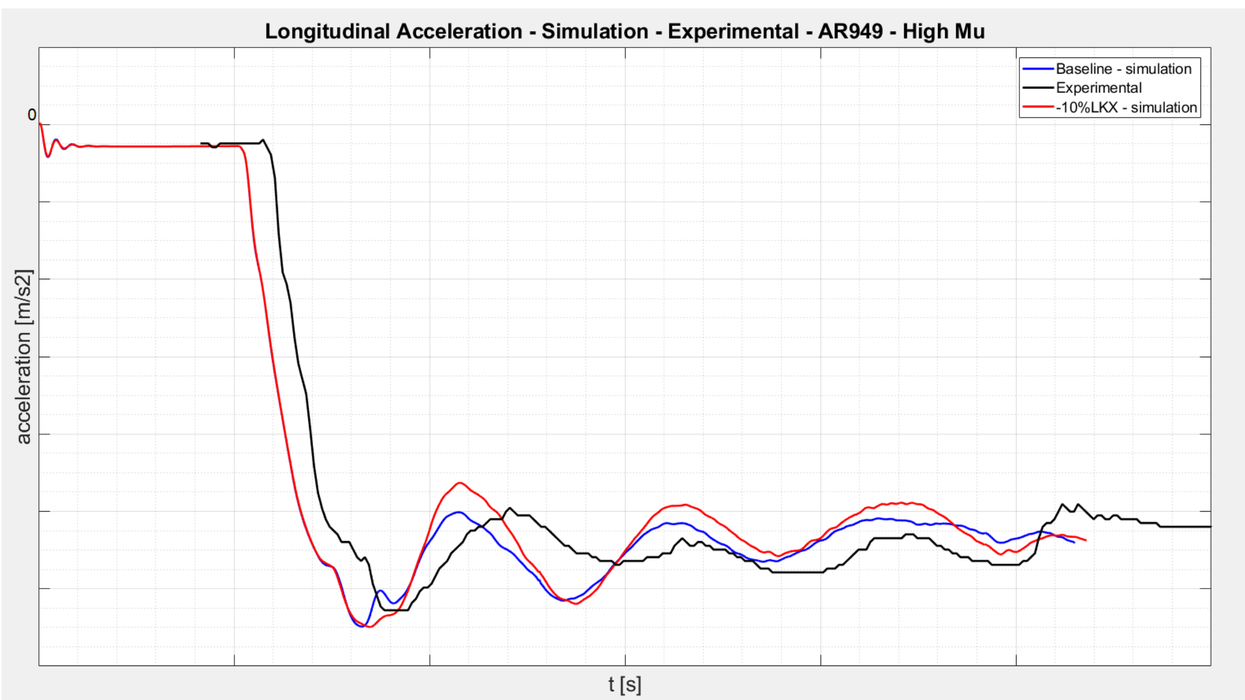


Figure 10.7.3 – Vehicle longitudinal acceleration – Baseline vs -10%LKX

- Braking distance (figure 10.7.4): It increased 1.47%.

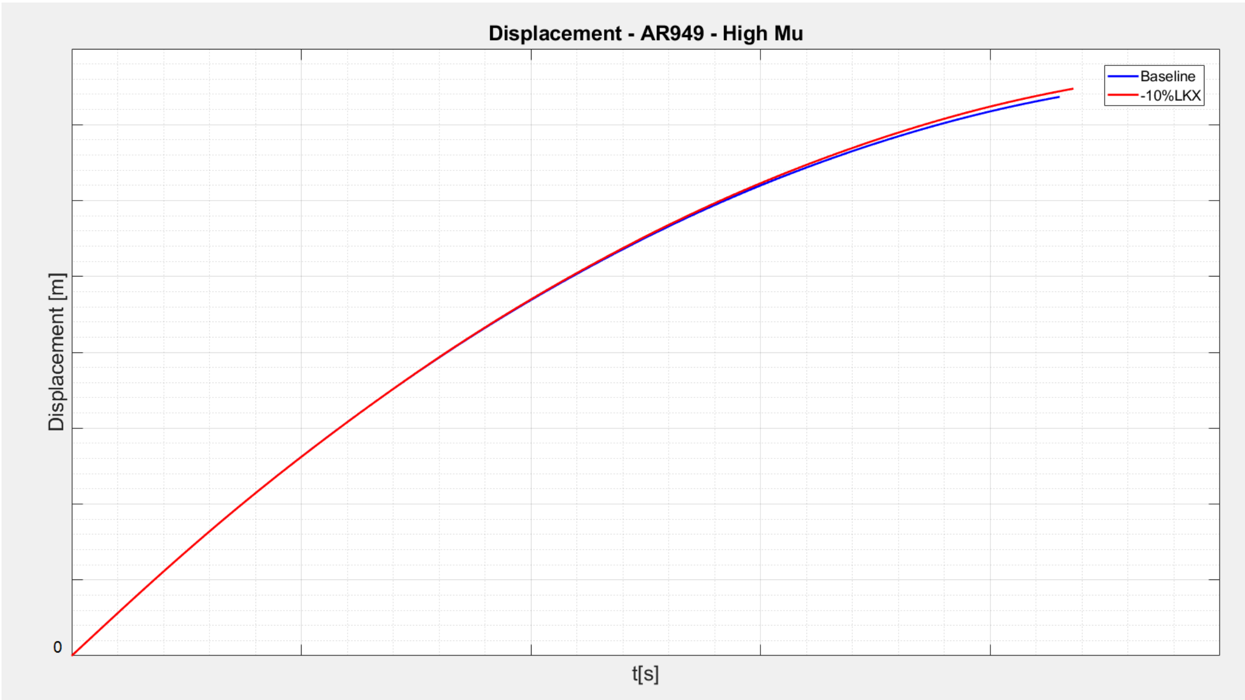


Figure 10.7.4 – Braking distance – Baseline vs -10%LKX

<u>OBS:</u> For a given Scale factor of F_x slip stiffness (LKX) of the tire, a certain slip threshold is set in order to obtain a maximum braking force. If requested to change the LKX (ex: change tires), a new optimal slip thresholds has to be set in order to obtain the maximum brake force of this new curve, otherwise if the system is tuned for a certain tire, and the tires are changed, the brake performance will decrease, increasing the brake distance. For example, increasing just the LKX of 10% made the braking distance to increase 0.95%, instead, if keeping this increase of LKX and finding an optimal slip threshold (-3%Sthr), this difference of braking distance reduced to +0.11%, as shown below.

The simulation with +10% LKX and -3%Slip Threshold is shown below:

- Estimated Wheel Speed (figure 10.8.1):

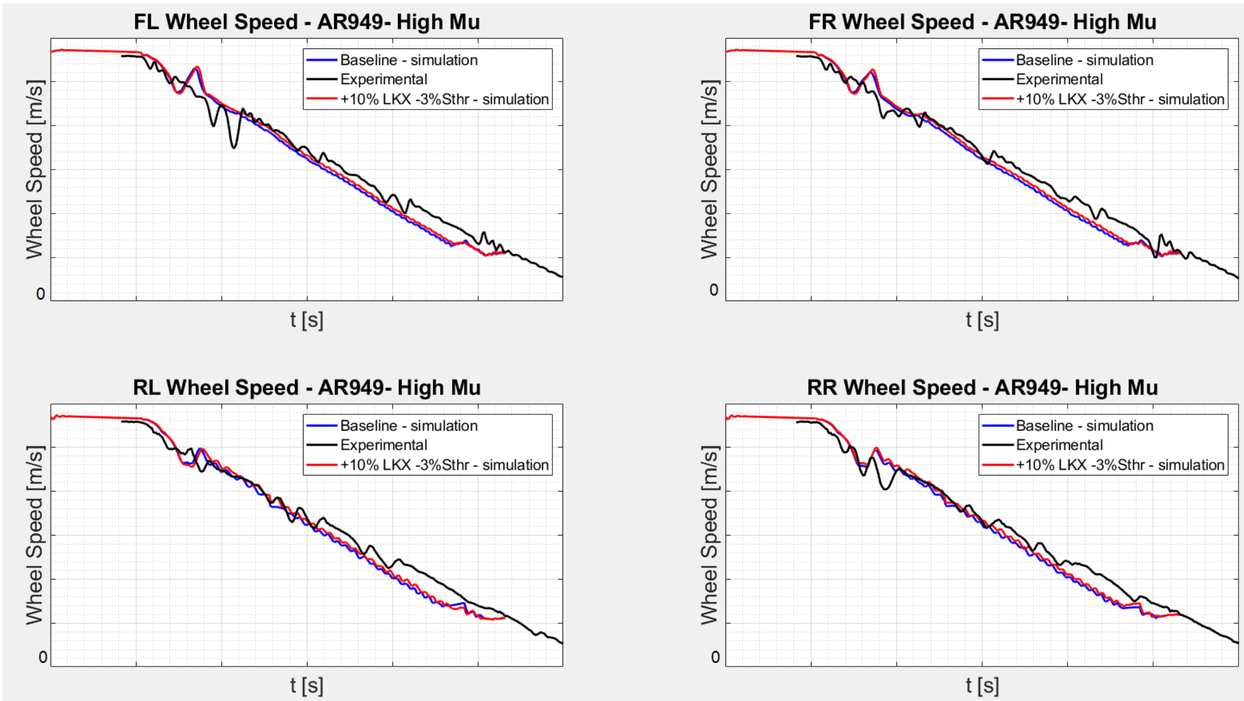


Figure 10.8.1 – Estimated wheel speed – Baseline vs +10%LKX -3%Sthr

- Brake pressure at the four corners (figure 10.8.2):

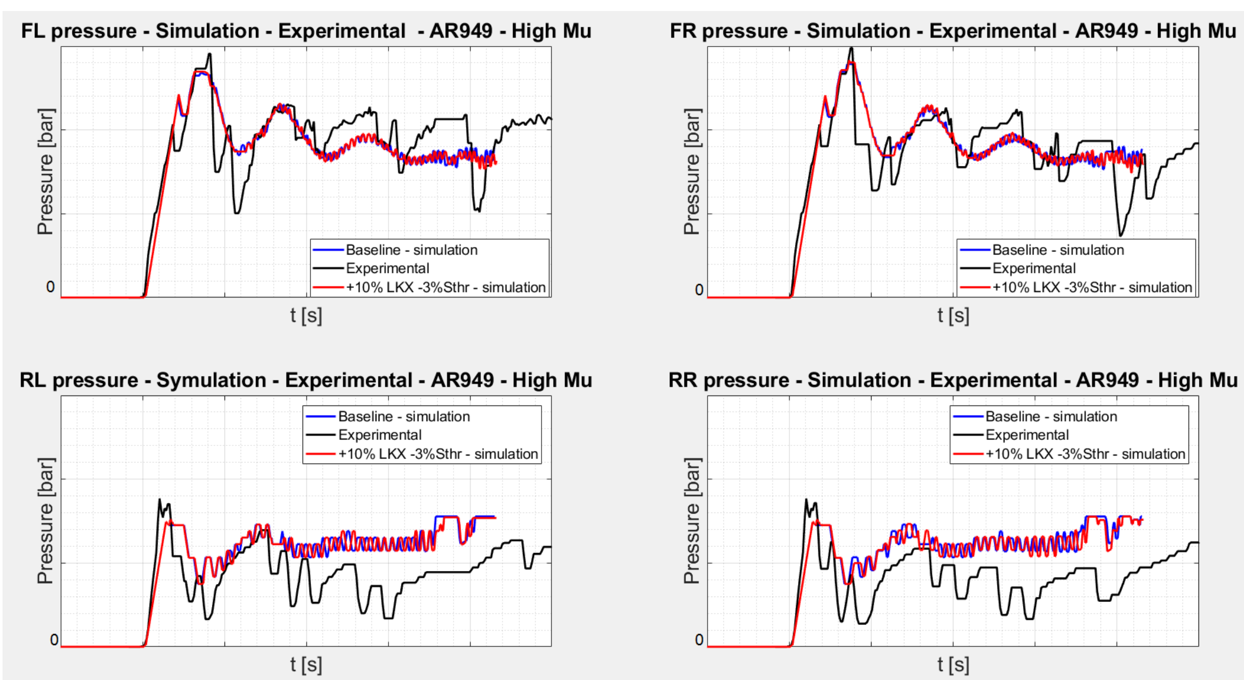


Figure 10.8.2 – Brake pressure on the four corners – Baseline vs +10%LKX -3%Sthr

- Vehicle longitudinal acceleration (figure 10.8.3):

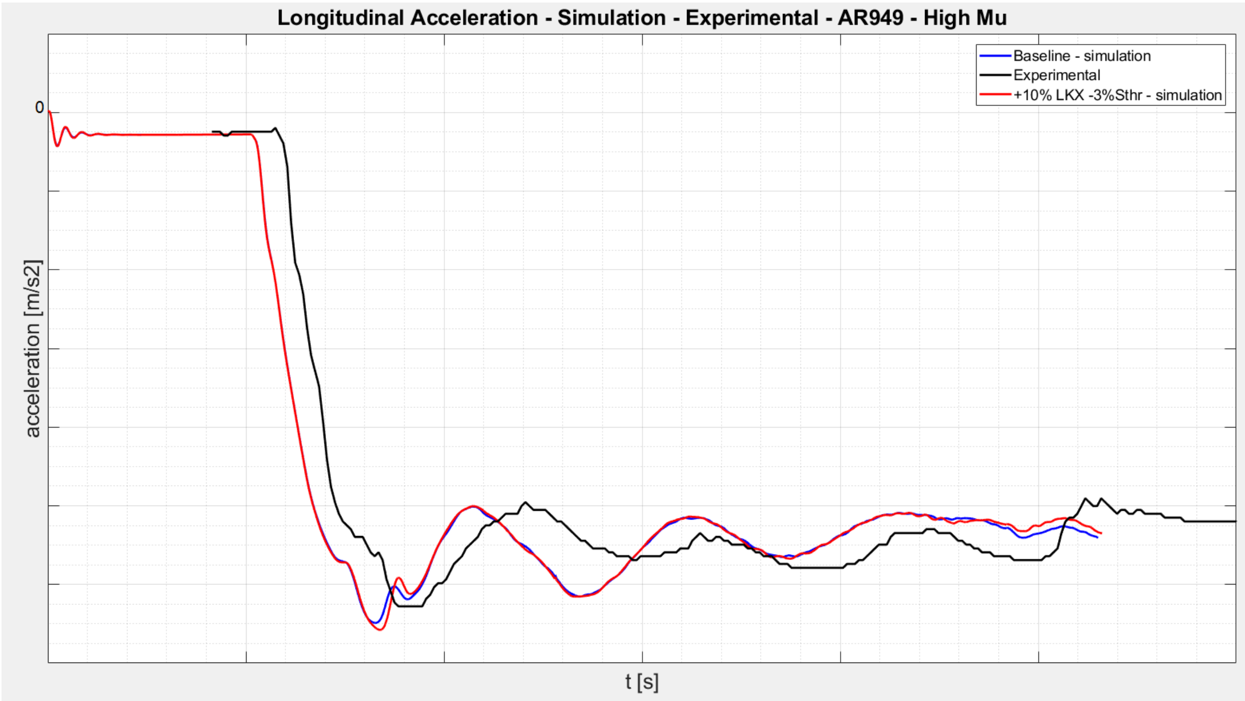


Figure 10.8.3 – Vehicle longitudinal acceleration – Baseline vs +10%LKX -3%Sthr

- Braking distance (figure 10.8.4): It increased 0.11%.

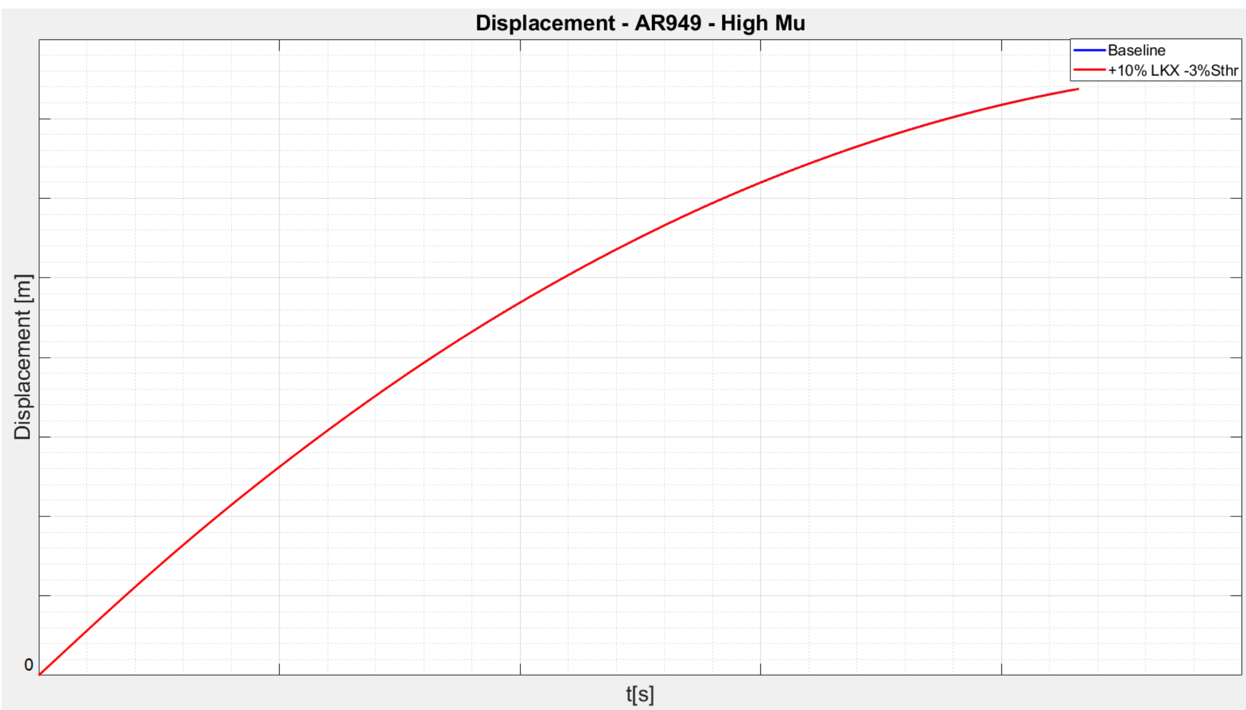


Figure 10.8.4 – Braking distance – Baseline vs +10%LKX -3%Sthr

10.4 Pad's Mu - Brake pad's coefficient of friction:

As expected, increasing the friction coefficient of the material of the brake pad the braking distance has decreased, and that when decreasing the friction coefficient the braking distance has increased. But actually this difference of braking distance in meters is not so relevant.

Changing the brake pad's mu influences on the main system pressure, and by consequence on the maximum brake pressure that can be reached on each corner. If the mu is increased, in order to reach the same desired brake force, less brake pressure is needed. And since the brake pressure on each corner is limited by the main system pressure, also the maximum brake pressure on the corners decreases.

When the brake pad's mu is decreased, the brake pressure in order to deliver the same brake force increases, so the brake pressure on each corner is less limited, allowing bigger brake pressures on the corners.

The results of the comparisons are shown in a graphical way for +/-20% Pad's Mu:

<u>+20% Pad's Mu:</u>

- Estimated wheel speeds (figure 10.9.1):

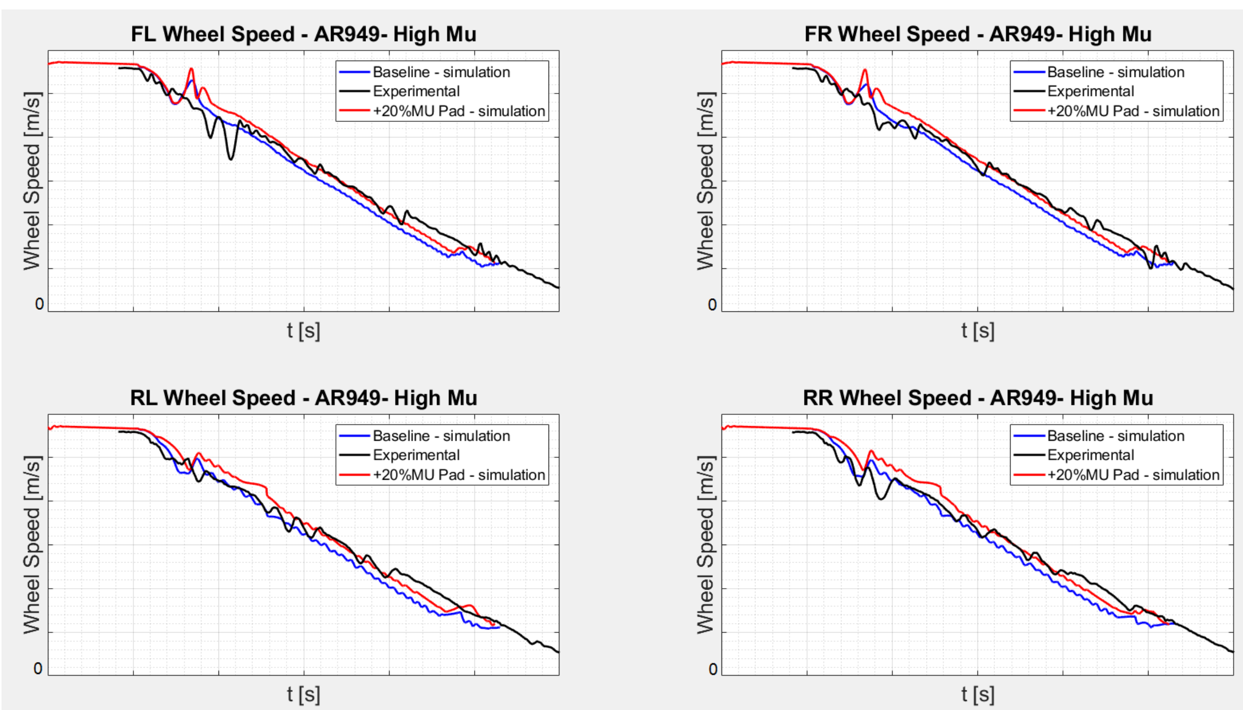


Figure 10.9.1 – Estimated wheel speed – Baseline vs +20% Pad's Mu

- Brake pressure at the four corners (figure 10.9.2):

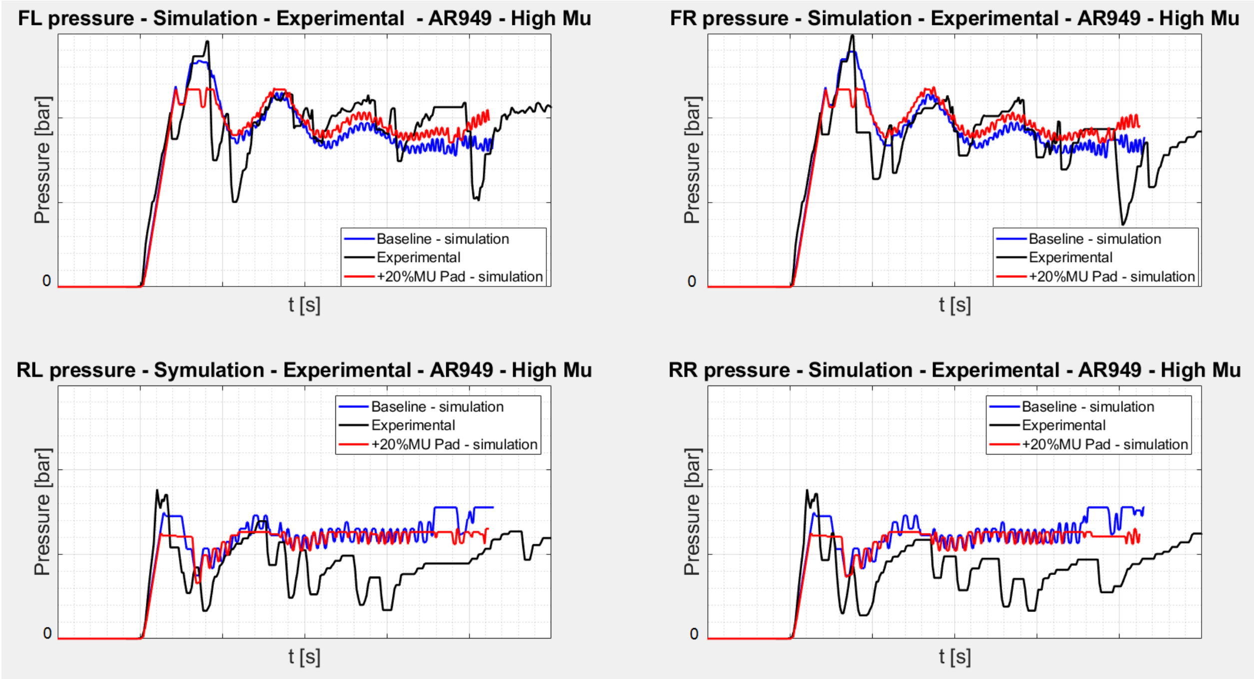


Figure 10.9.2 – Brake pressure on the four corners – Baseline vs +20% Pad's Mu

- Main System Pressure (figure 10.9.3):

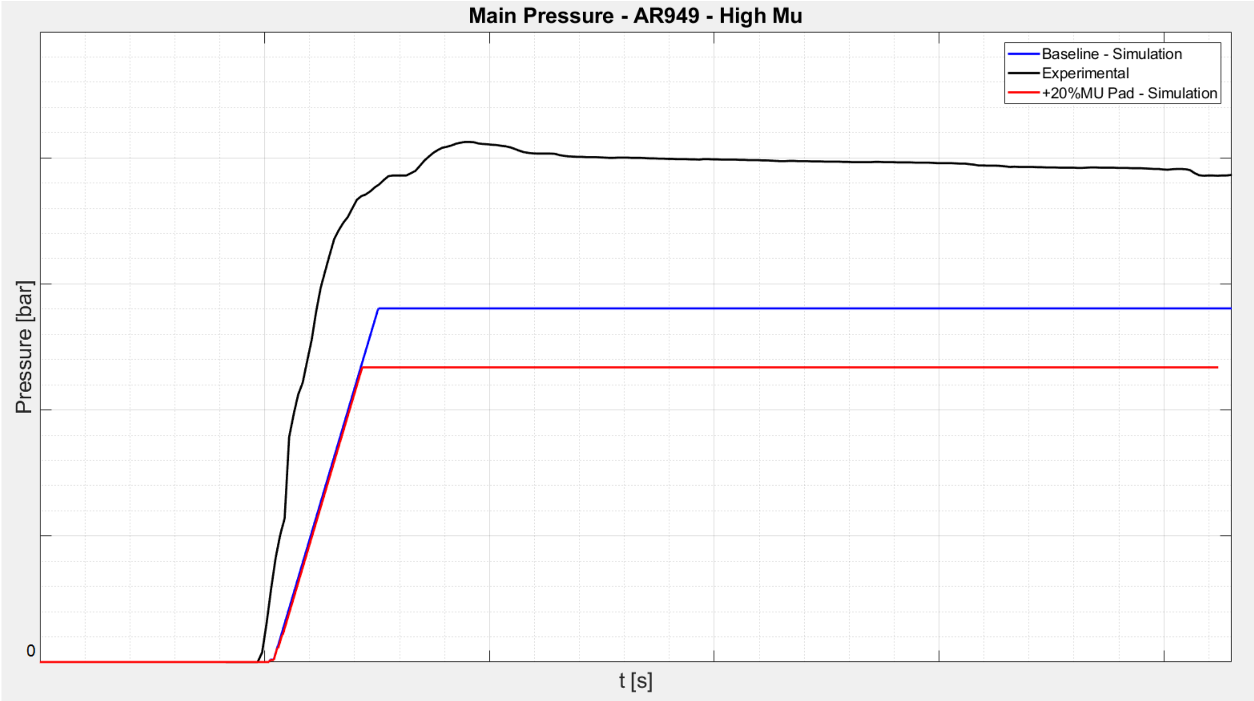


Figure 10.9.3 – Main System Pressure – Baseline vs +20% Pad's Mu

- Vehicle longitudinal acceleration (figure 10.9.4):

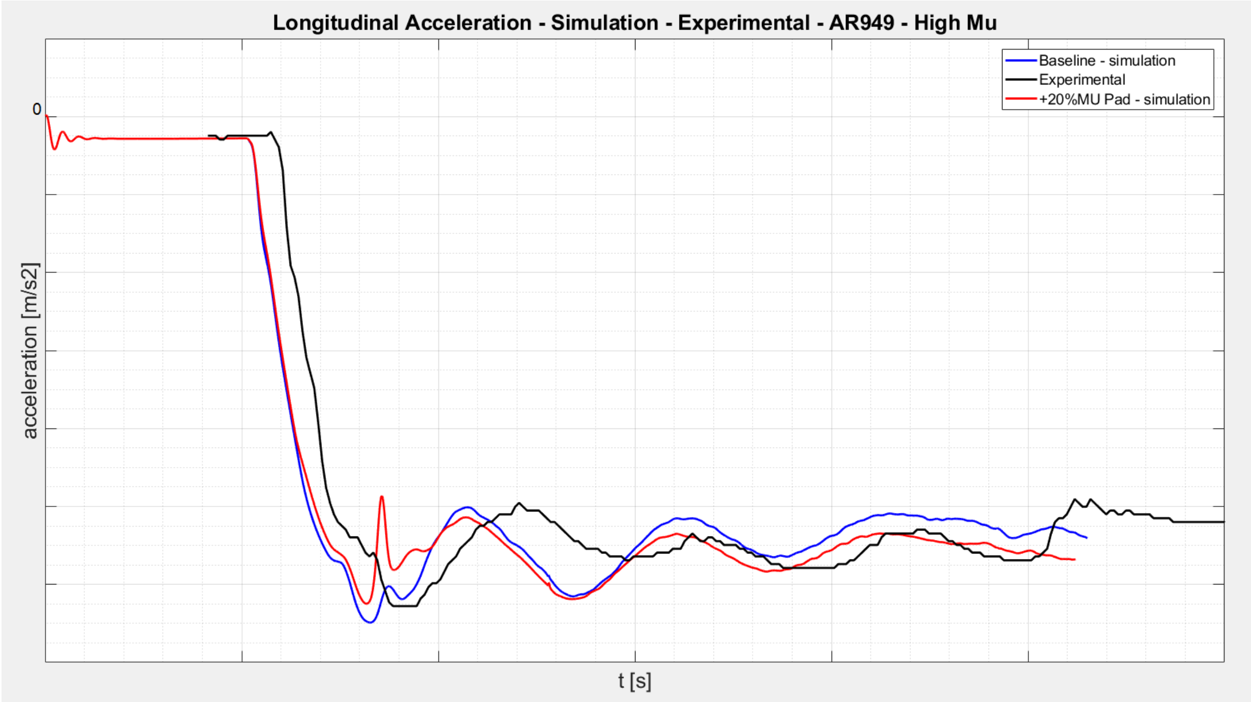


Figure 10.9.4 – Vehicle longitudinal acceleration – Baseline vs +20% Pad's Mu

- Braking distance (figure 10.9.5): It decreased 0.30%.

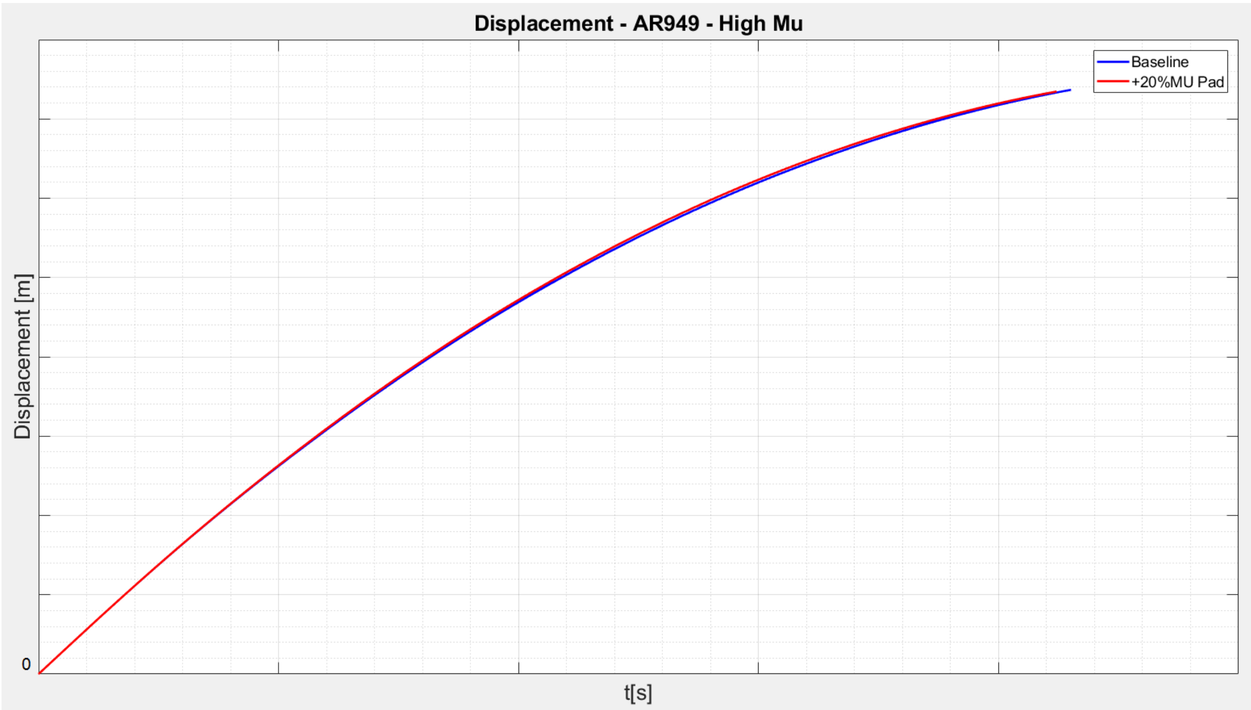


Figure 10.9.5 – Braking distance – Baseline vs +20% Pad's Mu

<u>-20% Pad's Mu:</u>

- Estimated wheel speeds (figure 10.10.1):

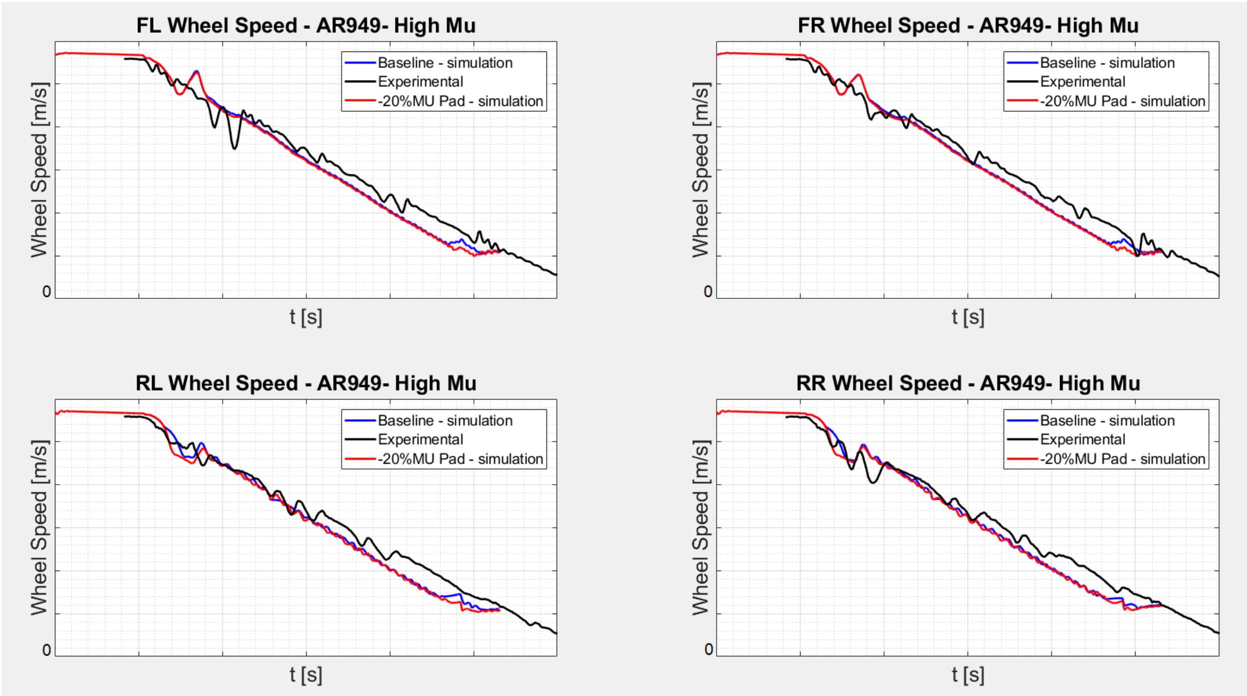


Figure 10.10.1 – Estimated wheel speed – Baseline vs -20% Pad's Mu

- Brake pressure at the four corners (figure 10.10.2):

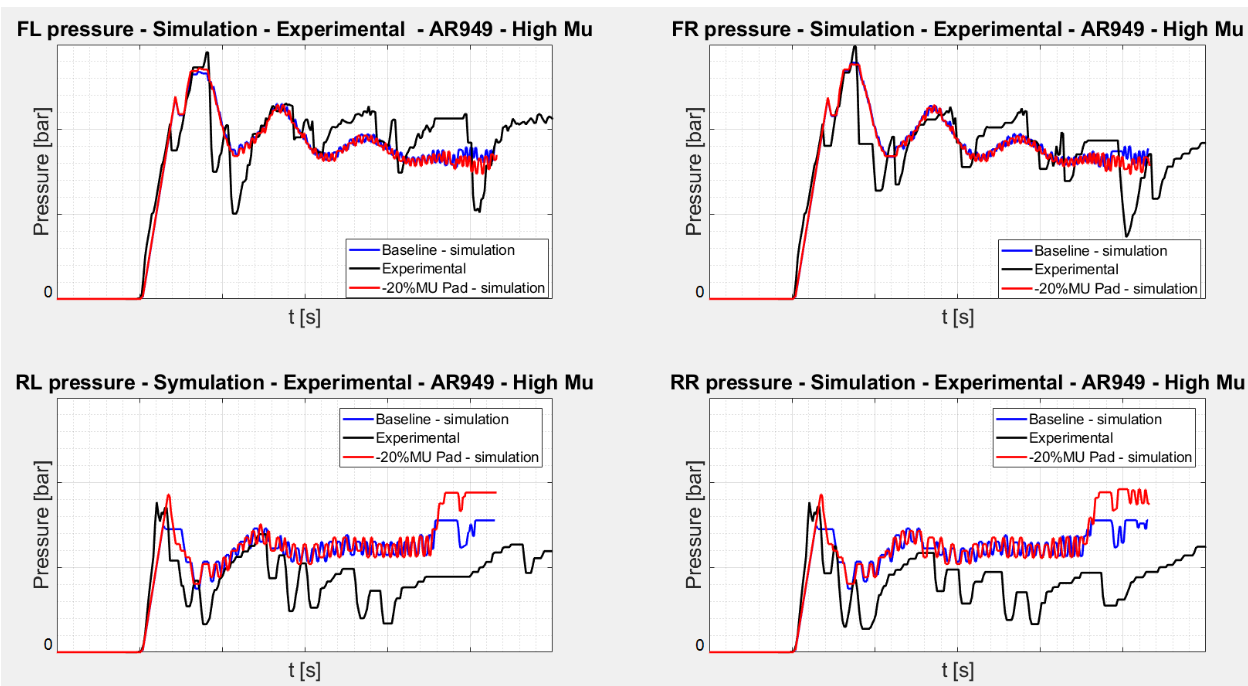


Figure 10.10.2 – Brake pressure on the four corners – Baseline vs -20% Pad's Mu

- Main System Pressure (figure 10.10.3):

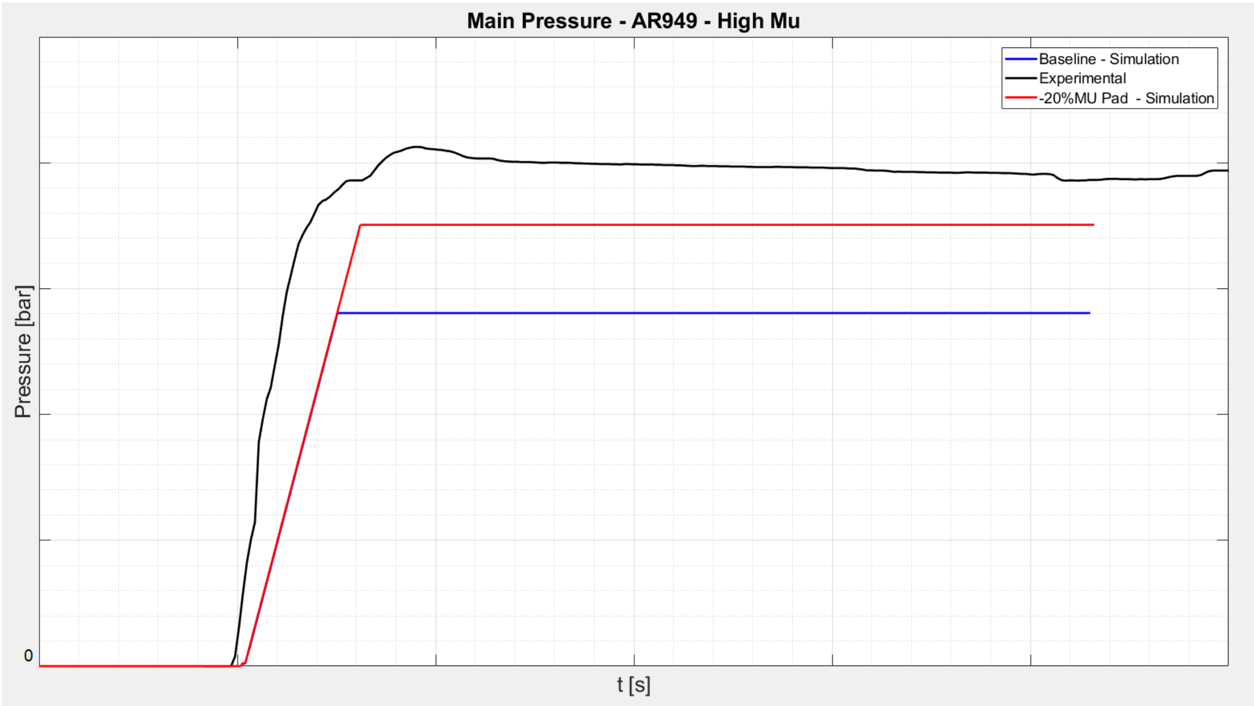


Figure 10.10.3 – Main System Pressure – Baseline vs -20% Pad's Mu

- Vehicle longitudinal acceleration (figure 10.10.4):

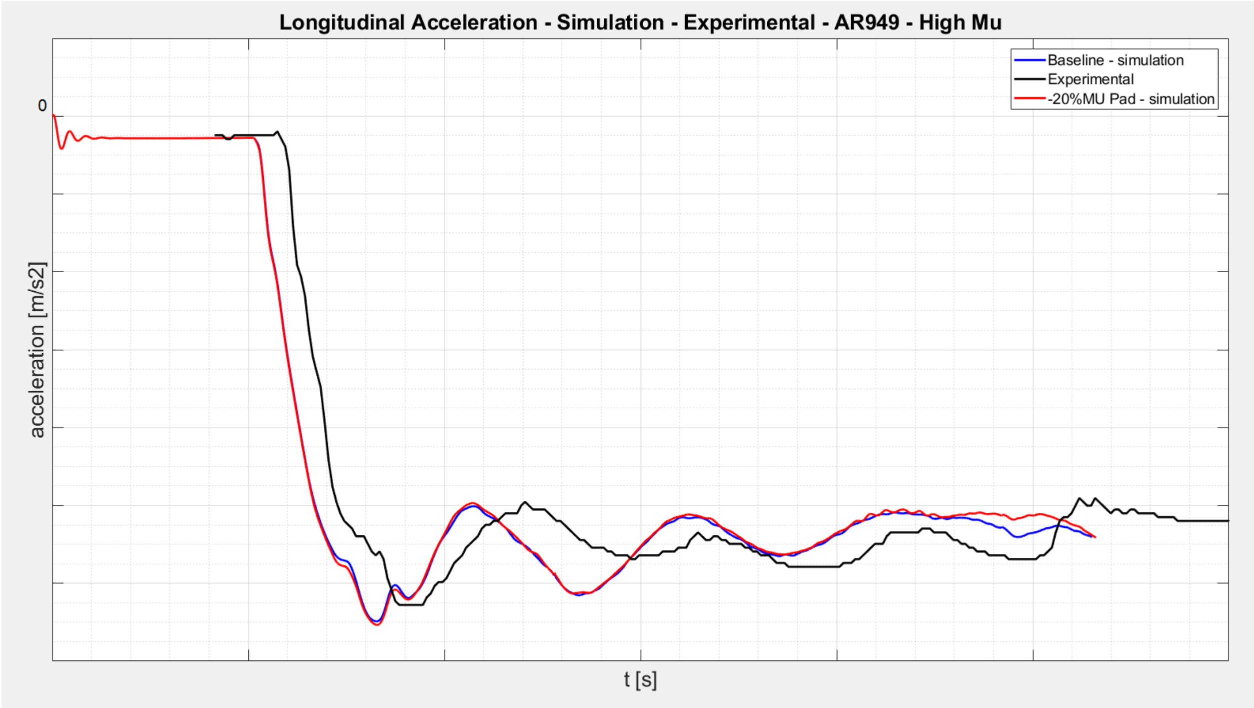


Figure 10.10.4 – Vehicle longitudinal acceleration – Baseline vs -20% Pad's Mu

- Braking distance (figure 10.10.5): It increased 0.19%.

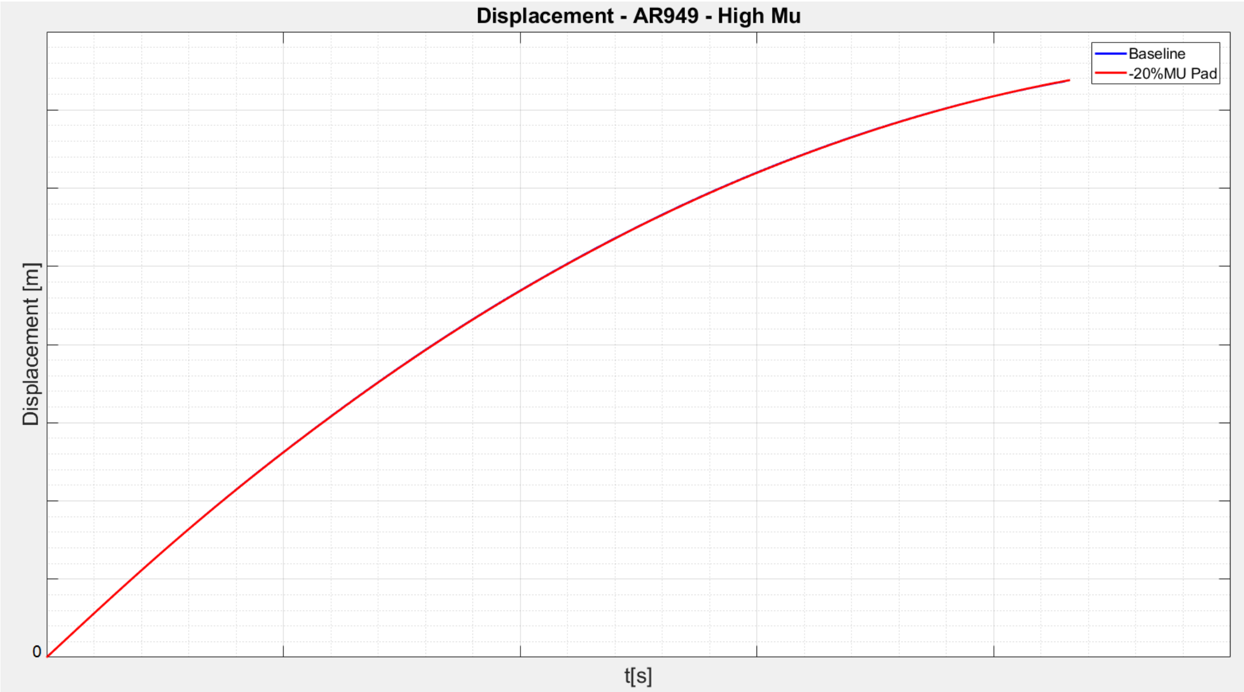


Figure 10.10.5 – Braking distance – Baseline vs -20% Pad's Mu

10.5 Slip Threshold

In order to evaluate the influence of the slip threshold target of the wheels and to verify if the chosen slip threshold target was a good objective, the slip threshold inside the control logics was modified to +/-20%. After the modifications, the braking distance increased for both positive and negative variations, it means that, for the tuned parameters set during the development of the thesis, the slip threshold used in the baseline simulation is a good target.

But it is good to remember that for different parameters values set, the slip threshold target needs to be changed in order to reach a good braking distance again.

The results of the comparisons are shown in a graphical way for +/-20% Slip Threshold:

<u>+20% Slip Threshold</u>

- Estimated wheel speeds (figure 10.11.1):

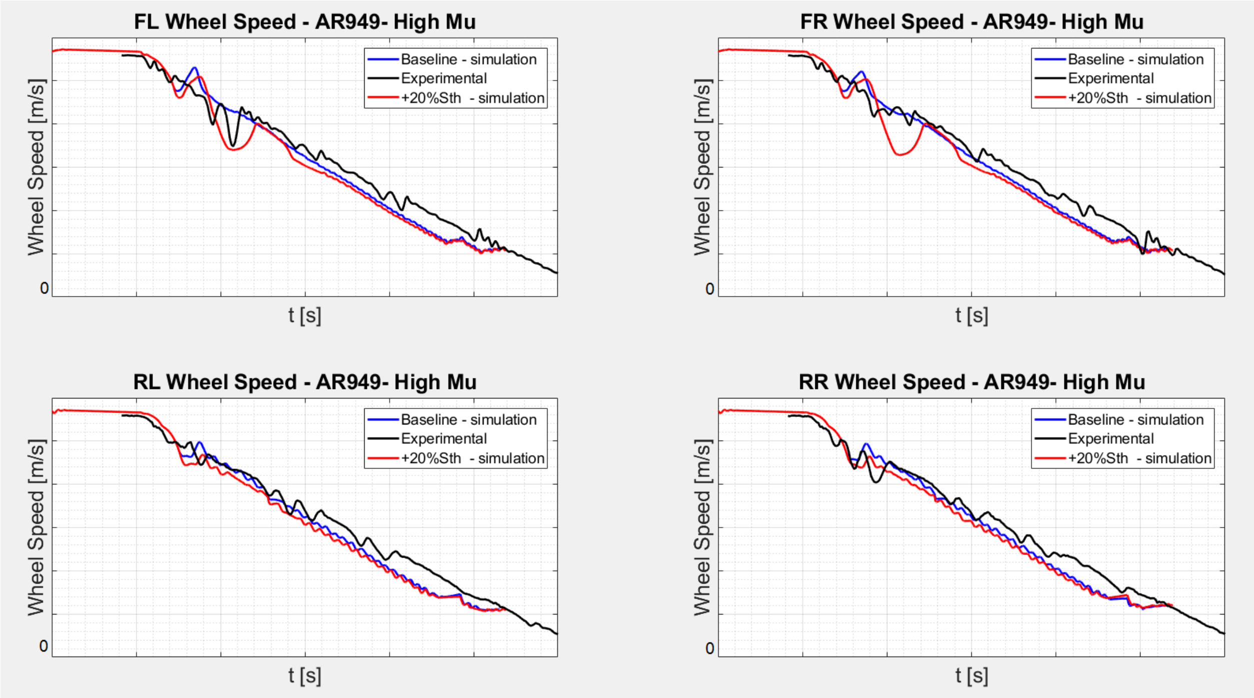


Figure 10.11.1 – Estimated wheel speed – Baseline vs +20%Sthr

- Brake pressure at the four corners (figure 10.11.2):

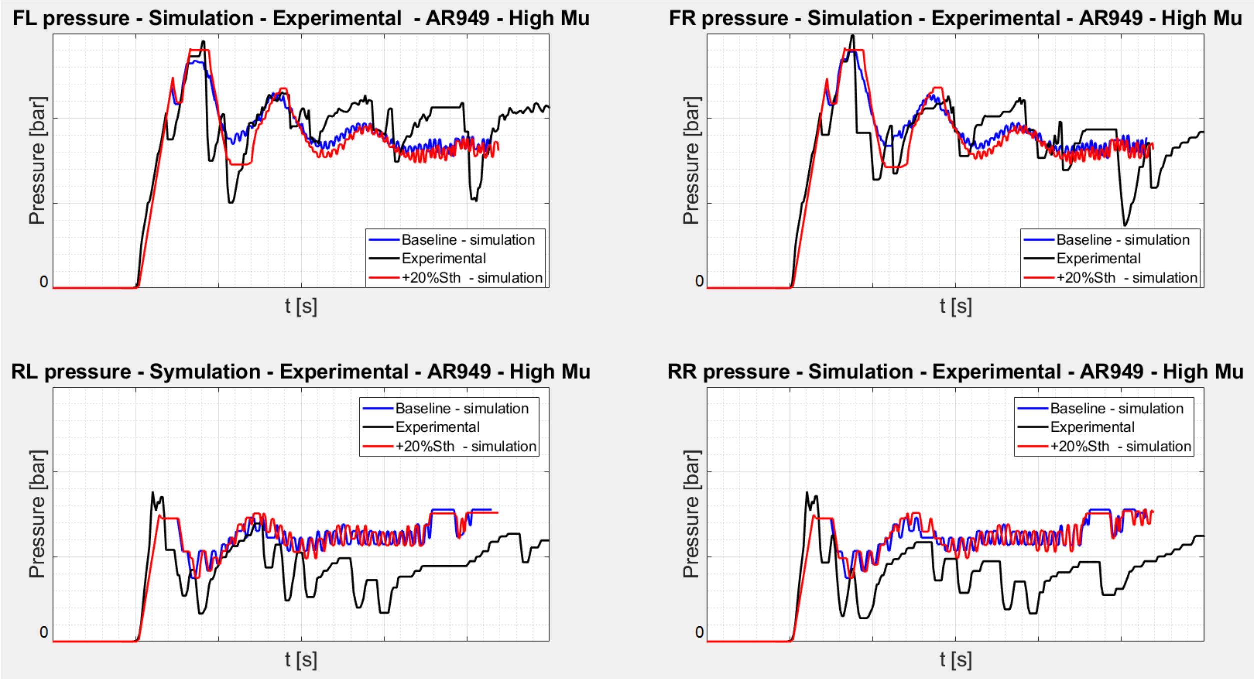


Figure 10.11.2 – Brake pressure on the four corners – Baseline vs +20%Sthr

- Vehicle longitudinal acceleration (figure 10.11.3):

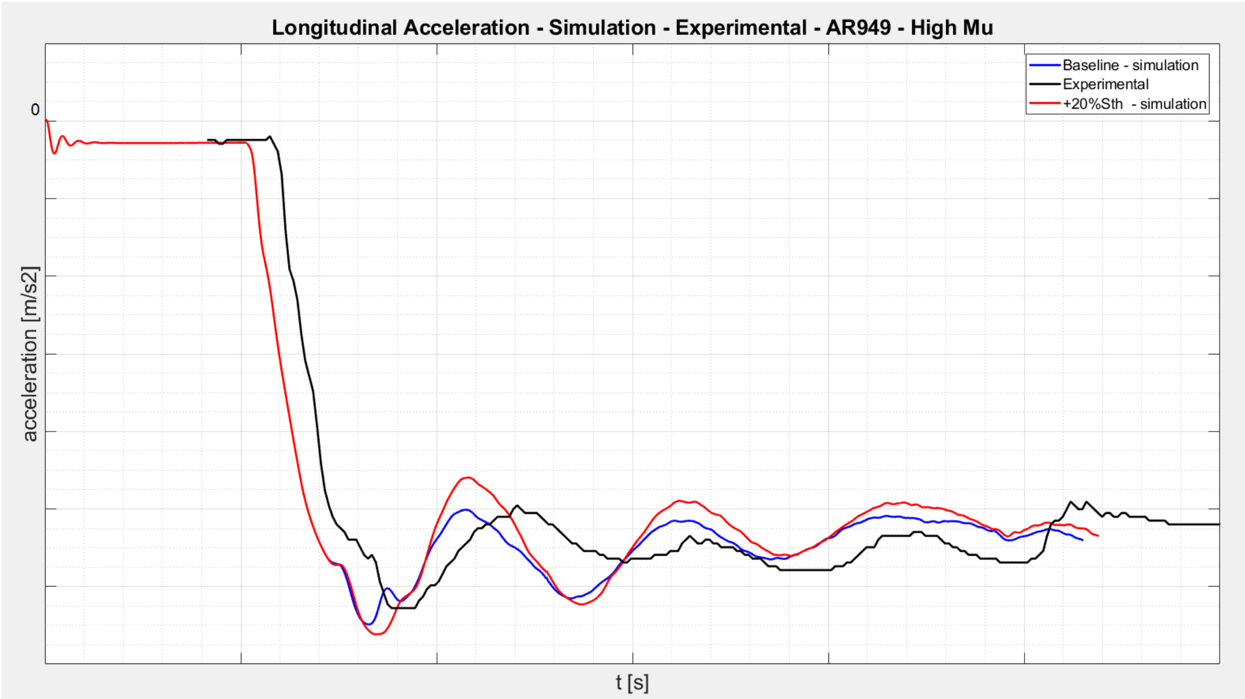


Figure 10.11.3 – Vehicle longitudinal acceleration – Baseline vs +20%Sthr

- Braking distance (figure 10.11.4): It increased 1.57%

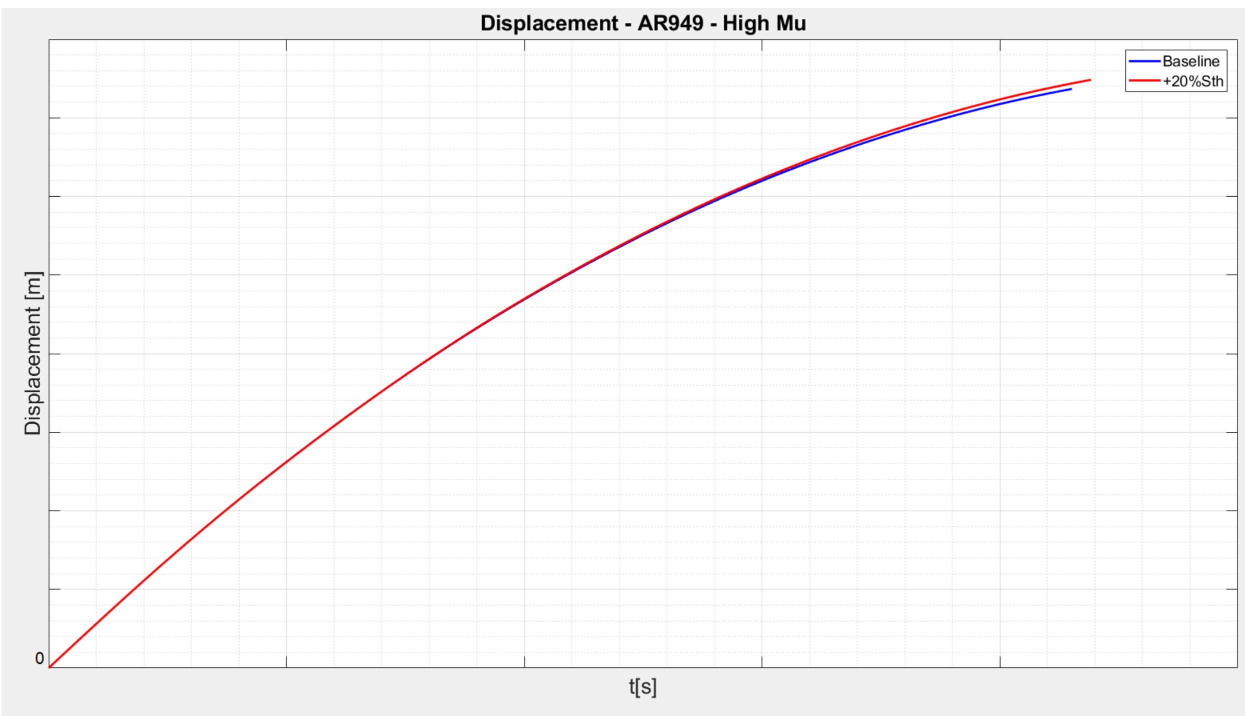


Figure 10.11.4 – Braking distance – Baseline vs +20%Sthr

-20% Slip Threshold

- Estimated wheel speeds (figure 10.12.1):

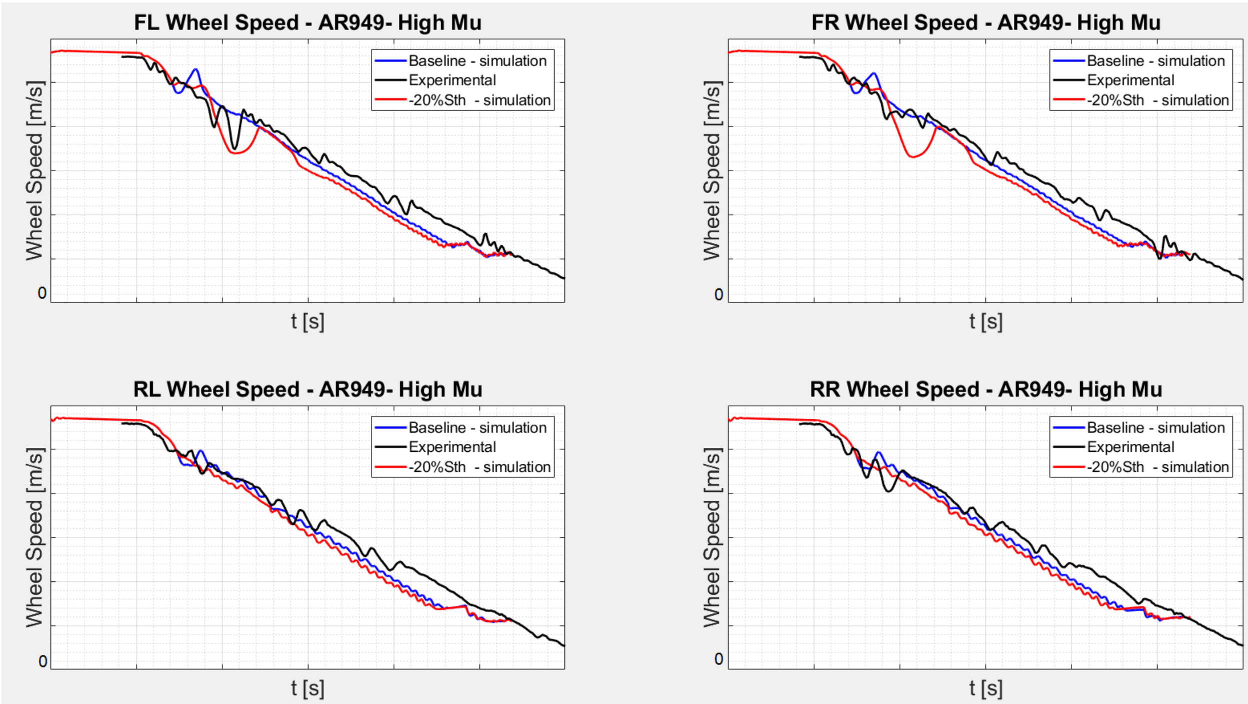


Figure 10.12.1 – Estimated wheel speed – Baseline vs -20%Sthr

- Brake pressure at the four corners (figure 10.12.2):

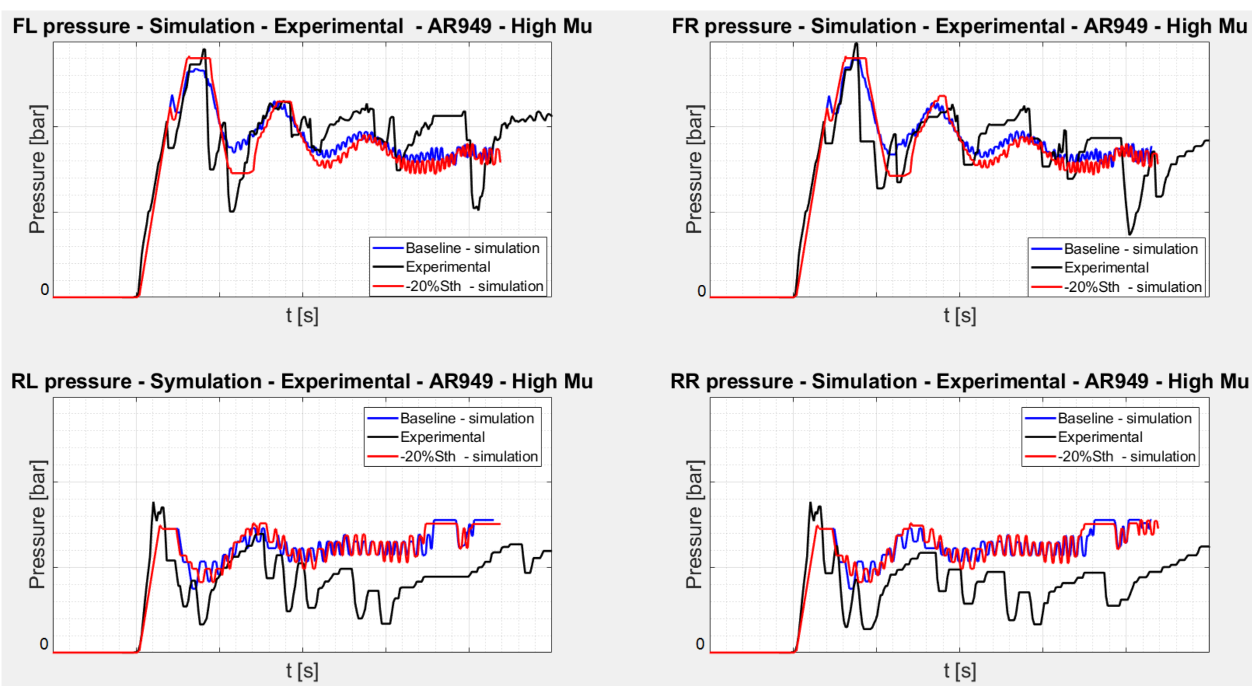


Figure 10.12.2 – Brake pressure on the four corners – Baseline vs -20%Sthr

- Vehicle longitudinal acceleration (figure 10.12.3):

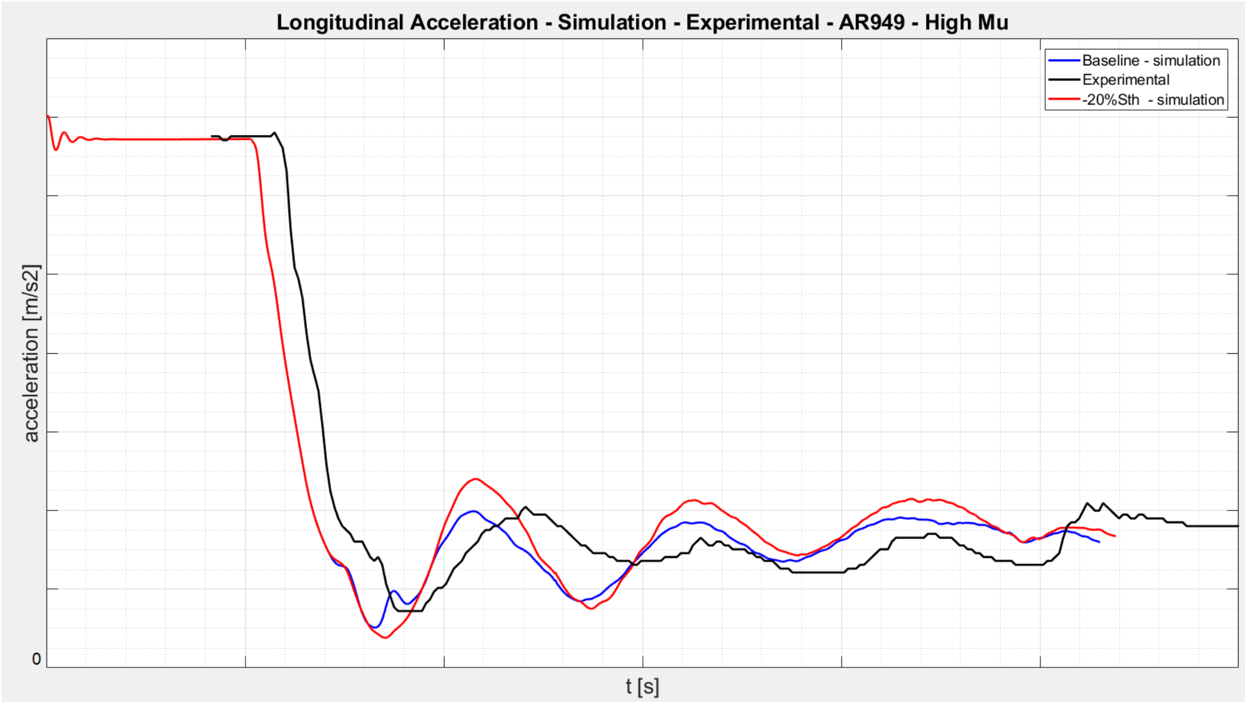


Figure 10.12.3 – Vehicle longitudinal acceleration – Baseline vs -20%Sthr

- Braking distance (figure 10.12.4): It increased 1.60%.

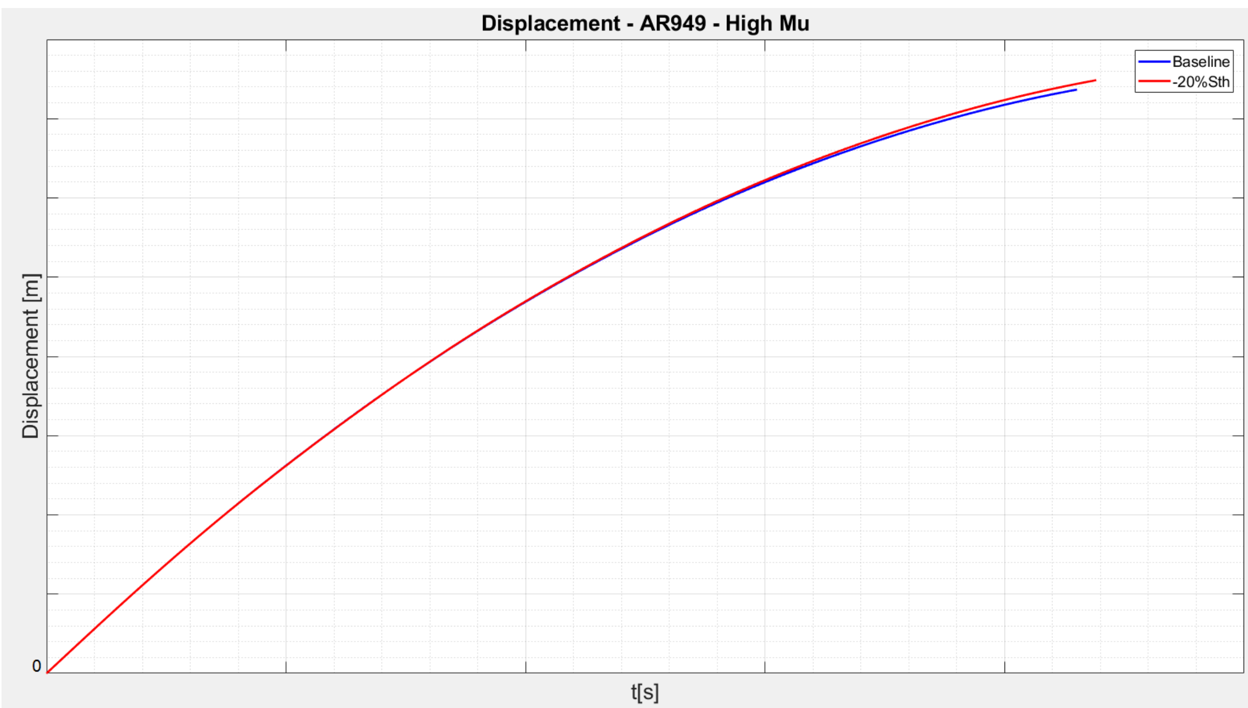


Figure 10.12.4 – Braking distance – Baseline vs -20%Sthr

10.6 Tire's Structural Longitudinal Stiffness

In order to evaluate the influence of the tire's longitudinal stiffness on the braking distance, it was varied of+/-10% on the files '.tir' for both front and rear tires of the AR 949 - Stelvio, in CarRealTime environment, then run once again in the Matlab-Simulink environment.

Varying the tire's longitudinal stiffness of +/-10% does not decrease the braking distance (it actually increased the braking distance), and did not improve any other parameter as the brake pressure in the four corners or estimated wheel speeds.

The results of the comparisons are shown in a graphical way for +/-10% Tire's structural stiffness:

<u>+10% Structural Longitudinal Stiffness:</u>

- Estimated wheel speeds (figure 10.13.1):

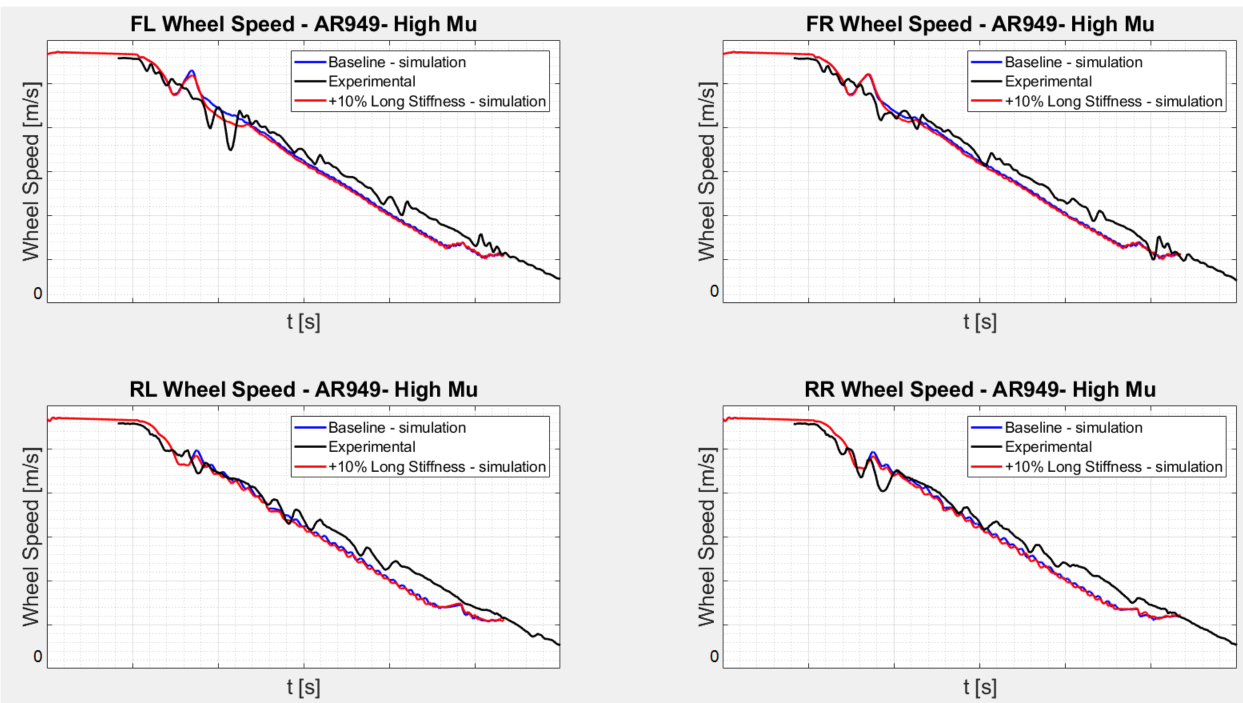


Figure 10.13.1 – Estimated wheel speed – Baseline vs +10% Structural longitudinal stiffness

- Brake pressure at the four corners (figure 10.13.2):

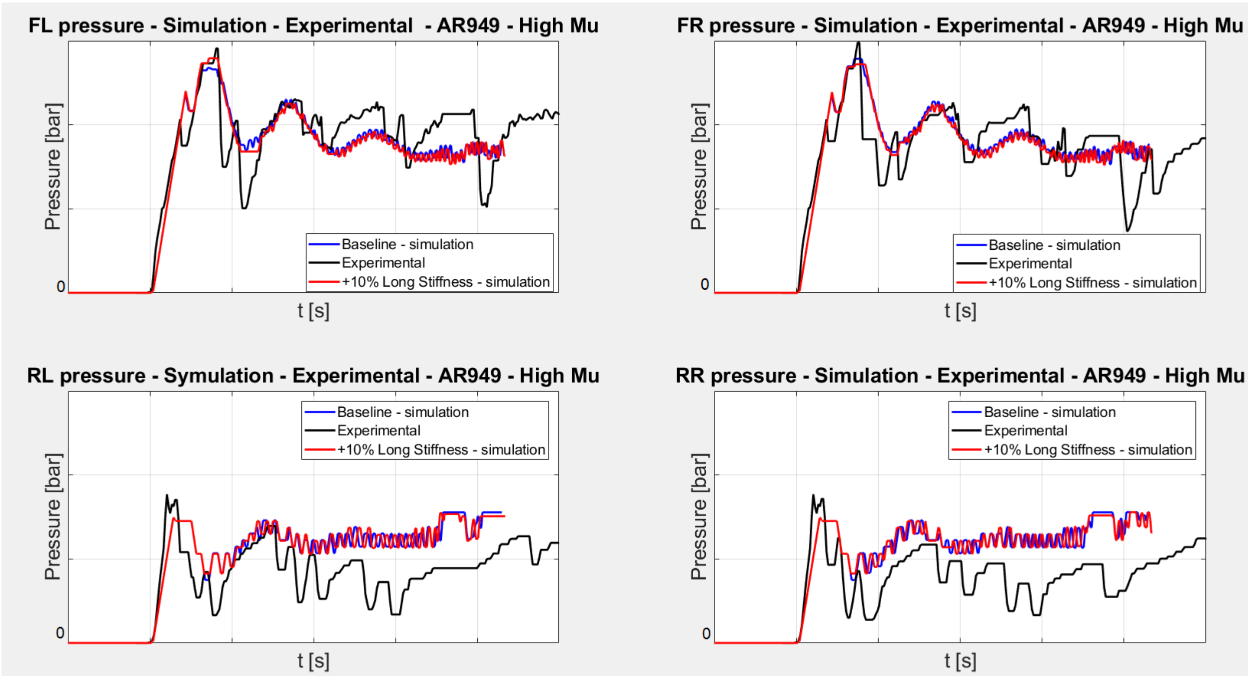


Figure 10.13.2 – Brake pressure on the four corners – Baseline vs +10% Structural longitudinal stiffness

- Vehicle longitudinal acceleration (figure 10.13.3):

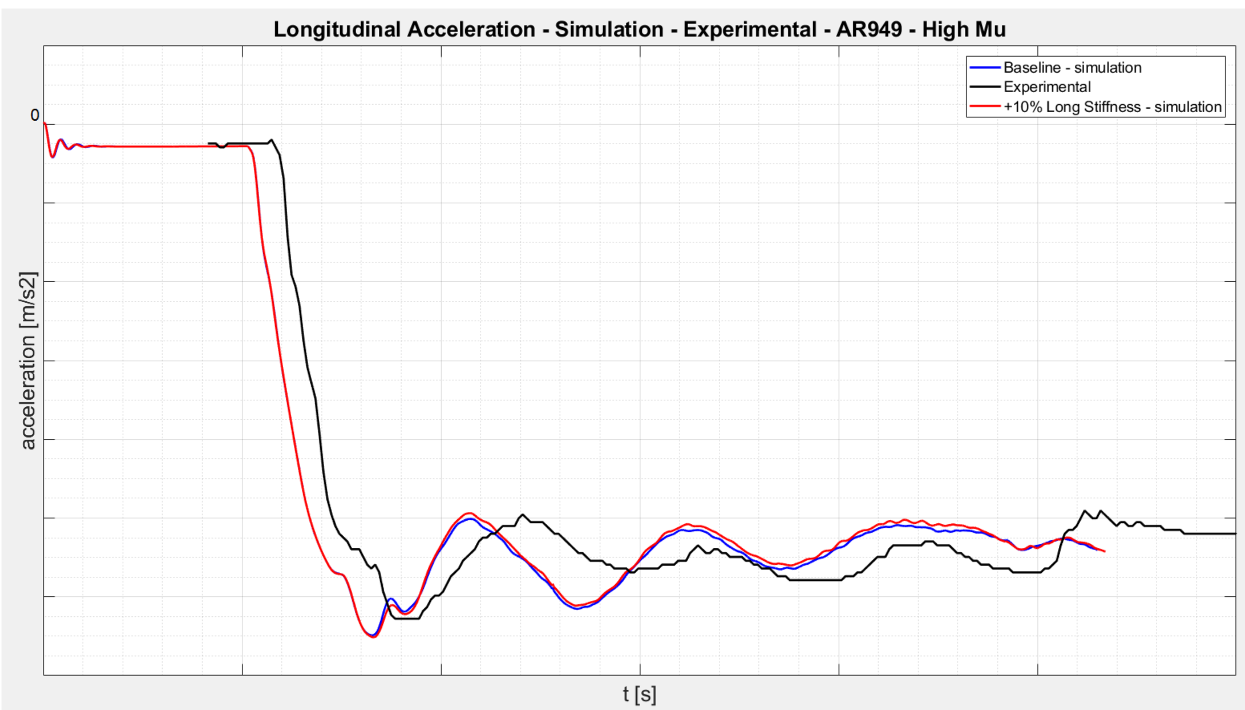


Figure 10.13.3 – Vehicle longitudinal acceleration – Baseline vs +10% Structural longitudinal stiffness

- Braking distance (figure 10.13.4): It increased 0.60%.

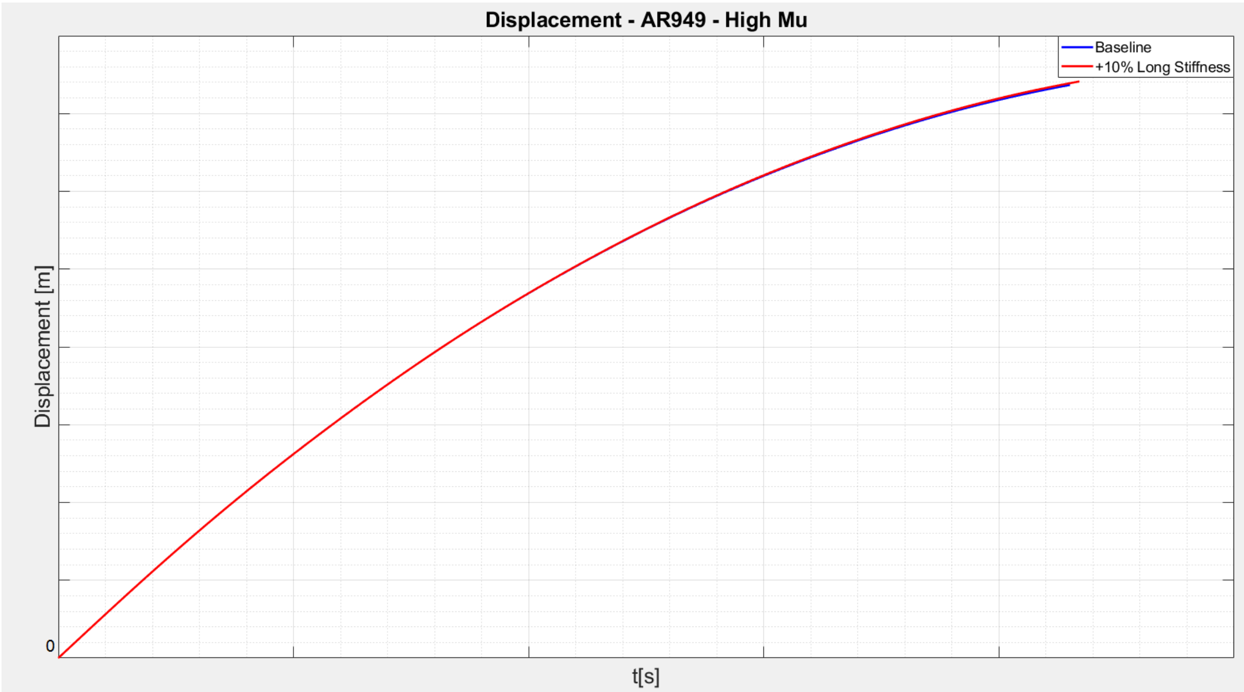


Figure 10.13.4 – Braking distance – Baseline vs +10% Structural longitudinal stiffness

-<u>10% Structural Longitudinal Stiffness</u>

- Estimated wheel speeds (figure 10.14.1):

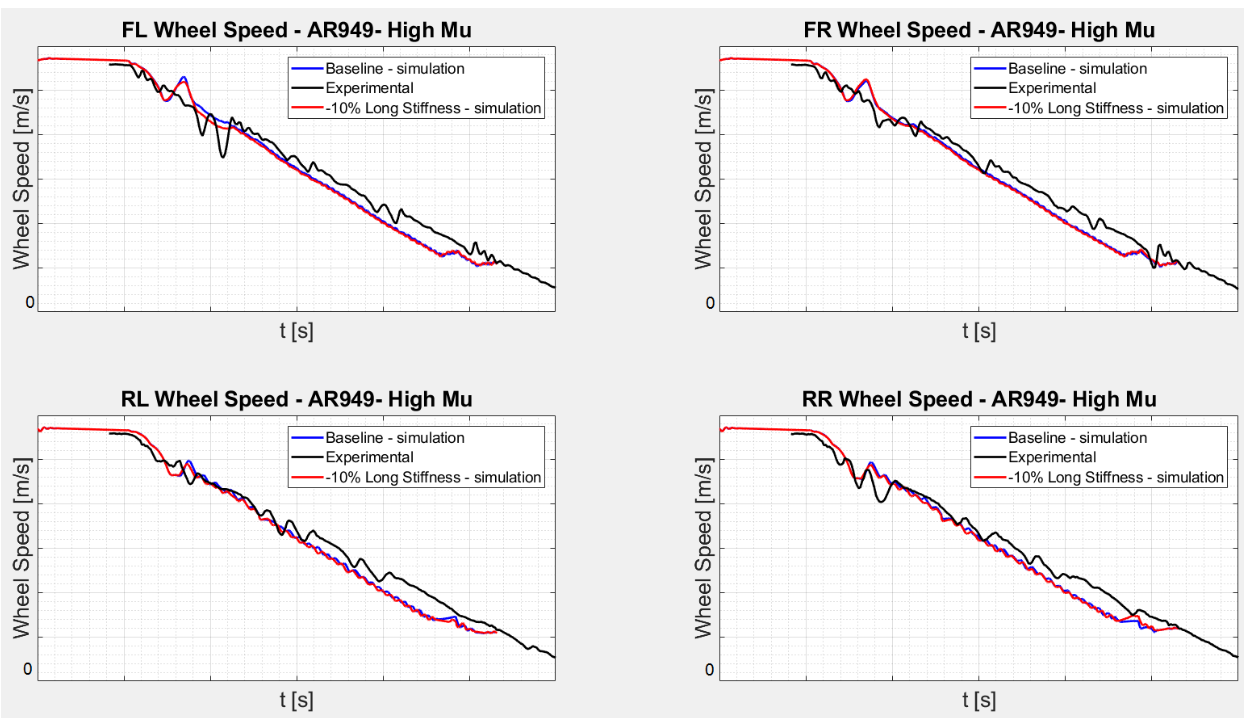


Figure 10.14.1 – Estimated wheel speed – Baseline vs -10% Structural longitudinal stiffness

- Brake pressure at the four corners (figure 10.14.2):

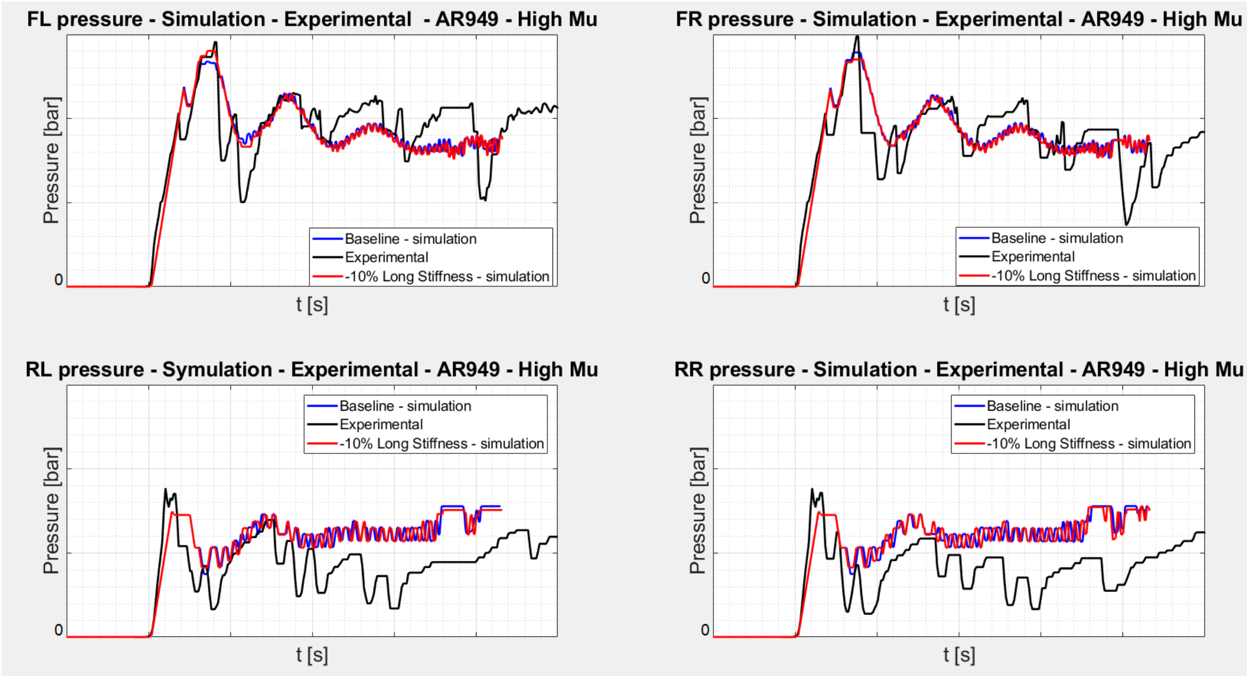


Figure 10.14.2 – Brake pressure on the four corners – Baseline vs -10% Structural longitudinal stiffness

- Vehicle longitudinal acceleration (figure 10.14.3):

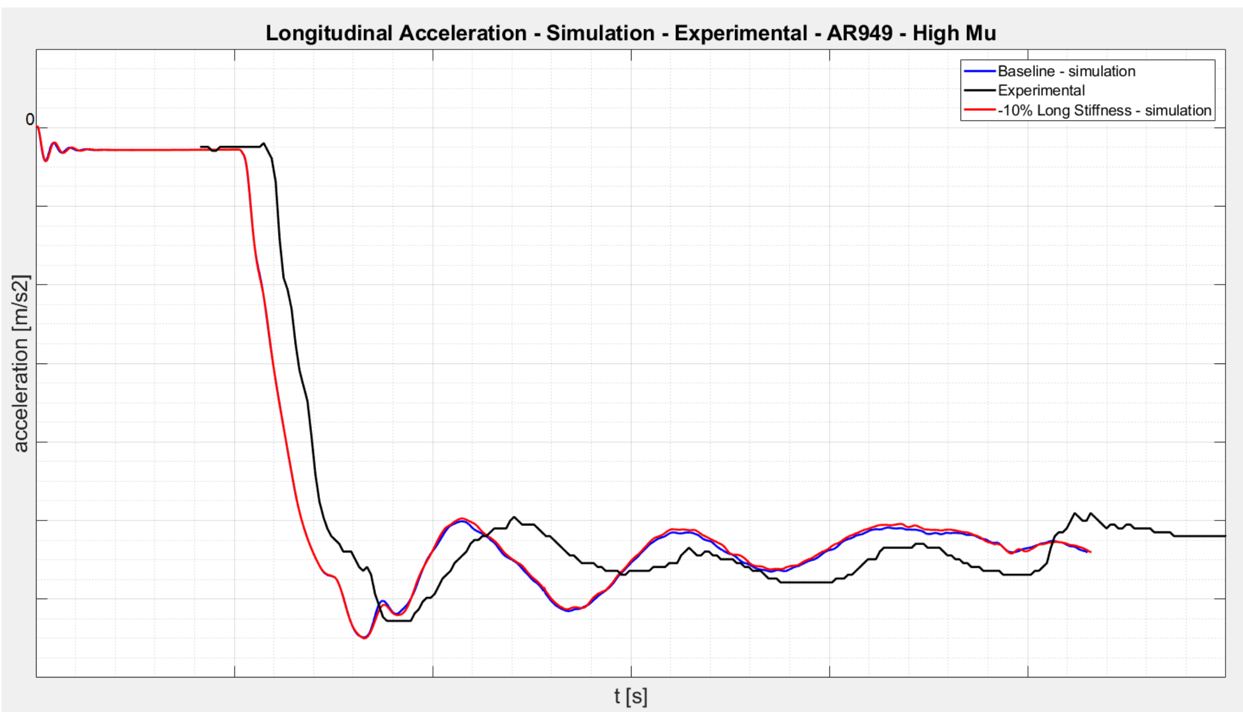


Figure 10.14.3 – Vehicle longitudinal acceleration – Baseline vs -10% Structural longitudinal stiffness

- Braking distance (figure 10.14.4): It increased 0.33%.

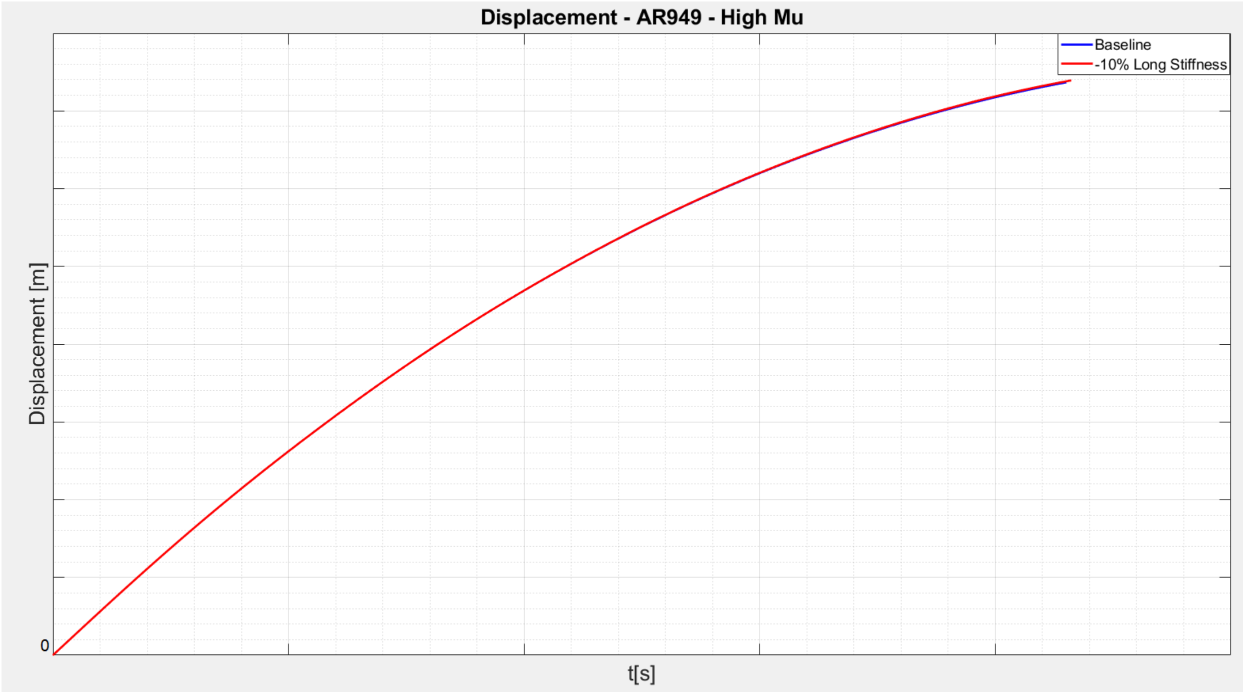


Figure 10.14.4 – Braking distance – Baseline vs -10% Structural longitudinal stiffness

10.7 Sensitivity Analysis - Overall results

The overall results of the variation of the braking distance influenced by the changes in the parameters during the sensitivity analysis are presented in the following histogram (figure 10.15):

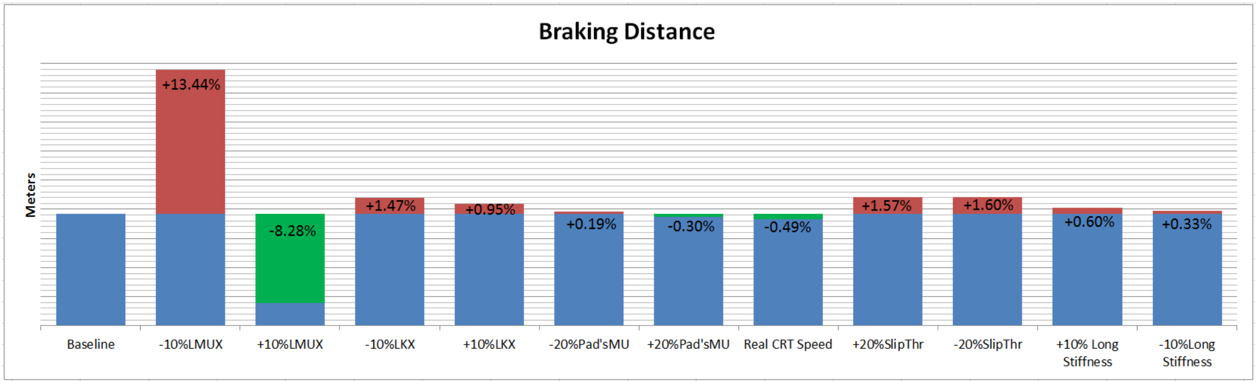


Figure 10.15 – Histogram Braking distance – Sensitivity Analysis - Overall results

It can be observed that:

- The increase of LMUX (of 10%) decreased the braking distance while the decrease of LMUX (of 10%) increased the braking distance, this means that increasing the tire's grip we have better braking performance;
- Both the increase and decrease of LKX (of 10%) increased the braking distance, this means that for this specific slip threshold, the LKX chosen is a good vale in order to obtain bigger braking forces;
- The increase of brake pad's mu (of 20%) decreased the braking distance while the decrease of the pad's mu (of 20%) increased the braking distance;
- Substituting the created speed estimator by the real speed from VI-CarRealTime, the braking distance decreases;
- Both increasing and decreasing the slip threshold (of 20%) increased the braking distance, this means that for the LKX previously chosen, the Baseline's slip threshold is a good target;
- Both increasing and decreasing the tire's structural longitudinal stiffness (of 10%) increased the braking distance.

And is good to notice that changing the LKX (tire characteristic) of a certain percentage, the slip threshold target (brake system logic parameter) has to be set again in order to obtain the maximum value of braking force.

This histogram helps to understand the parameters that influence more or less the braking distance, and the parameter that has the direct bigger influence on the braking distance is the tire's LMUX.

The same data of the histogram is now presented in a table form (table 10.1):

LMUX		
	-10%	**+10%**
Variation braking distance	+13.44%	-8.28%
LKX		
	-10%	**+10%**
Variation braking distance	+1.47%	+0.95%
PAD'S MU		
	-20%	**+20%**
Variation braking distance	+0.19%	-0.30%
SLIP THRESHOLD		
	-20%	**+20%**
Variation braking distance	+1.60%	+1.57%
STRUCTURAL LONGITUDINAL STIFFNESS		
	-10%	**+10%**
Variation braking distance	+0.33%	+0.60%
SPEED		
	CRT Real Speed	
Variation braking distance	-0.49%	

Table 10.1 – Braking distance variation – Sensitivity Analysis

11 Conclusions and Recommendation to further work

In conclusion, from the main objectives of this Master's thesis, the results are:

1. Validation of Matlab-Simulink brake model for both AR-949 (Stelvio) on dry road and AR-952 (Giulia) on wet road; with satisfactory results regarding vehicle speed estimation, vehicle acceleration, brake pressure on the four corners, estimated wheel speed, and braking distance;
2. The sensitivity analysis was useful in order to better understand how the braking distance is influenced with the variation of various brake logics parameters and structural tire parameters.

Since this thesis had a limited period of time inside the company to be developed, would be interesting as next steps:

- To validate the Matlab-Simulink model comparing with experimental results of braking maneuvers that don't enter on ABS.
- Develop logics of electronic stability program – ESP.

12 Bibliography

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Appendix A – Tire Load Capacity

Table A.1: Load capacity/air pressure category specified in the directives. The load capacity on the left – also known as 'load index' (LI) – applies for all passenger cars up to the speed symbol W; they relate to the minimum load capacity values up to 160 km h–1 at tire pressure 2.5 bar. Further criteria, such as maximum speed, handling etc., are important for the tire pressures to be used on the vehicle [9]. For LI values above 100, further load increases are in 25 kg increments:

LI = 101 corresponds to 825 kg,
LI = 102 corresponds to 850 kg etc. to
LI = 108 corresponds to 1000 kg.

Load index	*Wheel load capacity in kg with tyre pressure measured in bars* 1.5	1.6	1.7	1.8	1.9	2.0	2.1	2.2	2.3	2.4	2.5
69	215	225	240	250	260	270	285	295	305	315	325
70	225	235	245	260	270	280	290	300	315	325	335
71	230	240	255	265	275	290	300	310	325	335	345
72	235	250	260	275	285	295	310	320	330	345	355
73	245	255	270	280	295	305	315	330	340	355	365
74	250	260	275	290	300	315	325	340	350	365	375
75	255	270	285	300	310	325	335	350	360	375	387
76	265	280	295	310	320	335	350	360	375	385	400
77	275	290	305	315	330	345	360	370	385	400	412
78	280	295	310	325	340	355	370	385	400	410	425
79	290	305	320	335	350	365	380	395	410	425	437
80	300	315	330	345	360	375	390	405	420	435	450
81	305	325	340	355	370	385	400	415	430	445	462
82	315	330	350	365	380	395	415	430	445	460	475
83	325	340	360	375	390	405	425	440	455	470	487
84	330	350	365	385	400	420	435	450	470	485	500
85	340	360	380	395	415	430	450	465	480	500	515
86	350	370	390	410	425	445	460	480	495	515	530
87	360	380	400	420	440	455	475	490	510	525	545
88	370	390	410	430	450	470	485	505	525	540	560
89	385	405	425	445	465	485	505	525	545	560	580
90	400	420	440	460	480	500	520	540	560	580	600
91	410	430	450	475	495	515	535	555	575	595	615
92	420	440	465	485	505	525	550	570	590	610	630
93	430	455	475	500	520	545	565	585	610	630	650
94	445	470	490	515	540	560	585	605	625	650	670
95	460	485	505	530	555	575	600	625	645	670	690
96	470	495	520	545	570	595	620	640	665	685	710
97	485	510	535	560	585	610	635	660	685	705	730
98	500	525	550	575	600	625	650	675	700	725	750
99	515	540	570	595	620	650	675	700	725	750	775
100	530	560	590	615	640	670	695	720	750	775	800

Table A.1 - Load capacity/air pressure category

Table A.2: The tire load capacity shown in the ETRTO standards manual in the form of the load index LI is valid for V tires up to vehicle speeds of 210 km h–1; for W tires up to 240 km h–1 and for Y tires up to 270 km h–1. At higher speeds, lower percentages of the load capacity must be incurred; for VR and ZR tires, which are no longer made, these values were determined by vehicle and tire manufacturers [9].

Top speed of car ($km\ h^{-1}$)	*Tyre load capacity (%)* Speed symbol V	W	Y Tyres
210	100	100	100
220	97	100	100
230	94	100	100
240	91	100	100
250	–	95	100
260	–	90	100
270	–	85	100
280	–	–	95
290	–	–	90
300	–	–	85

Table A.2 - The tire load capacity shown in the ETRTO standards