Evaluation of the Impact of Electric Powertrain on the Performance of the Steering System

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ABSTARCT

The study of dynamic behavior of Electronic Power Assist Steering System [EPAS] is very critical to understand the overall handling of the vehicle. The appropriate feedback from the steering to the driver is crucial so that the driver get the precise steering feel in order to control the vehicle with ease during all kinds of maneuvers. In this Project Thesis, a complete Steering System dynamic model is developed and integrated with the Full Vehicle Chassis Model using a 1D Simulation Tool. The simulation is carried out as per the guidelines laid by the ISO 7401 Standard Criterion. The simulation is performed on three Vehicle Models in which a comparison of behavior of EPAS System between their Conventional & Electric Variants has been performed. A thorough implementation of various Vehicle, Geometrical & Kinematic SPMM Parameters and KnC Tables is done in the Vehicle Model run in 1D Simulation Tool. Finally, a list of Performance Aggregate Target [PAT] Parameters are calculated, using the output parameters obtained in Post Processing section of the 1D Tool. The resulting PAT parameter values are then compared with the Target Range Data and an appropriate reasoning is provided accordingly.

Key Words: EPAS, AMESim-1D, ISO7401, KnC, SPMM and PAT

1 INTRODUCTION

The design of the steering system has an influence on the directional response behavior of the motor vehicle. The function of the steering system is to steer the front wheels in response to the driver input commands in order to provide an overall directional control of the vehicle. For a maximum tire life, the steering system should maintain a proper tire road contact, and proper tire geometry with respect to the chassis both during cornering and straight ahead driving. The driver should be able to turn the vehicle with ease during low as well as high speed maneuver. The steering wheel connects by shafts, universal joints and vibration isolators to the steering gearbox, whose purpose is to transform the rotary motion of the steering wheel to the translational motion of the rack and pinion. The rack and pinion system consist of a linearly moving rack with the pinion gear turning over it's gear teeth, mounted on a forward cross member, which steers the left and right wheels directly by a tie rod connection. The tie rod linkage connects to steering arms on the wheels, thereby controlling the steering angle.

During the straight ahead rolling of the tires, the translational velocity cancels the rotational velocity of the tire, thereby keeping zero velocity at every contact point. But if the tires try to head ahead during turning, then the translational & rotational velocity will not cancel each other, thereby not allowing the velocity of the contact point to be zero. This phenomenon may cause the tires to slip. Hence to ensure smoother turning profile, the tires must turn effectively causing both the velocities mentioned above to cancel each other out. The tie rod connects the steering arm & the rack assembly thus facilitating both the translational & rotational motion. This satisfies the Ackerman Steering Criteria causing a turn without the occurrence of the slip.

2 LITERATURE REVIEW

2.1 EPAS System for Automobile

[1]MathiasWurges in his paper on New EPAS System, has presented a thorough analysis on all the components of the electronic power assist system and their functionality in detail. The different types of EPAS system technologies are explained appropriately. Different types of sensors used are stated clearly. Emphasis has been laid on the importance of Brushless DC Motor. Analysis is presented on the functional safety aspect during all kinds of maneuvers. Finally, identification of the safety hazards is clearly specified to enhance the safety aspect.[3]Thomas D. Gillespie, author of the book The Fundamentals of Vehicle Dynamics, provided a deep insight into the aspects of

Steady State Cornering, Steering System and the behavior of Tires. A thorough emphasis is given on the behavioral aspect of the vehicle during cornering. Concepts related to Handling, Ackermann Geometry, Understeer Gradient, Characteristic & Critical Speed and Roll Steer were thoroughly explained. The effects of wheel and tire geometry parameters on the behavior of the EPAS system was found to be extremely critical.

2.2 Modeling & Simulation of an EPAS System

[6]Hui Zhang et al. [Hui Zhang, Yuzhi Zhang, Jinhong Liu, Jing Ren &Yongjun Gao] in their paper on Modeling and Characteristic Curves of EPAS system, developed a mathematical model and analyzed the characteristic curves of EPS System. Along with the principles and structure of EPAS system, characteristics of EPS system are analyzed, including the typical power curves which deal with the relationship of the assist torque and the steering wheel torque. At last the EPS system's dynamic model and the computerized simulation model have been established in MATLAB/Simulink in teams of three aspects which are the mechanical steering without power system, EPAS system and EPAS system with PID control. The simulation results obtained specify that the actual current of the motor follows precisely the target current validating the designed assist characteristic. [10]Shailendra Kumar et al. [Shailendra Kumar, Virendra Kumar Verma and Ashish Gupta] in their paper on Mathematical Modeling of EPAS System, have developed a mathematical model for the EPAS control logic which integrates base-assist, damping, return, and inertia control logic. Forms of the EPAS system, according to the location of assist motor were discussed. A mathematical model for column assist steering system has been derived. A controller is designed for the model with the help of assist characteristics curves and PID action. Simulation has been carried out for the circular and moose vehicle test path.

[12]AhnNa Lee et al. [AhnNa Lee, JiHyun Jung, BonGyeong Koo and HangByoung Cha] in their paper on Steering Assist Torque Control Enhancing Vehicle Stability, have presented an analysis on the Unified Chassis Control System (UCC) of Electronic Stability Control (ESC) System and EPAS System for improving vehicle dynamic stability. UCC is observed to be improving the vehicle stability by steer intervention to assist driver's maneuverability. ESC detects vehicle-stability and driver-maneuvering through the sensor and processing signals and judges the intensity of assist torque needed. An emphasis is laid on the Unified Chassis Control which enhances the vehicle stability by several vehicle tests on Split-μ roads during under-steer and over-steer condition. [13]SpyridonZafeiropoulos and Stefano Di Cairano in their paper on Vehicle Yaw Dynamics Control by Torque-based Assist Systems, have investigated the control of torque-based steering assist systems for improving yaw rate tracking and vehicle stabilization. These systems being mechanically coupled with the driver, the specific constraints related to the driver-actuator interaction need to be enforced. These constraints were formulated to avoid excessive strain in the driver's arms. In order to achieve high control performance and constraints satisfaction, controllers were implemented based on linear and switched model predictive control, where different types of driver's steering feel constraints were enforced. The different controllers were evaluated in simulation maneuvers to analyze their capabilities and the impact of the constraints in terms of vehicle cornering, stabilization, and driver's steering feel.[14]R. P. Rajvardhan et al. [R. P. Rajvardhan, S. R. Shankapal and S. M. Vijaykumar] in their paper Effect of Wheel Geometry Parameters on Wheel Steering, did a thorough analysis on the handling and steering characteristics of a vehicle in a virtual environment with the help of multi-body system packages to save product development time and cost. The SUV model under consideration was validated by comparing simulation results with the standard graphs from literature. Using this model, maneuvers for different values of wheel geometry parameters, were simulated and the results obtained clearly stated appropriate reasoning for the different values of Castor, Camber, Toe in and Toe Out and KPI obtained.

3 PROBLEM DEFINITION

In this Report a thorough analysis is performed on the comparative behavior of the EPAS system in Conventional Variant vs. Electric Variant interms of the Performance Aggregate Target Parameters. Simulation of the Steering Module in integration with the 1D Vehicle Model with the help of a 1D Simulation Tool, assists in predicting the performance of the steering system during the early stages of development. This will also help in effective cascading of the components to achieve the desired result. A list of crucial SPMM parameters related to Steering, Tire and Suspension system which affect the behavior of steering response are implemented into the simulation. Finally, an appropriate reasoning is presented for the comparative analysis of the PAT parameters.

4 OBJECTIVE

1. To develop a Steering System Module and integrate the same into the Full Vehicle Model built into the 1D simulation tool.
2. To implement the SPMM parameters in the Conventional Powertrain in the initial phase and digitally assess the PAT parameters.
3. To replace the Conventional Powertrain with EV Powertrain and digitally reassess the steering PAT targets by implementing the SPMM parameters related to EV Powertrain.
4. Finally, to conduct a comparative analysis between the list of Performance Aggregate Targets given below:

   a. Steering Sensitivity [g/100 degSWA]
   b. Steering Angle Dead band [deg]
   c. Steering Wheel Torque[Nm]
   d. On Centre Yaw Gain [deg/sec/100 degSWA]
   e. Yaw Gain Dead band [deg]

5 MODELING METHODOLOGY

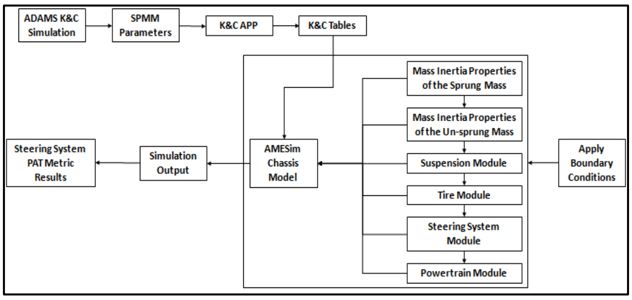


Figure 5.1: Modeling Methodology in AMESim

The 1D Simulation tool allows the simulation engineers to virtually assess and optimize the performance of mechatronic system models. It includes ready to use multi-physics libraries combined with the industry-oriented solutions that are supported by powerful platform capabilities, to let us rapidly create accurate Models and obtain appropriate simulation results.

5.1 Generation of Kinematics & Compliance Tables

K is the acronym for the word Kinematics, which represents the geometric and kinematic characteristics of the suspension. The variation in the Kinematics is related to the change in the suspension positioning parameters caused by the displacement. C is the acronym for the word Compliance; it represents the elastic kinematic characteristics of the suspension. The variation in the Compliance properties is related to the change in the suspension positioning parameters caused by the force. Compliance is a property of suspension springs, antiroll bars, elastomeric bushes and component deformation.

5.1.1 Generation of SPMM Parameters

Inorder to generate the Kinematics & Compliance Tables, firstly we need to generate the list of SPMM Parameters from the SVC & DAT file which are the output files from a 3D Simulation tool. The list of the SPMM Parameters is as given below:

Table 5.1: List of SPMM Parameters

Vehicle Parameters	**Unit**
Wheelbase	mm
Steering ratio	deg/deg
C-factor	mm/rev
Pinion radius	mm
Equivalent stiffness of steering system	N-mm/deg
Tire width	mm
Aspect ratio	%
Wheel diameter	inch
Tire radius	mm
Static loaded tire radius	mm
Front Track	mm
Front height of roll center	mm
Rear Track	mm
Rear height of roll center	mm
Geometric Parameters	**Unit**
Initial Toe Angle_Front	deg
Initial camber angle_Front	deg
Caster angle_Front	deg
Caster offset/trail_Front	mm
Kingpin angle_Front	deg

Kingpin offset_Front	mm
Initial toe angle_Rear	deg
Initial camber angle_Rear	deg
Kinematic Parameters	**Unit**
Bump Steer coeff_Front	deg/mm
Anti-dive rate	deg
Anti-squat rate	deg
Bump camber coeff_Rear	deg/mm
Roll camber coefficient_Rear	deg/mm
Mass Inertia Parameters	**Unit**
Sprung mass	kg
Total mass	kg
Front Sprung Mass	kg
Rear Sprung Mass	kg
Front spindle mass	kg
Rear spindle mass	kg
Front wheel mass	kg
Rear wheel mass	kg
Sprung mass inertia_Ixx	kg-m^2
Sprung mass inertia_Iyy	kg-m^2
Sprung mass inertia_Izz	kg-m^2
Sprung mass inertia_Ixy	kg-m^2
Sprung mass inertia_Ixz	kg-m^2
Sprung mass inertia_Iyz	kg-m^2
Wheel inertia_xx	Kg-m^2
Wheel inertia_yy	Kg-m^2
Wheel inertia_zz	Kg-m^2
CG Coordinate Parameters	**Unit**
ECIE_condition_X	mm
ECIE_condition_Y	mm
ECIE_condition_Z	mm

5.1.2 Kinematics & Compliance Application in 1D Simulation Tool

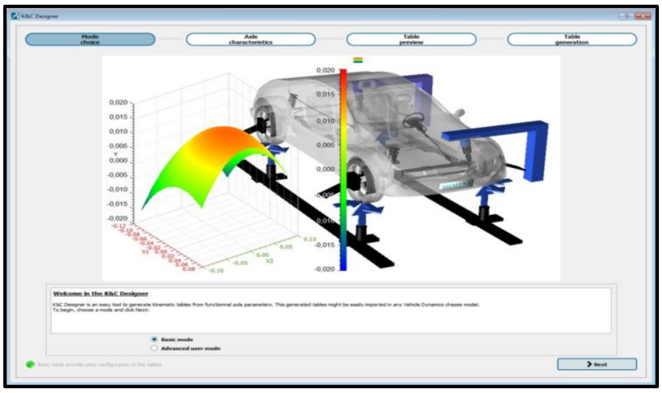


Figure 5.2: Kinematics & Compliance Application

K& C application is a part of 1D Simulation Tool. It takes the Vehicle, Geometric and Kinematic SPMM parameters mentioned above as the input in three subsections and delivers 20 K&C Tables. Kinematic tables are the "functional representation" for the definition of axle system geometry. They describe the variation of track width, wheelbase, steering angle, camber angle and self-rotating angle as a function of vertical wheel lifts.

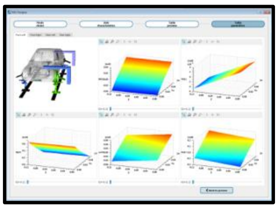
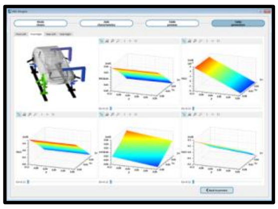
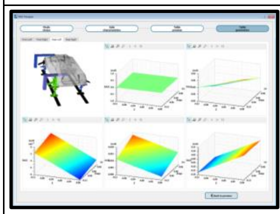
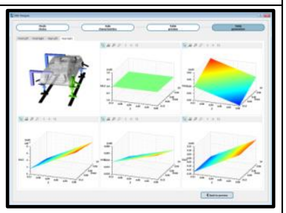
 Front Left Wheel	 Front Right Wheel
 Rear Left Wheel	 Rear Right Wheel

Table 5.2: Images of 20 K&C Tables

	Front Left	**Front Right**	**Rear Left**	**Rear Right**
Wheel	Xa11	Xa12	Xa21	Xa22

Recession				
Track Change	Ya11	Ya12	Ya21	Ya22
Steer	Delta1 1	Delta1 2	Delta21	Delta22
Camber	Epsilon 11	Epsilon 12	Epsilon 21	Epsilon 22
Spin	Eta11	Eta12	Eta21	Eta22

Table 5.3: List of 20 K&C Tables

5.2 Vehicle Model in 1D Simulation

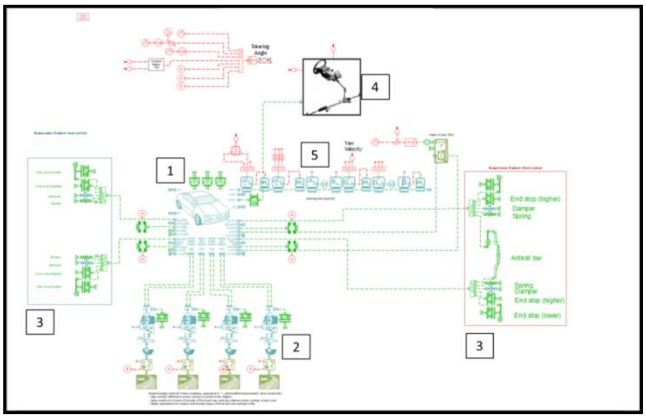
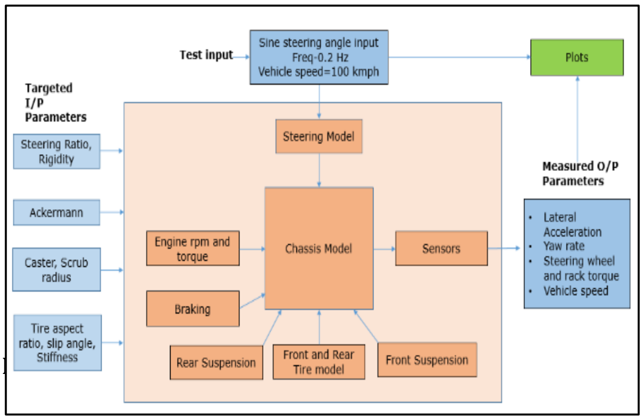


Figure 5:3: Vehicle 1D Model under Simulation

1. Chassis Module
2. Tire & Road Adherence Module
3. Suspension Module
4. Steering System Module



The diagram above represents the architecture of the full vehicle 1D Simulation Model. The main systems which are related to the vehicle dynamics such as Steering System, Suspension System, Tire and Road Adherence System and Sensor System are considered during the development of the model. These systems are then integrated with the Full Vehicle Chassis Module. The outputs from these systems such as Forces and Torque act on the Chassis and bring about variations in it's dynamics.

Standard Steady State On Centre Handling Simulation Procedure is followed and appropriate Boundary Conditions are applied. The Sensor Module calculates the Steering Wheel Angle, Steering Wheel Torque, Lateral Acceleration, Yaw Rate and Aligning Torque with respect to the test applied on EPAS System. Accordingly the Performance Aggregate Targets are plotted by using the above mentioned Parameters from Post Processing section from 1D Simulation Tool.

5.2.1 Steering System Module

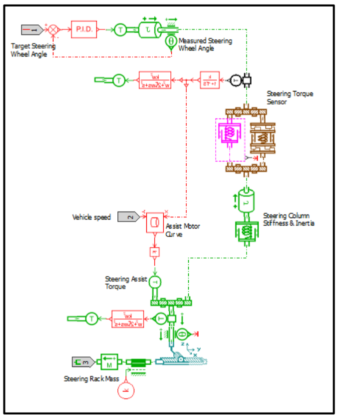


Figure 5:5: Steering System Module

An EPAS System uses a Bi-directional Brushless Motor, Electronic Controller, Rack and Pinion Gear System, Torsion Bar & Torque Sensor to provide an appropriate Steering Assist Torque. The motor will drive a gear that is connected to the steering column shaft or the steering rack. Sensors located in the steering column measures the following primary driver inputs: Steering Wheel Torque [Steering Effort], Steering Wheel Speed and Steering Wheel Position. The torque, speed and position inputs and vehicle speed signal are interpreted by the ECU. The controller processes the steering effort and steering wheel position through a series of algorithms for assist and return to produce the accurate amount of polarity and current to the motor. Other inputs that will affect assist and return are vehicle speed, engine speed and chassis control systems such as ABS and ESC.

The brushless DC motor uses a permanent magnet rotor and three electromagnetic coils to propel the rotor. A motor worm gear is used to drive the gear on the steering shaft or rack. The torque sensor monitors the twist of the torsion bar by measuring the change in magnetic flux generated by its position to the vanes located on the sensor stator rings. When the rotor moves, a change in magnetic flux will send a signal to an analog sensing integrated circuit (ASIC) that will process the signal and send the information to the controller's assist algorithm.

The assist torque is thus calculated by looking up the assistance curve table algorithms fed into controller unit. The curve-based assistance characteristics are helpful to realize continuous and uniform assistance and its curve shape can be adjusted according to real requirements.

5.2.2 Suspension System Module

The vehicles considered in this study has Macpherson strut in the front and twist beam assembly rear.The damper curve are used in damper model.Suspension model is connected to chassis via dynamic mechanical node which calculates the summation of all the forces from the suspension.

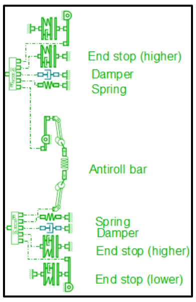


Figure 5:6: Suspension System Module

5.2.3 Tire & Road Adherence Module

This model generates the contact forces at the tire contact patch namely Longitudinal Forces, Lateral Forces and Aligning Torque.

5.2.3.1 Tire Kinematics Model

This model computes the location and velocity of the tire road contact point from the wheel centre. The model assists in calculating the vertical comfort by conducting a mathematical analysis using complex equations and lookup tables.

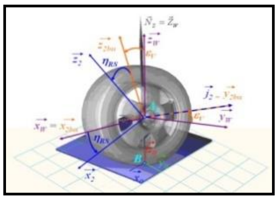


Figure 5:7: Tire Kinematics Model

5.2.3.2 Contact Area Belt Model

This model computes the slip quantities that will be implemented in the tire model. It calculates the longitudinal slip and side slip stiffness combined with the longitudinal and lateral stiffness. This model majorly handles the steering torque in car park maneuvers.

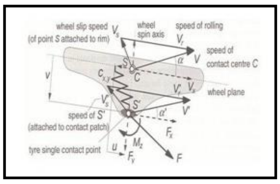


Figure 5.8: Contact Area Belt Model

5.2.3.3 Road Model

The road model gives access to all the ground inputs; the height, velocity, the ground normal vector and a constant adherence that equals 1.This normal vector is essential since it will lead to the calculation of the tire frame. In this study a simple flat road is considered for Steady State On Centre Handling Test.

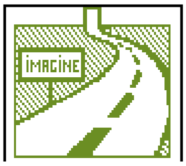


Figure 5:9: Road Model

5.2.3.4 Adherence Model

The adherence given to the tire model is a modulation of the signal input given by the user. It is multiplied by a modulation coefficient called "tire characterization reference adherence". This parameter of the sub-model refers to the adherence coefficient at which the tires models are defined. In this study the adherence coefficient is taken as 1.

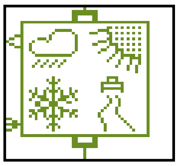


Figure 5.10: Aherence Model

5.2.3.5 Tire Model

It is a widely used semi-empirical tire model to calculate steady-state tire force and moment characteristics. It is used in the field of Vehicle Dynamics and is based on the Principles of Magic Formulae. This model was developed by H. Pacejka, from The Delft University of Technology. These Tire models compute the longitudinal and lateral forces as well as the X, Y and Z torques from the vertical load and slip coefficients.

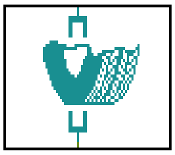


Figure 5:11: Tire Model

5.2.4 Chassis Module

The number of degrees of freedom of the system depends on the number of bodies and the nature of joints used for kinematic constraints. The chassis module is the central module of the 1D Vehicle Model, to which all other sub-modules mentioned above are connected.

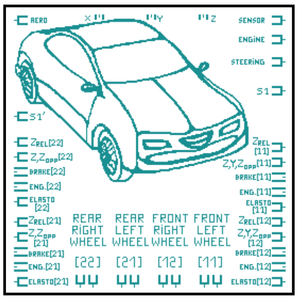


Figure 5.12: Chassis Module

The 18 DOF chassis model is a multibody system containing several pieces, Front Carbody Solid, Rear Carbody Solid, Steering Rack, Spindle, Wheel and all the Mechanical Joints between these elements.

5.2.4.1 Inputs required for Chassis Module

5.2.4.1.1 Kinematics & Compliance Tables

The inputs to the chassis module are the 20 Kinematic & Compliance tables which are generated through K&C Designer Application as discussed in the Section 3.1.2 for the Front Left(11), Front Right(12), Rear Left(21) & Rear Right(22) wheels.

5.2.4.1.2 Mass Inertia Properties

Mass and inertia properties contains the parameters for the following components:

1. The sprung mass i.e. carbody and steering rack
2. The four un-sprung masses i.e. spindles and wheels

5.2.4.1.3 Explicit State Initial Values

Explicit state initial values are the values of the state variables at the beginning of the simulation. To perform an initialization means, to set initial values for all the state variables of the model. Some of the parameters required for initialization are given below:

1. Absolute velocity of carbody-C.G.along X axis (longitudinal axis) which is the initial vehicle speed.
2. Absolute position of carbody C.G. along Z (vertical axis).
3. Pitch Euler angle.
4. Wheels rotary velocities depending on the vehicle initial speed.

6 SIMULATION RESULTS & ANALYSIS

6.1 Standard Sinusoidal Steer Test

1. Drive the vehicle at 100 kmph speed in straight line.
2. Apply one full period sinusoidal steering wheel input with a frequency of 0.2 Hz.
3. Take the data while steering wheel is rotated both to left and right. All data shall be taken in one direction followed by all the data in other direction.
4. Increase the steering wheel input stepwise up to a magnitude sufficient to produce the desired lateral acceleration. Standard acceleration level is 4 m/s^2.
5. Perform at least 3 test runs for each combination of speed and steering.

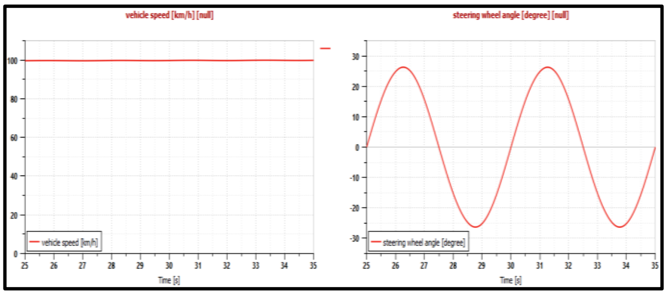


Figure 6.1: ISO_7401 Boundary Conditions Implemented InSteady State OnCentre Handling Test

The Steady State On Centre Handling Test undertaken in this Project is implemented upon by the Boundary Conditions which are a part of ISO 7401 Standard Criteria. The list of the Boundary Conditions is as follows:

1. Sinusoidal steering input is applied to the vehicle at 100km/hr on a straight road.
2. The steering wheel angle is increased till the lateral acceleration achieved is between 0.4g-0.5g.
3. The corresponding steering wheel angle measured is 35degrees.

6.2 Steering Sensitivity and Steering Angle Deadband

Steering Sensitivity[g/100 degSWA]: The variation in the lateral acceleration on a level road with respect to change in steering wheel angle in a given test condition is known as the Steering Sensitivity. It is calculated by taking a slope of the plot obtained between the Steering Wheel Angle vs Lateral Acceleration.

Steering Angle Deadband [deg]: Steering Angle Deadband is described as the phase wherein, any change in the position of steering wheel, brings about no observable response in position of the road wheels. It clearly states as to how much steering wheel angle is necessary to get a vehicle response in lateral acceleration. It is calculated at '0'g lateral acceleration.

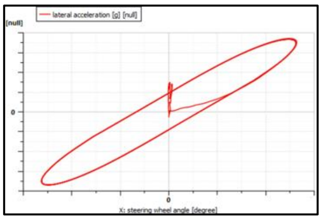
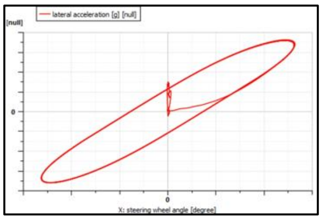
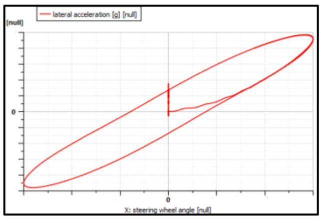
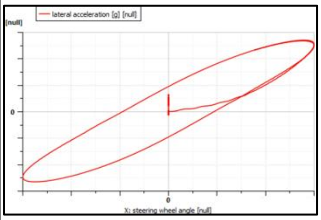
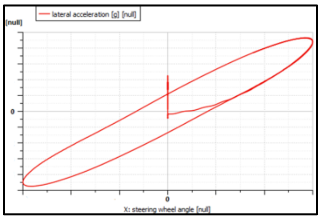
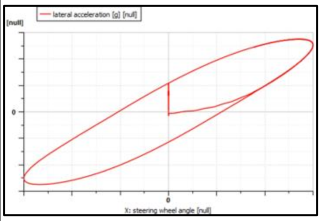
 Vehicle Model 1_ Conventional	 Vehicle Model 1_EV
 Vehicle Model 2_ Conventional	 Vehicle Model 2_EV
 VehicleModel 3_ Conventional	 Vehicle Model 3_EV

Table 6.1: Plots on Steering Wheel Angle vs. Lateral Acceleration_Conventional vs. EV_Vehicle Model 1, 2 & 3

6.2.1 Observation on Steering Sensitivity

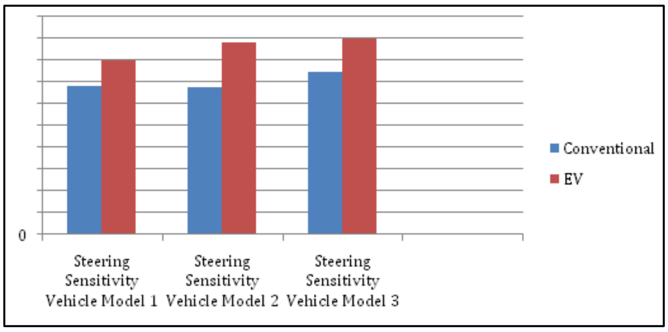


Figure 6.2: Comparative Analysis of Steering Sensitivity between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3

As observed in Figure 6.1, the Steering Sensitivity of the Electric Variant is higher as compared to it's Conventional Variant for the Vehicle Models 1, 2 & 3. Since the rear axle weight of the electric variant is higher,

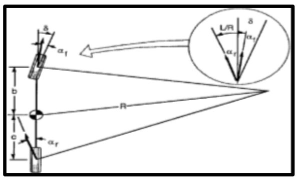


Figure 6.3: Bicycle Model in Vehicle Dynamics

hence the CG of the vehicle lies closer to the rear axle. By applying the principle of Bicycle Model from Vehicle Dynamics, we observe that, during cornering higher lateral forces are produced at the rear axle wheels causing higher slip angles at the rear. This in-turn reduces the under-steer gradient & helps the vehicle to maintain same curve radius, even at higher speeds with lower steering angles. Hence the steering sensitivity of the electrical variant is higher as compared to it's conventional variant for the Vehicle Models 1, 2 & 3.

6.2.2 Observation on Steering Angle Deadband

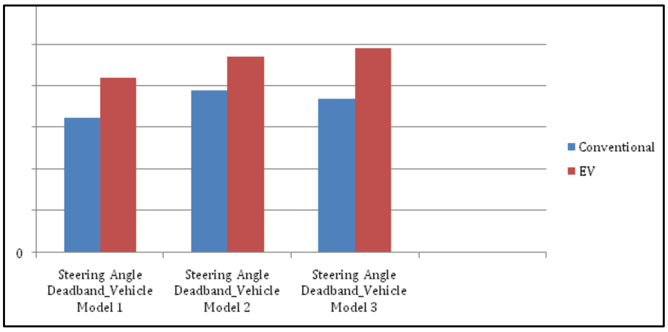


Figure 6.4: Comparative Analysis of Steering Angle Deadband between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3

As observed in Figure 4.4, the Steering Angle Deadband in the Electric Variant is higher as compared to it's Conventional Variant for the Vehicle Models 1, 2 & 3. A dead band is a region wherein, even though the steering wheel is being steered, no inputs are being recorded at the road wheels, so the vehicle continues on its recommended path. For a vehicle with a higher steering sensitivity any change in steering wheel angle would cause the vehicle to have a relatively higher lateral acceleration. Every steering wheel angle will generate different values of lateral acceleration, resulting in different self-aligning moments of the tires producing varying frictional forces in the steering system. The increased steering friction at the rack leads to the reduction in steering response; hence it might be one of the factors leading to steering angle dead-band being higher in electric variant.

6.3 Torsion Rate

Torsion Rate [Nm/100degSWA]: Torsion Rate is defined as the variation in the steering wheel torque with respect to change in steering wheel angle. It assists the driver in judging the stiffness level of the steering and analyzes the appropriate intensity of the torque required to turn the steering wheel. It is calculated by taking a slope of the plot obtained between the Steering Wheel Angle vs Steering Wheel Torque.

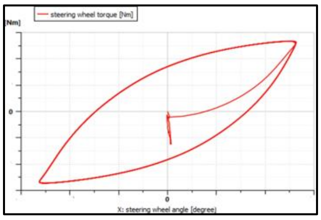
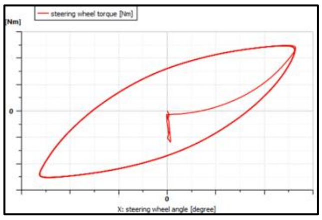
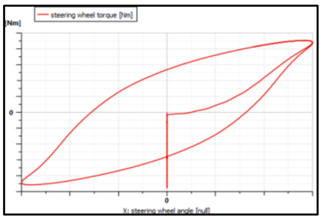
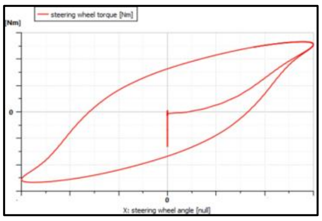
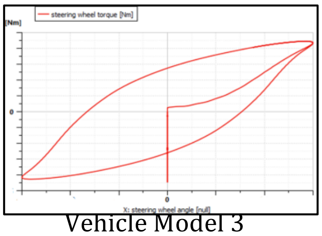
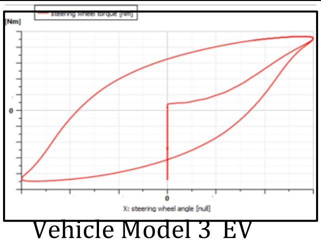
 Vehicle Model 1 _ Conventional	 Vehicle Model 1_EV
 Vehicle Model 2_ Conventional	 Vehicle Model 2_EV
 Vehicle Model 3	 Vehicle Model 3_EV

Table 6.2: Plots on Steering Wheel Angle vs. Steering Wheel Torque_Conventional vs. EV_Vehicle Model 1, 2 & 3

6.3.1 Observation on Torsion Rate

As seen in Figure 4.5, the Torsion Rate in Conventional Variant is observed to be higher as compared to it's Electrical Variant for the Vehicle Models 1 & 3.The ECU takes up the torque applied by the driver, steering wheel

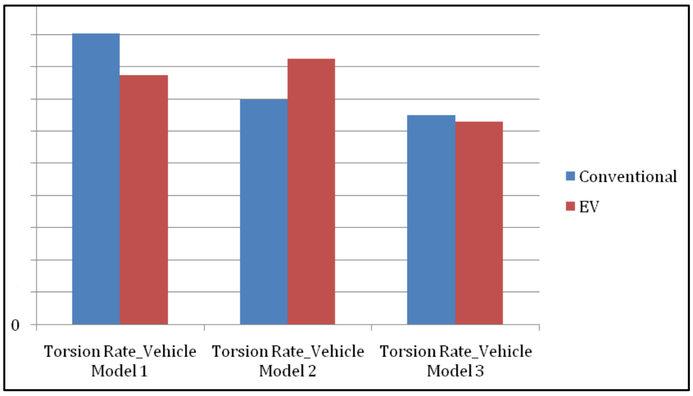


Figure 6.5: Comparative Analysis of Torsion Rate between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3

angle, steering wheel speed and vehicle speed signal & generates the appropriate steering wheel torque. Since the front axle weight of the conventional variant is higher as compared to it's electrical variant, the slip angles and the lateral forces developed at front wheel contact patch during cornering are higher. Hence the magnitude of the aligning torque generated at front wheel contact patch is higher in-case of conventional variant. The under-steer gradient for the conventional variant is higher as compared to it's electric variant making the vehicle to turn with higher steering wheel angles. Hence the torsion rate is observed to be higher in conventional variant compared to it's electric variant for vehicle models 1 & 3.

As an exception, the Torsion rate is observed to be lower in Conventional Variant as compared to it's Electrical Variant, in Vehicle Model 2. The ECU after taking the input signals as mentioned above, does a constant comparison between the steering wheel torque applied and the actual amount of torque required at the rack pinion as per the driving conditions. A cyclic comparison is done by the PID controller unit between the two torque values as mentioned above. Then the appropriate current signal is sent to the assist motor to provide the required assist torque. The assist torque is calculated by looking up the assistance curve table algorithms fed into controller unit. The curve-based assistance characteristics are helpful to realize continuous and uniform assistance and its curve shape can be adjusted according to real requirements. In this manner, the manipulation of the steering assist curves will help in improving the reduced torsion rate value in the Conventional Variant of Vehicle Model 2.

6.4 On Centre Yaw Gain and Yaw Gain Deadband

On Centre Yaw Gain [deg/sec/100 degSWA]: Yaw is an indication of a vehicle's rotation about it's vertical axis. On Centre Yaw Gain is the variation in the yaw rate with respect to change in steering wheel angle. It signifies the tendency of the vehicle to deviate from the vertical axis with variation in steering wheel angle. It is calculated by taking a slope of the plot obtained between the Steering Wheel Angle vs Yaw Rate.

Yaw Gain Deadband [deg]: Yaw Gain Dead-band is defined as the delay in the variation of yaw gain for a particular phase of changing steering wheel angle. Vehicle turning get affected due to this dead-band. It signifies as to how much steering wheel angle is necessary to obtain a yaw gain. It is calculated at '0' deg/sec yaw rate.

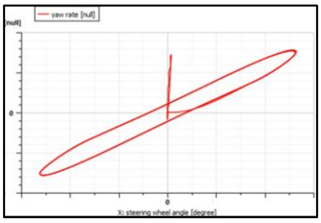
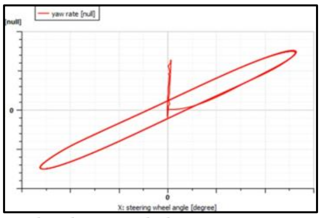
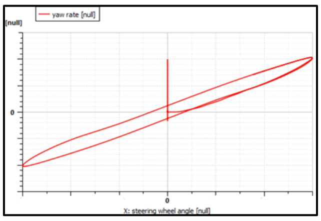
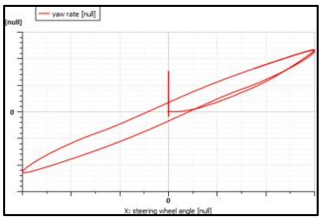
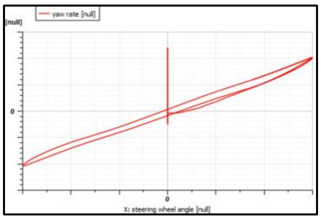
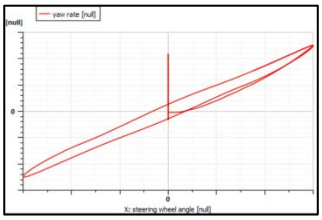
 Vehicle Model 1_ Conventional	 Vehicle Model 1_EV
 Vehicle Model 2_ Conventional	 Vehicle Model 2_EV
 Vehicle Model 3_ Conventional	 Vehicle Model 3_EV

Table 6.3: Plots on Steering Wheel Angle vs. Yaw Rate_Conventional vs. EV_ Vehicle Model 1, 2 & 3

6.4.1 Observation on On Centre Yaw Gain

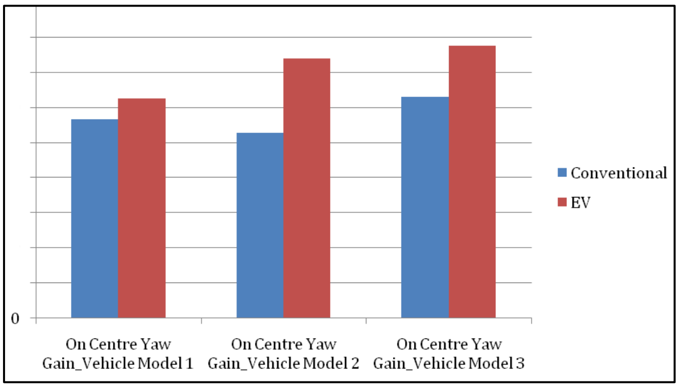


Figure 6.6: Comparative Analysis of On Centre Yaw Gain between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3.

As observed in Figure 6.5, the PAT parameter, On Center Yaw Gain is higher in Electrical Variant as compared to it's Conventional Variant for Vehicle Models 1, 2 & 3. As discussed in previous sections, since the rear axle weight of electric vehicle is higher due to the heavier battery pack, hence by applying the principle of "Bicycle Model-Vehicle Dynamics", the reduction in the under-steer gradient occurring, increases the turning rate of the electric vehicle. This makes the electric vehicle to turn at faster rate during a sinusoidal steer test, as compared to it's conventional variant. Yaw rate of the vehicle is majorly affected by the lane change occurring during cornering due to the dynamic load transfer occurring in the lateral direction of the vehicle. This in-turn contributes to the increase in the yaw rate of the electrical variant.

6.4.2 Observation on Yaw Gain Deadband

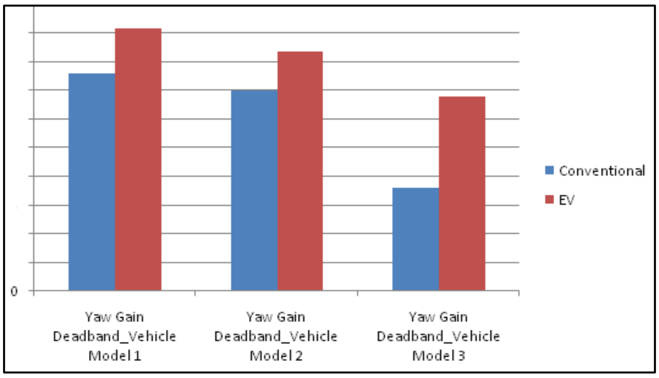


Figure 6.7: Comparative Analysis of Yaw Gain Deadband between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3.

As observed in Figure 6.6, the Yaw Gain Deadband is higher in Electrical Variant as compared to it's Conventional Variant for Vehicle Models 1, 2 & 3. Yaw Gain Deadband is the delay in the variation of yaw gain for a particular phase of changing steering wheel angle. Since the Electric Variant has a higher steering sensitivity, any variation in steering wheel angle would cause the vehicle to have a relatively higher lateral acceleration. Varying steering wheel angle values during cornering test will generate different values of lateral acceleration, resulting in different self-aligning moments of the tires producing varying frictional forces in the steering system. The turning rate of the electric variant being higher due to the higher rear axle weight, the increased frictional forces in the steering system leads to the reduction in yaw gain response.

6.5 Steering Wheel Torque @0_g and @0.3_g Lateral Acceleration

Steering Wheel Torque [Nm] is the torque applied on the steering wheel by the driver. Steering wheel torque at '0'g Lateral Acceleration signifies the friction in the steering system. Steering wheel torque at '0.3'g Lateral Acceleration is the measure of steering effort.

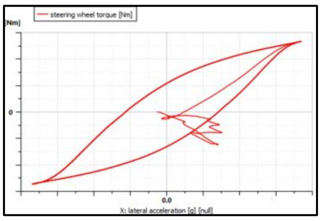
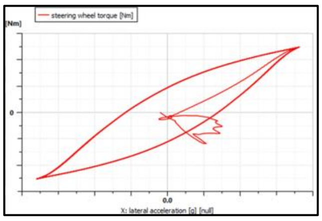
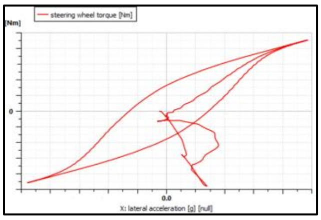
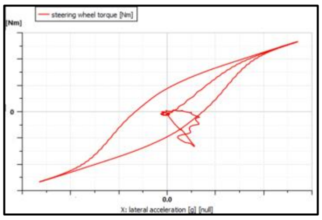
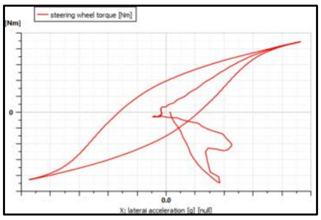
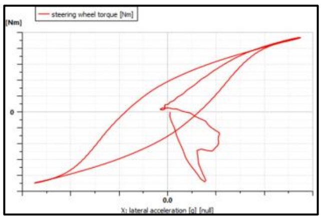
 Vehicle Model 1_ Conventional	 Vehicle Model 1_EV
 Vehicle Model 2_ Conventional	 Vehicle Model 2_EV
 Vehicle Model 3_ Conventional	 Vehicle Model 3_EV

Table 6.4 Plots on Lateral Acceleration vs. Steering Wheel Torque_Conventional vs. EV_ Vehicle Model 1, 2 & 3

6.5.1 Observation on Steering Wheel Torque @0g and @0.3g Lateral Acceleration

As observed in Figures 6.7 and 6.8, the Steering Wheel Torque at '0'g and '0.3'g Lateral Acceleration is observed to be higher in Conventional Variant as compared to it's Electric Variant for Vehicle Models 1 and 3. The under-steer gradient being higher in case of the conventional variant as compared to it's electric variant makes the vehicle to turn with higher steering wheel angles leading to higher values of Steering Wheel Torque. As an exception, the Steering Torque values at '0'g and '0.3'g Lateral Acceleration are observed to be lesser incase of the Conventional Variant for Vehicle Model 2.As discussed in section 4.2.1, the manipulation of the steering assist curves will assist in improving the steering torque values for Vehicle Model 2.

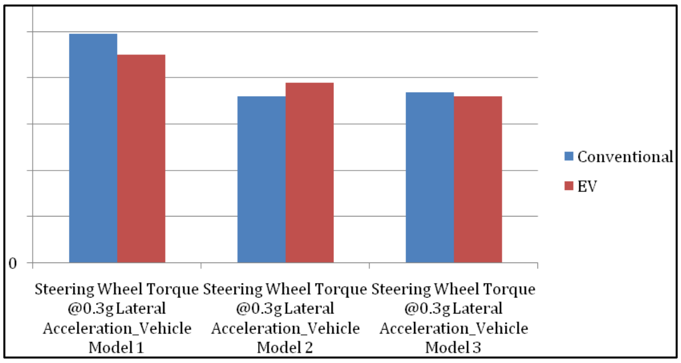


Figure 6.9: Comparative Analysis of Steering Wheel Torque @0.3g Lateral Acceleration between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3.

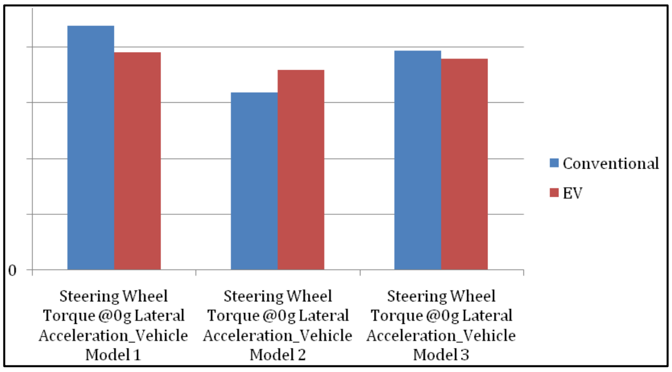


Figure 6.1: Comparative Analysis of Steering Wheel Torque @0g Lateral Acceleration between the Conventional vs. EV Variants for Vehicle Models 1, 2 & 3.

7 CONCLUSION

The parameter Steering Sensitivity is observed to be higher for the Electric Variants, since the reduction in understeer gradient makes the EV variant to turn with a lesser turning radius during the cornering test. The Steering Angle Deadband is observed to be higher in the Electric Variants. Due to the higher steering sensitivity in electric variants, varying steering wheel angles brings about varying values of lateral acceleration and aligning torques & the large frictional forces generated leads to the occurrence of higher steering angle deadband.

The parameter Torsion Rate is always higher in Conventional Variant. But the variation of torsion rate between the conventional & EV variants of vehicle model 2 can be regulated by the real time manipulation of the steering assist curves.

The parameter On Centre Yaw Gain is observed to be higher in Electric Variants majorly because the increase in the rear axle weight of the EV due to heavier battery pack, makes the vehicle to turn at a faster rate during the Sinusoidal Steer Test. The Steering Angle Deadband in the Electric Variants is observed to be higher due to higher frictional forces into the steering system due to higher steering sensitivity and faster turning rate.

The Steering Wheel Torque developed at '0'g and '0.3'g Lateral Acceleration are observed to be higher in Conventional Variants. Any reduction in the steering wheel torque in the conventional variants can be improvised by the real time manipulation of the steering assist curves.

8 FUTURE SCOPE

In this Project we performed a simulation test for Steady State On Centre Handling of the Vehicle. We may also carry out a simulation for different types of vehicle path which may include Circular and Moose test path.

Optimization of the K&C Parameters to improve the performance of the Suspension, Steering & Tire Geometry Systems will definitely assist in enhancing the Dynamic Behavior of the EPAS System.

PID controller which is a generic control mechanism is widely used in EPAS System, due to their low cost, inexpensive maintenance, as well as simplicity in controller design and operation. In order to optimize the dynamic behavior of the EPAS system, the implementation of Binary Coded Genetic Algorithm [BCGA] into the PID Controller System will help achieve the desired PAT Targets.

We may also implement Frequency Response Test, in which the responses to Sinusoidal Excitations are studied in the Frequency Domain. Bode Plots are used to detect amplification factor and Phase Delay of the model outputs as a function of frequency. Simulations are run with different configurations, by iteratively changing values for Cornering Stiffness and Weight Distribution, to detect which vehicle parameters influence vehicle handling at different frequencies.

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