HAN_UNIVERSITY OF APPLIED SCIENCES

HAN Minor Project– Advanced Vehicle Dynamics

Handling Model of the V-Tron Toyota C-HR

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Master Engineering Systems

Front Matter

Project Title: Handling Model of the Toyota C-HR

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Summary

This project was done in collaboration with V-Tron who wanted a validated handling model to aid in their quest for development of driver assistance systems. This study was based on a Toyota CH-R vehicle. V-Tron was interested in knowing how the vehicle behaves in transient and steady state situations. The model that was developed as a double track model with three degrees of freedom, namely the yaw, lateral and roll motions. Furthermore, it accounted for non-linear tire behavior. A well-established system modeling approach called the ‘4+1 approach’ was used to develop the model. The model was validated from the data obtained from track testing provided satisfactory results.

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1 Introduction

1.1 Background

Vehicle handling and control is an essential aspect of intelligent driver assistance systems, a building block of the upcoming generation of 'smart cars' (Riener, 2008). A vehicle's handling describes how well it responds to the driver's input. The handling characteristics of passenger vehicles are influenced by several factors such as tire stiffness, load shift, suspension geometry, road conditions etc. Intelligent driver-assistance systems are electronic systems that assist drivers in driving and parking functions. Through a safe human-machine interface, they increase car and road safety. This project is conducted in collaboration with V-Tron who wants to further research on driver assistance systems for their Toyota CH-R. For the development and testing of such assistance systems, vehicle dynamics models are necessary, which take into account all important influences on the vehicle handling with a sufficient accuracy. To explore the dynamic capabilities of vehicles, several software tools have been developed, thus helping engineers to understand the real time vehicle behavior. The basic idea is to build a handling model that allows to perform important dynamic analysis of the vehicle before the construction of a real prototype. A sufficient accuracy is not always guaranteed in the case of simplistic models used in many studies like single track model, where the vehicle center of gravity is assumed to be on road level. A similar single-track model was developed by another HAN student group who faced several issues during their project, producing an end result (model) with little accuracy. It was thus decided to develop a double track model, which considers the vehicle dynamics with the occurring yaw and roll of the vehicle body (Riegl, 2018). In other words, a twin track model that interacts with the ground at four contact points and includes non-linear tire behavior and load transfer. The model does not include the suspension system, thus keeping a reasonable level of complexity. This model is studied for both transient and steady state situations and is validated with data from literature. The double track model (Figure 1) was developed in MATLAB/Simulink.

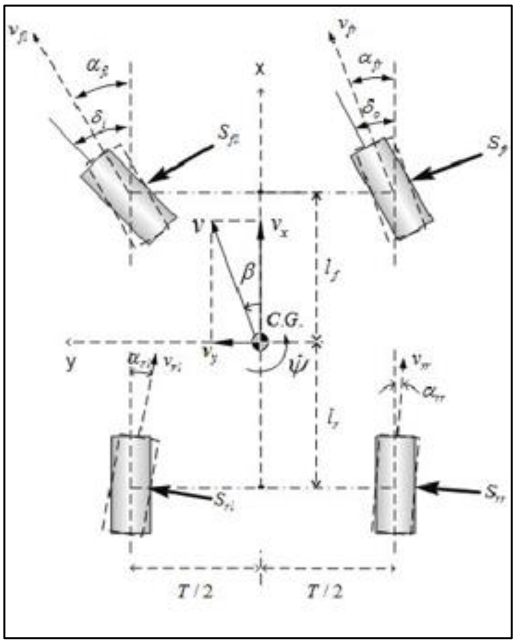


Figure 1: Schematic of double track handling model

***Source**: Publication - 'A novel method for indirect estimation of tire pressure'*

1.2 Problem Definition

Modern cars are moving towards a point where 'smart' technologies are increasingly used to make driving easier by providing several assistances. V-Tron, a firm that shares similar vision, aims to develop a driving assistance system for their Toyota C-HR. The problem is that the behavior of the before mentioned vehicle is unknown in driving conditions and it has to be known for the ADAS to assist the driver. The vehicle behavior cannot be simply perceived, but it can be studied by carrying out physical tests on the vehicle under different conditions and this would take a lot of time and resources. For a good prediction of vehicle behavior and resources saving, the development of a model is needed i.e., a validated double track handling model. This latter is expected to predict the vehicle handling behavior in steady state and transient state conditions.

1.3 Project Objective(s)

The project objective is to create a validated double track model accounting for both transient and steady state handling characteristics of the V-Tron Toyota C-HR with non-linear tire characteristics. The model further takes into account the roll behavior and lateral load transfer.

1.4 Research Questions

In order to achieve the project objectives, several important research questions were formulated. These were thoroughly analyzed and the solutions to these questions along with sufficient activities build the key elements needed to attain the objectives. The research question topics that were formulated are summarized below.

- How to implement the roll behavior and lateral load transfer in the double track model?
- What are the additional parameters that would influence the vehicle behavior?
- How to transform the defined equations of motion into a Simulink model?
- What modifications should be done to make the model results identical to test results?

1.5 Methodology and Approach

Literature review

The first major stride towards the project is to carry out a wide literature survey. This helps to analyze and identify the different vehicle dynamics equations concerning the double track model incorporating non-linear tire behavior, roll behavior and load transfer.

Modeling

The next step is to develop the model itself that encompass the essential core of the project. In order to develop the model, a '4+1' approach is used. This approach mainly consists of 4 steps. These are process definition, drawing the data flow diagram, writing the mathematical equations and development of the model. The model as developed from these equations has the necessary input, output and disturbances.

Testing and validation

Once the model is developed, it is followed by test development for validation. As a groundwork for testing, the DAQ hardware is to be installed in the V-Tron CHR. The right kind of tests are then determined to satisfy the project requirements. The test plan is developed as per the ISO standards. The 'bicycle model' is initially simulated using similar parameters to get a feel for the results. These results are then used as a benchmark for actual simulation for the double track model. The simulation results are then compared with results from these literatures to study the trends and similarities which gives an insight into the accuracy of the model.

2 Literature survey

This literature survey is a representation of the information needed to realize the double track model. This survey contains important information that briefly summarizes the basics for the modeling of a double track model. Subjects such as roll behavior, load transfer, nonlinear tire behavior are also considered and explained.

First and foremost, before moving forward in this project it is necessary to understand the working of different models under normal driving conditions. This paper (Lundahl) gives a comparison between 4 different vehicle models, where three of them are based on a single-track model with different tire models (linear model, Pacejka model and Pacejka model with lag) and the fourth is based on a double track model with load transfer and roll behavior consideration. Testing was performed for steady state circular behavior using constant radius and for transient behavior using a double lane change maneuver. A comparison of results between simulation and actual testing on a Volkswagen Golf IV showed that the Pacejka model (Magic Formula) for the tire behavior is suitable in the limits of normal driving conditions and gives the best result out of the 4 models. The double track model gave better results in comparison with the single-track models (Without consideration of nonlinear tire behavior).

Next step is to find the governing equations of this double track model. A paper by (Berntorp.K, 2013) gives the derivation of six degrees of freedom handling model where a double track model is used, which takes into account of the pitch and roll behaviors. The load transfer was also accounted for. Equations of motion for the 6 DOF were derived and then simplified by neglecting the pitch behavior due to the sole interest in roll behavior. These equations were written by applying the summation of forces acting on the car body in each direction as well as the summation of moments for yaw, roll and pitch. Pacejka models were used for both cases of pure slip and combined slip to calculate the lateral forces developed at the wheels level. The formulae for lateral tire forces were written as a function of vertical load and slip angle of the tire.

To gain further knowledge on roll behavior, this paper (Parczewski, 2017) talks about how controllability and stability of vehicle motion is influenced by the stiffness and damping of suspension elements, suspension kinematics motion and alter the wheel angles. The solutions of design for suspensions affect the load and wheel inclination while performing various vehicle maneuvers. The tire deflection was omitted. If the tire deflection is to be considered, then it will result in an increase in roll angle by a few percent for trucks to several percent for passenger cars and further increasing the complexity. Roll center and roll angle even depends on the kind of suspension used in the vehicle. The largest impact on roll angle is caused by the lateral acceleration, followed by roll center, and then roll stiffness.

The paper (Abe, 2017) talks about how the vehicle behaves under cornering if its roll axis is varied. During cornering, load at one side of the vehicle increases while it decreases on the other side. The roll center and roll moment have to be in equilibrium at the front and rear wheels in the plane perpendicular to the vehicle longitudinal direction. If the roll center height varies then the load transfer might be small or large depends on the difference between COG and roll axis.

Key points extracted from the literature survey

a. The paper by Mr. Lundahl gives purpose for considering double track model over the single-track model as it is a valid model and also provides quality results.
b. From Mr. Berntorp's paper 'Derivation of a six degrees-of-freedom ground-vehicle model for automotive applications' the equations of motions which govern the double track model has been borrowed and contemplation has been made on several assumptions. Longitudinal force, lateral force and yaw moment has been considered. Pitch motion was rejected because it was outside the scope of the project.
c. 'The influence of vehicle body roll angle on the motion stability and maneuverability of the vehicle', gives the formulas for the load transfer and the roll angle. For determination of roll angle, consideration has been made for centre of mass of the vehicle, roll stiffness at the front and rear axles of the vehicle. For load transfer - roll angle and height of roll centre has been reckoned.
d. The paper by Mr. Lundahl and the paper of Berntorp, reveal the accuracy of the Pacejka model in combination with the combined slip theory. From these two papers the decision was taken to adapt the nonlinear tire model.

3 Modeling

The modelling for the V-Tron Toyota CHR was done to be able to predict the vehicle behavior and use the results for development of ADAS system. To be able to get more realistic results, Double track handling model was developed taking in consideration the load transfer and the roll behavior of the vehicle. To replicate the non-linear tire behavior, the tire forces were derived using the MF Tire model for combined slip conditions.

The parameters of the vehicle are mentioned in the Table 1.

Parameters	**Value**
Wheelbase (l)	2.64 [m]
Front Track Width (t1)	1.55 [m]
Rear Track Width (t2)	1.57 [m]
Front Left Wheel Load (m11)	460 [kg]
Front Right Wheel Load (m12)	436.5 [kg]
Rear Left Wheel Load (m21)	322.5 [kg]
Rear Right Wheel Load (m22)	298.5 [kg]
Total Mass of Vehicle (M)	1517.5 [kg]
Steering Ratio (sr)	13.6 [-]
Tire Specifications	Michelin Primacy 3 225/50 R18 95V

Table 1: Vehicle Parameters

3.1 Center of Gravity calculation (CoG)

From the mentioned vehicle parameter, the position of the center of gravity could be calculated to be used in the modeling using the following equations:

$$a = \frac{(m21 + m22) \times l}{M}$$

$$b = \frac{(m11 + m12) \times l}{M}$$

For the location of COG in the lateral direction the following equation was used:

$$w = \frac{(m_{11} - m_{12})\left(\frac{t_1}{2}\right) + (m_{21} - m_{22})\left(\frac{t_2}{2}\right)}{M}$$

'w' is the offset of the lateral COG position from the vehicle longitudinal mid-plane. A negative sign shows that the center of gravity is located to the right side of the longitudinal centerline of the vehicle and a positive value indicated the same towards left.

The height of the center of gravity of the vehicle was not possible to calculate. The value used in the project is an estimated value obtained from research study of experiments done on other vehicle of similar class.

The position of the center of gravity of the vehicle was obtained as follow.

Longitudinal Position from the front axle (a)	1.0804 [m]
Longitudinal Position from the rear axle (b)	1.5596 [m]
Lateral Position from midplane (w)	0.0244 [m]
Vertical position from ground (h)	0.5 [m]

Table 2: Position of centre of gravity

3.2 Calculation of Yaw Moment of Inertia (J)

Yaw Moment of inertia of the vehicle was estimated by considering the vehicle as a cuboid. The moment of inertia of a cuboid is given by:

$$J = \frac{M(W^2 + L^2) \times 10^{-6}}{12} = 3144.5\ kg.m^2$$

3.3 Process Definition

To define the process, the inputs and outputs are defined, in addition to the goal of the modeling. This model does not consider any disturbances and various kinds of sensors can be implemented to measure the inputs and outputs. The goal of this model is to predict with good accuracy, the behavior and values of the outputs. The inputs and outputs are listed below:

3.3.1 Inputs

1. Vehicle longitudinal velocity
2. Steering wheel angle

3.3.2 Outputs

1. Lateral acceleration
2. Yaw rate
3. Roll angle
4. Lateral velocity

3.3.3 Sensors

As stated earlier some sensors are used for measuring the input and outputs (mainly required for the testing and validation phases of the project). These sensors be defined (in the process definition part of the modeling approach) with their respective variable. Table 3 defines the used sensors for this project.

Type of sensor	**Variables measured**
Accelerometers	Longitudinal and lateral acceleration
Yaw rate sensors	Yaw rate and angular velocity
GPS sensor	Vehicle position, roll angle, body slip angle

Table 3: Measurement sensors

3.4 Data Flow Diagram

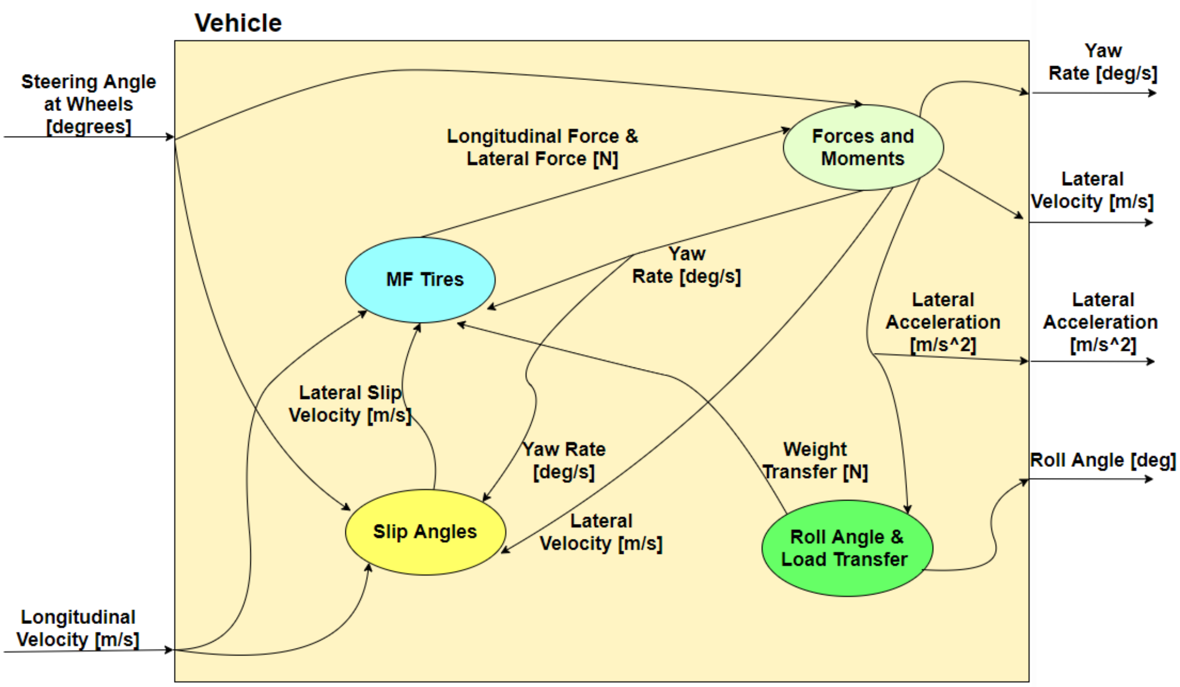


Figure 2: Data flow diagram for double track model

An elaboration of how to get the outputs from the inputs is described in addition to a complex data flow diagram (**Error! Reference source not found.**). Starting with the longitudinal velocity and steering a ngle at the steering wheel, the longitudinal velocity is converted into a wheel angular speed and fed to the nonlinear tire model with addition to other parameters and variables. The steering angle at the steering wheel is transformed into the steering angle of the wheels and fed to the Forces and Moment subsystem. The nonlinear tire model gives the forces at each wheel, the latter are fed to the Forces and Moment subsystem to calculate the lateral acceleration, the yaw rate and other outputs. The two inputs are also fed to the Slip Angles subsystem to calculate the slip angles of each wheel which is used to get the lateral slip speed of each wheel and fed to the nonlinear tire model. The lateral acceleration calculated from the Forces and Moment subsystem is fed to the load transfer and roll behavior subsystem to get the roll angle of the vehicle.

3.5 Equations of motion & MATLAB/Simulink model

The model has 3 degrees of freedom involving the lateral, yaw and roll motions. The modeling was done in 4 subsystems, the “slip angles” subsystem, the tire model subsystem, the “forces and moment” subsystem and the “load transfer and roll behavior” subsystem. Modeling was done using MATLAB and Simulink. Each part will be discussed in this section of the report.

3.5.1 Slip Angles Subsystem

This subsystem calculates the slip angle of each wheel following the below formulas:

$$\alpha_1 = \arctan\left(\frac{v_y + a\gamma}{v_x - \frac{t1}{2}\gamma}\right) - \delta \qquad \alpha_3 = \arctan\left(\frac{v_y - b\gamma}{v_x - \frac{t2}{2}\gamma}\right)$$

$$\alpha_2 = \arctan\left(\frac{v_y + a\gamma}{v_x + \frac{t1}{2}\gamma}\right) - \delta \qquad \alpha_4 = \arctan\left(\frac{v_y - b\gamma}{v_x + \frac{t2}{2}\gamma}\right)$$

Where $\alpha_1, \alpha_2, \alpha_3, \alpha_4$ are respectively the slip angles of the front left, front right, rear left and rear right wheels. $v_x, v_y, \gamma, \delta, t1$ and $t2$ are respectively the longitudinal speed, the lateral speed, the yaw rate, the steering angle of the wheel at the front and rear track widths. These blocks have the longitudinal and lateral velocities as inputs in addition to the vehicle yaw rate and the steering angle. The outputs of the block are respectively the slip angle of each wheel.

The calculation of the slip angles is essential to calculate the lateral slip velocities (V_{sy}) of the tires which is given by:

$$V_{sy} = V_x \times \tan(\alpha)$$

3.5.2 Tire Model Subsystem

In order to develop the tire model subsystem, MF Tire: CPI block was used. MF Tire as the name suggests stands for the standard Pacejka Magic Formula for tires. It provides fast and robust road-tire contact forces and moments which can be effortlessly simulated on the Simulink platform. The Pacejka model is regarded as a reliable model extensively tested and validated for various conditions.

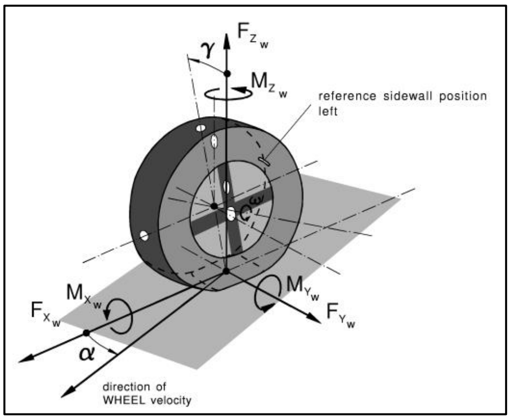


Figure 3: Sign convention

The MF Tire model uses ISO sign conventions as depicted in Figure 3. Detailed ISO force and moments and wheel slip are explained in Appendix -3. The model uses 'Tire Property File' which comprises all the parameters required for the tire. Some of these parameters were changed in accordance with the tire used. These include the tire physical specifications, nominal Load, inflation pressure etc. These tire spec are provided in Table 1. The nominal load on the tires were taken from the average load on all four tires. There are also a number of other essential parameters that influence the tire behavior. As these parameters are not easily available and can only be determined through several experiments, scaling factors were used as an alternative. These scaling factors were tuned to match the results from the test data to generate results as close to the actual results.

Four such tire blocks were used to simulate the behavior of all four wheels. These blocks have inputs of wheel angular velocity, longitudinal speed, lateral slip velocity, vehicle yaw rate, camber angle and normal load and forces in the X and Y direction as output. The key function of these blocks was thus to get the forces at the tire contact patch.

3.5.3 Forces and Moment Subsystem

The forces from each tire that was obtained using the tire model was used as input for this subsystem. The lateral and longitudinal forces on all four wheels were used to calculate the total force which is acting at the center of gravity of the vehicle.

This subsystem accounts for the 3 out of total 4 degrees of freedom studied for the vehicle. The subsystem is based on 3 key equations as follows:

$$\sum F_x = M(\dot{v}_x - v_y\gamma)$$

$$\sum F_y = M(\dot{v}_y + v_x\gamma)$$

$$\sum M_z = I_x\dot{\gamma}$$

Where the F_x, F_y and M_z are the longitudinal force, lateral force and yaw moment respectively. M, v_x, v_y, γ and I_z are the vehicle mass, longitudinal velocity, lateral velocity, yaw rate and moment of inertia respectively. F_x, F_y and M_z can be derived for the forces at the individual tires using the following equation:

$$\sum F_x = F_{x11}\cos\delta + F_{x12}\cos\delta + F_{x21} + F_{x22} - F_{y11}\sin\delta - F_{y12}\sin\delta + F_f + F_w$$

$$\sum F_y = F_{y11}\cos\delta + F_{y12}\cos\delta + F_{y21} + F_{y22} + F_{x11}\sin\delta + F_{x12}\sin\delta$$

$$\sum M_z = (F_{y11}\cos\delta + F_{y12}\cos\delta)a - (F_{y21} + F_{y22})b$$
$$+ (-F_{x11}\cos\delta + F_{x12}\cos\delta - F_{x21} + F_{x22})\, t/2$$

Thus, the forces and moments subsystem have inputs as the forces in the X and Y direction (in addition to the steering angle of the wheels) which then calculates lateral acceleration, lateral velocity and yaw rate.

3.5.4 Load Transfer and Roll Behavior Subsystem

To add the load transfer and the roll behavior to the vehicle model, certain parameters like roll stiffness of front ($k_{\phi f}$) and rear ($k_{\phi r}$), height of roll center axis from vehicle center of mass (h_s) has to be considered. From the centrifugal force acting on the vehicle due to the lateral acceleration of the vehicle, the roll angle of the vehicle could be derived using the following equation:

$$\phi = \frac{\ddot{y}Mh_s}{k_{\phi f} + k_{\phi r} - Mh_s}$$

Where 'M' is the total mass of the vehicle.

The load transfer due to this roll angle at the front and the rear axle can be defined as ΔW_f and ΔW_r. This load transfer can be defined with the help of the roll angle (ϕ) as:

$$\Delta W_f = \frac{\ddot{y}M}{d_f}\left(\phi + \frac{b}{l}h_f\right)$$

$$\Delta W_r = \frac{\ddot{y}M}{d_r}\left(\phi + \frac{a}{l}h_r\right)$$

Here h_f and h_r are the roll center axis height at the front and rear respectively. Also d_f and d_r are the front and rear treads respectively. The obtained load transfer is then used to change the load inputs to the tire model and the whole system run in closed loop.

The input of this subsystem is the lateral acceleration and other parameters specific to the vehicle and the outputs are the load transfer value at each wheel and the roll angle. This is a simplified version for the roll behavior of the car. More realistic version is much more complex, and a lot of other parameters are needed. For example, the suspension parameters.

4 Testing

For the vehicle test, the Toyota CH-R was taken to RDW test track in Lelystad. Two types of tests were performed, where multiple methods were used to obtain the test data. Tests will be performed as per ISO standards. The exact procedure will be followed for both tests. For the steady state circular behavior, the ISO 4138 fourth edition (2012) is followed, and the ISO 7401 third edition (2011) is followed for the lateral transient response test. Detailed procedure is explained in Appendix – 1

5 Results and Validation

5.1 Data Processing and Filtering

The double track handling model was simulated to evaluate the results and to assess the accuracy of the model. The data obtained from the testing was inclusive of noise and unwanted signals, hence it was an utmost necessity to filter the data for accurate analysis. A standard 'Butterworth filter' based on ISO 15037-1 standard is utilized.

Butterworth filter was utilized as it generates less peaking. The requirement to eliminate all the peaking from the filter is conservative (Ellis, 2012). Cut-off frequency is specified as a scalar or a two-element vector. The cut-off frequency, W_n must be in the range of 0.0 < W_n < 1.0, with 1.0 corresponding to half the sample rate. For digital filters, the cut-off frequencies must lie between 0 and 1, where 1 corresponds to the Nyquist rate, which is half the sample rate. To avoid phase shifts in the graph "filtfilt" command has been utilized.

A low pass butterworth filter was used in this particular case. All low pass filters have a certain cut-off frequency, above which the output voltage drops below 70.7% of its input voltage. The frequency at which the magnitude response is 3 dB lower than the value at 0 Hz, is known as cut-off frequency of a low pass filter. Hence, 0.707 value has been opted as a cut-off frequency (Electrical4U, 2021). If cut-off frequency increases more than the value mentioned, it may result in more peaking at the cut-off value. If the cut-off frequency is less than the mentioned value, it may result in higher attenuation at the cut-off value.

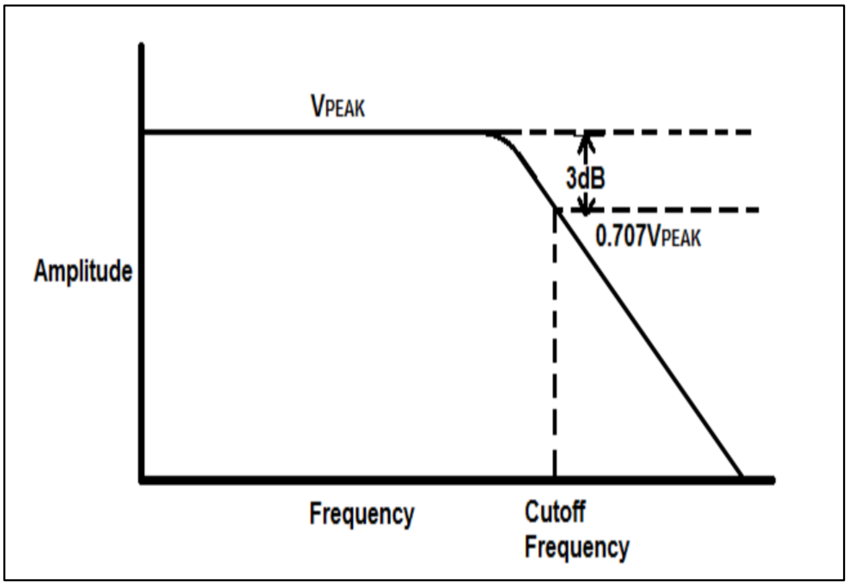


Figure 4: Cut-off frequency of a low pass filter (Electrical4U, 2021)

Once the cut-off frequency is obtained, it is implemented in a first order butterworth filter as well as in a second order filter. First order Butterworth filter is used to preserve the behavior of the graph better compared to the second order filter which actually resulted in more roll off. Figure 5 is the step input for steering angle.

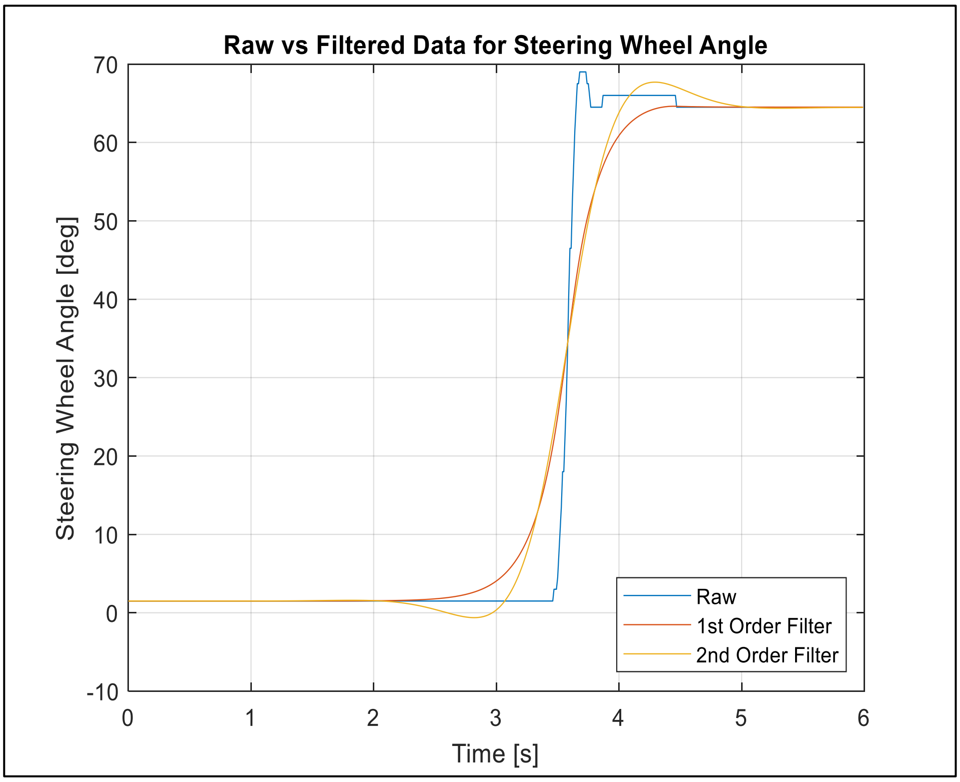
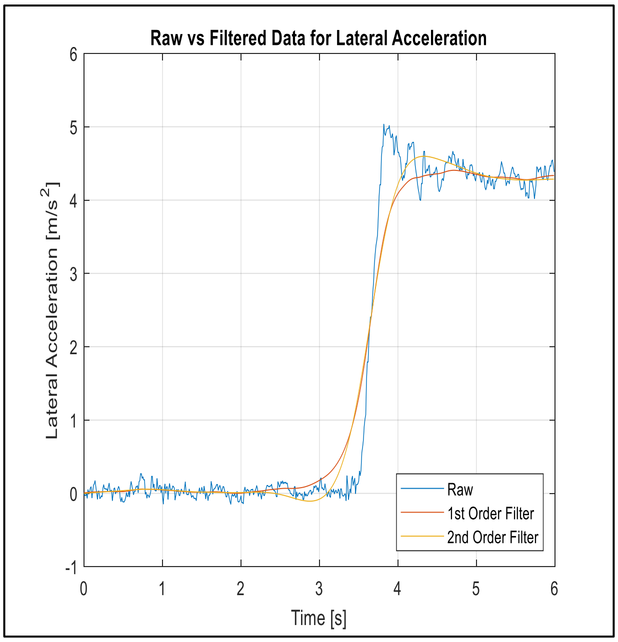


Figure 5: Different orders of butterworth filter vs raw test data

It is evident from Figure 5 that the first order butterworth filter will try to roll off the curve smoothly at cut-off value whereas the second order butterworth filter will provide a peaking value at cut-off value.



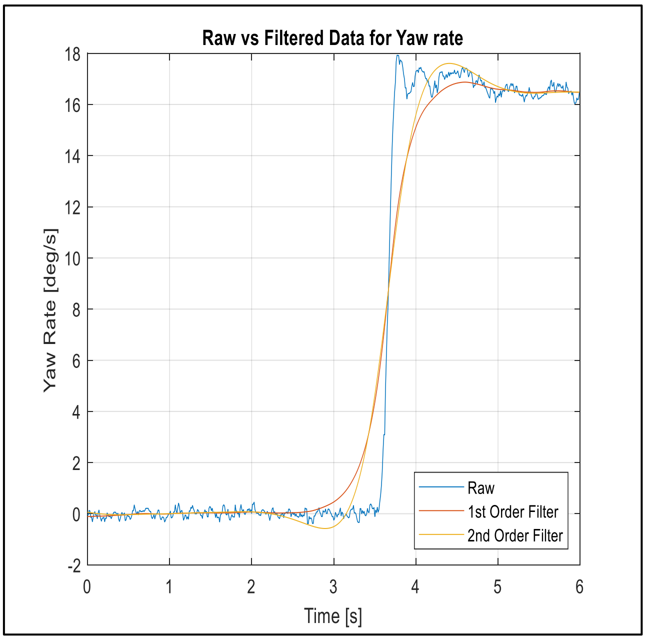


Figure 6: Graph for lateral acceleration and yaw rate for different order filters.

Simillarly for lateral acceleration and yaw rate in the Figure 6, the major difference between a 1st and 2nd order low pass filter is that at the stop band, roll-off will be twice in the second order filters.

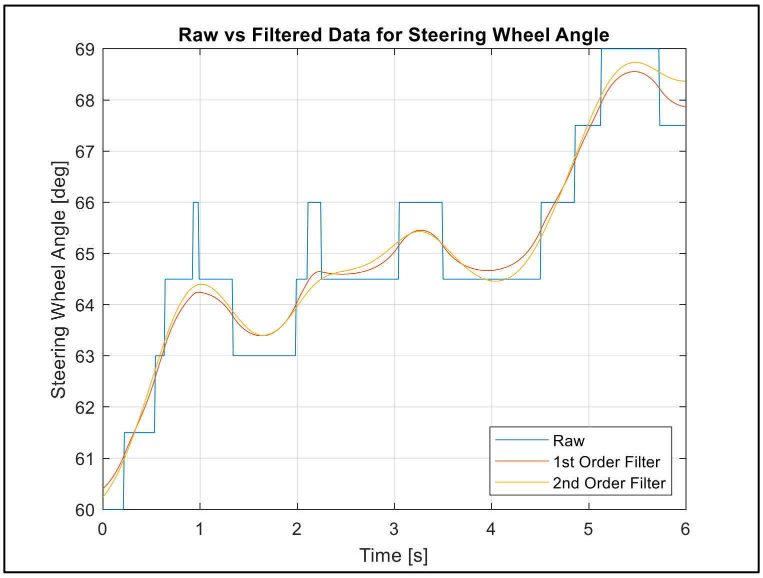


Figure 7: Graph for steering wheel angle for different order filters.

Figure 7 is the steady state data of steering angle. It contains too much of noise and the filtering data has to be within the standard deviation range, according to ISO-standards.

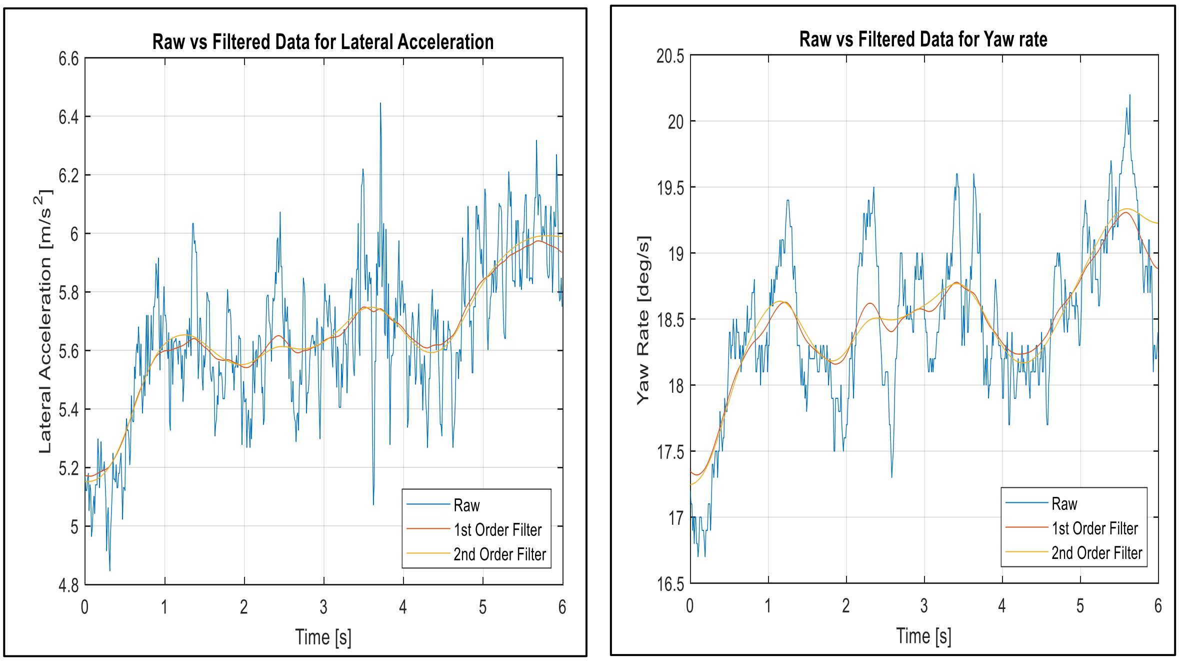


Figure 8: Plot for lateral acceleration and yaw rate for different order filters.

From Figure 8, lateral acceleration plots from both filter looks good due high noise and disturbances in the data, whereas in yaw rate, if observed carefully, the first order follows the data trajectory better than the second order by preserving it much better.

To sum it up, the first order butterworth filter ensures better consistency with the data after tuning the model for validation. Therefore, throughout this project first order butterworth has been implemented.

5.2 Parameter estimation

Parameters are constant values which creates a direct impact on the plant's (model) response based on certain inputs. In explicit conditions, some of the parameters are unknown during the course of modeling the plant. This causes an error or inaccuracy between the measured data and plant's output. Hence, it is essential to estimate these parameters accurately, in order to fine tune the model to obtain results which can closely replicate the measured data.

As the name suggests, 'parameter estimation' is a process of estimating parameters based on a set of data (input, output and plant model). Depending on these data, there are two ways in which the parameters can be estimated, which is described in the following sections.

5.2.1 Mean squared error

This method used in regression analysis, predicts the best fit line to the set of data available to it. The best fit line is obtained by squaring the difference between the data points and the regression line. As this method depends on the direction, squaring the difference is necessary to eliminate the negative sign. Adding all the squares and finding out the mean value of it gives the means squared error. The best fit line is obtained only when the means squared error is minimum.

$$MSE = \frac{1}{n}\sum_{i=1}^{n}(Y_i - \hat{Y}_i)^2$$

$MSE =$ Mean squared error
$n =$ number of data points.
$Y_i =$ Observed values.
$\hat{Y}_i =$ Predicted values.

5.2.2 Absolute error

This kind of error estimation calculates the absolute average distance between the modeled data and the measured data. This method of estimation is similar to the 'Mean squared error', the difference is that it ignores the direction of difference (takes the absolute value).

$$MSE = \frac{1}{n}\sum_{i=1}^{n}|x_i - \hat{x}_i|$$

$n =$ number of data points.
$|x_i - \hat{x}_i| = absolute\ error.$

One of the major disadvantages of using absolute error method is that it is not delicate towards the outliers. In the project, the data consists of many outliers as it is real time data and also the frequency of signal is too high. Hence this method of estimation was not considered while performing the parameter estimation in Simulink.

5.2.3 Parameter estimation in Simulink.

MATLAB add-ons consists of various packages and toolboxes which offers unique and wide range of problem-solving techniques over a large domain of engineering branches. One such toolbox used is the 'Simulink parameter estimator'. This toolbox offers the above two types of solutions. The toolbox asks for the parameters that has to be estimated and also the output of the model and the measured data. After choosing all the parameters which has to be estimated, the toolbox asks for an initial number for the search to begin, minimum and maximum values as the extrema which the parameters should not exceed.

Considering these as the input, the toolbox estimates all the various possible combinations of the parameters based on the input data points. In this method of parameter estimation in Simulink, mean squared error was used because of its advantage stated above with regards to absolute error.
For this project, to check the deviation of these parameters, parameter estimation was carried out for all the available speeds and both the turns. The parameters obtained during this are stated in the Table 4.
From Table 4, it is clearly observable that the values of these parameters for different speeds and turn is very adjacent to each other with not much difference between them. Thus, for each factor, the average value was chosen as the scaling factor to accommodate different speeds.

Factors	Left			Right			Average
	50 km/hr	60 km/hr	70 km/hr	50 km/hr	60 km/hr	70 km/hr	
Longitudinal friction	1.4	1.4	1.4	1.4	1.4	1.4	1.4
Longitudinal Slip Stiffness	1.0564	0.98271	1.139	0.89632	0.92279	0.92385	0.986845
Lateral friction	0.90492	0.9342	0.92523	0.75188	0.7687	0.7685	0.84223833
Cornering Stiffness	3.8117	3.8877	3.8985	4.0137	4.0141	4.0124	3.93968333
Camber Stiffness	0.011094	0.005989	0.003079	0.001989	0.000798	0.001241	0.00403149
Pneumatic Trail	1	1	1	1	1	1	1
Camber Torque Stiffness	1	1	1	1	1	1	1

Table 4: Parameter estimation table

5.3 Validation

In order to check the validity of the model for different scenarios, field track data from testing was used in addition to the data from another group who developed a single-track model. The double track model output was thus compared with two sets of test data to evaluate and assess the effectiveness of the model.

5.3.1 Step Steering input

i. Left turn, 60-degree steering wheel angle, 50 km/hr.

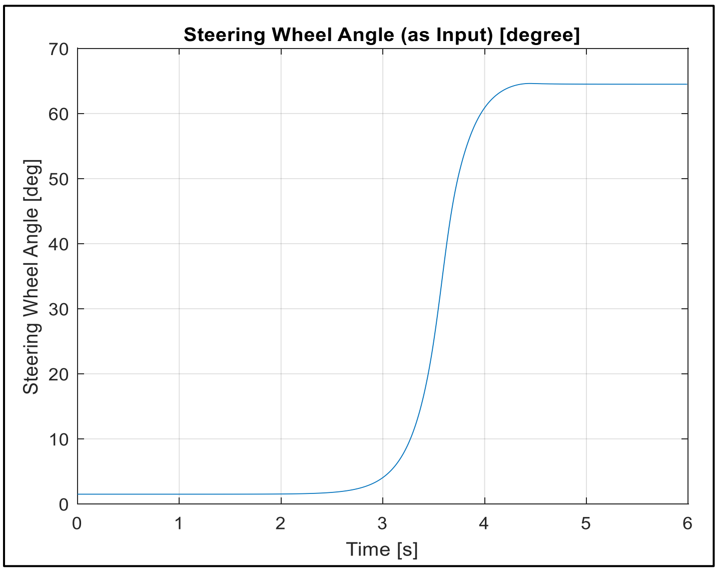


Figure 9: Step Input for left turn at 60 degree steering wheel angle and 50 km/hr

The model was simulated for step steer of 60° steering wheel angle to make a left turn and at a constant speed of 50 km/hr. The results were analyzed for lateral acceleration, yaw rate and roll angle. The results obtained is compared against the test data.

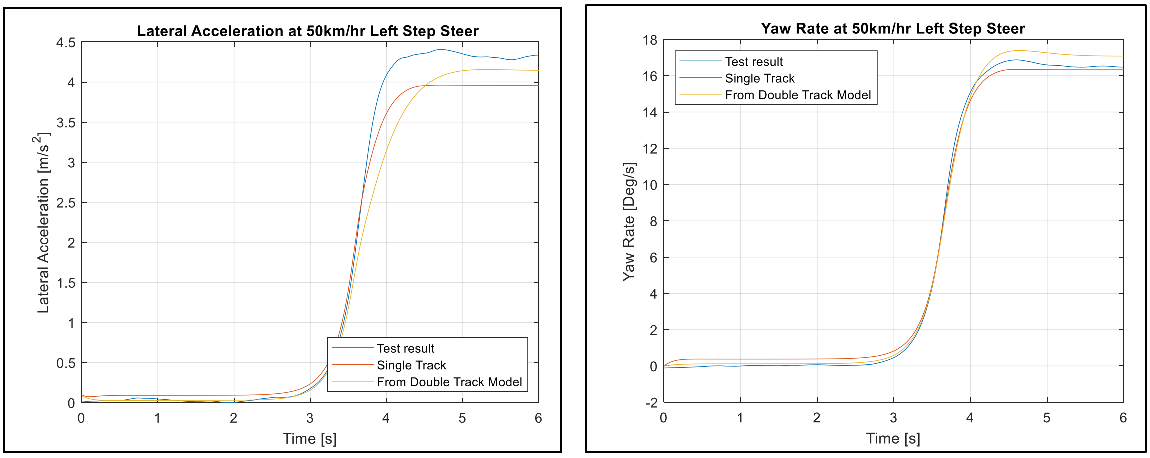


Figure 10: Lateral Acceleration and Yaw rate for 60° steering wheel angle step Input

represents the results of testing and simulation of both single and double track models in terms of the lateral acceleration, the double track model shows close behavior to actual results with two main differences. The model transient response is slower than the actual vehicle due to delay in development of tire forces because of late slip angle generation at rear axle and the final steady state value of the test result (4.4 m/s^2) is little higher than the modeled result (4.35 m/s^2).
As per the yaw rate is concerned, the transient behavior of the double track model traces the obtained test result, while the final value from the model (16.4 deg/s) at steady state condition is higher than the test data result (16.1 deg/s).

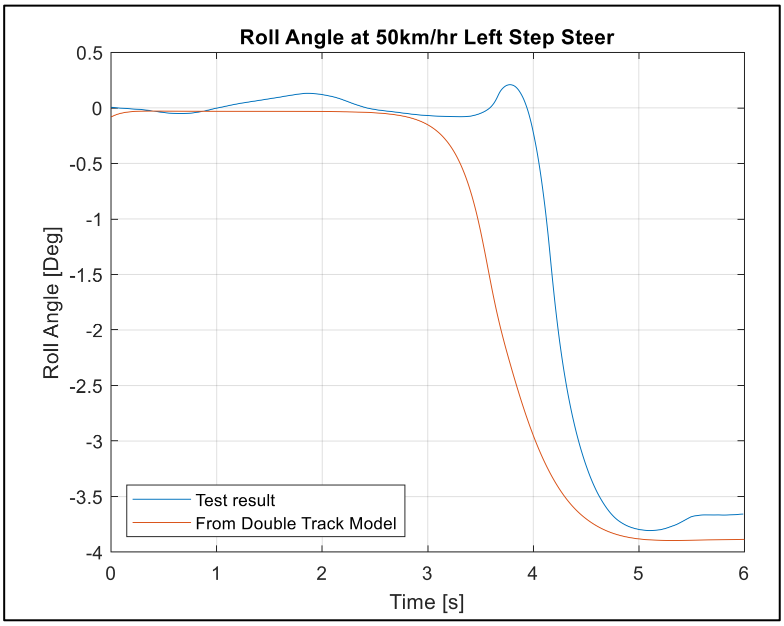


Figure 11: Roll rate for 60° steering wheel angle step Input

The roll angle output closely matches with the actual vehicle output but with a slight delay. Since roll angle is dependent on lateral acceleration therefore delay in transient behaviour can also be seen here.

5.3.2 Right turn, 60-degree steering wheel angle, 70 km/hr.

Simulation was similarly performed for a right turn but at an increased speed. A step steer input of 60° was given at the steering wheel to make the vehicle yaw in the right direction at a constant speed of 70 km/hr was given. The results obtained are present down below.

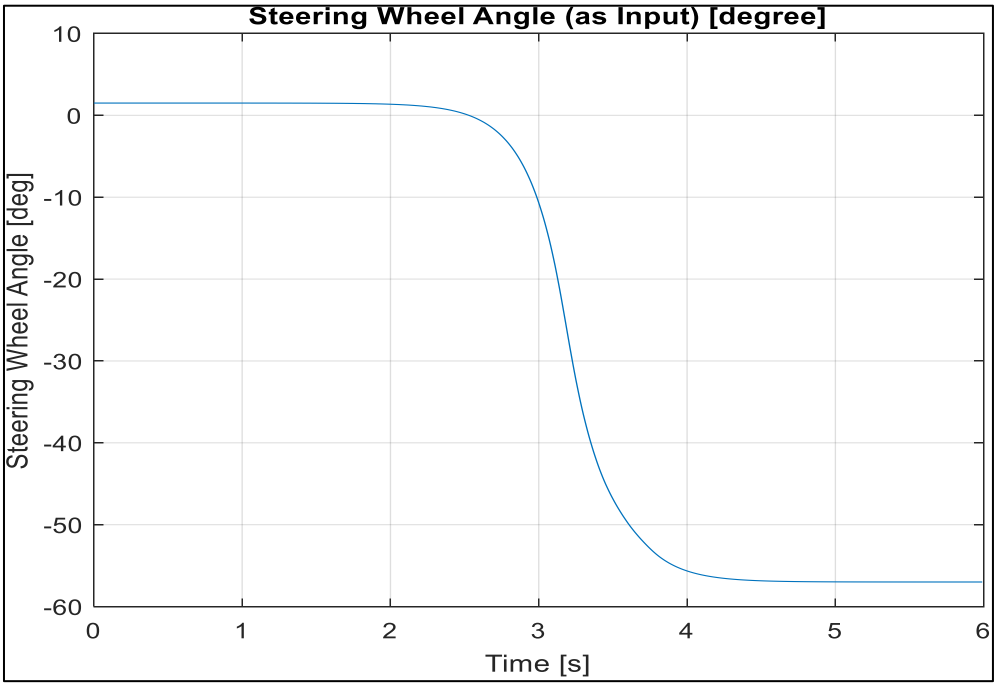
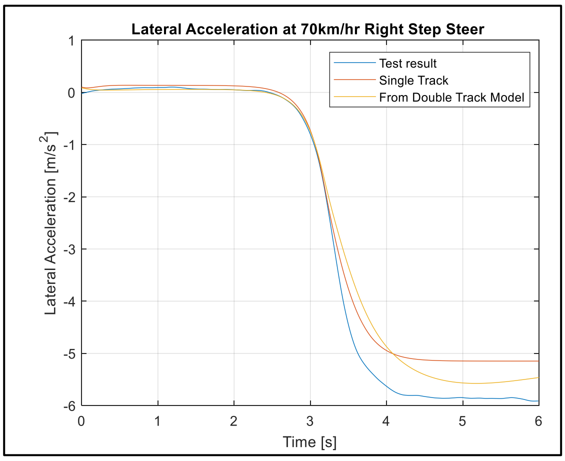


Figure 12: Step input for right turn



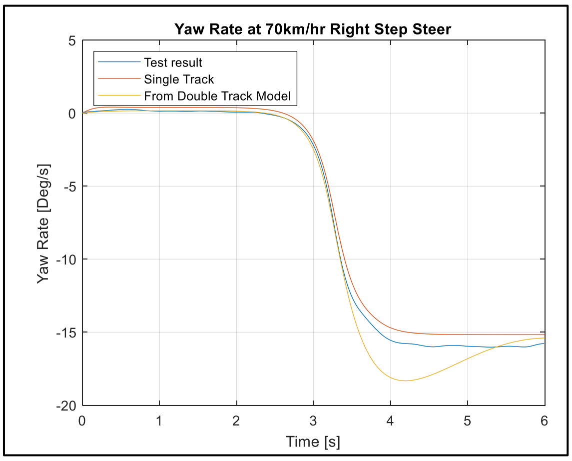


Figure 13: Lateral Acceleration and Yaw rate for 60° steering wheel angle step input

The results yielded similar results in terms of lateral acceleration and yaw rate in comparison to the test data. Yaw rate has an overshoot compared to test data. The test data is filtered using first order butterworth filter hence peaking at cut off value will be eliminated as it conserves smoother roll off.

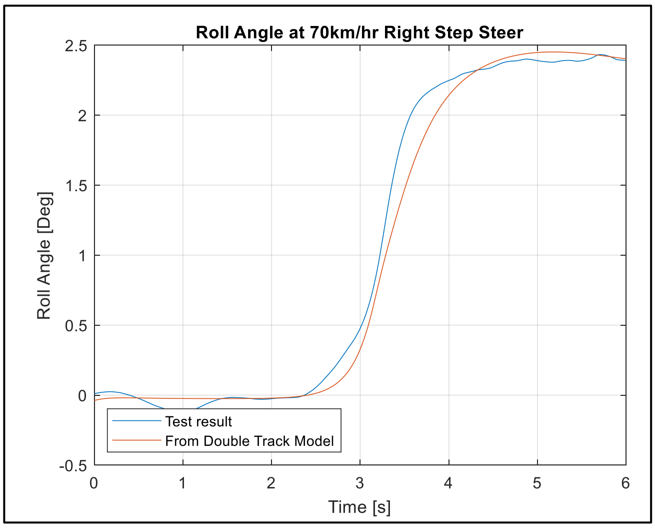


Figure 14: Roll rate for 60° steering wheel angle step input

5.3.3 Steady State Input

Steady state test was performed at constant velocity of 60 km/hr. Figure 15 shows the steering wheel and longitudinal velocity as an input and the output as lateral acceleration and yaw rate are represented in Figure 16 respectively.

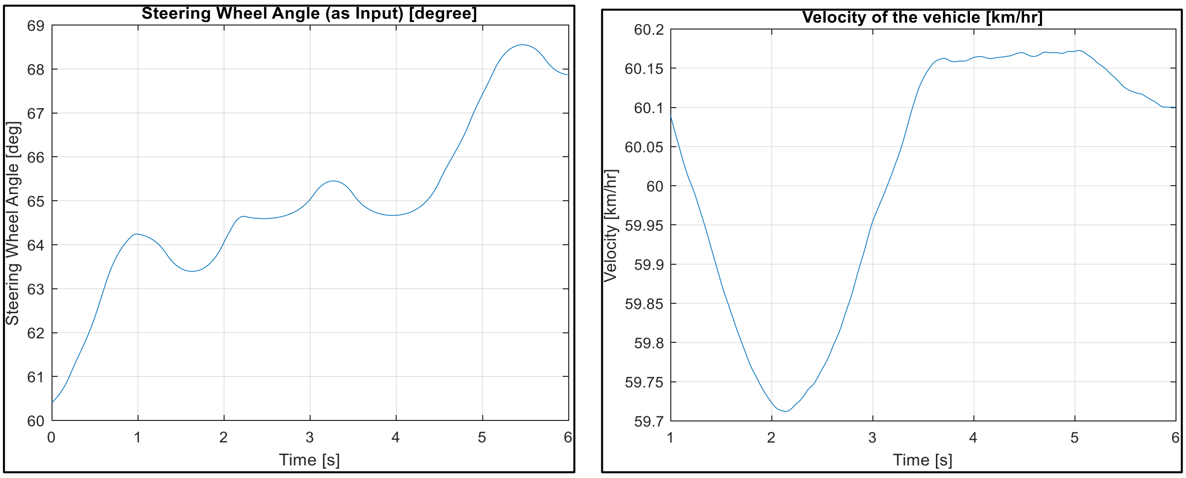


Figure 15: Steering input for steady state and longitudinal velocity of a vehicle

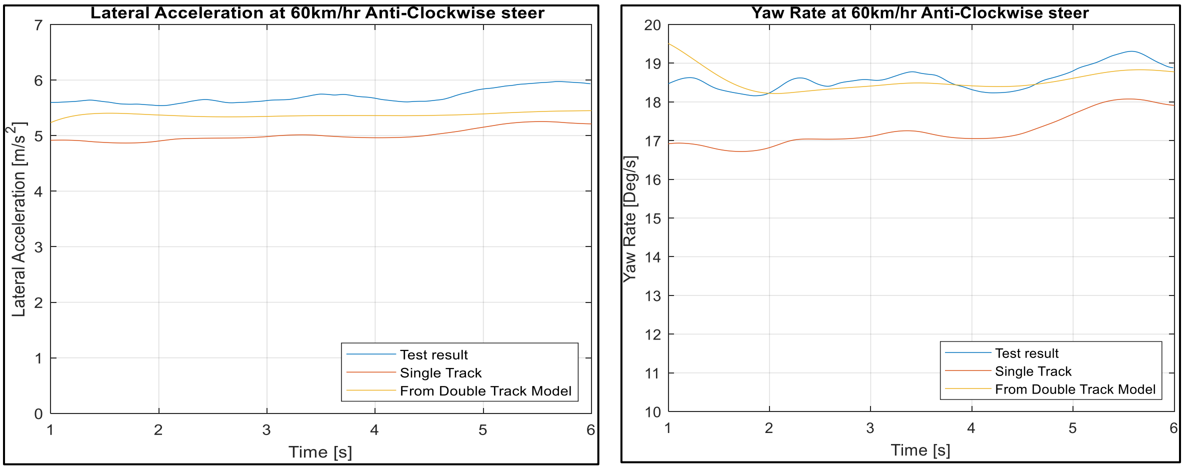


Figure 16: Lateral acceleration and Yaw rate for steady state

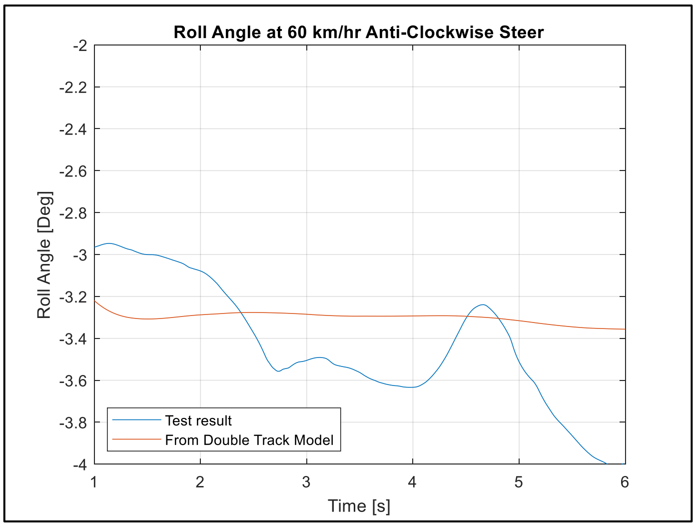


Figure 17: Roll angle at 60km/h for steady state

The reason to start the data from time 1 [s] is because in simulation model takes time to generate forces, so start from zero hence is omitted. For the steady state test, the lateral acceleration graph shows that the double track model is able to predict the behavior better than the single track model, the steady state value of the lateral accleration of double track is (around 5.4 m/s^2) also even for the yaw rate it is same (18.9 deg/s). The fluctuations in the yaw rate and the lateral acceleration is due to the non constant velocity as seen in Figure 16. The reason that single track model does not match with data is because of the implementation of linear tyre model. Both models were simulated for a constant speed of 60 km/hr.

Parameter			Test data			Simulated data		
	Symbol	Unit	Left turn	Right turn	Average	Left turn	Right turn	Average
Lateral acceleration peak response time	$T_{aY,max}$	s	1.74	1.57	1.655	2.29	2.1	2.195
Yaw velocity peak response time	$T_{\dot{\psi},max}$	s	1.83	1.58	1.705	1.7	1.72	1.71
Overshoot value of lateral acceleration	U_{aY}	-	0	0	0	0	0	0
Overshoot value of yaw velocity	$U_{\dot{\psi}}$	-	2.3	0	1.15	1.7	1.4	1.55

Table 5: Step input response data summary

6 Handling behaviour

6.1 Understeer Gradient

From the graph of steering angle versus lateral acceleration, the understeer gradient, K_{us} can be obtained from the slope of the graph in the linear range. From the incremental definition, we can find K_{us} as,

$$\delta = \frac{L}{R} + K_{us}\frac{a_y}{g}$$

The Ackerman steering angle is given by $\frac{L}{R}$. While $K_{us}\ or\ \eta$ is the understeer gradient in the equation mentioned above. Steady-state circle test with constant radius needs to be done in order to plot this graph.

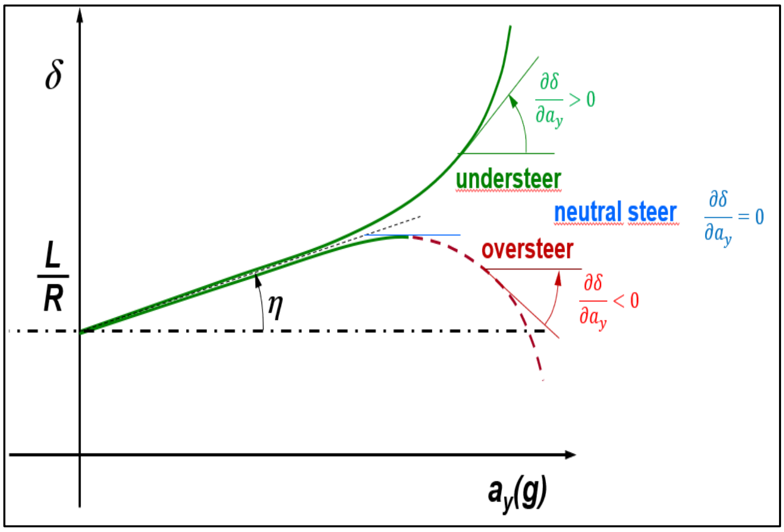


Figure 18: Plot of the steer angle δ as a function of the lateral acceleration (Pauwelussen, 2015)

From the steady-state circle test data, we obtained plenty of data points for steer angle and lateral acceleration for different speeds. By using curve fitting tool, we can choose the best polynomial plot for the data. Thereby, using linear least square estimator, the best fit for the data points is obtained. The Figure 18 shown below is for steering angle (δ) versus lateral acceleration (a_y in g's).

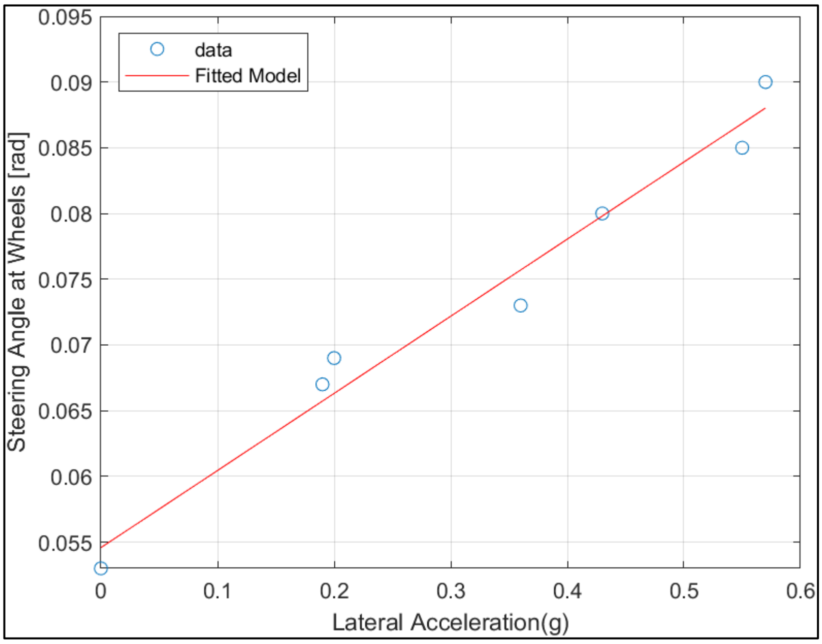


Figure 19: Plot of the steer angle δ as a function of the lateral acceleration for dataset

The understeer gradient, $K_{US}/\ \eta$ was found out by taking slope in linear range or by using the general equation for straight line for the above plot.

By taking the slope of the above data fit curve, the understeer gradient was founded to be 0.06 and is rounded off to 2 decimals. By using straight line equation, we know

$$y = mx + C$$

Where 'm' is the gradient of the straight line. Therefore,

$$y = 0.065x + 0.053$$

If we round of to 2 decimals understeer gradient will be 0.06 and where, the Ackerman steering angle at the wheels is 0.05.

Even from simulation the understeer gradient can be obtained and is shown in Figure 20.

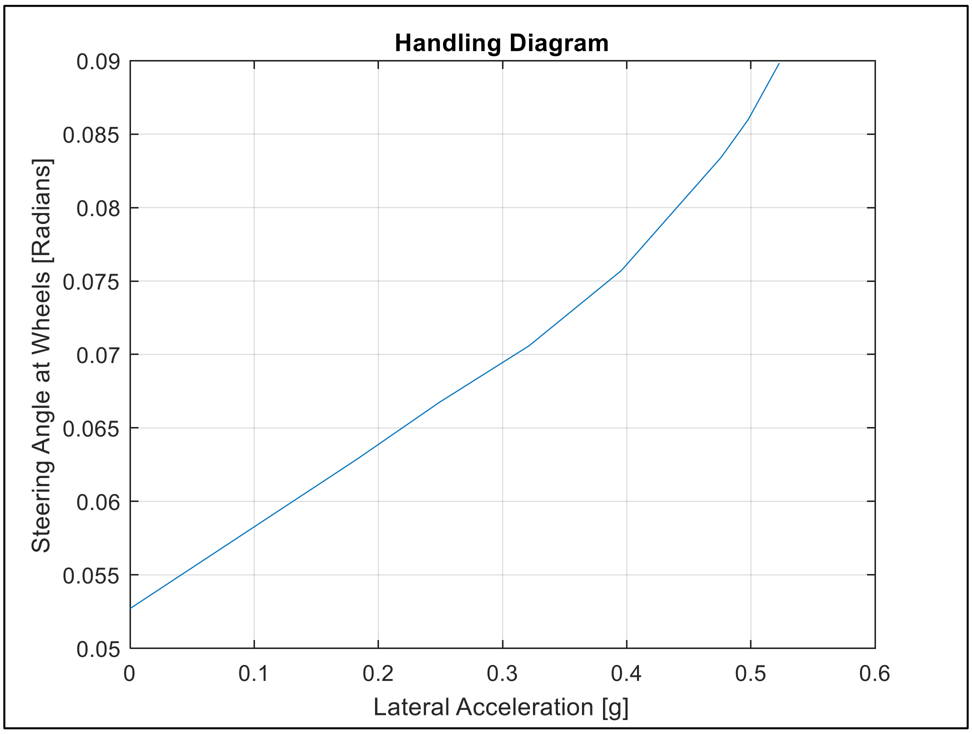


Figure 20: Steering angle at wheels vs lateral acceleration including non linear behavior.

The value obtained for understeer gradient from simulation is approximately 0.05. The simulation value is actually less compared against the test data because for the test data, curve fitting tool is used to obtain the slope, as it minimizes the sum of area of the data points to plot it linearly across all the data set.

Since the understeer gradient value is greater than zero from the incremental definition. Hence it can be concluded that the vehicle handling behavior is understeer.

$$\mathit{Understeer:}\ \frac{d\delta}{da_y} > 0$$

6.2 Yaw rate gain and Characteristic Speed

The yawrate gain can be expressed as the ratio between yaw velocity and the steering angle of a vehicle. Yawrate gain increases linearly with speed until it reaches a particular speed, this speeds is called as 'the characterstic speed' for an understeered vehicle. From Figure 21 as shown below (Pauwelussen, 2015).

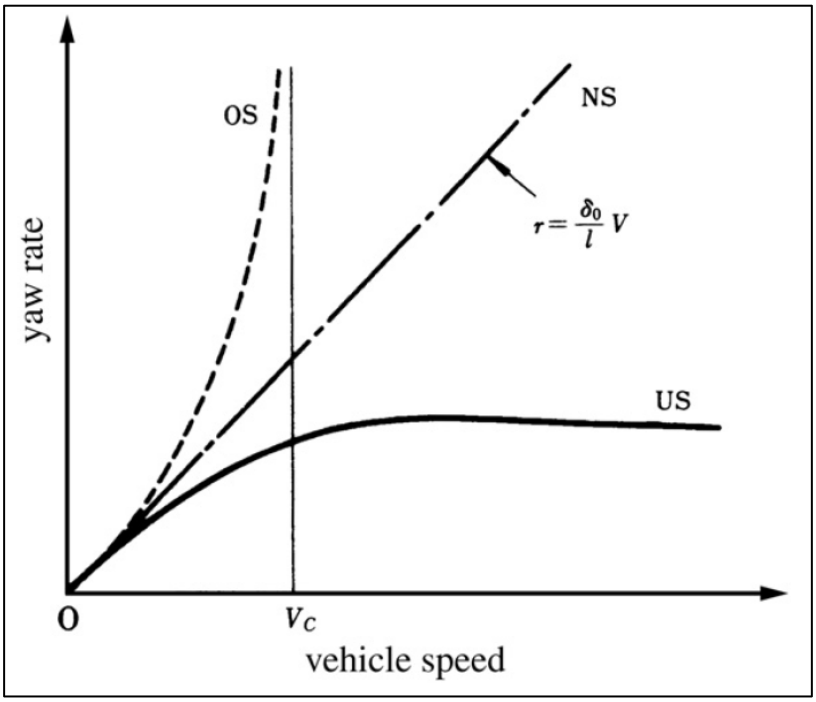


Figure 21: Characteristic speed obtained from yaw rate vs speed (Abe, 2017).

Using the validated model, simulation was carried out for different speeds and was plotted against Yaw rate gain shown in Figure 22. The yaw rate gain increases linearly for low velocities and then starts to decrese after reaching the peak value at velocity of 21m/s (75.6km/hr).

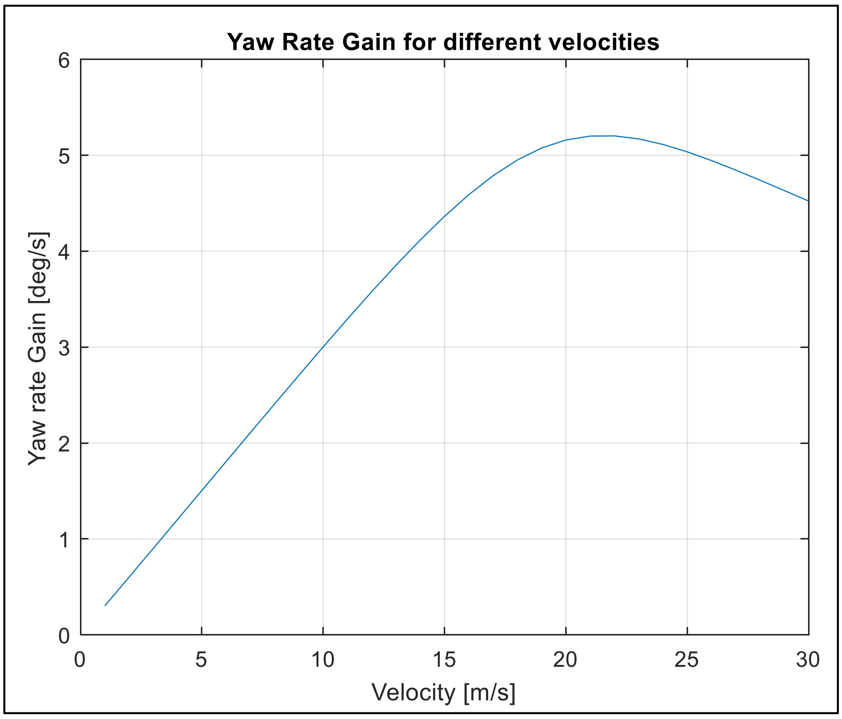


Figure 22: Yaw rate gain vs velocity Simulated data.

By definition, characteristic speed is the speed at which the steering angle needed to negotiate a turn is equal to twice the Ackerman angle (Genta, 2017).By extending the steering angle to twice the Ackerman around 0.104 rad, the corresponding lateral acceleration is found to be 8.2 m/s^2.

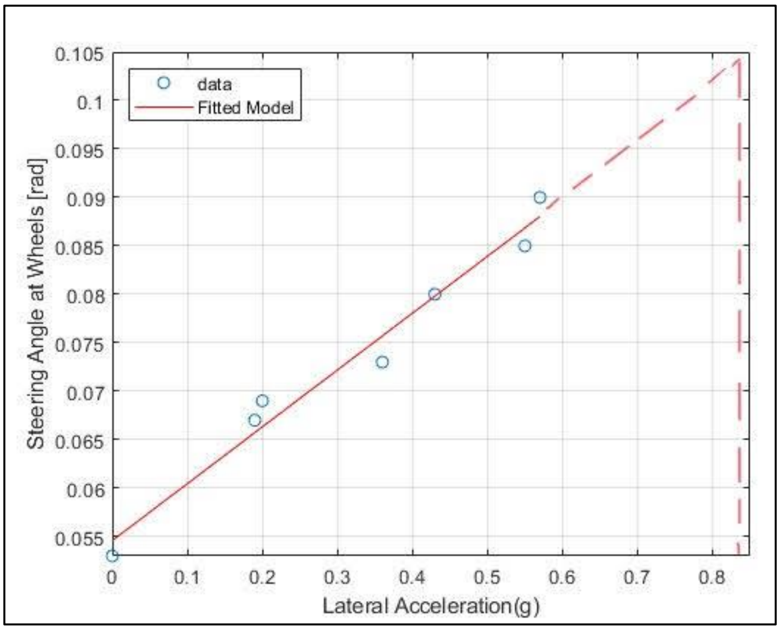


Figure 23: Best fit line for handling behavior graph (steering angle vs lateral acceleration)

The other way of finding characteristic speed by using the equation shown below (Genta, 2017),

$$V_{char} = \sqrt{\frac{gl}{K_{US}}}$$

Therefore, the characteristic speed is around 72km/h.

6.3 Sensitivity Analysis

6.3.1 Changing Mass of Vehicle

For this analysis, to study the change in behavior of the vehicle, the mass of the vehicle was increased and decreased by 20%. This change in the mass of vehicle was achieved by increasing/decreasing the load on each wheel by 20%.

As the mass of the vehicle increases, the tendency of the vehicle to turn into the corner decreases due to its higher inertia. This increases the understeer characteristic of the vehicle. The same can be observed in the graphs in Figure 24. For light weight vehicle, the lateral acceleration and yaw rate is higher than that for a heavy weight vehicle.

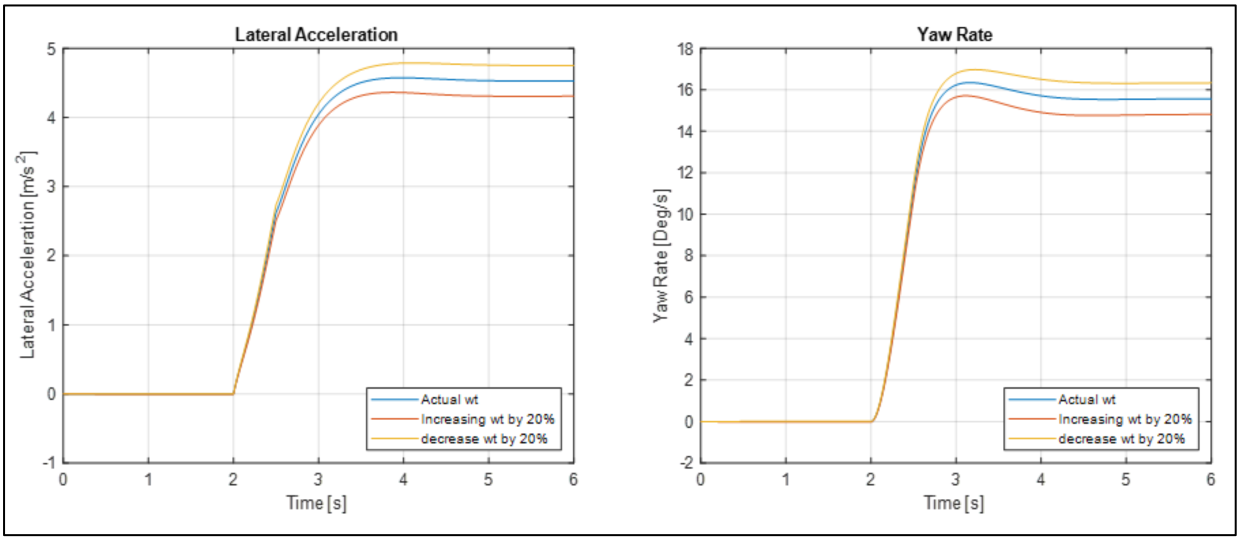


Figure 24: Lateral acceleration and yaw rate graph for $\pm 20\%$ change in mass of the vehicle.

6.3.2 Changing COG of the Vehicle

For this analysis, the load on the front axle was increased by 20%, which would cause the COG of the vehicle to shift towards the front axle. Then the load on rear axle was increased by 20% setting the load on the front axle as initial values. This causes the COG to shift towards the rear axle. The results for lateral acceleration and yaw rate of the vehicle were plotted under these conditions.

On increasing the normal force (F_z), the tires get less efficient. When the front wheels are more loaded, the normalized lateral force (F_y/F_z) of front axle decreases. This increases the understeer behavior of the vehicle. In the case, of more loaded rear axle, the normalized lateral force at front axle increases. This decreases the understeer characteristic of the vehicle. This is the reason why the lateral acceleration and yaw rate have lower value when the front axle is more loaded.

When the load on the front axle increases, the overall Fy on the axle reduces and thus to compensate for the loss of Fy, we need to increase the front axle slip angle (by increasing the steering angle) which will eventually increase the Fy on the front axle. And this increase in front axle slip angle results in increase in the understeer behavior of the car.

From the yaw rate graph in Figure 25, increasing load at rear axle causes the steady state value to shift higher. This is an oversteer property and increasing the load at front axle causes the steady state value to shift lower which is the property of an understeer vehicle.

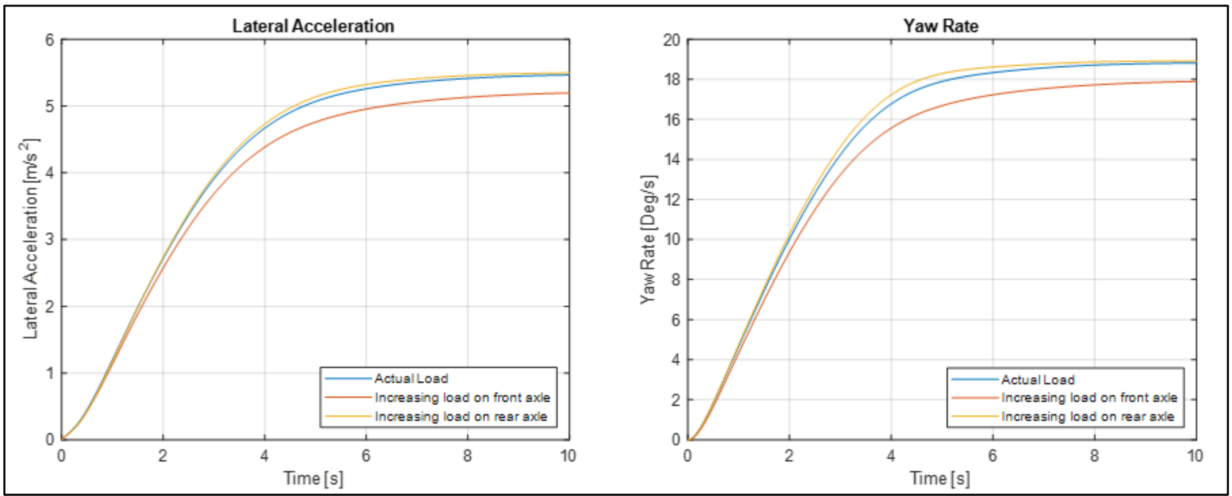


Figure 25: Lateral acceleration and yaw rate graph for increase load on front and rear axle.

7 Conclusion

In this project, a double track handling model for V-Tron Toyota C-HR was investigated. It was decided to consider a double track model with nonlinear tire behavior, load transfer and roll behavior. Such decision was made as to improve the single-track model done previously. The stated considerations were studied through the literature survey, then the model was developed and simulated. The results obtained from this model are satisfactory with respect to the double track model and the actual testing results. The goal was to predict the behavior of the V-Tron Toyota C-HR in steady and transient states. The trend of the outputs should be the same also the values should be as accurate as possible. Although majority of the few graphs gave excellent results, there were few graphs which gave below satisfactory results. It can be concluded that some modifications and calibration should be made to the double track to improve the simulation results in terms of response time and accuracy of the values. In addition, the single-track model previously developed gave inferior results when compared against the double track model at the present stage. Some recommendations for future work are stated in the following order to improve the double track model.

8 Recommendations

1) The height of the CoG should be measured (Using the ECE regulations) to get better model accuracy. In this project the height is estimated, such estimation could be far from the actual value and this has a major effect on the results.
2) Decimals in the parameters should be taken into account to fix the minor lateral forces present in the tire subsystem model leading to a trend in the results.
3) Calibrating the parameters using test data is a necessity for future works, especially that such model can be used for the nonlinear behavior of the vehicle and in a bigger lateral acceleration range where nonlinearities exist, and load transfer has a major influence.
4) Better methods or algorithms maybe developed and implemented to find the unknown parameters, other than parameter estimation in Simulink.
5) Testing on tires to obtain the accurate tire property parameters to get better results.

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10 Appendix – 1

Test plan

1. Introduction

The V-Tron Toyota C-HR is a modified version of the Toyota C-HR available at V-Tron company. The original Toyota C-HR is a subcompact crossover SUV with two doors weighting around 1500kg.The handling behavior of the V-Tron C-HR is not simply perceived and therefore a model and testing are needed. Testing of this vehicle will help in improving the model by making it more accurate and more specific to the V-Tron Toyota C-HR.

Two Types of testing are required in this situation, the steady state circular behavior and the lateral transient response. These two tests do not cover the behavior of the car for all driving conditions but give the necessary information towards better understanding of the handling behavior.

During testing, the inputs to be considered are the steering wheel angle, car velocity (controlled by the throttle position), while measurements will be taken for the outputs in the form of lateral acceleration and yaw rate. Other outputs, needed for model validation, like body slip angles in addition to lateral forces (developed at the wheels level) will be calculated.

2. Objectives and Tasks

a. Objectives

The main goal of these tests is to understand the vehicle behavior and to log the variation of the physical parameters related to vehicle handling. This data will be used later to improve and validate the handling model made for the vehicle.

b. Tasks

Testing will require performing various tests for multiple times and data logging/acquisition. The main tasks considered in this plan are:

- Preparing the vehicle for testing.
- Checking if the measurement and data acquisition systems are functional and calibrated
- Documenting the track and weather conditions.
- Driving the vehicle as per the ISO Standards (Performing the test).
- Logging the data in specific periods when the car is in the required conditions.
- Keeping up with the schedule and making sure to complete all the required tests.
- Acquiring data, plotting, and analyzing it.
- Test reporting

3. Scope

The V-Tron Toyota C-HR will be tested to visualize and document its handling behavior during cornering and maneuvers

Testing of the V-Tron Toyota C-HR consists of driving the vehicle on track and logging the parameters related to handling behavior and needed in the modeling.

The tests covered in this plan are:

- Steady State Circular Behavior.
- Lateral Transient Response.

There are various methods to perform these tests. Each method provides different information related to the handling behavior of the vehicle. Therefore, multiple ways were chosen in a manner to acquire as much information as possible about the handling behavior.

4. Testing Strategy

The strategy followed for the tests consists of completing one test with all the considered methods before moving to the second. In such strategy, all the testing conditions should be valid, and measurements will be considered only in this case. This way the handling behavior of the vehicle is acquired in a proper way for each of the various specific cases.
The tests will be done in an alternative way (i.e., Considering both turning possibilities) to reduce tire wear and heat concentration in one part of the wheel. Example: If the first test considers driving the car in a circular trajectory while turning to the right, the next one will be the same while turning to the left.

5. Hardware Requirements

The Hardware used in these tests is acquisition systems and software rather than machines and benchworks. The hardware needed is as mentioned below:

- A data logging and acquisition hardware (DAQ).
- Sensors connected to the DAQ for measuring the previously mentioned inputs and outputs.
- CAN bus (Already implemented in the vehicle), V-Box and data conversion system.
- Memory stick to transfer data from the DAQ to a computer.

6. Environment Requirements

Environmental conditions have a major impact on the test results. Weather and track conditions should be satisfactory to consider that testing was done properly. These conditions are split into track conditions and weather conditions.
The track conditions:

- Dry track.
- Smooth and not containing irregularities/obstacles that could lead to vertical disturbances.
- There should be no water spots on the vehicle trajectory.
- Moderate asphalt temperature.

The weather conditions:

- Low to null wind speed.
- No precipitations.
- Moderate temperature to prevent fast tire heating.

Any variation in these conditions will lead to low-quality tests results and, in some conditions, unusable data.

7. Tests Description and Sequence

Two types of tests will be performed, where multiple methods are used to obtain the required variables. In these tests, some methods are chosen from which the required data can be gathered. A description of the tests and the methods used are written below.
Tests will be performed as per ISO standards. The exact procedure will be followed for both tests. For the steady state circular behavior, the ISO 4138 fourth edition (2012) is followed, and the ISO 7401 third edition (2011) is followed for the lateral transient response test.

a. Test description

A short description is carried out to give a better understanding on how the tests will be performed. First the steady state circular behavior test is described followed by a description of lateral transient response test.

I. Steady State Circular Behavior Test

This test is done to study the vehicle handling behavior during cornering mainly and to acquire the steering characteristics of the vehicle (Understeered, neutral steered or oversteered). The steady state circular behavior requires driving the car with either constant speed, constant steering wheel angle or on a fixed circular trajectory with constant radius. It is called steady state because the car velocity should be constant to exclude any additional behavior related to accelerating or braking. Three methods can be followed to perform this test and are as following:

- Steady sate circular behavior with constant trajectory radius:

Description: The trajectory radius is kept constant means the vehicle follows a fixed circular trajectory, the speed is kept constant and the driver figures out the steering angle needed to keep the car on track once this is achieved, data is recorded for 3 seconds for all parameters.

The starting car speed is the minimum possible to get the Ackermann steering angle, afterwards the velocity is increased slowly between one testing speed and the following. The limit is achieved when the lateral acceleration of $0.4 m/s^2$ is reached.

As mentioned previously the three methods follow the same strategy by fixing one parameter, variating the second and measuring the third.

II. Lateral Transient Response Test

This test in done to study the vehicle handling behavior for the cases of lane switch of maneuvers. The overall trajectory in this type of test is a straight line and the velocity is kept constant as much as possible.

The methods used in this test will give results in time and frequency domain. Having the behavior in both domains will give a better understanding of the complete handling behavior for various inputs. In the case of the considered tests, the random input test will cover the frequency domain while the first two will cover the time domain.

- Transient response with step input:

Description: The car is driven straight with constant speed, a step input at the level of the steering wheel is made by turning the steering wheel to a specific angle as fast as possible and keeping the steering angle constant. Measurements starts once constant speed is achieved and ends after 3 seconds from achieving constant steering wheel angle.

- Transient with double lane change input:

Description: With the help of ISO 3888-2 the double lane change maneuver will be done. The car is driven straight with constant speed, a double lane input at the level of the steering wheel is made by turning the steering wheel to a specific angle then in the opposite direction to reach the negative of this angle and then back to angle 0 degrees and this again mirrored similarly for double lane change. This input should be done in 3 or 4 seconds maximum. Measurements starts once constant speed is achieved and ends after 2 seconds from achieving 0 degrees steering wheel angle.

Measurements are taken for a combined time of 12 minutes; each logging should be for at least 30 seconds. Lateral acceleration should not exceed $4 m/s^2$.

b. Testing sequence

Testing will start with the steady state circular behavior, once finished the lateral transient response will follow. The number of tests for the latter is higher than the first and excessive tire wear could happen. The order of the tests is chosen based on this reason.
A detailed sequence is found in the “Test Day” document.

8. Control Procedures

Some preparation should be done before testing to make sure that the acquired results are useful and describe the actual behavior of the vehicle.

- Documentation of the vehicle specifications: Mass, load and pressure at each wheel, tire track, wheelbase.
- Documentation of the tyre specifications and condition.
- Documentation of the sensors and measurement equipment accuracy and specifications.
- Documentation of the weather and road conditions.
- Check the calibration of the measurement equipment.

Other conditions should be taken care of during the whole testing period

- Tires temperature should be kept in the range of normal driving conditions.
- Tests should be carried out in the normal driving conditions and should never reach extreme limits.

Whenever overheating of the tires is noticed, testing will stop until the tires cool down.
In case of incident, if the problem can be fixed easily, some adjustments will be made, and testing can be resumed otherwise testing will be postponed to another day if possible. If not, testing will be cancelled.
In case of change of weather conditions, testing will be carried out until it is no longer possible to keep testing within the conditions specified in the ISO standards. The change of weather will be documented.

9. Parameters

Various parameters are present in the equations of motion for the handling behavior of a vehicle, the Handling model also contains these parameters. For this reason, a good knowledge of the parameters’ values should be acquired. This knowledge allows the team to make improvements to the model and validate it. The values of these parameters could only be acquired by measuring and logging them during testing. The necessary parameters will be logged while parameters with less priority will be calculated from the acquired measurements.

a. Parameters to be logged

The essential parameters leading to a good model will be logged and they are divided into input parameters and output parameters as following

I. Input parameters:

- Steering wheel angle
- Longitudinal velocity

II. Output parameters:

- Longitudinal acceleration
- Lateral acceleration
- Yaw rate
- Roll angle
- Lateral velocity

- Slip angle

10.Roles & Responsibilities

Roles and responsibilities are split between the testing team where each member has specific tasks to accomplish. Once these tasks are properly done, testing can be considered as successful. The main tasks and responsibilities are as mentioned below.

- Preparing the Vehicle: V-Tron team
- Implementing the DAQ and V-Box: To be done at the HAN ARLA Workshop by all
- Driving the car: Andrea Cremona
- Checking the DAQ and calibration: Andrea Cremona and Manish Varma Raathimiddi
- Documenting the road and weather conditions: Mathew Prasanth
- Keep up with the time schedule and make sure all tests are done: Sarvajith Guru Prakash
- Start/ stop Logging the data in the optimal periods during testing: Manish Varma Raathimiddi
- Acquiring the data into a computer: Yash Sinha
- Test reporting: Sarvajith Guru Prakash

11.Schedule

Proper time schedule required to conduct test plan has been mentioned/explained in the separate document named test day.

12.Dependencies

Testing will depend on:

- Track road conditions
- Weather conditions
- The presence of all persons assisting in the realization of the tests
- The availability/Readiness of the V-Tron Toyota C-HR equipped with the required tools

13.Risks/Assumptions

Some unexpected situations can lead to cancelation of the testing in the specified dates. These are as following:

- Bad weather conditions
- Strict lockdown measures related to COVID-19 pandemic, where testing is prohibited
- Inability of the main tests performing members to be present in the specified day

If testing cannot be done in the specified date, another date will be considered (of possible) otherwise testing will be canceled and the team will rely on papers to validate/verify the model.

11 Appendix – 2

Test Day

Brief explanation has been already given in test plan, need to follow these steps accordingly to achieve proper results and to finish test as per time scheduled.

1. Vehicle Parameters:

The below shown vehicle parameters has been provided by project supervisor.

Parameters	Value
Left hand wheel of the front axle (m1)	460 [Kg]
Right hand wheel of the front axle (m2)	436.5 [Kg]
Left hand wheel of the rear axle (m3)	322.5 [Kg]
Right hand wheel of the rear axle (m4)	298.5 [Kg]
Total mass (m)	1,517.5 [Kg]
Wheelbase (l)	2640 [mm]
Front track width (b_f)	1550 [mm]
Rear track width (b_r)	1570 [mm]
Vehicle length (L)	4360 [mm]
Vehicle width (W)	1795 [mm]
COG position from front axle (a)	1080.4 [mm]
COG position from rear axle (b)	1559.6 [mm]
COG position from bottom/top (z)	500 [mm]
Tire specifications (for all 4 wheels)	Michelin Primacy 3 225/50 R18 95V

Table 6: Parameters of Toyota CH-R

2. Pre-checks:

The goals of pre-check are to focus on identifying and fixing problems by reducing measurement error, and ultimately improving data quality. The prechecks will be done partially at the HAN and the other part related to connecting the antennas to the V-box will be done just before entering the track. The parameters to check are:

- Tire pressure on 4 wheels.
- Fuel level.
- Offset and calibration.
- No additional weight
- Warm-up test.

3. Time schedule:

On test day, a basic time management possible tasks or actions which are intended to take place in the given sequence of chronological order. The time mentioned here might vary a little in case on unexpected circumstances. The testing sequence will begin with a warmup and finding the Ackermann steering angle then proceeding to the lateral transient response and steady state circular tests.

	Activities	Schedule	Time allotted [min]
1]	Arrival time	09:00 AM	-
2]	Warm-up test	09:00 - 09:30 AM	30
3]	Lateral transient response test	09:30 – 10:30 AM	60
4]	Steady state circular test	10:30 – 11:30 AM	60
5]	Re-take test (In-case if any error occurs)	11:30 - 12:00 PM	30
6]	Departure time	12:00 PM	-

Test Number	Input type	Steering angle [Degrees]	Longitudinal Velocity [km/h]	Expected Lateral Acceleration [g]	Time	Duration [min]	Maneuver time [s]	Number of trials
1	Step	60	50/60/70	0.36/0.46/0.55	09:30	30	60	4
2	Double lane switch	115/125	30/40	0.3/0.6	10:00	30	60	4
3	Steady state circular (R=50m)	50/60/70	35/50/62	0.2/0.4/0.6	10:30	60	30	6
4	Spare for retesting				11:30	30		

Table 7: Test day activities and schedule

The step input of the transient response can also be considered for the steady state circular as the maneuver will results in the same car trajectory.

4. Logbook:

Note: Multiple copies of printout will be carried for both tests (Steady state and transient response test).

I. Steady state circular test:

Date	06-04-2021
Start time	09:30
End time	10:30
Direction (mark what is applicable)	☒ Clockwise ☒ Anti-clockwise
Test conditions (mark what is applicable)	Test track ☒ Dry ☐ Wet Weather ☒ Sunny ☒ Cloudy ☐ Rainy ☒ Windy
Desired speed	50, 60 and 70kmph
Test track radius	50m
Data acquisition (name)	
General comments (If needed)	

Table 8: Log book for teady state circle test.

- **Test conditions (steady state circular test):**

Test method	constant	varied	Measured/calculated
Constant radius	Radius	Speed	Steering-wheel angle

Measurables:

The following variables shall be measured.

1. Lateral acceleration.
2. Steering-wheel angle.
3. Yaw rate.
4. Longitudinal velocity.
5. Lateral velocity.
6. Longitudinal acceleration.
7. Roll angle.

II. Transient response test:

Date	06-04-2021
Start time	10:30
End time	11:30
Direction (mark what is applicable)	☒ Clockwise ☒ Anti-clockwise
Test conditions (mark what is applicable)	Test track ☒ Dry ☐ Wet Weather ☒ Sunny ☒ Cloudy ☐ Rainy ☒ Windy
Type of steering input (mark what is applicable)	☒ Step ☐ Sinusoid ☐ Random ☐ Continuous sinusoid
Test method (mark what is applicable)	☒ Time domain ☐ Frequency domain
Desired speed	
Data acquisition (name)	
General comments (If needed)	

Table 9: Log book for transient test.

5. Test table (Transient response and steady circular):

No. of trials	Test Type	Turn	Steering wheel angle	Speed[km/h]	Test Done	Data Checked
1	Step	Right	60	50	Yes	Yes
2	Step		60	60	Yes	Yes
3	Step	Left	60	70	Yes	Yes
4	Step		60	80	Yes	Yes
5	Double lane change	Left	120	30	Yes	Yes
6	Double lane change		125	40	Yes	Yes
7	Double lane change	Right	118	30	Yes	Yes
8	Double lane change		140	40	Yes	Yes
9	Steady state	Clockwise	51	35	Yes	Yes
10	Steady state		58	50	Yes	Yes
11	Steady state		69	60	Yes	Yes
12	Steady state	Anti-Clockwise	55	35	Yes	Yes
13	Steady state		64	50	Yes	Yes
14	Steady state		73	60	Yes	Yes

Table 10: Overall test data obtained

12 Appendix -3

The standard used in this project is based on ISO standards.

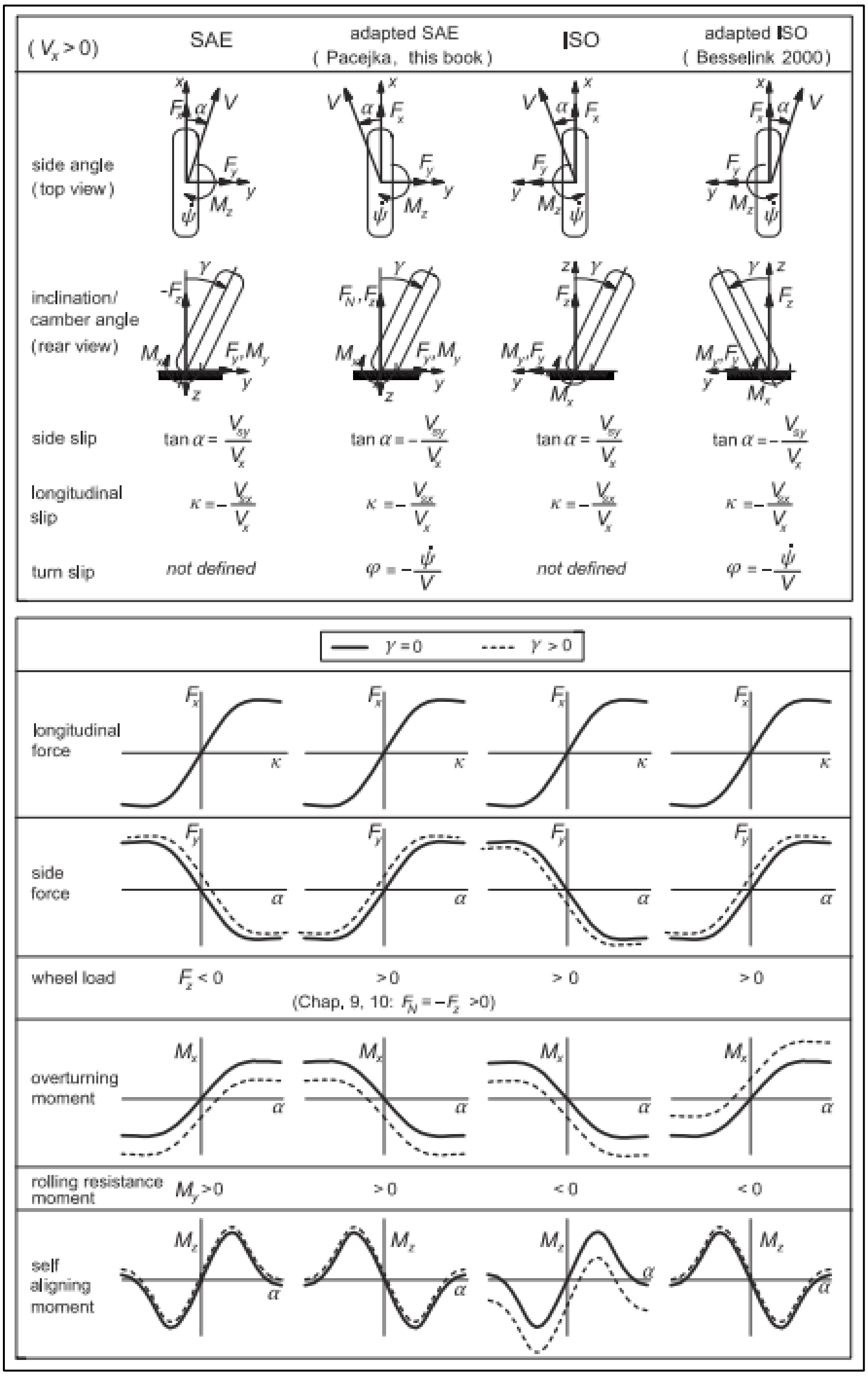

Figure 26: Sign convention for Force & Moment & Wheel Slip (Pacejka, 2013)