Politecnico di Torino

Master Degree in Mechatronic Engineering

Master Degree Thesis

Modeling and Torque Split Control Strategy development of a 4WD-Hybrid Vehicle for Rally Application

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Abstract

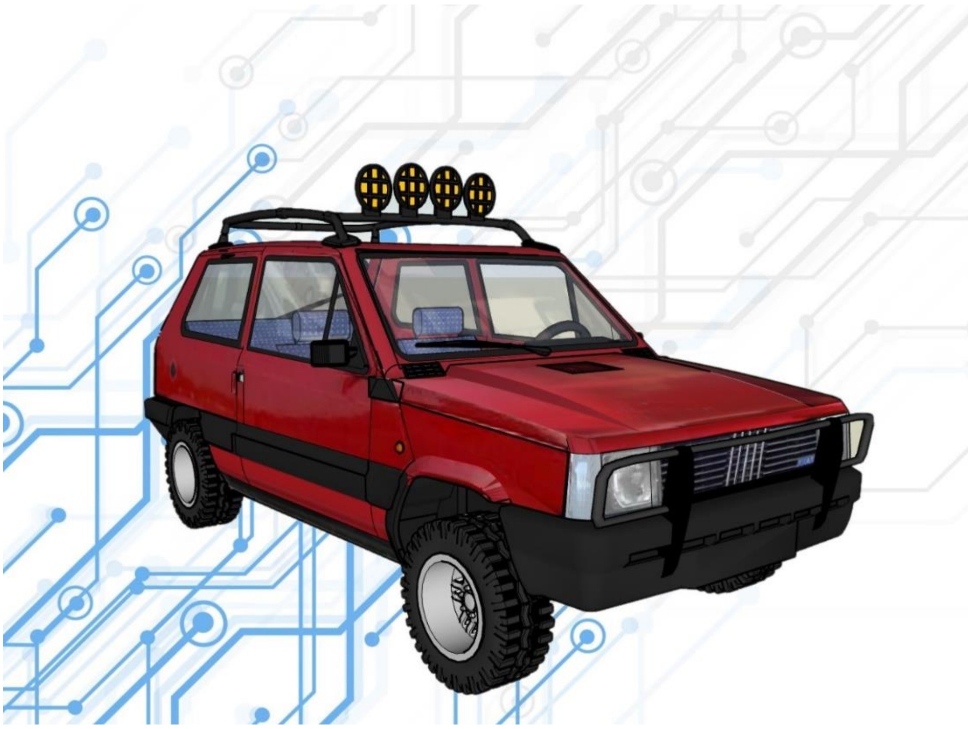
The aim of this thesis work is the conversion of a Fiat Panda into a hybrid vehicle, for the purpose of its participation to the “Panda Raid” event in the year 2020.

The Panda Raid is a long-distance amateur rally that takes place annually in March, where crews compete on board of Fiat Panda vehicles produced before 2003. Three thousand kilometers from Madrid to Marrakesh, on desert trails across endless landscapes, seven stages and approximately 400 cars involved, these are the numbers of the Panda Raid.

The Panda considered in the project is the variant 1108 i.e. cat. 4X4 produced from the ’95 to the year 2003.

The proposals of the hybrid conversion will see the addition, to the original engine, of an electric motor on the rear axle, a generator coupled to the engine on the front axle and a small battery. Consequently, a design with Independent axes will be obtained.

Once both the original and the hybrid vehicle will be modeled, a control strategy of the motors will be found and analyzed according to the simulations obtained on a specific kind of path, designed to replicate the average track faced during the Panda Raid.



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Chapter 1

Introduction

1.1 Thesis objective

In the last years, the adoption of hybrid technology on the vehicles is growing more and more for its powerful flexibility in addition to the several vantages it brings along with it, such as the regeneration mechanisms and their less dependence on fossil fuels. However, nothing prevents you to adopt this technology on an already existing car to reap its benefits in a competition, and this is exactly what will be done.

In this thesis work, a vintage vehicle was chosen to be modified because of the possibility to customize it with relative ease, in fact it doesn't adopt advanced communication technologies through its elements, like the CAN Network.

The development process of the car will follow the guidelines of the V-shaped design, allowing the modeling of the entire vehicle in the Simulink* environment with the opportunity to perform simulations of the obtained vehicle functioning.

**(Simulink, developed by MathWorks, is a graphical programming environment for modeling, simulating and analyzing multidomain dynamical systems. Its primary interface is a graphical block diagramming tool and a customizable set of block libraries. Simulink is widely used in automatic control and digital signal processing for multidomain simulation and model-based design.)*

1.2 The Panda Raid

The competition is reserved for the historic Panda models, in both the variant two-wheels drive and 4x4. The planned route from Madrid to Marrakech includes seven stages characterized by different surfaces to run through. Indeed, it probably means to face dirt roads, gravel and fine sandy dunes which could jeopardize the vehicle's mechanics.

Figure 1.1: Photo taken from the Panda Raid event, 2018.

Moreover, in the event the use of technology is not allowed: the navigation involves only the road book provided by the organization integrated with the map and the compass. The crews do not run to reach the finish line as soon as possible but the pilot and his co-pilot must reach pre-established stages within the preset time.

1.3 Project Requirements

During the Panda Raid the vehicles have to deal with:

- Long distances
- Roads difficult to go over
- Inclined paths

Thus, the vehicle must satisfy the following features:

- Ability to face long distances
- High mechanical resistance
- Ability to overcome highly inclined surfaces

(Huge performances in acceleration and speed are not necessary for this purpose.)

In order to meet all the previous conditions, the proper simulations are needed to test the vehicle subsequently modeled according to them. For this reason, the simulation model will include:

1. A 3 Degrees of Freedom model (3DOF) which considers the **longitudinal movement** of the vehicle, the **vertical position** of the axes and the following **pitch** derived. The lateral movement of the vehicle is useless for this kind of study, so it is neglected.
2. The possibility to re-create a specific condition of the ground covered through the respective adhesion coefficient and rolling resistance, similar to the ones faced in the competition.
3. The opportunity to include an inclination profile to be followed, simulating sand dunes and small obstacles.

Restrictions to follow

The proposals to modify the powertrain of the vehicle will not going to change the:

- Original Engine
- Gearbox
- Differentials

This means that it will not vary the **maximum Speed** of the vehicle, which will remain close to the 130 km/h, and the **maximum Torques** tolerable for the front and rear axles, because these are constraints depending on the design of the mechanical devices already present on the vehicle and can't be different by construction.

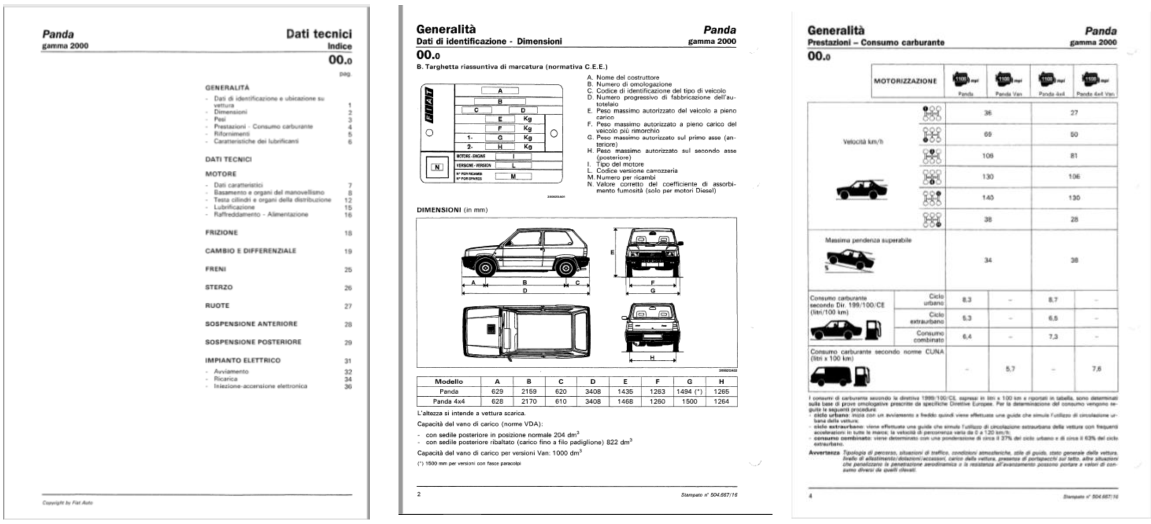
Observation: the vehicle considered allow you to supply power to all the four wheels, it means that the entire torque can be also transferred just on a single axes in particular conditions, thus the transmission resistance is designed to support the maximum engine torque through all the components.

1.4 Vehicle description

An example of the FIAT Panda 4x4 first series on which the study can be conducted is shown in the images (figure 1.2).

Figure 1.2: Panda 4x4 variant example, front and back view.

The vehicle specifications are taken from the manual attached to this document, of which some sample pages are given (figure 1.3):


Panda
gamma 2000
Dati tecnici
Indice
00.0

Generalità
Dati di identificazione - Dimensioni
Panda
gamma 2000
00.0

Generalità
Prestazioni – Consumo carburante
Panda
gamma 2000
00.0

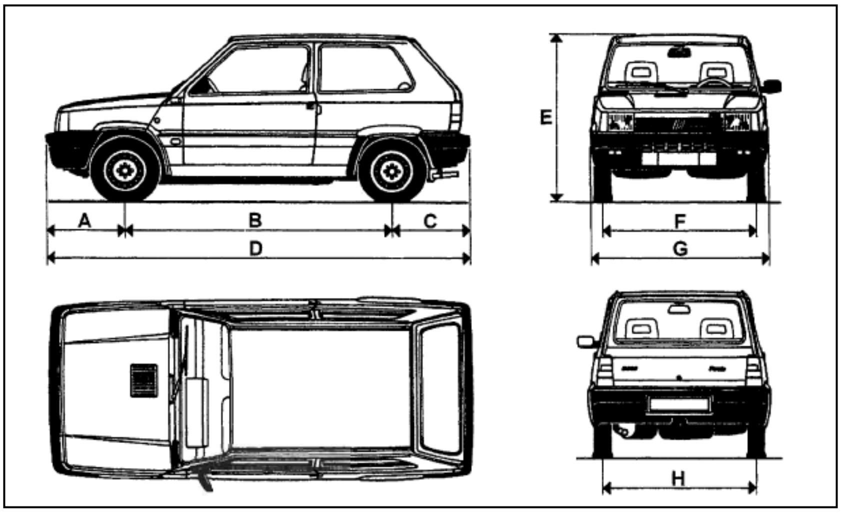
Figure 1.3: Samples of Panda series 2000 Datasheet.

The vehicle is described below, analyzing the following aspects:

- Weight and Dimensions
- Engine
- Transmission

1.4.1 Weight and Dimensions

A scheme of the vehicle's external dimensions is given below. The table shows the vehicle dimensions and the unladen mass (with liquids) before the modifications (figure 1.4).



Size	A	B	C	D	E	F	G	H	Max Mass [Kg]
Dimension [mm]	628	2170	610	3408	1468	1260	1500	1264	1200

Figure 1.4: Vehicle's external dimensions and mass.

The mass must then be modified in the simulations considering the weights of batteries and electric motors.

In addition, the maximum tolarable axle loads are given in the manual:

	MOTORIZZAZIONE		1108 mpi Panda	1108 mpi Panda Van	1108 mpi CITIVAN	1108 mpi Panda 4x4	1108 mpi Panda Van 4x4	1108 mpi CITIVAN 4x4
Carichi massimi ammessi sugli assi ■	asse anteriore		580	580	580	590	590	590
	asse posteriore		630	630	630	680	680	680

Figure1.5: Panda 4x4 1108cc maximum load per axis.

As shown in the figure (1.5), the maximum loads that can be tolerate for the 4x4 variant are:

- 590 kg on the front axis.
- 680 kg on the rear axis.

1.4.2 Engine

The panda is equipped with the 1108 FIRE thermal engine which operates according to an Otto cycle and 4-stroke. The engine characteristics are shown in the table (figure 1.7), while the engine characteristic curves are shown in the figure. The power curve (figure 1.6) shown is the one obtainable when the engine is overhauled and run-in (50 hours of operation), without fan, with exhaust silencer and air filter, at sea level. Injection is integrated electronic, MPI - I.A.W. type. Weber Marelli, there is an electric pump immersed in the tank with fuel pressure regulator set at 3 bar.

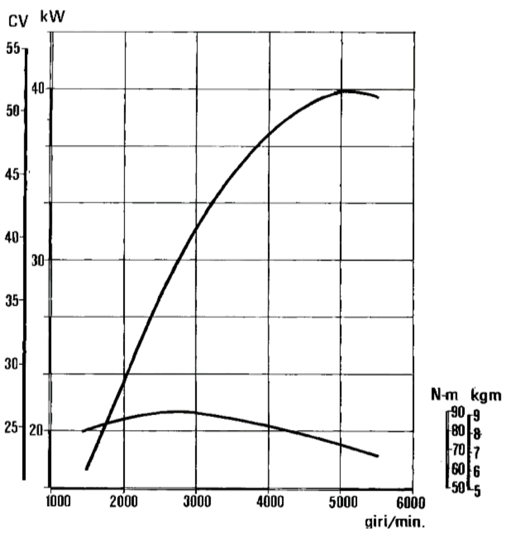


Figure 1.6: Power and Torque curve wrt Engine speed.

Cylinders	4 in-line
Capacity [cm^3]	1108
Compression ratio	9,6 ±2
Maximum Power KW (CV)	40 (54) at 5000 rpm
Maximum Torque (Nm)	88 at 2750 rpm

Figure 1.7: Engine characteristics.

1.4.3 Transmission

The transmission system allows the flow of power from the thermal engine to the driving wheels, which can be 2 under normal conditions, or 4 by using a lever located near the gearbox which, if raised, allows the activation of the 4x4 with transmission of the driving torque also to the rear axle (figure 1.8).

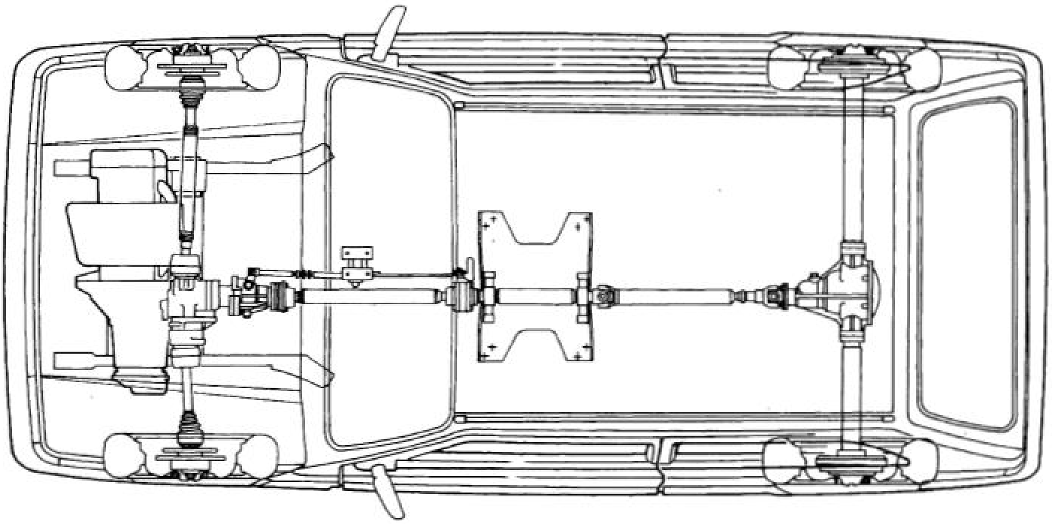


Figure 1.8: Panda 4x4 Transmission scheme.

This system has been designed and developed with the collaboration of the Steyer-Puch.

At the exit of the gearbox was placed the traditional cylindrical torque that moves the front differential, but in this case, however, next to the cylindrical gear there is another conical that moves a short longitudinal shaft, in practice, it is like if the differential is connected with two final transmissions: a cylindrical and a conical. At the end of the longitudinal shaft there is a coupling sleeve which, at the command of the lever in the cockpit, connects the transmission to the shaft which transmits the motion to the rear wheels.

The transmission shaft is divided into three parts: the mobile front one has the task of compensating the movements of the motor group and has two simple sliding joints at the ends; the short central segment is fixed by means of a support connected to the platform; at the rear end of this shaft the final section is connected, by means of a cardan joint, which, in turn, is connected to the differential with another cardan joint. The third part of the drive shaft, due to the rather large movement of the rear axle, also has an extension joint.

The following tables show the reduction ratios of the gearbox gears for each gear, the reduction ratio of the cylindrical torque with the differential wheel, the bevel gear reduction ratio and finally the reduction ratios on the wheels obtained by multiplying the first two transmission ratios (figure 1.9).

DIFFERENZIALE - RINVIO		Panda 1108 mpi	Panda 4 x 4 1108 mpi
Rapporto coppia cilindrica di riduzione		16/57 (3,562)	11/60 (5,455)
Rapporto sulle ruote	1	13,924	21,324
	2	7,868	11,215
	3	4,791	6,939
	4	3,469	5,335
	5	2,728	3,988
	R	13,600	20,331
Rapporto ingranaggi	1	3,909	3,909
	2	2,158	2,056
	3	1,345	1,272
	4	0,974	0,978
	5	0,766	0,731
	R	3,818	3,727
Rapporto	coppia cilindrica di riduzione	–	14/41 (2,929)
	coppia cilindrica di riduzione	16/57 (3,562)	11/60 (5,455)

Figure 1.9: Panda 4x4 Transmission Datasheet.

Chapter 2

Simulation Model

In the following paragraphs of this chapter it will be analyzed the Simulink model used for the simulations, the modeling has been realized according to the so called “V-shaped” design logic.

2.1 V-shaped diagram

A model is based on an original system and it reflects its relevant properties in order to be used in place of it for several purposes. The model-based software design according to the V-shaped diagram is a graphical representation of the process development of an item. It summarizes the main steps to follow during the item’s design (figure 2.1).

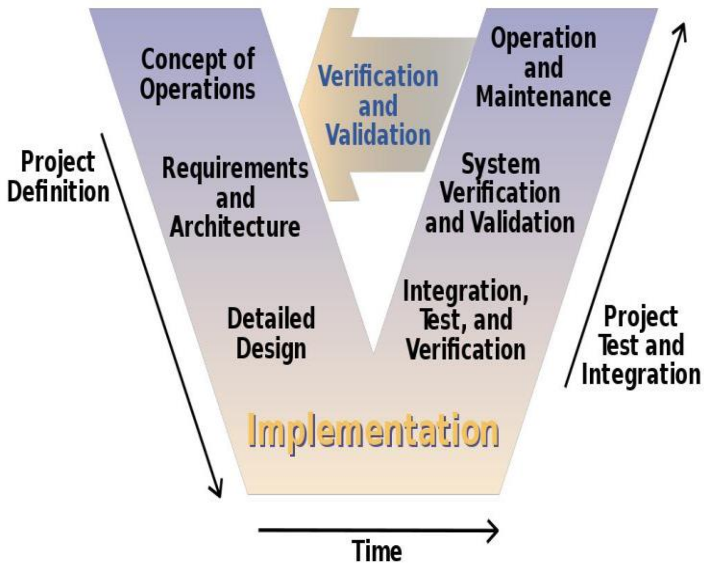


Figure 2.1: V-Sha2ped Diagram for model-based design.

The diagram can be divided into three main sections:

- Project definition, where the system is analyzed from its requirements point of view and its design is translated into a suitable software with a simulation tool.
- Implementation, here the software is implemented as a code.
- Test and integration, the code obtained is analyzed for testing the correct implementation of the original model on the final hardware.

Model in the Loop section

This thesis work doesn't provide for the implementation of the software into a code, thus the study will concern only with the "model in the loop" design, where the model exists entirely in the native simulation tool (Simulink environment in this case). It is good for the control algorithm development and the simulation of the system (a 3dof vehicle).

2.2 Panda 1108cc 4x4 1° gen. Model

The definition of the simulation environment starts with the modeling of the original vehicle Panda 1108cc 4x4 (figure 2.2), then new blocks will be added to the description in the section related to the hybrid project.

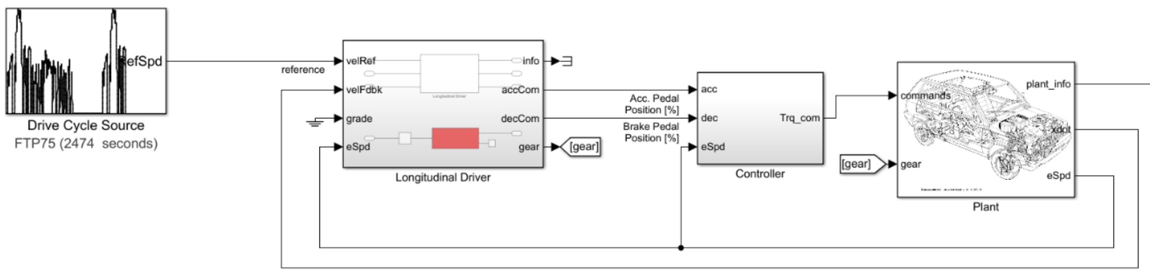


Figure 2.2: Final Simulink model of the original vehicle.

The scheme is divided into 4 main blocks:

- Track (with a Driving Speed Profile).
- Driver.
- Control block.
- Vehicle.

2.3 Track

2.3.1 Driving Speed Profile

One of the elements without which the model cannot work is the Driving Speed Profile, necessary to give to the driver a reference in speed to follow during the simulation, so it works as a target to be achieved and, measuring the potential differences between the reference and the speed reached, it can be determined if a particular vehicle model is better than another one.

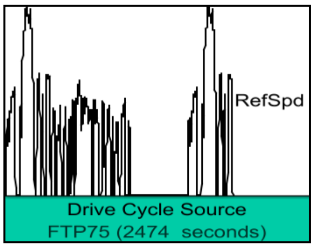


Figure 2.3: Example of Driving Speed Profile Block.

The Simulink block "Drive Cycle Source" (figure 2.3) can generate a standard or specific guide loop created ad hoc to simulate a specific path. The block defines a reference speed profile in output.

However, in order to make the path imposed more realistic, other information can be added to the track, as the inclination grade of the surface and an accurate level of adhesion and rolling resistance related to the kind of path selected.

2.3.2 Inclination profiles

The inclination profiles will be modelled by a variation of the ground inclination angle during the simulation period. For this reason, it can be simulated whatever kind of obstacle depending on the simulation parameters.

As long as the input given is an inclination measured in grade changing in time, the height and the size of the hill to face will be directly proportional to the actual speed of the vehicle, so it can be obtained hills or very small obstacles just combing speed and duration of inclination value imposed.

The necessary input will be generated within the vehicle block and will be shown later.

2.3.3 Adhesion coefficients

The dynamics of the tires in contact with the ground is based on the Pacejka model.

Hence, to simulate the effect of friction between the ground and the wheels of the vehicle, the parameters of the "Magic formula Constant Value" of the tyre model will be modified, as explained later (paragraph 2.6.3), according to the asphalt conditions (figure 2.4):

COEFFICIENTS →	D-peak value	C-shape factor	B-stiffness factor	E-curvature factor	a	b	c	α	β
SURFACE ↓									
Dry Asphalt	1	1.9	10	0.97	0.013	0	0	0	1
Wet Asphalt	0.82	2.3	12	1	0.015	0	0	0	1
Soil/Sand	0.5	2	10	1	0.05	0	0	0	1
Snow	0.3	2	5	1	0,025	0	0	0	1
Ice	0.1	2	4	1	0.013	0	0	0	1

Figure 2.4: Magic Formula Parameters for several surfaces.

The two most effective parameters are the D-peak value and the rolling resistance coefficient which, put together with the remaining ones allow to obtain different dynamics in the adhesion and in the rolling resistance to the variation of the surfaces considered.

These values are proposed by default in the Simulink environment, they are typical sets of constant Magic Formula coefficients for common road conditions, except for the ones related to the wheel Dynamics in the sand, properly chosen.

2.3.4 Selection of Soil/Sand Coefficients

For the soil/sand surface, the appropriate values for the parameters C, B, E have been chosen accordingly to those previously assigned in cases of wet and snow asphalt.

Figure 2.5: Sample of the kind of track faced during the Panda Raid.

The a-rolling resistance coefficient, selected to simulate the contact between the wheels and the soil, is obtained looking at the worst case among the Gravel and the Dirt Road surfaces (figure 2.5 v figure 2.6).

A variable value, instead, is assigned to the coefficient D of adhesion (figure 2.7), in order to consider the variation of adhesion during the simulation in a range of [0.4-0.6], considering a mean value of 0.5 derived from the Dirt Road value rounded down (decreasing this value it's examined a worse adhesion condition).

Surface	D-Peak friction coefficient	a-Rolling resistance coefficient
Asphalt or concrete	~ 0.9	0.013
Asphalt or concrete	~ 0.8	0.015
Gravel	0.6	0.025
Dirt road	0.55	0.05
Ice	0.1	_

Figure 2.7: Table for Soil Parameters determination.

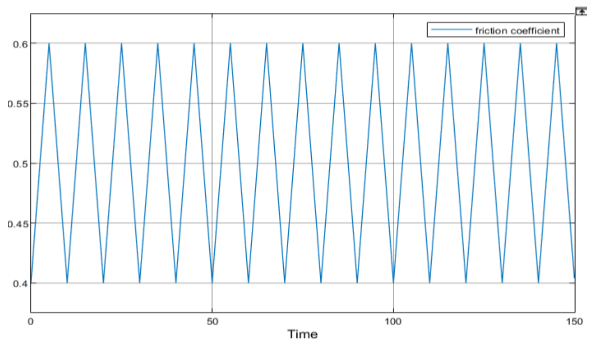


Figure 2.6: D coefficient variation during the simulations.

2.4 Driver

Normally the driver bases its commands of acceleration and deceleration on the pedals depending on the speed and gears suggested by the copilot, the track and the vehicle's responses. Thus, the driver model has been developed considering these information.

The driver block is composed of two different blocks joined together:

- the **Longitudinal Driver** available in the Simulink environment
- an external **Gear Selection** logic properly customized.

2.4.1 Longitudinal Driver with PI Speed-tracking control

The Simulink virtual driver (figure 2.8) receives the input:

1. The reference Speed profile
2. The speed of the vehicle
3. The inclination of the road

Output returns:

1. Acceleration command
2. Deceleration command

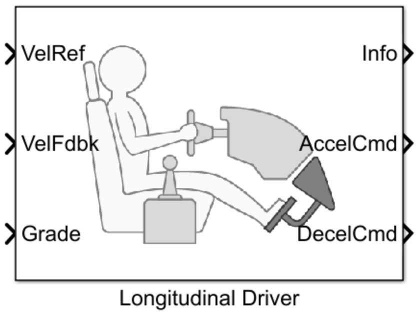


Figure 2.8: Simulink Longitudinal Driver Block.

The block implements a longitudinal speed-tracking controller, modeling the dynamic response of a driver who wants to follow a specific velocity profile during the simulation.

Thus, based on reference and feedback velocities, the block generates normalized acceleration and braking commands that can vary from **0** through **1**.

Setting the control type to PI, the block implements proportional-integral (PI) control with tracking windup and feed-forward gains.

To calculate the speed control output, the block uses this equation.

$$y = \frac{K_{ff}}{v_{nom}} v_{ref} + \frac{K_p e_{ref}}{v_{nom}} + (\frac{K_i}{v_{nom}} + K_{aw} e_{out}) \int e_{ref} dt + K_g \theta$$

equation 2.1

where:

$$e_{ref} = v_{ref} - v$$

$$e_{out} = y_{sat} - y$$

The equation uses these variables:

vnom	Nominal vehicle speed	[m/s]
Kp	Proportional gain	_
Ki	Integral gain	_
Kaw	Anti-windup gain	_
Kff	Velocity feed-forward gain	_
Kg	Grade feed-forward gain	[1/ °]
θ	Grade angle	[°]
y	Nominal control output magnitude	_
ysat	Saturated control output magnitude	_
eref	Velocity error between actual and reference vehicle speeds	[m/s]
eout	Difference between saturated and nominal control outputs	_
v	Velocity feedback signal	[m/s]
vref	Reference velocity signal	[m/s]

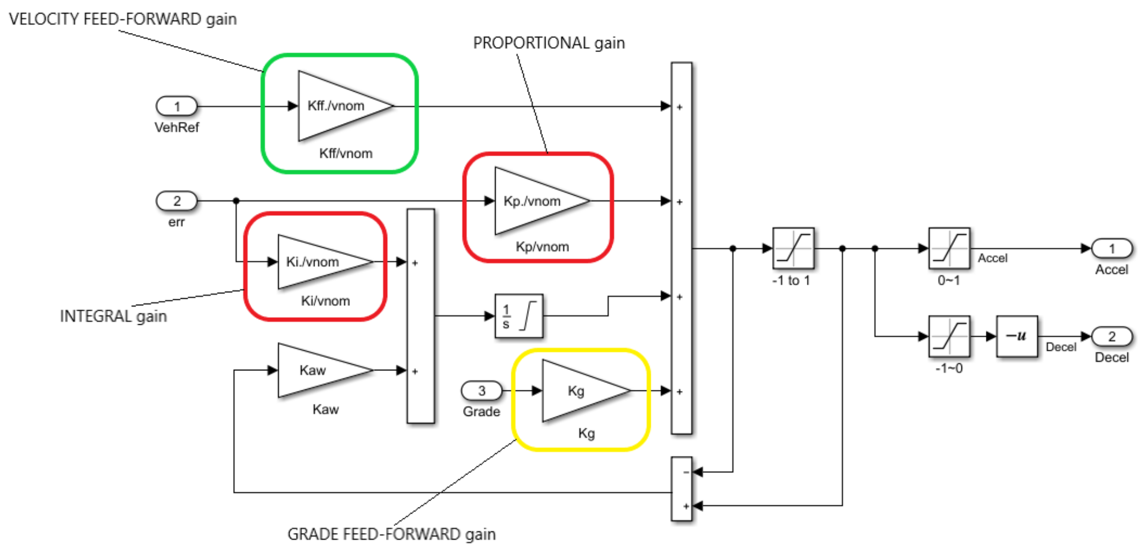


Figure 2.9: Longitudinal Driver Block, Internal Logic.

The main part of the longitudinal driver logic is shown above (figure 2.9) and it illustrates all the elements described in the equation 2.1.

The upper one is a velocity feed-forward gain, so it considers only the velocity desired. On the contrary, the elements in the middle depend on the error computed between the velocity desired and the actual one reached by the vehicle, so a feedback is needed. The lower gain, as the first one, works only with the velocity value desired.

By choosing these gains, the user gets a bigger or smaller sensibility in the response of the driver to the velocity reached and the inclination of the path (this thesis work does not include this kind of study, for this reason the Simulink default parameters have been used).

2.4.2 Gear Selection

The gear selection logic (figure 2.10) takes as input the engine speed and returns the inserted gear.

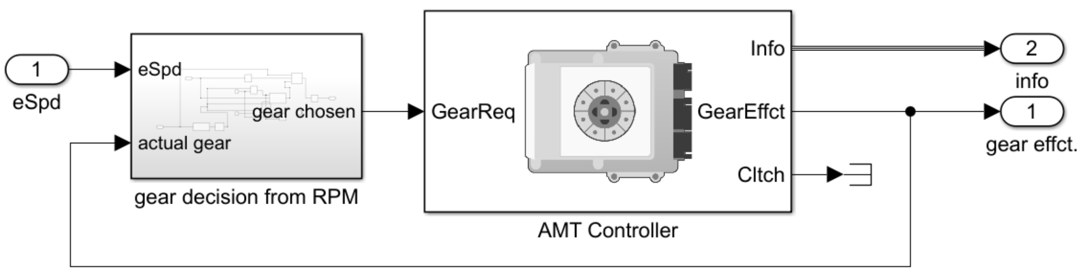


Figure 2.10: Gear selection Logic.

In detail the gear selection system includes:

A decision group that decides which gear has to be used according to the number of revolutions of the engine (left)

An AMT controller (Automated Manual Transmission) that determines the command to the clutch according to the required gear and the time necessary for the activation of the clutch itself. The estimated time to actuate the clutch is 0,2 seconds.

The decision group on the left selects a gear:

- Higher if the motor RPM exceed a certain threshold. ▲
- Lower if the motor revolutions are lower than a smaller threhold. ▼

This logic has been designed for a kind of guide that uses the engine in a speed range that guarantees good torque values, found between the 2000 and the 4000 RPM (figure 2.11)

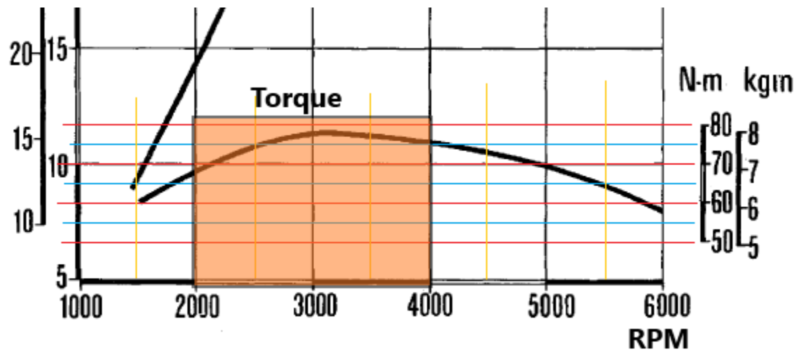


Figure 2.10: Engine Torque curve, optimal range.

2.5 The control system

The control system is a command group of the vehicle that takes in input:

- Acceleration command.
- Deceleration command.

It gives in output:

- Torque command to the engine and a command for the braking system.

The logics used in the control block are represented in the figure 2.12:

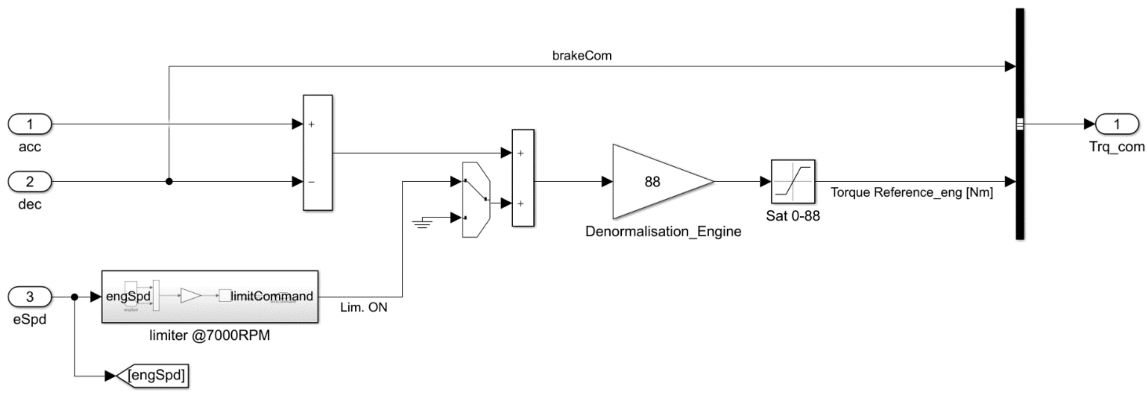


Figure 2.11: Engine Controller Logic designed, 4x4 vehicle.

The acceleration command is normalized according to the capacity of the engine (in this case it is capable to provide a maximum torque value of 88 [Nm]) and it will be sent directly to the engine block, working as a reference. The reference value of torque requested for the engine will be surely higher than the one actually reached by it that, as the name says, will be the real one which will act on the car.

The braking command has been modeled as a signal broadcasted by the torque command but, obviously, it will not be sent to the engine but to the braking system implemented in the wheels.

Importance of the Control Block

The control block allows the user to implement every kind of logic to control the car, in fact adding functions to this block it will be possible to make a smart use of the car's engines based on every signal available in the model.

To give an example of the control possibilities allowed, it has been inserted in the figure above (2.12) a limiter designed for the engine, which works taking as input the engine speed feedback coming from the vehicle.

2.6 Vehicle

The vehicle block represents the plant of the model, it has to reproduce in the best way possible the dynamics involved in the functioning of the vehicle, it follows that it will simulate:

- the transient phenomena of the combustion.
- the reduction ratio chosen by the driver together with all the rotational speed of the other elements.
- the contact between wheel and ground; the dynamics of the chassis etc.

The scheme of the vehicle model is shown in the figure 2.13 while the figure 2.14 shows the designed logic for all the vehicle model elements except for the engine.

Figure 2.12: Scheme of the Vehicle Model designed.

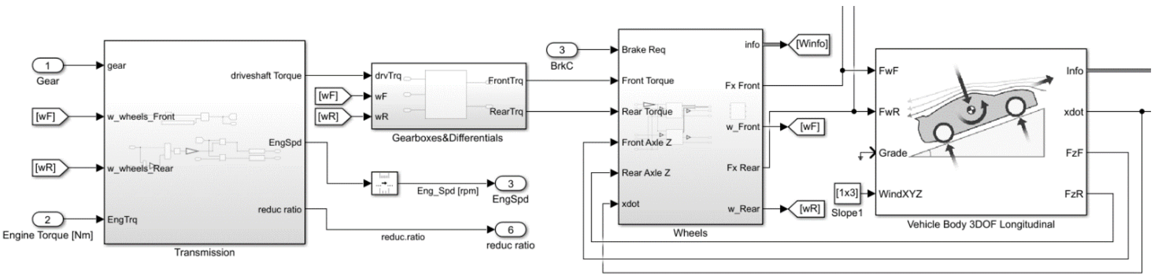


Figure 2.13: Vehicle Logic designed for transmission, wheels, dynamics.

2.6.1 Engine

This block (figure 2.15) takes in input:

- Engine torque command.
- Engine rotational speed.

It returns:

- The actual torque supplied by the engine.

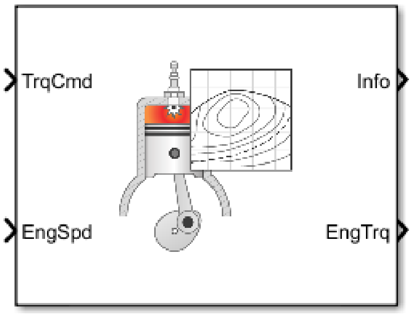


Figure 2.14: Spark Ignited Engine Block.

In order to model the engine to operate in accordance with the 1108 Fire engine that the Panda mounts (figure 1.6), the following parameters had to be estimated:

- Firing map
- Non Firing Map
- Air Mass Flow Rate
- Fuel Mass Flow Rate

Firing Map

This has been obtained from the torque curves found in the technical manuals.

Non Firing Map

This is the operating curve of the motor with zero torque drive. As there is no map available, the motor's resistant torque values are estimated.

Air Mass Flow Rate

It is calculated as:

$$cycle\ per\ second = \frac{rpm}{2 * 60}$$

Note: Two turns of the crank are required for a thermodynamic cycle.

$$Air\ Mass\ Flow\ Rate = \ engd * \rho_{air} * cycle\ per\ second$$

Where:

- $engd$ is the engine capacity: 1,108 * $10e^{-3}$ [m^3].
- ρ_{aria} is the air density N.C.: 1,225 [kg/m^3] at 1atm, 15 Celsius degrees.

Fuel Mass Flow Rate

A perfect blend with an Air-to-Fuel ratio of 14.7 was assumed.

$$Fuel\ Mass\ Flow\ Rate = \frac{Air\ Mass\ Flow\ Rate}{14{,}7}$$

After the mapping, the engine model will work according to the graph in the figure 2.16.

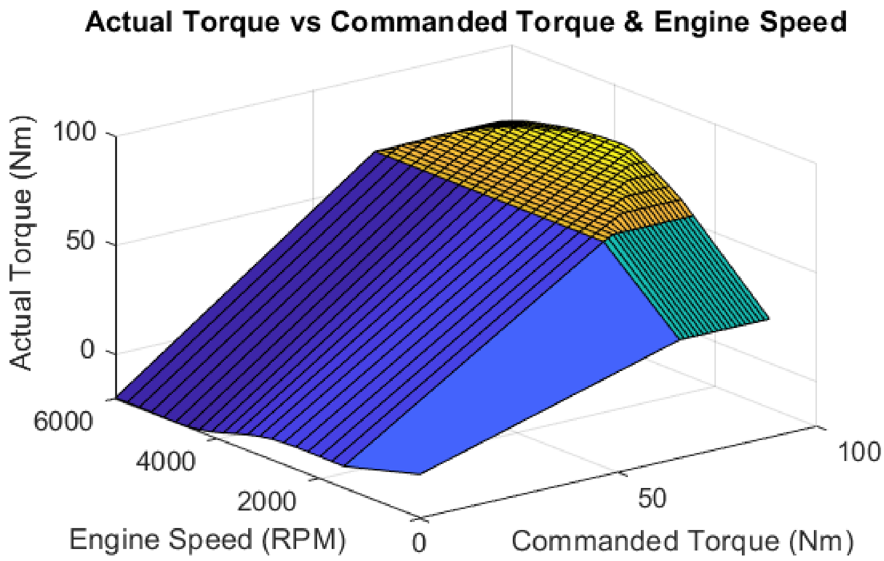


Figure 2.15: Engine Mapp Designed for Panda 4x4 1108cc.

2.6.2 Transmission

To make the model more readable and clearer for the user, the transmission model has been divided into two groups: Gearbox and Differentials (figure 2.17).

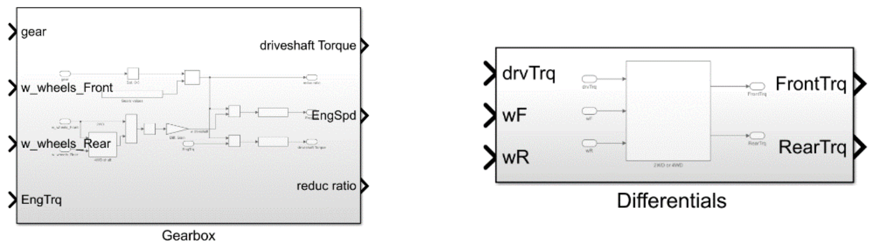


Figure 2.16: Gearbox and differentials blocks.

Gearbox

The block takes in input:

- the Engine Torque.
- The gear selected.
- The speed of the front wheels.
- The speed of the rear wheels.

It gives in output:

- The Engine Speed.
- The torque on the drive shaft.
- The reduction ratio related to the selected gear.

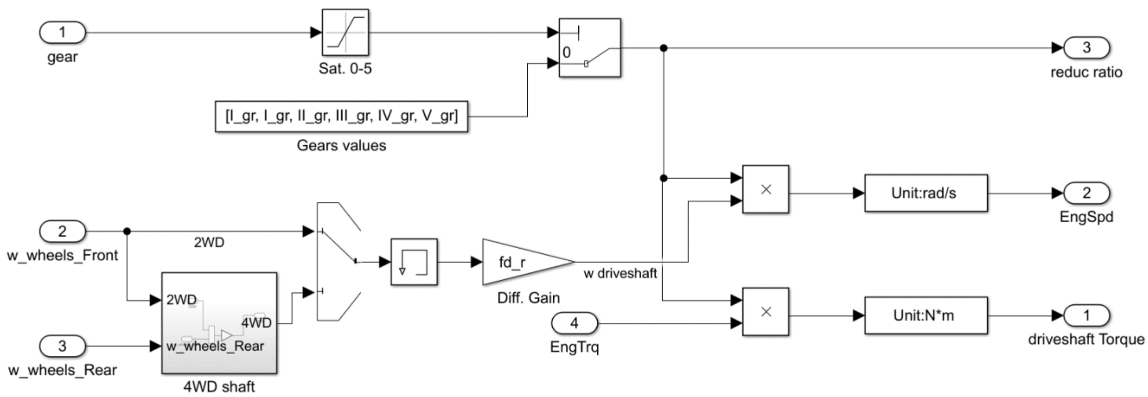


Figure 2.17: Logic designed for the gearbox.

The internal logic (figure 2.18) selects the reduction ratio value according to the gear inserted and performs the necessary multiplication on the driveline. The values of reduction ratios used within this module derive from the original car manual, shown in the preceding paragraphs.

This logic changes beteween the 4x2 and 4x4 versions through a lever, since in the first one the feedback from which the engine speed is derived is just the velocity of the front wheels while the rear wheels are conducted and totally separated from the driveline.

Differentials

The group of differentials is shown in the figure 2.19.

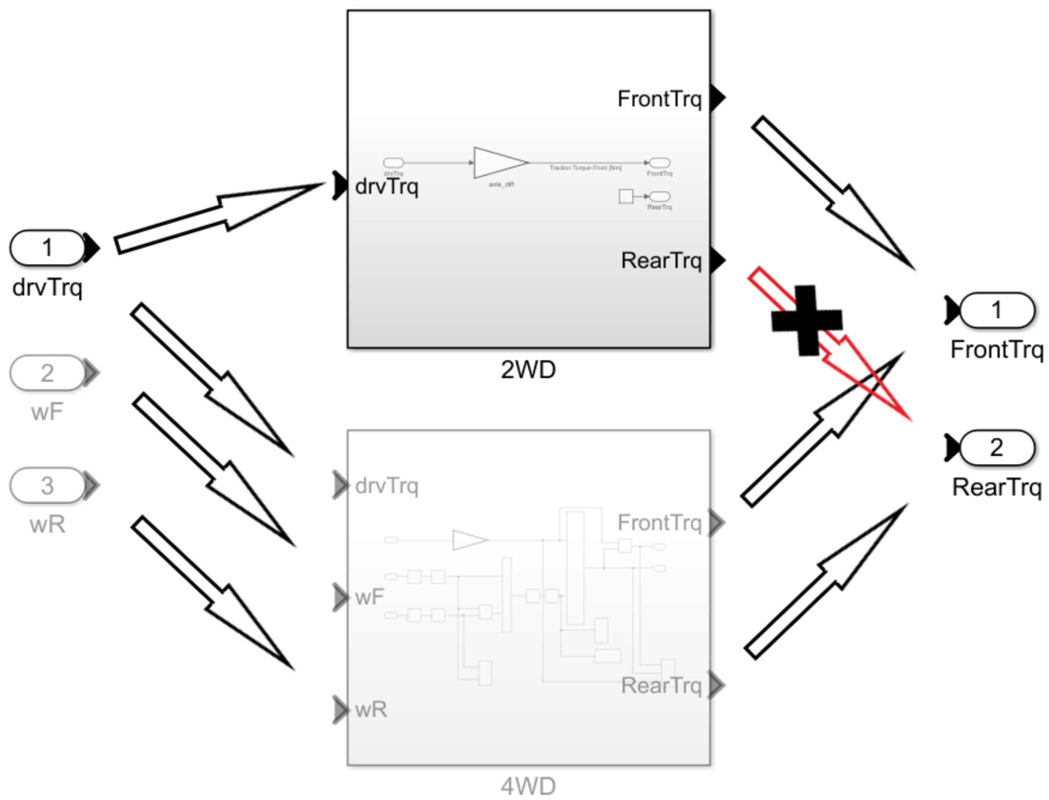
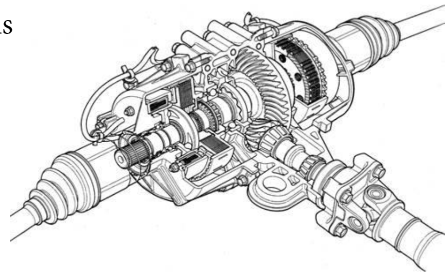


Figure 2.18: Two model variants, 2WD v 4WD.

Consequently, the block of the differentials changes too depending on the 4x2 or 4x4 traction chosen for the simulation. Here the logic is totally different between the two models, so the user has to select the proper variant to use within the main block.

The block takes in input:

- The Torque of the driveshaft
- The speed of the front wheels and the rear wheels



It gives in output:

- The Torque on the Front axle
- The Torque on the Rear axle

When the only front wheel drive (**4X2**) is inserted, the torque supplied by the engine will only move the front differential, so in the presence of low grip the front wheels could turn at speeds higher than those imposed by the ground. The torque given the rear axle will be grounded, so it has always a null value in this kind of simulation.

If **4x4** traction is inserted, the differentials will have to work at the same speed by construction, it follows that the axle with greater resistance in motion will affect the movement of the other one and it will be distributed more torque on it although, in ideal adhesion conditions, the driving torque on each axle will be exactly equal to half of the available one. Further features of this transmission system will be analysed in the following paragraphs.

2.6.3 Wheels

The wheel simulation block (figure 2.20) receives the input:

- The braking request command.
- The Torque on the front axle.
- The Torque on the rear axle.
- The vertical forces on the front and rear axle.

Output returns:

- The longitudinal force on the front axle.
- The longitudinal force on the rear axle.
- The Front and Rear wheel rotation speed.

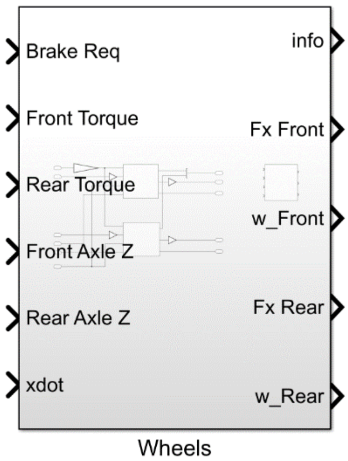


Figure 2.19: Wheels block.

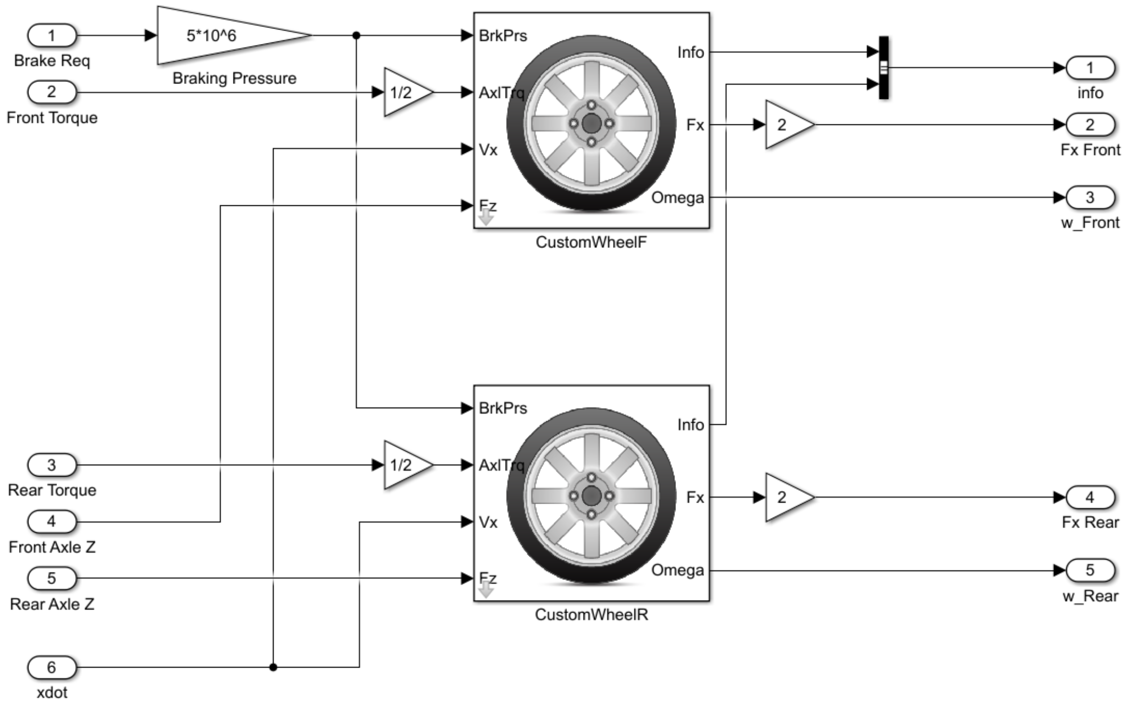


Figure 2.20: Internal logic designed for the wheel model.

It can be noticed that here are modeled just two of the four car wheels, this is because a longitudinal model is used and we don't care about the difference in rotation through the left and right wheel, but it is considered one wheel per axis, it follows that the rotation of these wheels corresponds to the speed of the rear and frontal differentials.

The block so developed allows to convert the braking command in a new higher value which considers approximately the pressure of the braking system that actually acts on the wheels.

Both signals of the torque acting on the wheel and the longitudinal force obtained takes into account that in this model each wheel used represents an entire axis, so there are a couple of gain element to fix this aspect.

The vertical loads data, coming from the vehicle model as feedback, are not divided because the car block has been made aware of the "single tire per axis" logic and it gives in outuput the informations on the vertical loads already divided by a factor of 2.

Wheel model

This is, without any doubts, both the most complex and important block needed for this thesis work.

The longitudinal wheel block, available in Simulink, implements the longitudinal behaviour of an ideal wheel according to the "Pacejka Magic Formula Tyre Model" [fonte], a widely used semi-empirical tyre model to calculate steady-state tyre force and moment characteristics for use in vehicle dynamics studies.

In this model the combined slip situation is designed from a physical point of view. It implements the stationary function of the tyre, the so called magic formula:

$$F_x = f(\kappa, F_z),$$

where *Fx* is the longitudinal force supplied by the wheel, based on **Vertical Load *Fz*** (in Newton) and **Wheel Slipping *k***. The Magic Formula typically produces a curve that passes through th origin *k=Fx=0,* reaches a maximum and subsequently tends to a horizontal asymptote.

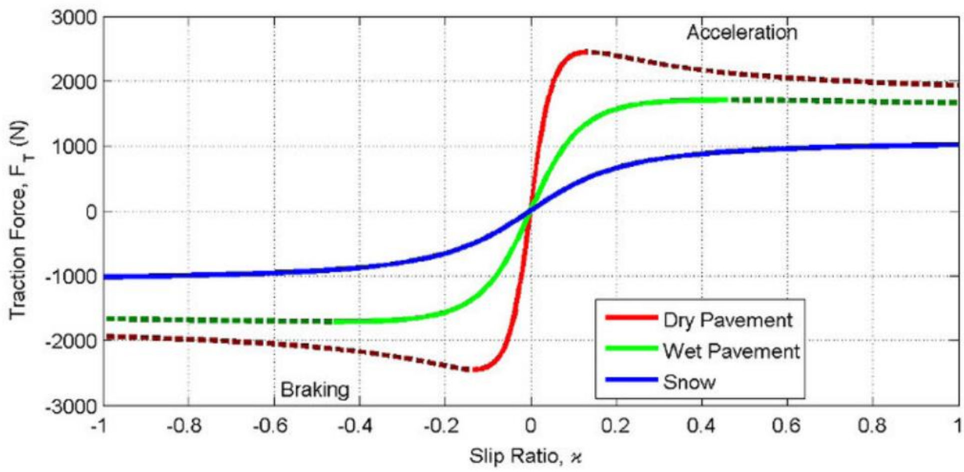


Figure 2.21: [example traction curves obtained on different surfaces according to the Pacejka m.f. (for a fixed value of the load Fz)].

As mentioned, the longitudinal force depends on both the vertical load and the slipping coefficient, but is highly more profitable to link the behaviour of this function to a set of adimensional unitary coefficients [B, C, D, E], extending the study in such a way it can be flexible to every kind of surface taken into account (like shown in the figure 2.22). Then the Magic Formula is so characterized:

$$Fx = f(\kappa, Fz) \ = FzDsin(Ctan^{-1}[\{B\kappa - E[B\kappa - tan^{-1}(B\kappa)\ ]])$$

equation 2.2

The angular coefficient of f in $\kappa = 0$ is $BCD \cdot Fz$.

Coefficient **D** represents the peak value of the force, since it is directly proportional to the vertical load, like a normal friction coefficient.

The shape factor **C** controls the limits of the range of the sine function appearing in the formula and thereby determines the shape of the resulting curve.

The factor **B** is the stiffness factor and it determines the slope at the origin of the curve.

The factor **E** is introduced to control the curvature at the peak and its horizontal position too.

Going back to the general dynamic of the wheel, the radius chosen for the tyre is 0.27 m and it has been estimated. Other parameters are selected through the wheel block interface in the following configuration:

- Longitudinal force: Magic formula constant Value
- Rolling resistance: Pressure and velocity
- Vertical motion: Mapped stiffness and damping
- Brake type: Disc

The block calculates the inertial response of the wheel subject to:

- Braking torque or acceleration.
- Rolling resistance.
- Contact with the ground on the pneumatic-road interface.

The input torque is the sum between the engine torque on the axle, the braking torque and the traction torque associated with the tyre.

$$T_i = T_a - T_b + T_d$$

equation 2.3

The traction torque is self-calculated within the block implementing the wheel traction force and the rolling resistance with first order dynamic equations. The rolling resistance has a time constant parametrized in terms of wheel size at rest (L_e).

$$\dot{T}_d = \frac{\omega R_e}{L_e + \omega R_e}\left(F_x R_e + M_y\right)$$

equation 2.4

2.6.4 Dynamics

The module implements the 3 degrees of freedom dynamics of the vehicle (figure 2.23).

It takes in input:

- the total forces exchanged between wheels and ground.
- the road inclination (expressed through the inclination grade).

It gives in output:

- the vehicle speed.

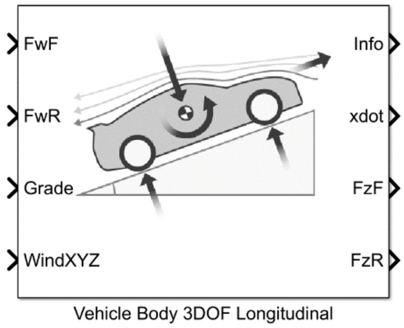
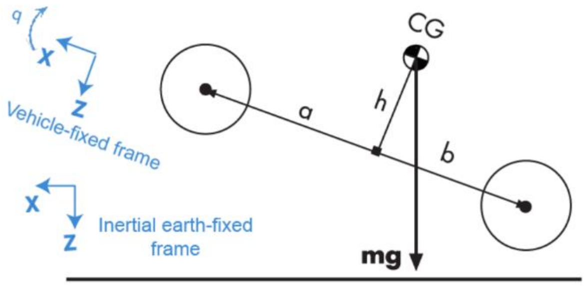


Figure 2.22: Longitudinal Vehicle block.

3 degrees of freedom Body 3DOF

It is a rigid body model with configurable axle stiffness, to calculate:

1. Longitudinal movements
2. Vertical movements
3. Pitch



The block accounts for body mass, aerodynamic drag, road incline, and weight distribution between the axles due to acceleration and the road profile.

The block uses rigid-body vehicle motion, suspension system forces, and wind and drag forces to calculate the normal forces on the front and rear axles. The block resolves the force components and moments on the rigid vehicle body frame:

$$F_x = FwF + FwR - Fd,x - Fsx,F - Fsx,R + Fg,x$$

$$F_z = Fd,z - Fsz,F - Fsz,R + Fg,z$$

$$M_y = aFsz,F - bFsz,R + h(FwF + FwR + Fsx,F + Fsx,R) - Md,y$$

Rigid-Body Vehicle motion

If the vehicle is traveling on an inclined slope, the normal direction is not parallel to gravity but is always perpendicular to the axle-longitudinal plane. The weight of the vehicle acts through its center of gravity (CG) and depending on the inclined angle, it pulls the vehicle to the ground and either forward or backward.

The block uses the net effect of all the forces and torques acting on it to determine the vehicle motion. The longitudinal tire forces push the vehicle forward or backward.

$$\ddot{x} = \frac{F_x}{m} - qz$$

$$\ddot{z} = \frac{F_z}{m} - qx$$

$$\dot{q} = \frac{M_y}{I_{yy}}$$

$$\dot{\vartheta} = q$$

Suspension System Forces

The block uses nonlinear stiffness and damping parameters to model the suspension system.

The front and rear axle suspension forces are given by:

$$F_{S_F} = N_F\left[F_{k_F} - F_{b_F}\right]$$

$$F_{S_R} = N_R\left[F_{k_R} - F_{b_R}\right]$$

The block uses lookup tables to implement the front and rear suspension stiffness and damping. To account for kinematic and material nonlinearities, including collisions with end-stops, the stiffness tables are functions of the stroke. Similarly, to account for nonlinearities, compression, and rebound, the damping tables are functions of the stroke rate.

$$F_{k_F} = f(dZ_F)$$

$$F_{k_R} = f(dZ_R)$$

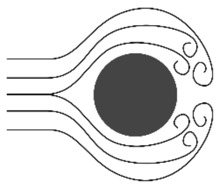
$$F_{b_F} = f(d\dot{Z}_F)$$

$$F_{b_R} = f(d\dot{Z}_R)$$

Wind and Drag Forces

The block subtracts the wind speeds from the vehicle velocity components to obtain a net relative airspeed. To calculate the drag force and moments acting on the vehicle, the block uses the net relative airspeed:

$$F_{d,x} = \frac{1}{2TR} C_d A_f P_{abs} (\dot{x} - w)^2$$



$$F_{d,z} = \frac{1}{2TR} C_l A_f P_{abs} (\dot{x} - w)^2$$

$$M_{d,y} = \frac{1}{2TR} C_{pm} A_f P_{abs} (\dot{x} - w)^2 (a + b)$$

Note: no one component of the wind force has been considered in the simulation..

The equations use these variables:

Fx	Longitudinal force on vehicle	[N]
Fz	Normal force on vehicle	[N]
My	Torque on vehicle about vehicle-fixed *y*-axis	[Nm]
FwF, *FwR*	Longitudinal force on front and rear axles along vehicle-fixed *x*-axis	[N]
Fd,x, *Fd,z*	Longitudinal and normal drag force on vehicle CG	[N]
Fsx,F, *Fsx,R*	Longitudinal suspension force on front and rear	[N]
Fsz,F, *Fsz,R*	Normal suspension force on front and rear axles	[N]
Fg,x,*Fg,z*	Longitudinal and normal gravitational force on vehicle along vehicle-fixed frame	[N]
Md,y	Torque due to drag on vehicle about vehicle-	[Nm]
a,*b*	Distance of front and rear axles, respectively, from the position of vehicle CG	[m]
h	Height of vehicle CG above the axle plane along vehicle-fixed *z*-axis	[m]
FsF, *FsR*	Front and rear axle suspension force along vehicle-fixed *z*-axis	[N]

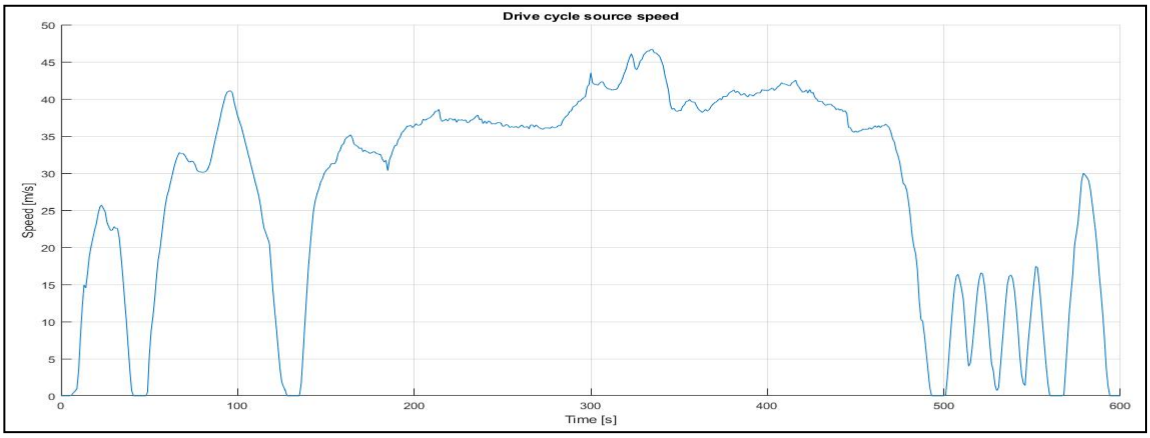
ZwF, *ZwR*	Front and rear vehicle normal position along earth-fixed *z*-axis	[m]
Θ	Vehicle pitch angle about vehicle-fixed *y*-axis	[rad]
m	Vehicle body mass	[kg]
NF, *NR*	Number of front and rear wheels	_
Iyy	Vehicle body moment of inertia about the vehicle-fixed *y*-axis	[Kg*m^2]
$x, \dot{x}, \ddot{x}$	Vehicle longitudinal position, velocity, and acceleration along vehicle-fixed *x*-axis	[m,m/s,m/s^2]
$z, \dot{z}, \ddot{z}$	Vehicle normal position, velocity, and acceleration along vehicle-fixed *z*-axis	[m,m/s,m/s^2]
FkF, *FkR*	Front and rear wheel suspension stiffness force along vehicle-fixed *z*-axis	[N]
FbF, *FbR*	Front and rear wheel suspension damping force along vehicle-fixed *z*-axis	[N]
dZF, *dZR*	Front and rear axle suspension deflection along vehicle-fixed *z*-axis	[m]
$d\dot{Z}F, d\dot{Z}R$	Front and rear axle suspension deflection rate along vehicle-fixed *z*-axis	[m/s]
Cd	Frontal air drag coefficient acting along vehicle-	_
Cl	Lateral air drag coefficient acting along vehicle-	_
Cpm	Air drag pitch moment acting about vehicle-fixed *y*-axis	_
Af	Frontal area	[m^2]
Pabs	Environmental absolute pressure	[Pa]
R	Atmospheric specific gas constant	_
T	Environmental air temperature	[K]
wx	Wind speed along vehicle-fixed *x*-axis	[m/s]

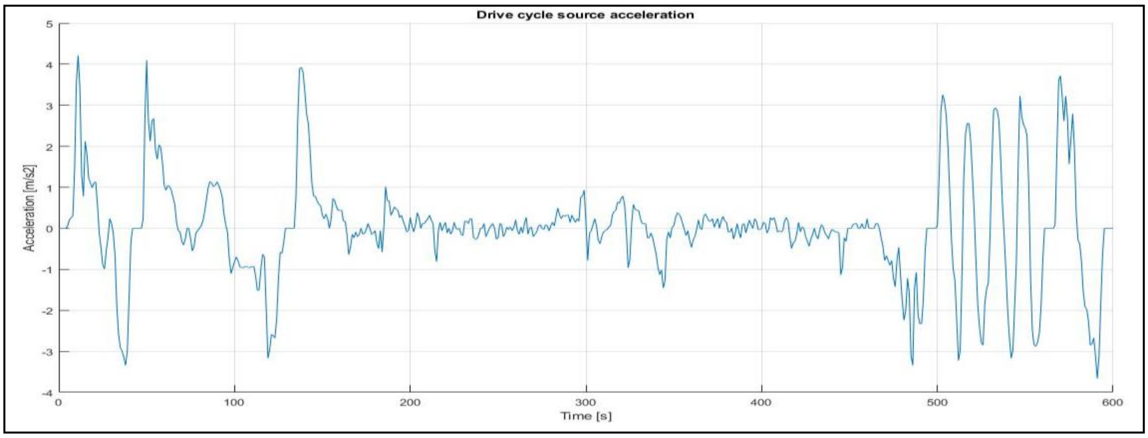
Chapter 3

Model validation and introduction to the simulation environment

In this chapter, the Panda 1108cc 4x4 model is validated, highlighting the speed, power and other parameters of the vehicle, verifying that they are consistent with those expected. Moreover, to give evidence of the potentialities of the instrument used, simulations with different input parameters are proposed.

For the cited purpose, a guide cycle (normally used for homologation) is selected to highlight the vehicle's performance limits. The **US 06 Guide Cycle** is chosen, multiplied by a factor of 1.3:





As shown above, the cycle so imposed has these characteristics:

- Peak speed of 170 km/h.
- Peak acceleration of 4.2 m/s^2.
- Duration of 600 seconds.

3.1 Simulation on Dry Asphalt

Firstly, we begin with a simulation on dry asphalt, in order to allow the car the maximum grip when it transfers to the ground all the available torque, consequently, according to the equations of Pacejka implemented in the model, the parameters of adhesion will be:

Surface	B	C	D	E
Dry Asphalt	10	1.9	1	0.97

Some of the data provided by the simulation are listed and useful for the validation of the model:

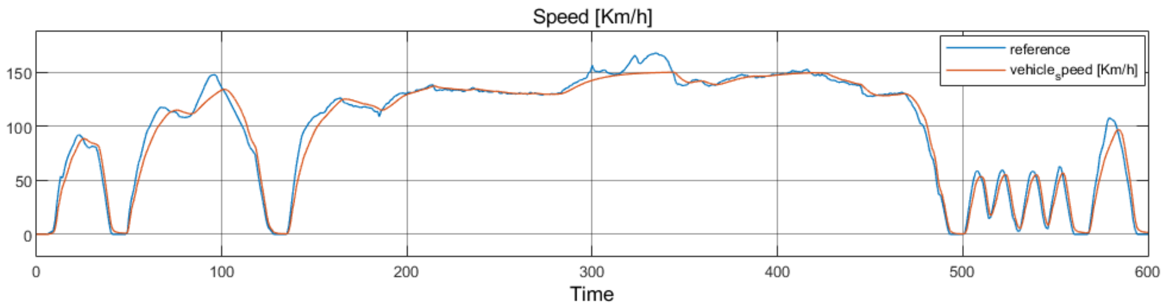


Figure 3.1: 4x4 vehicle model speed following the US 06*1.3 profile.

The maximum speed reached by the vehicle is 150 km/h and, considering that the mechanical inefficiencies of the transmission have not been taken into account, the data is reliable.

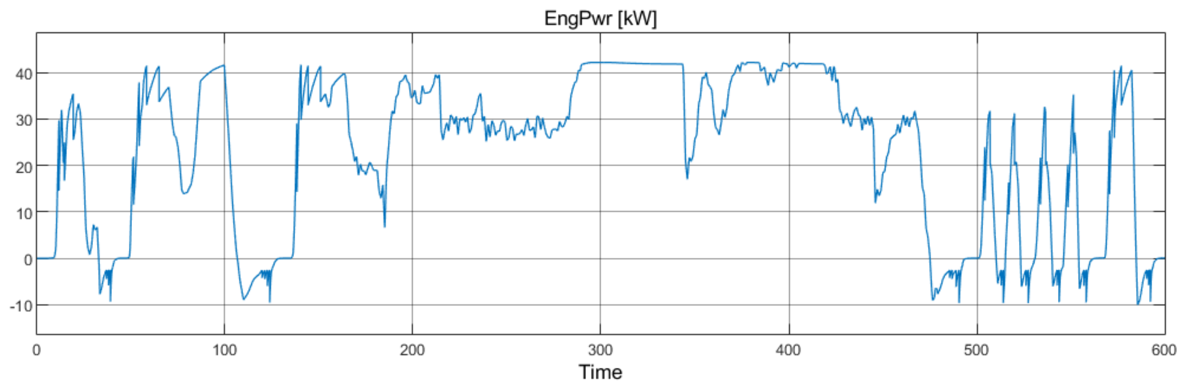


Figure 3.2: Engine Power supplied by 4x4 vehicle.

The maximum power supplied by the engine is just over 40 kWatt (54 hp) as expected.

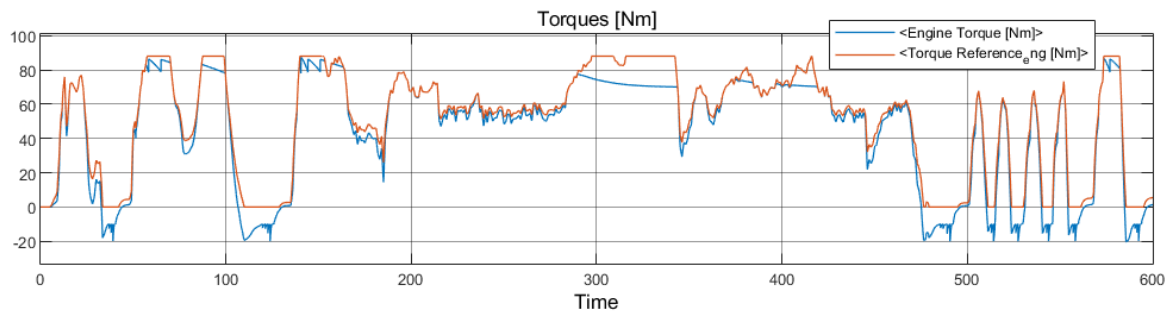


Figure 3.3: Difference between Engine torque requested, and torque supplied.

The maximum torque supplied by the motor is 88 Nm as expected, it is also possible to notice a clear difference between the torque required and the output, since the dynamics of the thermal motor are simulated too.

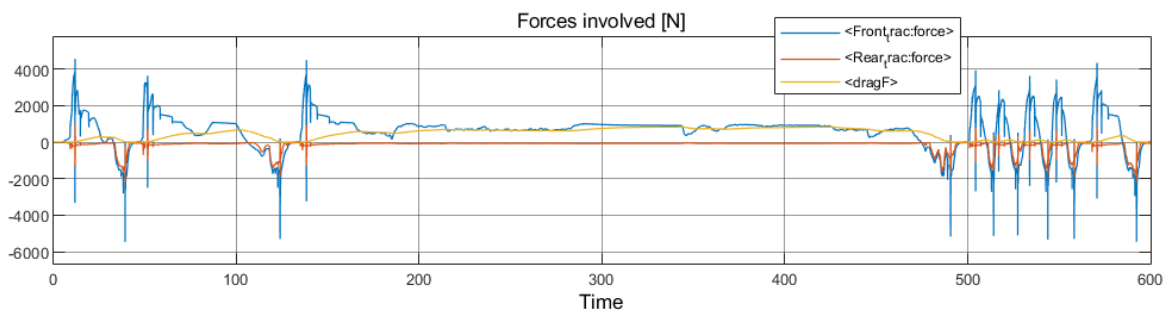


Figure 3.4: Forces acting on the vehicle.

The only axle in traction is the front one, while the force peak of approximately 4500N from it reached is justified as follows:

57Nm (motor torque at peak times)

* 3,909 (1st gear ratio)

* 5,455 (diff. Ratio)/0.27 m (wheel radius)

= 4.501 Nm

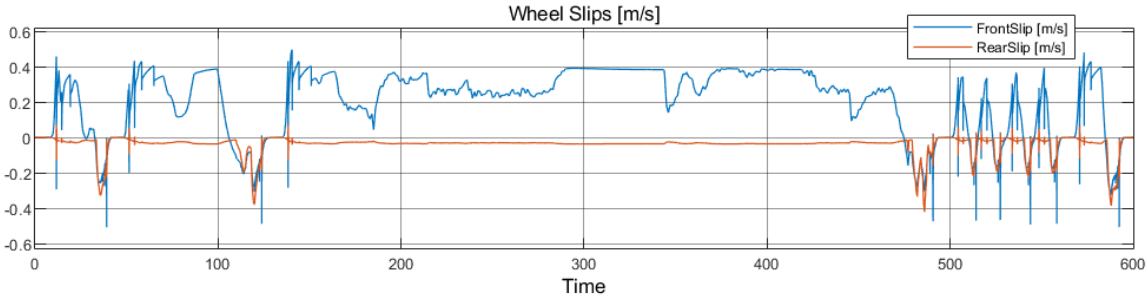


Figure 3.5: Wheels slip values on front and rear axles.

Since the front wheels are the only driving wheels (in this simulation) the single value of positive slip, useful to the grip of the tyre in traction, is provided by the front axle, instead the rear wheels involve a negative slip because they are carried.

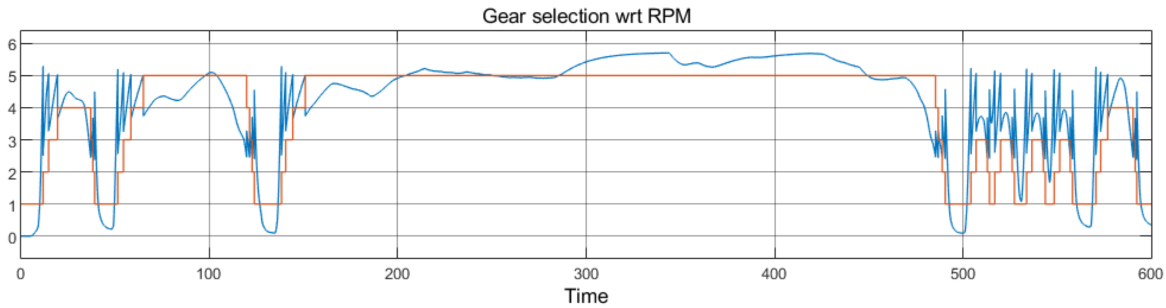


Figure 3.6: Selection of gears according to the engine speed.

Finally, there is a graph with the gears changing according to the revolutions of the engine in which we obtain a selection of the upper and lower gear respectively for a number of revolutions greater than 5000rpm and less than 3000rpm as programmed.

3.2 Simulation on Irregular Ground

This simulation is still made on dry asphalt but with the addition of a non-zero inclination profile during the path, in order to model a series of periodic bumps of 30 ° angle that the vehicle must face:

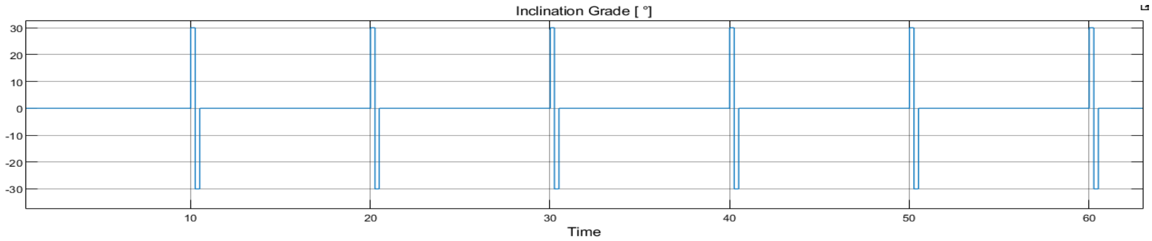


Figure 3.7: Example of the inclination profile.

The obtained changes are almost instantaneous, so we'll get a reference profile similar to the starting one with the addition of periodic bumps every 10 seconds.

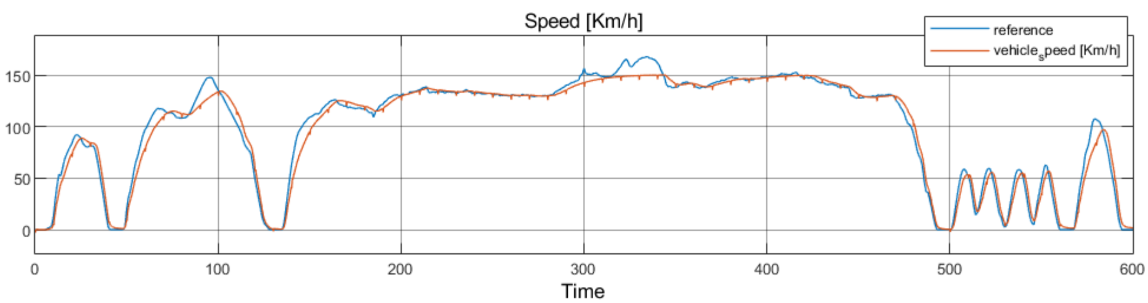


Figure 3.8: 4x4 vehicle model speed facing the new track obtained.

The performance of the vehicle speed shows the presence of obstacles. The average speed, as well as the other data, is rather unchanged than the previous test.

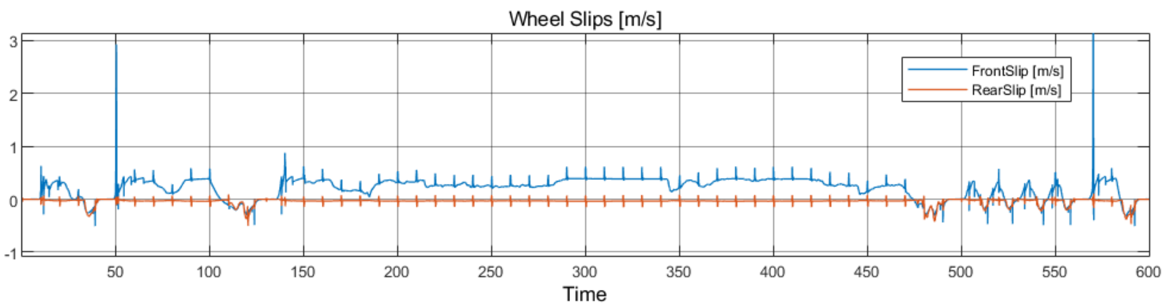


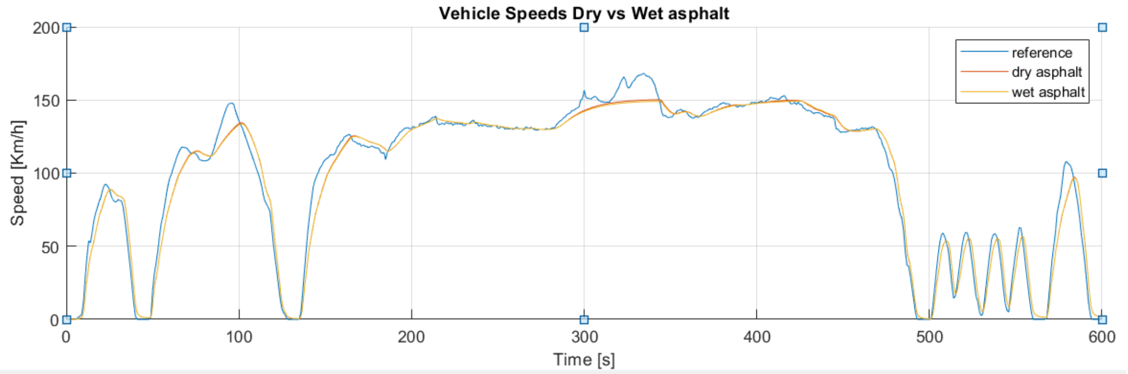
Figure 3.9: Wheels slip values obtained with bumbs.

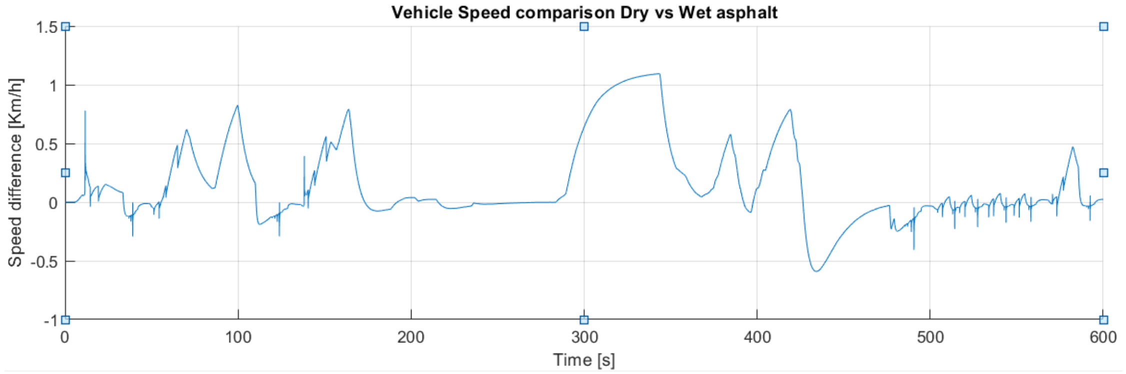
The most interesting result is shown by the tyres slip values, there are peaks in the presence of bumps also for the carried wheels, but the higher values are still reached by the wheels in traction.

3.3 Simulation on Lower Adhesion soil

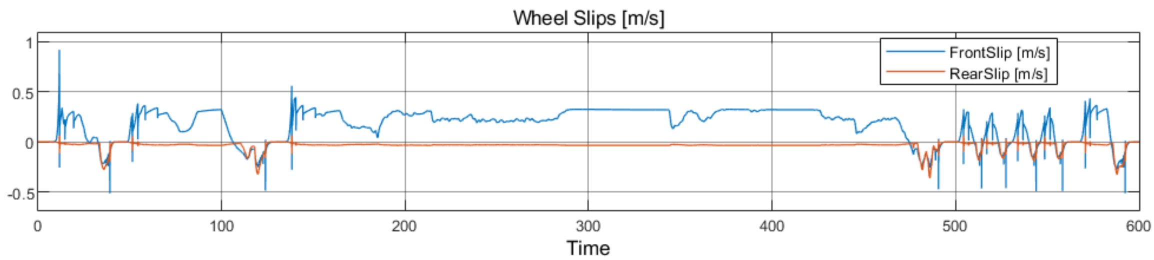
The following simulation is carried out on a flat surface but with a lower adhesion, for this reason we choose the coefficients of adhesion that reproduce the case of wet asphalt.

Surface	B	C	D	E
Wet Asphalt	12	2.3	0.82	1





The graph above shows the loss of vehicle speed due to the lower adhesion of the ground. However, the differences are not so consistent because the decrease of the adhesion itself is small too, and it is still enough to transfer all the needed torque on the ground.



It can be seen a considerable increasing of tire slipping compared to the simulation made on dry asphalt.

Chapter 4

4x4 Traction Activation, simulation and considerations

In this chapter we choose to insert 4x4 traction on the Panda 1108cc vehicle in order to highlight differences and advantages compared to the front wheel drive mode.

4.1 Key aspects of the 4x4 mode

To give a first look to the 4x4 features, in the following simulation a moderate speed profile is used (US 06 * 0.6) but to be followed on a low-adhesion ground, such as an icy one.

The Pacejka parameters selected will be:

Surface	B	C	D	E
Ice	4	2	0.1	1

Figure 4.1: Vehicle speed with and without 4x4 traction, iced surface.

These results already show a clear difference between the two models performing the profile imposed.

The vehicle with 4x4 traction activated is totally capable to follow the imposed profile, on the contrary the vehicle with front wheel drive easily reaches the limit of the possible traction conditions on the ground thus simulated.

It is also possible to note that for the front wheel drive vehicle the limited acceleration is the only element that prevents the correct execution of the test, since both vehicles are able to achieve the maximum speed imposed, while during the braking they have the same performance as the braking is still distributed on all the wheels, unlike traction.

Below are shown the differences in tyre slipping in the two cases considered:

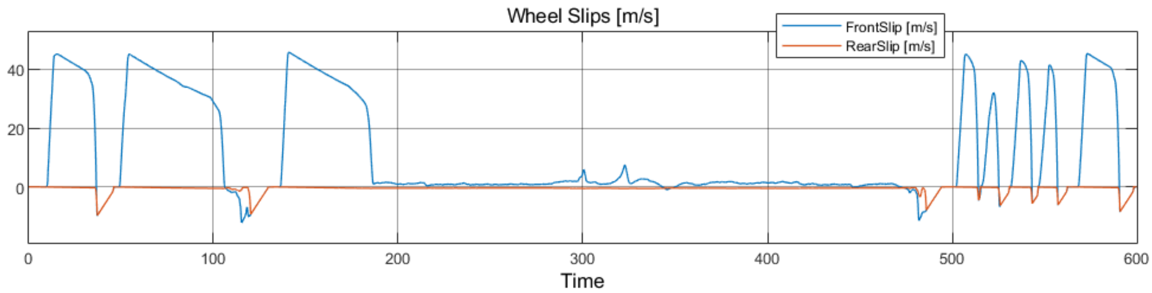


Figure 4.2: Wheels slip values without 4x4 traction, iced surface.

With **NO 4x4 traction,** the vehicle works in a high-slip condition, even at low speeds, but only on the front axle, the only one in traction (of course these are extreme working conditions exclusively used to compare the two models).

Figure 4.3: Wheels slip values with 4x4 traction, iced surface.

With **4x4 traction**, the wheels have the same degree of slippage, because they are kinetically connected, resulting in a more homogeneous and distributed traction of the vehicle on both axes and also of the tire slip values clearly lower.

4.2 4x4, 4WD, AWD, a closer look

Often, these terms are used interchangeably, but while it seems like they do the same thing, they are in fact very different, and the differences affect their functionality.

4WD

4WD is generally accepted as a car or more typically a larger SUV (Sports Utility Vehicle) that uses a driver selectable system that mechanically engages the drive to all four wheels.

Driveline dynamic behavior

4WD system allows to send one quarter of the engine torque to each wheel constantly (in ideal conditions) and needs three differential gearboxes, two for the axes and a central one.

Through the differentials the engine torque is split equally but, in this configuration, the wheel with smaller grip will impose the torque value to be transmitted. This means that a slipping wheel will let the other receive a null torque, so the possibility to lock the differentials is introduced.

Locked 4WD driveline (4x4 mode)

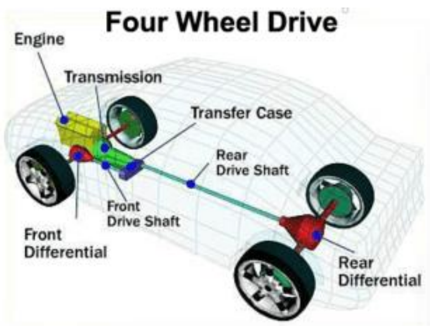


Figure 4.4: Locked 4WD driveline scheme.

(the configuration proposed, figure 4.4, shows a driveline provided with a “transfer case”, this item provides the same kind of transmission given by a locked differential, adding the possibility to reduce the transmission ratio in the driveline → lower gears)

This system sends power to all four wheels equally and without vectoring (controlling the division of power delivery between the wheels or axles), meaning each wheel will spin at the same constant rate as all the others.

In this configuration the entire engine torque can be transferred on a single wheel if the other three are slipping on the ground.

This means that there is direct mechanical link between front and rear axles with no mechanism to allow any difference in the number rotations of the front and rear axles. Thus, when the 4x4 vehicle turns a corner, because the steering radius is different for front and rear axles, the tires on the axle with the smaller radius of turn must be able to slip on a loose slippery ground surface. If the ground surface is not slippery and the tires do not slip too, then the driveline (axles and propeller shaft etc.) will twist and stress will be induced. This is known as 'wind up' and ultimately if the twist cannot be dissipated the vehicle will no longer be able to move as it becomes 'locked up'. This will generally only happen at lower speeds on ground surfaces with no slip. At higher speeds or on slippery road surfaces, the tire is able to slip and the 'wind up' is released.

In conclusion, this means that when 4WD vehicles are driven on normal road surfaces, 4x4 mode must be deselected.

AWD

Vehicles that are designed for normal road, with occasional dirt or mild off road, generally use permanently engaged AWD systems. These vehicles drive all of the wheels all of the time, so the system must include a mechanism that is generally a limited slip differential or an electronically controlled clutch to allow a rotational difference between front and rear axles.

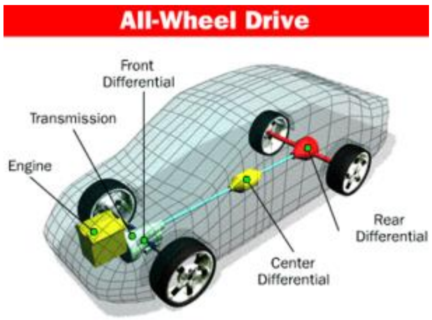


Figure 4.5: AWD driveline scheme.

Driveline dynamic behavior

AWD is all about varying the amount of power to each wheel, either by physically with differentials or electronically by brake vectoring (where the brakes are used to slow down a specific wheel due to traction loss), so it sends power away from the slipping wheels when it detects a loss of traction.

Pros

This has the active safety advantage of always having twice the grip of a driver selectable 4WD system. This means that in the unexpected situation where the corner is more slippery than expected, or when immediate traction is required to move safely into merging traffic, All-Wheel Drive is already engaged, and the required level of traction is available to safely negotiate the situation.

In conclusion, the AW-Drive is the one suggested for permanent use.

The following table resumes the characteristics described (figure 4.6).

4WD with locked differentials (→ 4x4)	AWD
Driver decides when to use it.	Works all the time.
Best traction in off-road conditions.	Increased grip and control under all road conditions.
Can't be used in all conditions.	Not as good in extreme off-road conditions.

Figure 4.6: Pros and cons between 4WD and AWD traction systems.

Panda 4x4 1108cc

From a mechanical point of view, the Panda traction system normally has to be treated as a common front-wheels drive transmission, otherwise it corresponds to a locked differential transmission when the 4x4 is activated with the proper lever by the driver.

4.3 Distribution of the Torque, Panda 4x4 1108cc case

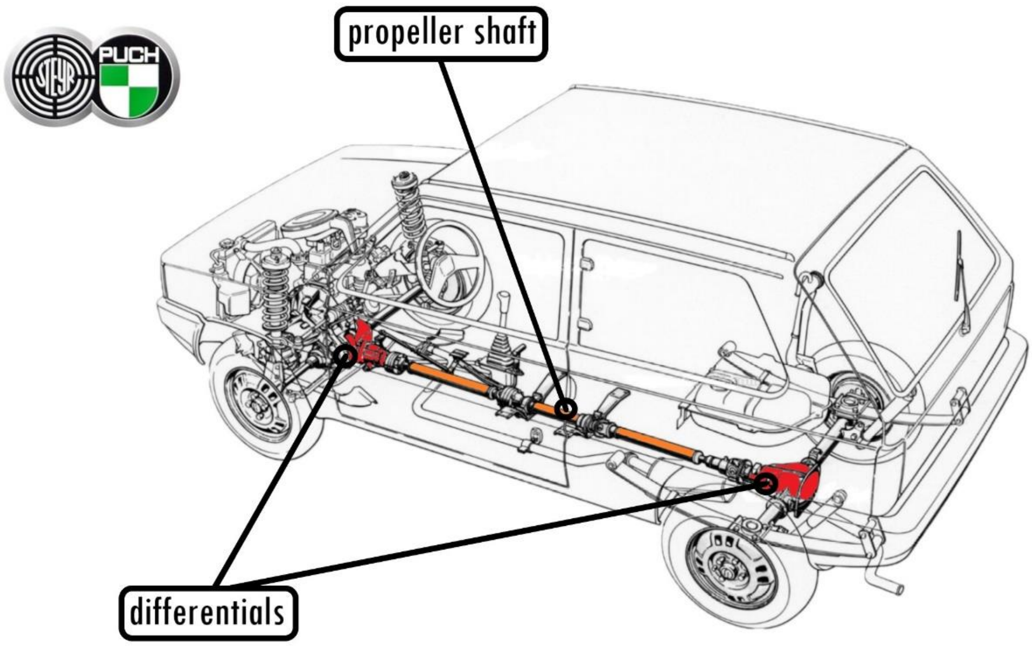


Figure 4.7: Panda 1108cc 4x4 traction scheme with focus on the transmission shaft.

The peculiarities of the transmission employed on this vehicle have already been faced in the preceding paragraphs from the mechanic point of view. Specifically, the distribution of the torque on the vehicle Panda 1108cc 4x4 is structured in the following way:

When the 4x4 traction is inserted, the front and rear axles are kinetically connected, i.e. the velocities of the respective differentials will be the same.

The only element interposed between the differentials is the transmission shaft (figure 4.7), divided into three sections, which is responsible for the transferring of the engine torque to the rear axle and for the handling of any differences in adhesion between the axles.

The reasons why the two axes can have different conditions of adhesion depend on the dynamics of the ground-pneumatic contact (paragraph Wheels 2.6.3) and they are:

- Different load on the axle (resulting from the position of the vehicle's center of gravity, ground inclination, acceleration or deceleration)
- Different coefficient of adhesion, if the soil is uneven and irregular (maybe due to the presence of a mug puddle, breach on the track, etc.)

So, in several situations it can be that one of the two axes obstructs the rotation of the other one. In these cases, since it is not possible to have different speed between the differentials, the torque supplied to the axle with greater resistance in motion will be consequently greater than one half delivered by the engine. A borderline case is one in which an axis is totally free to rotate, this situation would see the entire torque discharged on the reciprocal axis.

For a more explicative analysis we proceed with simulations under different conditions, focusing on the distribution of torque between the axes.

Dry Asphalt

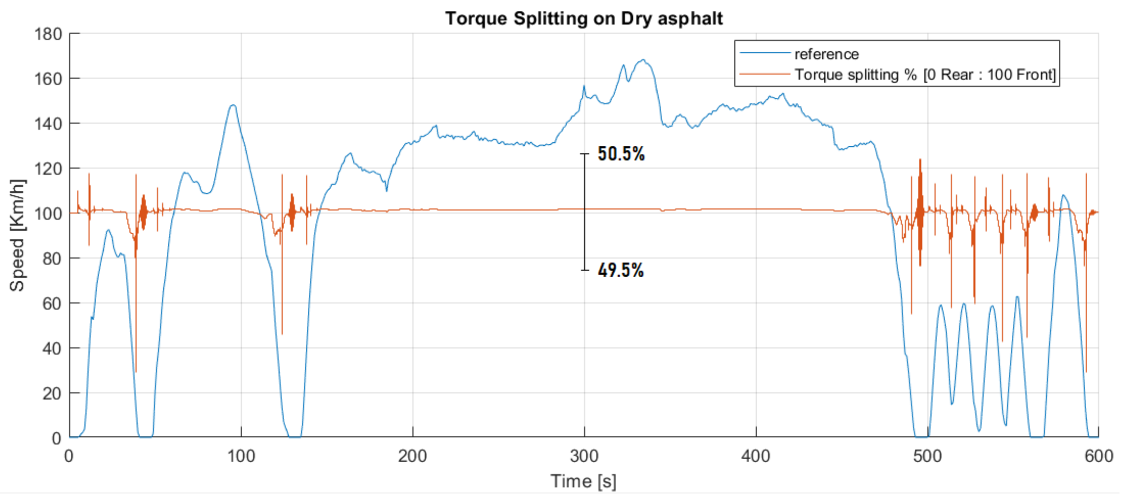


Figure 4.8: 4x4 vehicle, torque splitting through the axles, dry asphalt.

Wanting to analyze the distribution of the torque on the axes in detail, it is reported in the previous graph a simulation with speed profile US 06 (* 1.3) on dry asphalt, linked with the percentage of distribution between the front axle and rear.

It can be easily inferred that in conditions of perfect adhesion, like the one analyzed, the distribution of the torque between the axes is to be considered constantly of kind **50:50** even in presence of strong accelerations.

In cases such this, therefore, the difference in weight between front and rear axle and the pitch of the vehicle have a negligible influence on the distribution of the torque, since the model always works in conditions of total adherence compared to the torque available on the axes and there is no possibility of slipping for the tires which could lead to a torque difference on the propeller shaft.

Wet Asphalt

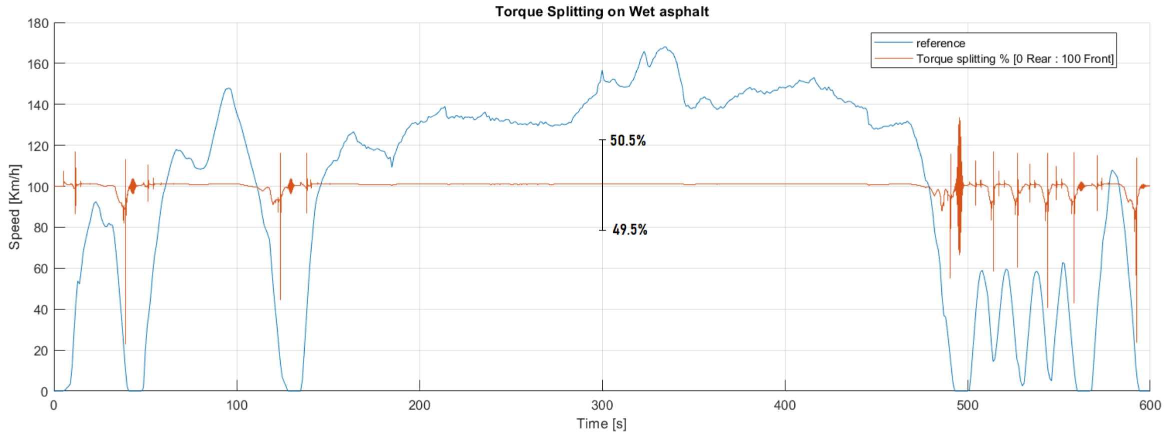


Figure 4.9: 4x4 vehicle, torque splitting through the axles, wet asphalt.

In the wet asphalt test there is no particular difference, in fact both in these conditions and in the previous one the vehicle maintains a permanent torque division between the axes of 50:50.

Iced Surface

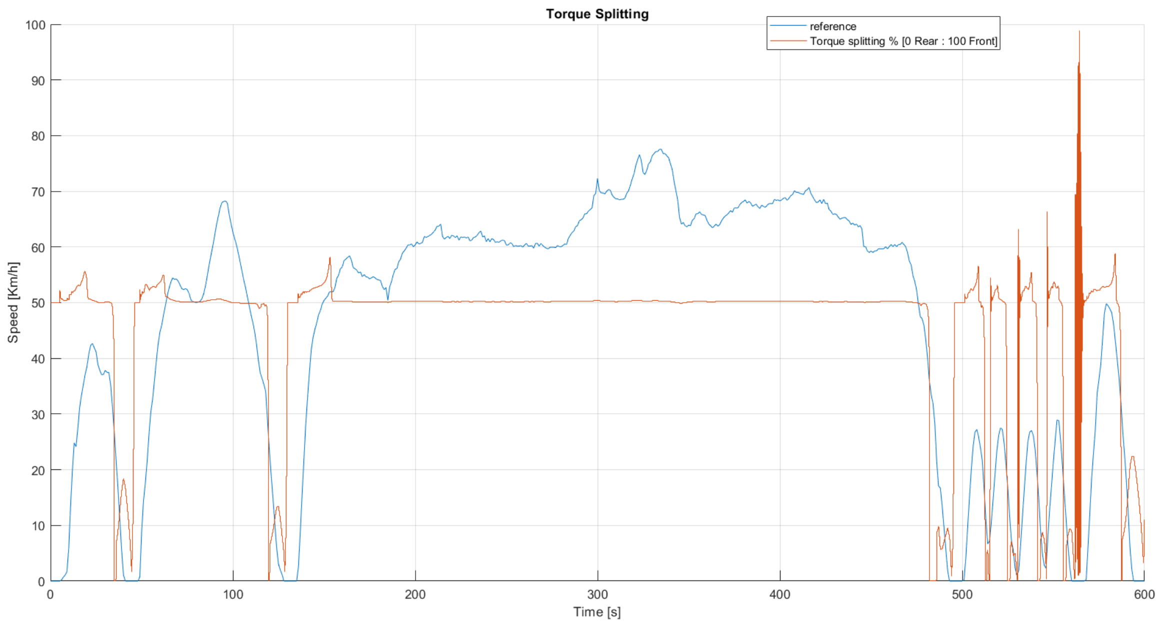


Figure 4.10: 4x4 vehicle, torque splitting through the axles, iced surface.

In low-adhesion simulations, like the one on icy ground, the difference in torque discharged on the axes is evident.

At **constant speed**, except for a few oscillations, each axle is moved by about 50% of the engine torque.

In **acceleration,** the torque supplied to the front axle easily exceeds a value of **60%** because the center of gravity of the car is closer to the front chassis and the lower vertical load on the rear axle causes a consistent slippage when more torque is supplied in these conditions.

When **braking**, again for the weight difference, the rear wheels are the first to reach the complete clamping (the pressure of the braking system is too high for this kind of surface) and this implicates a higher resistant torque ("Motor brake") discharged on the rear axle.

Chapter 5

Hybrid conversion Design Proposals & Torque Split Strategies

This chapter illustrates the several solutions employed on the hybrid vehicles, explaining the selection criteria related to the final technologies chosen for the car that has been object of this work, with advantages and constraints associated with them.

5.1 Mild Hybrid Electric Vehicle Architectures (MHEV)

The positioning of the main components of a hybrid electric system on the vehicle can be understand and classified by architecture, topology or configuration. Since the only mechanical link between the electrical system and the rest of the vehicle is done through the electric machine, the MHEV architecture is basically defined by the **position of the electric machine** and the type of connection with the powertrain / drivetrain (belt, integrated or gear mesh).

The electric machine can be placed, concerning the other powertrain components, in five major points:

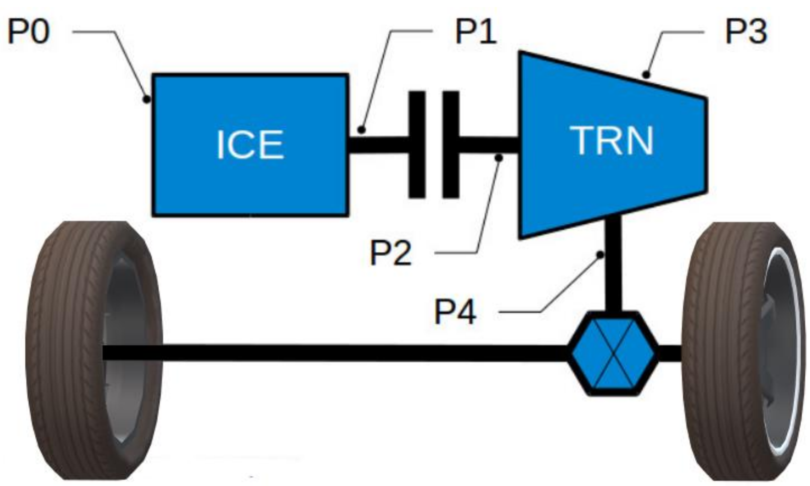


Figure 5.1: Powertrain elements scheme.

Looking into the positioning of the components on the vehicle, a brief description of the electric machine connection points is done in the table below.

P0	The electric machine is connected with the internal combustion engine (ICE) through a **belt**, on the front-end accessory drive (FEAD)
P1	The electric machine is connected **directly** with the crankshaft of the internal combustion engine
P2	The electric machine is side-attached or **integrated** between the internal combustion engine and the transmission (TRN); the electric machine is decoupled from the ICE and it has the same speed of the ICE (or multiple of it)
P3	The electric machine is connected through a **gear mesh** with the transmission; the electric machine is decoupled from the ICE and its speed is a multiple of the wheel speed
P4	The electric machine is connected through a **gear mesh** on the rear axle of the vehicle; the electric machine is decoupled form the ICE and it's located in the rear axle drive or in the wheel's hub

The final solution adopted on the hybrid vehicle designed is the Belt Starter Generator Architecture, so it is the single one analyzed in this thesis work.

5.1.1 Belt Starter Generator Architecture (P0)

Also known as BiSG from belt integrated starter generator, this mild hybrid topology is the most cost effective due to the limited impact on the existing vehicle architecture.

On a hybrid electric vehicle application, there are two major cost drivers: the impact on the existing powertrain components and the high voltage battery. To minimize the integration costs, the vehicle and transmission architecture should be kept the same as for a conventional vehicle. Thus, the easiest way of achieving a minimum cost is to integrate the electric machine into the already existing engine accessories belt drive, by replacing the 12V alternator (generator).

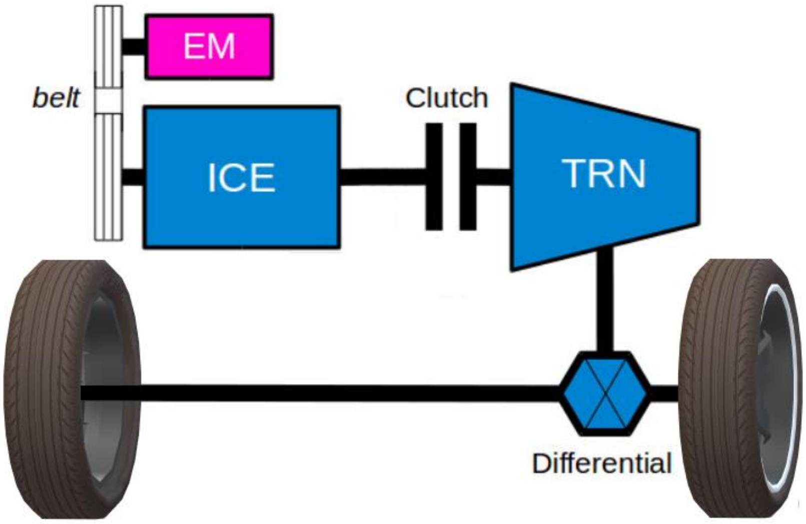


Figure 5.2: P0 architecture scheme.

In the BiSG architecture, the internal combustion engine (ICE) and the electric machine cannot be separated, they are mechanically linked through the accessory belt.

Therefore, one of the disadvantages of this configuration is that, the engine friction torque will be a **parasitic loss** for the electric machine when it gives boost torque and when it's recuperating electrical energy.

The main characteristics with the pros and cons of this design are:

Electric machine performance	Maximum torque (at crankshaft): **up to 50 Nm** (with belt pulley ratio multiplication, e.g. 2.8) Maximum power: **12 … 14 kW** Continuous power: **2.5 … 3.5 kW** Efficiency: **up to 85%**
Hybrid modes (functions)	Idle Stop & Start Moving Stop & Start Engine load shift Torque assist (fill) Energy recuperation Brake regeneration
Advantages	Low cost of integration Air or liquid cooled electric machine Integrated inverter (with electric machine) Size modularity for the electric machine

	Speed / torque ratio possible between electric machine and ICE results in lower Power demand from the electric machine
Disadvantages	Limited torque capacity due to belt drive Energy recuperation affected by engine friction losses
Overall characteristics	Torque Boosting Capability: **Medium** (limited by belt slip, durability) Electrical Energy Recuperation: **Medium** (due to engine losses) Driveability Improvements: **Medium** (due to limited torque boost) Electrical Creep / Drive: **Not possible** (due to limited torque and belt drive) Packaging: **Easy components integration** with limited impact on other components System Efficiency: **Medium** (mainly due to belt-drive integration on the FEAD)

Repercussions on the Transmission Belt

The BiSG MHEV architecture has a significant impact on the design of the **Front End Accessory Drive (FEAD)**. The belt durability needs to be increased to sustain higher torque and more engine off/on cycles. The variable belt tensioners have to provide:

- increase tension during cranking and boost (torque from electric machine to engine)
- increase tension during recuperation (torque from engine to electric machine)
- reduce tension during normal driving (in order to reduce friction losses)

There is also a significant impact on the noise, vibrations and harshness (NVH) of the engine and on the durability of the main bearing of the engine's crankshaft.

Furthermore, the majority of BiSG MHEV applications adopts a 48Volt electric system, still using the 12V starter. The reason is that the cold engine start, especially after a long period of inactivity, demands a high electric machine torque (due to high engine friction). This is a limitation on the BiSG because the amount of torque which can be transmitted is limited by the belt slip. With an improved design of the FEAD belt and increased durability, the 12V starter can be removed and all its functions performed by the electric machine.

5.2 Panda 1gen Hybrid Design

As mentioned before, given the particular kind of competition the car will face, the new hybrid vehicle has to provide an increment of the efficiency, due to the huge distances, and an increment of the total power because of the rough terrain.

The first requirement is satisfied with the adding of the generator but for the second one the use of a bigger electric motor it's mandatory.

Thus, the final result of this design includes a kind of technology applied in the recent field of MHEV cars coupled to a classic electric motor on the rear axle, classifying the car architecture in the category of Full-Hybrid cars (figure 5.3).

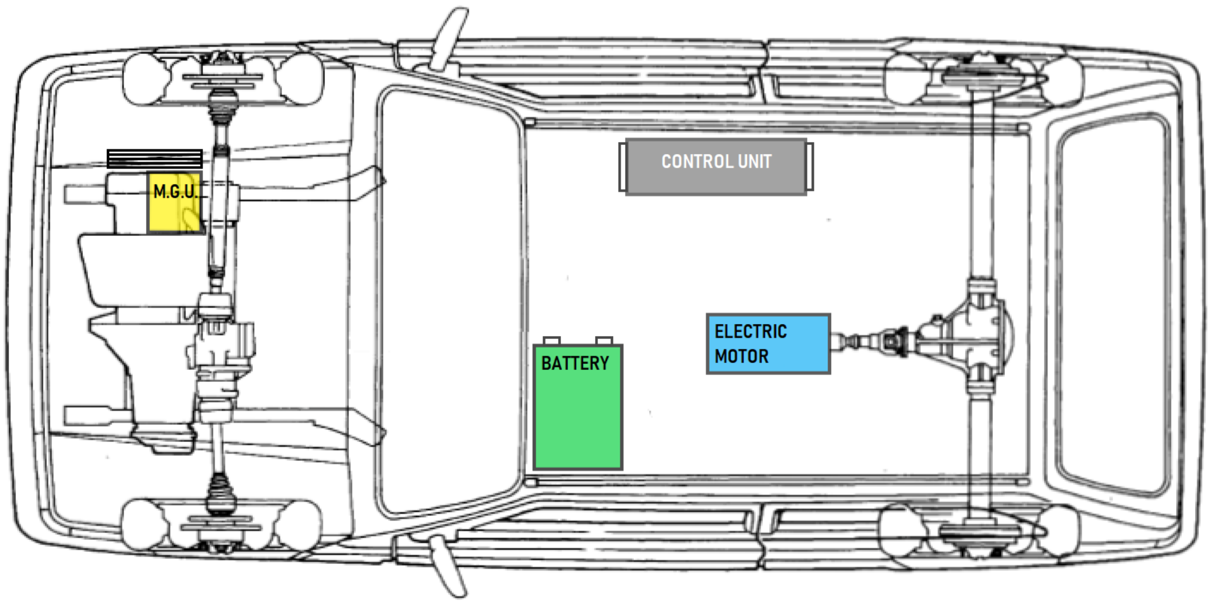


Figure 5.3: Panda 1108cc Final Hybrid Architecture.

Each axle gets its own motor, so the four wheels are always powered but there's no mechanical connection between the front and the back of the car. This improves traction and performance while lighting the vehicle because there's no more need of propeller shaft.

Looking at the requirements of this new vehicle, designed for a competition, many of the possible functions available with the obtained architecture will be not implemented because useless for the purpose (like the idle start & stop).

The electric machine on the front axle will work as a generator, during vehicle deceleration and regenerative mode, and as a motor (to assist the engine) during vehicle accelerations. Due to the employing of a 96 Volt electric system, the alternator will be removed, and all its functions performed by the electric machine.

The electric motor on the rear axle will be designed to work constantly with the thermal motor to provide a four-wheel drive traction.

The battery dimension chosen will be of small capacity in order to keep the vehicle lighter.

The manual transmission will be kept, and it will be necessary the installation of several sensors on the vehicle, like the ones which scan the gear used by the driver in each moment (this thesis work does not include this kind of study).

Summarizing the improvement to the car architecture mentioned before, the vehicle will have the following components:

1. Belt-Drive Starter-**Generator** with a nominal power of **7kW** [max 10kW] and **reduction ratio of 2** in the coupling with the engine.
2. **Electric motor** with a nominal power of **15kW** [max 25kW] and **reduction ratio of 10.5** through the rear axle.
3. High voltage battery [96V].
4. DC/DC converter, because of the low and high voltage batteries.

5.2.1 Modeling of the electric components

Given the above, the implementation of the hybrid model previously discussed requires the adding of a proper electric plant which includes the battery, the electric motor and the generator (figure 5.4).

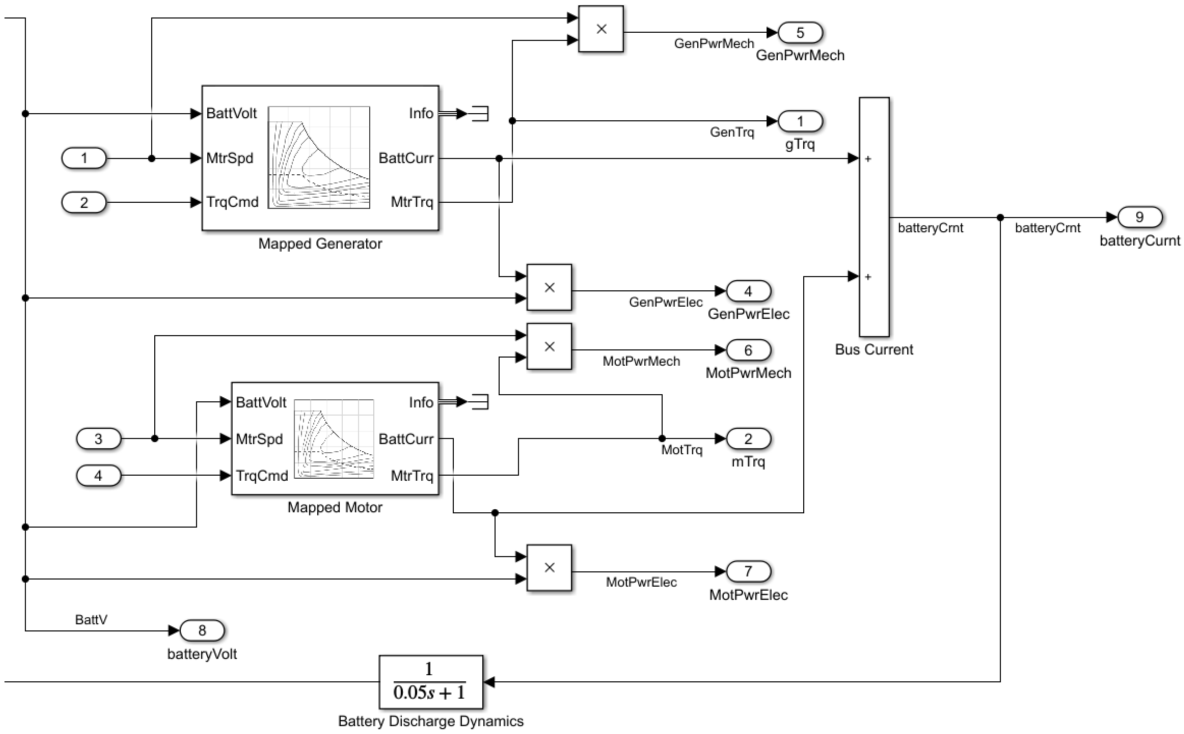


Figure 5.4: Electric Plant Model

The electric plant designed reproduces a 96 Volt system, constant (ideal condition), equipped with a 2kWh battery with an initial state of charge of 60%. Both the electrical machines are designed through the default "Mapped Motor" blocks given in the Simulink Environment.

These blocks Implement a mapped motor and its drive electronics too, operating in torque-control mode, specifying the electrical torque range with the maximum motor power and torque values needed. The user can also specify electrical losses as a single operating point that estimates loss across the operating range, measured loss, or measured efficiency.

The electrical plant designed in this way, gives the user all the parameters of each single element, regarding the current and voltage levels of the system, the state of charge of the battery, the mechanical and electrical powers exchanged.

As for the drivetrain, the only inputs of this system are the commands of torque requested to the generator and the motor, coming from the controller, and the electrical machines rotational speed obtained as feedback from the model.

5.2.2 Controller and implementation of the control functions

Regarding the hybrid vehicle model, the main section of the system becomes the one of the controllers. Indeed, without the adoption of the proper control logics of the three motors employed, the efficiency of the final configuration of the vehicle could be lower than the efficiency of the starting one, making the entire work of design pointless.

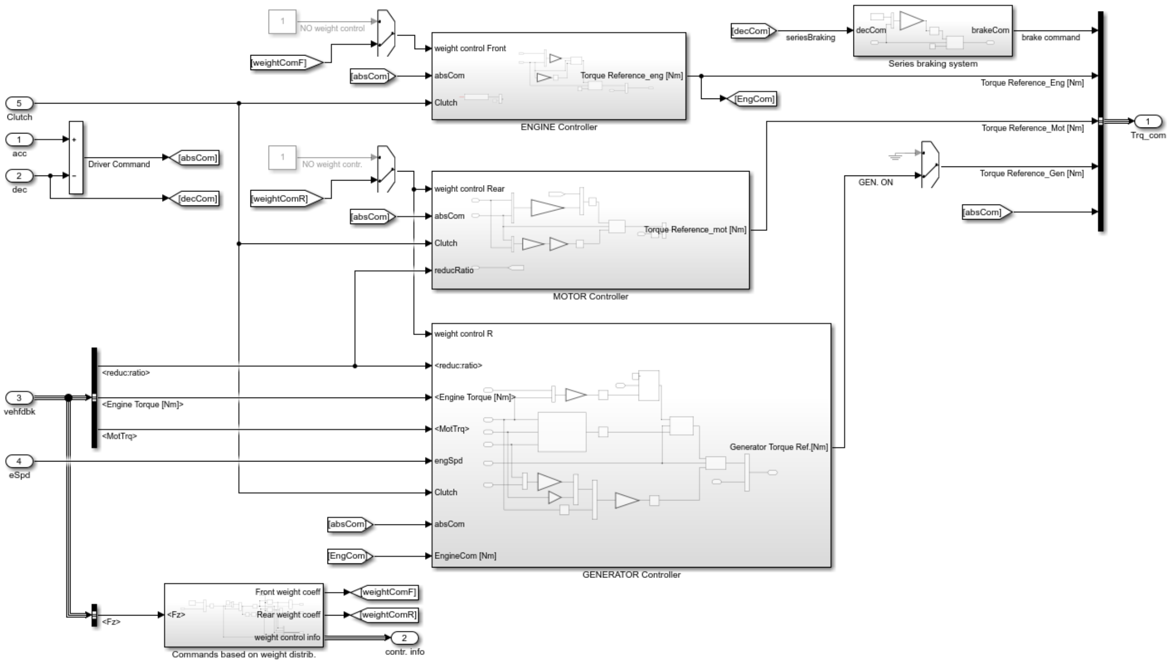


Figure 5.5: Hybrid vehicle controller logic designed.

The controller interface (figure 5.5) allows to activate or not the generator, according to the kind of simulation needed, and it makes also possible to select the activation of a particular control system for the vehicle traction, acting on the engine and the electric motor (analyzed in the following paragraphs).

From the scheme above, it can be observed that the 3 control blocks of the motors are quite distinct, but this doesn't mean they work separately. Indeed, each motor requires several system information for its functioning, which from the point of view of the model are simple data, but on the real prototype they will correspond to real measures acquired by the sensors installed on the vehicle.

At this stage, it's required the introduction of 2 important control parameters:

1. **Acceleration Threshold**
2. **Braking Coefficient**

Both the parameters refer to the driver's command, respectively of acceleration and deceleration. Looking closer, if the acceleration command will be lower than the Acceleration Threshold, the vehicle will work in regenerative mode, also called Engine Load Shift (look at the next paragraphs). While if the braking command will be lower than the braking coefficient, then the car will receive a braking force only from the electrical machines, excluding the mechanical braking system (see the paragraph 6.2.4.1 "Brake Regeneration").

The choice of these parameters, fixed during the simulations, will be crucial to understand if the dimensioning of the electric elements added to the vehicle is suitable to face a competition on terrain and sand, avoiding the discharging of the battery.

In the next paragraphs are analyzed accurately the control logics of each motor, stressing mostly the theories on which the several functioning modes of the hybrid vehicle are based.

5.2.3 Generator

The generator is the first motor analyzed, because it is the most flexible element of the hybrid vehicle.

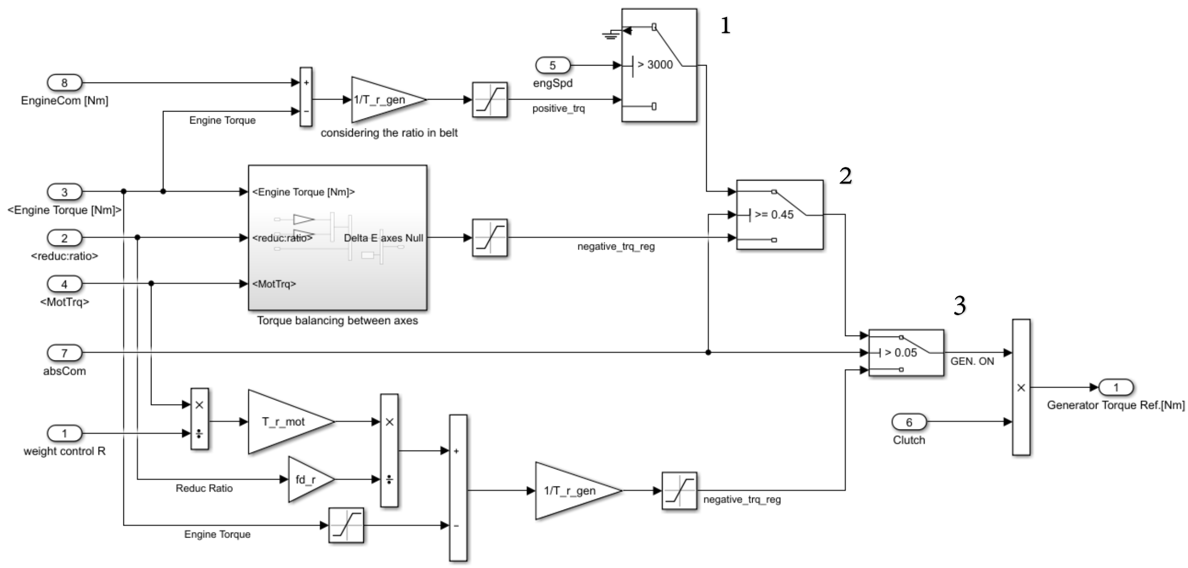


Figure 5.6: Generator Controller, internal logic designed.

First, the clutch signal was used to decouple the generator from the transmission when the driver is changing gear.

It is called “generator” due to its main purpose which is to recover energy applying a resistant torque on the thermal motor, both when it is required low traction torque (mechanism useful to keep the battery charged to use it when is more necessary) or simply during the regenerative braking.

Note: From the physical point of view, the generator is coupled with the engine considering the mechanical ratio between them previously chosen, modeling the belt architecture described before.

But, as it is well known, a generator is still an electrical machine, thus can be employed also as motor if necessary. This is the case of “Torque Fill” mode (see the next paragraph).

Before continuing the analysis, it’s necessary to clarify briefly the functioning modes of the generator, as shown in the previous scheme (figure 5.6).

It supplies:

1. **Positive Torque for Torque Fill**, when the engine speed is higher than 3000 RPM and the acceleration command is lower than the Acceleration Threshold.
2. **Resistant Torque for Regenerative Mode**, for acceleration command lower than the Acceleration Threshold, whatever the engine speed value is.
3. **Resistant Torque for Regenerative Braking**, for any value of the deceleration command.

5.2.3.1 Torque fill

The internal combustion engine generates torque through the crank mechanism (piston, piston pin, connecting rod and crankshaft). These components have mass and inertia (rotational and translational). Also, the air drawn into the engine has mass, therefore inertia. Because of these design constraints, the engine cannot deliver instantaneous torque. If the drive tips-in the accelerator pedal, it takes a while until the engine accelerates to the required operation point (torque and speed).

Electric machines, having only one moving part (rotor) and being governed by electromagnetics laws, can deliver instantaneous torque. A hybrid powertrain can benefit from the electric machines torque delivery in order to improve the overall dynamic performance of the vehicle.

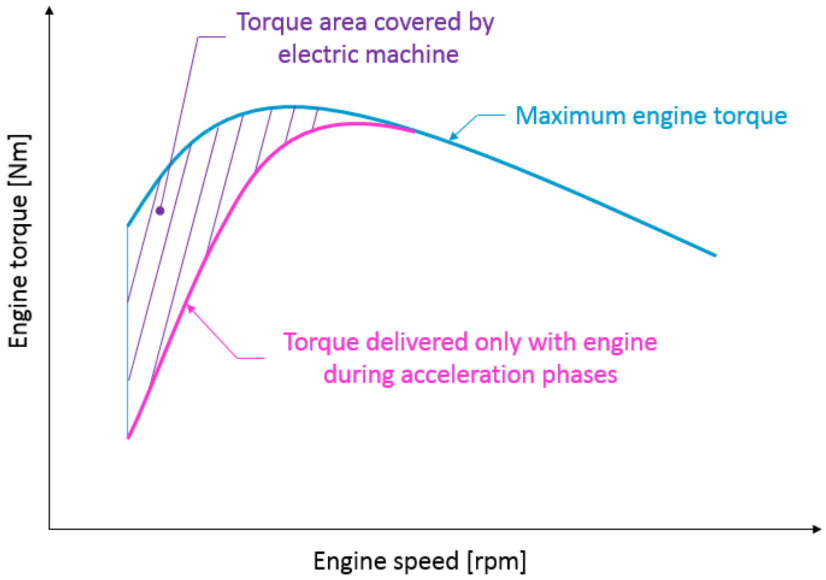


Figure 5.7: Example of Engine torque fill with electric machine.

Torque fill means to compensate (fill) the torque demand of the driver, which cannot be delivered by the engine, with the electric machine torque. Especially in the low speed range, an internal combustion engine has a significant torque lag (delay). If the engine is operating in this region and the driver demands high torque, the difference between what the engine can deliver and what the driver demands is compensated by the electric machine.

The torque fill function can be regarded as a torque assist function during the transient torque demand phases.

Application on the 1108 cc Engine

In the case of the hybrid vehicle designed, the engine can supply a torque up to 88 [Nm], but this value is reached only at the engine speed peak value of 3000 [RPM]. Thus, the torque fill logic considers that the generator has to provide torque on the transmission belt until the engine reaches this torque peak.

Furthermore, the generator command torque is derived from the difference between the engine actual torque delivered and the engine torque request, considering also the coupling ratio between the two motors through the belt (figure 5.6).

5.2.3.2 Engine load shift

The Brake Specific Fuel Consumption (BSFC) [g/kWh] of an internal combustion engine (ICE) is the ratio between the hourly rate fuel consumption [g/h] and the engine power [kW].

When the vehicle is driving at a constant speed, the engine runs at a specific **operating point** (speed and torque) which might not be at highest BSFC value. On a P0 MHEV architecture, from the drivability point of view, the total powertrain torque (at the crankshaft) must fulfill the driver's torque request. This means that the same crankshaft torque level can be maintained with different torque values for the generator and internal combustion engine.

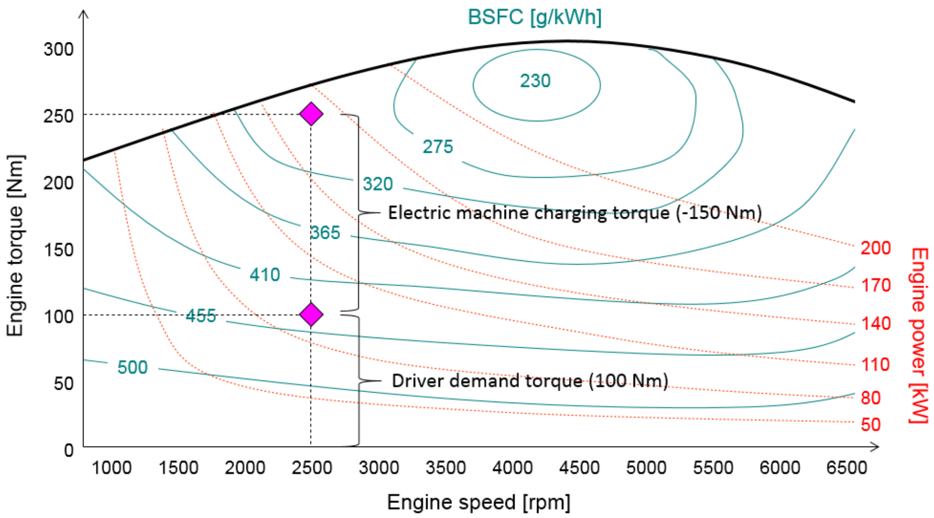


Figure 5.8: Example of Engine Load Shift application.

For example (figure 5.8), if the driver demands 100 Nm of torque at 2500 rpm, to maintain a constant vehicle speed, the engine is running at a low efficiency, where BFSC is 455 g/kWh. In order to increase the efficiency of the engine, the electric machine is set in generator mode (and charge the battery), with a load torque of -150 Nm.

To compensate for the additional electric load, the engine torque is increased to 250 Nm. The same crankshaft torque level is maintained (250 – 150 = 100 Nm), increasing in the same time the efficiency of the engine, with the BFSC at 320 g/kWh. Thus, the engine load (torque) is shifted from 100 Nm to 250 Nm with an increase in efficiency*.

**(There are several constraints to engine load shift strategy, one of them concerns the exhaust gas emissions. At high loads, the internal combustion engines have significantly higher emission levels (NOx, particles) compared to medium or low loads. However, this thesis work aims to the hybrid vehicle's performances, leaving aside these aspects.)*

Application on the 1108 cc Engine

In the case of the hybrid vehicle designed, the engine torque command in this mode is more than twice the one given to the electric motor, so then the first will work close to the highest operating points even at low speed. On the other hand, the electric motor will require lower energy than the normal functioning, consequently the energy recovered by the generator will be greater than the energy spent.

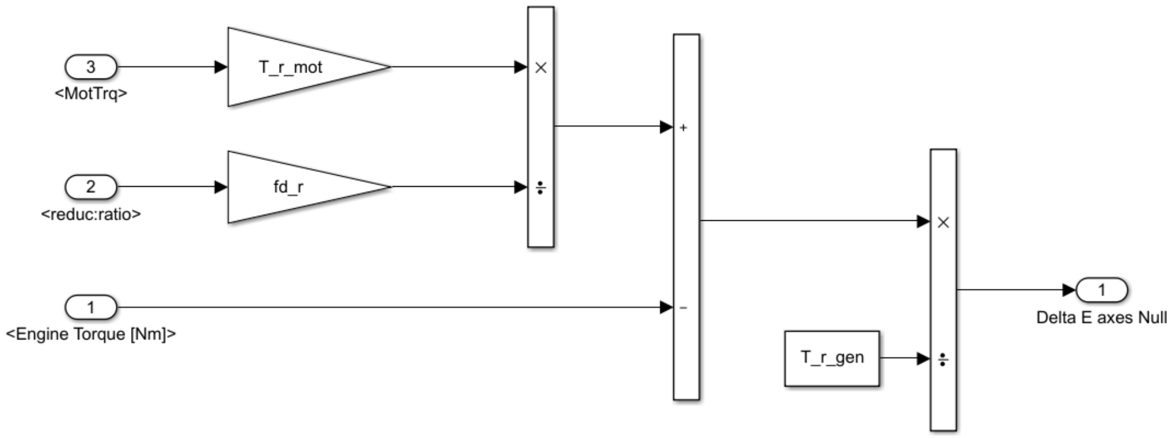


Figure 5.9: Logic for the generator command which returns a null Torque gap between axles.

Furthermore, considering the project requirements, the logic of the generator torque command for the load shift leads to the balance of the torque on both the axles (figure 5.9), so then the total torque supplied will reach the same values of the standard functioning. This condition can be obtained, of course, only up the maximum power deliverable by the electric machine.

The first of the following equations returns to the generator the command named "Delta E axes" for the purpose discussed. This value is set equal to zero, obtaining the second equation which returns the generator command for Engine Load Shift:

$$Delta\ E\ axes = MotTrq * T_{r_{mot}} - \left(\left(EngTrq + GenTrq * T_{r_{gen}}\right) * GearRatio\right) * DiffRatio = 0$$

$$\rightarrow\ GenCom = \frac{\left(\frac{MotTrq * T_{r_{mot}}}{DiffRatio * GearRatio} - EngTrq\right)}{T_{r_{gen}}}$$

As can be seen, in case the engine torque is greater than necessary the equation will return a negative command to the generator, which will apply a resistant torque on the transmission belt, reducing that torque gap on the axles. In the following table are given the elements of the equations above.

$MotTrq$	Motor actual torque value supplied [feedback]
$EngTrq$	Engine actual torque value supplied [feedback]
$GenCom$	Torque Command given to the Generator which returns the axes balance
$T_{r_{mot}}$	Reduction ratio between the motor and the wheels
$T_{r_{gen}}$	Reduction ratio between the generator and the engine in the transmission belt
$GearRatio$	Actual value of gear ratio engaged by the driver [feedback]
$DiffRatio$	Front differential reduction ratio

5.2.4 Motor

The electric motor provides only two functioning modes, one for traction and one for regenerative braking, respectively the upper and lower part of the logic shown in figure 5.10.

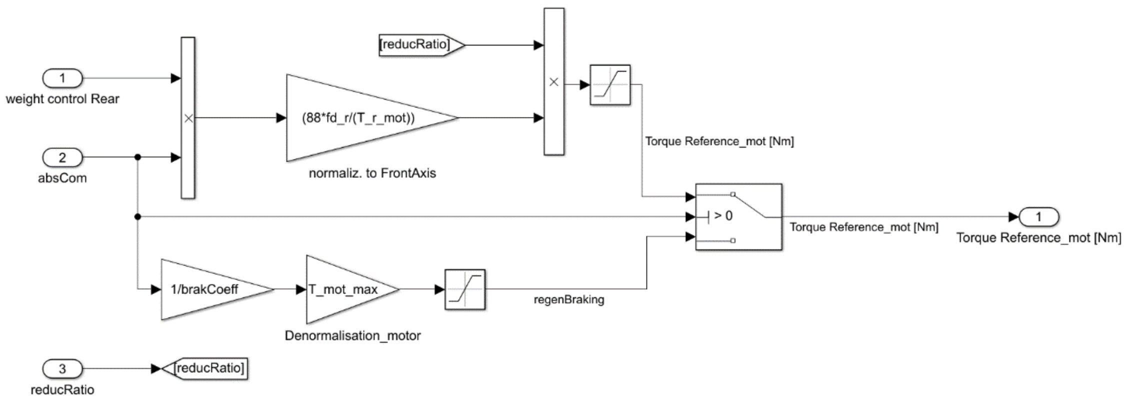


Figure 5.10: Motor Controller, internal logic designed.

Similarly to what has been done for torque balancing in the generator controller, also for the motor command it's compulsory to take into account the reduction ratio of the gears selected by the driver, the maximum torque values supplied from both the engine and the motor, and the several fixed reduction ratio through the driveline.

As a result, during the vehicle acceleration the motor will supply a torque value such that the final torques on the axles will be balanced.

Now the clutch signal is used to stop the motor when the driver is changing gear, however, a minimum torque is always requested from the motor, while driving, to ensure continuity in the vehicle movement .

Concerning the regenerative braking mode, it's necessary to clarify the logic adopted on the electric motor, being the same one implemented on both the electrical machines of the hybrid vehicle designed.

5.2.4.1 Brake Regeneration

One of the main features that hybrid vehicle powertrain technology brings with is the regenerative braking, converting a portion of a car's forward motion back into electrical energy to be used later.

Regenerative braking takes the electric motor and drives it in such a way that instead of behaving as a motor driving the wheels, it behaves like a generator and, on the contrary, it is driven by the wheels.

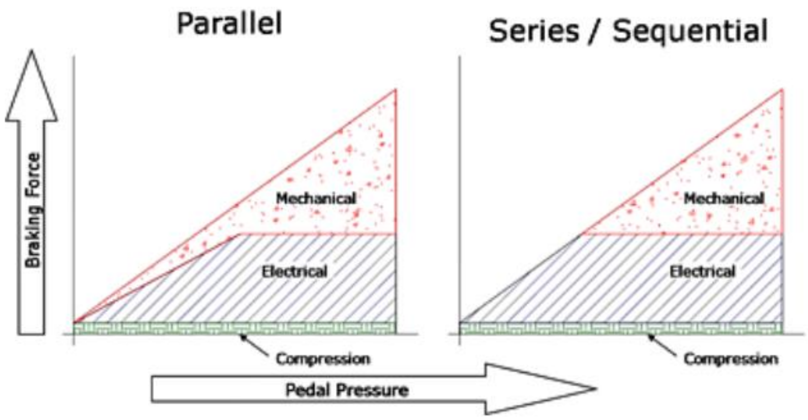


Figure 5.11: braking force behavior in Parallel and Series braking systems.

There are a few methods or strategies to do this. In each technique not all of the forward motion is returned to electrical energy. Some is consumed by the internal combustion engine due to the engine friction torque, a good portion is converted into electricity, while the remaining is converted into heat by the brake pads (if used).

Nevertheless, all the strategies can be summarized in two main categories, the Parallel and the Series (or sequential) Regenerative Braking (figure 5.11).

Between the two methods, parallel and series (or sequential), parallel is the easiest to control because the braking consist in a linear functioning. Pedal pressure increases both mechanical and electrical braking forces. Series systems must hold off the mechanical braking until the regenerative braking mechanism has supplied all the available resistant torque.

This is not so simple to achieve, since the instantaneous activation of the mechanical brakes should result moderate as less as possible in order to provide continuity in the overall braking effect.

The regenerative braking logic designed for this project belongs to the Series variant.

Given the above, the regenerative command assigned to the rear axle motor will return an increasing resistant torque until the brake pedal signal reaches the **Braking Coefficient** imposed, starting from that point the motor will supply the maximum resistant torque available even for higher values of the brake signal. Conversely, the crossing of the braking threshold will lead to the mechanical brake's actuation.

5.2.5 Engine

The engine control of the hybrid vehicle differs from the one of the starting model only for the command adopted in case of engine load shift that is, how it has been said before, higher than the standard one.

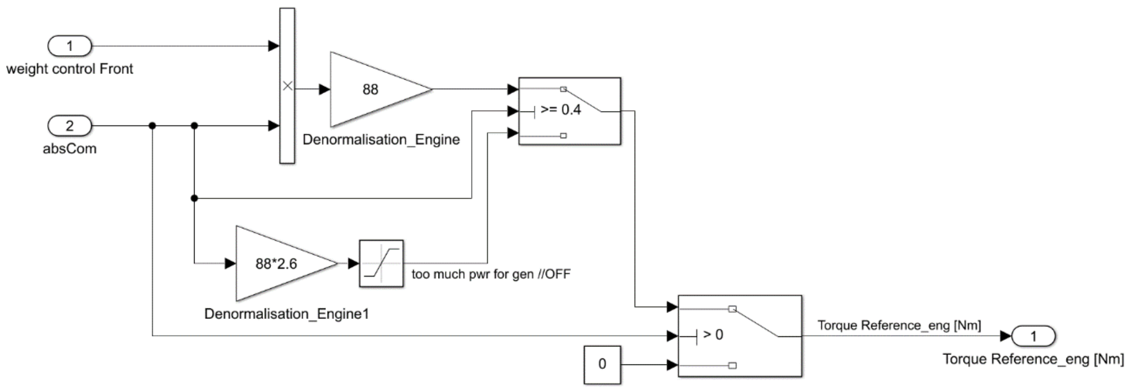


Figure 5.12: Engine Controller, internal logic designed.

The logic shown in the figure 5.12 returns a command of:

- **Standard Torque**, when the acceleration command is higher than the Acceleration Threshold. The torque is proportional to the acceleration command and normalized to the maximum torque deliverable.
- **High Torque**, when the acceleration command is lower than the Acceleration Threshold. The torque value requested here is higher than the double of the one

requested from the electric motor (2.6 times greater in the case shown), in order to lead to the engine load shift.

- **Null Torque**, when driver's command is null or negative. But in this case the actual torque delivered by the engine will be not equal to 0. Indeed, this is the case of "Engine Braking" where the engine supplies a certain value of friction torque depending on the speed which it is running at.

  As already said in the previous paragraphs, this leads to a **parasitic loss** for the vehicle, because the amount of friction torque supplied from the engine cannot be converted into energy delivered to the battery.

Torque Split Strategy Truth Table

The several modes which the vehicle can work with are summarized in the following table:

Mode	INPUT		ENGINE	MOTOR	GENERATOR	Braking System
Acceleration	Acc > AccThres	RPM > 3000	Standard Torque ↑	Standard Torque ↑	✗	✗
Torque Fill	Acc > AccThres	RPM < 3000	Standard Torque ↑	Standard Torque ↑	Filling Torque ↑	✗
Engine Load Shift	0 < Acc < AccThres	_	Optimal Torque ↑	Standard Torque ↑	Regener. Torque ↓ (Based on ΔE)	✗
Regenerative Braking	-BrakCoeff < Acc < 0	_	Engine Braking Torque ↓	Regener. Braking Torque ↓	Regener. Braking Torque ↓	✗
Series Braking	Acc < -BrakCoeff	_	Engine Braking Torque ↓	Regener. Braking Torque ↓	Regener. Braking Torque ↓	✓

Legend:

↑ = *Traction Torque*

↓ = *Resistant Torque*

= *Charging Battery*

= *Discharging Battery*

5.3 Regenerative Braking with Torques Balance

By assigning the same deceleration command to the electrical machines, excellent results are obtained from the energy point of view, through the recovery of energy on both axes.

However, this type of control does not ensure a good grip of the vehicle on all 4 wheels, for the following reasons:

- the electrical machines used have different powers;

- the generator is connected to the engine, so it too can vary its resistant torque according to the gears chosen by the driver;

- the engine produces a considerable friction torque due to the motor brake effect, so this unavoidable problem must also be taken into account.*

**Note: to acquire this data on the real vehicle it will be sufficient to obtain it from the rotation speed of the engine, the only variable on which it depends.*

For these reasons, it was decided to balance the torques at the braking axles, respectively at the rear and at the front one, by acting on the torque control at the generator and leaving the one at the electric motor unchanged:

$$MotTrq * T_{r_{mot}} = \left(GenCom * T_{r_{gen}} + EngTrq\right) * DiffRatio * GearRatio$$

the control torque at the generator will then be:

$$GenCom = \frac{\frac{MotTrq * T_{r_{mot}}}{(DiffRatio * GearRatio)} - EngTrq}{T_{r_{gen}}}$$

$GenCom$	Generator Input Command	
$MotTrq$	Actual Motor torque supplied	[feedback]
$EngTrq$	Actual Engine Friction supplied	
$T_{r_{mot}}$	Reduction ratio between the motor and the wheels	
$T_{r_{gen}}$	Reduction ratio generator-engine through the transmission belt	
$GearRatio$	Actual value of gear ratio engaged by the driver	[feedback]
$DiffRatio$	Front differential reduction ratio	

Implementation

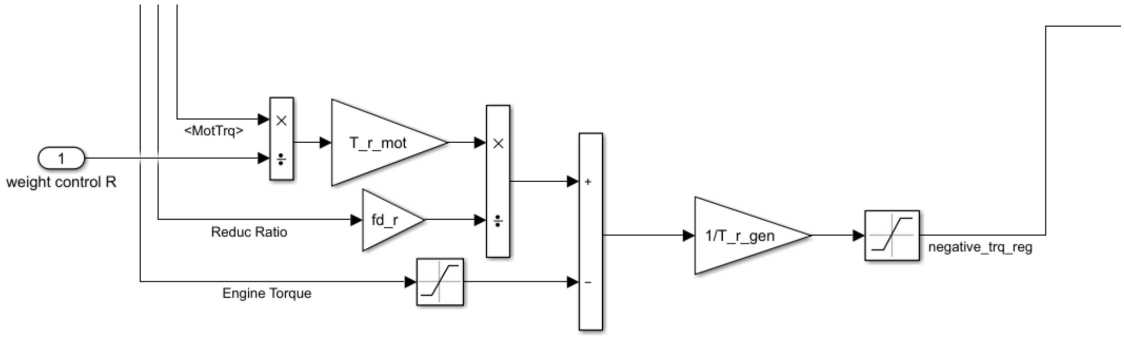


Figure 5.13: Generator Controller, Torque Balance logic for Braking Regeneration command.

5.4 Weight Control

The tires pulling force is directly proportional to the vertical load present on the axis considered (equation 2.2). The vertical load depends on:

- distribution of weights on the vehicle;
- acceleration;
- grade of the road.

Consequently, the fluctuation of the load on the axles is chosen within the control of the motors, in order to obtain a more homogeneous distribution of the torques, which considers the variation of all the parameters mentioned before in a single solution.

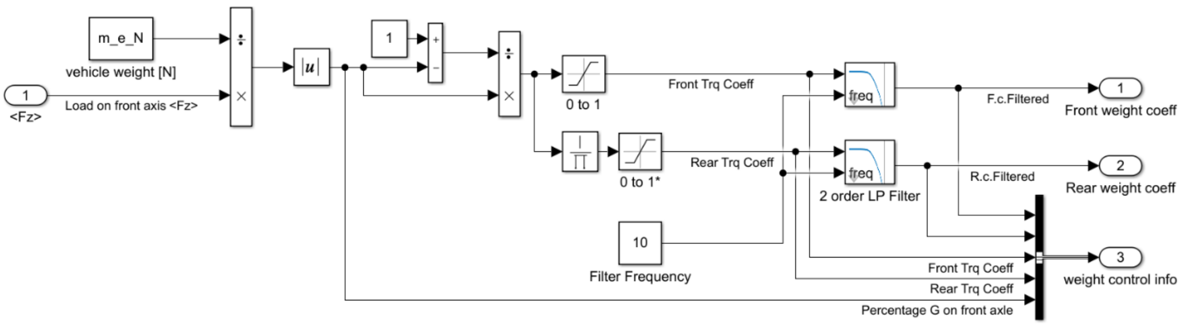


Figure 5.14: Weight Control Logic designed.

The logic designed (figure 5.14) takes in input the vertical load on the front axis and returns the weight percentage present on both the axles considering the vehicle total mass of the vehicle, known and fixed except for any future developments.

Once these values have been obtained, the logic allows to obtain coefficients at the output that are multiplied by the torque commands to the motors that lead to:

- Motor torque commands unchanged if the weight distribution is perfectly balanced (50:50).
- Limited torque control command, on the lightest axis, so that its command torque is proportional to the ratio between its load and the one present on the reciprocal axis, which instead will receive the maximum torque command available.

The coefficients of use of the torque are unitary, therefore for "0" there will be no torque and with "1" there will be the maximum torque that can be assigned.

Using a weight control, you get that the most limited axle is the rear one, in fact, except for the strong accelerations, more than 50% of the weight of the vehicle is unloaded on the front axle and the controls to the motors are limited accordingly.

From the logic used, it can be seen that it was necessary to introduce second-order filters to obtain more damped controls with respect to the rate of the vehicle's centre of gravity.*

**Note: discrete Lowpass filters with 10 rad/s cut-off freq., sample time of 0.01sec.*

As shown in the figure (5.5), the selection of the weight-based control is done through the interface of the logic inside the control block, acting on the respective blocks of the engine and motor.

A rather simple solution to implement this type of logic, on the real vehicle, is to install sensors that detect the lengthening of the suspensions during the race, then through the knowledge of their dynamics you can easily trace the vertical load present at any time on the tire.

A demonstration of the application and operation of these logics will be made in the following paragraphs.

Chapter 6

Simulations

In this chapter several simulations are performed in order to validate the proper functioning of each implemented logic.

To begin, the results of the global functioning of the hybrid vehicle are shown, while in the following paragraphs all the control logics developed will be validated.

The simulations foresee the use of two simulation conditions with the following characteristics:

- Dry Asphalt, flat road, speed profile US 06[*1.2] (figure 6.1).
- Soil/Sand, inclined profile, speed profile US 06[*0.8] (figure 6.2).**

**The speed profiles have been selected so that they can be properly followed by the hybrid vehicle under the selected conditions.

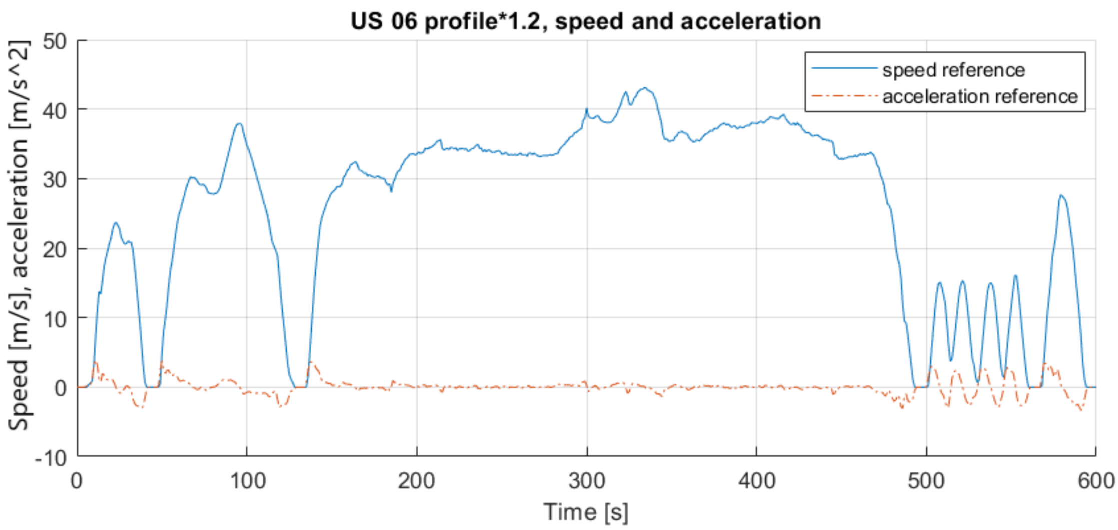


Figure 6.1

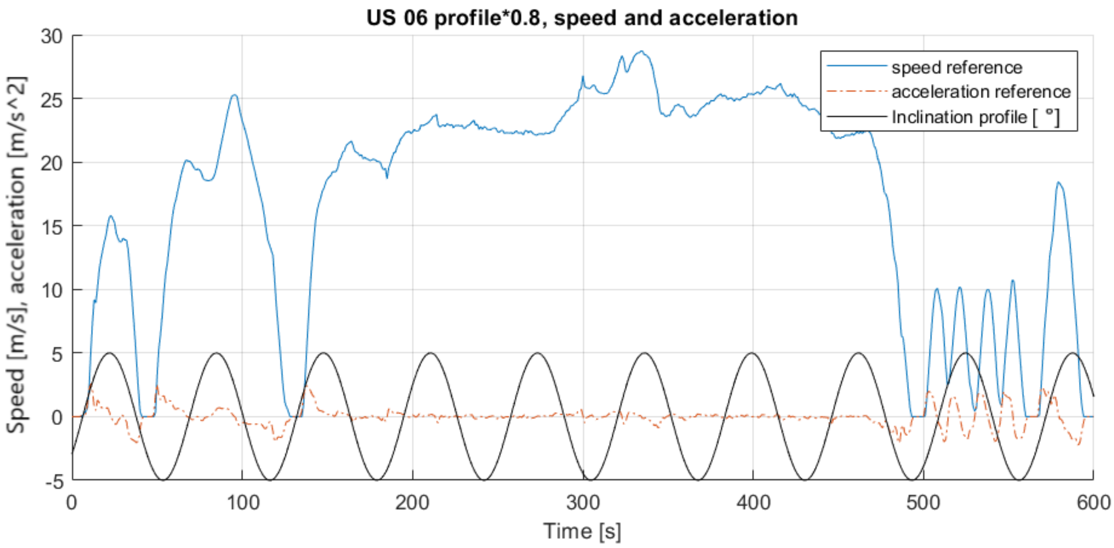


Figure 6.2

In addition, both types of simulation will be performed with variables:

- AccThres = 0.45
- BrakCoeff = 0.3
- **Engine command = 3*Motor Command**, during the Engine Load Shift mode.

In order to make the exposure clearer, only the most interesting simulation parameters to be analyzed in relation to each type of logic adopted will be reported. However, the model obtained allows to acquire further parameters useful to the user.

6.1 Torque Split Strategy, Global Test

Test on Dry Asphalt

Through the following simulation, it is possible to observe how the obtained model is able to adequately follow the imposed speed profile (figure 6.3).

Firstly, it's important to underline the changing through the different modes the vehicle works with.

The vehicle adopts all the operating modes, except for the Torque Fill, which only operates for a few moments. This happens because during the gear changing the thermal engine works at a speed close to the operation threshold of the Torque Fill, which is then immediately exceeded making the control inactive.

The battery charge status shows how the battery recharges during the Engine Load Shift and braking phases while discharging during the Acceleration Mode. The vehicle completes the simulation with 10% more energy, which means that under such conditions the regeneration threshold may even be lowered.

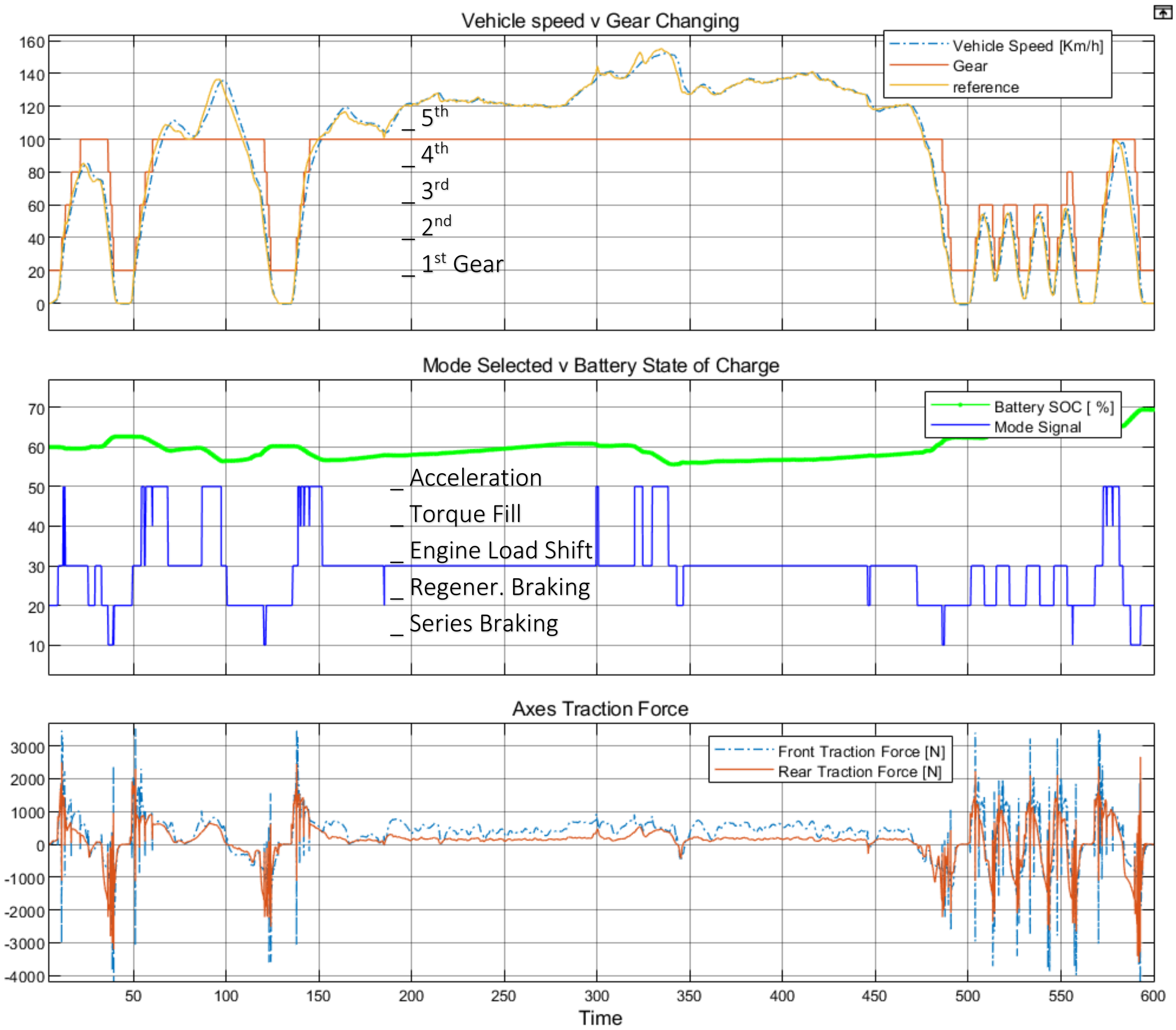


Figure 6.3

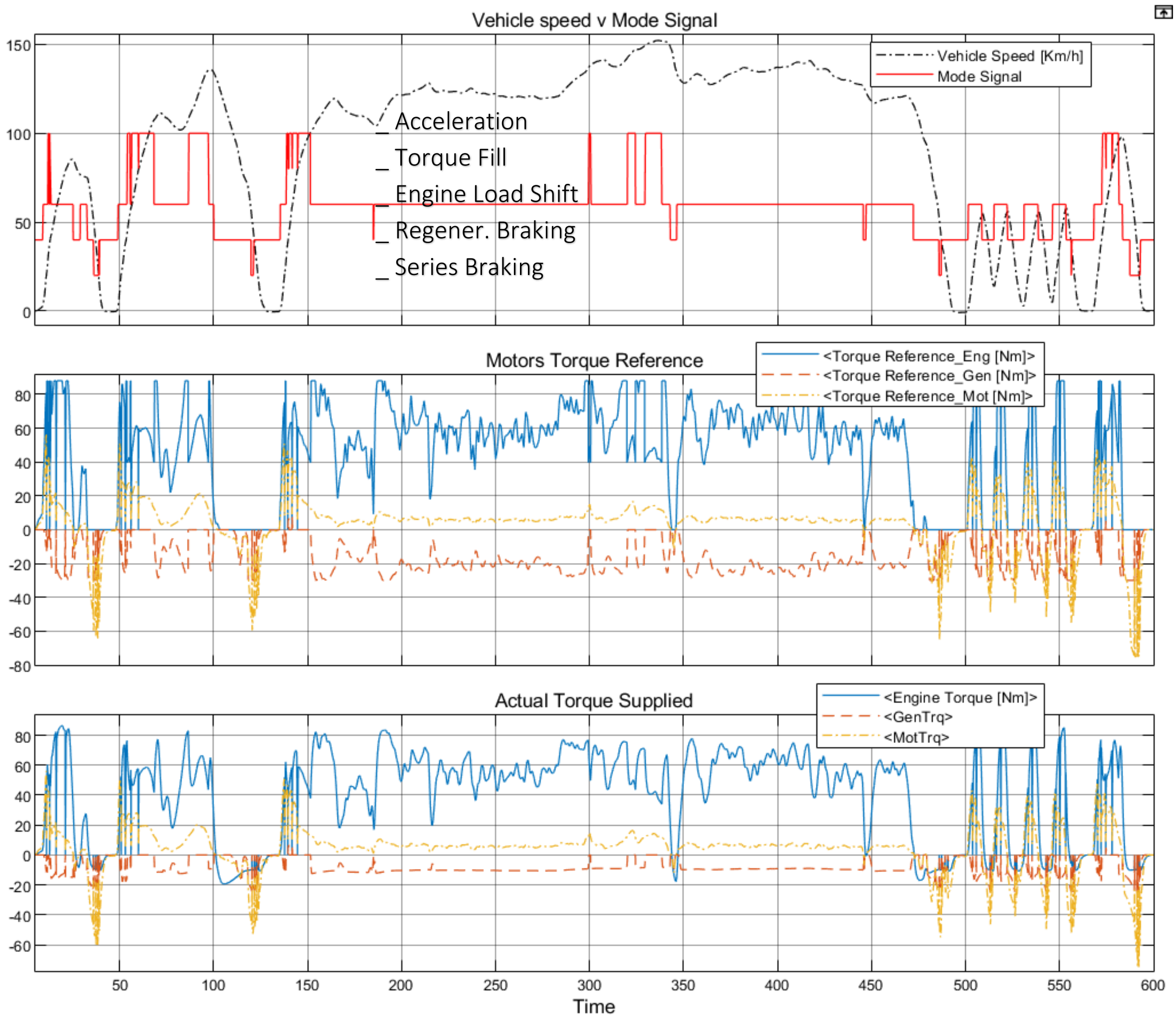


Figure 6.4

The graphs in figure 6.4 show how the operating modes act on the commands of the motors. For instance, in regenerative mode the thermal motor produces a much higher torque than the electric motor.

Another important aspect is the considerable difference between the commands assigned to the motors and the values of torque delivered. In particular, the thermal engine delivers maximum torque only at the optimal operating points (paragraph 2.4.2), while the generator often works in power saturation, except for the use of low gears (situation in which it can supply enough torque to the wheels).

Test on Soil

In the case of simulation on soil, the vehicle is not able to follow the imposed profile perfectly, especially when facing slopes in acceleration (figure 6.5).

Under these conditions, the battery ends the simulation with a charge percentage lower than the 20% of the starting value. This is caused by the vehicle's shorter permanence in regenerative mode. Thus, the regeneration threshold can be simply raised for better results.

The force values given to the axles are higher than the ones of the asphalt test, due to the higher rolling resistance and the introduction of an inclination profile.

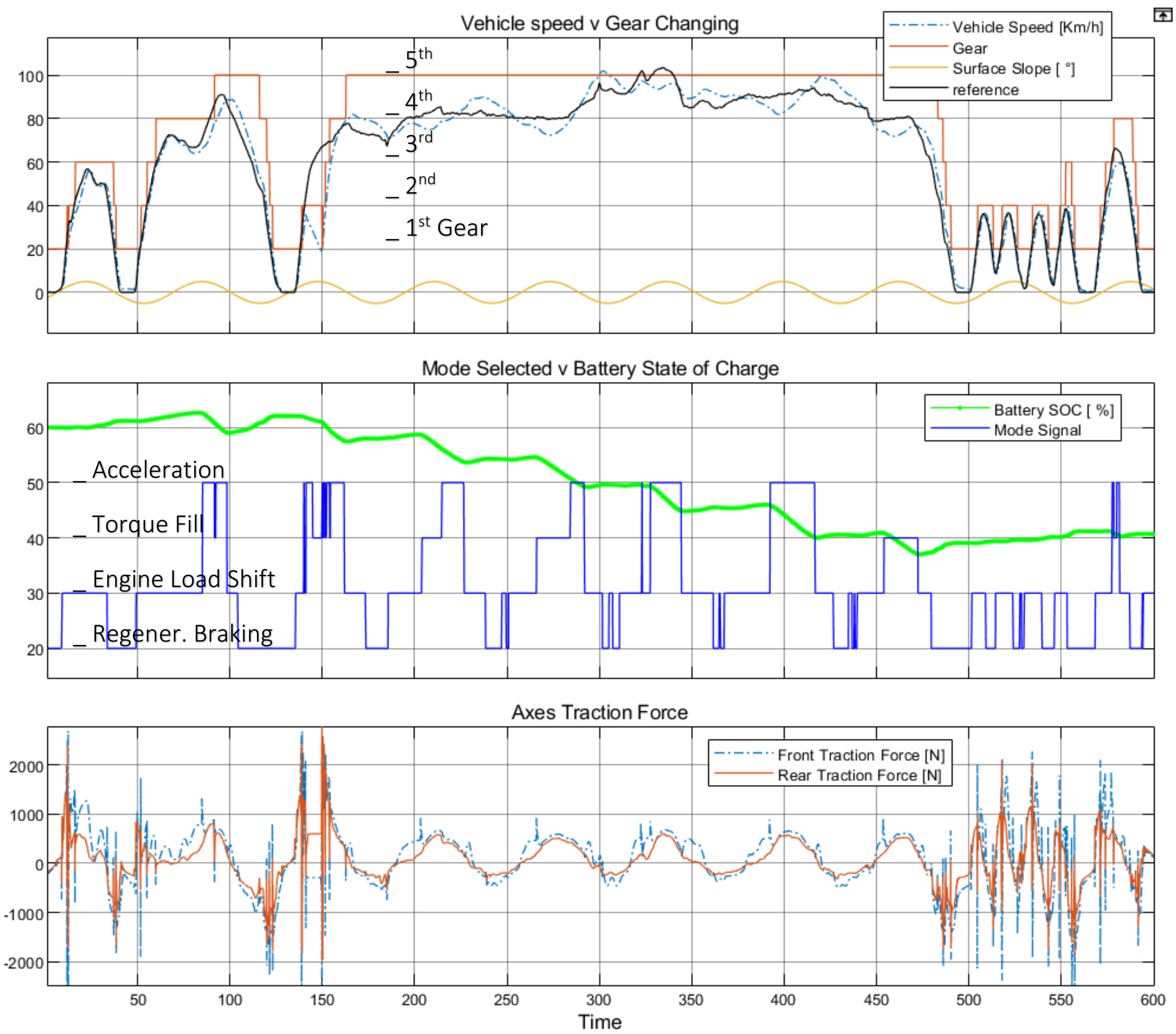


Figure 6.5

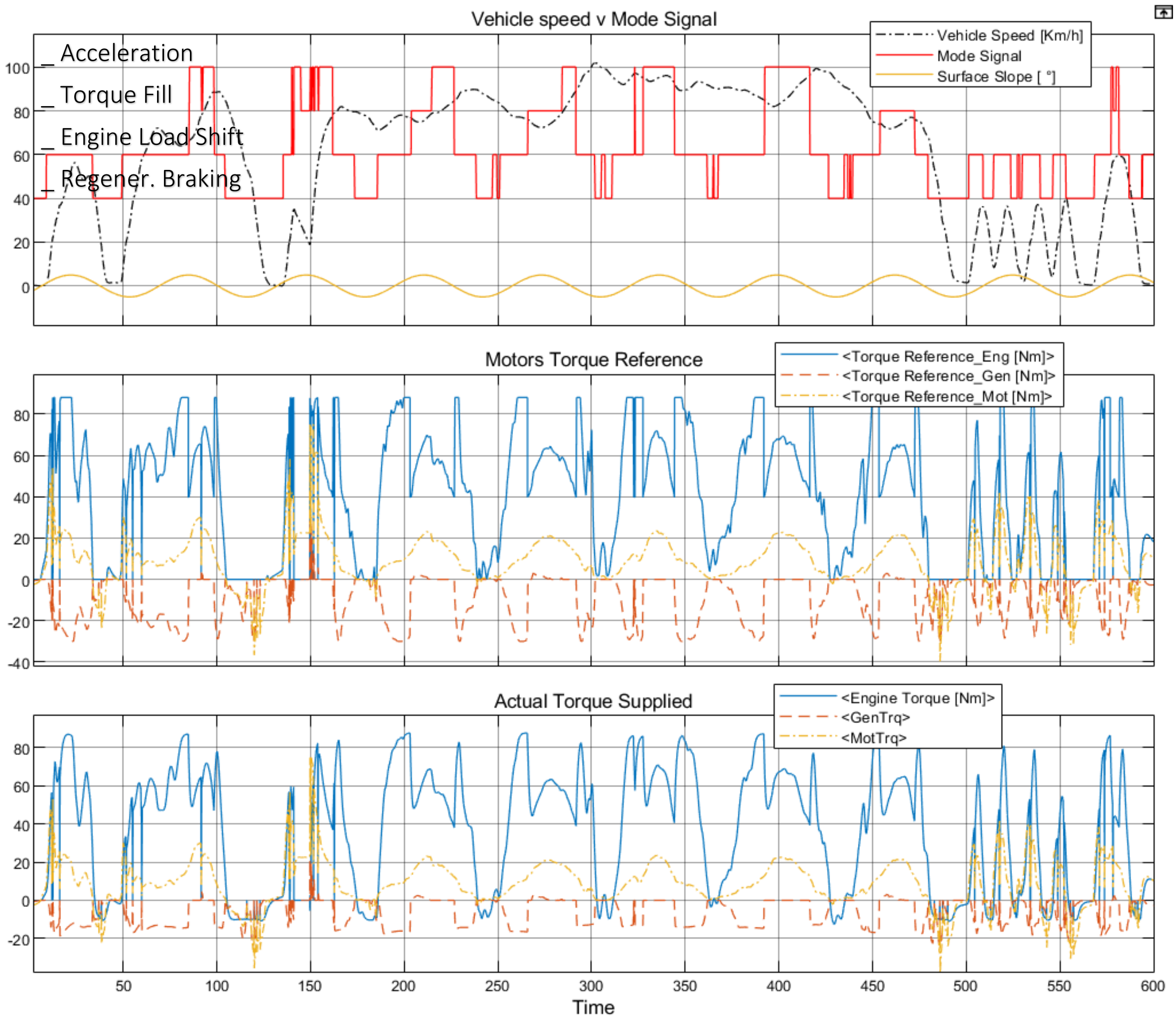


Figure 6.6

Equally, also in this case there is a difference between the torque commands given to the motors and the values of the torque actually supplied, especially for the generator (figure 6.6).

6.2 Torque Balance Control Validation

The axle torque balance is imposed in all vehicle operating modes except mechanical braking.

To test the functioning of axes torque balancing, the control of vertical loads must be deactivated. In fact, the load control acts on the torques supplied to the axles in proportion to the distribution of the vehicle's weights, but for correct operation it is necessary first to verify that the torque controls to the axles to be limited are equal.

Test on Dry Asphalt

From the graph (figure 6.7) it can be seen that, during the acceleration, in Engine Load Shift mode and during regenerative braking, the input torques to the wheels have similar values (torque distribution 50:50), demonstrating the functioning of the implemented logics.

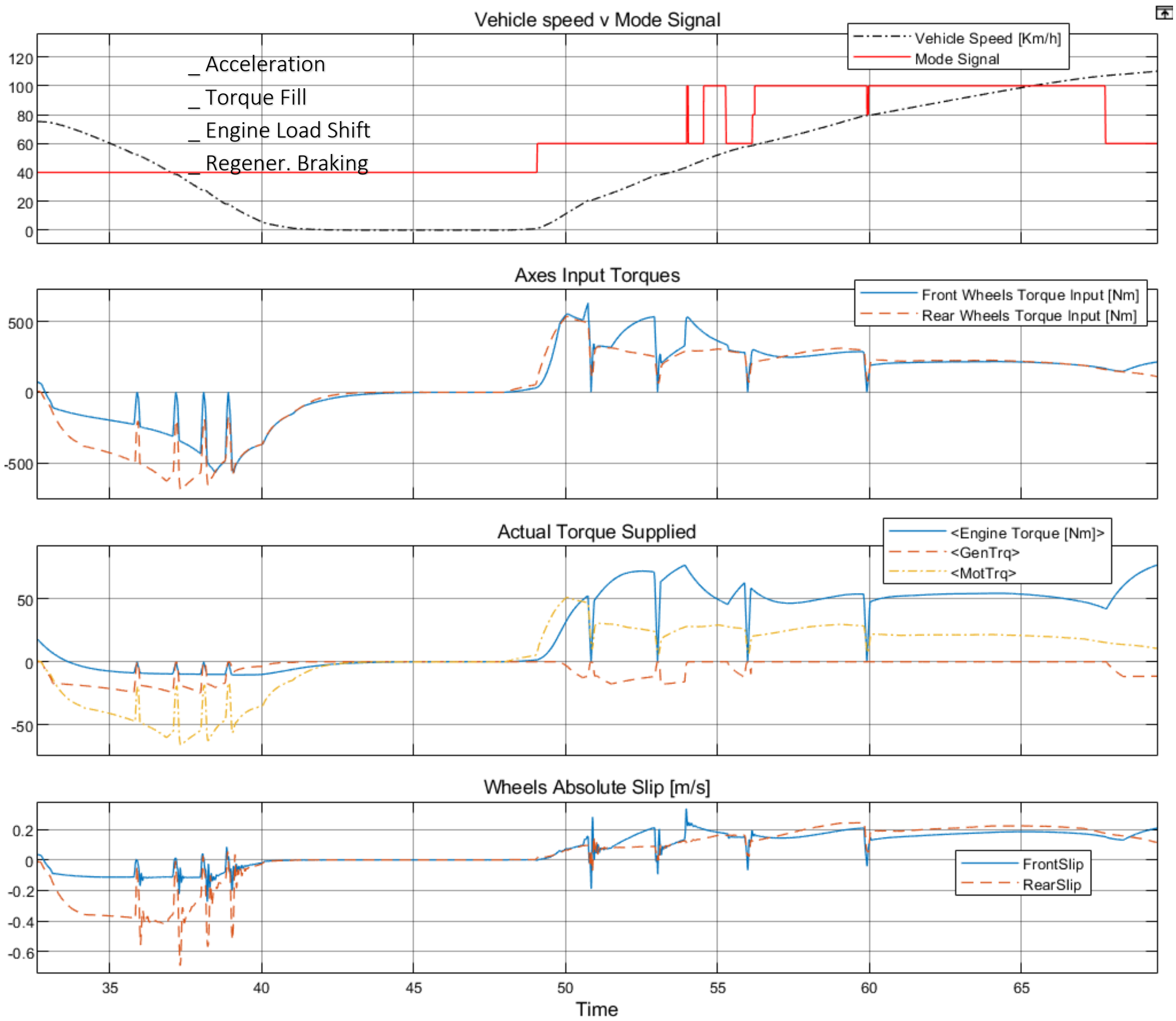


Figure 6.7

The presence of sections in which the values differ is due to the power saturation of the generator, therefore its torque is not sufficient to bridge the torque gap between the axles during the engine load shift and to balance the braking torque of the electric motor when using gears higher than the second (despite the passive help of the engine brake).

It can also be noted that for the same torque inputs to the axles you get slip values of the front and rear wheels different, this result justifies the need to use a different torque distribution from 50:50 to get a better traction. The greater deviations in the slip in case of braking compared to the engine load shift values are caused by the greater torque imposed on the wheels during braking and by the greater unbalance of the vehicle obtained in those conditions.

Test on Soil

The same conclusions can also be drawn in the soil test (figure 6.8). In this case we choose to focus more attention on the difference in the slip of the axles under acceleration with a torque distribution of 50:50. It is also evident that, due to slight decelerations, the engine friction constitutes a strong disturbance on regenerative braking and does not allow energy recovery.

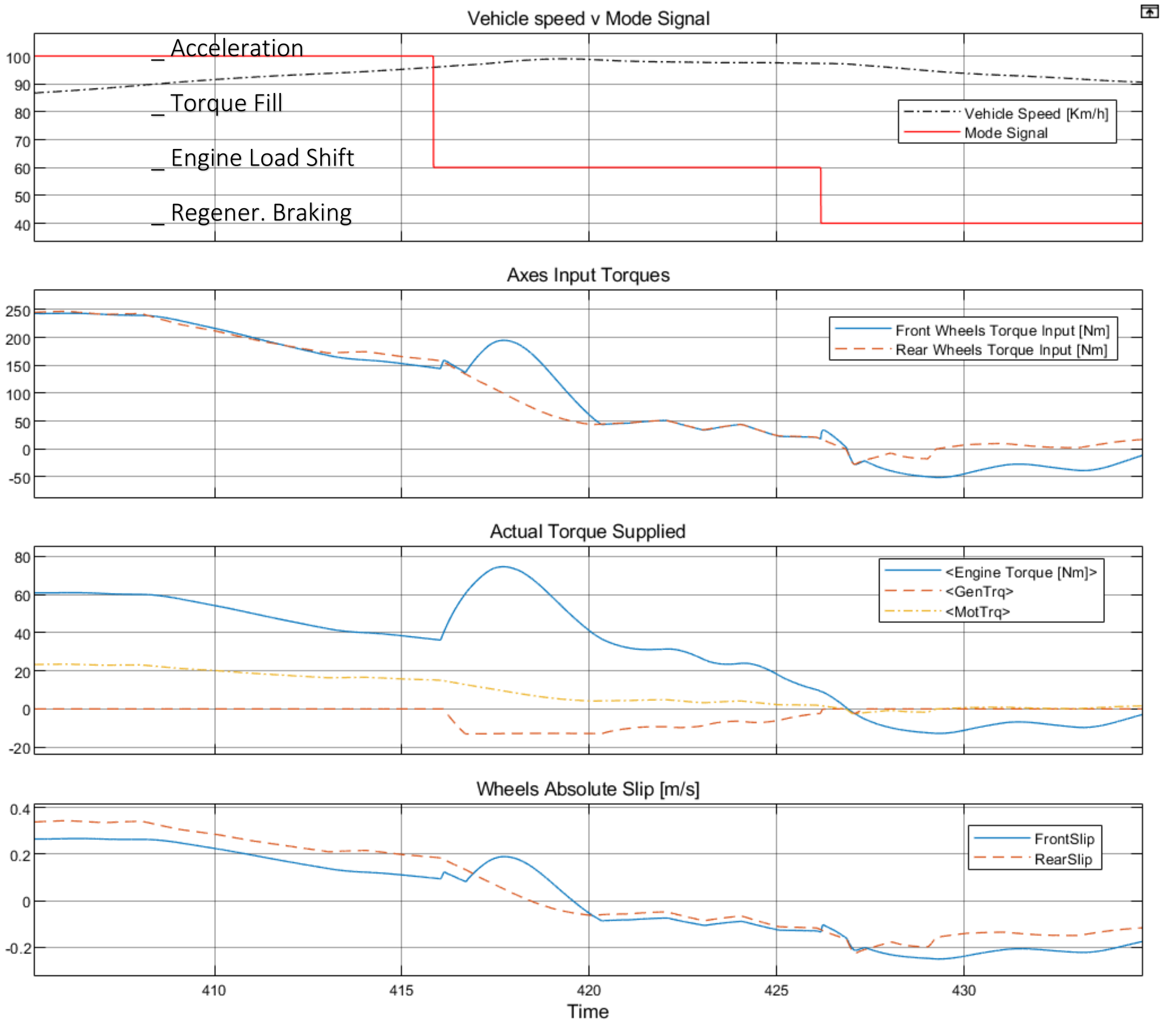


Figure 6.8

Test on Soil, example 2

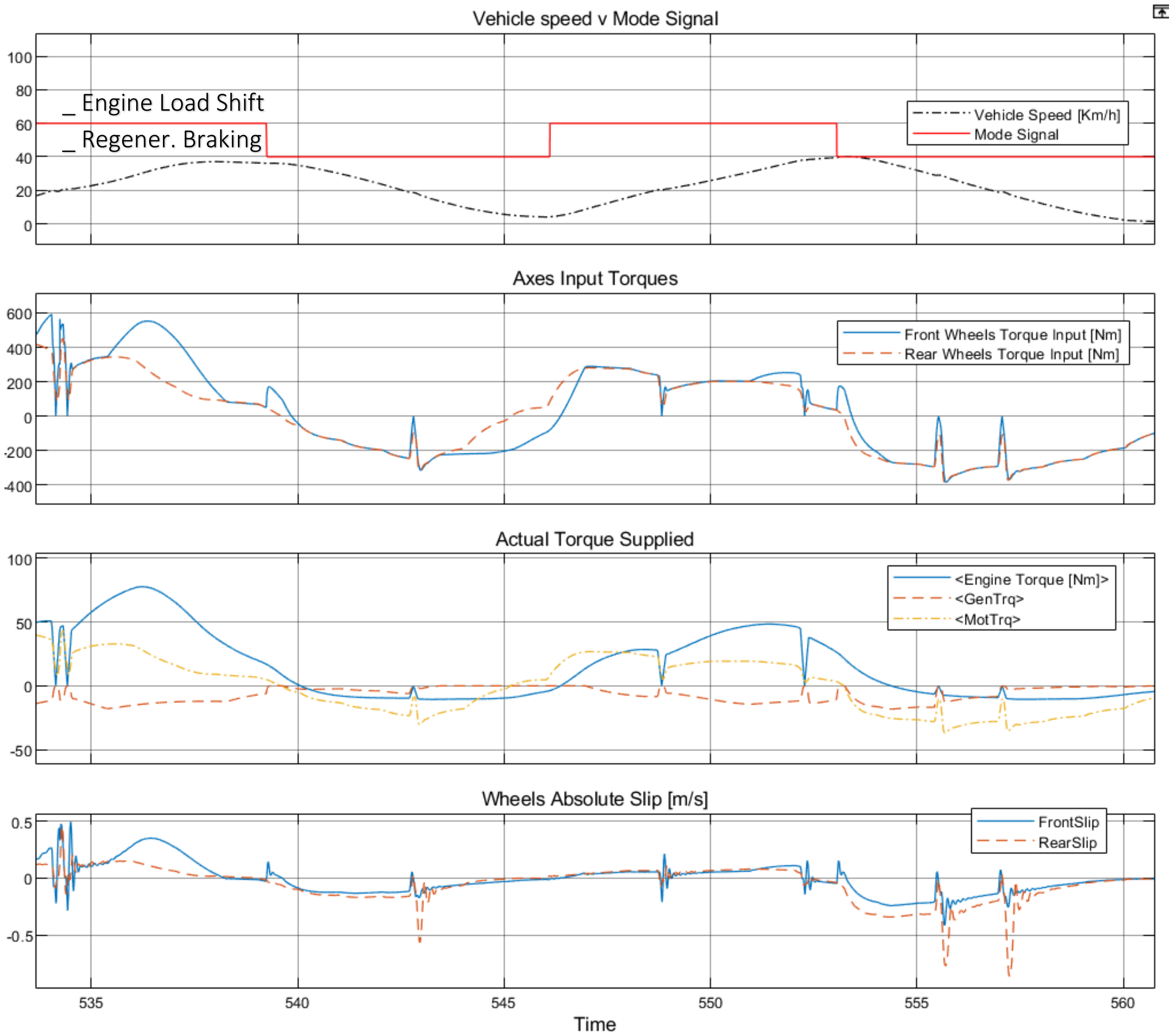


Figure 6.9

Another example on soil is given, in order to stress the need of a better splitting of the torque than the 50:50, especially under braking (figure 6.9).

6.3 Weight Control Validation

The weight control operates only in acceleration and braking (no Engine load shift).

The weight control gives excellent results but does not act significantly in acceleration because the car has the center of gravity moved forward (55%) while it is particularly useful in regenerative braking (figure 6.10). However, the control has effect only on light braking because once the mechanical brakes are applied, their action prevails over the electric motors in the total braking effect.

The only implicit disadvantage of this control is that when active the vehicle brakes more with the front axle and we have a lot of more energy lost for the engine friction effect.

Test on Dry Asphalt

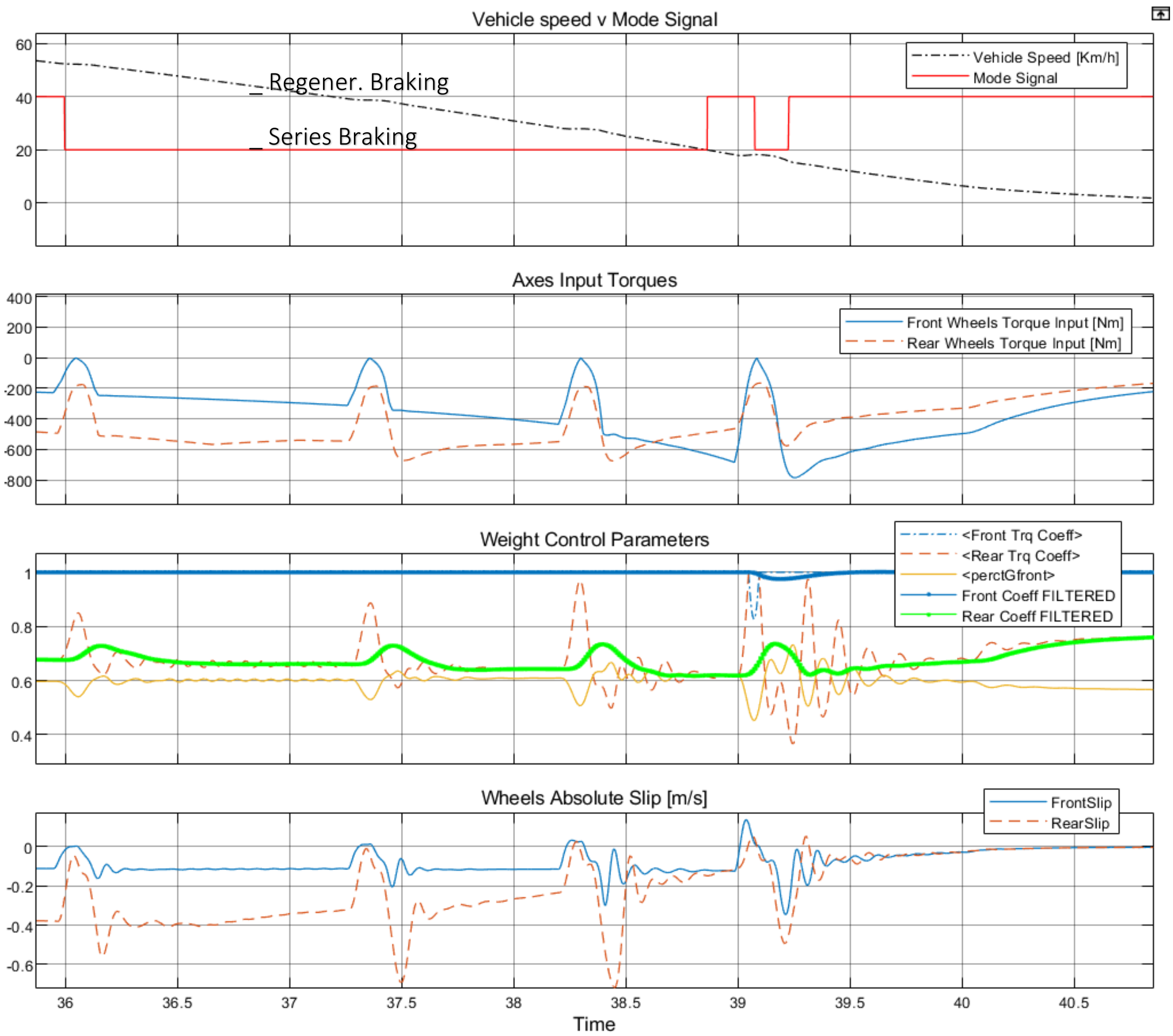


Figure 6.10

Test on Soil

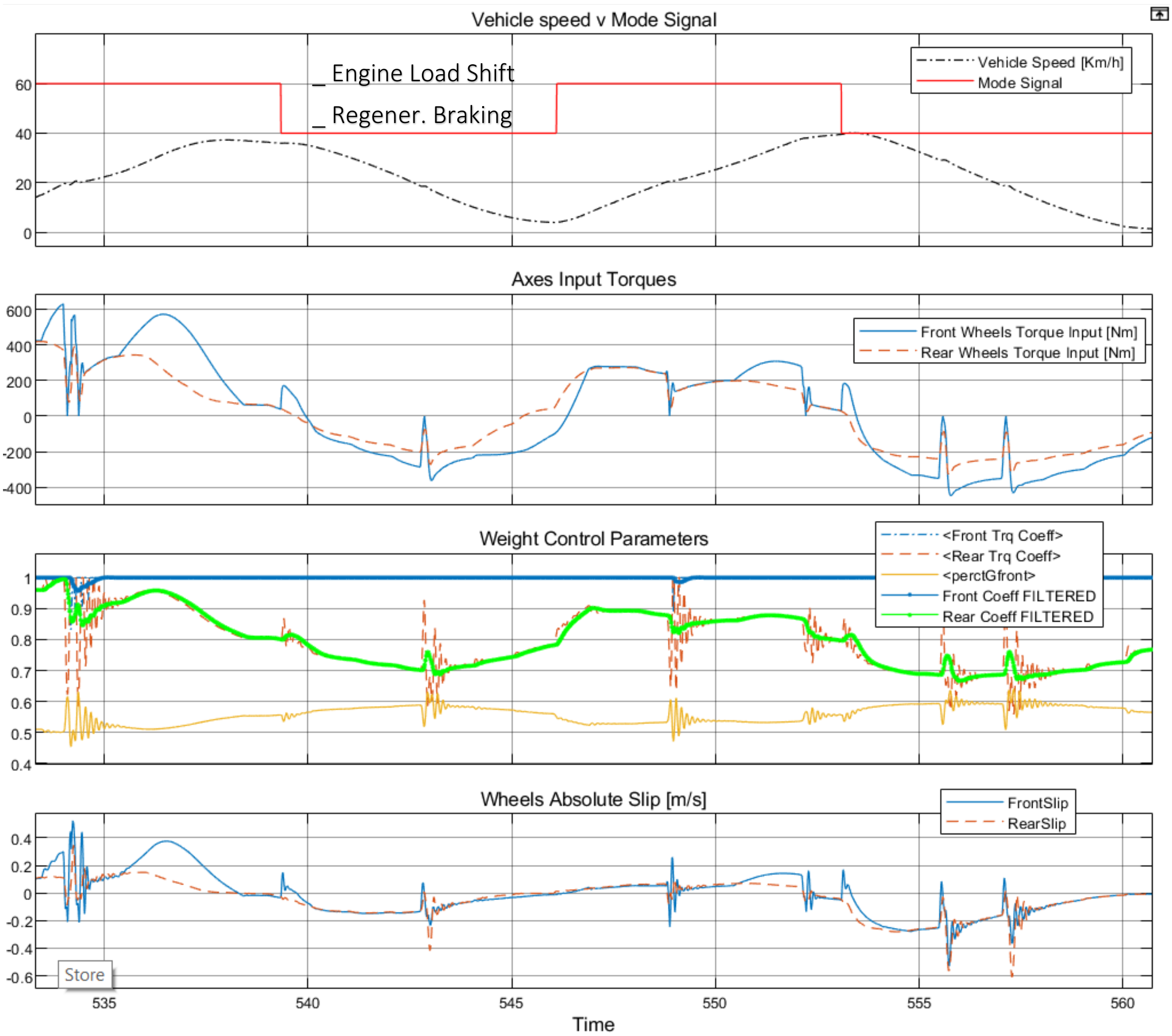


Figure 6.11

Good results are also obtained in the soil test (figure 6.11). Now the contact between wheels and soil produces the same slip on both axles using different torques in input.

Test on Soil, Limit Case

In order to validate better the weight control in acceleration, here we decide to run a simulation on soil changing the vehicle balance and moving its center of gravity to the rear axle, for a final design which sees the 55% of the load on the rear axle in stationary conditions.

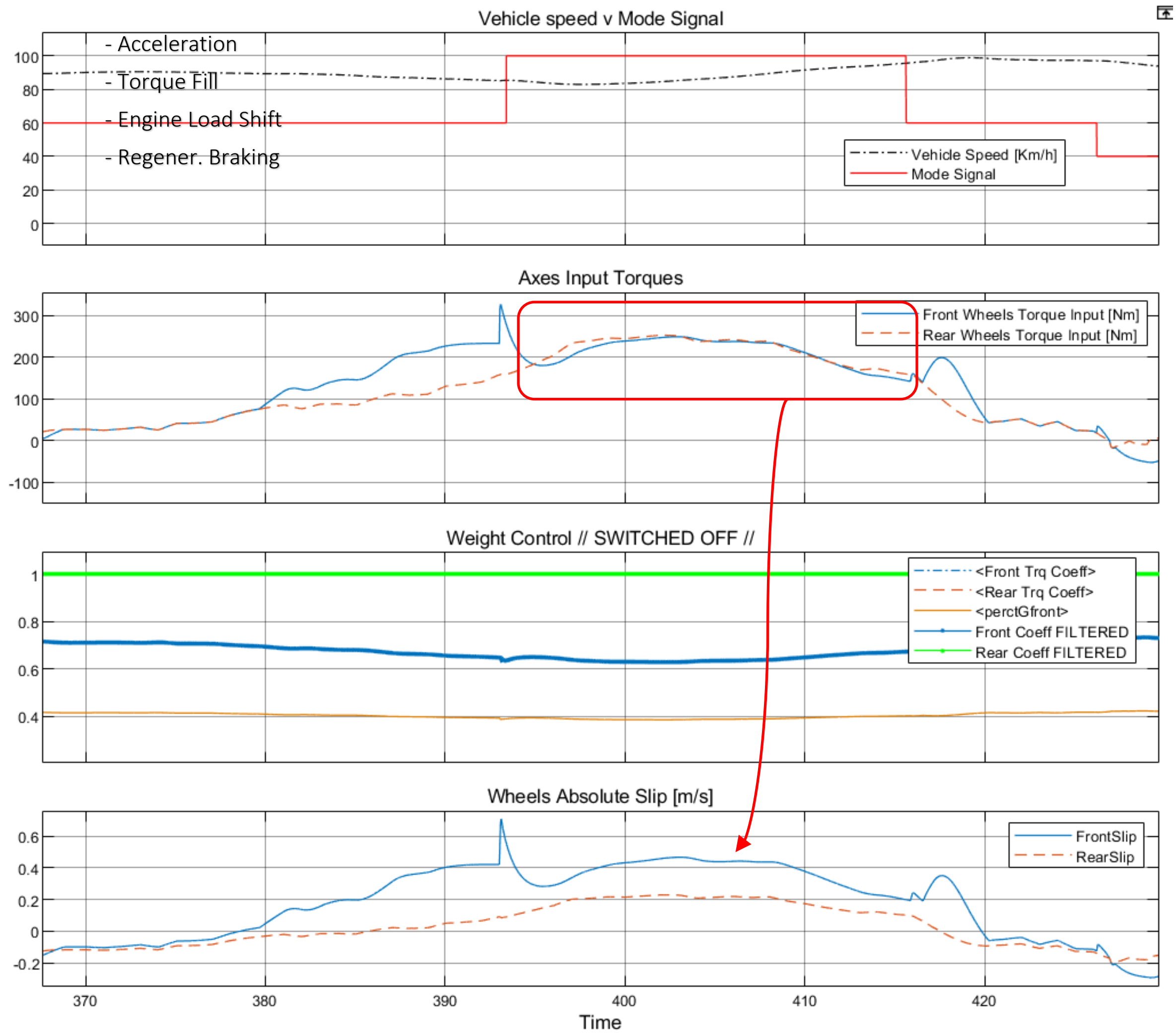


Figure 6.12

Now we can clearly see that the lighter axis is the front one, mostly in acceleration, and without a control which takes into account the vertical control we obtain very different values in terms of wheel's slip (figure 6.12) and, consequently, for the vehicle adhesion on the track.

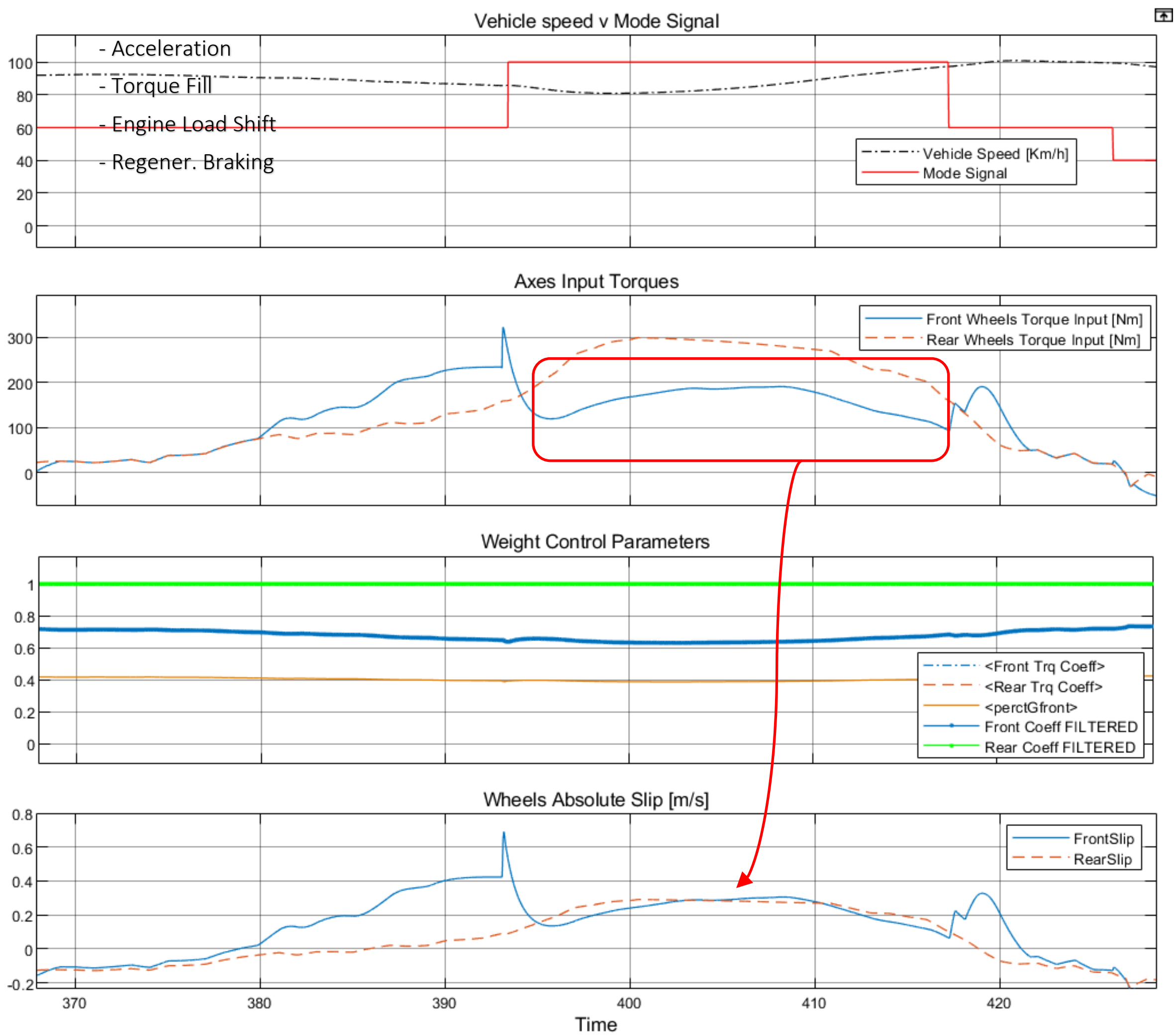


Figure 6.13

As counterevidence we activate the weight control running the previous kind of simulation (figure 6.13) and we can conclude that, if the final hybrid vehicle will have a bigger load on the rear axle, then the weight control will be mandatory for a good adhesion on the track.

6.4 Performance Benchmarking

Here the new and old model are compared on a Bang-Bang profile. Thus, the speed profile given in input to the driver makes it to accelerate as much as possible to run the two vehicles at the maximum speed they can reach.

Test on Dry Asphalt

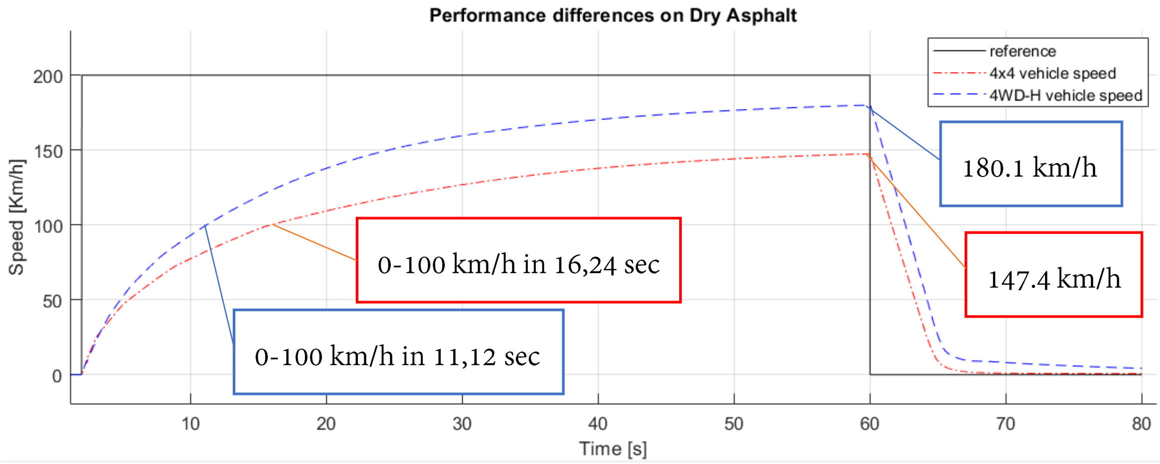


Figure 6.14

The performances obtained on asphalt by the two models are summarized in the figure 6.14. (Note that in these simulations we do not consider any kind of friction present in the driveline).

Test on Soil, same weights

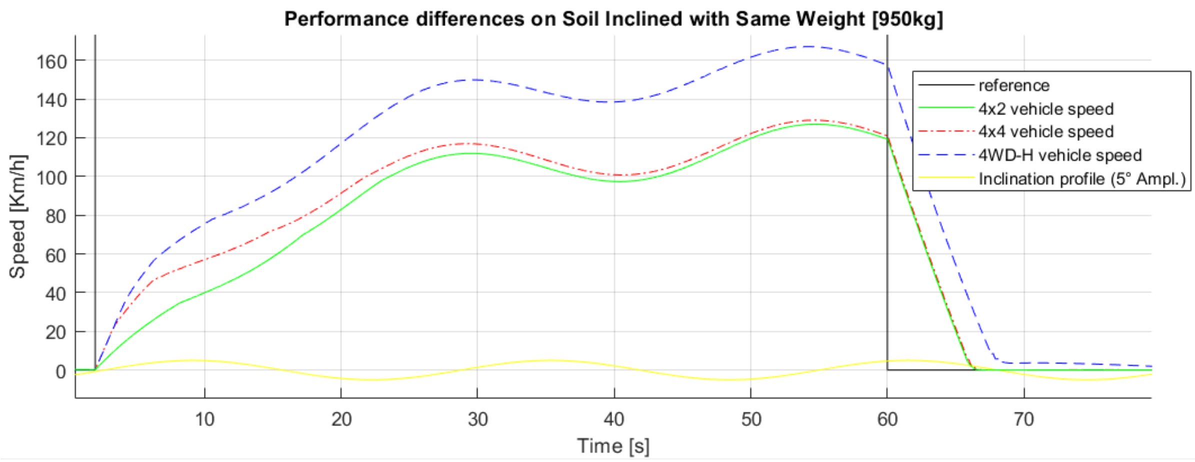


Figure 6.15

In this case (figure 6.15) we decide to run a simulation with the hybrid vehicle obtained, the original vehicle with 4x4 traction system and the 4x4 vehicle with the integral traction deactivated too, which makes it a two-wheel drive vehicle. The weight is the same for all the vehicles.

This test brings out several results, but the first and most obvious is that the best vehicle to face the path is the hybrid one. Additionally, we can see that the four-wheel drive traction is necessary on soil, mostly at the start, because in this condition the 2 wheel-drive vehicle loses a lot of traction than the 4x4 model. Moreover, both the hybrid and the 4x4 vehicle show a good adhesion at the start, but after few seconds the original vehicle reveals to have too much less power than the hybrid one.

Test on Soil, different weights

In order to better understand how much the values of power, weight and the kind of traction are essential and if one of these factors prevails over the others, in the global functioning of the vehicle, we run a simulation on soil with different weight for each vehicle model used.

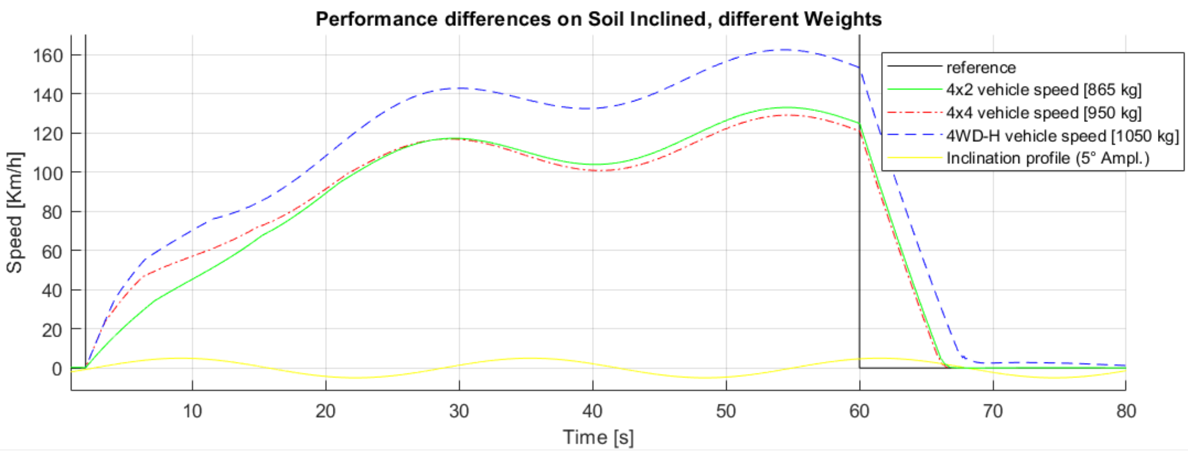


Figure 6.16

As can be seen (figure 6.16), the hybrid vehicle is still the fastest even being the heaviest vehicle at the same time. Moreover, the simulation stresses the effectiveness of the four-wheel drive traction due to the better performance of the 4x4 vehicle in acceleration. Indeed, the lightest vehicle with a two-wheel drive traction shows to be slightly faster only at high speed.

6.5 Energetic Analysis

Finally, we perform an energetic analysis on the hybrid model obtained. Firstly, we run a simulation with the normal functioning of the vehicle, then we disable the regenerative braking in order to know how much energy was saved in the previous test and measuring the level of the efficiency accordingly.

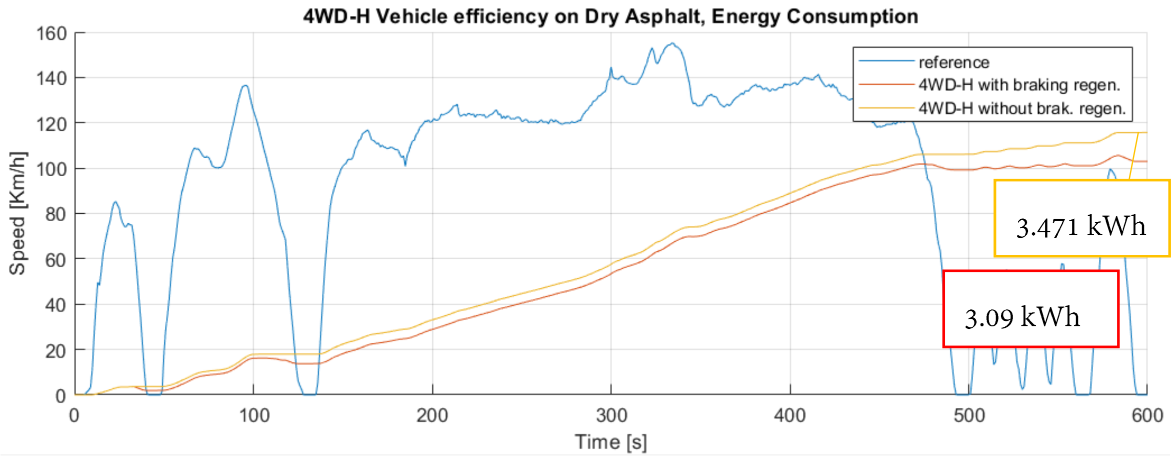


Figure 6.17

Given the energy spent by both the models, the efficiency of the hybrid vehicle for this kind of simulation on asphalt reaches the 10.9% (figure 6.17).

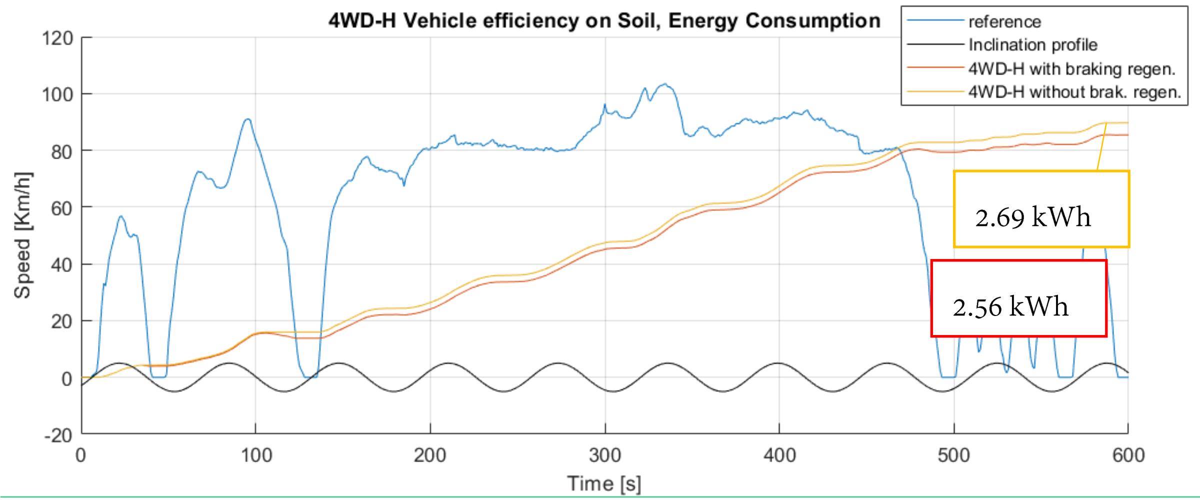


Figure 6.18

The vehicle efficiency on soil reaches the value of 4.8% (figure 6.18), so the half the one achieved in the previous test. This result is due to the lower braking requests but mostly because of the higher rolling resistance which slows down the vehicle, making the use of braking system less necessary than usual.

Chapter 7

Conclusions

The aim of this work was to realize the model of a hybrid vehicle starting from the Fiat Panda 1108cc 4x4, for the purpose of its participation to the Panda Raid event in the year 2020.

The starting vehicle has been modeled in the Simulink environment, and consisted in a longitudinal dynamic with three degrees of freedom to account for the longitudinal movement of the vehicle and its pitch. The model included the mapping of the real engine and the values of all the driveline's ratios taken from the datasheets.

The wheels model is based on H. B. Pacejka's theory, leading to an advanced representation of the contact between the wheels and the ground. This was done because of the important influence of the tire dynamic in the overall vehicle behavior, accordingly to the kind of surface it will face.

The proposals of the hybrid conversion included the addition, to the original engine, of an electric motor on the rear axle, a generator coupled to the engine on the front axle, and a small battery in order to keep the vehicle lighter. Consequently, a driveline based on independent axes has been obtained.

After the proper parameters to simulate the race path have been found, the importance of four-wheel drive was studied, especially on low adhesion surfaces. For this reason it has been fundamental to analyze the difference in adhesion obtained with the adoption of the 4x4traction on the original vehicle, in order to design a kind of control on the new hybrid vehicle with independent axes, which could achieve the same result or do even better.

Consequently, a control based on the vertical loads of the axes has been designed and implemented, along with a torque split strategy which could ensure the recovery of the energy during the simulation.

The final design of the hybrid vehicle sees the electric machine on the front axle working as a generator during vehicle deceleration and regenerative mode, and as a motor (to assist the engine) during vehicle accelerations. The electric motor on the rear axle has been designed to work constantly with the thermal motor to provide a four-wheel drive traction.

The hybrid model designed shows to manage well the state of charge of the battery, according to the impact of the energy recovery selected for the simulation.

It has been found that, given the hybrid architecture chosen, the vehicle reaches often the power saturation of the generator, therefore in those conditions its torque is not sufficient to satisfy the control requests.

The weight control gives excellent results but does not act significantly in acceleration because the car has the center of gravity placed forward, while it is particularly useful in regenerative braking. At the same time, it has been demonstrated that modifying the vehicle in such a way that it moves its center of gravity backwards, then the weight control becomes even more necessary for a complete adhesion of the vehicle during acceleration.

It is also evident that, during slight decelerations, the engine friction constitutes a strong disturbance on regenerative braking and does not allow a good recovery of the energy. It follows that, the only implicit disadvantage of this control is that when active the vehicle brakes more with the front axle and we have a lot of more energy lost for the engine friction effect.

Finally, the simulations on soil stress the effectiveness of the four-wheel drive. Indeed, a lighter vehicle with a two-wheel drive traction shows to be only slightly faster at high speed, but it loses a lot of traction in acceleration.

7.1 Future developments

Starting from the results obtained in this thesis project, other analysis can be conducted to avoid the functioning in power saturation of the generator.

Thus, a tradeoff should be found in the torque splitting strategy during the regeneration mode, between the generator demanded torque and the torque supplied by the electric motor, in order to still keep the battery charged during the race.

Otherwise a more powerful generator should be selected, leading to further consideration on the vehicle weight that would increase.

The model can be also modified in order to take care of effects that have been neglected up to now, like the dissipative phenomena which affect the real vehicle, leading to a more realistic model.

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