Torque Vectoring Differentials and Their Control

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Abstract

This thesis focuses on clutch type torque vectoring differentials and their control strategies, it dedicates to show effects of torque vectoring differential. Different solutions of clutch type torque vectoring differential and their layout schemes, characteristics (outputs on both axles, torque difference between two outputs, clutch torque, speed difference, etc) and working principles will be discussed using velocity diagrams. Then one solution will be studied in detail and a control logic is designed to optimize the performance of the differential, especially its effects on vehicle lateral behaviors. MATLAB and IPG CarMaker for Simulink are the main tools to analyze and compare these behaviors of vehicle with and without a torque vectoring differential.

Keywords

Torque vectoring differential, Vehicle behaviors, Control strategy, CarMaker, Simulink

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1 Introduction

1.1 Background

As automotive industrial develops rapidly, the importance of stability of a vehicle during cornering is being perceived by more and more people. Shibahata introduced the direct yaw control (DYC) in 1992[1] and then Motoyama applied the technology to an experimental vehicle[2]. But this technology has a disadvantage of impairing the dynamics of the vehicle because it is based on imposing braking force on wheels of the vehicle. To solve this problem, Ikushima and Sawase proposed the concept of torque vectoring differential (TVD) in 1995[3]. Then the first vehicle with a TVD was produced by Sawase and Sano in 1999[4].

For a traditional open differential, the output torque from the engine is always split equally to the two driving wheels. But for a TVD, as Figure 1 shows, a torque will be transferred from the inner wheel to the outer one while the vehicle is cornering. Thus it allows better stability without impairing dynamics.

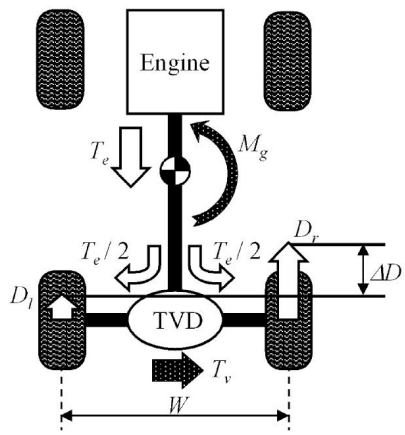


Figure 1 Torque transfer of a TVD

Torque vectoring differential can be classified into three kinds: clutch-type, brake-type and electric-powered. This thesis focuses on the clutch-type differential. Most clutch-type torque vectoring differentials are made of one or two clutches, differential gears, and some planetary gear sets. To classify and analyze the characteristics of them, velocity diagrams, which are often used to represent automatic transmissions, are also utilized here[5].

1.2 Classification of torque vectoring differentials

1.2.1 Definition of velocity diagram

To analyze characteristics of TVD and represent them clearly and briefly, velocity diagrams are used here, which is defined as follows. It is possible to consider a simple planetary gear set (PGS) with two degrees of freedom shown in Figure 2(a). It consists of three rotational elements, namely sun gear (SG), carrier (CA) and ring gear (RG). With given rotational speeds of two arbitrary elements, that of the other element can be found out.

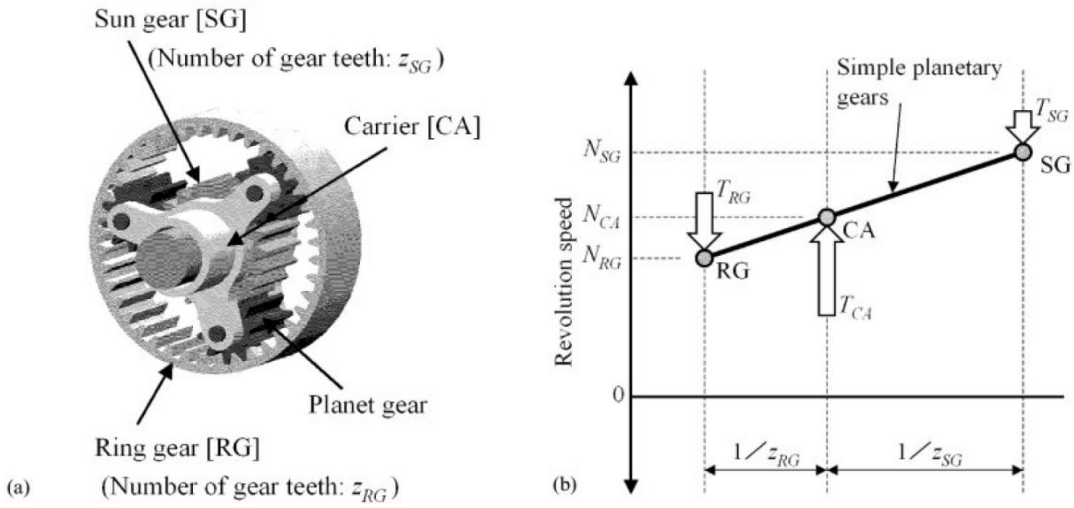


Figure 2 Simple PGS: (a) mechanism (b) velocity diagram

Figure 2(b) shows the velocity diagram of the PGS, where rotational speeds of three elements are represented by three points, and the vertical and horizontal axes represent rotational speed of elements and the ratio of rotational speed of each element to that of the carrier respectively. When CA is fixed, the rotating directions of RG and SG are always opposite, thus the points of RG and SG are placed on horizontally opposite sides with respect to the point of CA. Represent number of teeth and rotational speed of an element using N and Z respectively, we have:

$$\frac{N_{RG}}{N_{CA}} \propto \frac{Z_{CA}}{Z_{RG}} \propto \frac{1}{Z_{RG}} \quad (1)$$

So the horizontal distances between each element and carrier is proportional to the inverse of the number of teeth of each element, and thus it can be concluded that all the elements of a PGS is on a straight line which represents the PGS.

To indicate torques on these elements using T_{RG}, T_{CA} and T_{SG}, arrows are showed in **Figure 2**(b). It is possible to write down force equilibrium on both directions and moment equilibrium about point CA when all the elements are rotating at constant speeds:

$$T_{CA} = T_{SG} + T_{RG} \quad (2) \qquad T_{SG}\frac{1}{Z_{SG}} = T_{RG}\frac{1}{Z_{RG}} \quad (3)$$

Then torques on SG and RG can be calculated.

1.2.2 Velocity diagram of TVD

Similarly, the velocity diagram of a TVD, which consists of a differential gear unit represented by the lever i-o, a three-cluster gear unit consists of three main gears and a three-gang counter gear with a fixed carrier C and two sets of wet clutches C1, C2, can be drawn. Three main gears are connected to the differential case I and two hollow shafts HS1, HS2 respectively. And clutches C1, C2 can transfer torque between HS1 and outer output, HS2 and outer output respectively. Thus this TVD can be thought as a combination of two simple PGSs: PGS i-I-o and PGS C-HS2-I-HS1, which intersects at the point I. Besides, the numbers of all the gears of the three-cluster gear unit inside the differential are Z_{SL}, Z_{SC}, Z_{SR}, Z_{PL}, Z_{PC}, and Z_{PR} respectively, as shown in Figure 3.

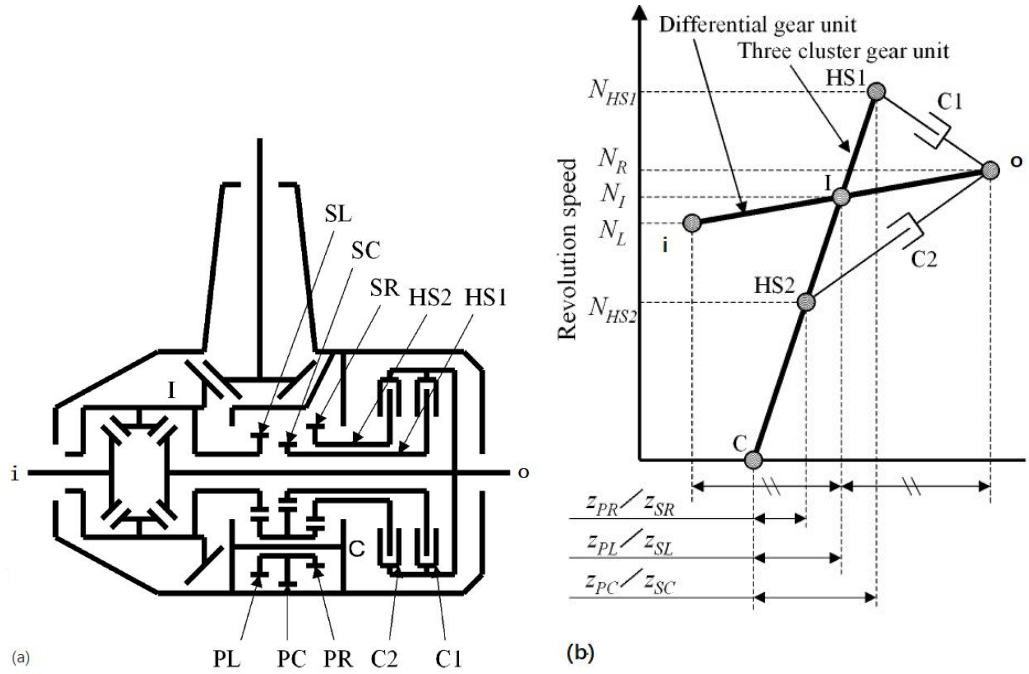


Figure 3 TVD: (a)schematic diagram (b)velocity diagram

And similarly, taking the velocity of differential input I as a reference, revolution speeds of all the elements and their ratios to the speed of I are represented on the velocity diagram.

1.2.3 Classification of TVD

To minimize the total clutch loss as well as for simplicity, here it is possible to consider a one-way TVD, which can only transfer torque from one side (namely the inner side) to the other side (the outer side) using one clutch while the vehicle is making a turn. This clutch-type TVD consists of four elements: inner output **i**, input **I**, and outer output **o**, which make up the differential, and speed increase element of the input, **HS1**.

Before classifying different TVD, an assumption of pure rolling motion of tires should be made: assume that a passenger car (PC) with an all-wheel drive mechanism (AWD) and a torque vectoring differential on the rear axle, for example, a Mitsubishi Lancer Evolution IV, is turning a corner with a low enough vehicle lateral acceleration, roughly lower than 0.2g (about 1.962 m/s^2), where g is the gravitational acceleration, so that the difference between speeds of two driving wheels increases as the lateral acceleration increases constantly, as shown in the left part of **Figure 4**. Because in this situation, the motions of tires are pure rolling without slipping relative to the ground. Thus the speed difference between driving wheels is just the function of rolling radius of the wheel with a lateral acceleration in the range 0-0.2g.

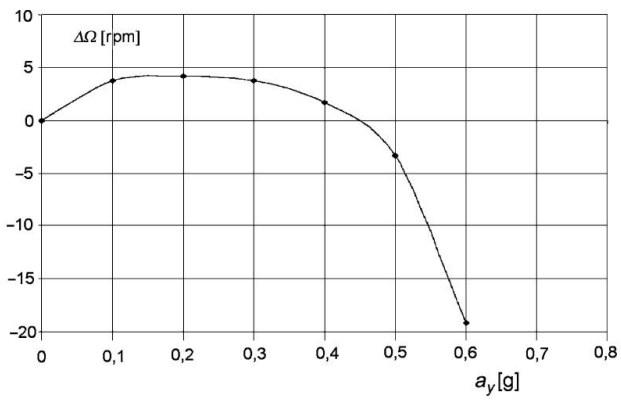


Figure 4 Speed difference between two driving wheels as a function of lateral acceleration

To name varieties of TVD, three letters connected by a hyphen are used here: the letter on the right-hand side of the hyphen means the torque receiver element which receives torque from the transceiver element, represented by the letter on the left-hand side of the hyphen, and another letter describes increase or decrease in speed using H or L respectively. Theoretically, there are 12 kinds of TVD, namely 12 combinations of different transceivers, receivers and transfer directions: there are 3 possibilities for the transceiver: L, I, R, and thus the receiver can be one of the remaining two choices. Additionally, the speed of HS1 can be increased or decreased.

However, the speed of the outer element connected by the clutch C1 must be higher than that of the inner element so that the torque can be transferred appropriately, then the number of different combinations is reduced to 6, listed as follows.

Type	Transceiver element	Receiver element	Speed variation
IH-O	I	R	High(increase)
OH-I	L	I	High
OH-O	L	R	High
O-IL	L	I	Low(decrease)
I-OL	I	R	Low
O-OL	L	R	Low

Table 1 All the possible types of a clutch-type TVD

To understand the mechanisms of all the types of TVD, the outputs of the differential, all the elements inside the differential as well as their rotational speeds, the torque transferred by the clutch and the power dissipated by the differential will be analyzed.

For example, the type IH-O is discussed as follows:

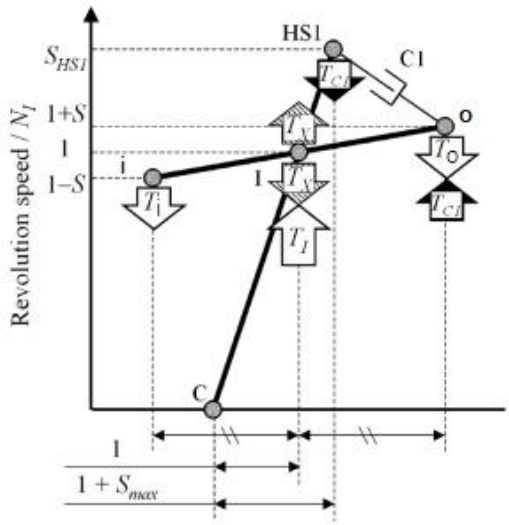


Figure 5 Velocity diagram of type IH-O TVD

As Figure 5 shows, input torque T_I of the differential is split into T_i on the inner output and T_o on the outer output. When the clutch C1 between HS1 and the outer output is activated, a clutch toque T_{C1} is transferred from HS1 to the outer output, and two reactive torques with the same amount of T_X appears on the bars i-o and C-HS1 respectively.

Under the above assumption, we have the relationship: $N_o - N_i = \frac{W}{R} N_I$ (4), where W represents the driving wheel track, R represents the rolling radius of driving wheels, and N is the rotational speed. Besides the mechanism of differential is: $N_I = \frac{N_o + N_i}{2}$ (5),

Thus we have the differential speed ratio: $S = \frac{N_o - N_i}{N_I} = \frac{W}{2R}$ (6)

More generally, rotational speed of each element can be represented in such a way: $S_i = \frac{N_i}{N_I}$

(7) where the character i means the element inside the differential.

A clutch-type TVD generates the differential speed of the clutch by a gear set with a fixed gear ratio. Once S_o becomes greater than S_{HS1}, the TVD transfers the torque to the opposite direction from that intended. To solve this problem, it is possible to design a gear ratio to make S_{HS1} large. However, by doing this, as the differential speed of slip clutch becomes large when a vehicle runs straight, another problem occurs: the heat caused by the clutch friction becomes large, too. Thus the minimum cornering radius that the TVD should function is determined. In the case that the revolution speeds of point R and HS1 are equal, the differential speed ratio S, is called the maximum acceptable differential speed ratio S_{max}[5]

From definition of S_{max} we have $S_{HS1} = S_o = S_{\max} + 1$ (8)

Thus by using S_{max} as a design parameter, various TVDs can be represented in the velocity diagram without considering the actual gear teeth numbers while the value of S_{max} can be calculated according to the differential structure and the velocity diagram. For example, by comparing the horizontal axes of Figure 3 and Figure 5, we have $1 + S_{\max} = \dfrac{Z_{PC} / Z_{SC}}{Z_{PL} / Z_{SL}}$ (9)

So, $S_{\max} = \dfrac{Z_{PC} Z_{SL}}{Z_{SC} Z_{PL}} - 1$

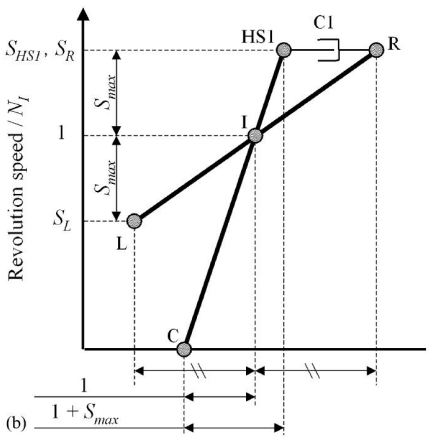


Figure 6 Velocity diagram of type IH-O TVD when $S=S_{max}$

Then for the bar i-o we have force equilibrium along vertical direction: $T_i + T_X + T_o = T_I + T_{C1}$ (10)

And moment equilibrium about point I: $T_i = T_o - T_{C1}$ (11)

For the bar C-HS1, moment equilibrium about point C: $T_X = T_{C1}(1 + S_{\max})$ (12)

$T_i + T_{C1}(1 + S_{\max}) + T_o = T_i + T_{C1}$ (13) $T_i + T_{C1}(1 + S_{\max}) + T_o = T_I + T_o - T_i$ (14)

Then the torques on both driving wheels are: $T_i = \dfrac{T_I}{2} - \dfrac{1 + S_{\max}}{2} T_{C1}$

(15)

$$T_o = \frac{T_I}{2} + \frac{1 - S_{\max}}{2} T_{C1}$$

(16)

And the difference between two torques is:

$$\Delta T = T_o - T_i = T_{C1} \quad (17)$$

From the similarity of triangle, we have: $\frac{S_{HS1}}{1} = \frac{1 + S_{\max}}{1}$ (18)

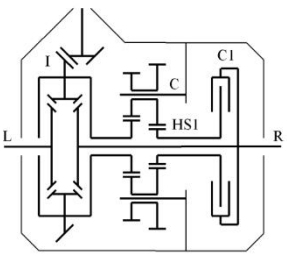
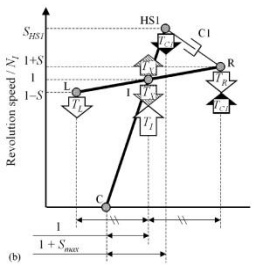
Speed difference between two elements connected by the clutch C1:

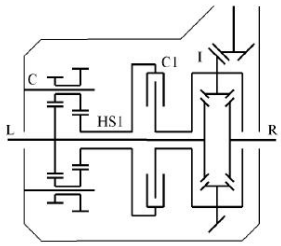
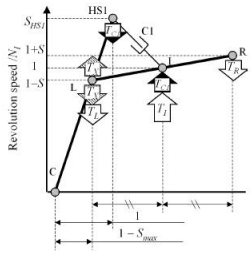
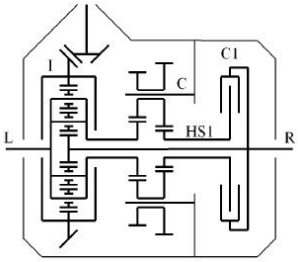
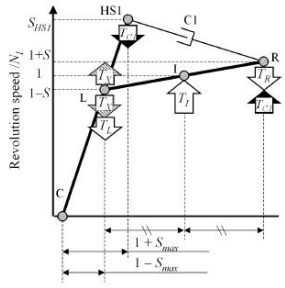
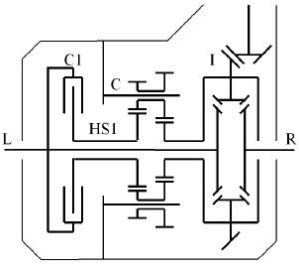
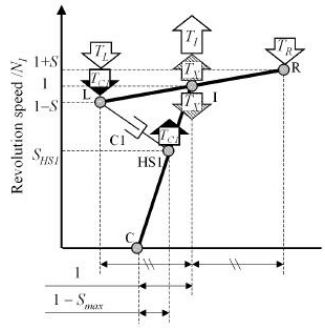
$$\Delta S_C = S_{HS1} - S_R = (1 + S_{\max}) - (1 + S) = S_{\max} - S \quad (19)$$

Thus the power dissipated by the differential can be calculated:

$$P_{LOSS} = T_C \Delta S_C N_I = \Delta T (S_{\max} - S) N_I \quad (20)$$

Similarly, with an AWD Mitsubishi Evolution IV that is turning a corner with a lateral acceleration lower than 0.2g, as assumed above, all the 6 possibilities of a clutch-type TVD and indices related to them are calculated using the same procedure and listed as follows.

IH-O			$T_L = \frac{T_I}{2} - \frac{1 + S_{\max}}{2} T_{C1}$ $T_R = \frac{T_I}{2} + \frac{1 - S_{\max}}{2} T_{C1}$ $T_{C1} = \Delta T$ $\Delta S_C = S_{\max} - S$ $P_{LOSS} = N_I (S_{\max} - S) \Delta T$ $S_{\max} = \frac{Z_{PC} Z_{SL}}{Z_{SC} Z_{PL}} - 1$

OH-I			$T_L = \frac{T_I}{2} - \frac{1+S_{\max}}{2(1-S_{\max})} T_{C1}$ $T_R = \frac{T_I}{2} + \frac{T_{C1}}{2}$ $T_{C1} = (1-S_{\max})\Delta T$ $\Delta S_C = \frac{S_{\max} - S}{1-S_{\max}}$ $P_{LOSS} = N_I(S_{\max} - S)\Delta T$ $S_{\max} = 1 - \frac{Z_{PR}Z_{SL}}{Z_{SR}Z_{PL}}$
OH-O			$T_L = \frac{T_I}{2} - \frac{1+S_{\max}}{1-S_{\max}} T_{C1}$ $T_R = \frac{T_I}{2} + T_{C1}$ $T_{C1} = \frac{1-S_{\max}}{2}\Delta T$ $\Delta S_C = \frac{2(S_{\max} - S)}{1-S_{\max}}$ $P_{LOSS} = N_I(S_{\max} - S)\Delta T$ $S_{\max} = \frac{Z_{PC}Z_{SL} - Z_{PL}Z_{SC}}{Z_{PC}Z_{SL} + Z_{PL}Z_{SC}}$
O-IL			$T_L = \frac{T_I}{2} - \frac{1+S_{\max}}{2} T_{C1}$ $T_R = \frac{T_I}{2} + \frac{1\text{-}S_{\max}}{2} T_{C1}$ $T_{C1} = \Delta T$ $\Delta S_C = S_{\max} - S$ $P_{LOSS} = N_I(S_{\max} - S)\Delta T$ $S_{\max} = 1 - \frac{Z_{PR}Z_{SL}}{Z_{SR}Z_{PL}}$

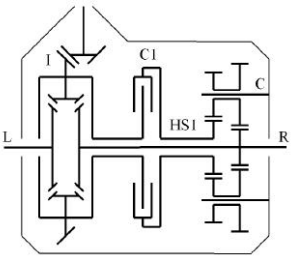
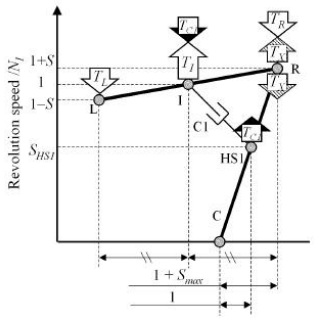
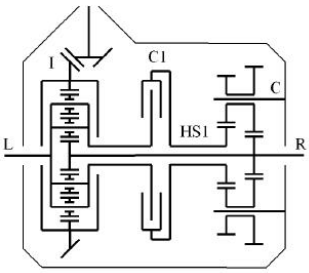
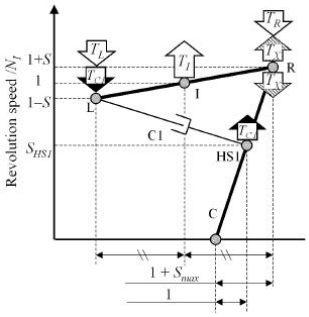
I-OL			$T_L = \frac{T_I}{2} - \frac{T_{C1}}{2}$ $T_R = \frac{T_I}{2} + \frac{1-S_{\max}}{2(1+S_{\max})} T_{C1}$ $T_{C1} = \Delta T(1+S_{\max})$ $\Delta S_C = \frac{S_{\max} - S}{1+S_{\max}}$ $P_{LOSS} = N_I(S_{\max} - S)\Delta T$ $S_{\max} = \frac{Z_{PC}Z_{SL}}{Z_{SC}Z_{PL}} - 1$
O-OL			$T_L = \frac{T_I}{2} - T_{C1}$ $T_R = \frac{T_I}{2} + \frac{1-S_{\max}}{1+S_{\max}} T_{C1}$ $T_{C1} = \frac{1+S_{\max}}{2}\Delta T$ $\Delta S_C = \frac{2(S_{\max} - S)}{1+S_{\max}}$ $P_{LOSS} = N_I(S_{\max} - S)\Delta T$ $S_{\max} = \frac{Z_{PC}Z_{SL} - Z_{PL}Z_{SC}}{Z_{PC}Z_{SL} + Z_{PL}Z_{SC}}$

Table 2 Clutch-type TVDs and their characteristics

1.3 Summary

From the table it can be concluded that for all the types of TVD, the speed difference between elements connected by the clutch: $\Delta S_C \propto \frac{S_{\max} - S}{1+S_{\max}}$ (21)

The torque transferred from the inner wheel to the outer one is proportional to the clutch torque:

$\Delta T \propto T_{C1}$ (22)

The power dissipated by TVD is: $P_{LOSS} = N_I(S_{\max} - S)\Delta T$ (23)

Frictional torque transferred by the clutch is controlled by the axial force F_a between the clutch discs, while its sign is imposed by the speed difference between the inner and outer clutch discs, with the clutch fiction surface radius r_c, number of active friction surfaces n_c and friction coefficient of the friction surface μ_F, friction torque is $T_{C1} = F_a r_c n_c \mu_F = T_{req}\mu_F$ (24), T_{req} is the hydraulic torque acting on the clutch.

And then, $P_{LOSS} \propto (S_{\max} - S)F_a$ (25)

In conclusion, as the cornering radius R increases, differential speed ratio S decreases, power dissipated by TVD will increase.

As all the possibilities of one-way TVD are listed above, a complete TVD should have another clutch to achieve torque transfers in both directions. For example, the complete form of the IH-O type should be the structure shown in Fig3(a), which is the first TVD in the world made by Mitsubishi and applied to Mitsubishi Lancer generation IV. Similarly, all the possibilities mentioned above can be extended to the corresponding complete structure by adding the other clutch. This Mitsubishi one will be analyzed further and modelled in the following.

Following the same procedures applied to type IH-O mentioned above, parameters related to another clutch C2 in the case of turning to the opposite direction with respect to the previous one can be calculated.

$$T_R = \frac{T_I}{2} - \frac{1+S_{\max}'}{2}T_{C2} \quad (26) \qquad T_L = \frac{T_I}{2} + \frac{1-S_{\max}'}{2}T_{C2} \quad (27) \qquad T_{C2} = \Delta T \quad (28)$$

$$\Delta S_C = S_{\max}' - S \quad (29)$$

$$P_{LOSS} = N_I(S_{\max}' - S)\Delta T \quad (30) \qquad S_{\max}' = 1 - \frac{Z_{PR}Z_{SL}}{Z_{SR}Z_{PL}} \quad (31)$$

In conclusion, all the parameters related to the Mitsubishi AYC (type IH-O TVD) is listed as follows

	Case 1	Case 2
Inner wheel	$T_L = \frac{T_I}{2} - \frac{1+S_{\max}}{2}T_{C1}$	$T_R = \frac{T_I}{2} - \frac{1+S_{\max}'}{2}T_{C2}$
Outer wheel	$T_R = \frac{T_I}{2} + \frac{1-S_{\max}}{2}T_{C1}$	$T_L = \frac{T_I}{2} + \frac{1-S_{\max}'}{2}T_{C2}$
ΔS_C	$S_{\max} - S$	$S_{\max}' - S$

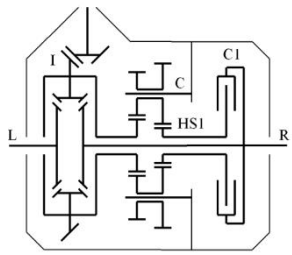
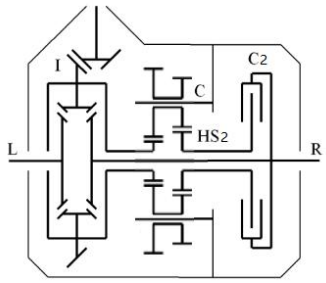
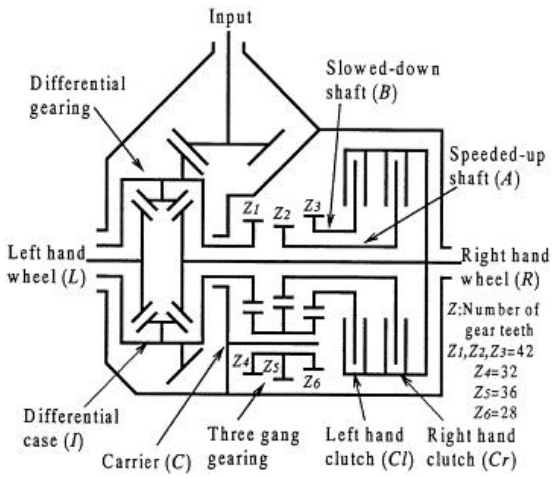
P_{LOSS}	$N_I(S_{\max}-S)\Delta T$	$N_I(S_{\max}'-S)\Delta T$
Layout		
Complete layout (combination of cases 1 and 2)		

Table 3 Characteristics of Mitsubishi AYC (type IH-O) differential

With all the numbers of gears given by [10]:

$Z_{SL}=Z_{SC}=Z_{SR}=42$ $Z_{PL}=32$ $Z_{PC}=36$ $Z_{PR}=28$

It's clear to see that $S_{max}=S_{max}'=0.125$, which corresponds to the condition $0.1<S_{max}<0.2$ mentioned previously. Thus this value is reasonable.

2 Introduction to some control strategies

In this part, several control strategies that were used to improve vehicle handling and traction performance in other papers will be introduced.

2.1 TVD actuator control

As Figure 7 shows, this control strategy was based on estimation of yaw rate using a bicycle model[6].

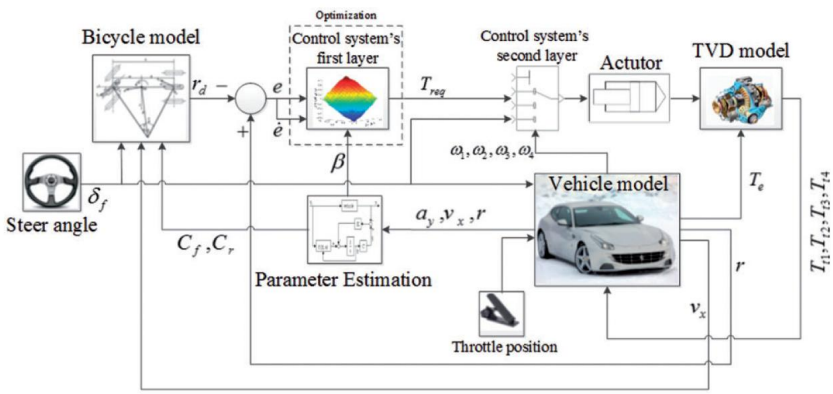


Figure 7 Schematic diagram of S.Jafaari & K.Shirazi's control system

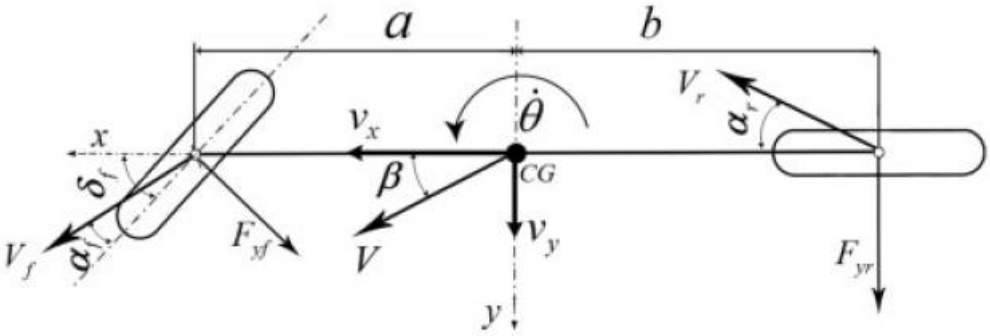


Figure 8 Bicycle model

2.2 Integrated vehicle dynamics control via TVD and ESC

In this paper, TVD is used to improve handling performance only under nearly normal and steady-state driving conditions, and ESC controller is activated under emergency conditions or when the TVD can't work normally, for example, when the friction coefficient between wheels and ground is very low. The selection and activation of the actuator(TVD & ESC) and estimation of parameters are performed by a three-layer system[7].

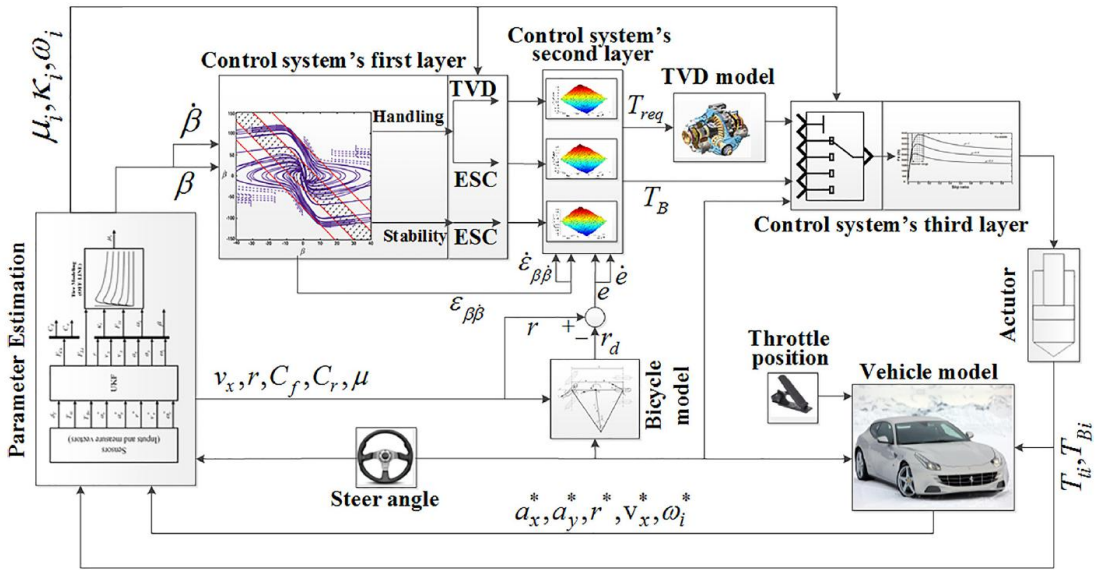


Figure 9 Schematic diagram of integrated control system

2.3 Hardware-in-Loop control

This hardware-in-loop system (HILS) is divided into software block, which mainly focuses on the controller of TVD using Simulink, and hardware block[8]. Two kinds of verification operations are introduced: double lane change maneuver(DLCM) and slalom maneuver(SM). In DLCM, the vehicle accelerates to 120km/h, then it turns to another lane and runs for a period before returning the first lane. In SM, the vehicle accelerates to 75km/h, then follows a sinusoidal path through several cones,which are placed with a certain distance.

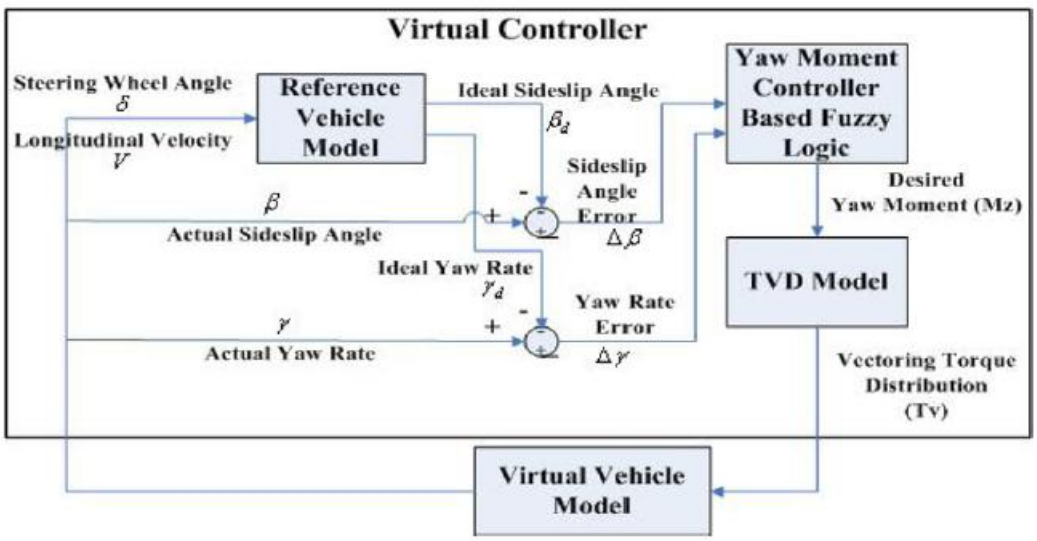
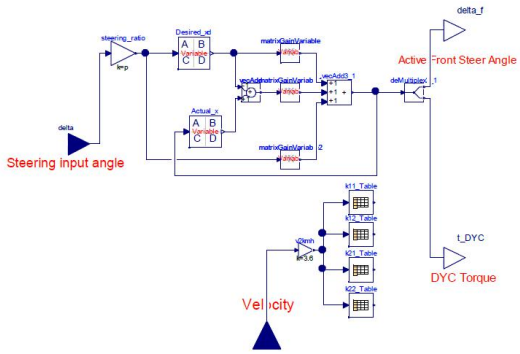


Figure 10 Structure of the virtual controller of the HILS

2.4 Integrated control of Front Steering and TVD

This control system was applied to an electric vehicle (EV) with both an active front steering (AFS) and a TVD to obtain the desired vehicle dynamics [9]. It is based on a single track model, a full vehicle model and two controllers: a PI controller for longitudinal dynamics and a LQR controller for lateral dynamics.



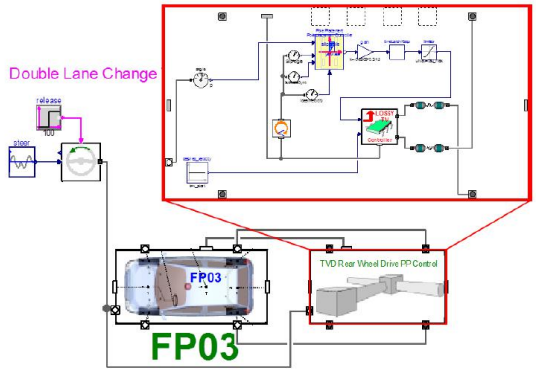


Figure 11 Modelica model of the controller Figure 12 Structure of full vehicle model

Comparison among different control strategies

	Pros	Cons
Actuator control	Great performance in handling without reduction in longitudinal acceleration or speed	Low stability in terms of critical and emergency situations
Integrated control with ESC	Improved performance in lower and different friction coefficient conditions and critical situations	More complex and expensive structure
HIL control	Improved handling and stability, low energy loss, better comfort	More complex and expensive structure
Integrated control with AFS	Better stability and improved performance	Limited actuator range of AFS

Table 4 Comparison among different control strategies

3 Construction of TVD model and control strategy

To analyse vehicle behaviors easily, it's possible to use a software to simulate different tests of

vehicles. In this thesis, IPG CarMaker for Simulink is used to perform this function.

IPG CarMaker is a software developed specifically for testing passenger cars and light-duty vehicles. With this software, it's possible to accurately model real-world test scenarios, including the entire surrounding environment. CarMaker is an open integration and test platform and can be applied throughout the entire development process: from model-to software-to hardware-to vehicle-in-the-loop[12].

CarMaker for Simulink integrates IPG CarMaker entirely into MATLAB/Simulink. The highly optimized and robust features of CarMaker were added to the Simulink environment using an S-function implementation and the API functions that are provided by MATLAB/Simulink. CarMaker for Simulink is a closely linked combination of two applications, resulting in a simulation environment that has both good performance and stability [13].

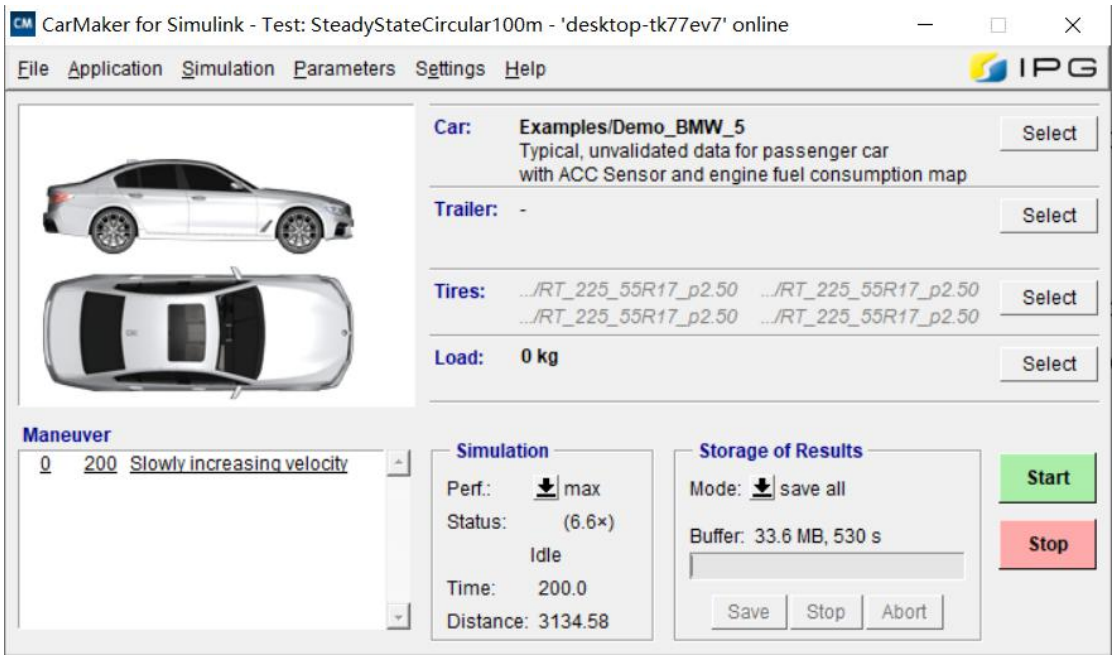


Figure 13 Main interface of CarMaker

With CarMaker for Simulink, all the following procedures are performed with choosing BWM 5 as the test vehicle in CarMaker and related vehicle data are those for BWM 5. Information related to this model is attached into the appendix.

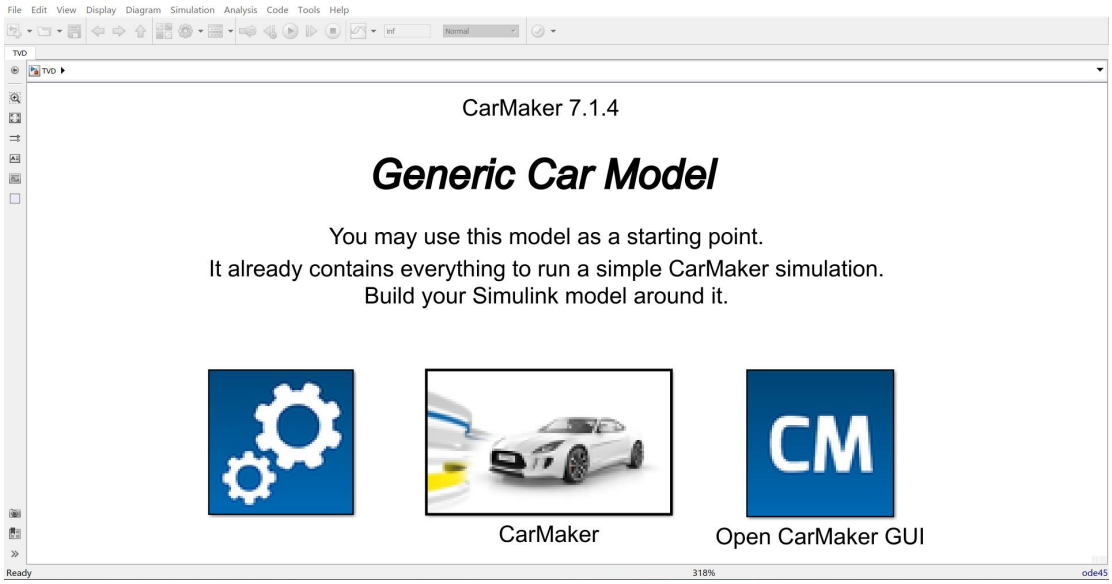


Figure 14 CarMaker for Simulink main interface

Into the 'CarMaker' block, several blocks which construct a whole vehicle model can be seen. In this thesis, we focus on the 'PowerTrain' block in 'IPG Vehicle' block.

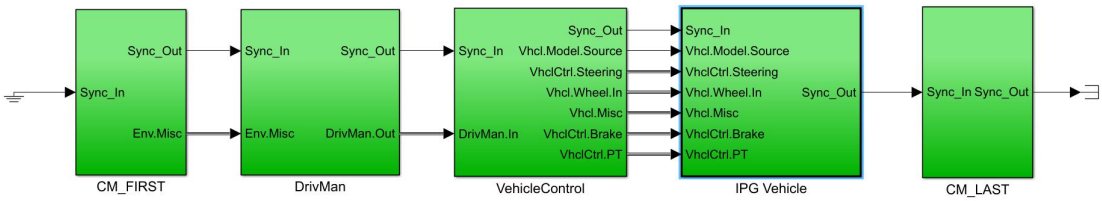


Figure 15 Configuration of CarMaker block

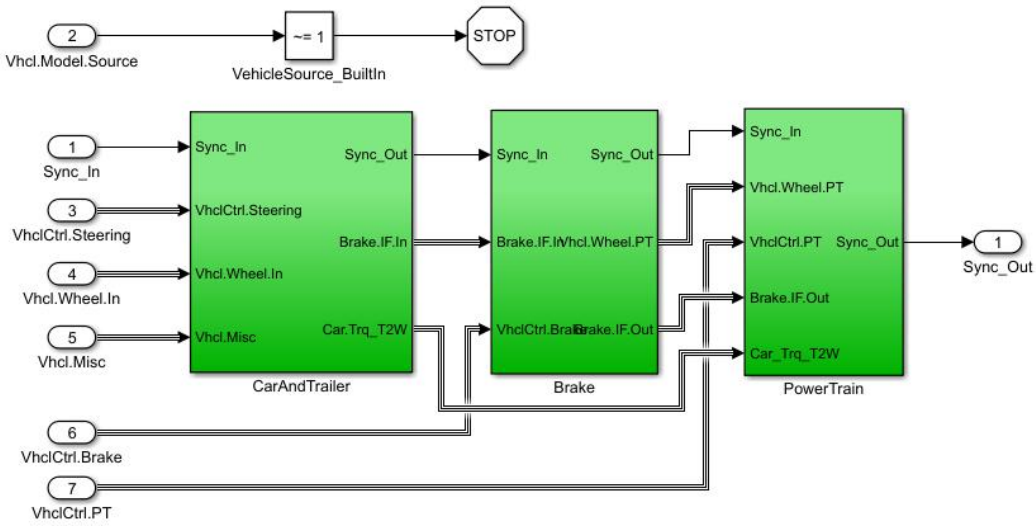


Figure 16 Configuration of ‘IPG Vehicle’ block

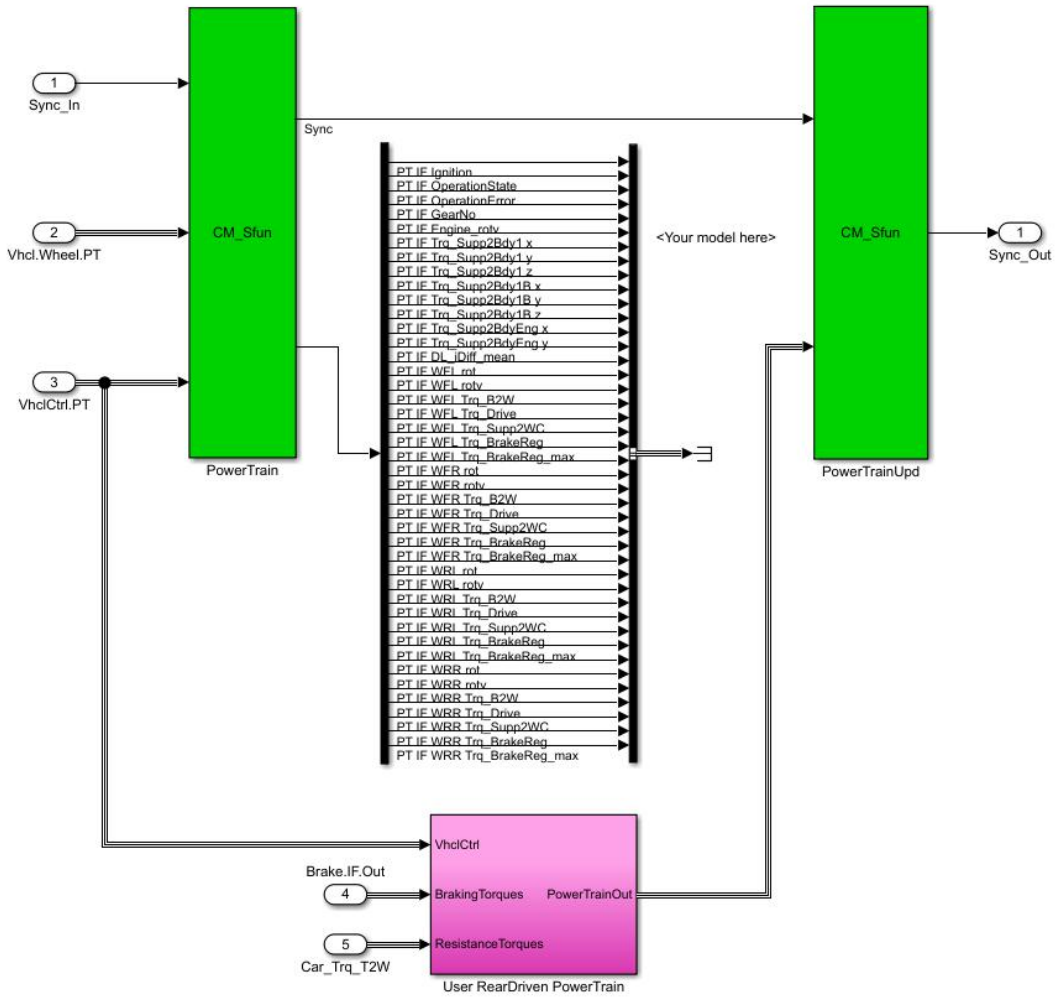


Figure 17 Configuration of ‘PowerTrain’ block

Instead of using the original PowerTrain block, a user defined rear-driven powertrain block is applied to implement the torque vectoring action with disabling the original powertrain (outputs of the original powertrain are connected to a terminator).

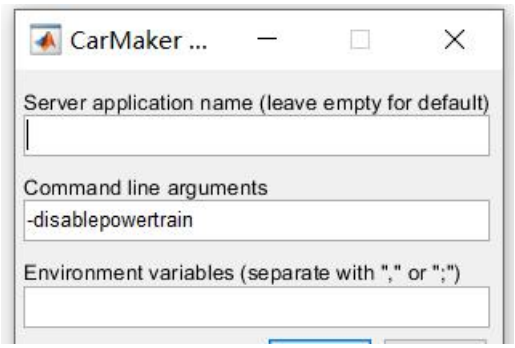


Figure 18 CarMaker model configuration

3.1 Overview of the system

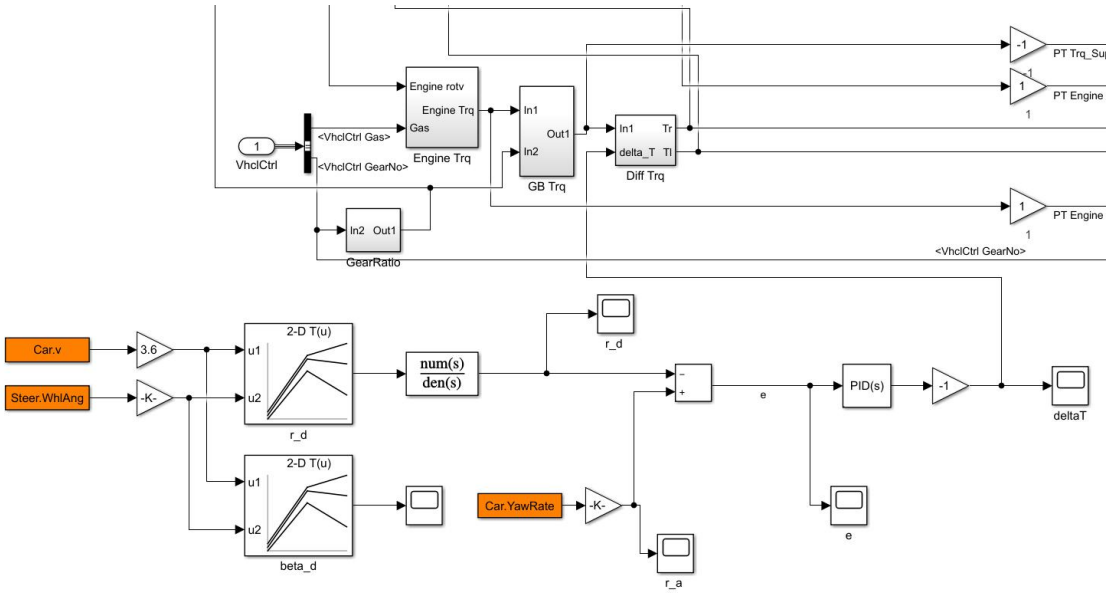


Figure 19 Overview of user defined powertrain

The left part of the model is used to generate a reference yaw rate using a look-up table. The reference yaw rate is compared with the actual yaw rate which can be read through the sensor, then the difference between two yaw rates is inputted into a PID controller to get a desired torque difference ΔT between two driving wheels, namely the clutch torque of the differential. This output ΔT may be positive or negative according to the sign of error. When e>0, it means that the actual yaw rate is higher than the desired one, it's necessary to decrease the difference between driving torques on two driving wheels other than amplify it. While e<0, the actual yaw rate is smaller than the desired one, so it's necessary to increase the difference between torques on driving wheels. From the above, a block for reversing the sign of ΔT is necessary.

Actually, it's impossible to impose a negative torque on the clutch. In our case (Mitsubishi AYC), there are two clutches in the torque vectoring differential cage so that torque could be transferred in both directions. And we are always talking about the outer wheel. When the value of ΔT is positive, a torque is transferred from inner wheel to the outer one. Once ΔT becomes negative, it means a torque with value equal to the absolute value of ΔT is transferred from the outer wheel to the inner one.

Besides ΔT, another two inputs of the TV block (inside the block 'Diff Trq') are rotational speeds of two driving wheels which are read through sensors.

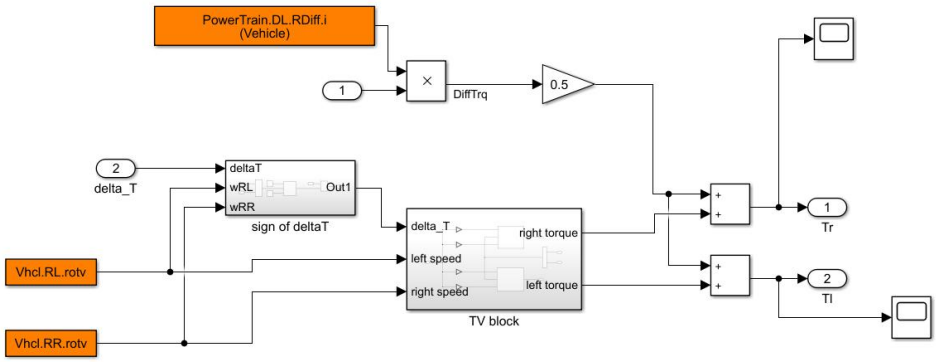


Figure 20 Configuration of block 'Diff Trq'

Output of PID controller ΔT should be applied another sign block which is determined by the relationship between two driving wheel speeds. If $\omega_{RR}>\omega_{RL}$, the sign of torque difference should be the same as that of ΔT, while $\omega_{RL}>\omega_{RR}$, the sign of clutch torque should be opposite with that of ΔT.

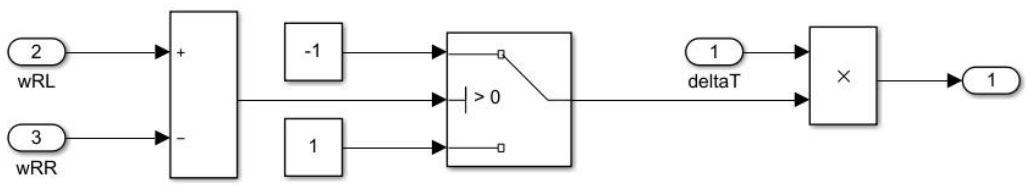


Figure 21 Determination of sign of torque difference

The upper left part in Figure5 is used to calculate the input torque of differential: multiplication of transmission ratio from differential input shaft to cage (through 'Read CM Parameter' block) and torque output from gearbox. Then this torque is split equally to two driving shafts if an open differential is mounted on the vehicle. In this case, split torques are accumulated with two additional torques from TV block, and then the torques are applied to driving wheels respectively.

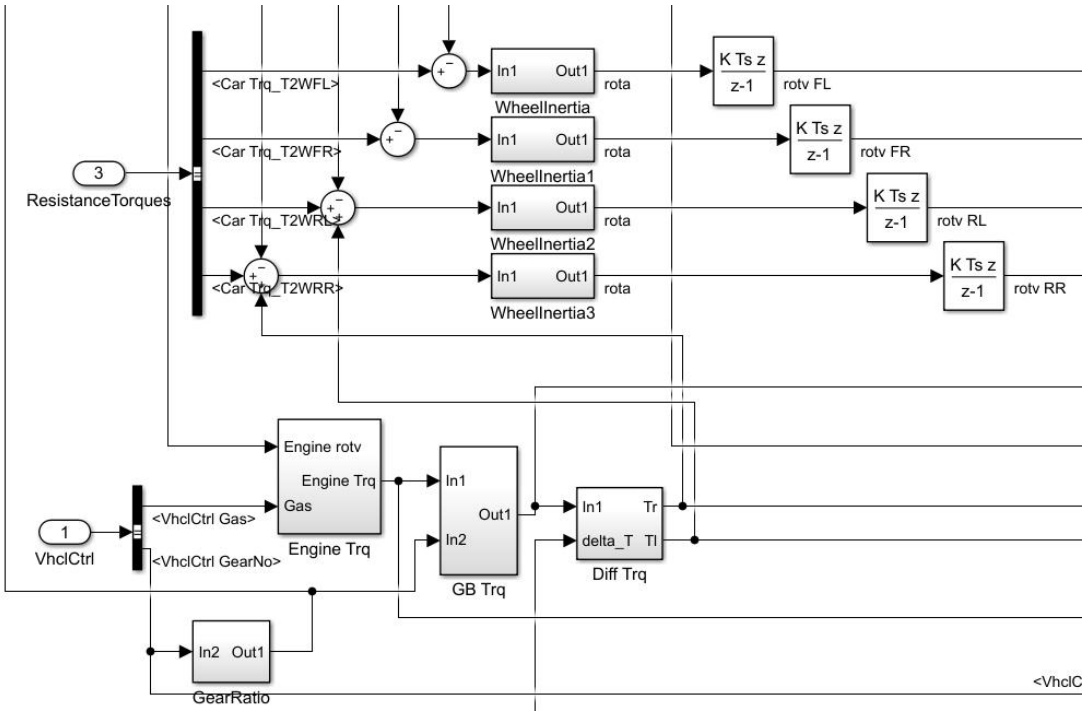


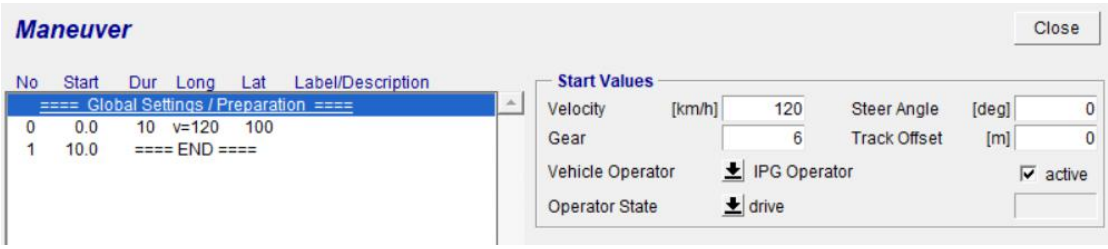
Figure 22 Application of torques

Then accumulated torques outputted from 'Diff Trq' block are applied to driving wheels to get corresponding wheel speeds.

3.2 Construction of look-up tables

To provide a reference yaw rate for the control system, a 2D look-up table with two inputs: vehicle speed and steering wheel angle and an output: yaw rate, is required.

To generate required data, several ramp steer maneuvers with vehicle speed equal to 20km/h, 40km/h, 60km/h, 80km/h, 100km/h and 120km/h are simulated, as shown below.



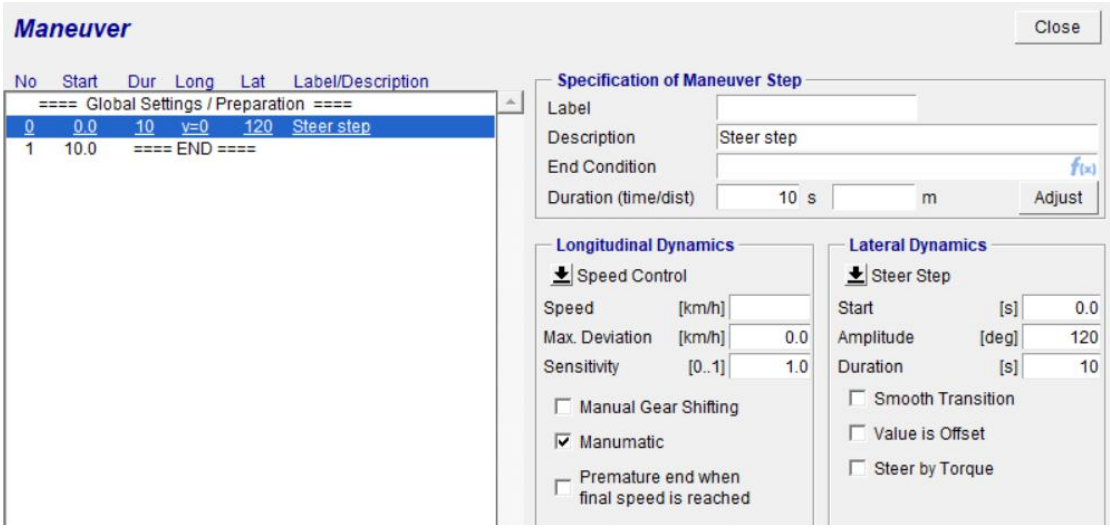


Figure 23 Ramp steer maneuver

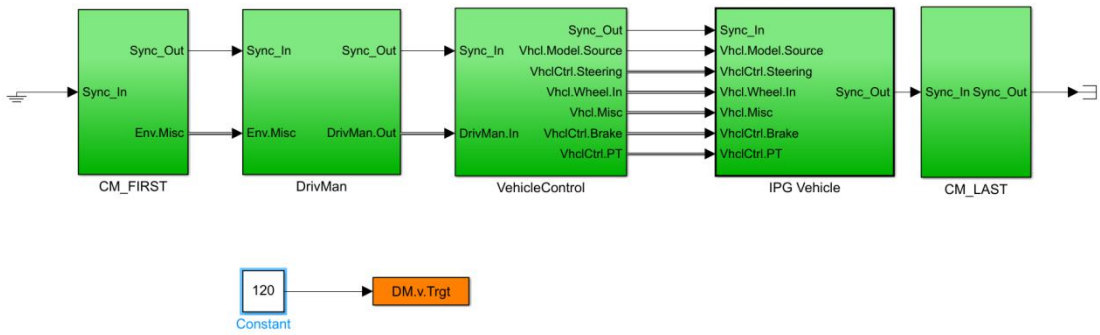


Figure 24 Setting of testing speed

Results: Time histories of steering wheel angle, speed, side slip angle, yaw rate, lateral acceleration, engine torque, engine speed, gear number of tires are shown below.

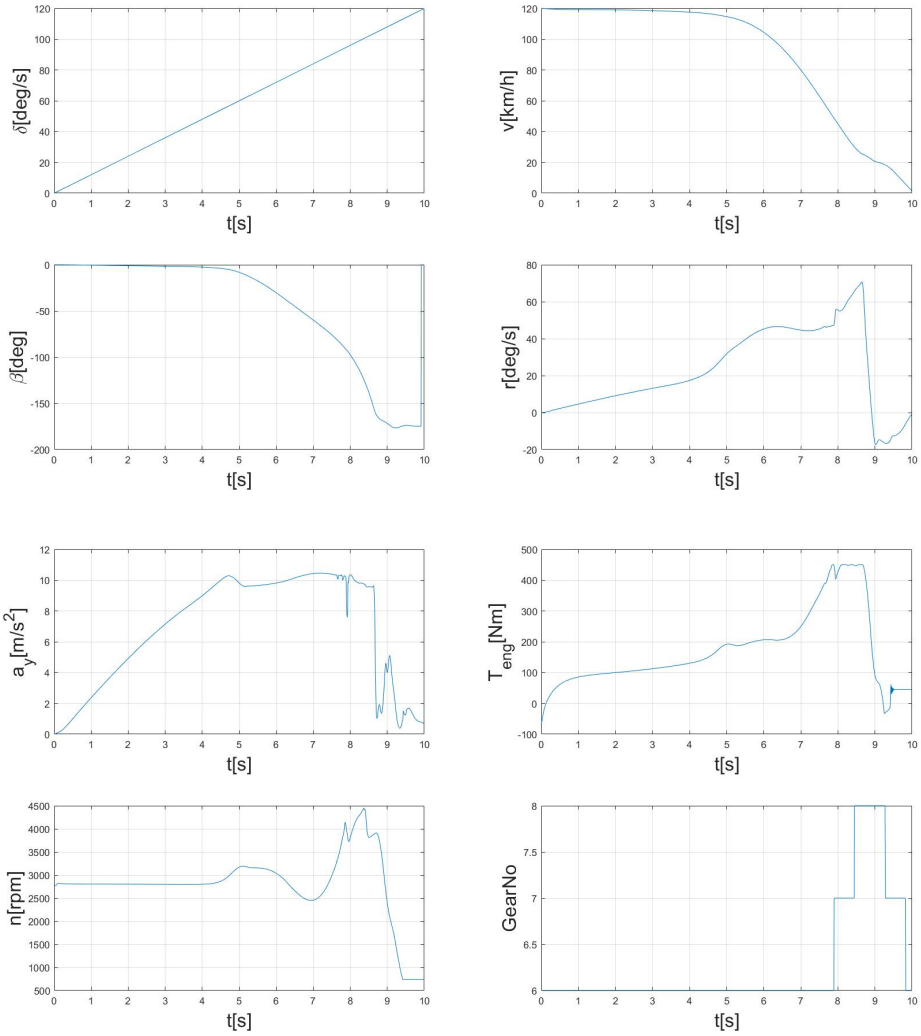
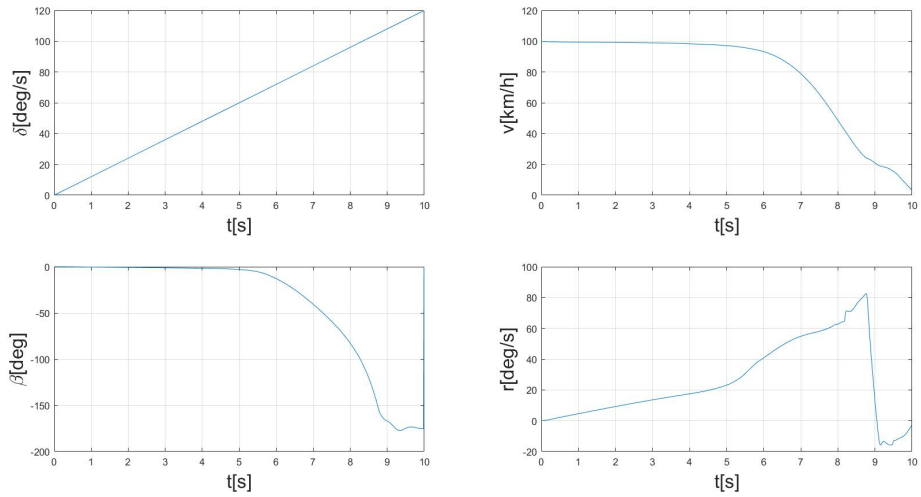


Figure 25 v=120km/h



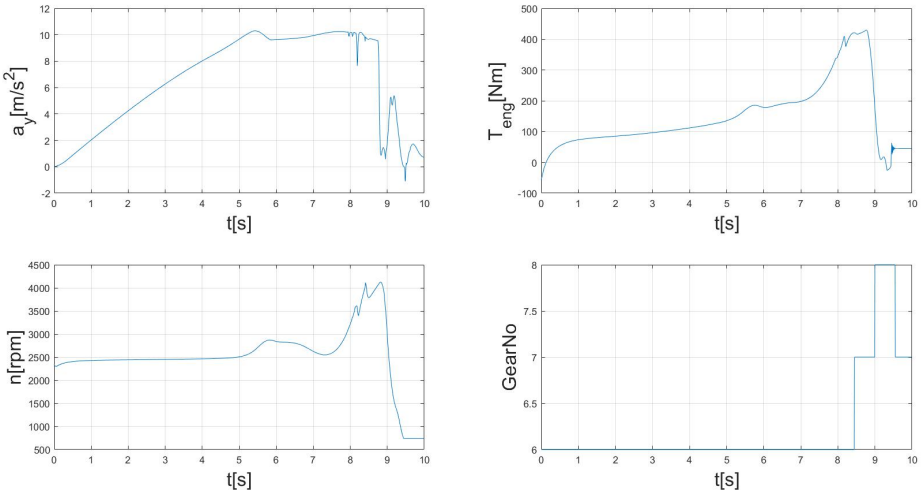


Figure 26 v=100km/h

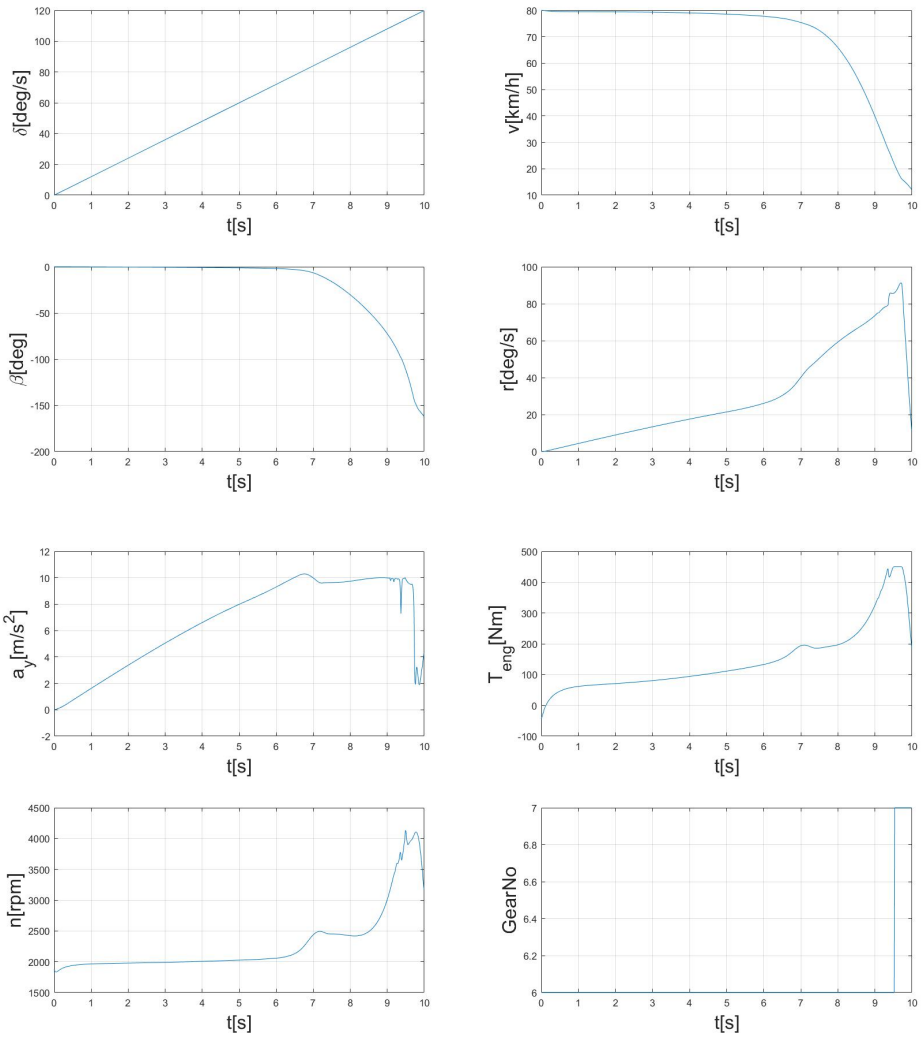


Figure 27 v=80km/h

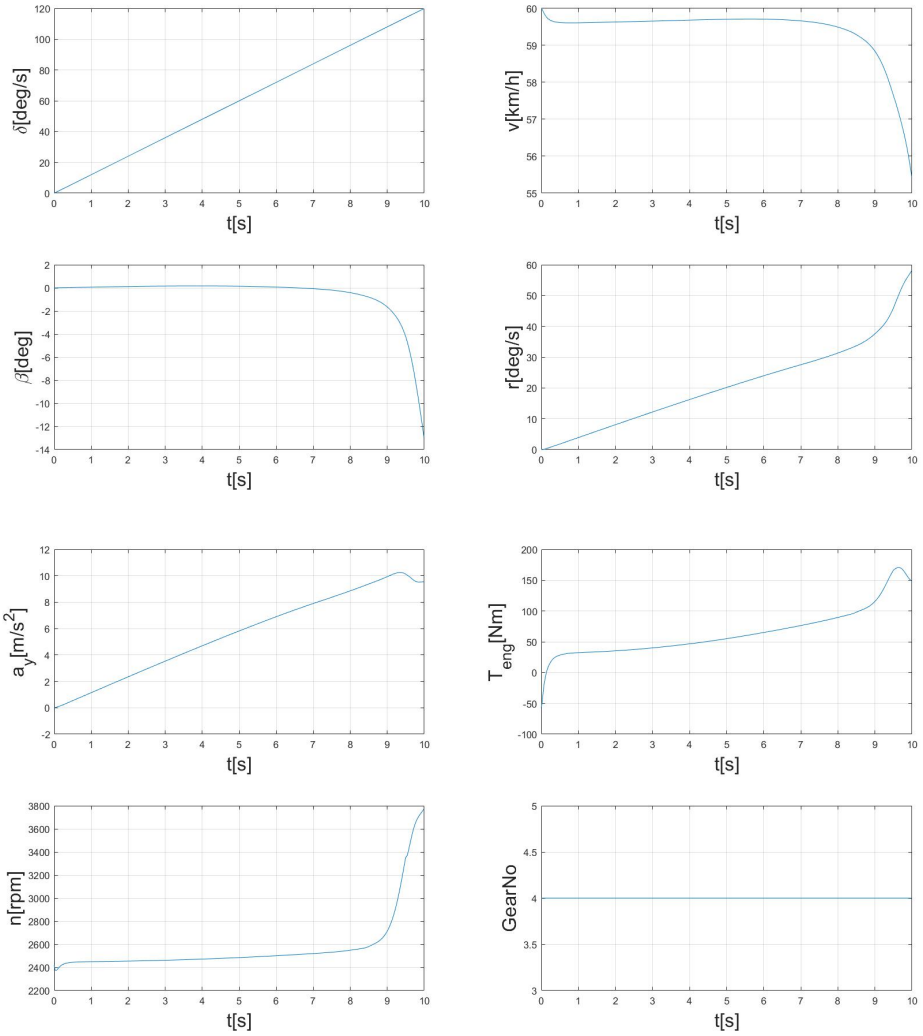
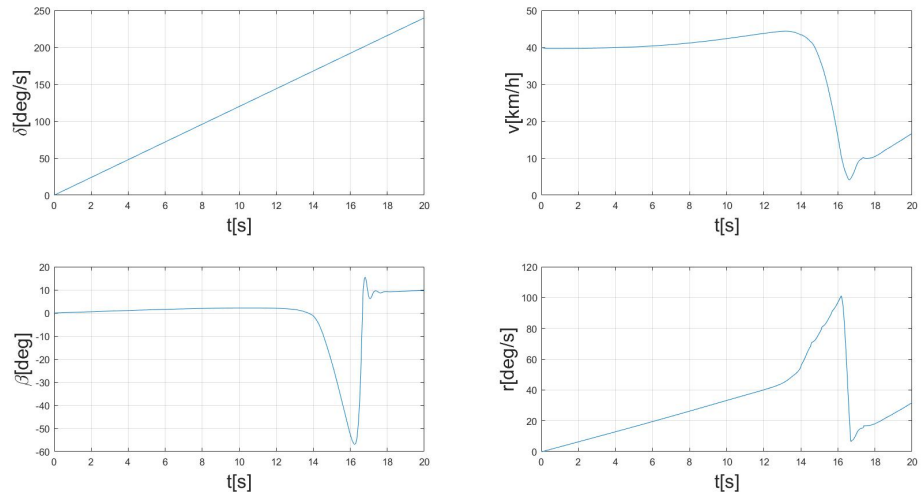


Figure 28 v=60km/h



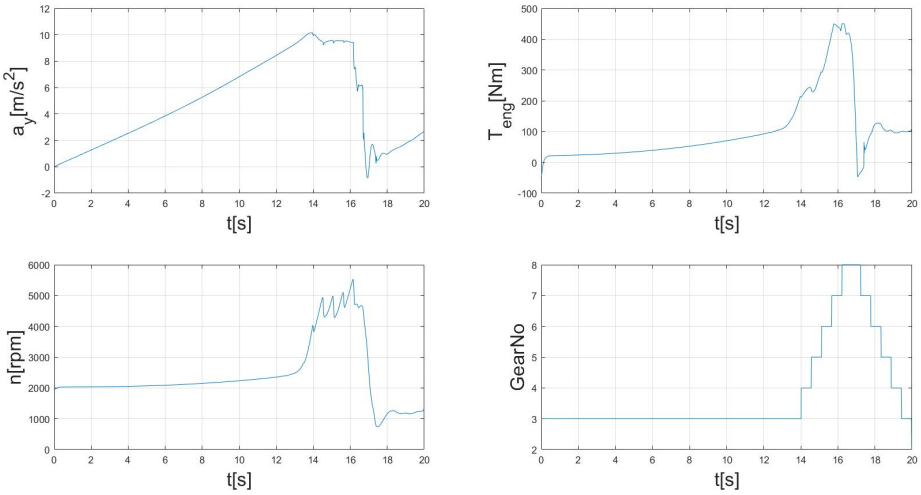


Figure 29 v=40km/h

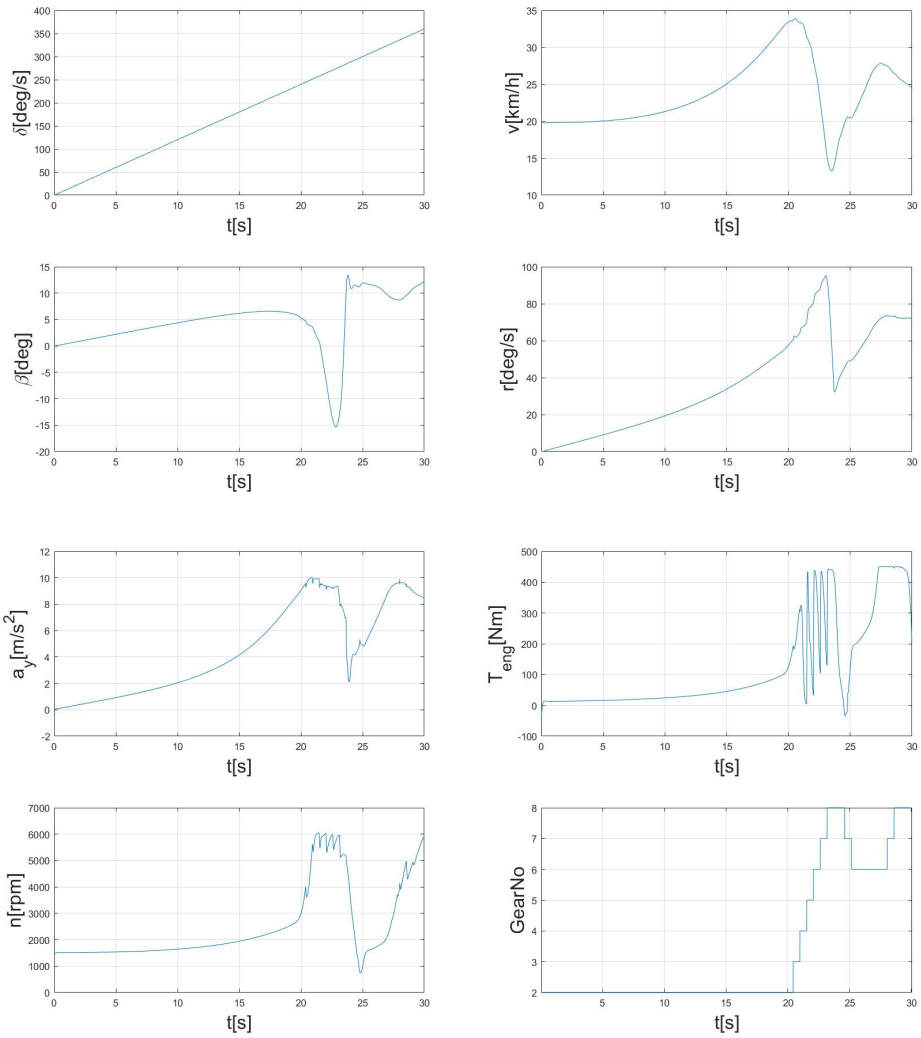


Figure 30 v=20km/h

It can be seen that for each of the simulations, after a specific period, one of the tires reaches the saturation state, the vehicle speed is no longer constant and the lateral acceleration begins to fluctuate. By focusing on the periods where gear ratio and vehicle speed are constant and lateral acceleration increases with steering wheel angle, it's possible to get a plot to show relations between steering wheel angle and yaw rate for different speeds.

To construct a look-up table, dimensions of arrays of yaw rates for different speed should be the same. It's possible to construct a matrix with 6 rows and 1914 columns: where for v=40km/h-120km/h, the data after saturation of tires are reached for each speed equal to the last data before saturation (here it's supposed that the reference yaw rate keeps constant after lateral acceleration exceeds 8m/s^2). Then the curves for look-up table become:

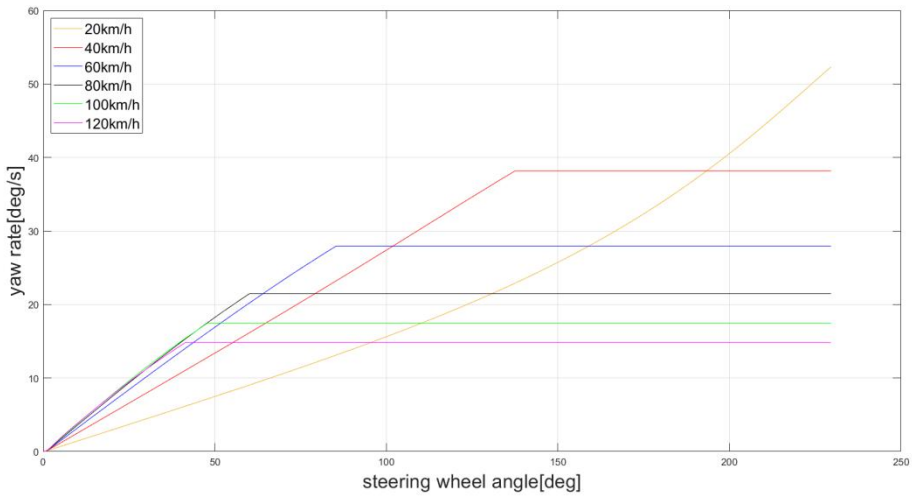


Figure 31 Curves for look-up table

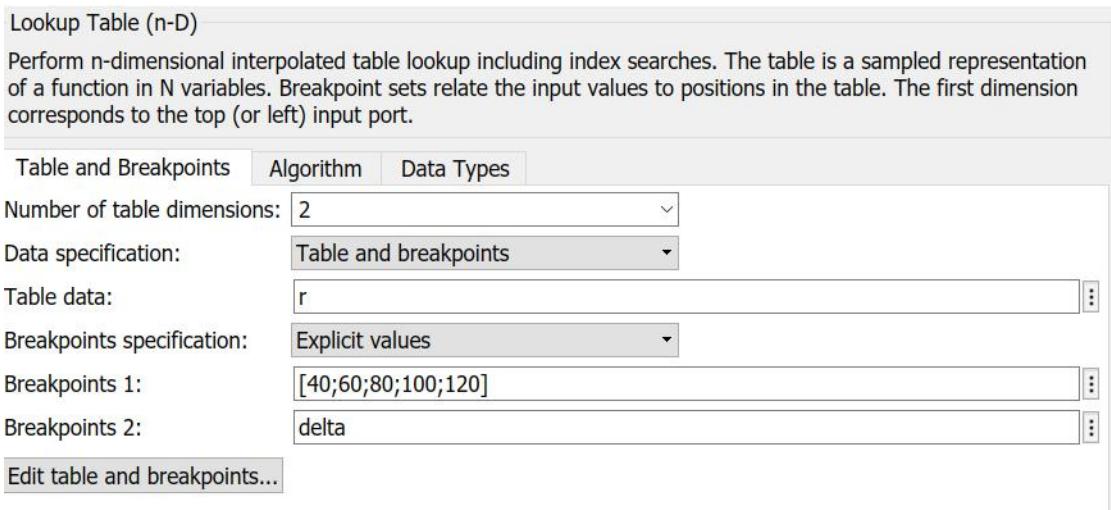


Figure 32 Construction of 2D look-up table for yaw rate

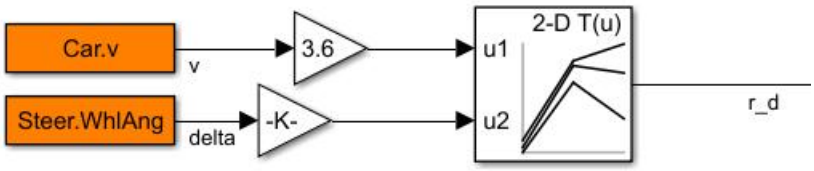


Figure 33 Configuration of 2D Look-up table for yaw rate

Here the unit of vehicle speed inputted to the look-up table is km/h, while that in CarMaker is m/s, so a gain of 3.6 is required. And the unit of steering wheel angle is deg while that in CarMaker is rad, so a gain of 180/π is required, which is also necessary for the real-time yaw rate (rad/s to deg/s).

A similar look-up table for side slip angle, which is used to monitor how far it is from steady-state conditions, is constructed as follows:

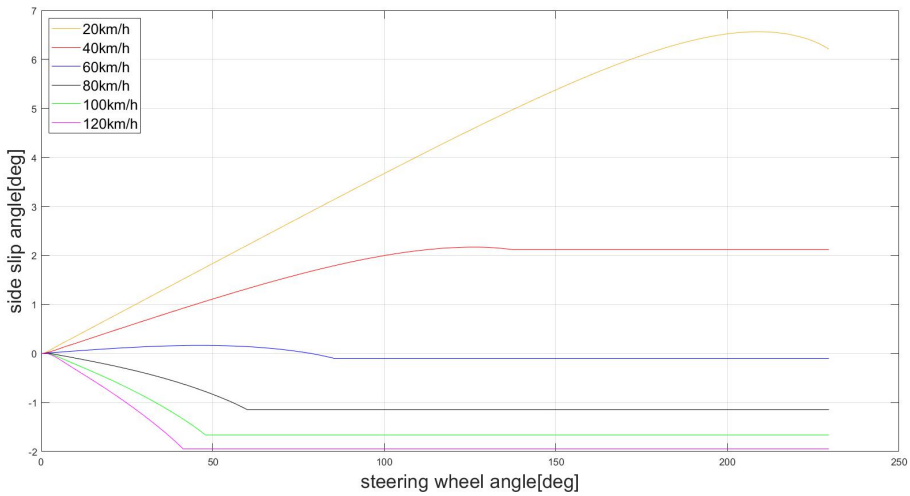


Figure 34 Curves for look-up table for side slip angle

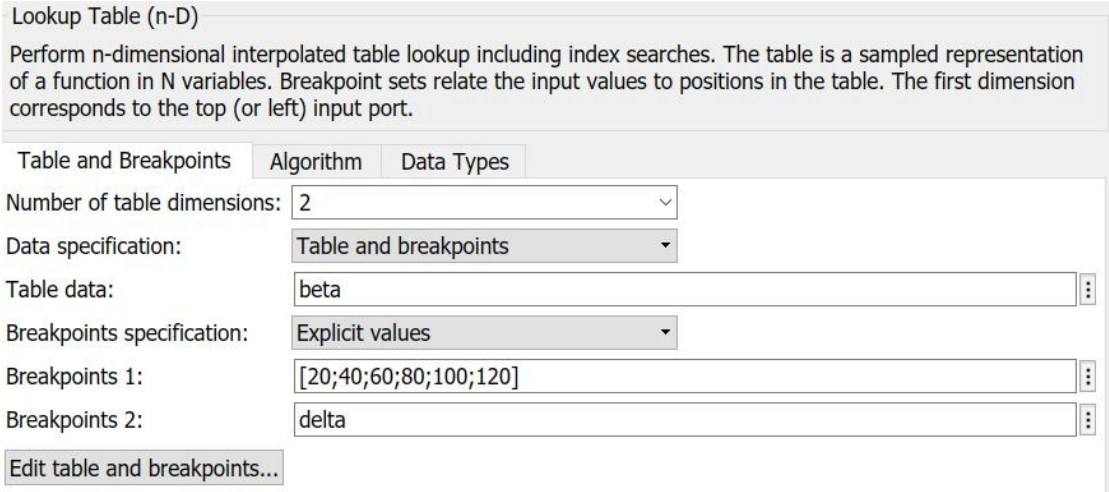


Figure 35 Construction of look-up table for side slip angle

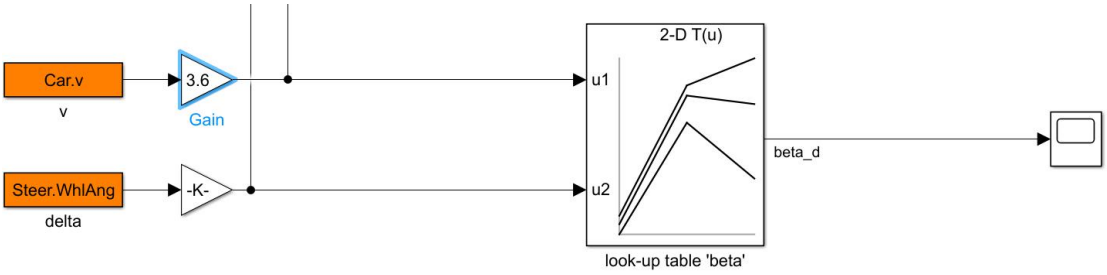


Figure 36 Configuration of 2D Look-up table for side slip angle

3.3 Construction of low pass filter

A low pass filter with cut-off frequency in the range 1-5 Hz is added downstream the look-up table for yaw rate in order to avoid saturation of PID controller.

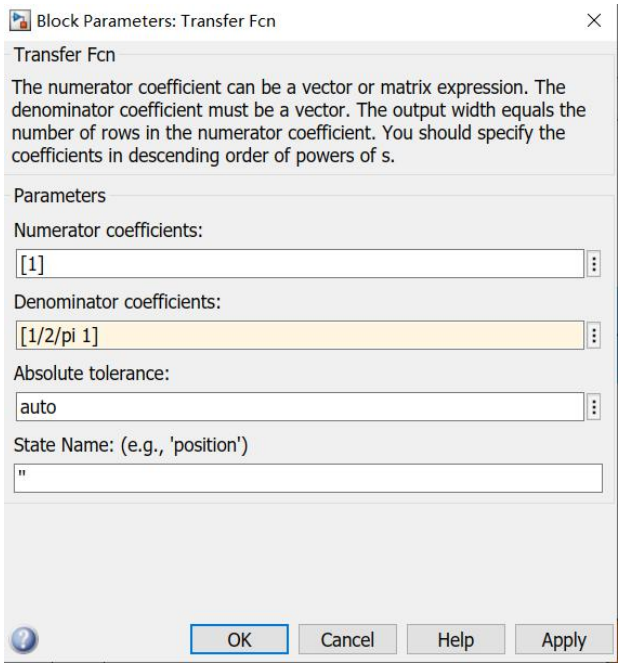


Figure 37 Transfer function block

A transfer function block is used as a low pass filter to avoid the saturation of PID controller.

The numerator is set to 1, while the denominator is [τ 1], where $\tau=1/(2\pi*f_c)$. Here f_c is the cutoff frequency, which is 1Hz for slow maneuvers and 5Hz for dynamic maneuvers.

3.4 Construction of torque vectoring block

This block has three inputs: ΔT from PID controller, speeds of two driving wheels from sensors. And the two outputs of this block are vectoring torques to be applied to two driving wheels.

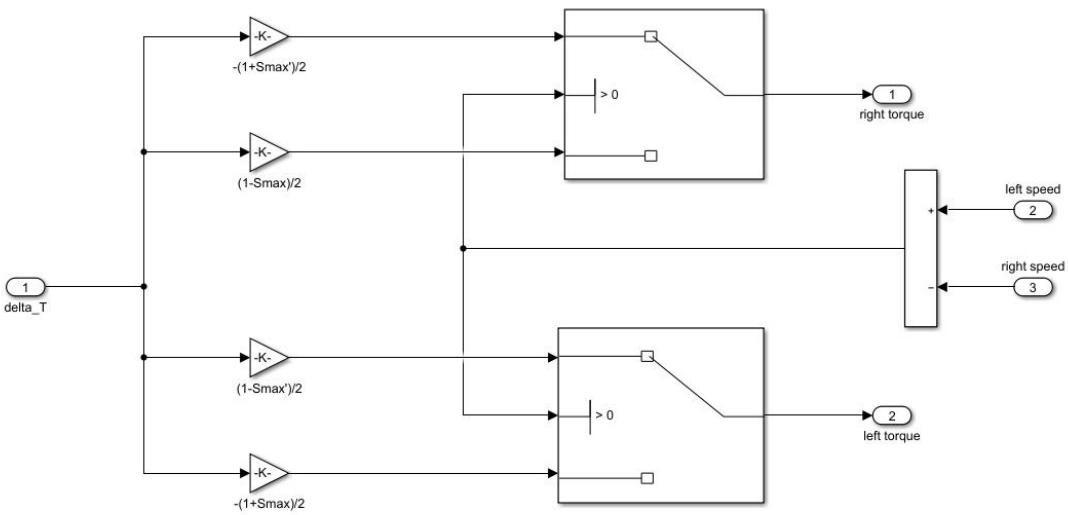


Figure 38 Configuration of torque vectoring block

Speeds of driving wheels are compared and then the result is inputted to the 'switch' block with another two inputs: vectoring torques for two wheels. As defined in Table 3, when the left speed ω_L is higher than the right one ω_R, additional torques distributed to both sides are

$$\Delta T_L = \frac{1 - S_{\max}'}{2} \Delta T \quad (32)$$

$$\Delta T_R = -\frac{1 + S_{\max}'}{2} \Delta T$$ (33) where the maximum acceptable differential speed ratio S_{max}' equals 0.125.

When $\omega_R > \omega_L$, additional torques distributed to both sides are

$$\Delta T_L = -\frac{1 + S_{\max}}{2} \Delta T \quad (34) \qquad \Delta T_R = \frac{1 - S_{\max}}{2} \Delta T$$ (35) where the maximum acceptable differential speed ratio S_{max} equals 0.125, too.

The differential block contains both conditions ($\omega_L > \omega_R$ and $\omega_R > \omega_L$) and the outputs of the block are left and right additional torques.

3.5 PID controller

Torque vectoring differential defined by CarMaker

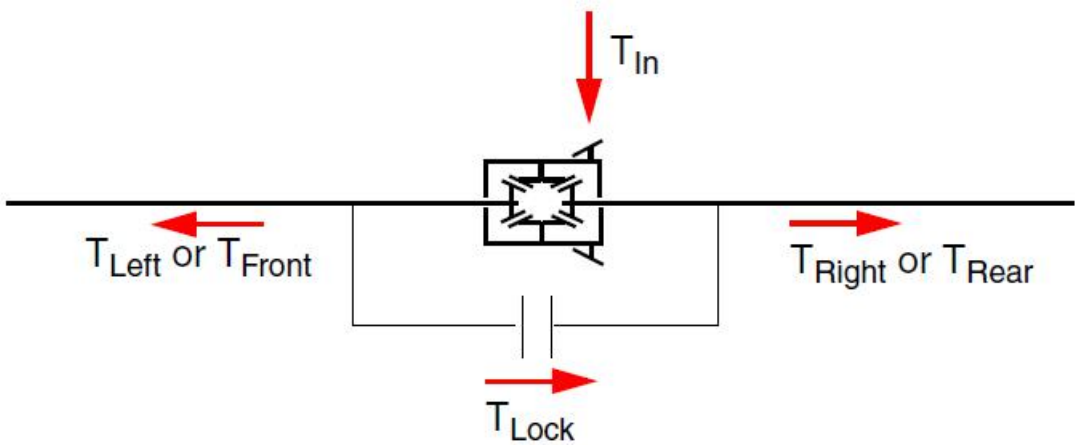


Figure 39 Principle of torque vectoring coupling in CarMaker

The locking torque depends on the torque distribution

$$T_{Lock} = T_{In}(1 - 2TrqRatio) \quad (36)$$

Where T_{In} is the input torque of the coupling (torque vectoring differential), and the torque ratio is defined by $TrqRatio_left = T_L / T_{In}$ (37), where T_L is the torque distributed to the left.

Besides the input torque is the sum of torques to left and right if there is no loss.

$$T_{In} = T_L + T_R \quad (38)$$

Thus we have $T_{Lock} = T_{In}(1 - 2T_L / T_{In}) = T_{In} - 2T_L = T_R - T_L$ (39), namely the torque difference between two driving wheels, ΔT defined in previous paragraph and the Simulink model.

So it's reasonable to choose ΔT as the output of PID controller.

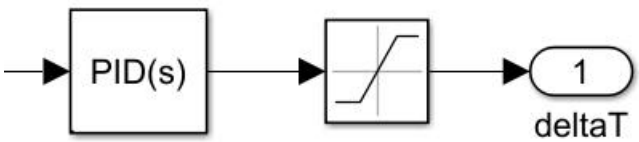


Figure 40 PID controller

Besides a saturation block is applied to the output of PID controller to limit the maximum value of ΔT to 5000Nm.

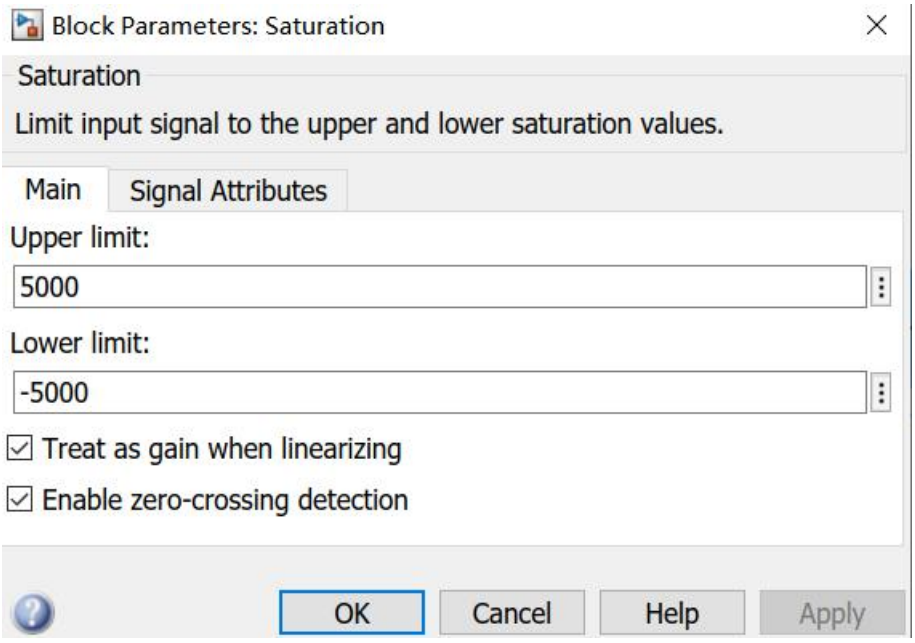


Figure 41 Saturation block

3.6 Tuning of PID controller

To tune the PID controller, several transient tests should be simulated, such as step steer tests.

A step steer test of 50 degrees of amplitude in 0.125s with speed equal to 80km/h is simulated.

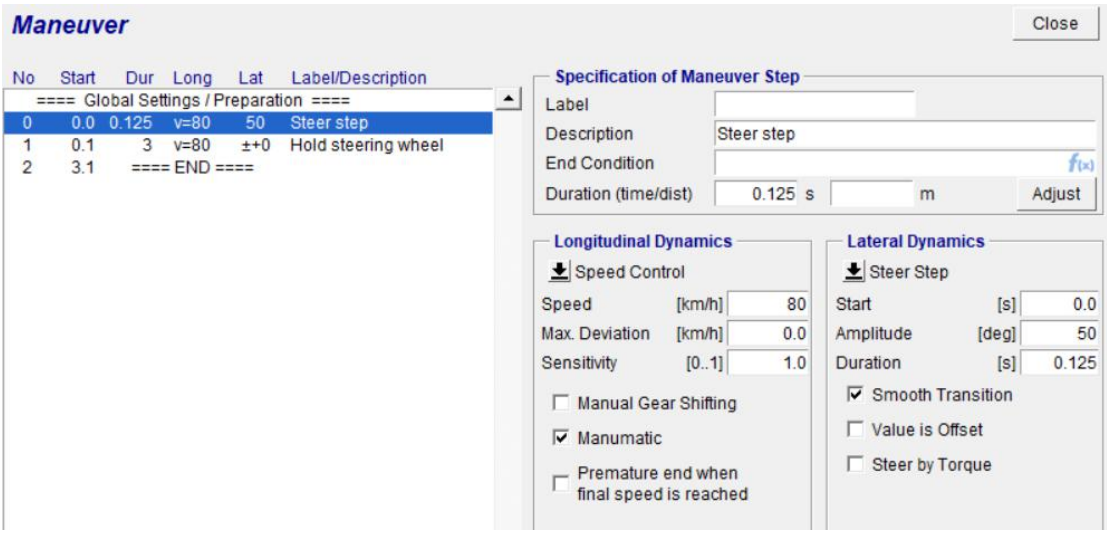


Figure 42 Step steer test

For step steer test, the cutoff frequency of transfer function is modified to 5Hz.

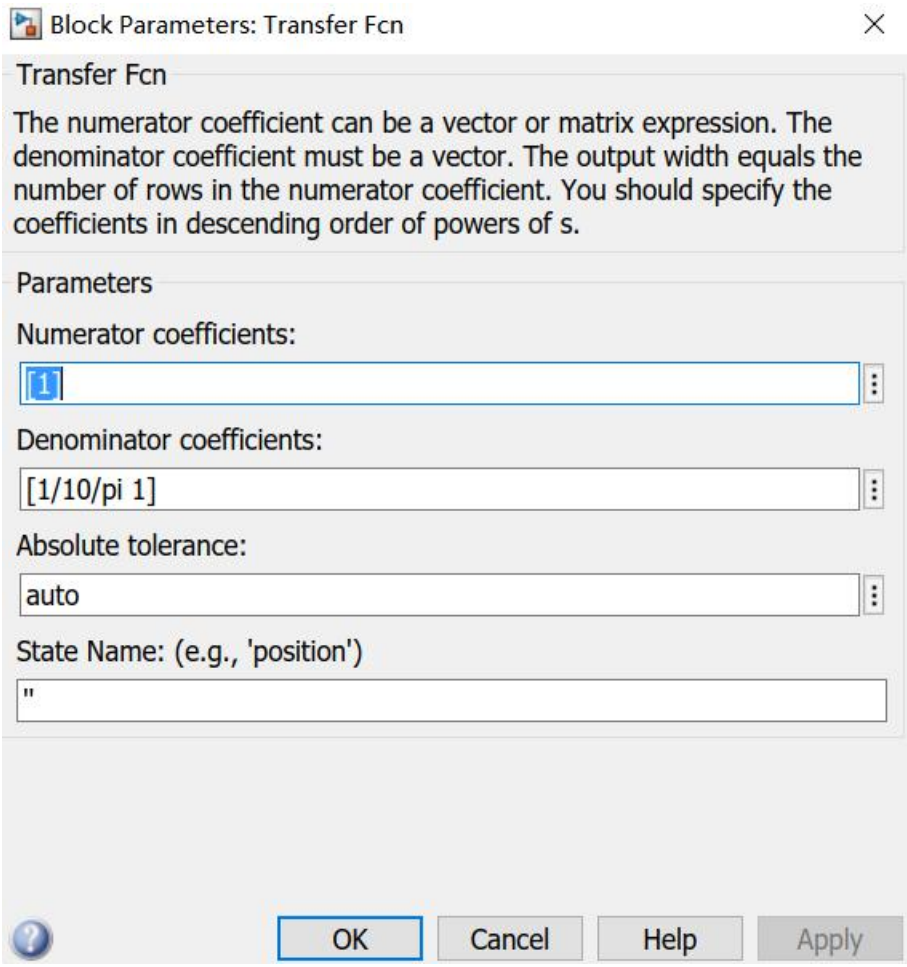


Figure 43 Transfer function block

3.6.1 Results of open differential

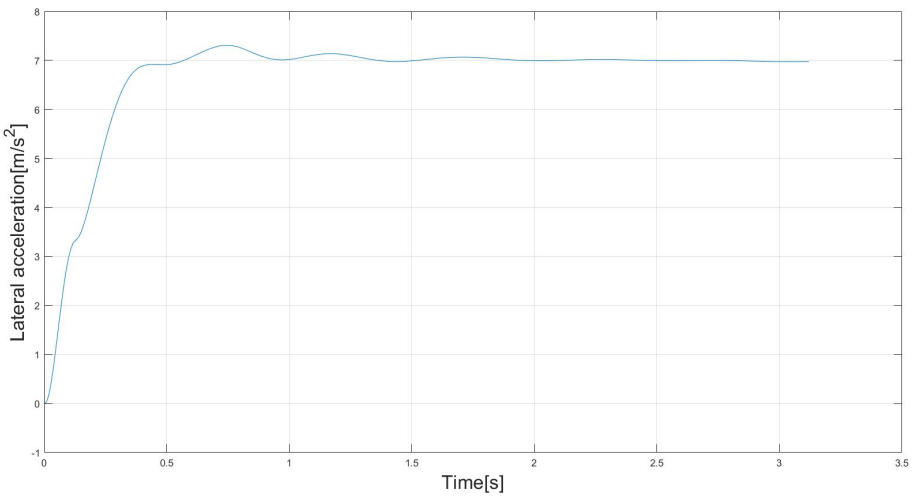


Figure 44 Lateral acceleration versus time for open differential

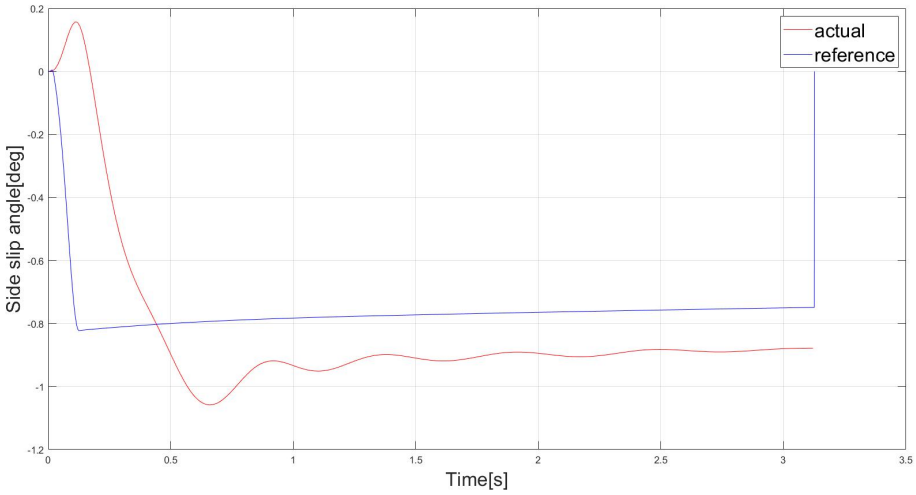


Figure 45 Side slip angle versus time for open differential

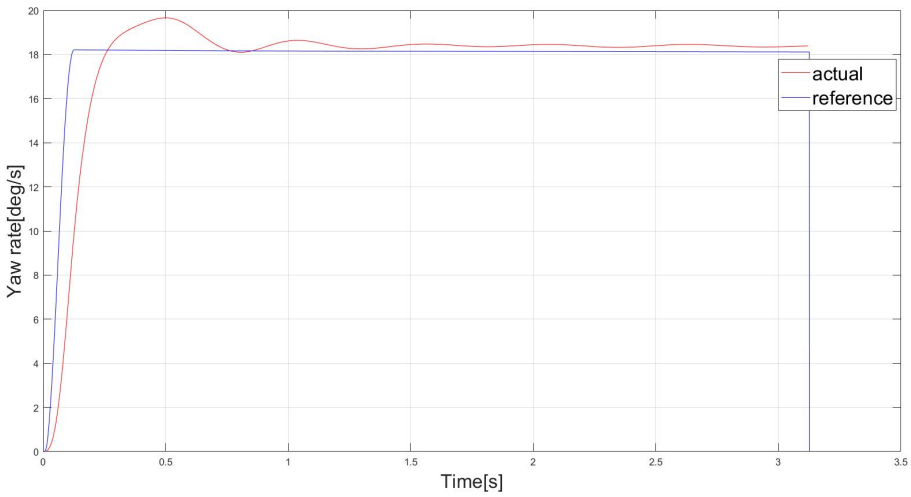


Figure 46 Yaw rate versus time for open differential

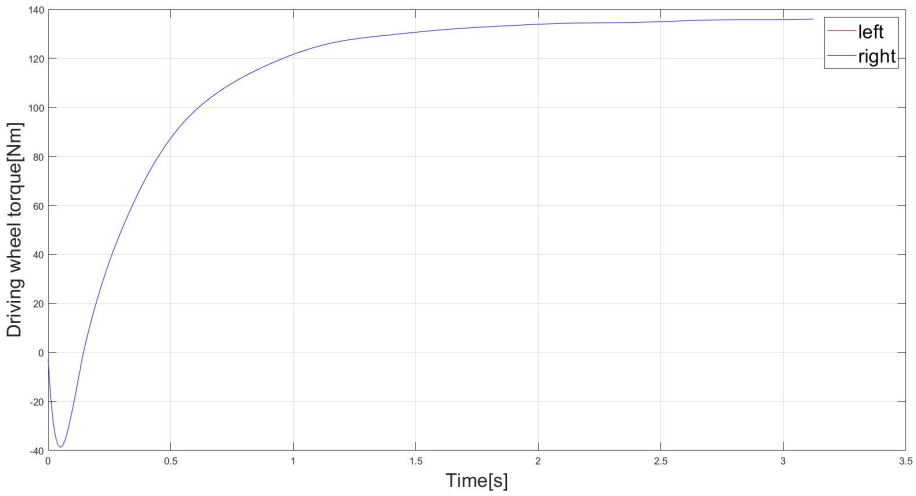


Figure 47 Driving wheel torques versus time for open differential

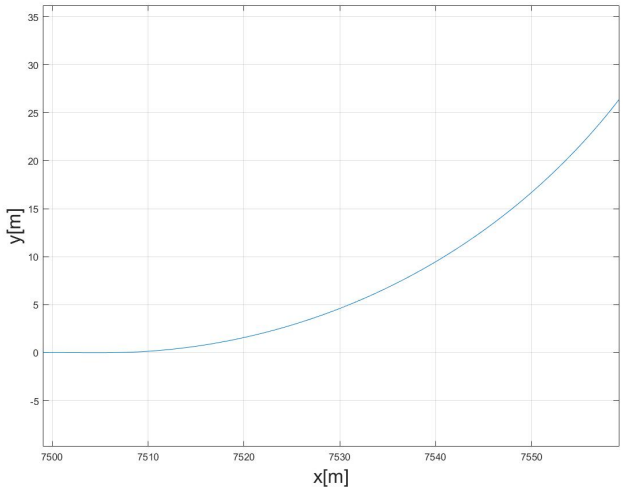


Figure 48 Trajectory for open differential

3.6.2 Results of torque vectoring differential with different settings of PID gains

A. P=400-1600 I=D=0

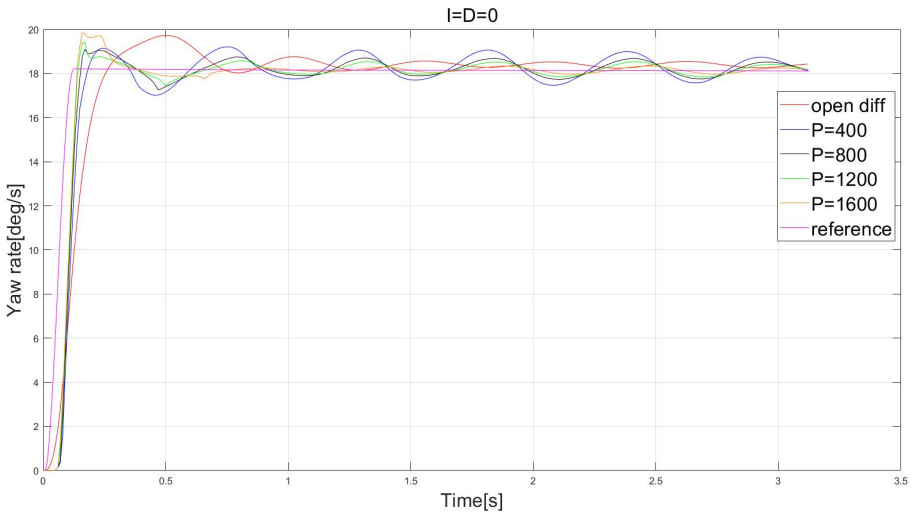


Figure 49 Yaw rate for different P

For P=400-1600, curves of yaw rate show decreasing amplitudes of oscillations while the rise time is increasing. When P=1200, a trade-off between rise time and oscillations could be found, so P_{opt}=1200.

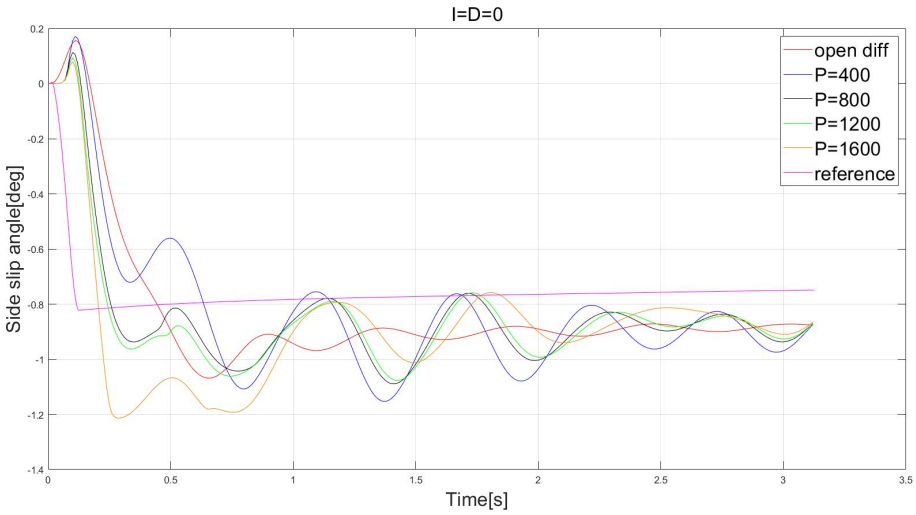


Figure 50 Side slip angle for different P

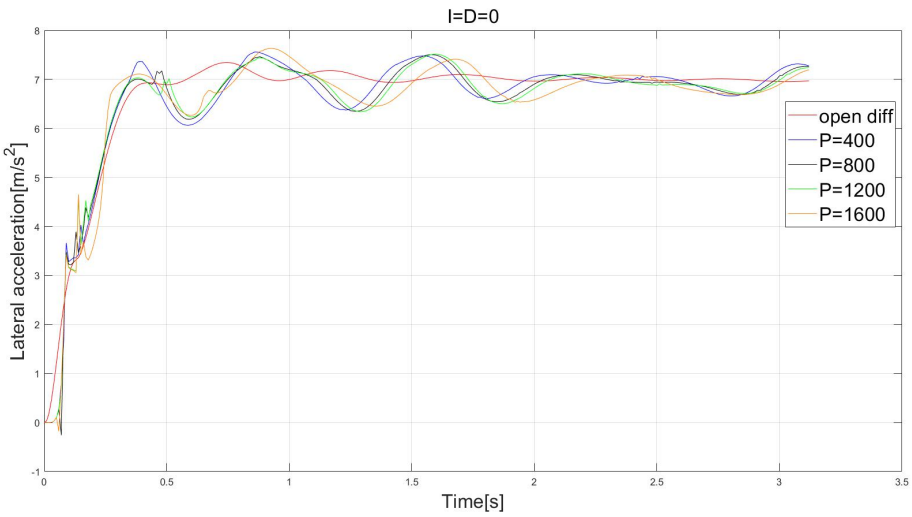


Figure 51 Lateral acceleration for different P

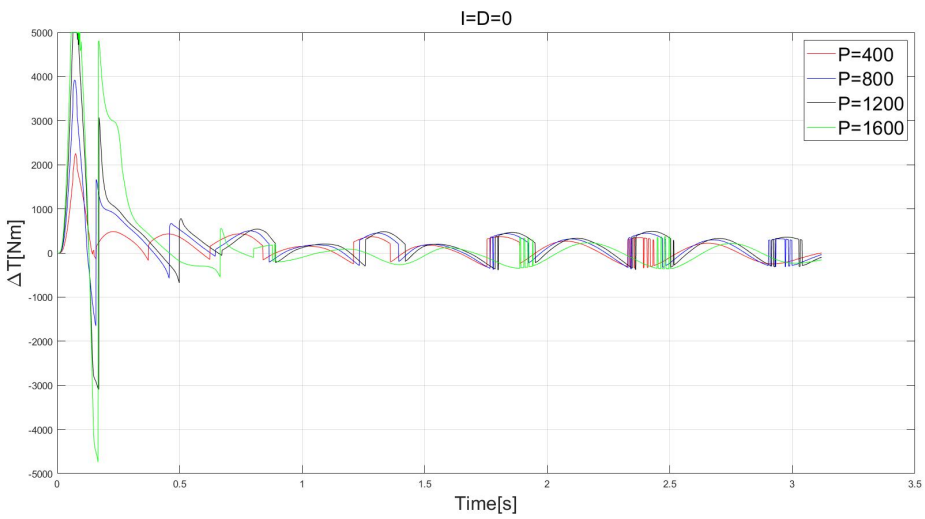


Figure 52 ΔT for different P

ΔT is defined as the torque transferred to the right wheel. When ΔT is negative, the torque is transferred from the right wheel to the left. Transferred amount is the absolute value of ΔT.

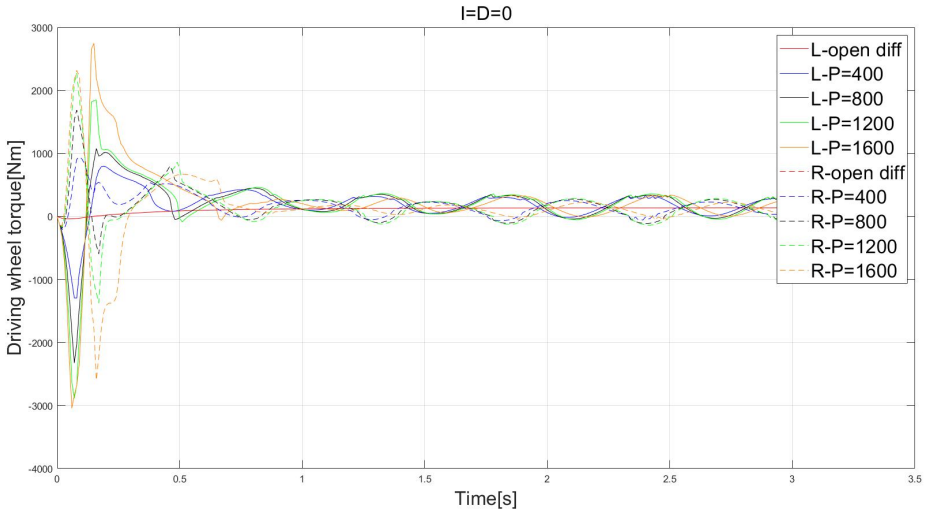


Figure 53 Driving wheel torques for different P

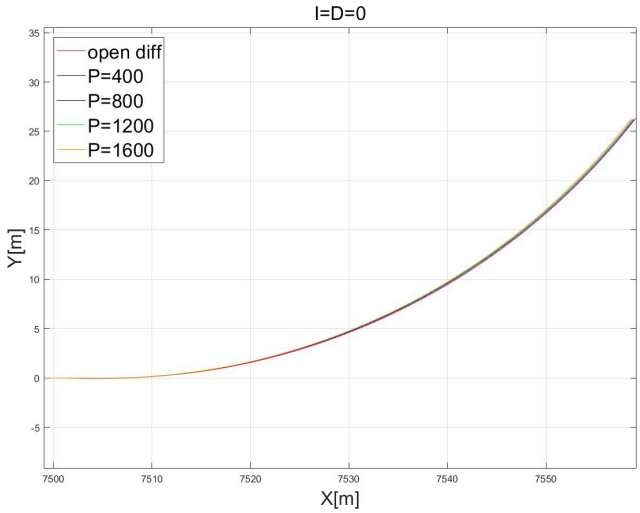


Figure 54 Trajectory for different P

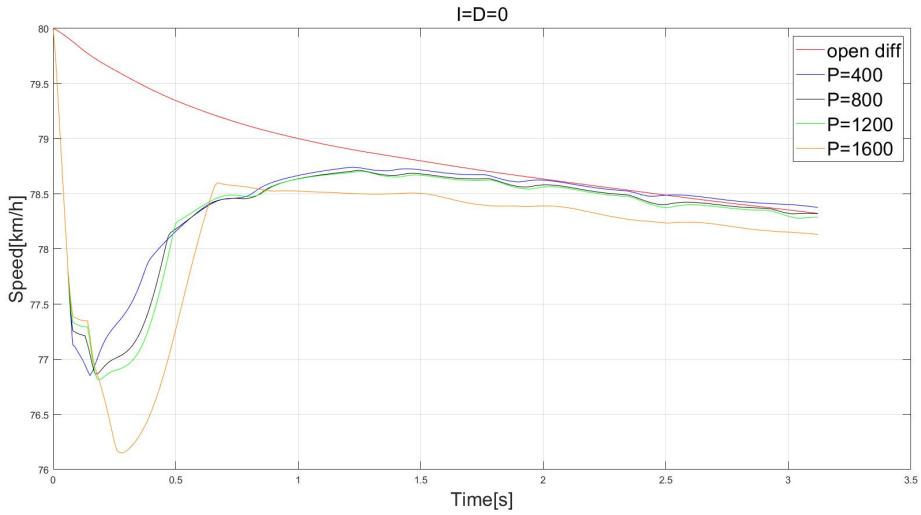


Figure 55 Vehicle speed for different P

B. P=1200, I=100-3000, D=0

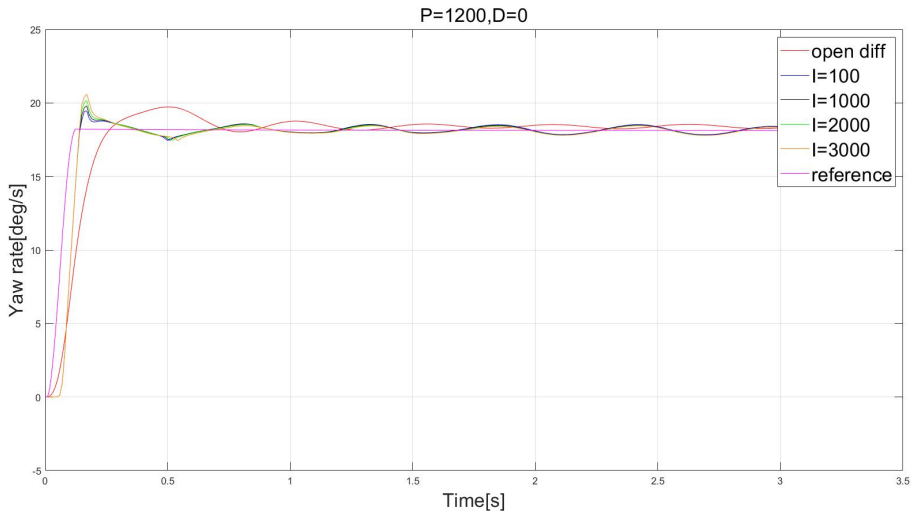


Figure 56 Yaw rate for different I

I doesn't contribute too much to yaw rate.

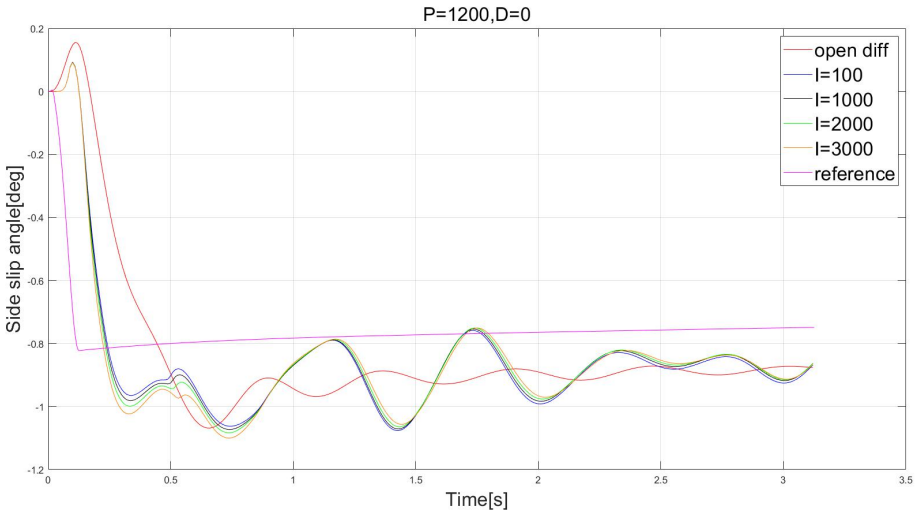


Figure 57 Side slip angle for different I

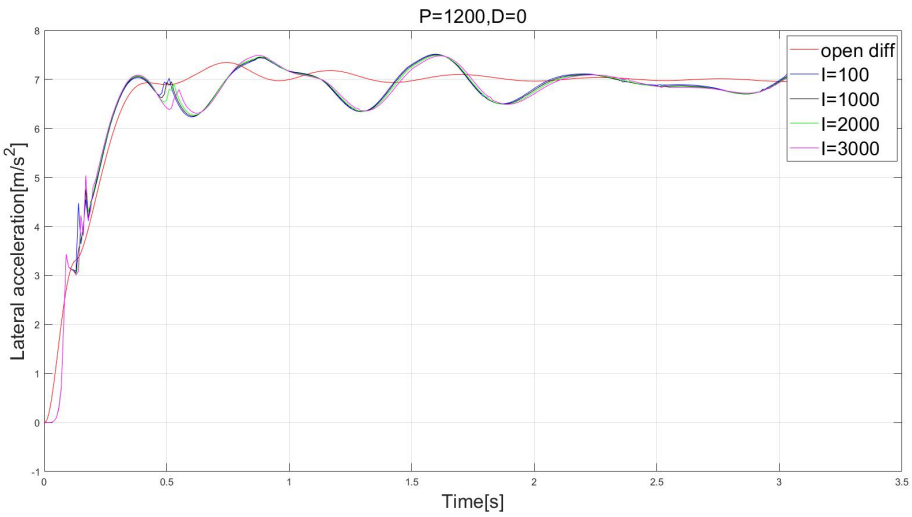


Figure 58 Lateral acceleration for different I

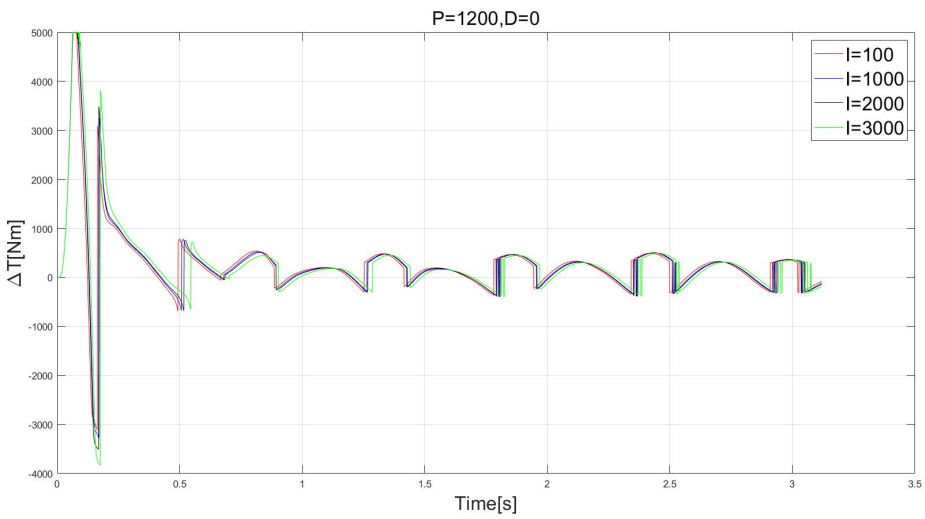


Figure 59 ΔT for different I

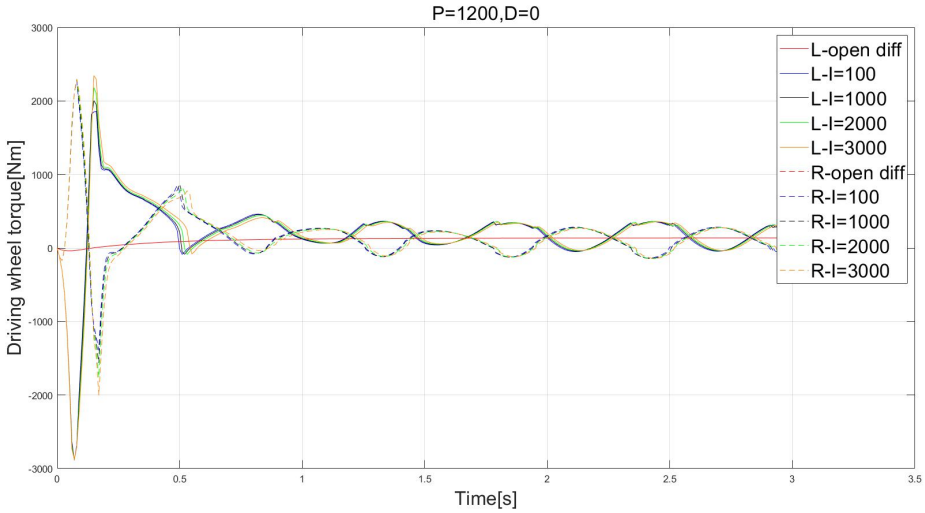


Figure 60 Driving wheel torques for different I

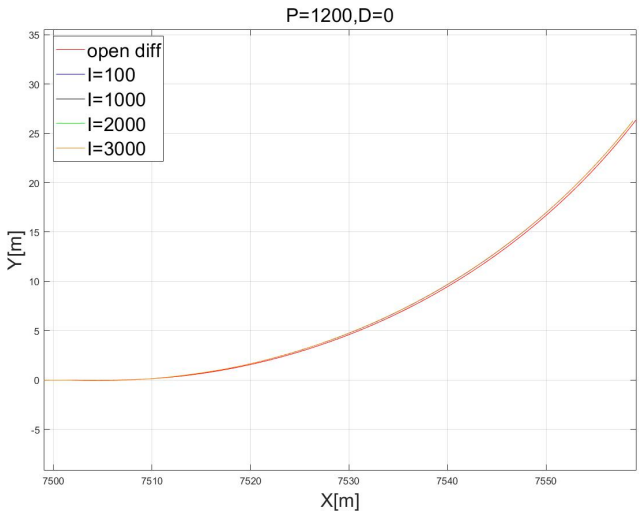


Figure 61 Trajectory for different I

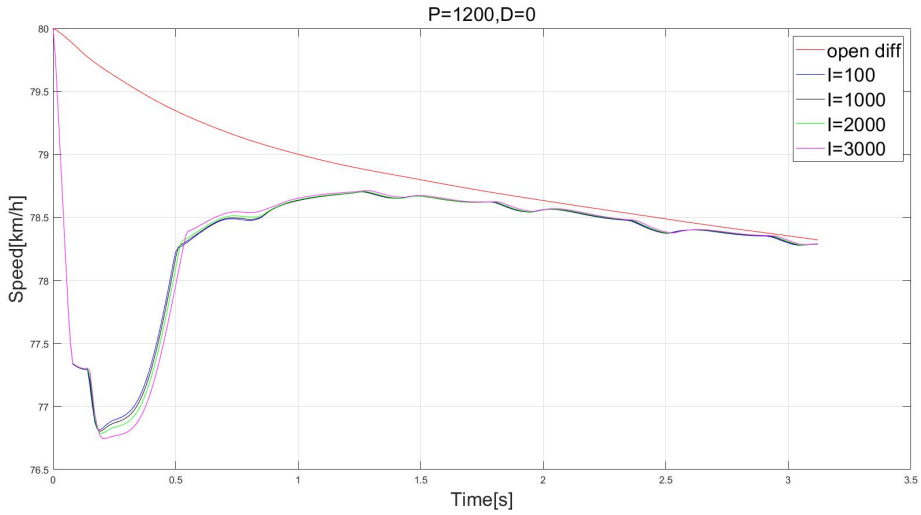


Figure 62 Vehicle speed for different I

From curves of yaw rate and side slip angle, it can be seen that I doesn't contribute positively to the vehicle behaviors. As I increases, the difference between actual and reference side slip angles even become higher. So I_{opt}=0.

C. P=1200, I=0, D=20-100

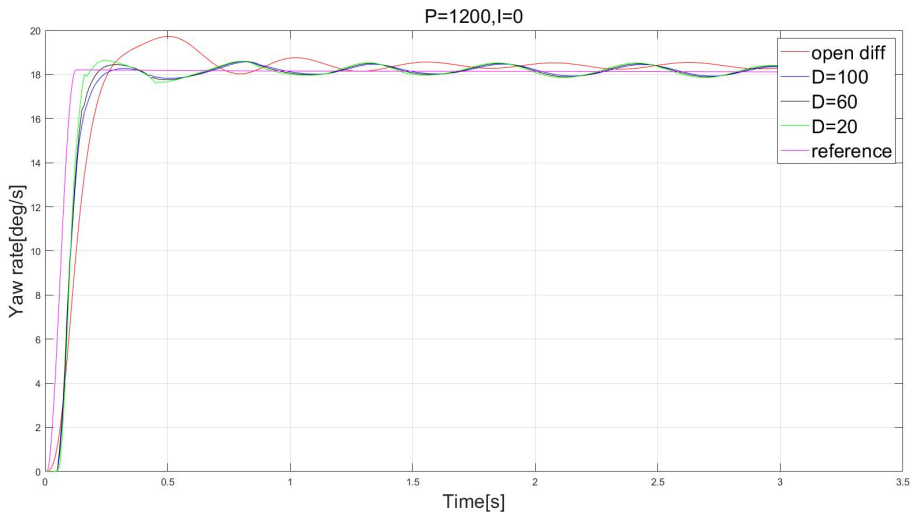


Figure 63 Yaw rate for different D

D doesn't contribute too much to yaw rate either. As D continues to increase, some oscillations occur.

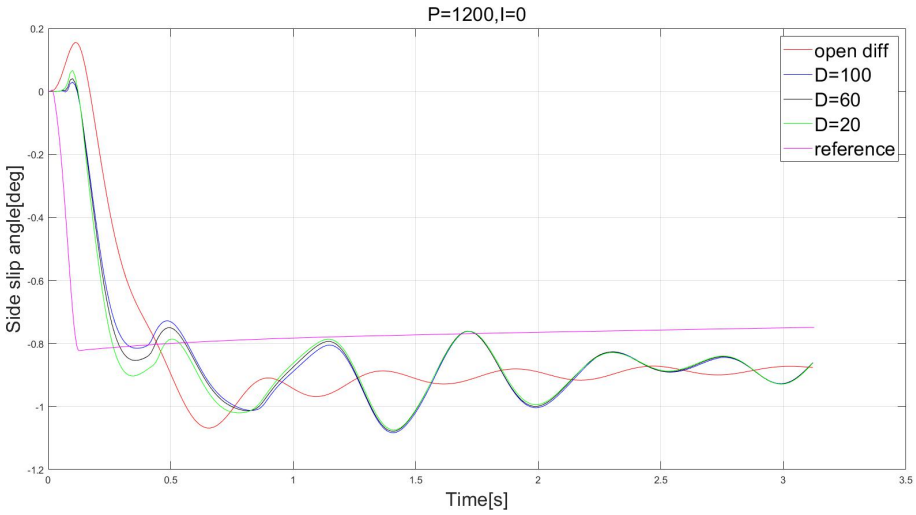


Figure 64 Side slip angle for different D

D_{opt}=0.

By comparing results above, the best settings of PID gain is P_{opt}=1200, I_{opt}=0, D_{opt}=0.

4 Effects of different gain at the output of reference yaw rate

A steady-state test, for example, steering pad test is used to show the effects of different gains.

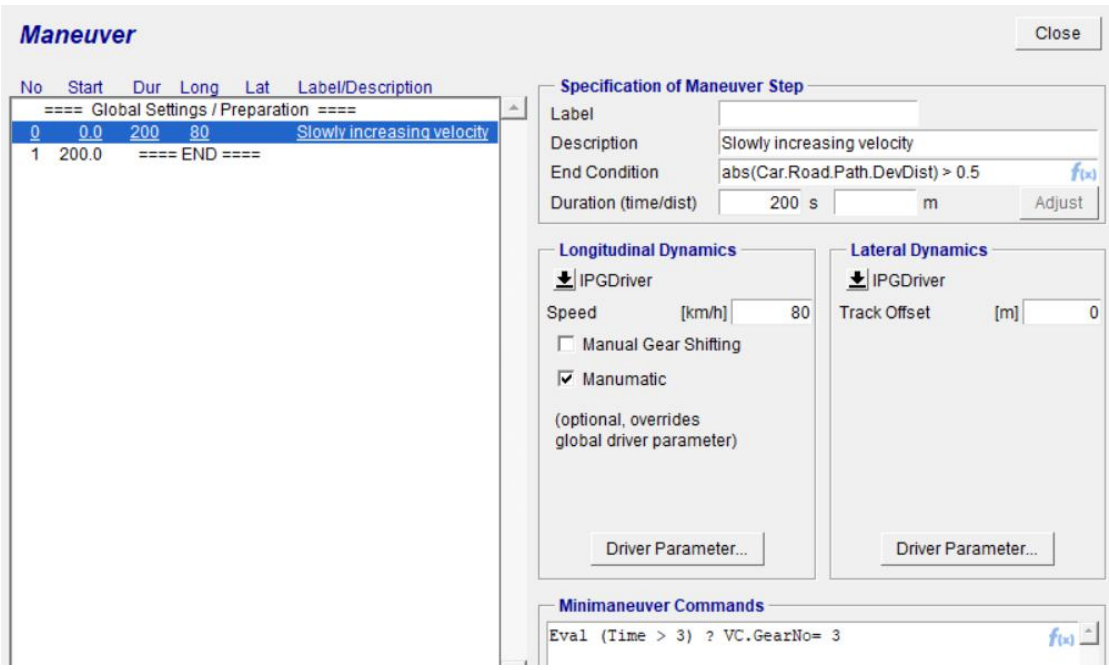


Figure 65 Steering pad test maneuver

The radius of steering pad test is 100m, and the cruising speed is 80km/h (22.2m/s), thus the lateral acceleration is about 5m/s^2.

4.1 Results of open differential

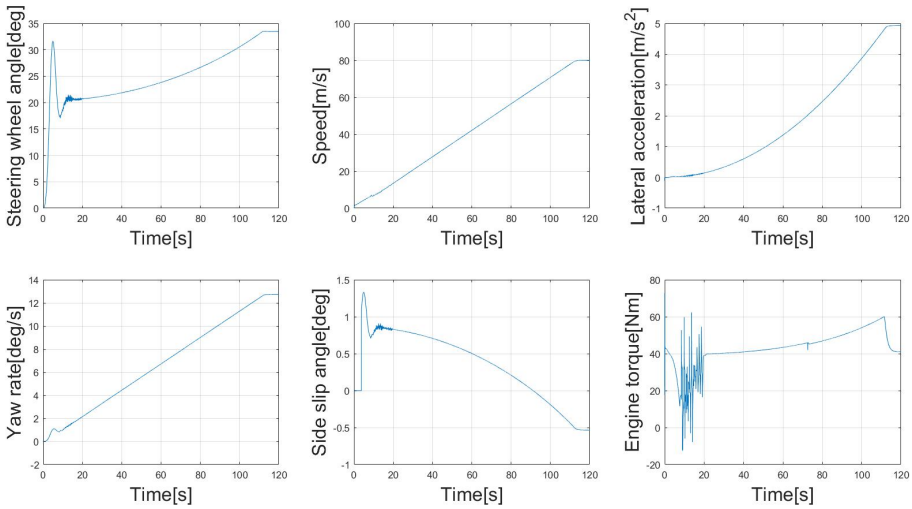


Figure 66 Results for open differential

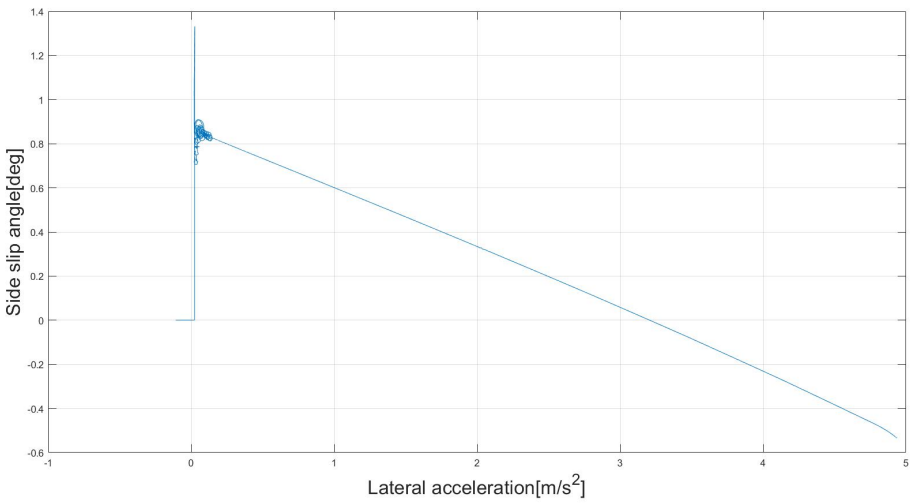


Figure 67 Side slip angle versus lateral acceleration

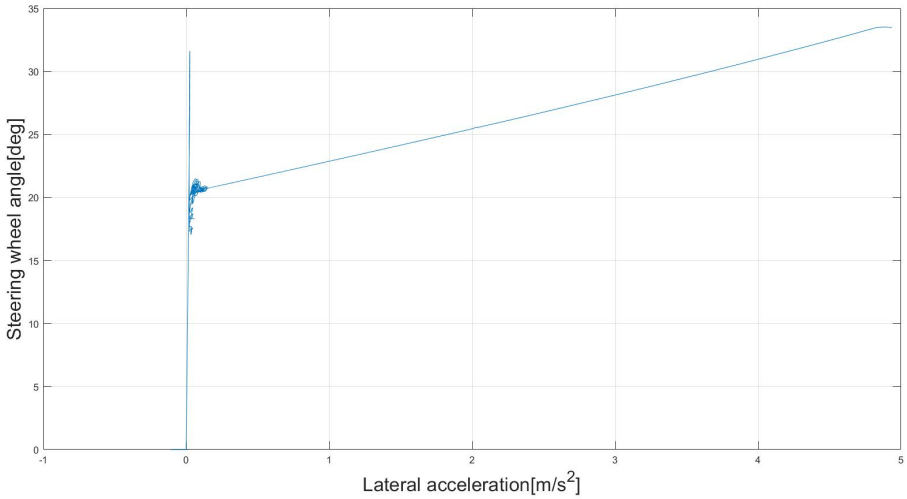


Figure 68 Steering wheel angle versus lateral acceleration

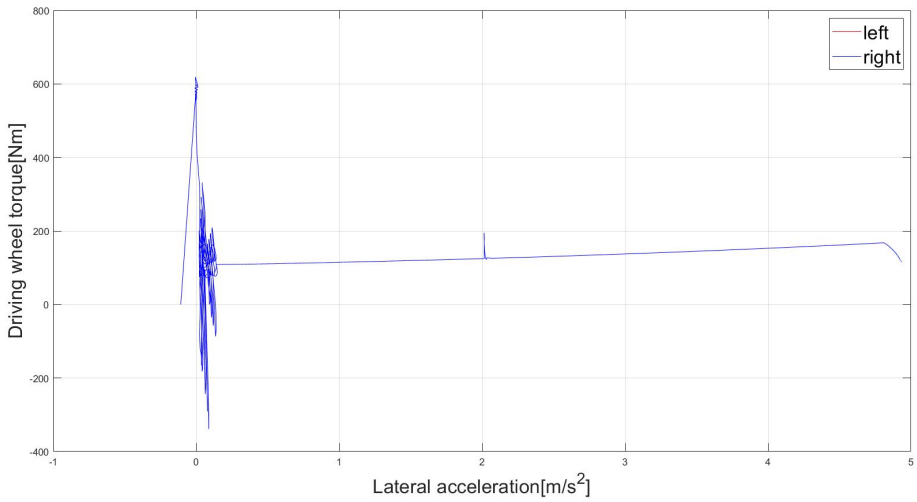


Figure 69 Driving wheel torques versus lateral acceleration

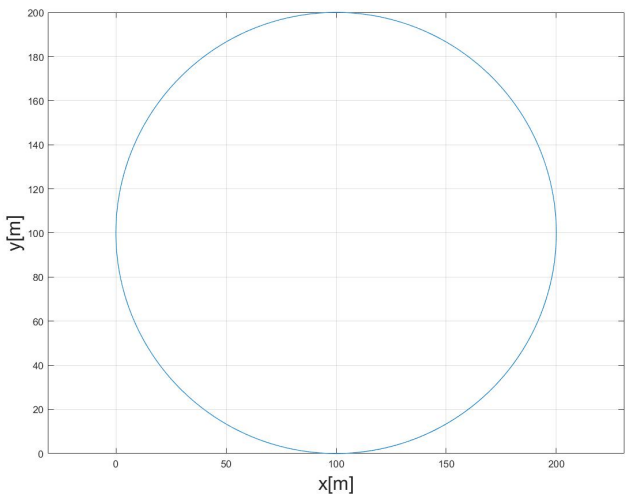


Figure 70 Trajectory of vehicle

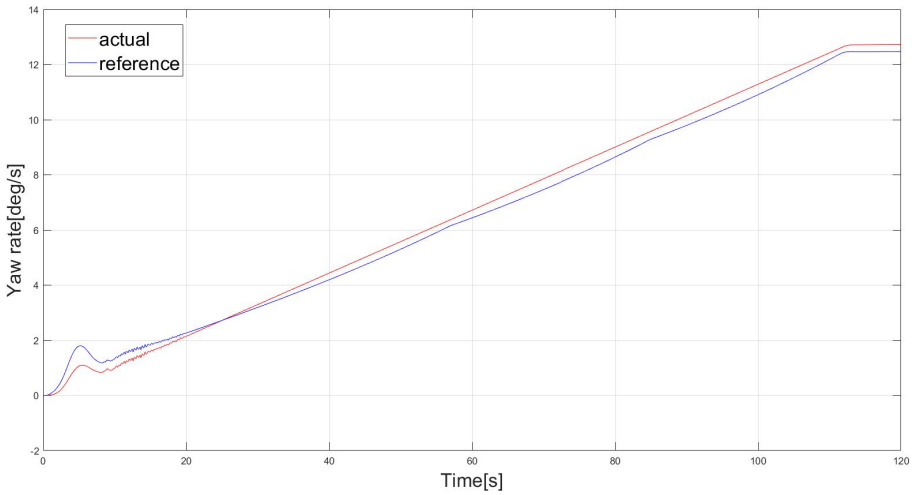


Figure 71 Actual and reference yaw rates

4.2 Torque vectoring differential with a gain at the output of look-up table

A gain is added at the output of the look-up table of reference yaw rate. The value of gain is set to 0.7, 0.9, 1.1 and 1.3 respectively.

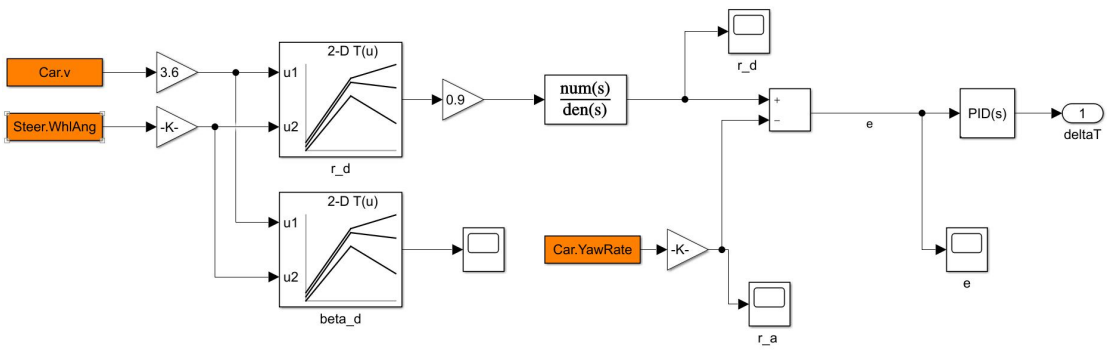


Figure 72 Addition of a gain at the output of look-up table

The curves of steering wheel angle and side slip angle versus lateral acceleration are shown below.

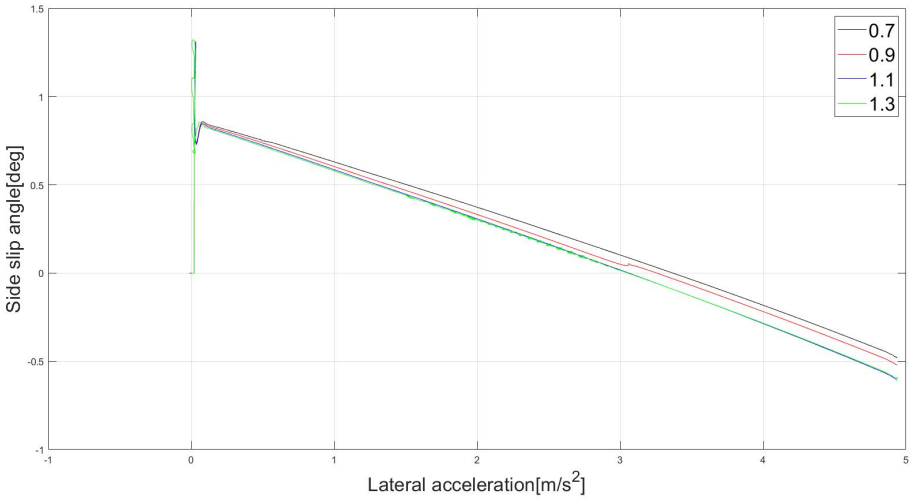


Figure 73 Comparison of side slip angle versus lateral acceleration for different gains

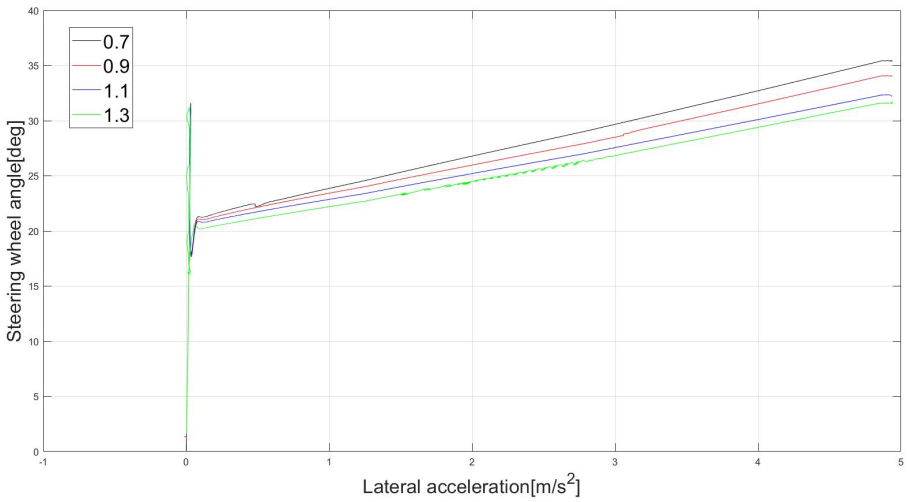


Figure 74 Comparison of steering wheel angle versus lateral acceleration for different gains

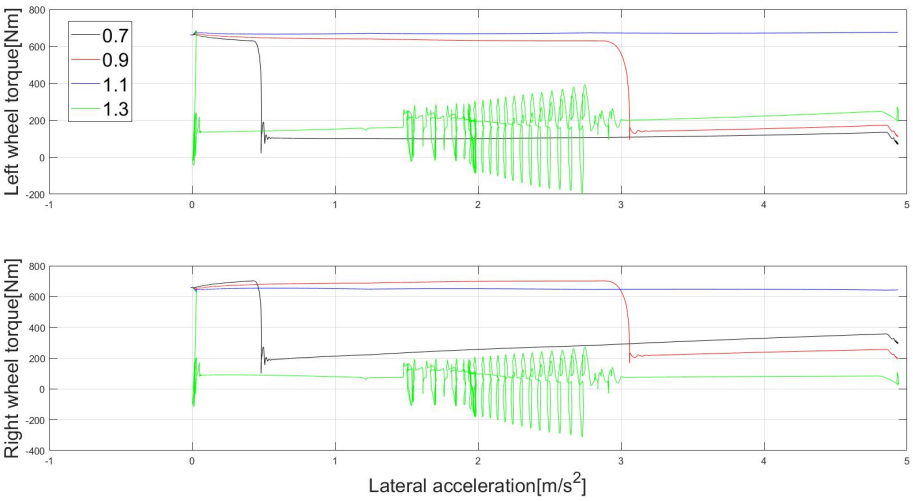


Figure 75 Comparison of driving wheel torques versus lateral acceleration for different gains

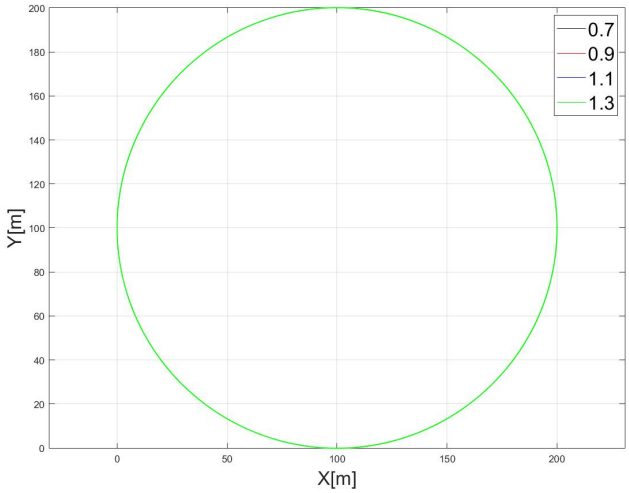


Figure 76 Comparison of trajectories for different gains

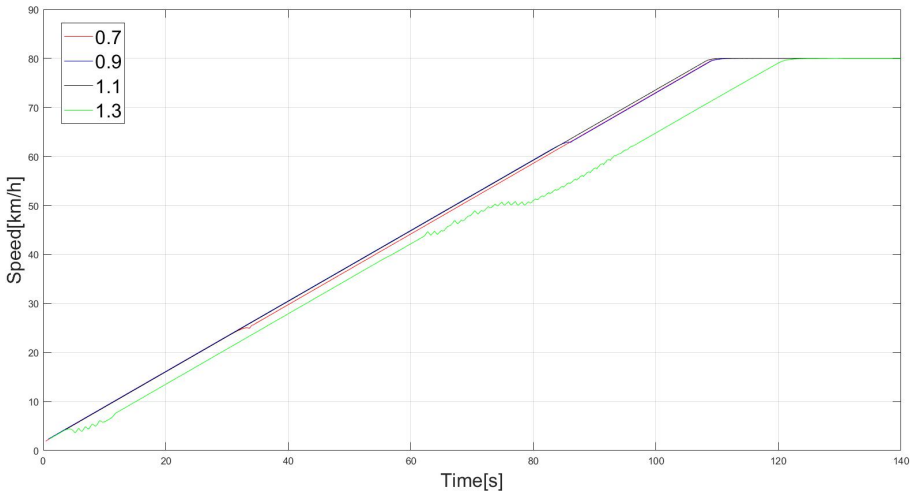


Figure 77 Speed versus time for different gains

Results show that with different request of yaw rate, the understeer gradient shows evident variations.

5 Conclusion

From results of simulations above, it ca be seen that a torque vectoring differential allows a vehicle to achieve better behaviors: such as yaw rate closer to reference, more stable variations of side slip angle and lateral acceleration, which lead to a better stability of vehicle.

However, torque vectoring has some limitations, such as limited performance under extreme operations. It would be better to use a torque vectoring system and some other system, for example, ESP, as discussed in the introduction of existent control strategies. Though the complexity would be another problem.

Appendix

Introduction to the CarMaker model

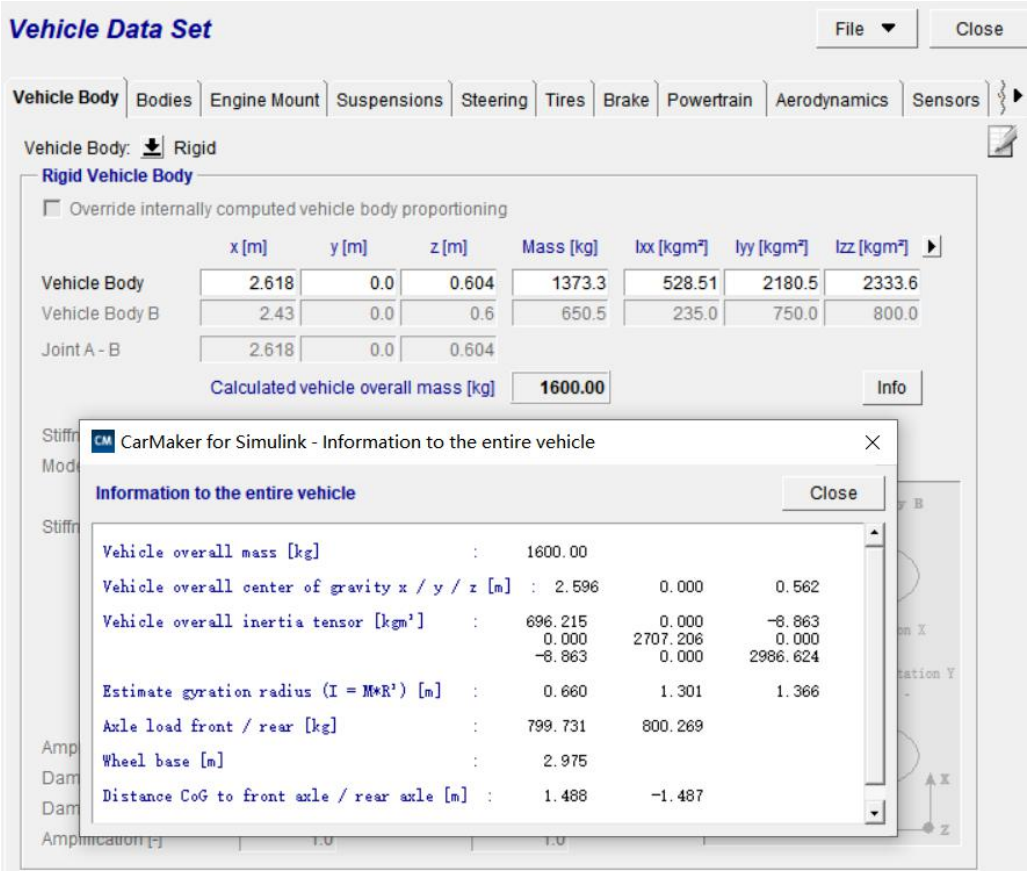
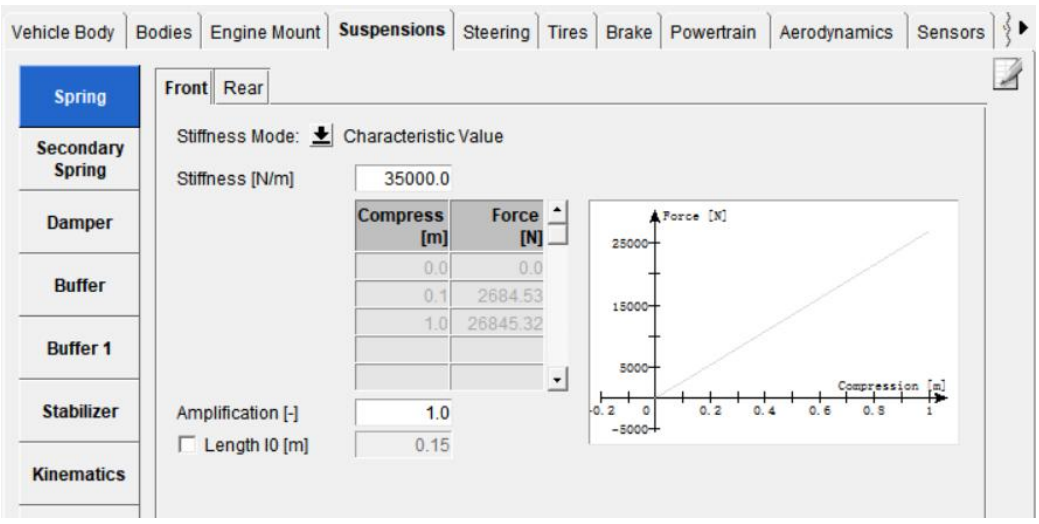


Figure 78 Basic data for CarMaker model

From the figure above, it can be seen that vehicle overall mass m=1600kg, wheel base l=2.975m, distances from to front and rear axles are a=1.488m and b=1.487m respectively, and moment of inertia J_z=2986.624kgm^2.



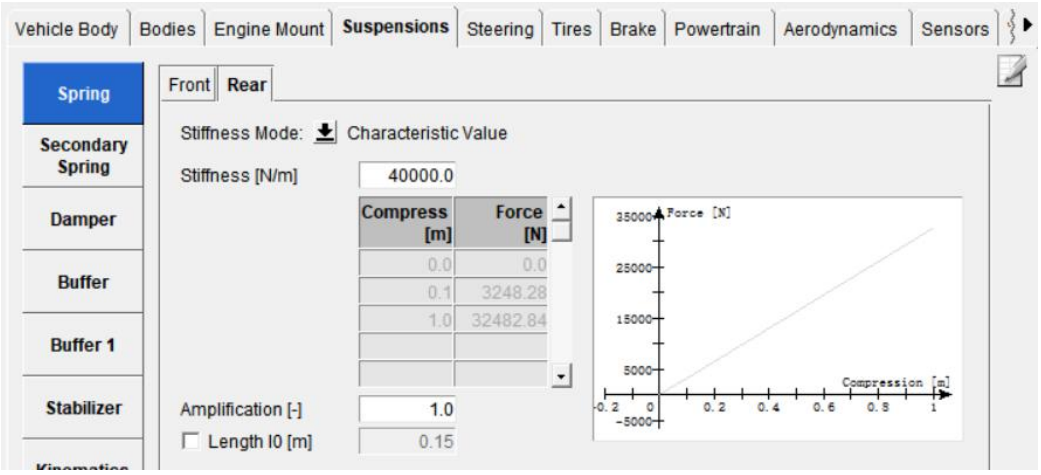
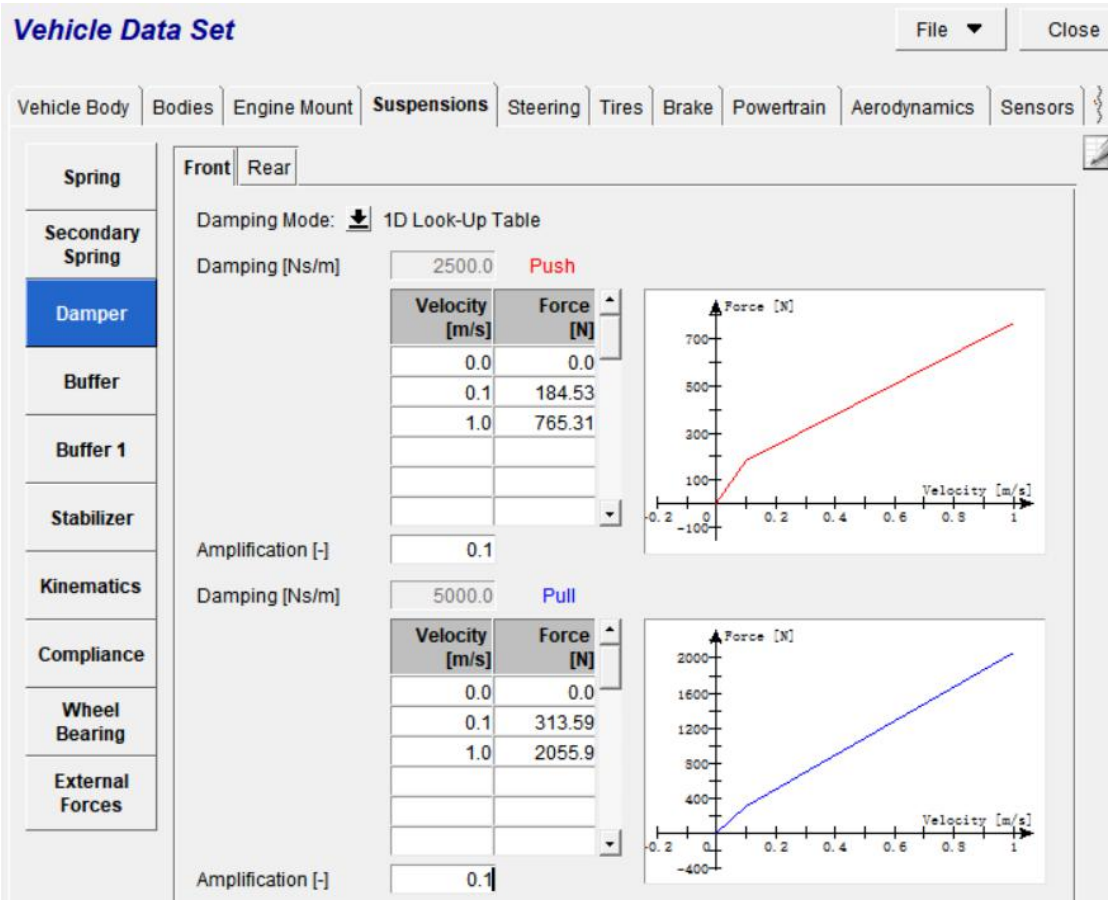


Figure 79 Spring characteristics for suspensions

Stiffness of springs for front and rear suspensions are 35000N/m and 40000N/m respectively, while the secondary springs are not specified.



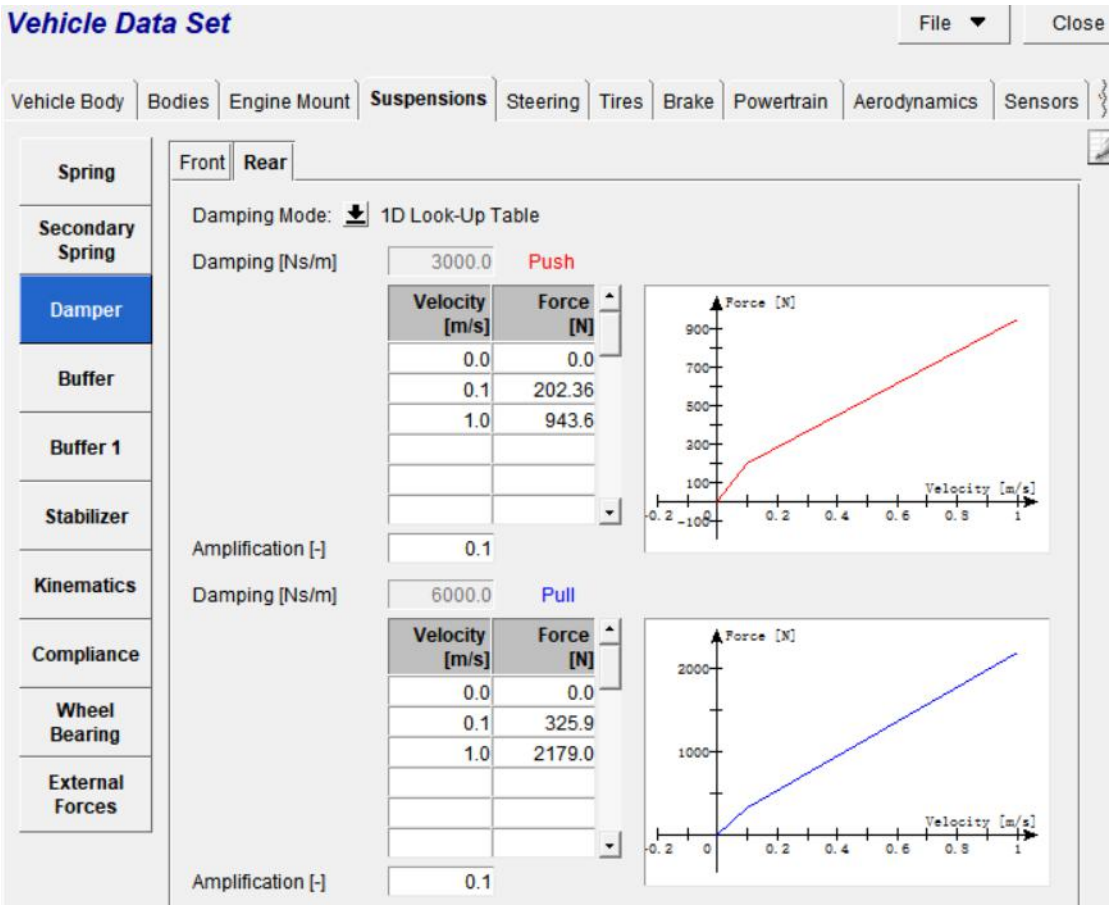


Figure 80 Damper characteristics for suspensions

For both the front and the rear dampers, the damping coefficients for the pushing and pulling actions follow two 1D look-up tables respectively, which are 2023.6Ns/m and 3259Ns/m for the first phase respectively, 823.6Ns/m and 2059Ns/m respectively for the second phase. The amplification factors for both front and rear dampers are 0.1.

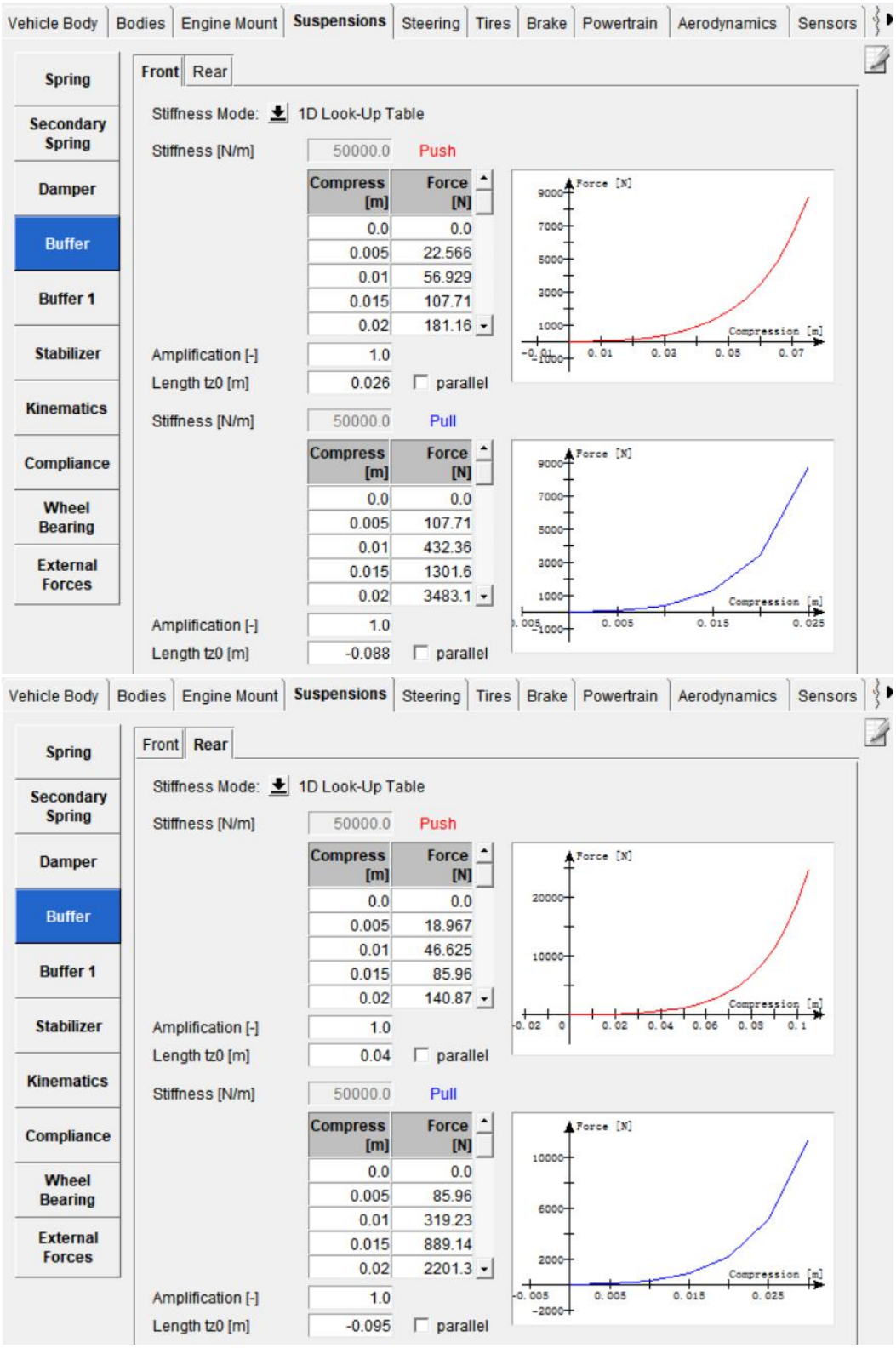


Figure 81 Buffer characteristics for suspensions

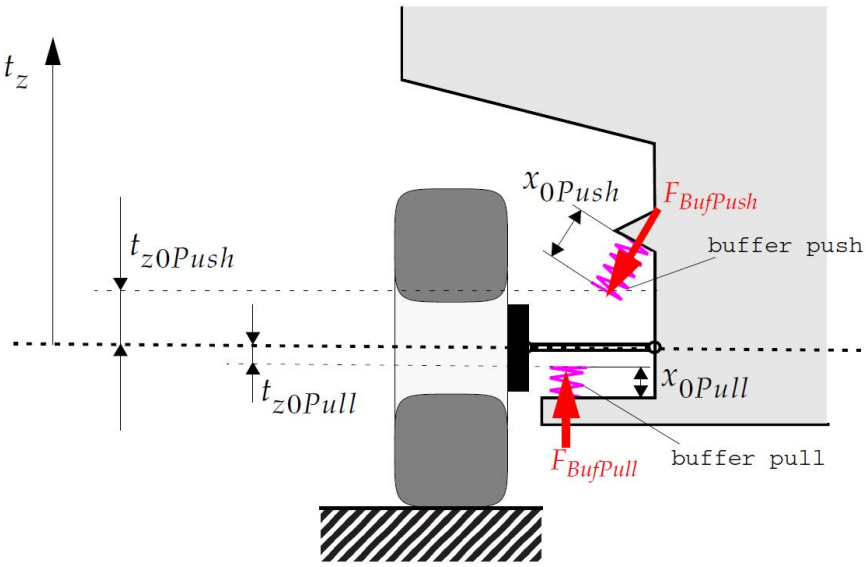
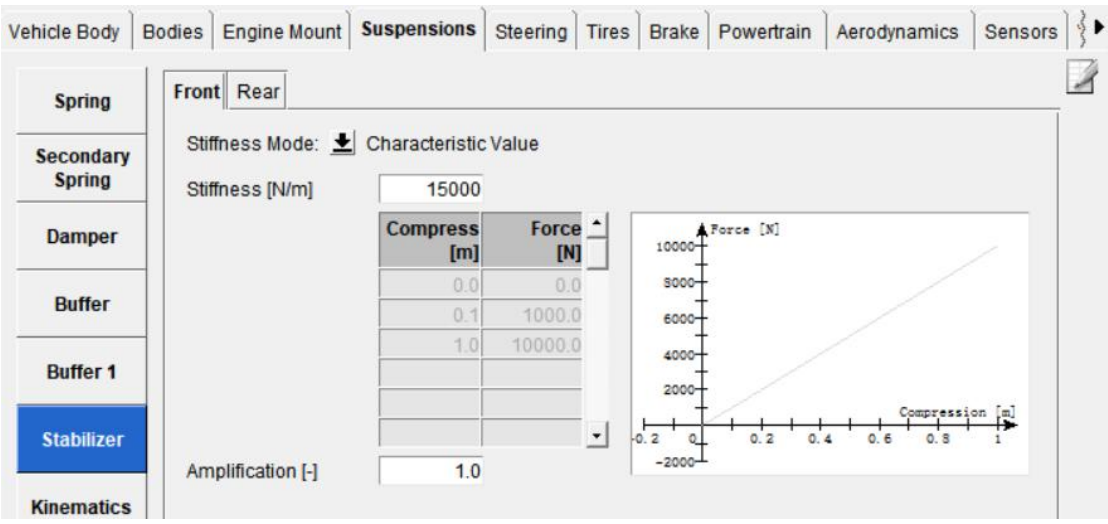


Figure 82 Buffer pushing and pulling

The pushing buffer and pulling buffer for both front and rear suspensions are specified following several 1D look-up tables shown in the figures. And the initial distances from wheel center to buffers tz0 are defined, too. These values define the active ranges of buffers.



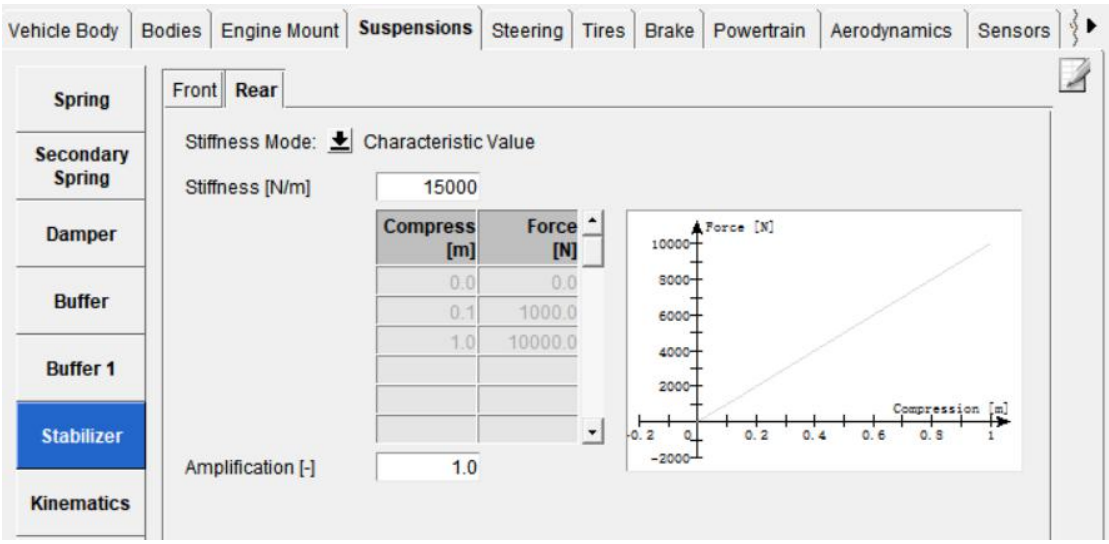


Figure 83 Stabilizer characteristics for suspensions

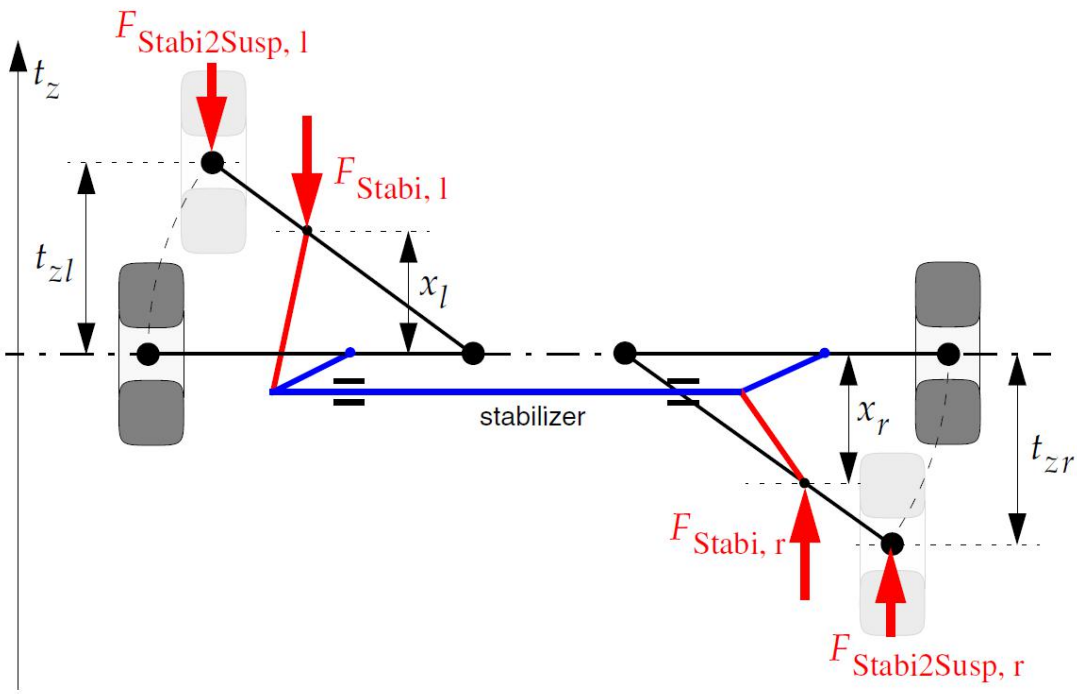


Figure 84 Stabilizer model with deflection length difference

The left and right parts of the stabilizer have the same stiffness which is 15000N/m.

The stabilizer deflection difference x^* is defined by a deflection length difference, $x^* = x_r - x_l$, and the wheel compression ratio is $t^* = t_{zr} - t_{zl}$, thus the stabilizer stiffness is defined as $F_{Stabi,l} = amp * f(x^*) = amp * c_{Stabi} * x^*$, where amp is the amplification factor, which is set to

1 under normal circumstances. And c_{Stabi} is the stabilizer stiffness. The deflection lengths, x_l and x_r, are distances between wheel centers before and after deflection. Thus the stiffness is calculated on the wheel centers [11]

Similarly, for stabilizer model with deflection angle difference, the stiffness is calculated through $M_{Stabi,l} = amp * f(x^*) = amp * c_{\alpha Stabi} * x^*$, and $M_{Stabi,r} = -M_{Stabi,l}$, where M_{Stabi} is the moment on the stabilizer bar, and the stabilizer deflection angle difference is $x^* = \alpha_r - \alpha_l$. Under this condition, the stabilizer works as a torsion spring [11].

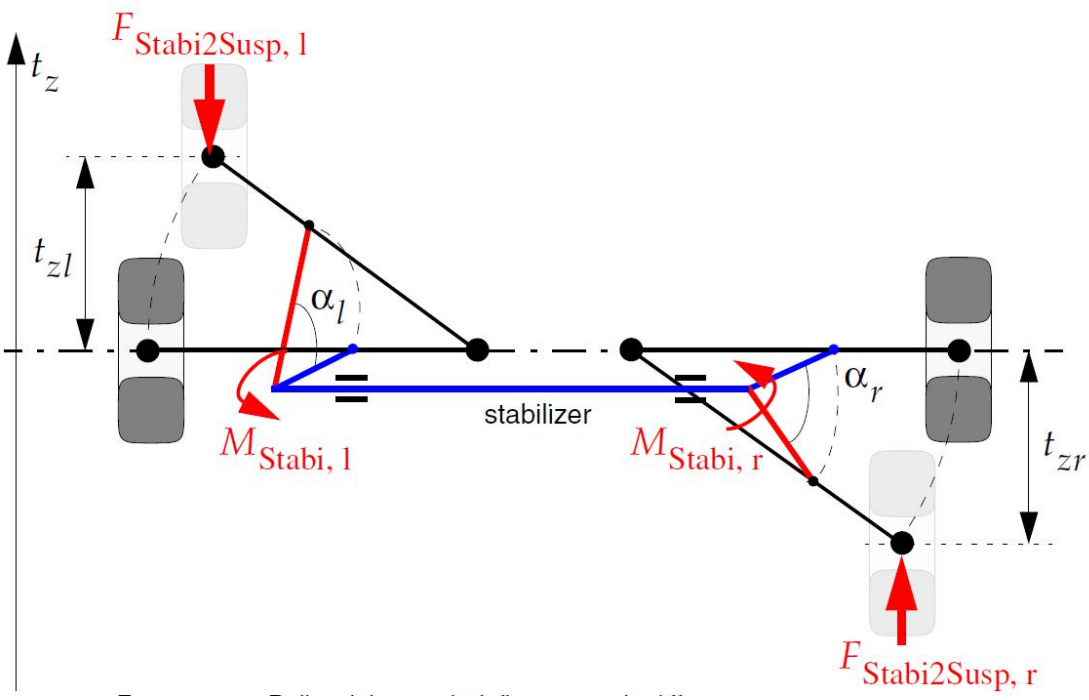


Figure 85 Stabilizer model with deflection angle difference

The front suspension model is a McPherson suspension defined using an SKC file, which is extracted from ADAMS.

Figure 86 Selection of the front suspension

SuspF.Kin.0.L.Data		tx	ty	tz	rx	ry	rz	lSpring	lDamp	lStabi
0	0	-0.0493203	-0.0267901	-0.0756322	-0.0895282	-0.1182488	0.7314551	0.2390546	0.2390788	-0.0636221
0	1	-0.0425011	-0.0215444	-0.0750905	-0.0734704	-0.1003294	0.6062151	0.2395857	0.2396098	-0.0648872
0	2	-0.0361675	-0.0177499	-0.0746994	-0.0616399	-0.0836123	0.4996348	0.2399862	0.2400105	-0.0658530
0	3	-0.0301281	-0.0149132	-0.0744073	-0.0525695	-0.0676147	0.4039669	0.2403016	0.2403258	-0.0666231
0	4	-0.0242902	-0.0127967	-0.0741895	-0.0455514	-0.0521001	0.3154218	0.2405535	0.2405777	-0.0672474
0	5	-0.0186013	-0.0112663	-0.0740319	-0.0401865	-0.0369311	0.2317872	0.2407542	0.2407784	-0.0677540
0	6	-0.0130286	-0.0102399	-0.0739259	-0.0362298	-0.0220201	0.1516164	0.2409112	0.2409354	-0.0681601
0	7	-0.0075509	-0.0096649	-0.0738660	-0.0335249	-0.0073076	0.0738817	0.2410293	0.2410535	-0.0684768
0	8	-0.0021543	-0.0095073	-0.0738486	-0.0319709	0.0072478	-0.0021981	0.2411116	0.2411359	-0.0687114
0	9	0.0031703	-0.0097459	-0.0738715	-0.0315056	0.0216757	-0.0772527	0.2411601	0.2411844	-0.0688683
0	10	0.0084281	-0.0103690	-0.0739335	-0.0320955	0.0359962	-0.1518183	0.2411759	0.2412002	-0.0689500

a) kinematics and compliance for left wheel

SuspF.Kin.0.L.Data		tx	ty	tz	rx	ry	rz	lSpring	lDamp	lStabi
0	0	0.0384716	0.0223798	-0.0751491	0.0585885	0.1198827	0.6150913	0.2405642	0.2405885	-0.0678194
0	1	0.0336798	0.0193203	-0.0748381	0.0511941	0.1061895	0.5335983	0.2407572	0.2407815	-0.0682165
0	2	0.0287860	0.0167143	-0.0745737	0.0450923	0.0923546	0.4545083	0.2409109	0.2409351	-0.0685235
0	3	0.0238052	0.0145336	-0.0743530	0.0401900	0.0784054	0.3772663	0.2410279	0.2410521	-0.0687466
0	4	0.0187479	0.0127576	-0.0741737	0.0364185	0.0643585	0.3013756	0.2411103	0.2411345	-0.0688903
0	5	0.0136209	0.0113721	-0.0740341	0.0337299	0.0502218	0.2263751	0.2411594	0.2411836	-0.0689575
0	6	0.0084281	0.0103690	-0.0739335	0.0320955	0.0359962	0.1518183	0.2411759	0.2412002	-0.0689500
0	7	0.0031703	0.0097459	-0.0738715	0.0315056	0.0216757	0.0772527	0.2411601	0.2411844	-0.0688683
0	8	-0.0021543	0.0095073	-0.0738486	0.0319709	0.0072478	0.0021981	0.2411116	0.2411359	-0.0687114
0	9	-0.0075509	0.0096649	-0.0738660	0.0335249	-0.0073076	-0.0738817	0.2410293	0.2410535	-0.0684768
0	10	-0.0130286	0.0102399	-0.0739259	0.0362298	-0.0220201	-0.1516164	0.2409112	0.2409354	-0.0681601

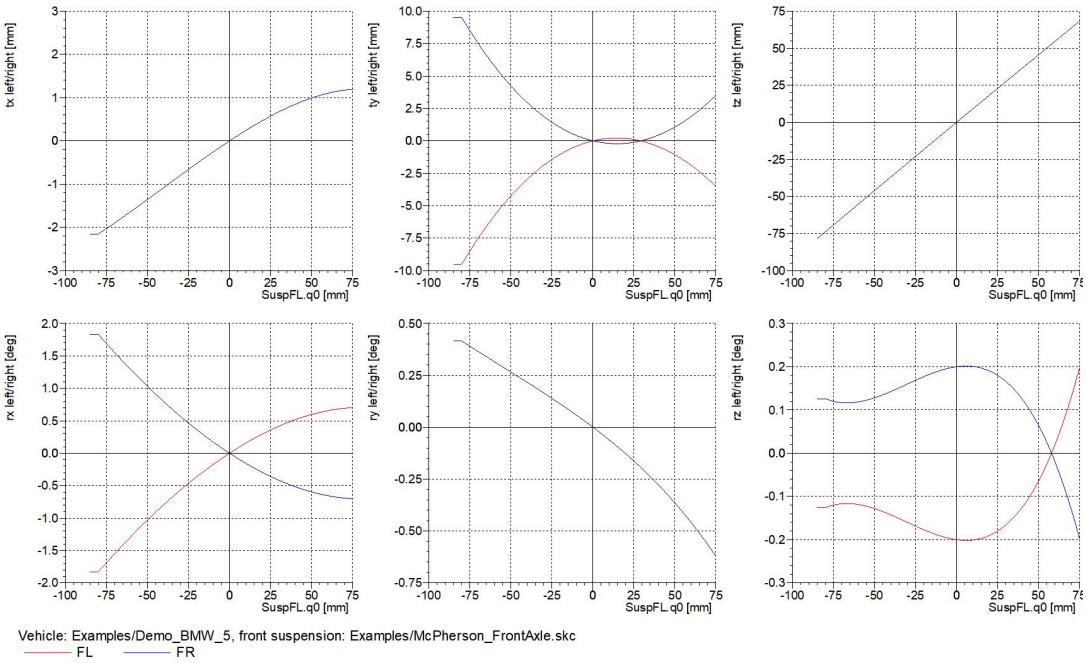
b) kinematics and compliance for right wheel

Figure 87 Kinematics and compliance of front suspension

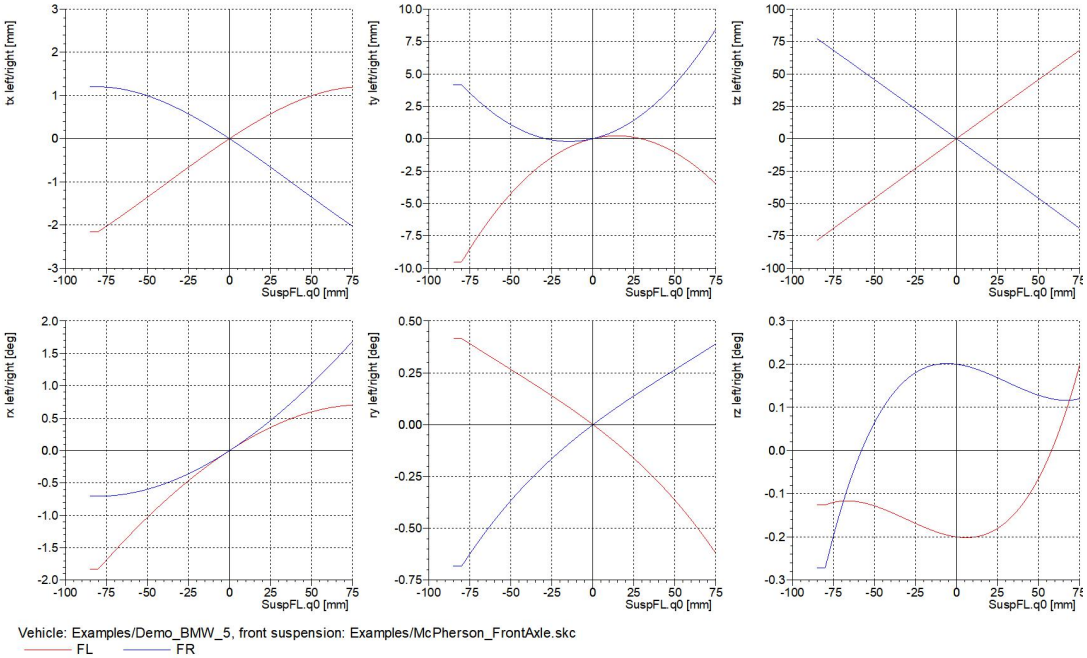
Here tx, ty and tz are translations of wheel center along three directions, while rx, ry and rz are rotations around three axes, namely camber, spin, and toe angles, as shown in the following figures with their positive directions. The other three quantities are deformations of spring, damper and stabilizer, respectively. Unit of translation is mm and that of rotations is deg.

These two tables show the kinematic values and compliance of the 1D+1D mappings depending on the DOFs wheel compression and steering rack displacement for the front axle. In Figure 22, the upper one 'SuspF.Kin.0.R.Data', shows information of the front right wheel, and the lower one shows information of the front right wheel. From the figures, toe angles and camber angles for static rotation can be found, which are under the condition when both Arg0 and Arg1 equal 0. For the left wheel, the static toe angle is r_{zL}=-0.6989466E-01 deg, and the static camber angle is r_{xL}=0.6783830E-03 deg. For the right wheel, the static toe angle is r_{zR}=-0.7612878E-01 deg, and the static camber angle is r_{xR}=0.2215086E-02 deg. And the kinematic gradients are roughly dr_x/dt_z=6.95deg/m, dr_z/dt_z=1176.06deg/m.

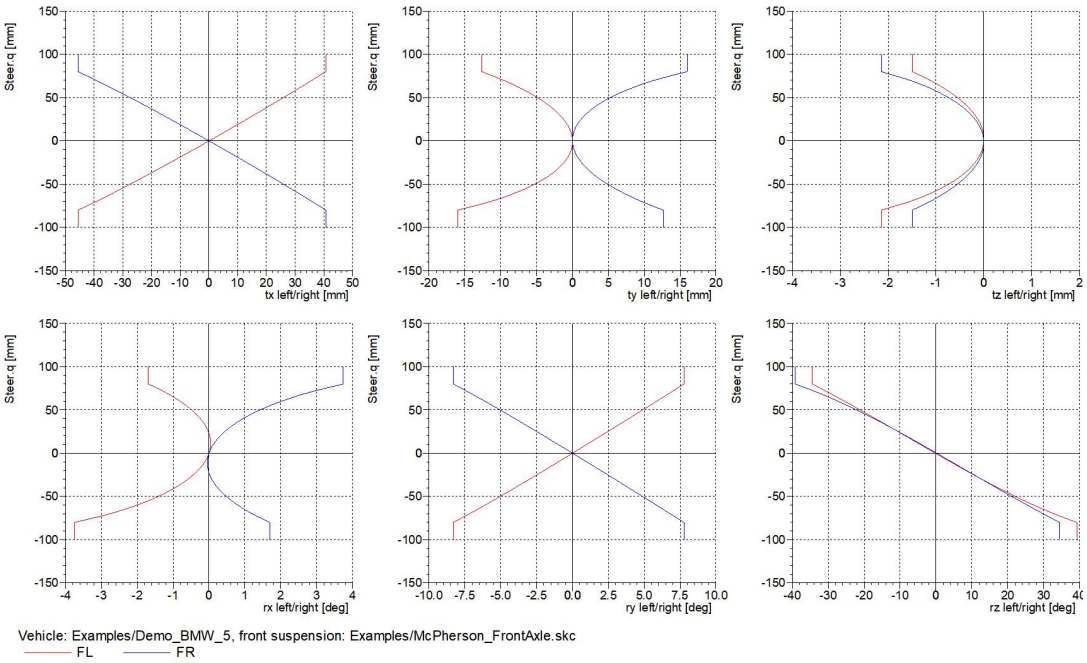
Some plots for front suspension kinematics are shown as follows, q0,q1 and q2 are the generalized coordinates: wheel compression, wheel compression of the opposite wheel and the rack displacement.



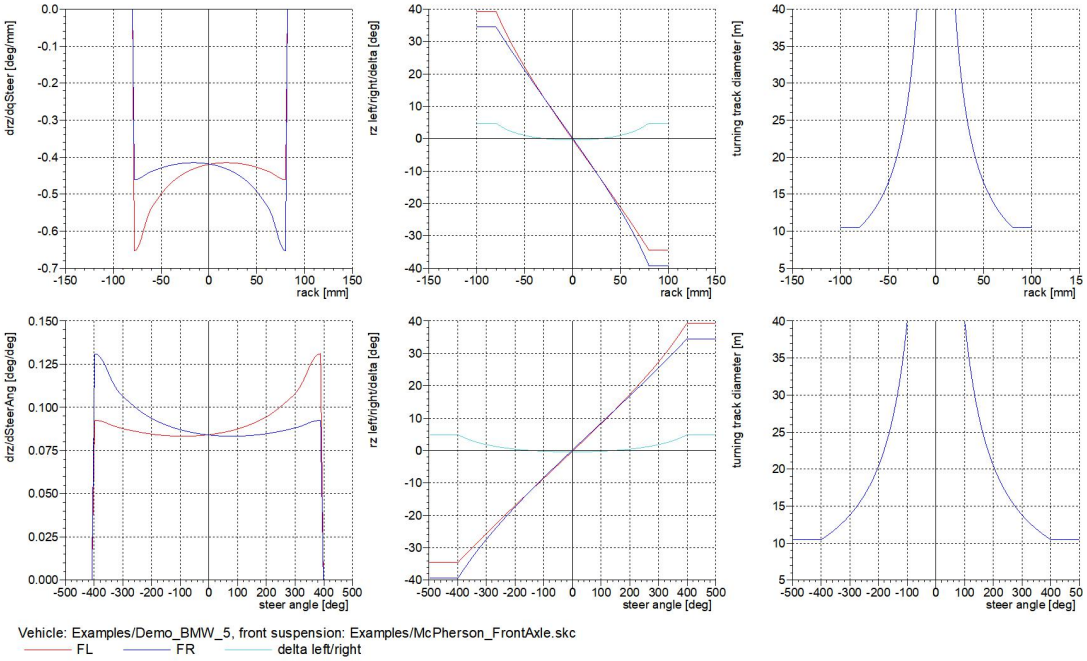
CarMaker 7.1.1 Model Check	Kinematics & Compliance: Parallel Compression Kinematics	



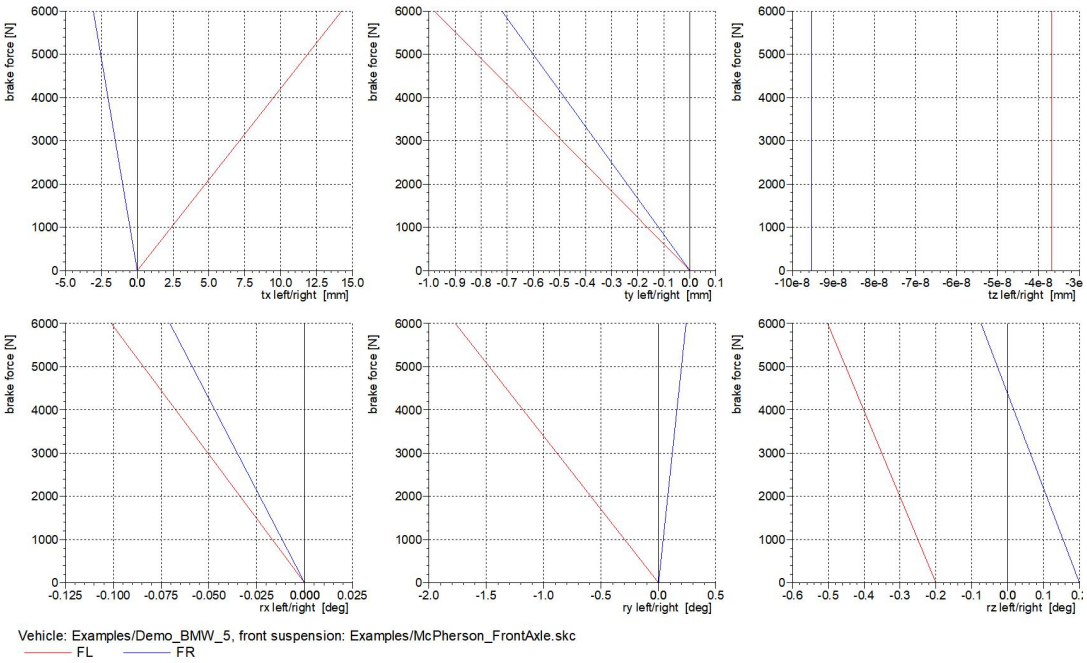
CarMaker 7.1.1 Model Check	Kinematics & Compliance: Antiparallel Compression Kinematics	



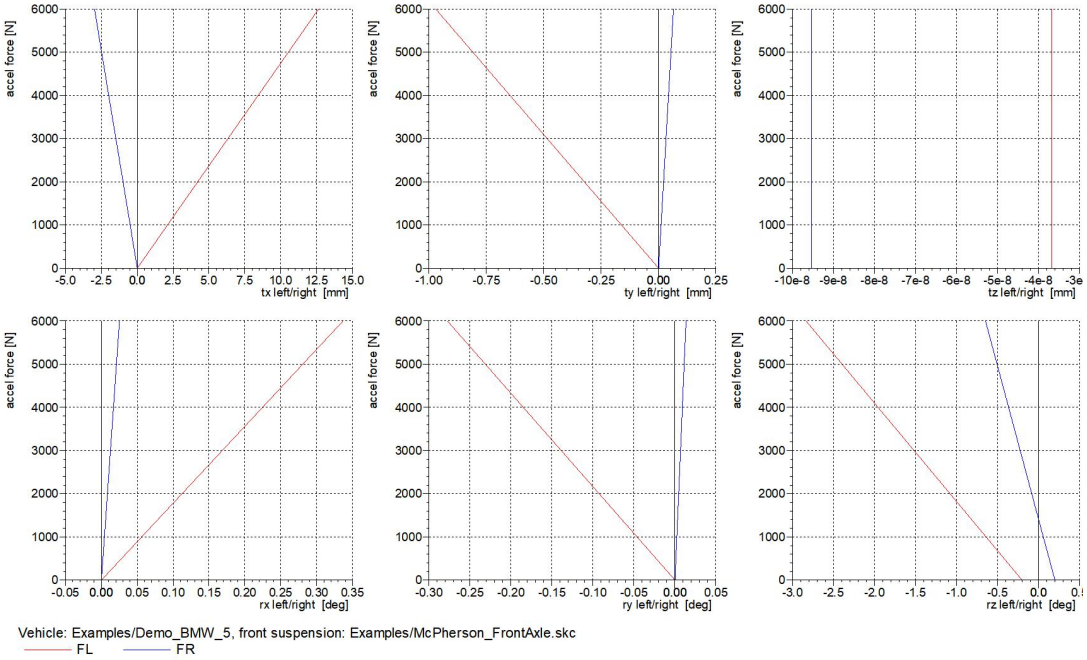
CarMaker 7.1.1 Model Check	Kinematics & Compliance: Steering Kinematics	



CarMaker 7.1.1 Model Check	Kinematics & Compliance: Steering Kinematics	



CarMaker 7.1.1 Model Check	Kinematics & Compliance: Braking	



CarMaker 7.1.1 Model Check	Kinematics & Compliance: Accelerating	

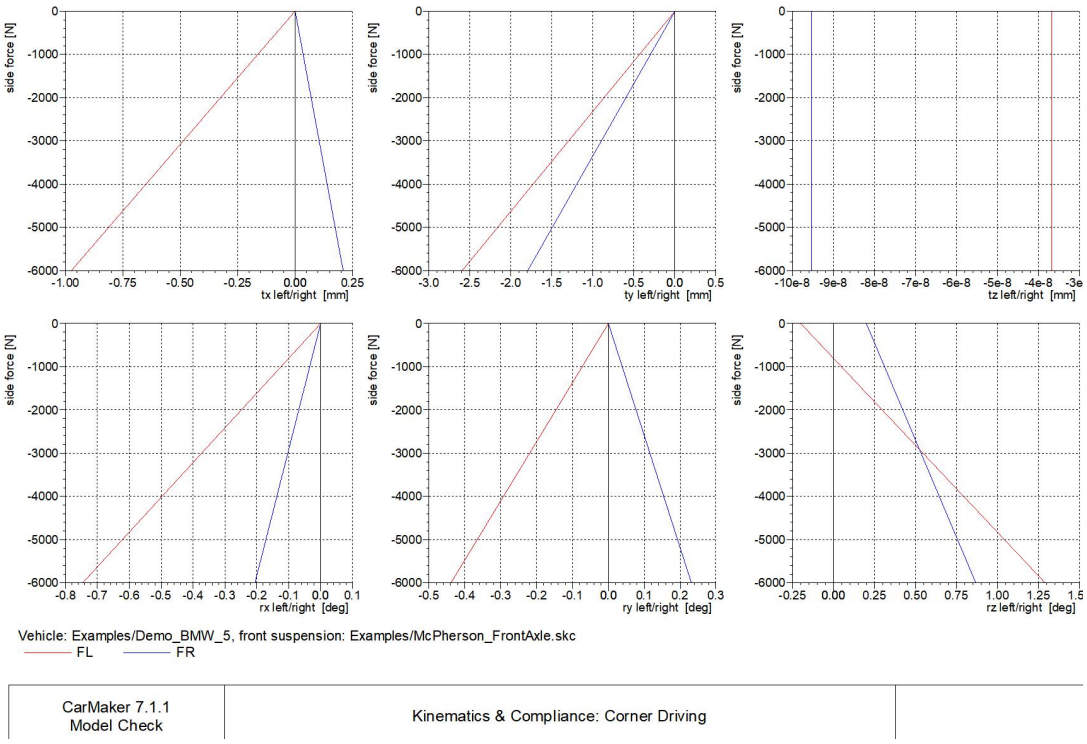


Figure 88 Front suspension kinematics and compliance

While for the rear axle, the relations are defined by a linear 2DOF model.

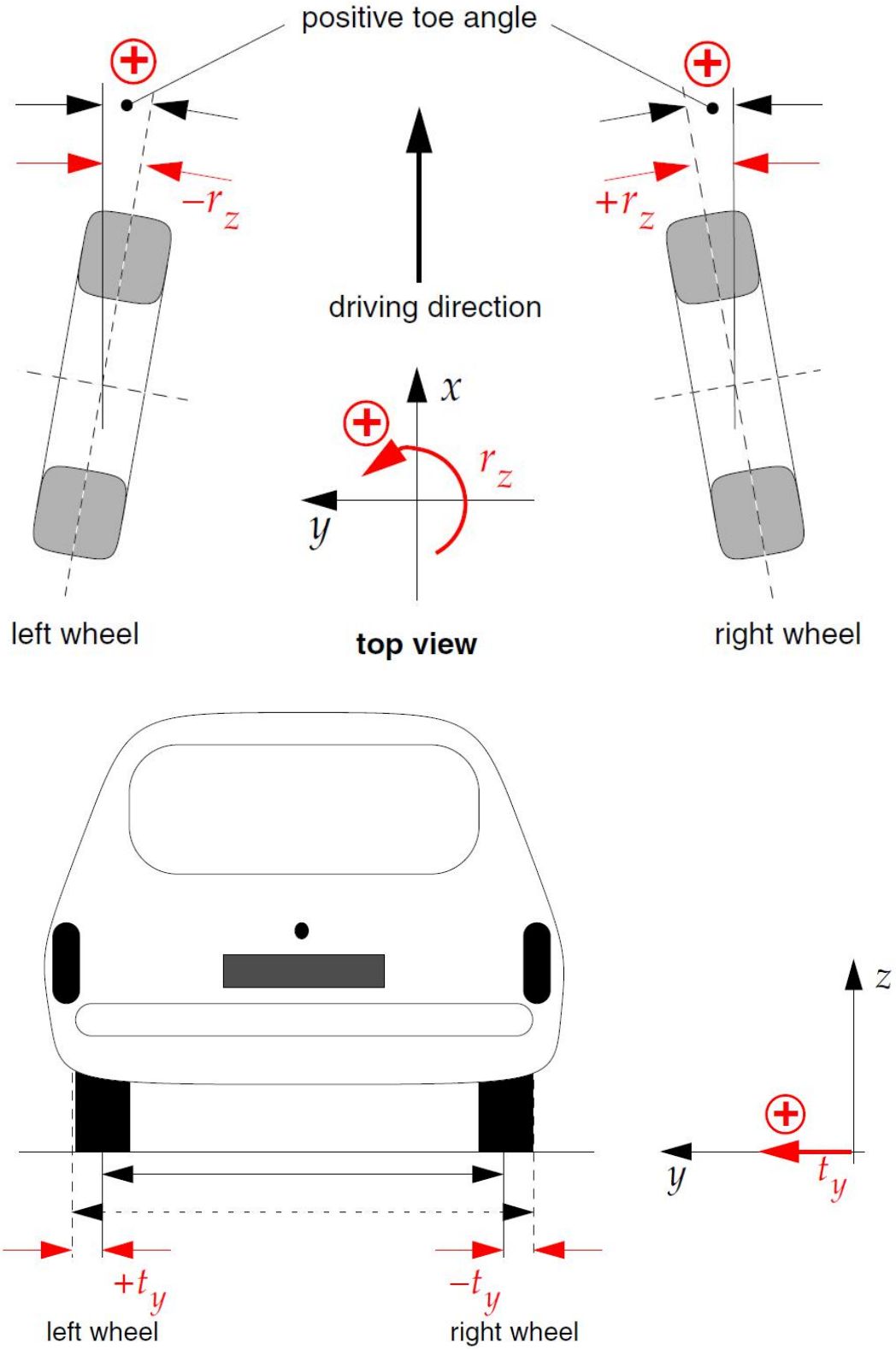


Figure 89 Definition of wheel DOFs

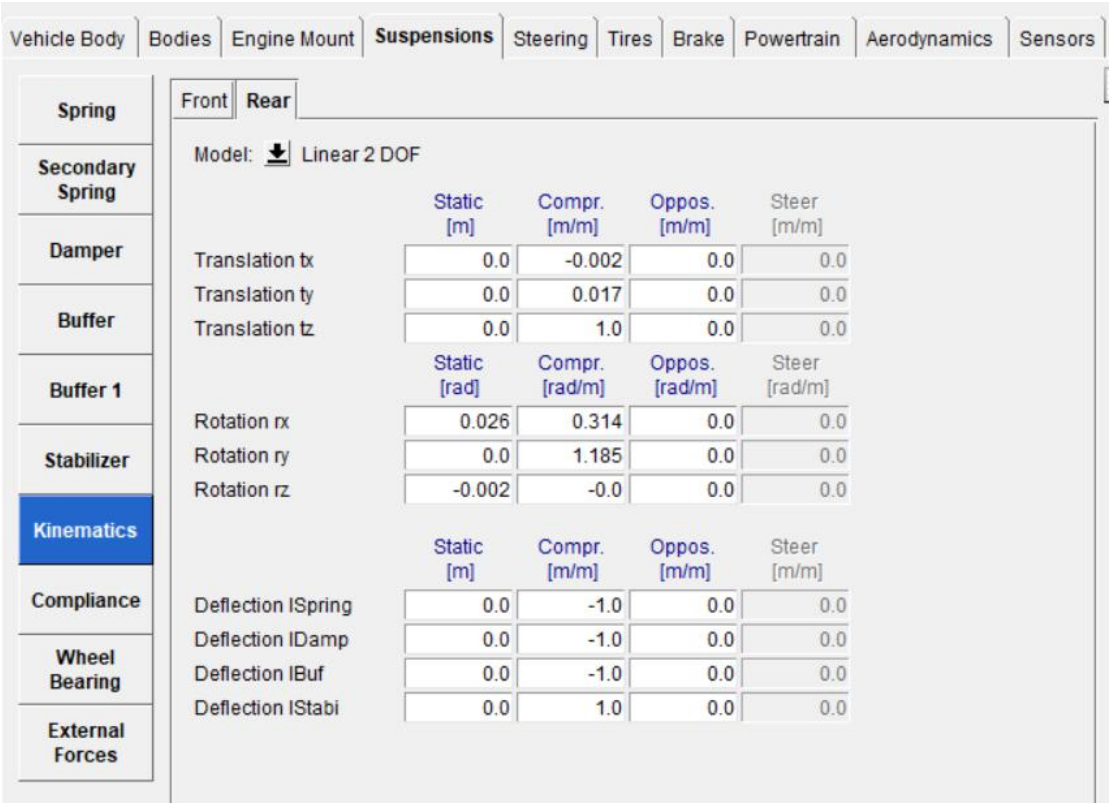


Figure 90 Kinematics and compliance of rear suspension

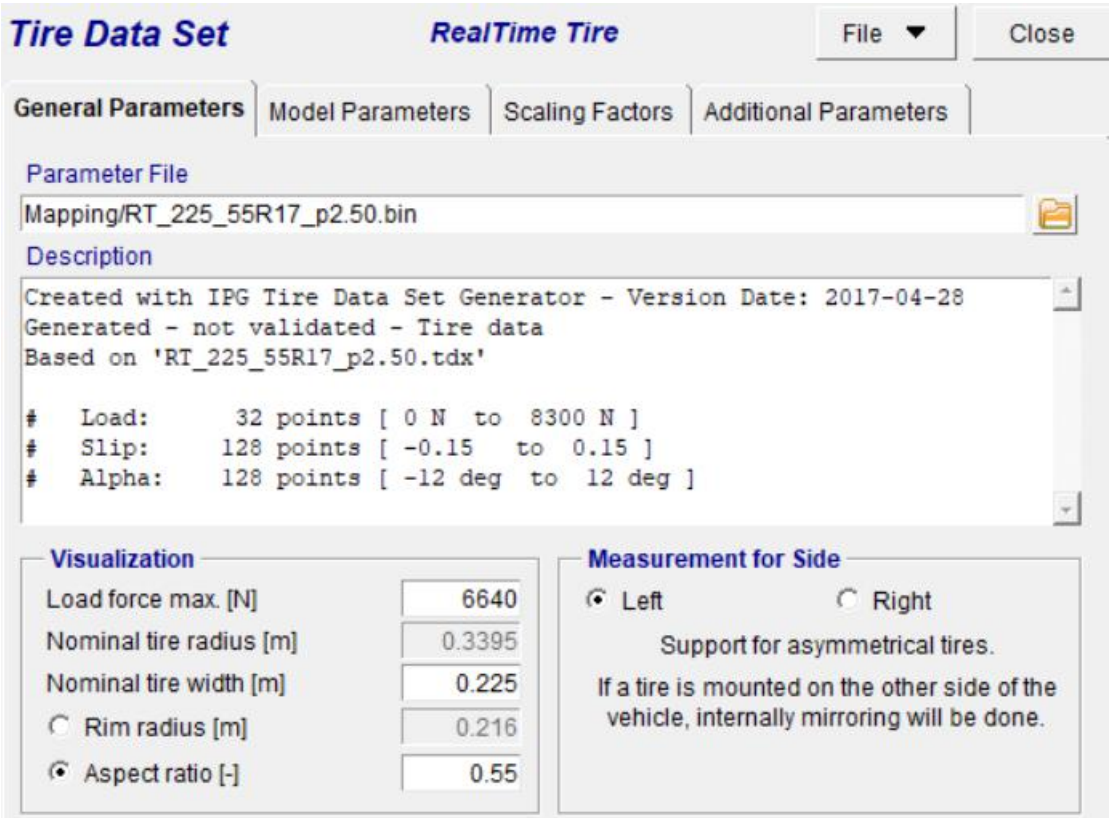
The wheel bearing and external forces are not defined.

Figure 91 Steering gear ratio of the model car

The rack travel to steering pinion angle is a certain value, 86.911rad/m, this number can be used to calculate the steering ratio.

Tires of the model car are selected as shown in the figure.

Figure 92 Tires of the model car



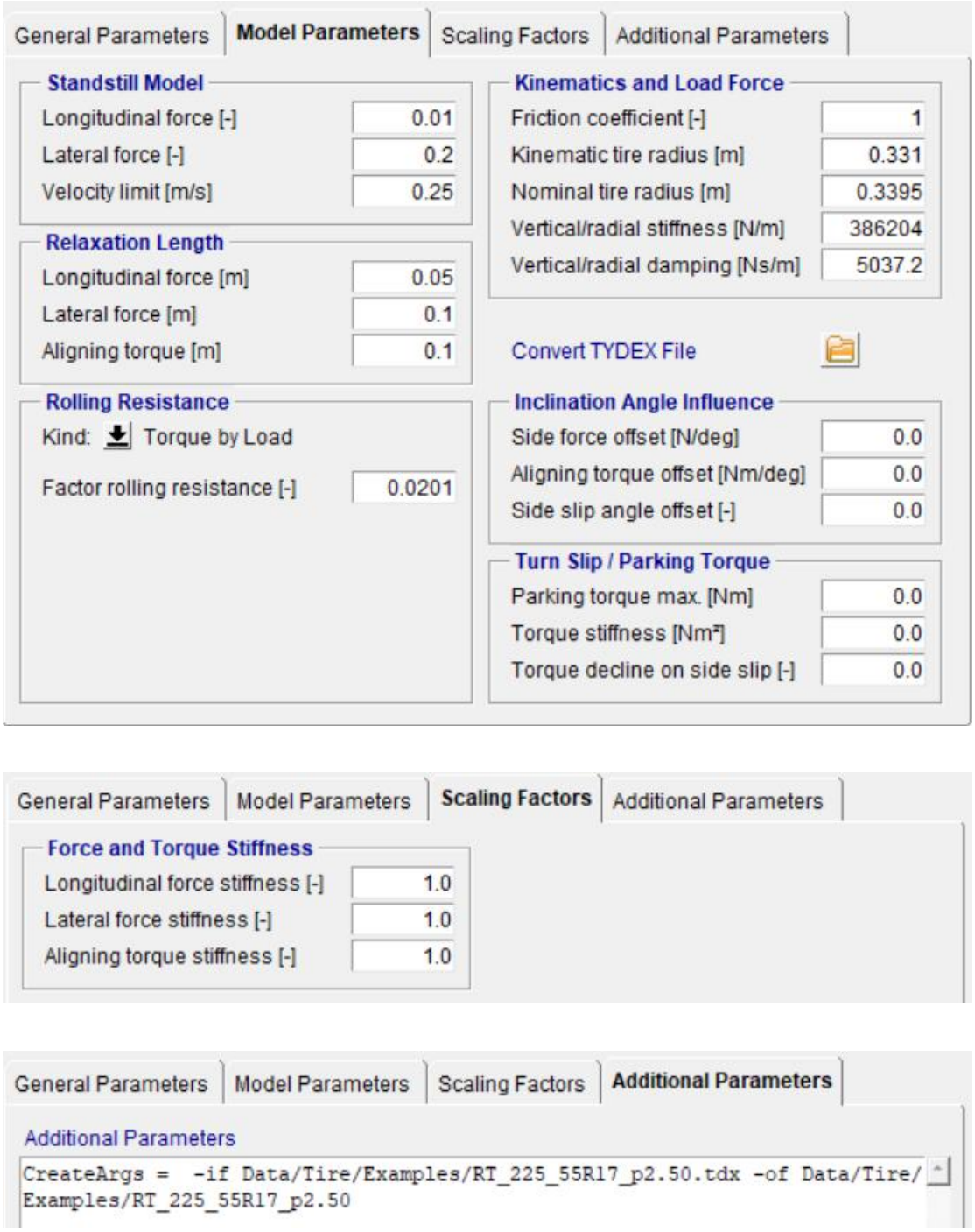


Figure 93 Parameters related to tires

In this figure, all the parameters and properties related to the tire used in this model is defined, such as ranges of load and deformation, dimensions of the tire, resistances, etc.

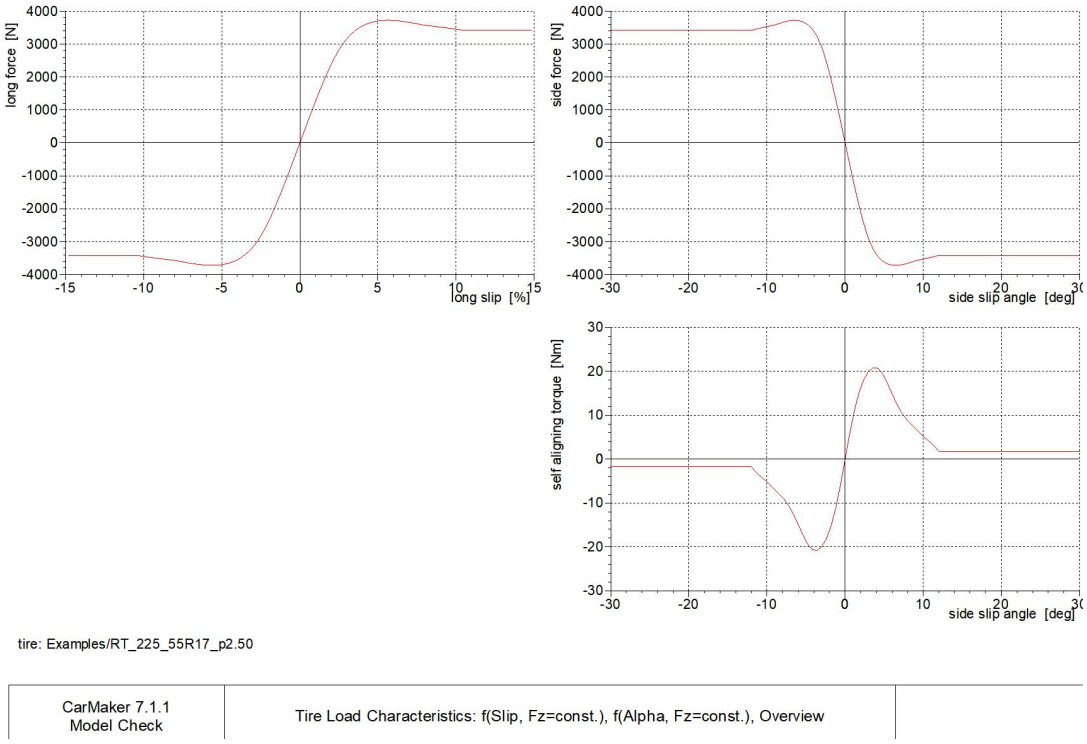
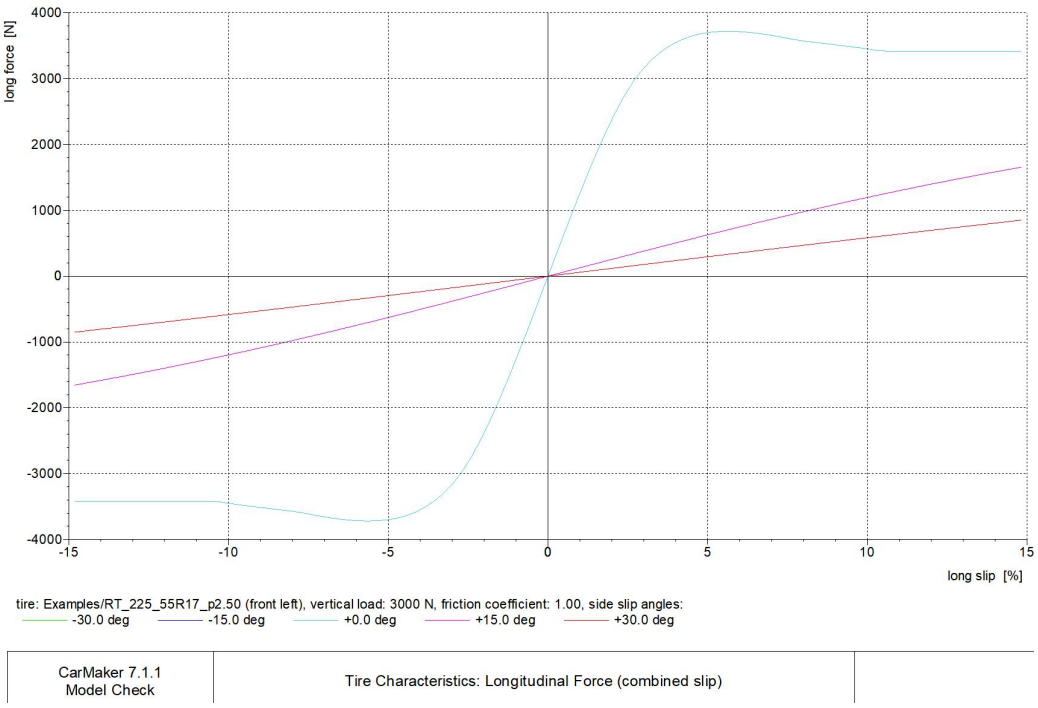
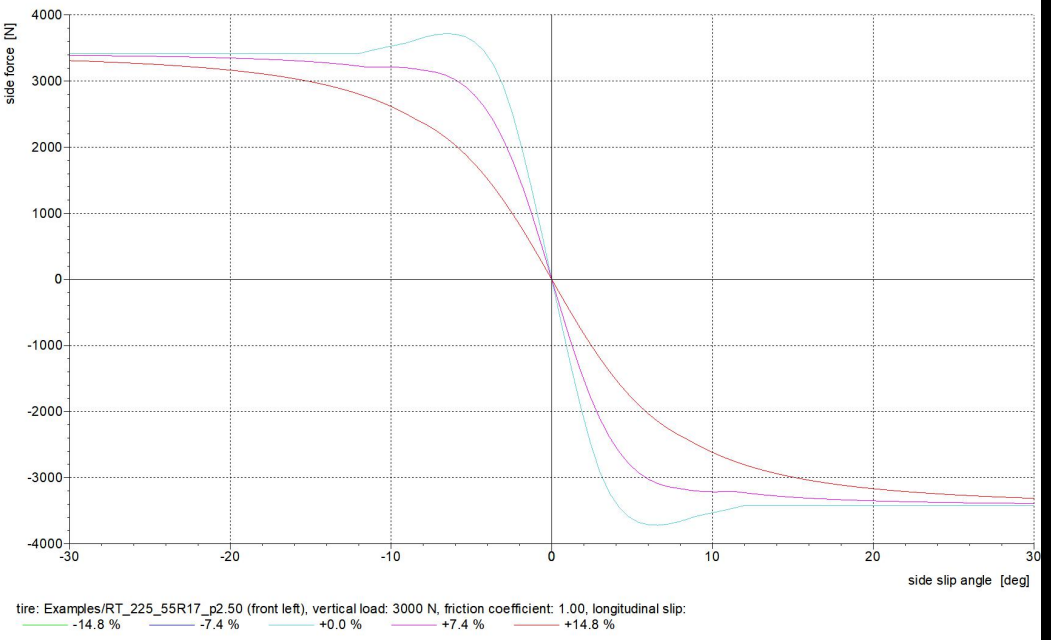


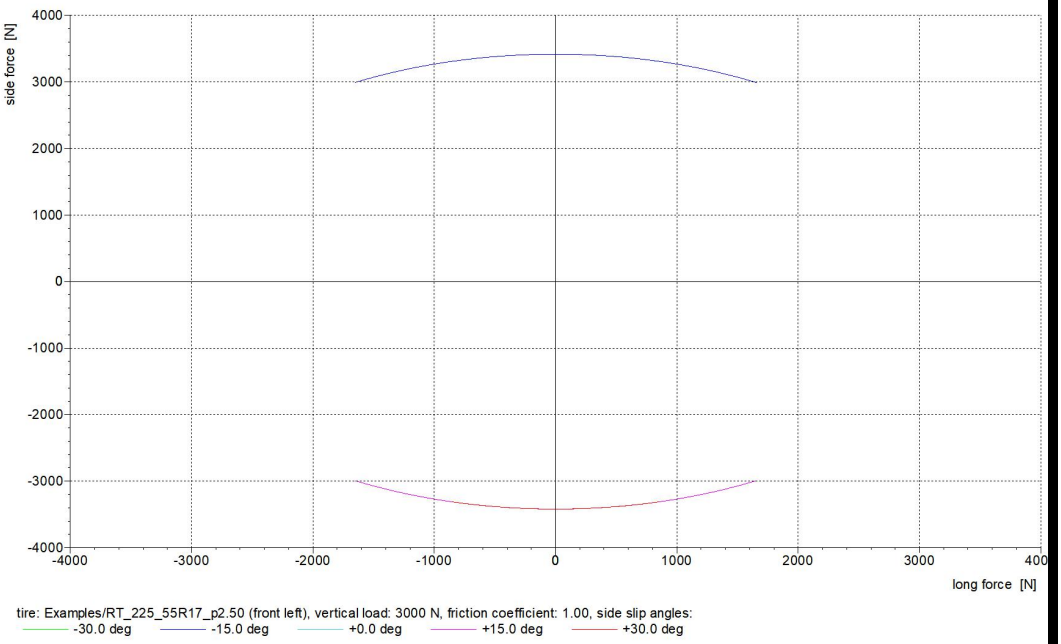
Figure 94 Tire load characteristics

This figure shows relationships between forces along three directions and corresponding deformation.

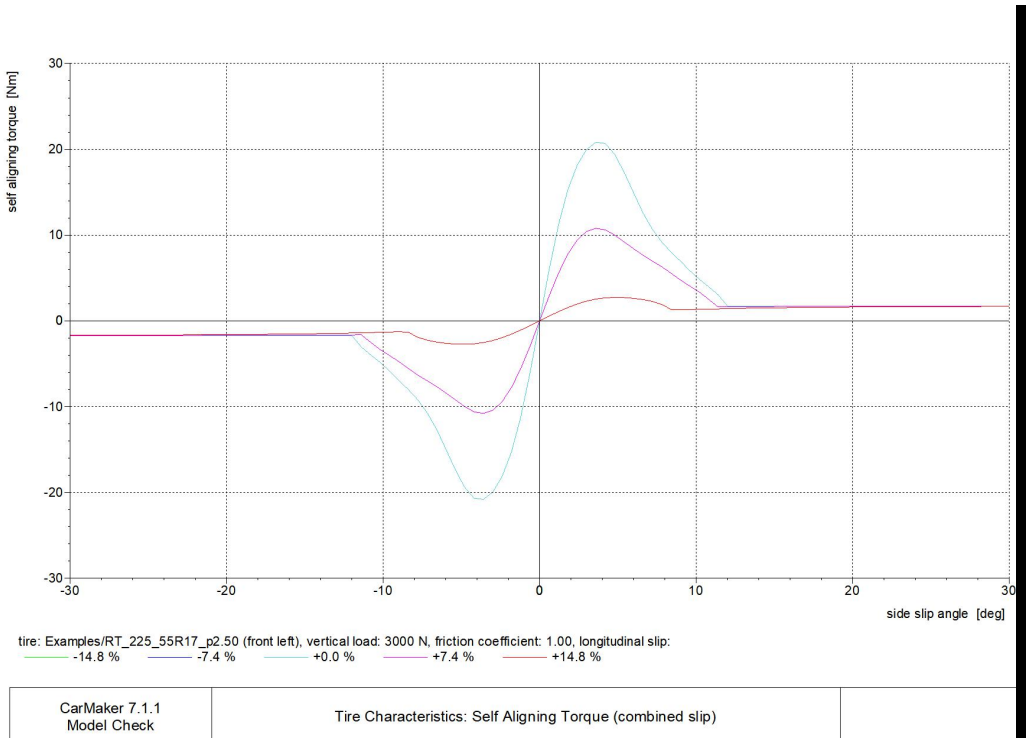




CarMaker 7.1.1 Model Check	Tire Characteristics: Side Force (combined slip)	



CarMaker 7.1.1 Model Check	Tire Characteristics: Side Force vs Longitudinal Force (combined slip)	



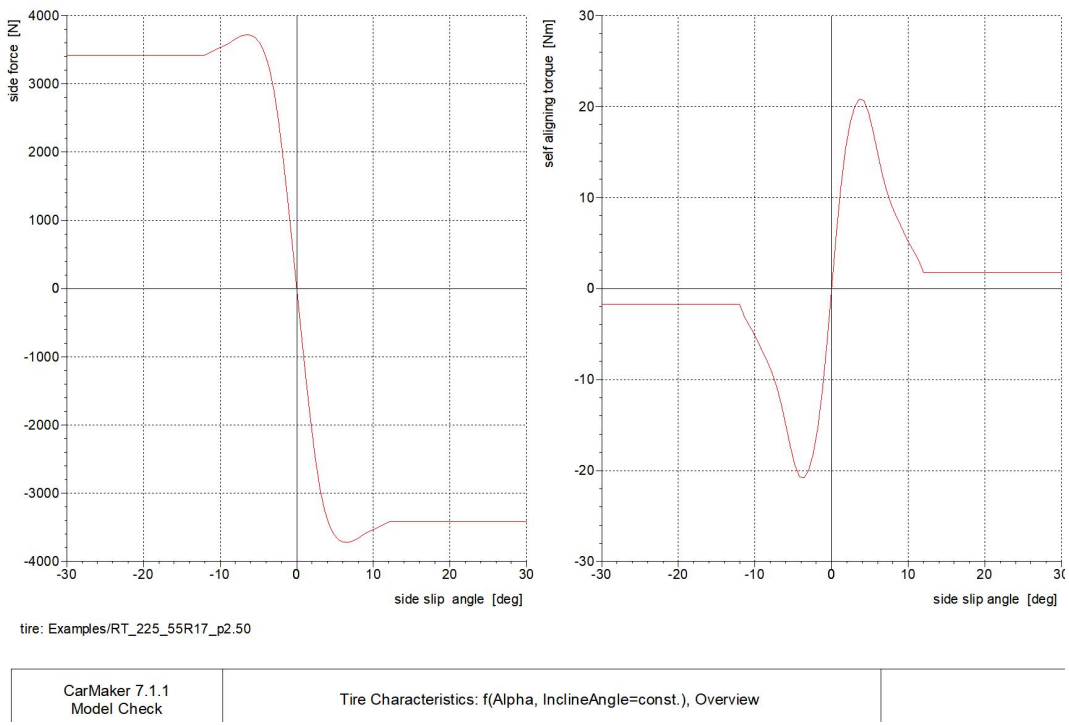
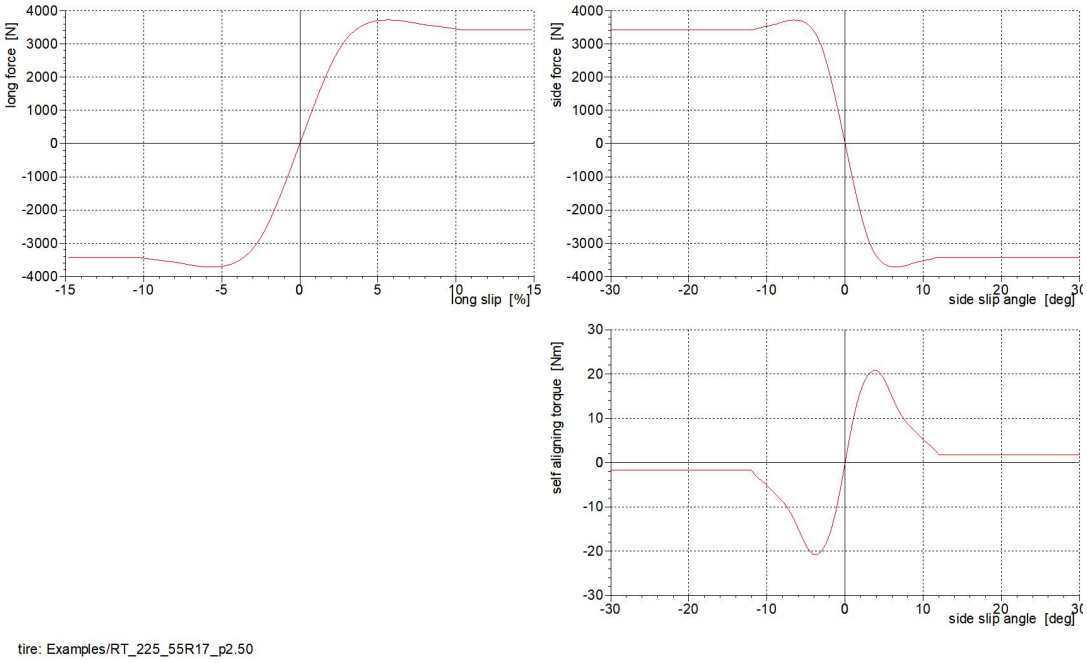


Figure 95 Tire characteristics

These figures show relationships between forces along x- and y-directions, between forces and deformation along every direction, and between self aligning torque and slide slip angle, respectively.



CarMaker 7.1.1 Model Check	Tire Friction Characteristics: f(Slip, mu=const.), f(Alpha, mu=const.), Overview	

Figure 96 Tire friction characteristics

In this figure, all the curves are generated with friction coefficient equal to 1.

Through the model parameter check block, it's possible to generate a text file and several figures besides those for tires.

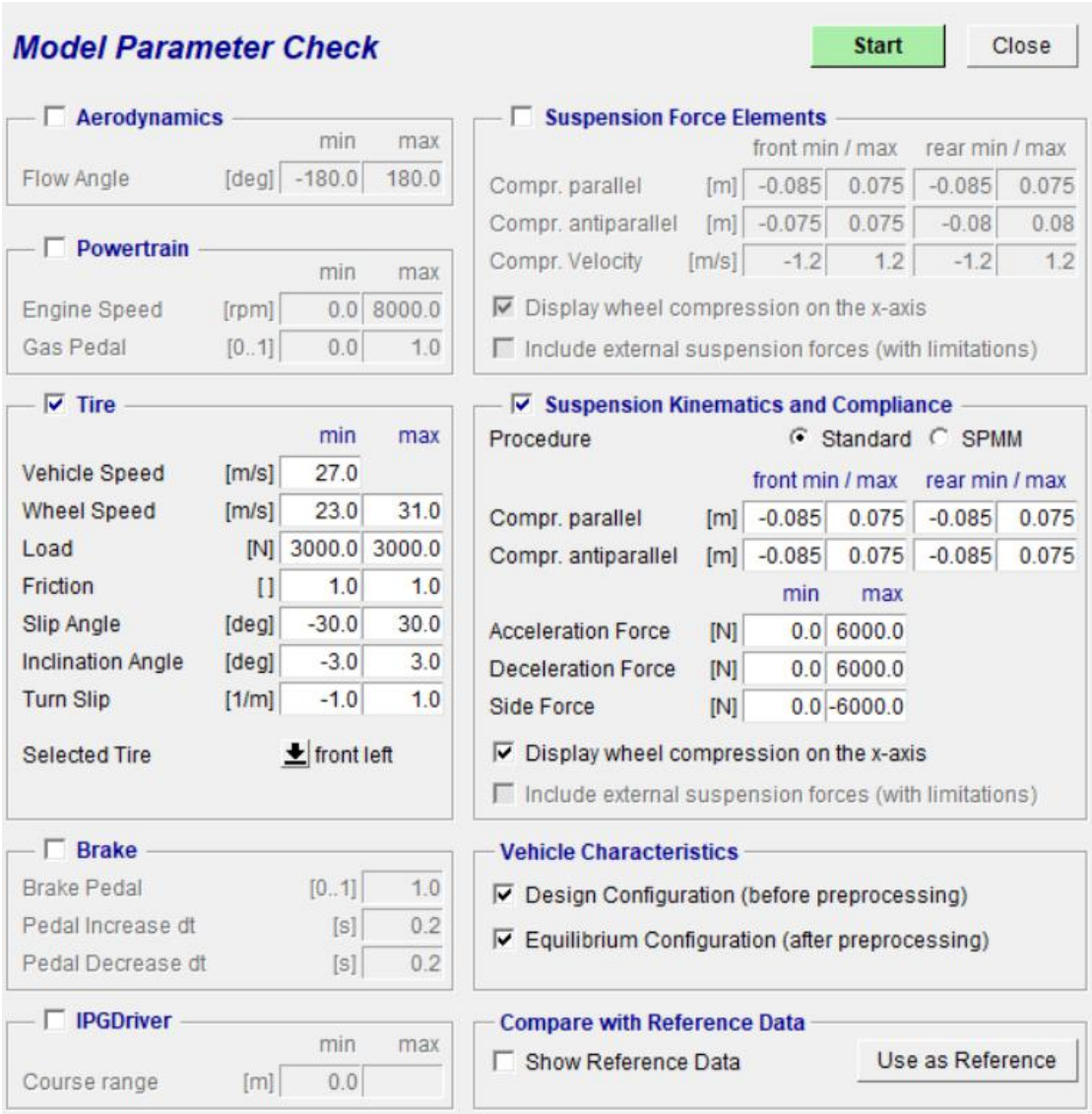


Figure 97 Model check interface

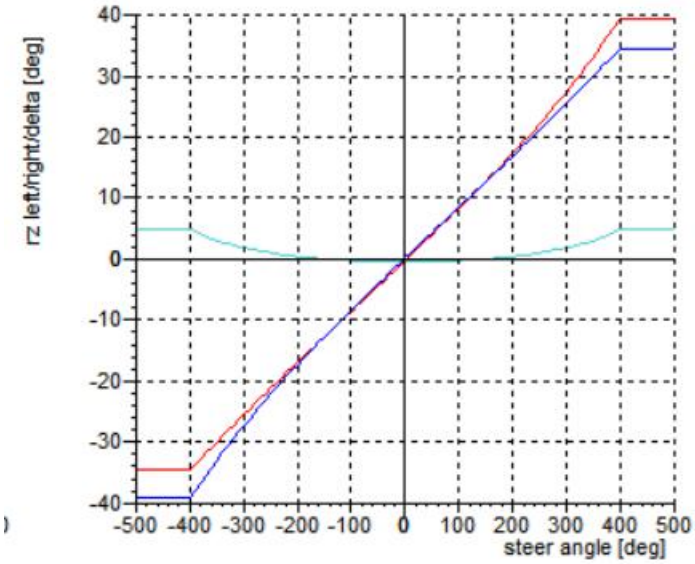


Figure 98 Estimation of steering ratio

Steering ratio is the ratio of steering wheel angle over steering angle at the wheel. It can be calculated as the ratio between steering gear ratio and the ratio of steering angle at the wheel over steering rack travel, where steering gear ratio is the ratio of steering wheel angle over steering rack travel. In this case, it's possible to read the steering gear ratio while another ratio is not show. So it's better to estimate the steering ratio according to the figure. From this figure, the steering ratio which is the ratio of steering wheel angle over steering angle of the vehicle, can be estimated by calculating the reciprocal value of the slope of the curves near the origin. The value is about 12.