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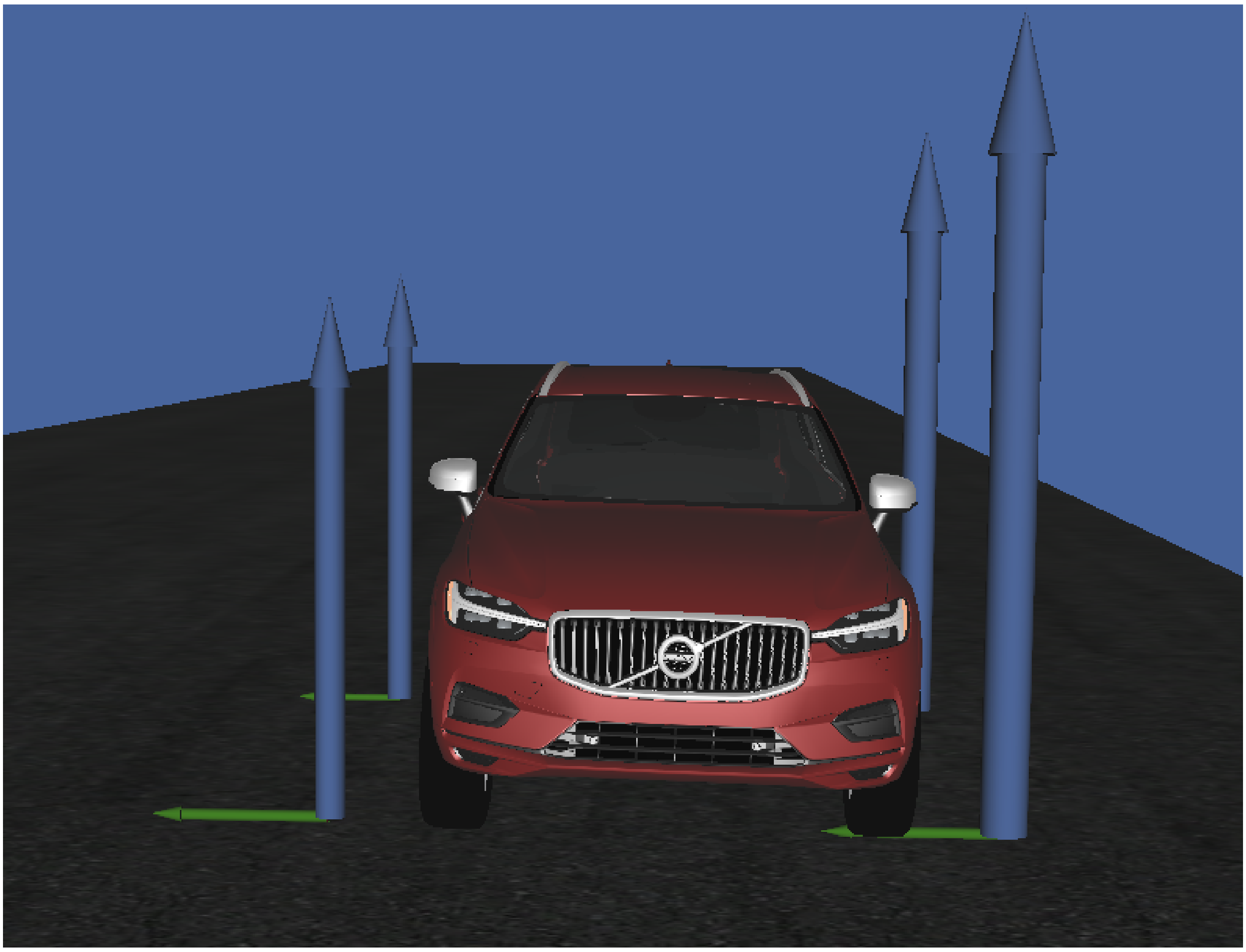

The effect of active forces on Wheel Suspension Kinematic Design parameters

Master's thesis in Automotive Engineering

CHINTALAPUDI ADITHYA REDDY
MANIKANTA VENKATESH

Department of Mechanics and Maritime Sciences
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2020

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Chintalapudi Aditya Reddy
Manikanta Venkatesh

Academic Supervisor:
Bengt J H Jacobson
Professor and Group leader-Vehicle Dynamics
Department of Mechanics and Maritime Sciences
Chalmers University of Technology

Examiner and Academic Supervisor:
Ingemar Johansson
Professor of the Practice, Vehicle Engineering and Autonomous Systems
Department of Mechanics and Maritime Sciences
Chalmers University of Technology

Industrial Supervisor:
Johan Ericson
Mechanical Systems Architect, Suspension Architecture and Structure
Wheel Suspension, Volvo Car Corporation

Technical Advisor:
Per Carlsson
Team Manager, Suspension Kinematics and Tuning
Wheel Suspension, Volvo Car Corporation

Master's Thesis 2020:15
Department of Mechanics and Maritime Sciences
Division of Vehicle Engineering and Autonomous Systems
Chalmers University of Technology
SE-412 96 Gothenburg
Telephone +46 31 772 1000

Cover: Illustration of a simulated Double Lane Change maneuver in VI-CarRealTime

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MANIKANTA VENKATESH
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Abstract

For passenger cars, passive suspension systems are designed to adapt to a wide range of driving conditions and hence, vehicle suspension design is associated with certain trade-offs between several vehicle attributes (ride comfort and road holding, to name one). Active suspensions however, gives the potential to minimise some of these trade-offs while still being able to adapt to almost all driving conditions.

This master thesis work aims at studying the effects of active spring and active damper forces on suspension kinematic parameters through full vehicle simulations. Further, possible modifications to the suspension geometry such as removing anti-roll bar component, having a zero anti-geometry (anti-lift, anti-dive and anti-squat) for front and rear suspension and changes to the suspension kinematic targets in the presence of active forces are also investigated.

The methodology adapted in the thesis work is an iterative process of comparing the performance of active suspension performance against the passive suspension baseline. This "baseline" performance of the passive suspension system is established by simulating a chosen number of driving maneuvers with a baseline suspension geometry and setup as that of a current production car. The active forces are then introduced in the suspension system and the same chosen number of driving maneuvers are simulated to compare the suspension performance. Further, the third set of simulations are performed for an active suspension system with varied kinematic targets and modifications to suspension geometry.

The results from these three sets of simulations are compared, which provides insights to the possible modifications that can be done to the suspension geometry in the presence of active forces. The main outcome is the suspension behavioral patterns in full vehicle simulations with active suspension, with varying parameters of suspension design. The report also details about the results from the driving simulator which are correlated to the simulation results. A case study is discussed in detail towards the final chapters, regarding the effect of active damping forces on vehicle steering demands.

Keywords: active suspension, vehicle attributes, full vehicle simulations, varied kinematic targets.

Acknowledgements

We would like to express our deep sense of gratitude to our academic supervisors from Chalmers, Ingemar Johansson and Bengt J H Jacobson for their continuous guidance and support throughout this thesis. Their knowledge and subject matter expertise is invaluable and helped us overcome the difficulties we faced during this thesis work.

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We are extremely grateful to Per Carlsson for providing us the opportunity to conduct our master thesis project in the Wheel Suspension department at Volvo Car Corporation. Per was very helpful in giving us right technical directions and providing us necessary technical and administrative support wherever needed, to carry out this thesis work.

Our special thanks to all other employees at Volvo Car Corporation who helped us during our thesis work. We thank them for their support in providing us the necessary data and also guiding us during the course of this thesis.

Finally, we would like to thank all our friends and family for their constant love and support.

Chintalapudi Adithya Reddy & Manikanta Venkatesh
Gothenburg, October, 2020

Abbreviations

VCC - Volvo Car Corporation
CAE - Computer Aided Engineering
PSD - Power Spectral Density
CoG - Center of Gravity
ARB - Anti-Roll Bar
SLA - Short-Long Arm(Suspension)
DLC - Double Lane Change
CRC - Constant Radius Cornering
ISO - International Standards Organisation
FFT - Fast Fourier Transform
RCH - Roll Center Height
IC - Instantaneous Center
RC - Roll Center
RMS - Root Mean Square
DoF - Degree of Freedom

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1
Introduction

1.1 Background and problem description

The development of a suspension with active dampers has potential to provide the suspension designer greater freedom to change the suspension kinematics. Passive suspension systems are developed to operate in a wide range of driving conditions and hence, it comes with certain compromises or trade-offs between several design factors and vehicle dynamics attributes. In passive suspension systems, suspension kinematics and suspension geometry control some of the forces that are applied, but this tends to compromise other attributes. Development of an active suspension system has the potential to reduce these compromises in some cases. Further, active suspension systems will improve ride comfort and vehicle dynamics significantly as compared to passive suspension systems.

By adding active dampers and active spring forces to the passive suspension system there are opportunities to improve the overall suspension performance and attributes in passive kinematic geometries. There are also opportunities to simplify the kinematics and suspension geometry and still improve the overall suspension performance. In other words, some trade-offs during suspension design could be less significant in case of active systems as compared to its passive counterpart.

However, to make best use of active dampers and active spring forces, it is important to have a control strategy to suit the attribute targets. It is also in the best interest to develop the complete suspension system to suit the integration of active dampers, thereby making the overall suspension design more efficient.

1.2 Scope

The thesis work aims to investigate the aforementioned advantages of the active suspension in a passenger vehicle by analysing parameters through full vehicle simulations for different driving maneuvers. Further, a thorough comparison and analysis of the active suspension system will be done by comparing it with it's passive counterpart. This analysis will help in making possible simplifications to suspension kinematics or suspension geometry. Changes to the kinematics and hard points that are feasible when replacing the passive suspension dampers with an active suspension actuator will be investigated. The concept design for an active suspension system where the effects of active forces will be considered when the kinematics is developed.

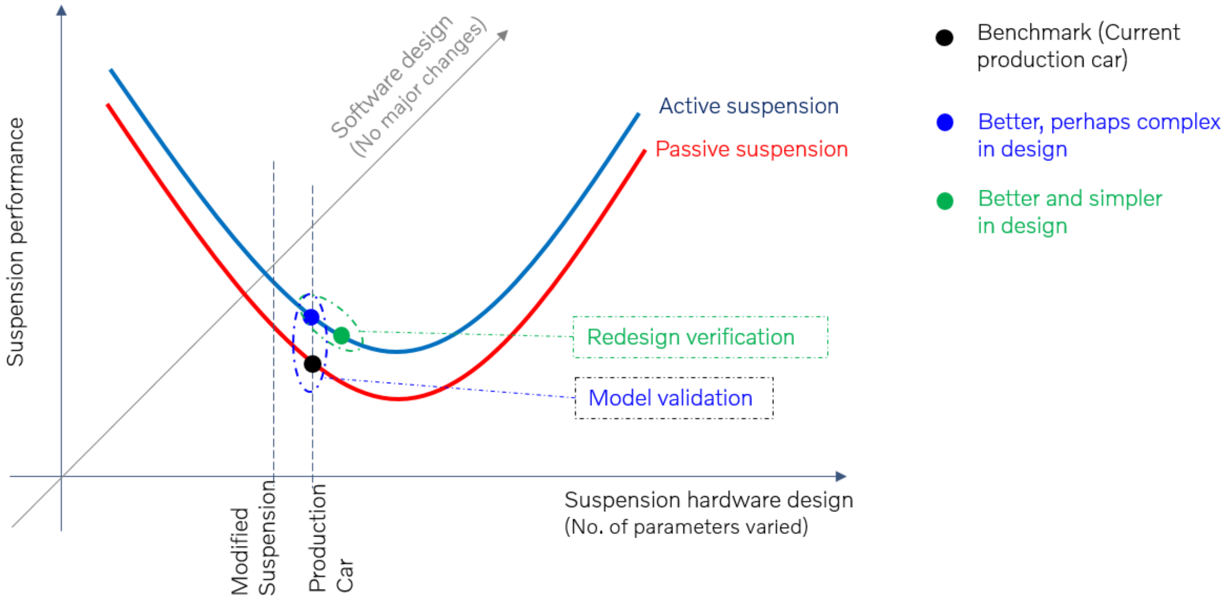

Figure 1.1: Illustration of the thesis objectives

The thesis objective can be better illustrated and explained with the help of figure 1.1. The plot can be regarded as a measure of suspension performance of passive and active suspension systems. X-axis indicates the parameters that will be varied during a suspension design/tuning and Y-axis indicates the change in suspension performance due to a change in parameter in X-axis.

- Black point : Indicates the current production car having a passive suspension system with ARB, passive springs and dampers
- Blue point: Indicates a standard active suspension when only passive dampers are replaced by active dampers and no other change in suspension kinematics
- Green point - Indicates a "tuned" active suspension when passive dampers are replaced by active dampers and suspension kinematics are modified and/or simplified.

The main objective is to compare green with the black point (which is regarded as the baseline for measuring suspension performance) and make conclusions based on these comparisons. A comparison between Black and Blue point will be made to validate the co-simulation model which is used for full vehicle simulations with an active suspension system.

This thesis work focuses on the active suspension kinematics design and not on the development of controller for active suspension. For example, one of the major thesis deliverable is to investigate the possibility of removing the anti-roll bar when active dampers are added to the design and the corresponding change in suspension parameters to compensate for the removal of anti-roll bar, if required.

1.3 Deliverables

The main goal of this thesis work is to analyse the effects that active forces has on the wheel suspension and further look into the possibility of simplification of suspension kinematics and geometries.

- Present the baseline performance for the selected passive suspension systems - SLA(Short-Long Arm) front suspension and Integral link rear suspension in the selected driving conditions.This baseline performance will serve as a reference to compare the active suspension system performance in full vehicle simulations.
- Present the analysis of suspension performance when active damping and active spring forces are introduced.
- Suggest possible changes and simplifications to the suspension kinematics and geometries when active dampers are introduced.
- Compile tools and engineering design guidelines for the integration of active dampers and active spring forces.
- To document all the observations and methodology chosen in the form of final thesis report.

1.4 Limitations

To keep the thesis scope focused on suspension kinematic analysis and not on the controller development, some limitations are applied to the controller. These, along with other limitations pertaining to thesis work are described below:

- The vehicle model used in the full vehicle simulations will be based on the complexity provided by VCC. These models are exported from MSC ADAMS/Car to VI-CarRealTime software. The actuator models used for simulations are also obtained by VCC from a development partner. Further, the co-simulation model consisting of active suspension controller and the vehicle model is provided by VCC. Fidelity of these exported models will not be validated. Also, these VI-CarRealTime models contain the vehicle behavioural parameters in the form of lookup tables or gradients and is not a Multi-body Simulation model.
- Compliance analysis will not be performed after a change in suspension kinematics.
- A chosen number of parameters will be analysed for the vehicle performance but will be kept the same for both passive and active systems.
- Limited number of maneuvers will be studied to assess the suspension performance.
- Effect of different road-tire friction coefficients on full vehicle simulations will not be studied. All simulations will be run for high friction conditions.
- The controller model and the actuator models are obtained from the stakeholder and development partner. These controllers will only be tuned as per

the thesis work requirements and no significant changes will be made.
- The performance of different control strategies will not be investigated.
- Physical testing of the controller will not be performed.
- The hardware implementation of the controller and the energy request of the controller will not be part of the scope and hence, will not be studied.

1.5 Stakeholder

This master thesis work is carried out at the Wheel Suspension department of Volvo Car Corporation (VCC), with Per Carlsson and Johan Ericson as Industrial supervisors. Academic supervisors are Ingemar Johansson and Bengt J H Jacobson from Chalmers University of Technology, Department of Mechanics and Maritime Sciences.

1.6 Methodology

The following methodology will be followed during the course of this project :
- A thorough literature study on the suspension kinematics and active suspension systems .
- Study the suspension performance for front SLA and rear Integral link suspension system, with a transverse leaf spring in the rear.
- Establish a baseline performance for the selected passive suspension systems by studying the vehicle performance in different vehicle dynamics and ride comfort simulations.
- Include active damping forces in selected suspension systems and analyse the improvements to the suspension performance by studying the different maneuvers in vehicle dynamics and ride comfort simulations.
- Investigate if the kinematics and the suspension geometries can be simplified and still improve the suspension performance relative to the passive system when active dampers and active forces are introduced. These include, for example, investigating the possibility of removing anti-roll bar for an active suspension system and tuning the hardpoints further.

1.7 Planning

The road plan for the thesis work is established by using a Gantt chart, indicating the list of tasks to be performed and the timelines for those tasks. This document was continuously updated after progress review meetings which were held every week. Gantt chart served as a good means to prioritise the tasks, keep track of activities and the time available to complete the planned tasks.

2

Theory

This chapter summarises the literature survey conducted during the course of this thesis work. It briefly includes the classification of suspension systems in terms of active and passive components and information on certain suspension design factors which will be the focus of this thesis work.

2.1 Vehicle Suspension

A vehicle suspension is a system of control arms and other linkages that control the kinematics of the wheel and springs-dampers that control the isolation and damping. Suspension systems enables vertical movement of the wheel assemblies relative to the vehicle body and is responsible for maintaining contact with the road at all times during driving scenarios. Suspension also controls the kinematics of the wheels to give the desired suspension performance.

The vehicle suspension not only has to provide good ride comfort to the driver and passengers in the vehicle, but also enable good handling and steerability of the vehicle in all driving scenarios. These two parameters, like many others that a suspension design engineer comes across, are always at odds with each other. Therefore, a compromise or a "trade-off" has to be made between such parameters when designing a suspension system.

One way to classify a suspension is based on how the suspension is controlled i.e., Passive, Semi-active and active suspension systems. Another way is based on the suspension layout which has several types but the thesis work focuses on a Double Wishbone, SLA(Short-long arm) front suspension and Integral Link rear suspension. These are further explained in subsequent sections.

2.1.1 Passive suspension

In case of passive suspension systems, the vertical movement of the wheel assemblies is dictated by the road profile and the suspension is controlled internally without the use of an electronic equipment. The word "passive" implies that no energy is added to the system. Most traditional suspensions are passive in nature. The vertical movements in the passive suspension is controlled by a spring and a damper where the specification is selected to suit attribute targets for the vehicle. Some suspension components relevant to this thesis work are explained in the subsequent sections.

2.1.1.1 Linkages

Linkages are the mechanical connections between the chassis and the wheel assemblies. The main purpose of linkages is to control the movement of the wheel assemblies and to give the expected kinematics and structural performance of the suspension systems [2]. Linkages consists of joints at the endpoints, which can be either ball joints or bushings. Bushings are used to isolate and give the expected elasto-kinematics performance with low friction.

The co-ordinates of these joints called as Hardpoints when represented in the 3D space form the suspension geometry. Hardpoints are the real design parameters of the suspension which dictates the length and inclination of linkages and also influences the dynamic behavior of the vehicle in driving maneuvers [2].

2.1.1.2 Springs

The vehicle can be categorised into sprung mass and unsprung mass. Sprung mass is the vehicle body and chassis, which is supported by springs. Unsprung mass constitutes of all other components which is not supported by springs, including suspension link arms and wheel assemblies.

Suspension springs are responsible to prevent the road shocks from being transmitted to the vehicle components and to maintain the desired ride height. The spring rate is selected to suit characteristics of the vehicle, low spring rate gives better ride comfort while stiffer spring rate gives better control. Springs compress or expand based on the jounce and rebound motion of the wheel. There are several types of passive springs available and the most common one in use is a coil spring due to the ease of packaging and simpler design as compared to other springs.

Some spring parameters that are important in suspension design are:

- $$\text{Spring motion ratio} = \frac{\text{Spring displacement}}{\text{Wheel vertical travel}} \tag{2.1}$$
- Spring rate: It is defined as the amount of force required to compress or expand a spring by one mm of length. It is also known as Spring constant or spring stiffness. The spring stiffness is selected to give the expected control of the vertical wheel movement and the characteristics of the vehicle.
- Spring load is selected to give the expected ride height for the vehicle.

2.1.1.3 Dampers

Dampers(also known as Shock absorbers) are responsible for damping the vehicle oscillations in order to provide better control, improved ride comfort and maintaining tire to road contact. Optimum damping is also necessary to have a good steering ability and to avoid wheel hop. The damper also controls the wheel travel geometrically [5].

Figure 2.1: Spring-Damper system from front suspension of a Volvo XC90[5]

Tires offer a small amount of damping(around 2%) but majority of damping has to be done by suspension damper [7]. The kinetic energy produced due to the vehicle oscillations is dissipated as heat by dampers. In case of passive dampers, damping force is partly dependent on the damper fluid velocity i.e., how fast the fluid moves in the damper tube.

Damper motion ratio which is defined by equation 2.2 is a important parameter which allows optimum damper performance. For a passenger vehicle, a damper motion ratio of 0.8 - 1.0 gives good damper functionality. While a damper ratio of 1 is good for reducing structural loads, it results in a long damper and a ratio greater than 1 often gives problems with the installation because of the length [5].

$$\text{Damper motion ratio} = \frac{\text{Damper displacement}}{\text{Wheel vertical travel}} \tag{2.2}$$

2.1.1.4 Anti-Roll bar

Anti-Roll bar (also known as stabilizer) is an elastic connection between the left and right wheel assemblies[2] and as the name suggests, it helps in reducing body roll. ARB increases the roll stiffness or the resistance to roll when the vehicle goes through a corner.

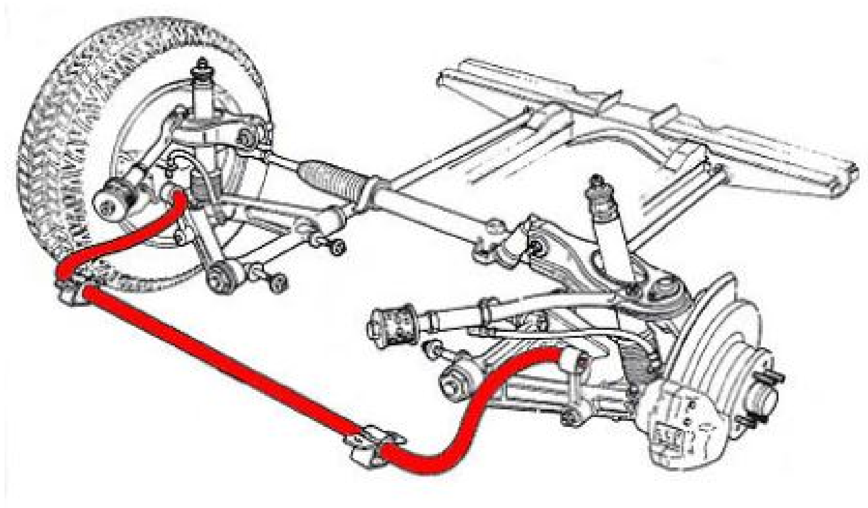

Figure 2.2: Anti-Roll bar for a passive suspension (highlighted in red)

Following are some of the functions of Anti-Roll bar:

- ARB reduces the body roll in corners. The amount of reduction in body roll depends on the total roll stiffness of the vehicle.
- It also helps in balancing the understeer and oversteer conditions of the vehicle by managing the vertical force distribution between front and rear outer tires[5]. Also, the understeer and oversteer behaviour can be tuned by modifying the ratio of front and rear roll stiffness. By increasing the proportion of front roll stiffness, the front outer tire is made to run at a higher slip angle and consequently, a lower slip angle at the rear, causing understeer. If the ratio of roll stiffness is increased at the rear, oversteer behaviour can be observed.

However, ARB is connected such that when one wheel is lifted, the wheel on the other side is also lifted. This results in transmitting forces from vertical movement on one side due to bump, to the other side and can cause reduced ride comfort. Also, having excessive roll stiffness causes the inside wheels to lift-off during hard cornering.

2.1.2 Semi-active suspension

In case of a semi-active suspension, the dampers are controlled electronically while using a conventional spring system (coil or air springs) to control the movement of the wheels. Although the dampers are "actively" controlled and can react to the road irregularities in milli-seconds, the movement of the wheel assemblies relative to the chassis is still dictated by the road profile. Hence, the name "semi-active" suspension system. The conceptual representation of passive, semi-active and active suspension is shown in figure 2.4.

2.1.3 Active suspension

In case of active systems, the suspension is controlled externally by using electronic active controllers and energy is added to suspension externally with the help of actuators. Active dampers produce damping force based on hydraulic tuning and the control system used.
There are three main types of active suspension systems, classified based on their operating bandwidth. Refer figure 2.3, for a classification in terms of energy request and available control bandwidth.
1. Load levelling suspension: 0.1 - 1 Hz
2. Slow-active suspension: 1 - 5 Hz
3. Fully active suspension: 20 - 30 Hz

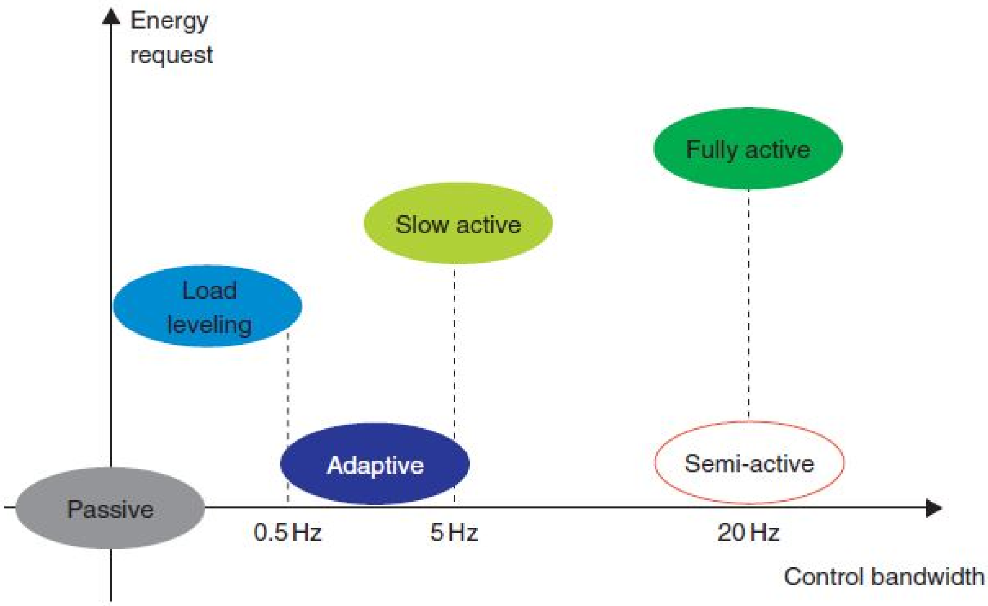

Figure 2.3: Graphical representation of active suspension system classification in terms of energy request with respect to the available control bandwidth[3]

Fully active suspension system is chosen for this thesis work as it provides a larger bandwidth and is able to react faster compared to semi-active suspension. The vehicle vertical dynamics can be broadly classified into body dynamics which is characterised by a bandwidth of 1 - 5 Hz and wheel dynamics which is characterised by a bandwidth of 15 - 20 Hz [3].

As seen from figure 2.3, fully active system also demands higher energy compared to other systems, which is expected. So, a compromise has to be made between active system performance and cost among other factors, when choosing the type of active system for the vehicle.

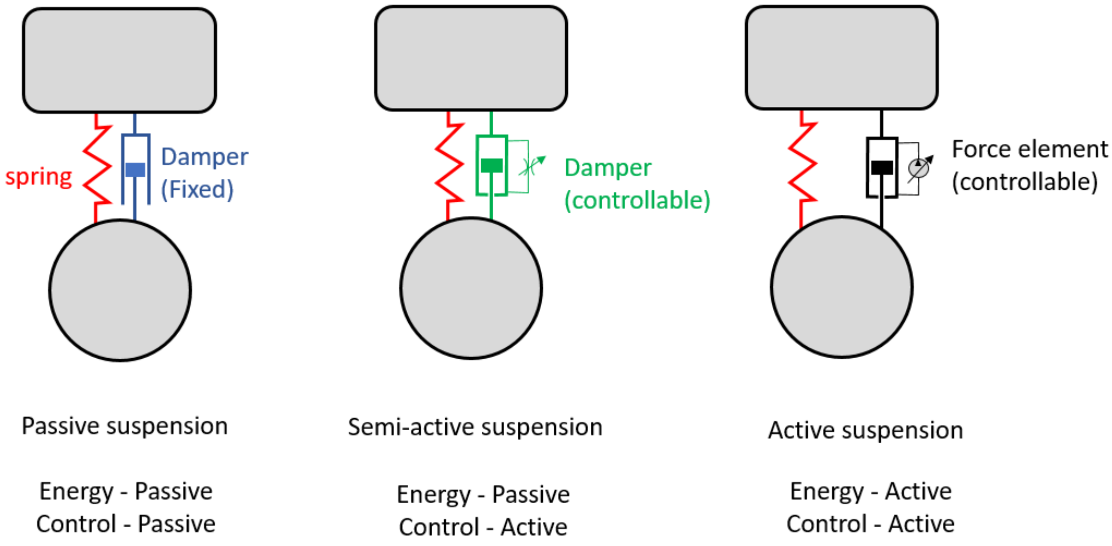

Figure 2.4: Conceptual representation of different suspension types based on the method of suspension control [2]

2.1.4 Suspension systems based on the layout

The vehicle model used in this thesis work has SLA (or double wishbone) suspension in front and Integral link suspension in the rear.

2.1.4.1 SLA suspension

Short-Long arm or Double wishbone suspension consists of two control arms (or transverse links) on either side of the vehicle body, which connects the chassis to the wheel assemblies. The name "double wishbone" comes from the presence of two wishbones controlling the wheels. In case of SLA suspension, the upper control arms are shorter than the lower ones to give the expected wheel control and camber gain and it falls in the Independent suspension category.

SLA suspension provides greater freedom in designing the suspension kinematics. Hence, it is used in vehicles with higher attribute targets for ride comfort and vehicle dynamics. The unequal length control arms influences angular movement of wheels during jounce and rebound, i.e., camber gain and track width variation. Due to the presence of shorter upper control arms, negative camber gain is obtained during jounce and positive camber gain in rebound. This essentially counter acts the camber gain caused by body roll. Another advantage of using SLA type of suspension is that is has greater freedom to select the intended anti-dive and anti-lift behaviour of the vehicle. This can be achieved by designing the pitch center to be over wheel center which reduces the squat on rear wheels (or lift on front wheels).

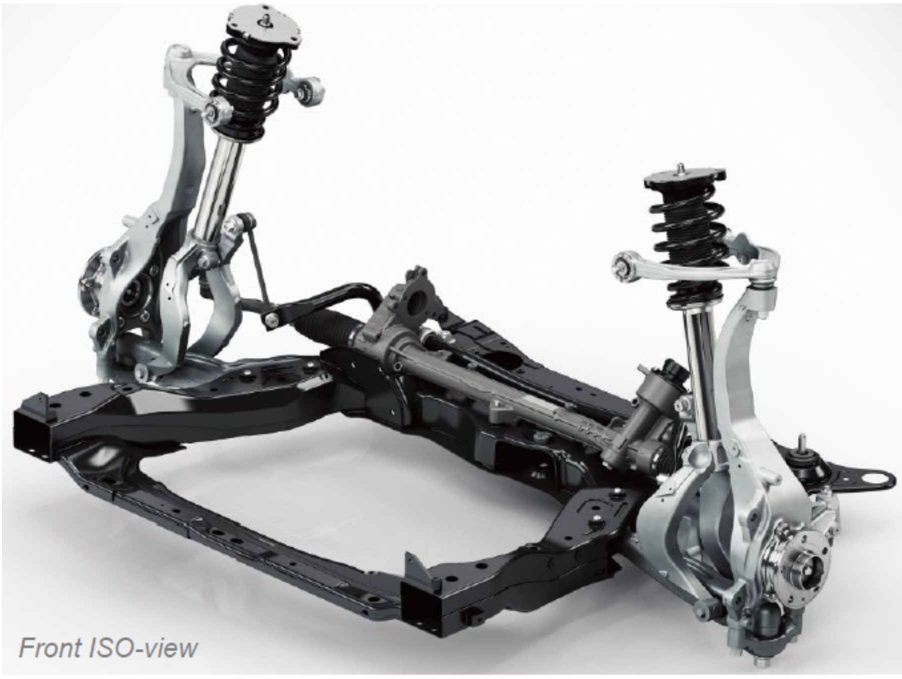

Figure 2.5: Short Long Arm suspension from a Volvo XC90 [5]

2.1.4.2 Integral link suspension

Integral link suspension offers several advantages such as high wind-up stiffness, better anti-lift and anti-squat behaviour of the vehicle and good road noise isolation[5].

Figure 2.6: Integral link suspension from a Volvo XC90 [5]

There are no trailing arms present, hence integral link suspension is easier to package. In case of a Volvo XC90, the suspension is packaged with a single upper control arm (refer figure 2.6). However, Integral link suspension is a complex suspension type which is difficult to optimise and hence difficult to tune and develop further.

2.2 Vehicle Suspension Design

Suspension design is often associated with compromises between various parameters for example, road handling and ride comfort. These trade-offs are well known for a passive suspension system but not for active suspension system. These can be better understood and data-driven decisions can be made with proper knowledge of suspension design parameters. Some of them are explained in this section.

2.2.1 Vehicle Co-ordinate system

There are different vehicle co-ordinate systems or axis systems available based on their intended use. The axis system used in vehicle design at VCC is illustrated in Figure 2.7, with positive X axis pointing to the rear, positive Y axis to the right, positive Z axis in the upward direction and the origin (0,0,0) in front of the vehicle and below ground.

Figure 2.7: Illustration of axis system used in vehicle design

2.2.2 Vehicle movements and DoF

The different movements of the vehicle can be classified as 3 translational movements and 3 rotational movements in the chosen co-ordinate system.

The translational movements being:

- **Heave:** Heave is the vertical movement of the vehicle in the Z-direction as a result of road irregularities.
- **Surge:** It is the longitudinal movement of the vehicle in X-direction during accelerating or decelerating the vehicle.
- **Sway:** Sway is the lateral movement of the vehicle in Y-direction as a result of centrifugal force during cornering maneuver.

The rotational movements being:

- **Pitch:** It is the rotation of the vehicle about Y-axis due to forces from acceleration and braking maneuvers. The sign convention is positive when the nose of the car moves upward (i.e., when accelerating) and negative when the nose of the car moves downward (i.e., during braking).
- **Roll:** It is the rotation of the vehicle about the X-axis arising due to bumps and during cornering maneuvers.
- **Yaw:** It is the rotation of the vehicle about Z-axis and depends on roll stiffness, weight distribution, brake bias and side winds, among other factors.

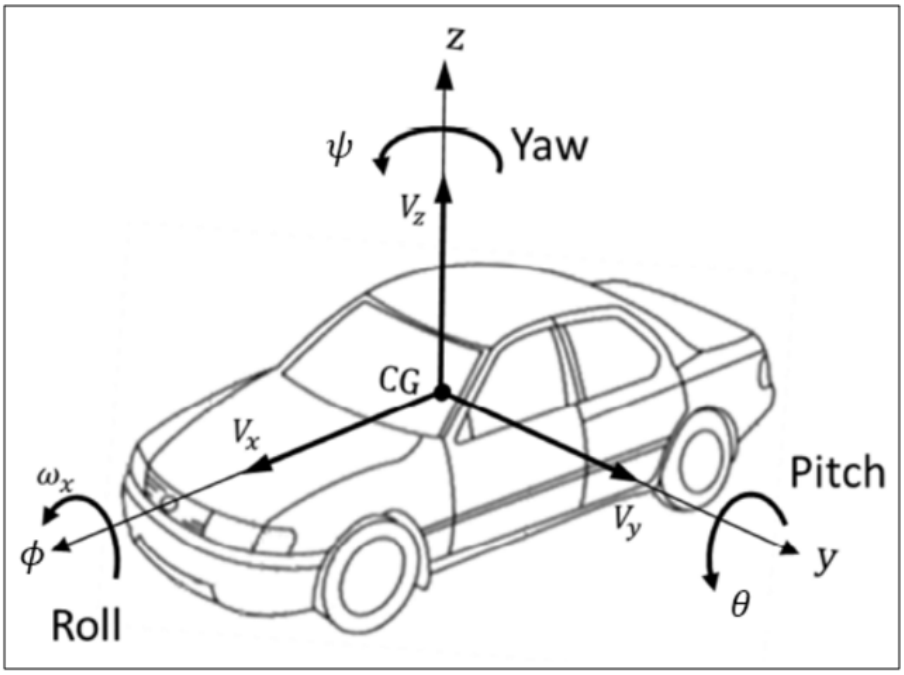

Figure 2.8: Illustration of translational and rotational movements of a car [8]

2.2.3 Suspension parameters

Some important suspension parameters that greatly influence the suspension performance and also vehicle behaviour in driving conditions are explained in this section. These parameters are of interest in this thesis work.

- **Camber angle:** It is the inclination of the wheel center line to the vertical plane, in front view. Camber angle is positive if the wheels are inclined or tilted outwards at the top and negative if the wheels are inclined inwards at the top.

  Camber angle usually ranges from -2° to +2°. In case of passenger cars, camber angle is less than 1°and for racing cars, it is usually a higher negative value. Camber angle contributes to lateral tire force during cornering and positive camber results in lateral wheel forces in the outboard direction [4]. Negative static camber angle with a negative camber gain helps in getting performance

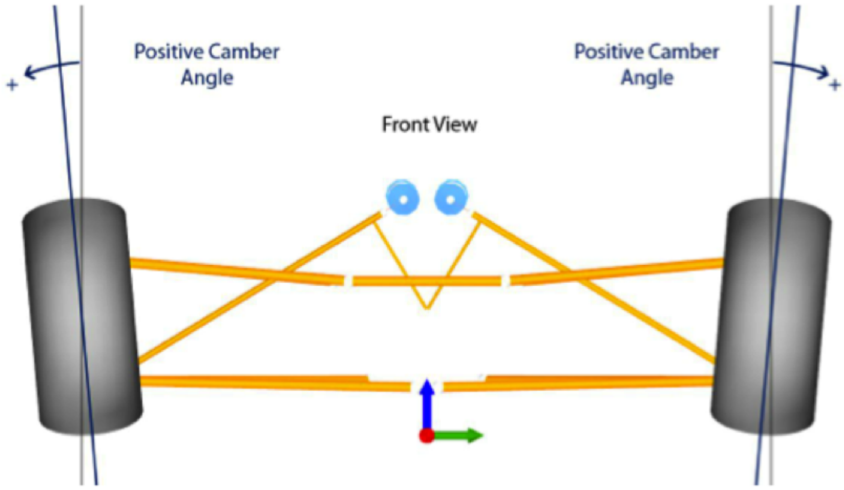

Figure 2.9: Illustration of camber angle [12]

during cornering. Negative camber also produces "camber thrust" which will significantly improve straight line driving. However, increased static camber regardless of positive or negative in value, will exacerbate tire wear.

- **Camber Gain:** It is defined as the change in camber angle with the vertical wheel travel. Negative camber gain implies that camber angle reduces during jounce and positive camber gain implies that camber angle increases during jounce. Camber gain is calculated by taking the slope of camber vs jounce curves.

- **Toe angle:** It is the inclination of the wheel center line to the vertical plane, in top view. Toe-in is when the wheels are pointing towards each other and toe-out is when the wheels are pointing away from each other.

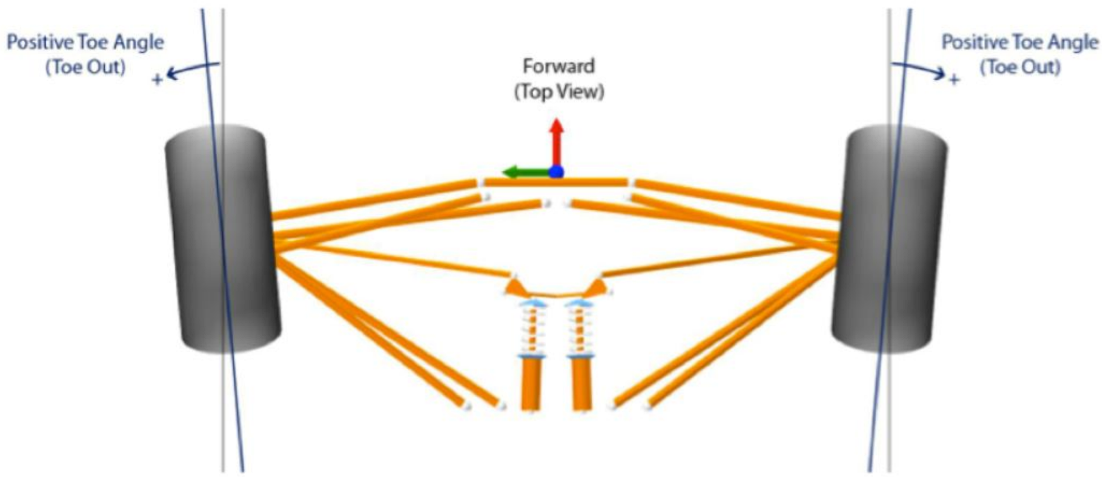

Figure 2.10: Illustration of Toe angle[12]

For front axle, toe-in improves straight line stability and toe-out improves cornering of the vehicle. High values of both will affect tire wear adversely, since excessive toe-in causes outer edge of the tires to wear excessively and increased toe-out will cause the inner edge to wear excessively. Although a zero toe angle is good for improving tire life, a small value of toe angle is added to the suspension design to improve steer-ability of the car.

- **Toe Gain or Bump steer:** It is defined as the change in toe angle during vertical wheel travel. Negative bumpsteer implies that the toe angle reduces during jounce and positive bumpsteer implies that the toe angle increases during jounce. Bumpsteer adversely affects the steering demands of the vehicle.

  It is desirable to have a low bumpsteer value in both front and rear suspension.

- **Roll Center Height:** The height at which Roll Center(RC) is situated from the ground. Roll axis is obtained by connecting front and rear roll centers. Roll centers and roll axis has a significant importance in vehicle handling. Roll Center height also determines the vertical load difference between front and rear axle.

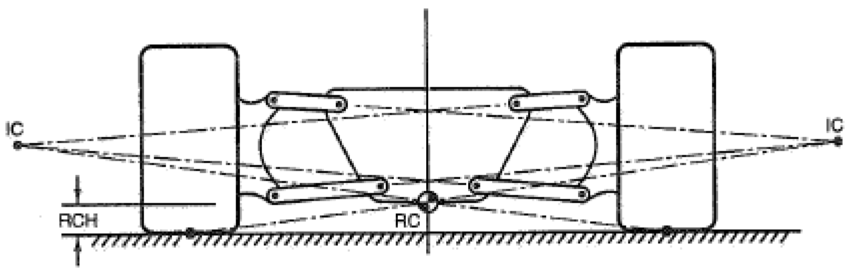

Figure 2.11: Illustration of roll center for a SLA front suspension[6]

Roll center height and roll center variation with the wheel travel is a compromise between the following parameters:
  - Desired vertical load distribution between front and rear axles which consequently influences understeer and oversteer behaviour of the vehicle.
  - Trackwidth variation with the vertical wheel travel

Another important aspect to consider during the suspension design is the difference between front and rear roll centers in the vertical direction when viewed in the XZ direction (refer figure 2.12). Rear RC should be slightly higher than the front RC in order to damp the yawing movements of the vehicle and have roll understeer characteristic.

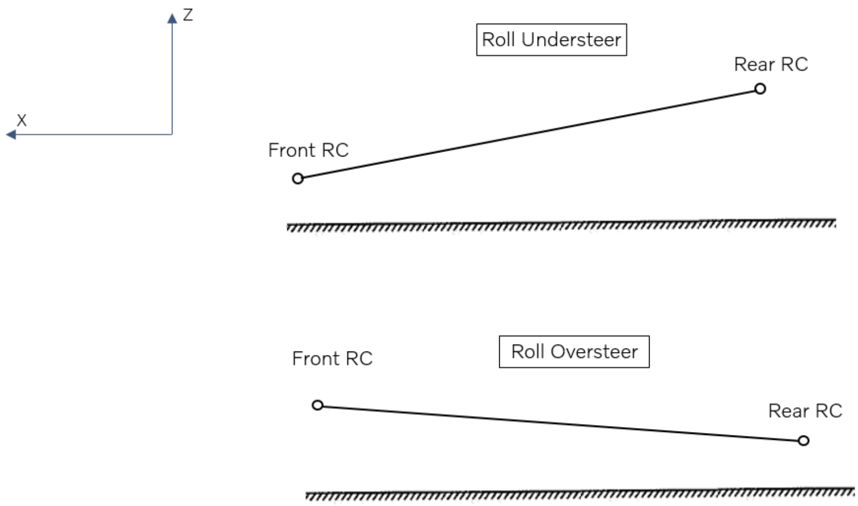

Figure 2.12: Roll understeer and Roll oversteer

Body roll during cornering depends on the distance between CoG and RC. Hence, a high roll center closer to CoG will help in reducing body roll as compared to a low roll center height.

- **Trackwidth variation:** Trackwidth is the distance between left and right centers of tire contact to ground, of the same axle [4]. A wider track width provides better driving behaviour and reduced body roll. Trackwidth variation is the variation in in trackwidth during vertical wheel travel. During jounce/rebound motions of the wheel, changes in camber and suspension kinematics might change the contact patch between tire and road. This results in variation of trackwidth since it is measured at the tire contact path.

  A higher value of trackwidth variation adversely affects the steering demands, rolling resistance and straight line stability of the vehicle.

- **Anti-geometry:** The anti- geometry is a form of suspension geometry both at the front and at the rear wheels which alters and controls the amount of wheel travel during acceleration and deceleration of the car. Suspension antis do not modify the steady-state load transfer at the tire contact patch but it changes the ratio of longitudinal and vertical load taken up by the control arms and suspension springs. If a suspension has 100% antis, then most of the load is taken up by the control arms and significantly less by the springs. Similarly, if a suspension has zero antis, then all longitudinal load transfer is taken by the springs and none by the control arms [6]. Anti-effects depends on the IC and the slope of Side view swing arm. Zero anti geometry is achieved by inclining the upper and lower control arms parallel to each other.

The anti-geometry can be further classified into anti-squat, anti-dive and anti-lift and is applicable for the front and rear axles as:

Front axle: Anti-lift and Anti-dive
Rear axle: Anti-lift and Anti-squat

- **Anti-lift:** Anti-lift geometry is used mainly on front wheel drive cars and four wheel drive cars. When an acceleration force is acting through the centre of gravity causing the nose of the car to raise and the rear to squat, the car tends to pivot about the COG. This causes the front suspension to expand and rear to compress aggressively. An anti-lift geometry reduces the wheel travel by producing an "opposite" force.

- **Anti-squat:** Anti-squat geometry features on the rear wheels. Anti-squat limits the amount of compression or vertical displacement of the rear wheels due to the acceleration of the car. Anti-squat geometry can be varied as a function of control arm geometry of the rear wheels.

- **Anti-dive:** Anti-dive geometry features on the front wheels of the car. It prevents the nose of the car from diving in during braking, when the car pivots due to the decelerating forces on the COG. In simpler words, it is the anti-squat feature on the front wheels.

2.2.4 Kinematic and Elasto-Kinematic analysis

Kinematics is the study of motion of rigid bodies, i.e., bodies which are non-deformable in nature and Elasto-kinematics is the study of motion of a mechanism by using flexible bodies.

So kinematic analysis in Suspension design implies that the ball joints (or bushings) have their designated degrees of freedom and the bushings are considered as rigid. In pure kinematic analysis, structural deformations of the control arms are not considered.

In Elasto-kinematic analysis, the bushings have more degrees of freedom and are considered as elastic bodies. Here, the structural deformations of the control arms are sometimes considered during analysis.

3
Methodology

3.1 Full Vehicle Simulations

This section explains the full vehicle simulations performed by selecting various driving maneuvers and for a combination of different vehicle speeds, varied suspension parameters and road conditions.

Certain vehicle suspension parameters/attributes can be better assessed using a specific maneuver and it is important to choose the right maneuver and road condition to assess the desired parameter. For example, the effect of anti-lift and anti-dive on vehicle handling can be better assessed in acceleration event and choosing uneven road for this test case can help assess ride comfort, since reduced antis can improve ride comfort on uneven road. Similarly, effect of roll center height variation on body roll can be better assessed in an evasive maneuver such as Double Lane Change (DLC).

Since there were certain parameters that will be changed, acting as input and certain attributes that are to be assessed for each maneuver, this would result in a greater number of simulations. Hence, a simulation planning spreadsheet (refer figure A.9), was prepared for a scientific and methodical approach.

The overall methodology can be better illustrated with the help of the flowchart in figure 3.1. The same parameter (for example, roll angle in DLC maneuver) is compared between passive and active system simulations to understand the trends in suspension performance variation.

Variation in suspension parameters is obtained by changing the kinematic targets and obtaining new hardpoints for these changed kinematic targets. The new suspension model generated will reflect this change in the full vehicle simulation. For example, a high roll center obtained through modifications to control arms hardpoints will be reflected in the full vehicle simulation with a reduction in the body roll angle.

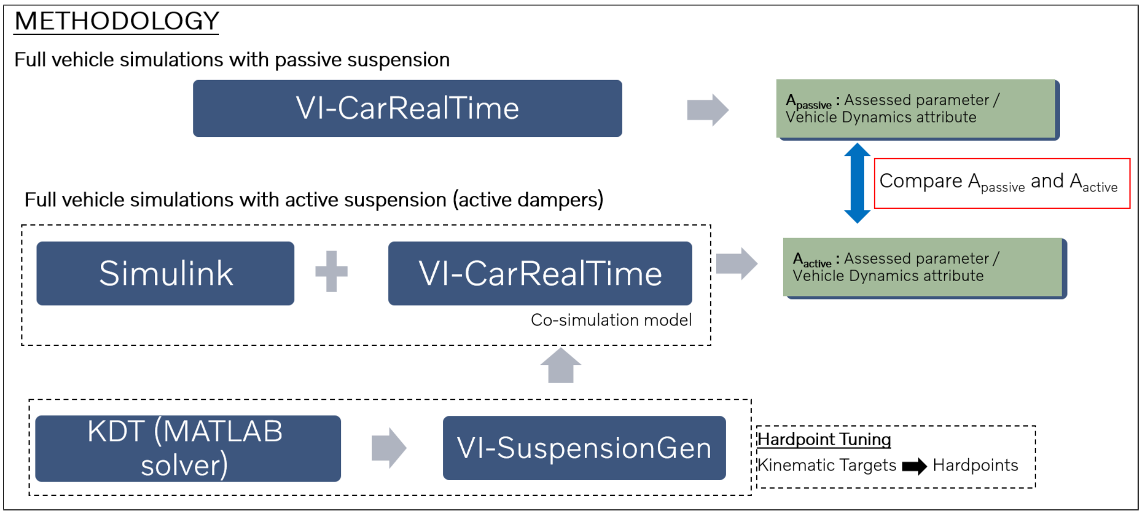

Figure 3.1: Methodology flowchart

3.1.1 Co-simulation model between VI-CarRealTime and Simulink

A simulation tool called VI-CarRealTime is used in the thesis work to evaluate the suspension performance in full vehicle simulations. The vehicle models compatible with VI-CarRealTime are imported from a multi-body simulation software MSC ADAMS. In these models, the suspension design parameters are in the form of look-up tables and thus speeding up the simulation time. To perform simulations with active suspension, the vehicle model is integrated with a controller in Simulink and a co-simulation model between VI-CarRealTime and Simulink is created (refer figure 3.2). Simulations involving passive suspension are performed only in VI-CarRealTime and not through the co-simulation model. The baseline vehicle models and co-simulation models are provided by VCC.

The suspension models can be further tweaked from the kinematic perspective to generate new look up tables for the suspension design parameters using an extension of the VI-CarRealTime software called VI- SuspensionGen. This allows for more iterations of suspension design while having the possibility to carry over other parameters of a full vehicle model like anti-roll bar, brake system, powertrain and so on.

Further, a vehicle data file is generated in the VI-CarRealTime and this file is used as an input to a simulink model where the passive damping curves are replaced by active damping lookup tables. The procedural steps to compare the performance of active and passive suspension is as follows:

- Import a Vehicle Model from MSC Adams to VI-CarRealTime.
- Setup a simulation event by creating a road and a driver model using the extensions of the simulation tool.
- Run the simulation to analyse the passive suspension performance.

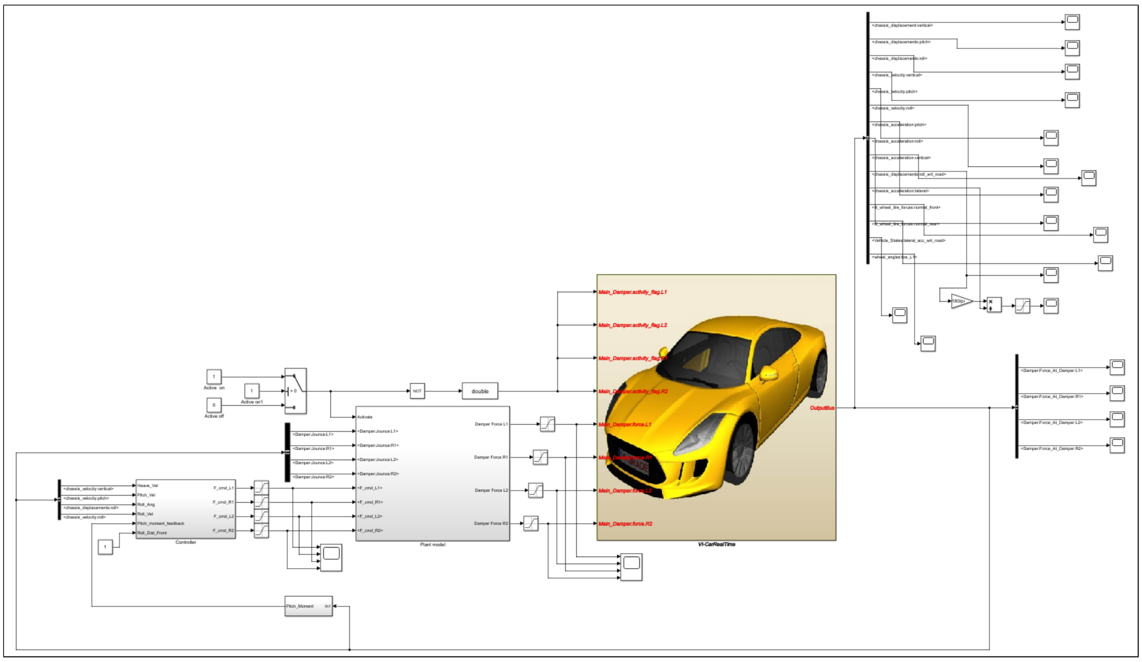

Figure 3.2: Co-simulation model between VI-CarRealTime and Simulink

- Generate a vehicle data file to be used for active suspension analysis.
- Using this vehicle data file as an input, an active suspension model is simulated in co-simulation model.
- Compare the results of active and passive suspension performance using an extension of VI-CarRealTime called VI-Animator.
- Tune the kinematics if required using VI-SuspensionGen and add the new lookup tables to the existing vehicle model. Run the simulations again.

3.1.2 Developing road models for full vehicle simulations

The simulation environment is mainly made up of a road profile and a driver model. The road profiles are built using an extension of the VI-CarRealTime software called VI-Road and an open source tool called OpenCRG. The following roads are developed for the different driving events used in this thesis work:

1. An uneven road profile of class D
2. Road with stochastically placed bumps
3. Flat road

3.1.2.1 Uneven Road Profile

The uneven road profile of class D used for this thesis work is developed according to the standards defied by ISO 8608 [9]. This standard classifies the road into eight different classes from A to H based on the harshness levels, with A being the smoothest and H being the harshest surface (refer figure 3.5).

The definition of unevenness is given in the frequency domain by the vertical displacement power spectral density (PSD). This data about the PSD for different road class can be a direct input in the VI-Road extension of the VI-CarRealTime software. The following data obtained from figure 3.5 is used to model a class D road.

- Omega 1 = 0.5
- Omega 2 = 18
- PSD(Omega 1) = 0.009
- PSD(Omega 2) = 0.0000009
- Phase angle between left and right unevenness = 180

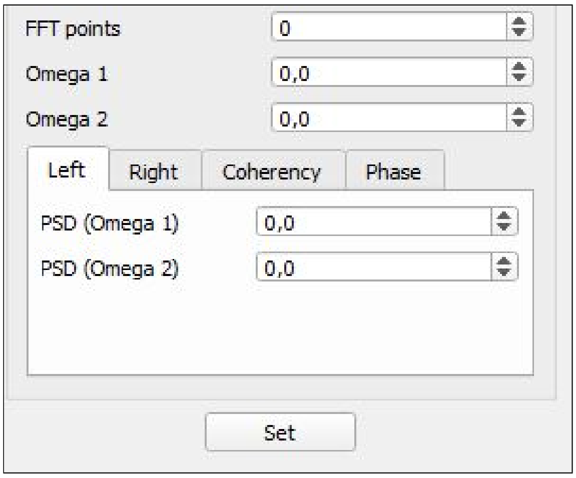

Figure 3.3: PSD Parameters input block in VI-Road

Road with uneven irregularities are used to analyse the effect of antis (anti-lift, anti-dive and anti-squat) on the suspension performance and ride comfort.
The resulting road profile can be seen in figure 3.4 where the road irregularities is of the range 0.04m to -0.02m in magnitude.

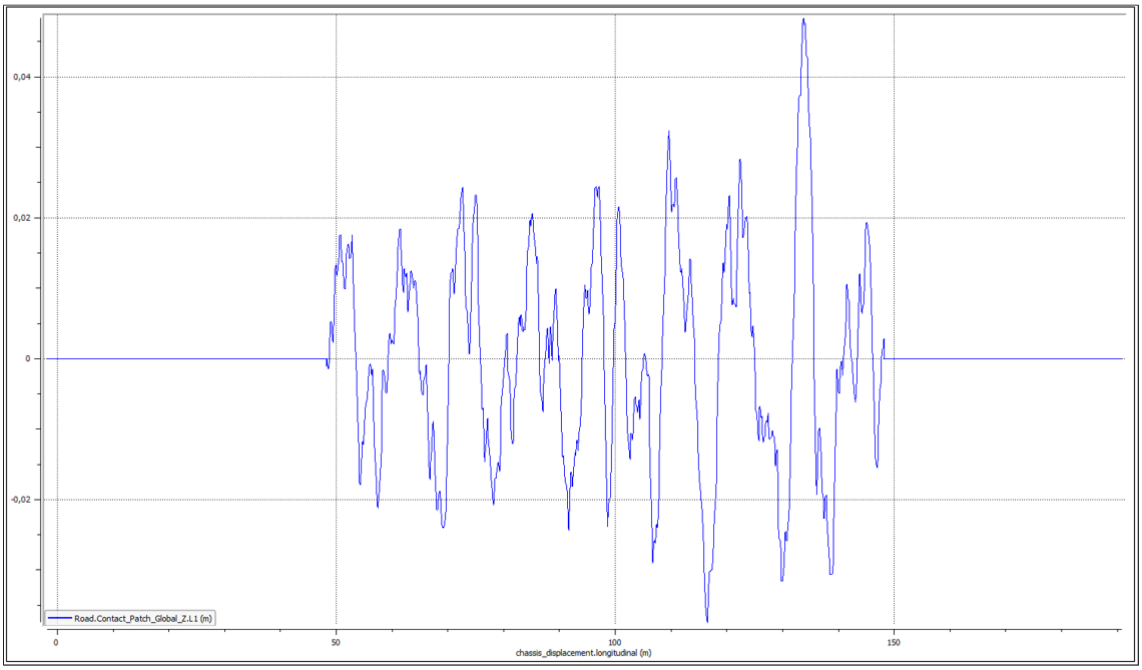

Figure 3.4: Uneven road profile of Class D according to ISO 8608

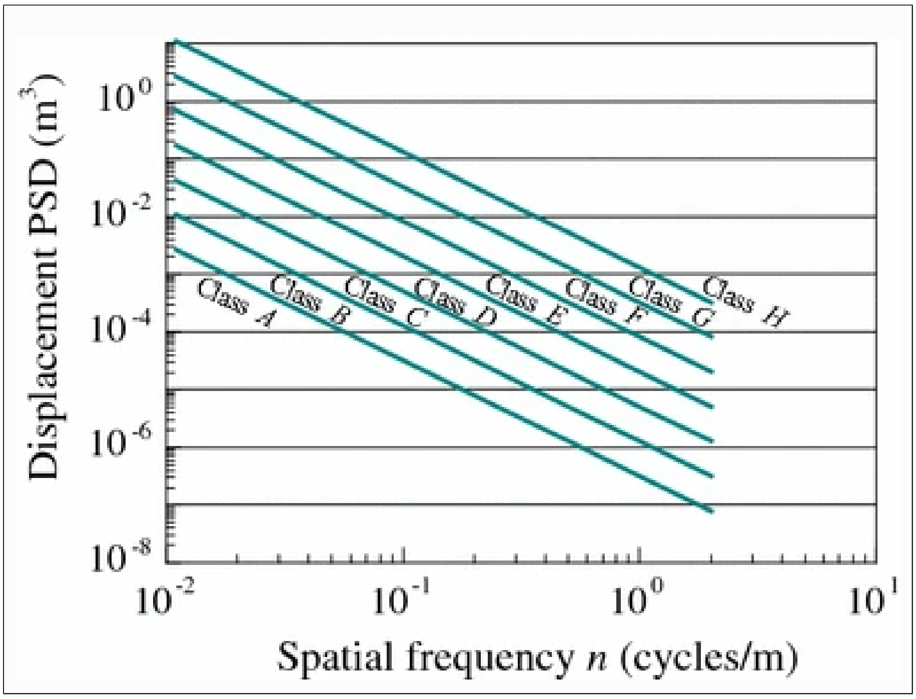

Figure 3.5: ISO 8608: Uneven road profile of Class D [16]

3.1.2.2 Road with stochastic bumps

As compared to the uneven road profile, this road profile consists of isolated bumps for left and right to induce roll motion and vertical movement in individual wheels. The road profile also consists of bumps to induce parallel wheel travel. This road profile is developed using the openCRG software package.

The road profile is visualized as a matrix in X and Z direction. The vertical elevation of the road in z direction is given as an input based for every pre-determined step size in x-direction. Further, the width of the road is specified in another matrix which allows to model bumps to individual left or right wheel. The road profile is as shown in the figure 3.6. This road profile helps to assess ride comfort and steering demands needed to keep the vehicle straight or to keep the vehicle in the desired path.

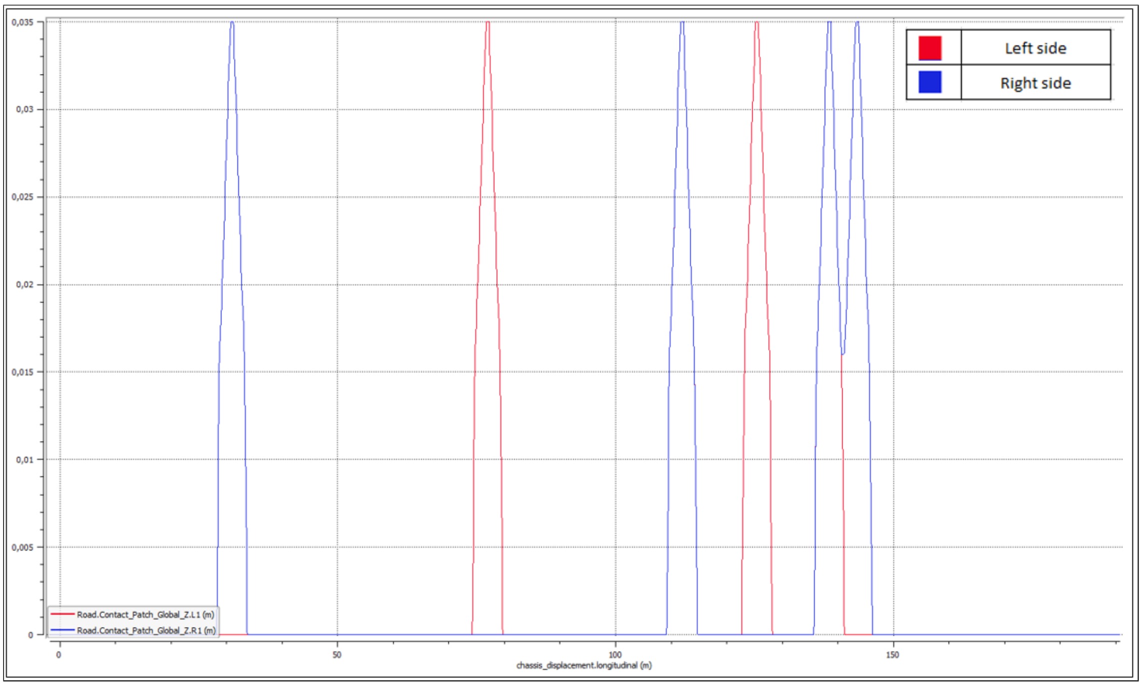

Figure 3.6: Road profile with stochastic bumps

3.1.2.3 Flat road

While simulating driving events like Double Lane Change, Constant Radius Cornering, Acceleration and Braking, a flat road without any irregularities are used. This is done in order to simplify the number of variables in the simulation and to have a detailed analysis of the suspension performance. A straight road of 200 meters length and 4 meters width is developed in VI-Road.

3.1.3 Developing suspension models after hardpoint changes

As mentioned in the previous sections, suspension models are generated as look-up tables through VI-SuspensionGen. A visualization of the SLA suspension is shown in the figure 3.7. Initially, the first set of hardpoints (baseline) are used to generate a reference model. Further, the kinematic splines are replaced when hardpoints are modified. This is an iterative process and hardpoint tuning process is further explained in the section Hardpoint Tuning. Although VI-SuspensionGen allows Kinematic and Elasto-kinematic analysis to generate suspension models, only Kinematic analysis is performed to generate the suspension models. Elasto-kinematic analysis is out of the scope of this thesis work.

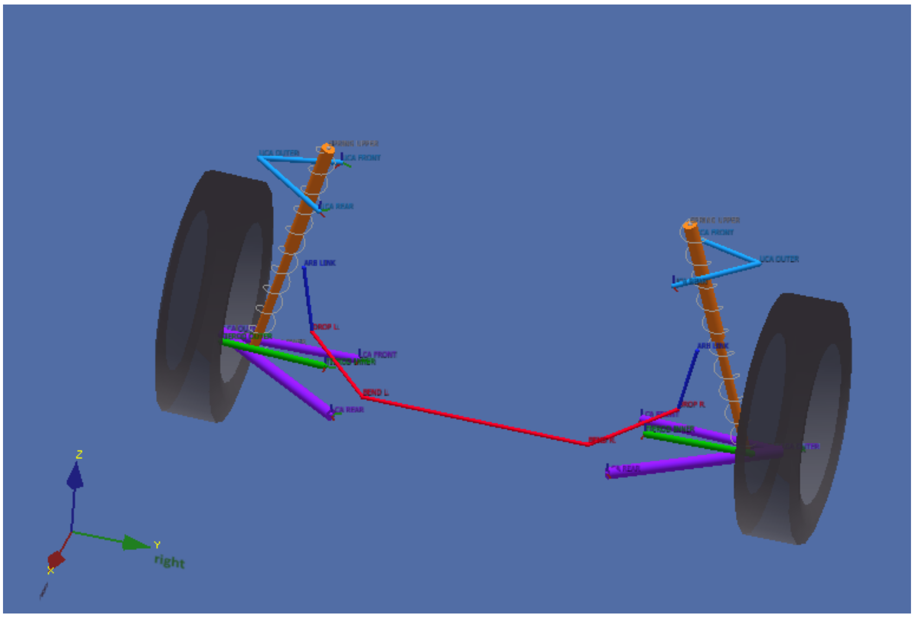

Figure 3.7: Visualization of SLA suspension in VI-SuspensionGen software

In suspension design, a change in hardpoints affects the kinematic behaviour of suspension and vice-versa. A "Forward method" is to specify the hardpoints and calculating the kinematic targets for the specified hardpoints. This results in several iterations to find the correct compromise between desired kinematic targets and feasible packaging solutions. However, a "Reverse method" is also available wherein the hardpoints are calculated with desired kinematic targets as input.

As per the defined thesis scope, packaging limitations are not considered. This allows moving the hardpoints freely in space and thus, "reverse method" was chosen to generate the new suspension models. These new models will be a result of change in hardpoint geometry due to change in kinematic targets. This was done using a MATLAB solver called "Kinematic Design Template (KDT)"[11], which generates new hardpoints based on kinematic targets and vice-versa, with the possibility of post analysis with various plots as a function of wheel center position and steer angle.

The new hardpoints generated were fed into the VI-SuspensionGen reference model to create updated suspension models, that can be used in full vehicle simulations. Compliance is kept unchanged for the new suspension models generated since compliance analysis is outside the scope of this thesis work.

Table 3.1 shows the different parameters that can be changed for SLA suspension.

The nomenclature of these parameters are explained briefly as:
Location - X,Y,Z Hardpoint co-ordinates in the global frame.
k - Linear translational stiffness in the bushing frame.
kt - Linear rotational stiffness in the bushing frame.
Orientation - Euler angles of bushing frame (rotation sequence "313 body").

Table 3.1: List of hardpoints and other properties that can be varied in VI-SuspensionGen for SLA suspension

Hardpoint name	**Location X,Y,Z (mm)**	**k (N/mm)**	**kt (Nmm/deg)**	**Orientation (deg)**
UCA front	✓	✓	✓	✓
UCA rear	✓	✓	✓	✓
LCA front	✓	✓	✓	✓
LCA rear	✓	✓	✓	✓
Tierod Inner	✓	✓	✓	✓
Tierod Outer	✓			
Wheel center	✓			
UCA outer	✓			
LCA outer	✓			
Spring Upper Mount	✓			
Spring Lower Mount	✓			
Damper Upper Mount	✓			
Damper Lower Mount	✓			
ARB link	✓			
Left Bend AR	✓			
Left Drop AR	✓			
Right Bend AR	✓			
Right Drop AR	✓			

3.1.3.1 Hardpoint Tuning

Hardpoint modifications are done based on desired kinematic targets. The list of kinematic targets with the percentage variations are given in Table 3.2.

Table 3.2: List of kinematic parameters and the corresponding variation

Suspension kinematic parameter	**Unit**	**Variation in the value**
Bumpsteer	deg/m	50% reduction
Roll Center Height	mm	±30 mm
Anti geometry (Anti-lift, Anti.dive and Anti-squat	%mm/mm	Values set to 0

- **Bumpsteer change:**
Bumpsteer is lowered by 50% compared to the baseline value. This is done by lowering the Tierod outer joint in front and toelink outer joint in the rear in Z-direction, with all other Hardpoints kept the same.

- **Zero-Anti geometry:**
The values of following Anti-geometries are changed to zero.
Front axle: Anti-lift and Anti-dive
Rear axle: Anti-lift and Anti-squat
This is done by changing the Z co-ordinates of Upper and lower control arms (UCA & LCA) to achieve a parallel setting between UCA & LCA. All other Hardpoints are kept the same.

- **Roll Center Height Variation:**
Variation in Roll center height is ±30mm.
High RCH: Raise UCA and LCA Z co-ordinates. Lower the Tierod outer joint in front and toelink in rear.
Low RCH: Lower UCA and LCA Z co-ordinates.

3.2 Active suspension controller architecture

The actuator models and the controller model for active suspension used in this thesis work are provided by VCC. These are further integrated with the simulation blocks from VI-CarRealTime to form the complete vehicle co-simulation model.

The controller as shown in the below figure takes heave velocity, roll velocity, roll angle and pitch angle as the inputs.

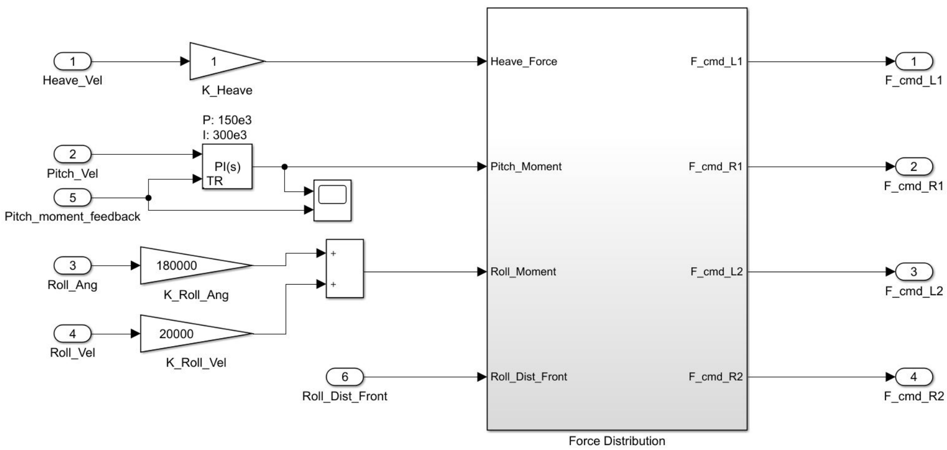

Figure 3.8: Controller Block in Simulink

The heave velocity input is further used to determine heave force through the corresponding damper movement. The pitch angle and the pitch moment are used to control the pitch of the vehicle and similarly the roll angle and the roll moment control the body roll. The sum or difference of these forces based on the directions at each individual wheel gives a force request command. This force request at each individual wheel is an output from the controller and an input to the actuator model provided by the development partner.

3.3 Driving Maneuvers

The driving maneuvers are chosen based on the parameters that are to be assessed. The process of creating driver model and integrating it with the road model into a simulation environment is explained in the first subsection. The chosen driving maneuvers, parameters varied and attributes assessed along with the simulation setup is explained in the subsequent sections.

3.3.1 Driving event setup in VI-Event Builder

The simulation event consists of a driver model and road profile. The driver model is setup using the extension VI-Eventbuilder of VI- CarRealTime. The driver model is constructed using pre-defined blocks such as steering input, transmission, path to be followed and so on, which are then assembled together into a model. Driver model for the Braking event is shown in the figure 3.9.

The driver models and the corresponding event setup depends on the nature of the driving maneuver selected and varies for each maneuver. Driver model setup for the chosen maneuvers in this thesis work is shown from figures A.5 to A.8 in Appendix.

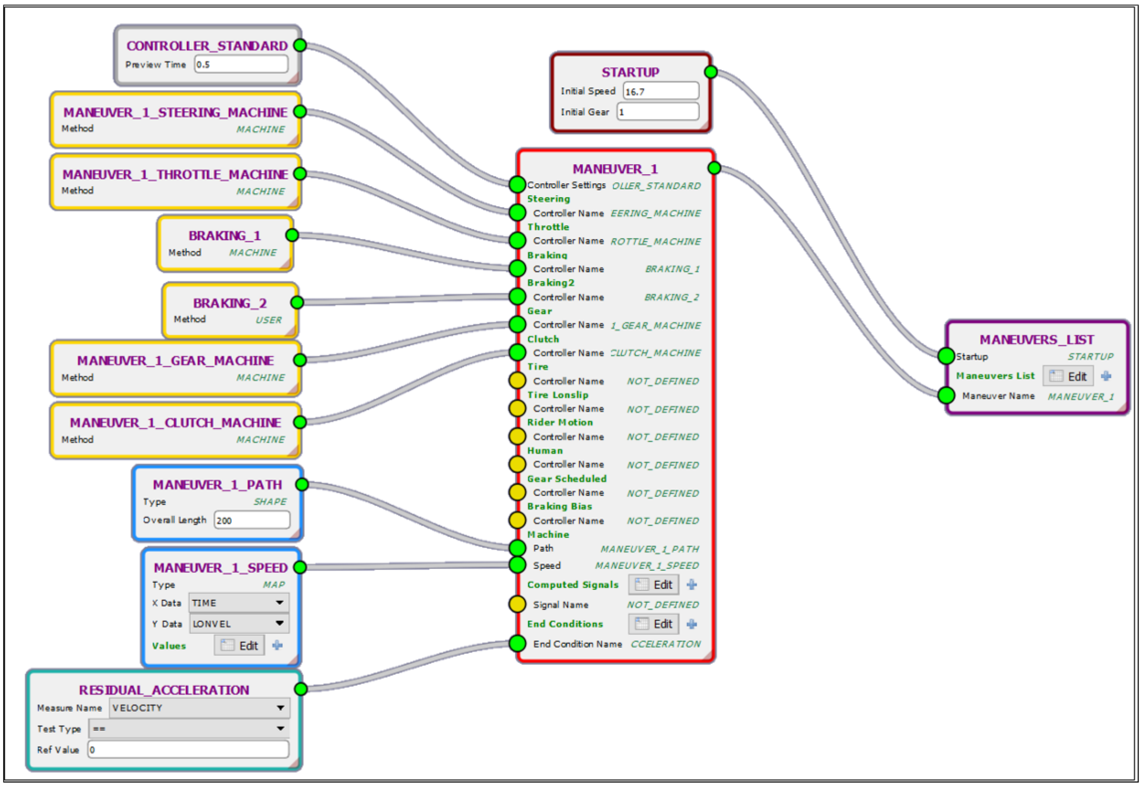

Figure 3.9: Defining the event for Braking in VI-Event Builder

3.3.2 Driving straight

This maneuver is performed to analyse the steering effects and vehicle pitch on an uneven road and a road with stochastically placed bumps. The simulated event requests the car to follow a straight line until the end of the road at a constant speed. Steering efforts required to keep the car in the desired straight line path along with other parameters of interest are logged. This data is useful to study the variation in the steering effects when moving over a bump or unevenness of the road. The parameters that are studied here are the roll angle of the vehicle, pitch and steering wheel angle.

Simulation setup:

- Vehicle speed: 40kmph
- Type of road: Uneven road of class D/road with stochastic bumps
- Length of the road: 200m
- Simulation time: 20s
- Starting gear: 1

3.3.3 Acceleration

This maneuver is performed to analyse the effect of suspension anti-squat when using an active suspension. In the acceleration event, nose of the car is lifted causing the front suspension to expand and the rear of car dips causing the rear suspension to compress. Suspension antis are added to the design to reduce the wheel travel in these cases, which produces an opposite force resisting compression. Higher the value of suspension antis, higher is the longitudinal and vertical load on the control arms. Removing antis from the suspension geometry delegates all loads on the springs and in the presence of active dampers, better control can be obtained. Also, removing suspension antis allows the suspension to move freely and therefore improves ride comfort when the vehicle is driven on uneven roads.

The acceleration event is performed in a 250m flat road simulation environment. The driver model is designed to accelerate from stand still to 100 kmph with gear shifts. The baseline simulations are performed with the standard passive and active suspensions and these are compared with the performance of the suspension without the anti-effects.

Simulation setup:

- Vehicle starting speed: 0kmph
- Vehicle final speed: 100kmph
- Type of road: Flat
- Length of the road: 200m
- Simulation time: Till the vehicle reaches 100kmph
- Starting gear: 1

3.3.4 Braking

This maneuver is performed to analyse the effect of anti dive and anti lift geometry when using an active suspension. In the braking event, nose of the car dips causing the front suspension to compress and the rear of car is lifted causing the rear suspension to expand. Also when the car is decelerating, there will be longitudinal load transfer from rear to front which is to be taken up by the control arms and springs. The reasoning for simulating this maneuver is similar to that of acceleration.

The braking event is performed in a 200m flat road simulation environment. The driver model is modelled to brake from 60 kmph to zero with maximum brake demand. The assessed parameters in this maneuver are the same as Acceleration event.

Simulation setup:

- Vehicle Initial speed: 60kmph
- Type of road: Flat
- Length of the road: 200m
- Simulation time: Till the vehicle approaches 0kmph
- Starting gear: 2

3.3.5 Double Lane Change

The double lane change (DLC) is a dynamic maneuver which is mainly performed to assess the handling of the vehicle. It is important to assess the dynamic handling of the vehicle which is also imperative to the safety of the driver and passengers. The parameter of interest here is the body roll angle and the variation in the roll angle observed when ARB is removed from the suspension.

The vehicle path of a standard double lane change is as shown in figure below. The path to be followed by the vehicle is developed according to the ISO 3888 standard [10]. The simulation of the car is performed in varying entry speeds to study the effect of speed on the performance of the suspension and further tune the parameters such as roll center height and actuator force, until roll reduction is achieved.

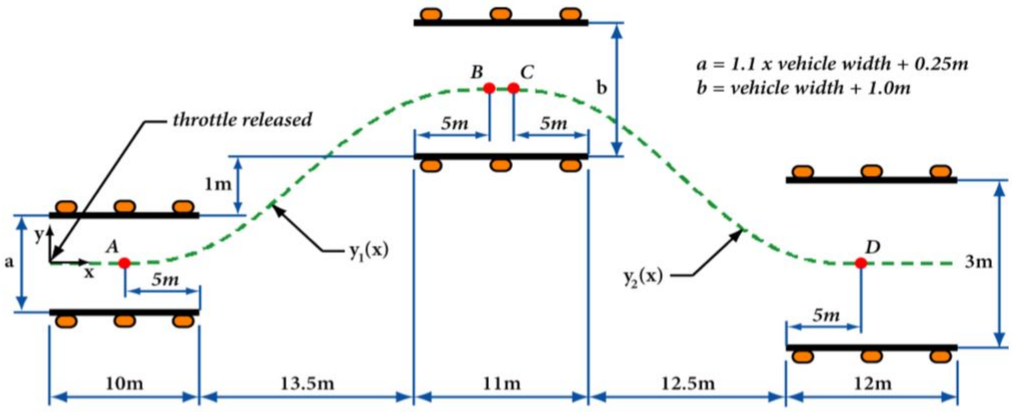

Figure 3.10: ISO 3888: Double Lane Change maneuver test track [15]

Simulation setup:

- Vehicle speed: 60 and 80kmph
- Type of road: Flat
- Simulation time: 20 seconds
- Starting gear: 2nd

3.3.6 Constant Radius Cornering

When analysing the roll behaviour of the car using active suspension it is considerate to analyse the transient behavior of the chassis roll angle. In Constant Radius Cornering maneuver, the car initially accelerates until a pre-determined speed is reached and then initiates the second stage of the maneuver where the driver model requests the car to follow the circumference of a 50m radius circle while maintaining the speed.

This scenario is mainly used in this thesis work to assess the lateral dynamics of the vehicle and to support the findings of double lane change maneuver. The passive and active baseline suspensions are compared with the suspension without ARB. The assessed parameter is the body roll angle of the vehicle.

Simulation setup:

- Vehicle speed: 60 kmph
- Type of road: Flat
- Path radius: 50m
- Starting gear: 2nd
- Simulation time: 120 seconds

4

Results

The results obtained in the full vehicle simulations will be summarised and presented in this chapter. Results will be categorised for each driving maneuver and as mentioned in the Introduction chapter (refer figure 1.1), with a comparison of a certain vehicle parameter/attribute between the established passive suspension baseline performance, active suspension baseline and active suspension with variation in a certain kinematic parameter. This comparison will give insights regarding the benefits of tuned suspension kinematics to match the active dampers. The simulations performed with a standard active suspension system will serve as a reference for model validation.

4.1 Driving straight

4.1.1 Driving straight on an uneven road profile - Class D

4.1.1.1 Baseline simulation results

Figures 4.1, 4.2 and 4.3 shows the results from simulations using passive baseline and active baseline suspension systems. To re-iterate, an active baseline suspension implies that passive damping forces are replaced by active damping forces with no change in the suspension kinematic targets.

From figures 4.1 and 4.2, it is evident that active suspension requires less steering demands and has lesser vehicle pitch compared to its passive counterpart.

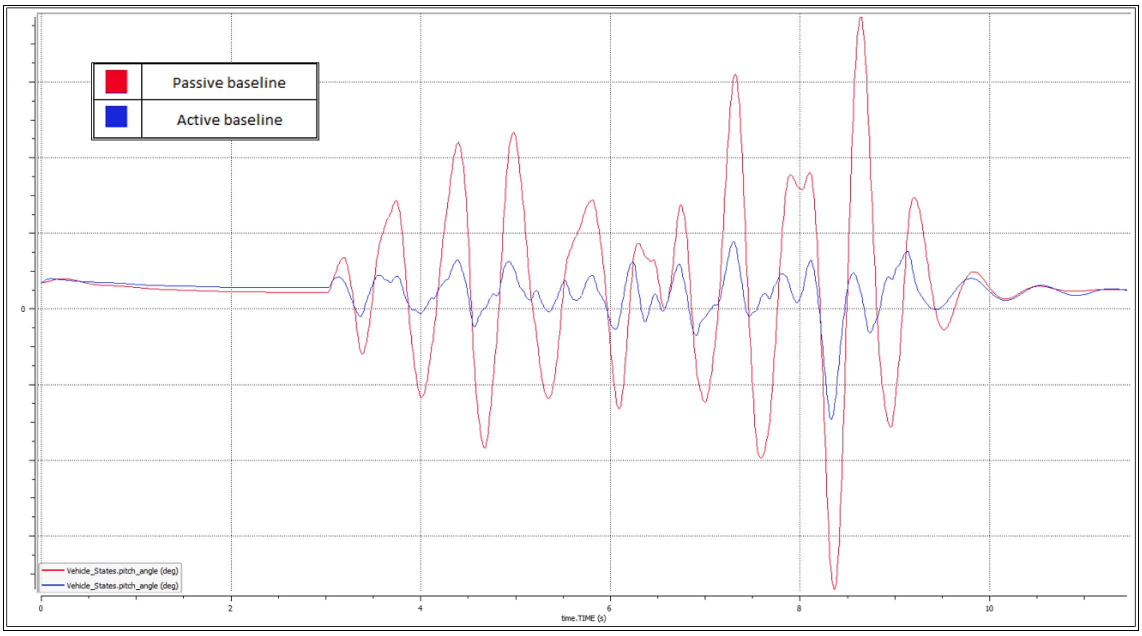

Figure 4.1: Vehicle pitch angle (deg) vs time (s) - Driving straight on an uneven road profile of Class D

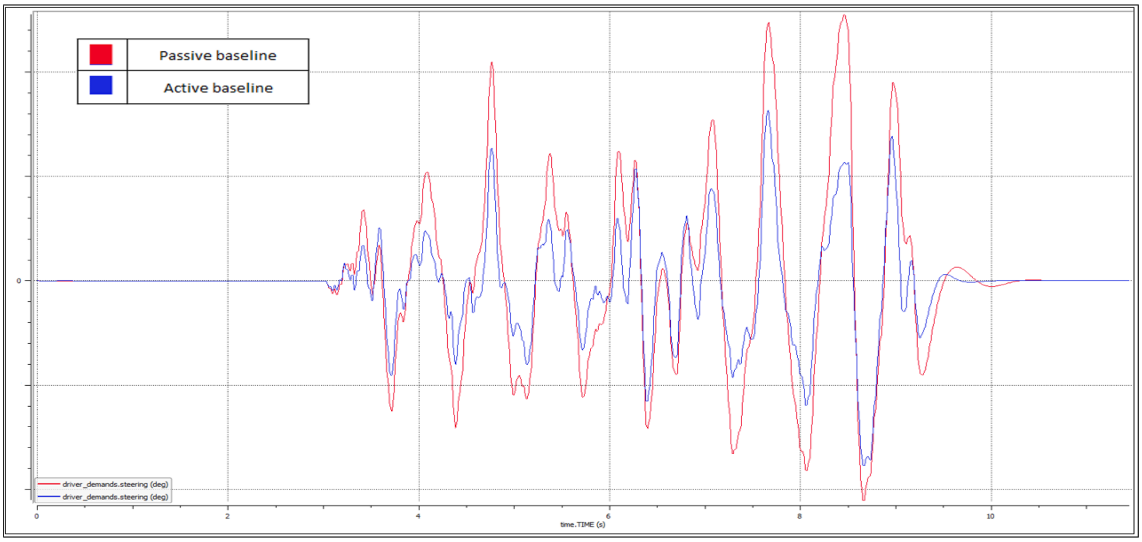

Figure 4.2: Steering wheel angle (deg) vs time (s) - Driving straight on an uneven road profile of Class D

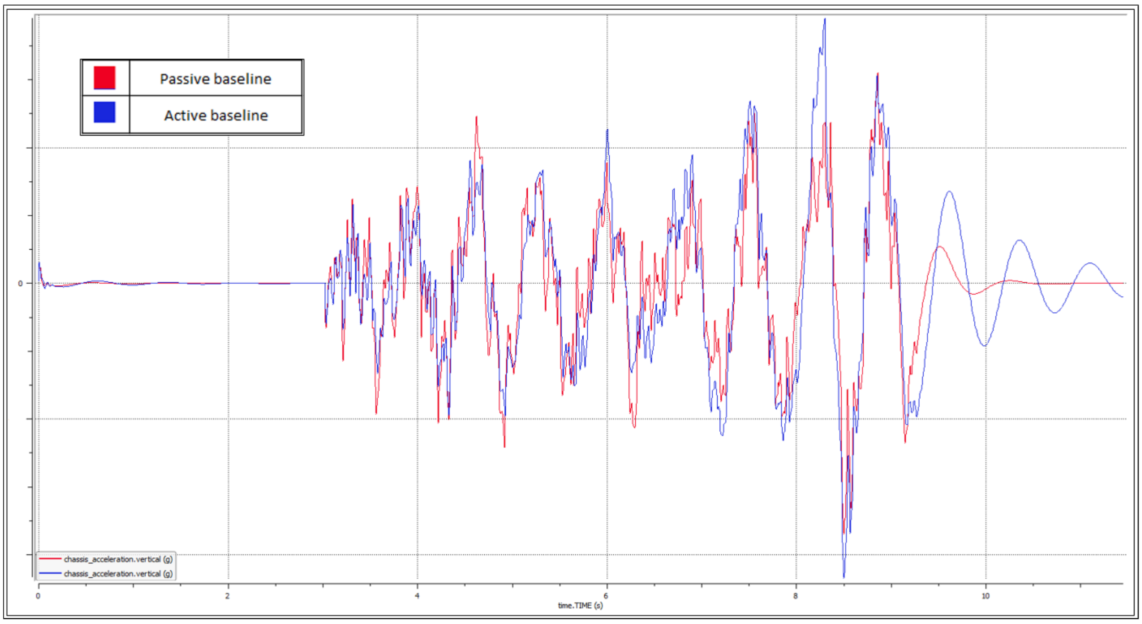

Figure 4.3: Vertical acceleration (g) vs time (s) - Driving straight on an uneven road profile of Class D

4.1.1.2 Simulation results with varied kinematic targets

Figures 4.4 to 4.8 shows the simulation results with varied suspension kinematic targets.

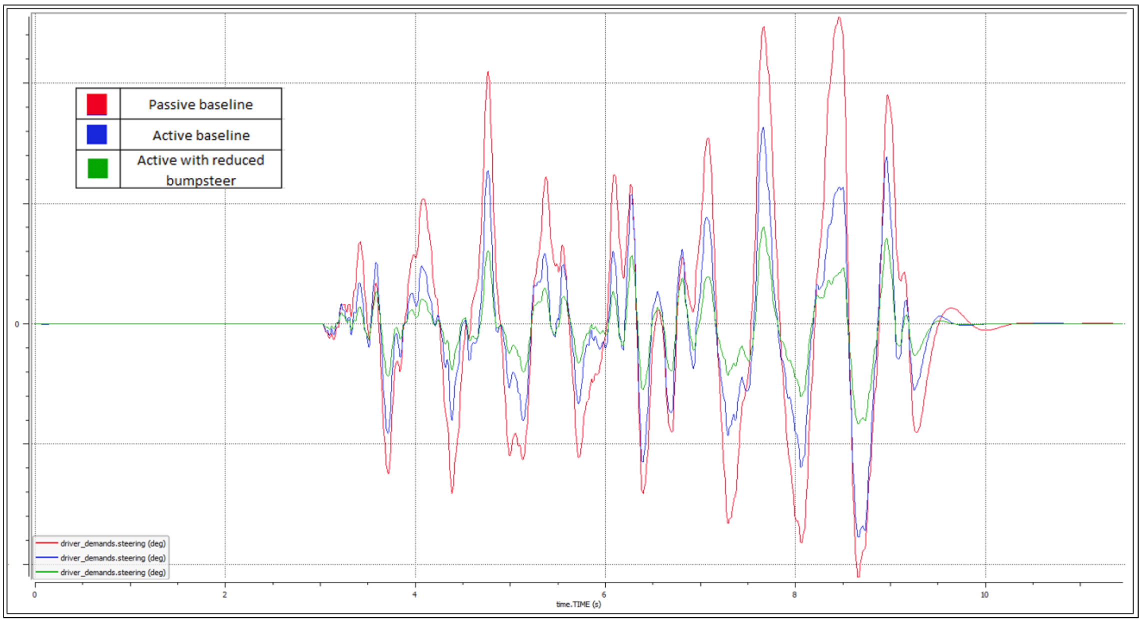

Figure 4.4: Steering wheel angle (deg) vs time (s) with bumpsteer reduction - Driving straight on an uneven road profile of Class D

It is evident from figure 4.4 that steering demands decrease significantly with a decrease in bumpsteer. This can be attributed to the fact that the vehicle becomes less sensitive as the toe-gain value is reduced by 50%.

It is also interesting to note that the steering demands reduces further when more energy is added to the vehicle suspension as seen from figure 4.5 when there is a 30% increase in the force limit that the actuator can inject into the suspension system.

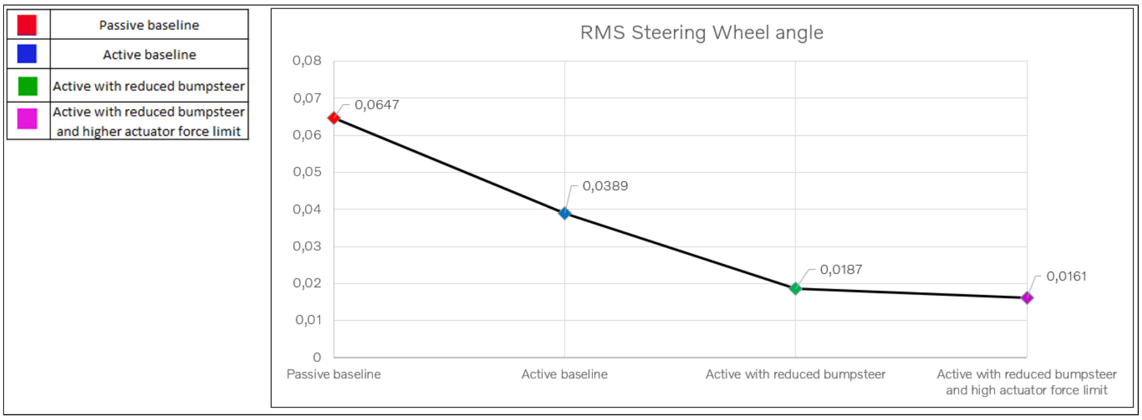

Figure 4.5: RMS values of steering wheel angle with bumpsteer reduction - Driving straight on an uneven road profile - Class D

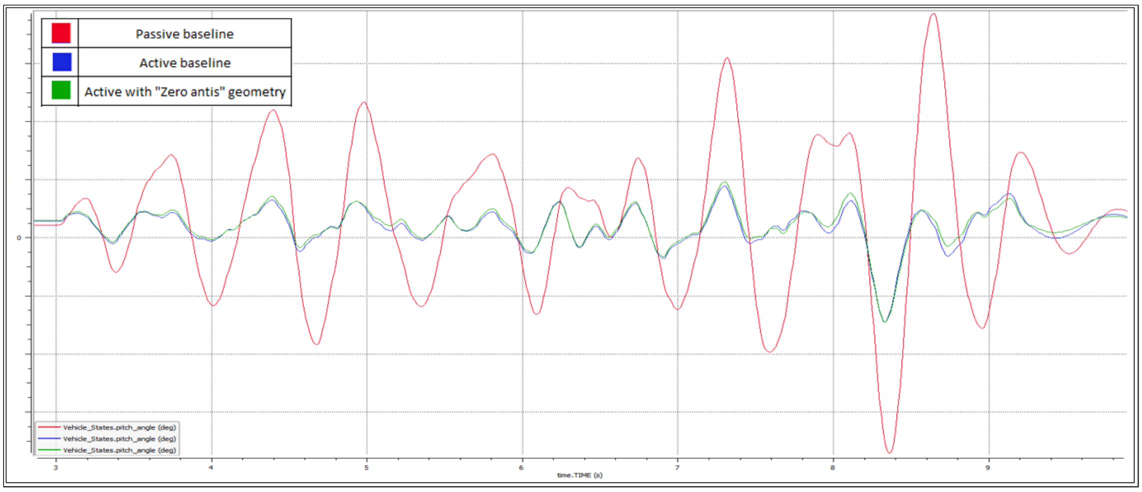

Figure 4.6: Vehicle pitch angle (deg) vs time (s) with "zero anti" geometry - Driving straight on an uneven road profile of Class D

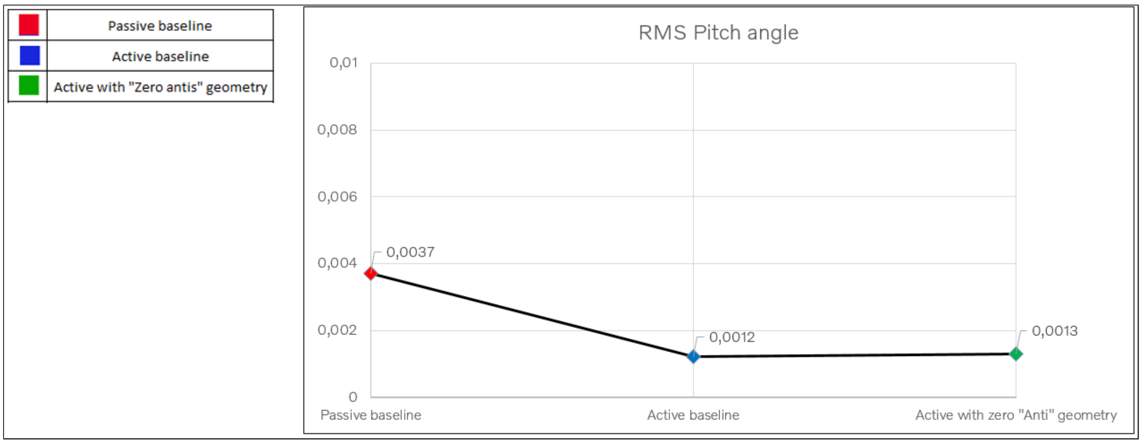

Figure 4.7: RMS values of vehicle pitch with "zero anti" geometry - Driving straight on an uneven road profile of Class D

Figures 4.6 and 4.7 show the suspension behaviour when the anti effects(Anti-squat, anti-lift and anti-dive) are removed from the suspension. No significant difference is seen in the values of vehicle pitch angles even when anti-effects are removed. However, removal of antis from the suspension does improve the vertical acceleration by a small margin as seen from figure 4.7. This can be attributed to the fact that zero anti geometry enables better wheel vertical movement.

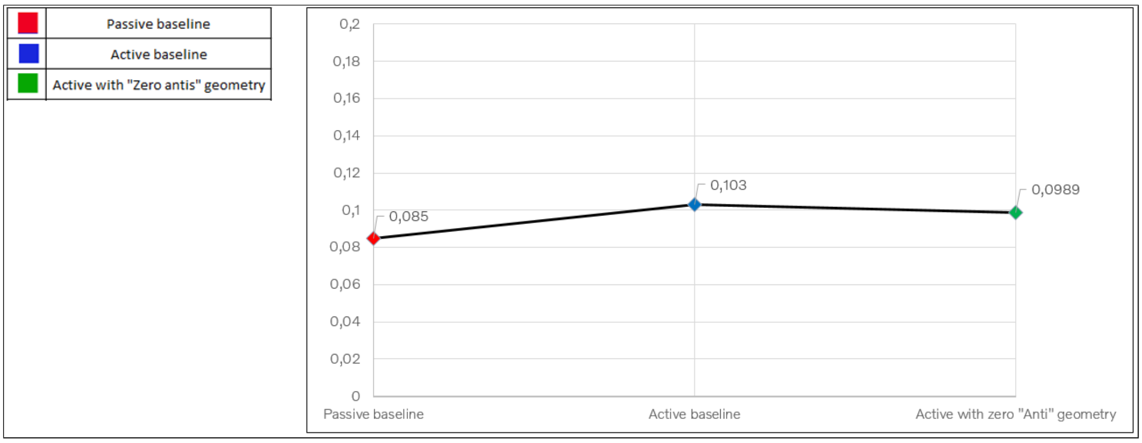

Figure 4.8: RMS values of vertical acceleration with "zero anti" geometry - Driving straight on an uneven road profile of Class D

4.1.2 Driving straight on a road with stochastic bumps

A similar trend in the suspension performance as explained in the previous section can be observed in the figure 4.9 wherein the vehicle pitch angle shows no significant variation with the removal of "anti" effects from the suspension. Another observation evident from the plot is improved vehicle pitch after reaching the peak values. These peak values result from the variation in road profile. Also, it can be observed from figures 4.11 and 4.12 that the steering wheel angles decrease with lesser values of bumpsteer.

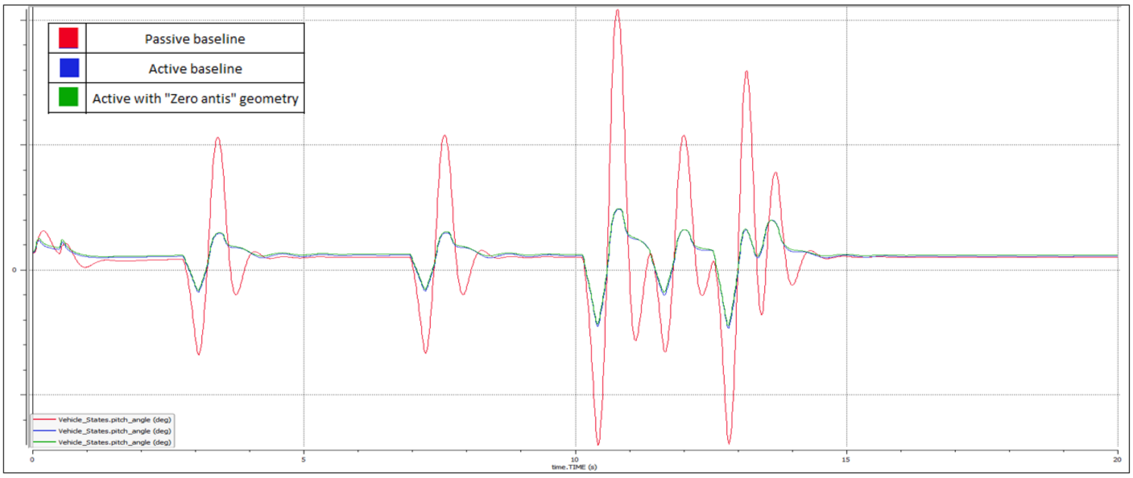

Figure 4.9: Vehicle pitch angle (deg) vs time (s) with "zero anti" geometry - Driving straight on a road with stochastic bumps

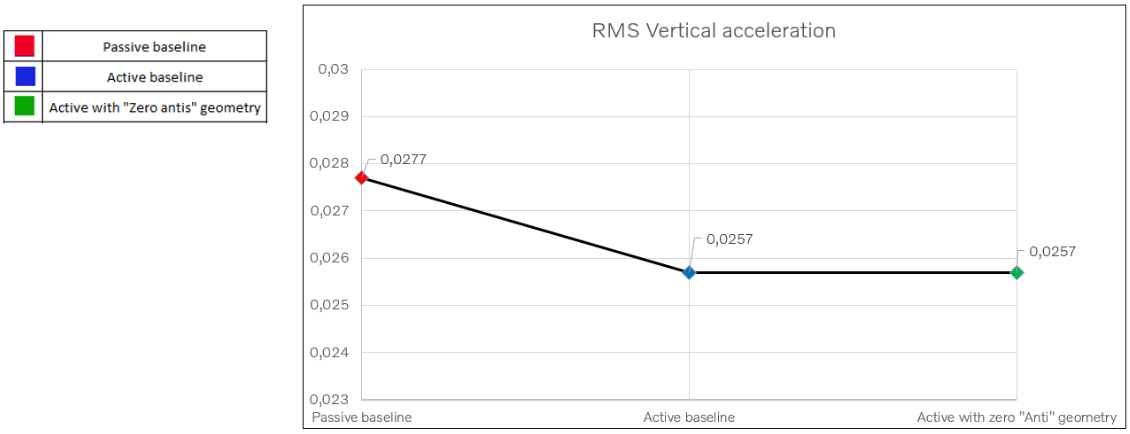

Figure 4.10: RMS values of vehicle pitch with "zero anti" geometry - Driving straight on a road with stochastic bumps

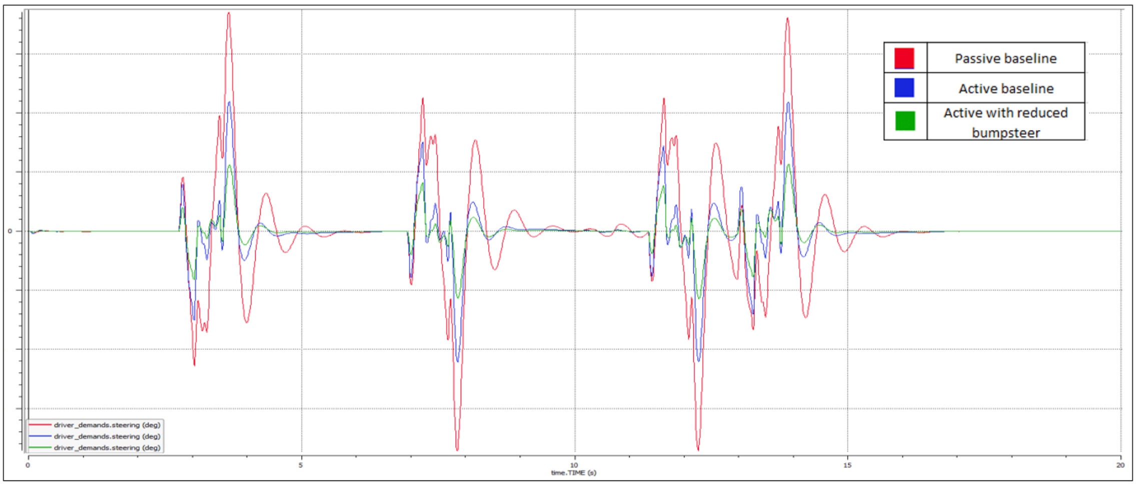

Figure 4.11: Steering wheel angle (deg) vs time (s) with bumpsteer reduction - Driving straight on a road with stochastic bumps

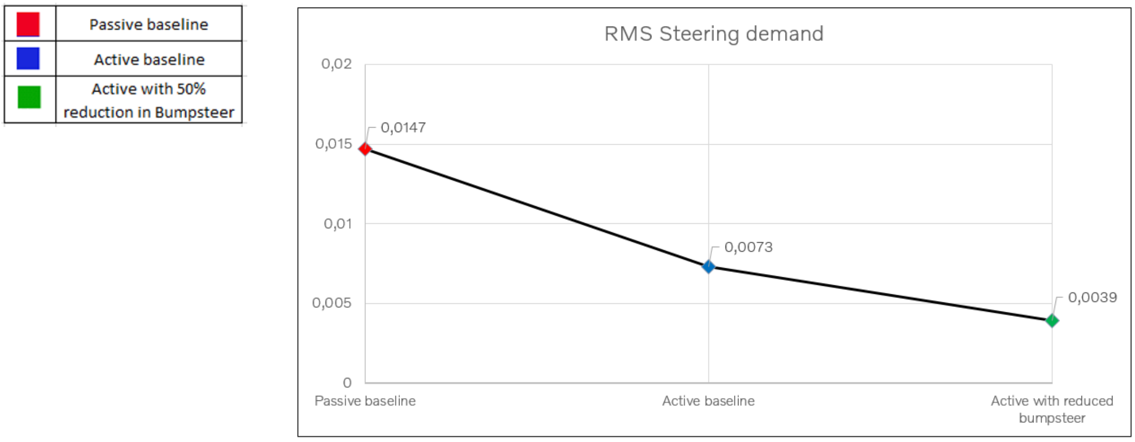

Figure 4.12: RMS values of steering wheel angles with bumpsteer reduction - Driving straight on a road with stochastic bumps

It is also interesting to note from figure 4.13 that body roll angle is lesser for the case without anti-roll bar. This is also crucial for an improved ride comfort when driving on a road with bumps, even more so in case of a "one-sided bump" where the forces due to the bump on one side of an axle will not be transmitted to the other side of the same axle, in the absence of ARB.

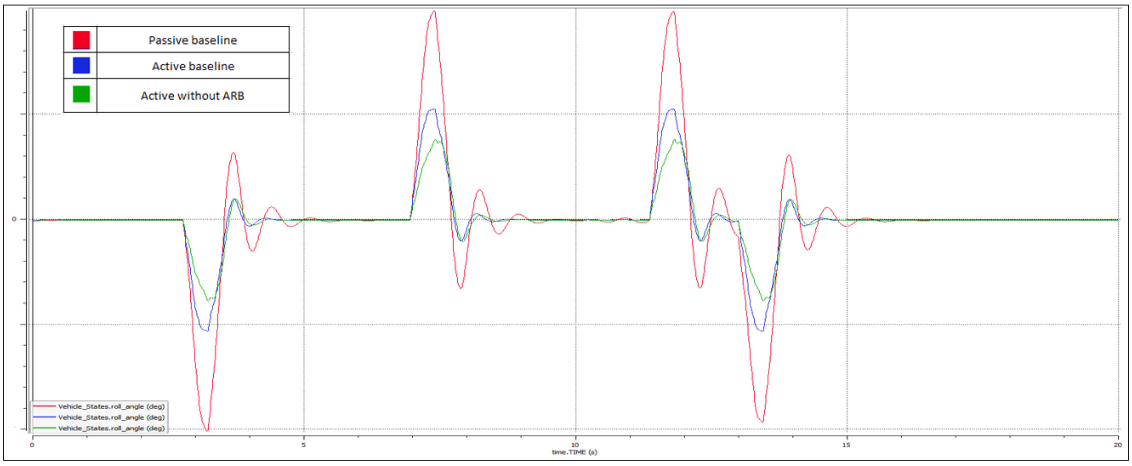

Figure 4.13: Body roll angle (deg) vs time (s) - Driving straight on a road with stochastic bumps

4.2 Acceleration

The acceleration maneuver is simulated to study the effects of variation in anti-squat and anti-lift geometry in the presence of active dampers. The values of all anti effects (anti-lift and anti-dive for front axle; anti-lift and anti-squat on rear axle) are set to 0 and active suspension model with zero anti geometry is used against the baseline suspension for comparison. The clear effects of active dampers can be seen in the figures 4.14 and 4.15 which is depicted by the reduction in peak values of pitch angle when an active suspension is used.

The intention behind removing the "anti" effects is to give more control to the active dampers for better utilization and also to simplify the suspension geometry giving more freedom to the suspension system architect.

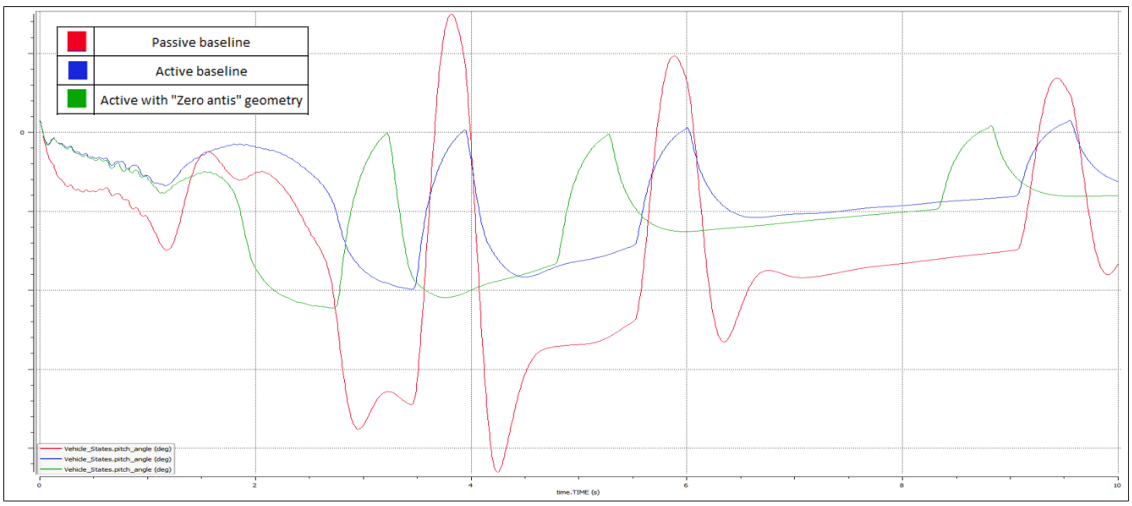

Figure 4.14: Vehicle pitch angle (deg) vs time (s) - Acceleration event

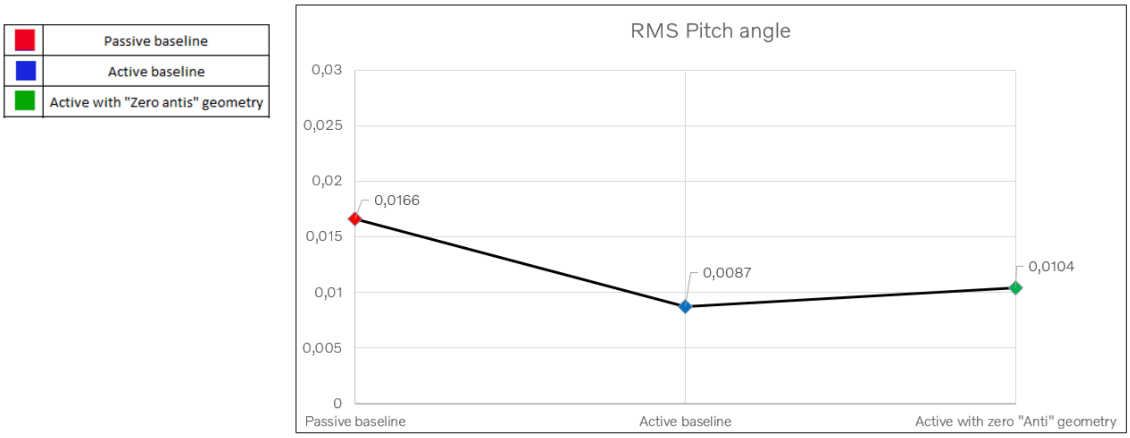

Figure 4.15: RMS values of Vehicle pitch angle - Acceleration event

4.3 Braking

As explained in the methodology section, the braking maneuver is used to understand the behaviour of the active suspension when the anti effects is set to 0. The clear effects of active dampers can be seen in the figure 4.16 and 4.17 which is depicted by the reduction in peak values of pitch angle when an active suspension is used. However, the pitch angle is increased by a small margin when "antis" are removed as seen from the RMS values and peak values.

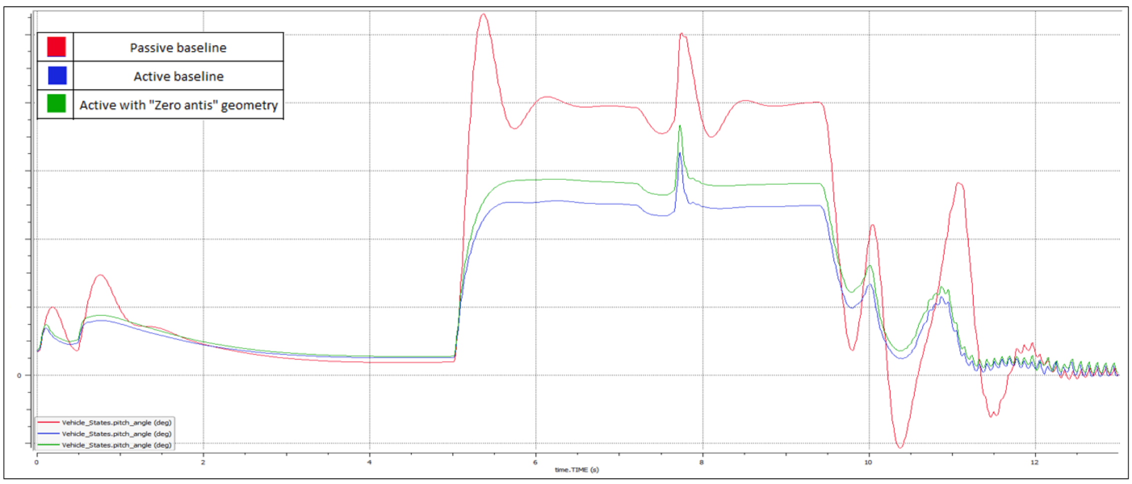

Figure 4.16: Vehicle pitch angle (deg) vs time (s) - Braking event

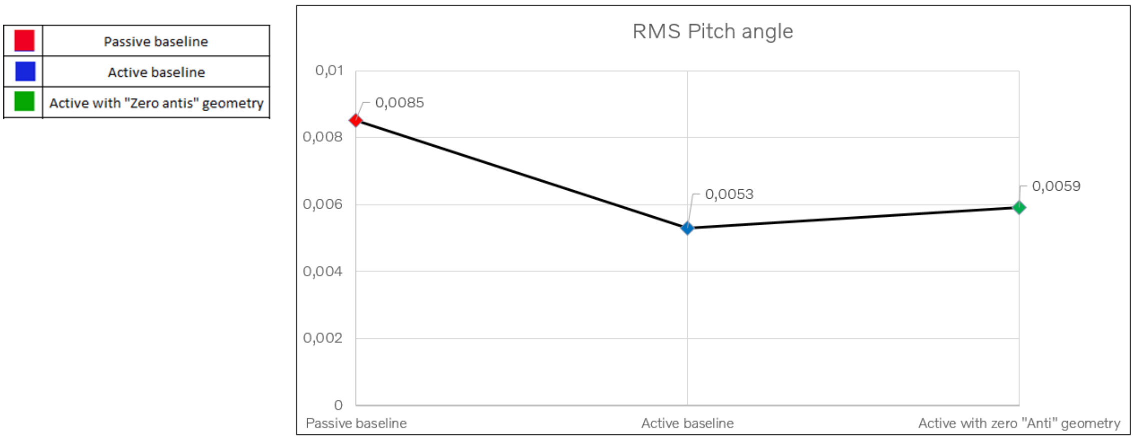

Figure 4.17: RMS values of Vehicle pitch angle - Braking event

4.4 Double Lane Change

An evasive maneuver like double lane change is used to analyse the handling characteristics of the car. The anti-roll bar plays a major role in reducing the body roll of the car with passive suspension.The active baseline suspension when compared to passive baseline suspension in the figure 4.18 performs better by a good margin with respect to the peak roll angle.

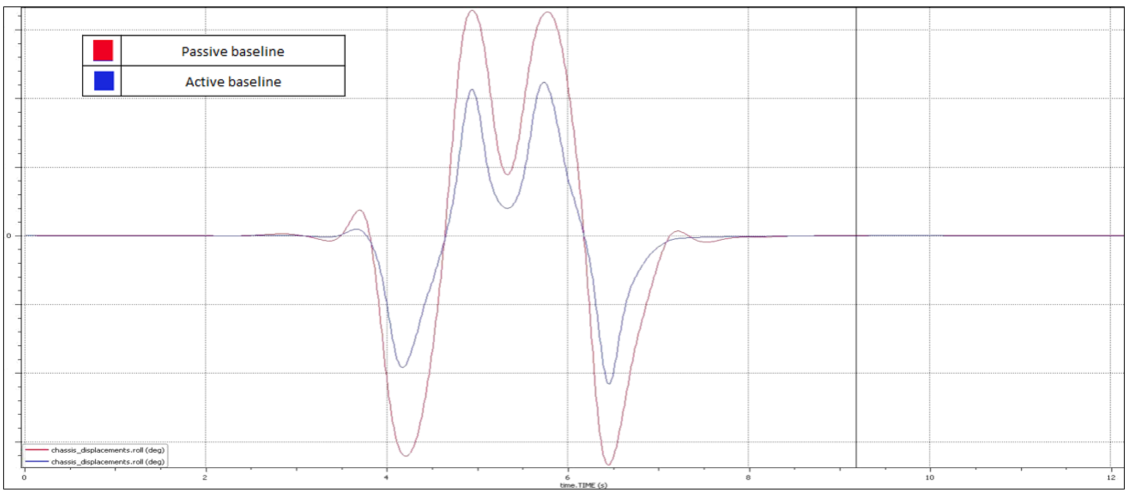

Figure 4.18: Body roll angle (deg) vs time (s) - Double Lane Change baseline simulation results

Further, a comparison is made between passive suspension with ARB and active suspension without ARB. It can be observed that the performance in the peak roll angle of the active suspension without ARB is almost equal to that of passive baseline suspension. This result (figure 4.19) indicates that the active dampers are capable of compensating for the absence of ARB and reducing body roll.

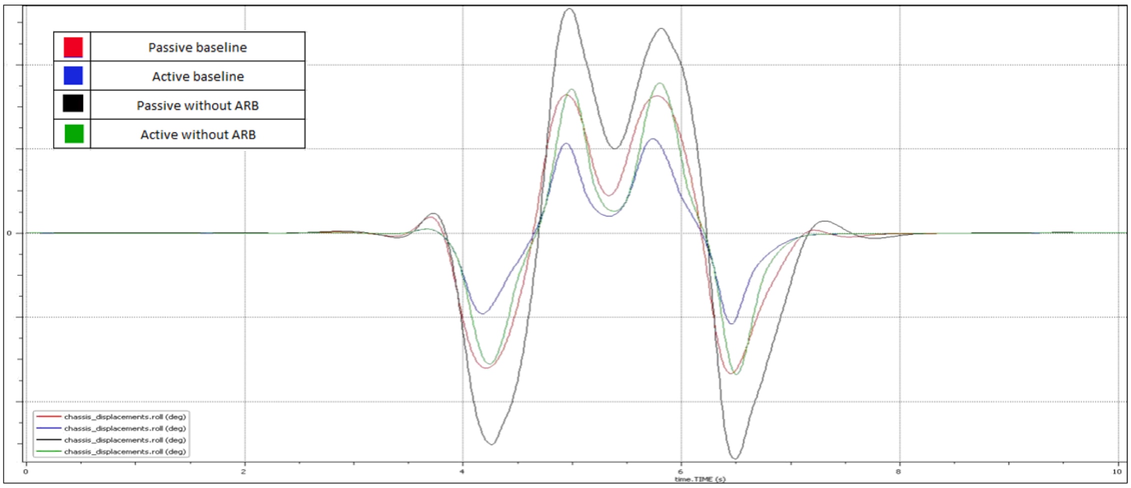

Figure 4.19: Body roll angle (deg) vs time (s) - Double Lane Change without ARB

The results shown in figure 4.20, shows the Roll center height variation with ARB. It is evident that high RC reduces body roll since the distance between the RC and CoG reduced in z-direction, when RCH is increased.

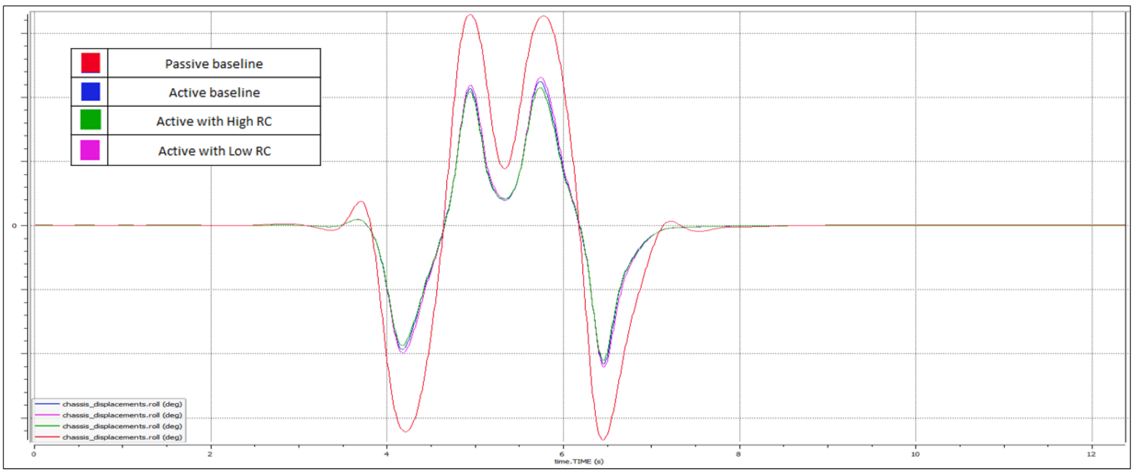

Figure 4.20: Body roll angle (deg) vs time (s) - Double Lane Change with ARB and Roll center height variations

Another parameter that was varied is the limit on the actuator force in the controller. When the actuator force limit is increased by 30% to the actuator limit set in the baseline simulations the active suspension performs better as shown in the figure 4.21. The cost of the increase in damper force needs further investigation and it is out of the scope of the current thesis work.

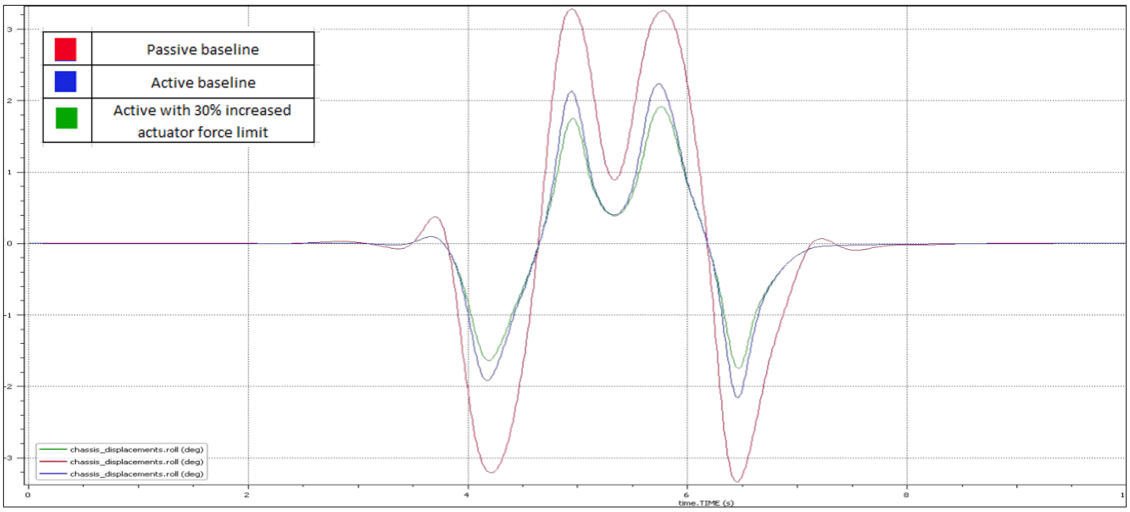

Figure 4.21: Body roll angle (deg) vs time (s) - Double Lane Change with ARB and a higher actuator force limit

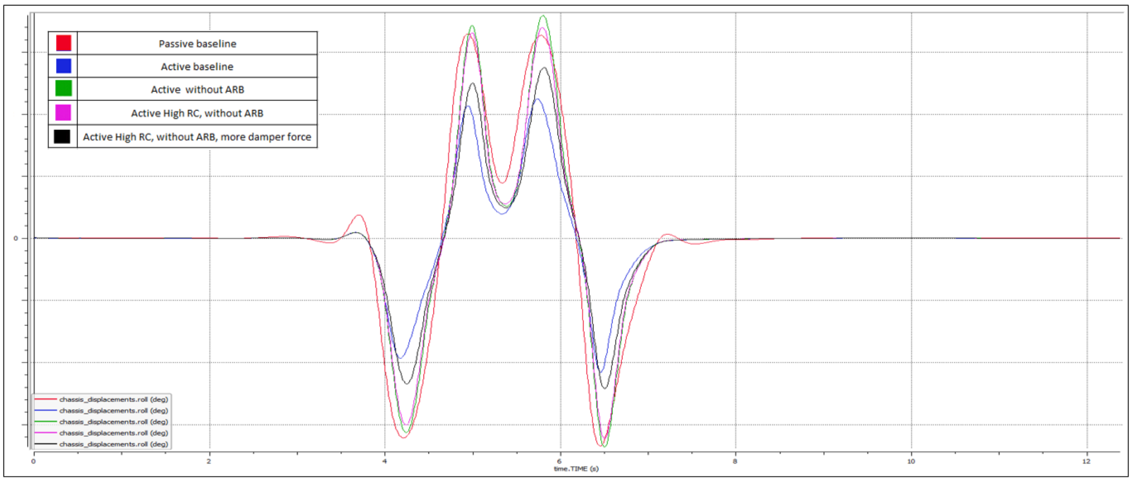

Figure 4.22: Body roll angle (deg) vs time (s) - Double Lane Change without ARB and with RCH variation

The combination of High RC and increased actuator force limit without an ARB is presented in figure 4.22.

4.5 Constant Radius Cornering

The simulation results from Constant Radius Cornering shows more promising results to remove ARB in the presence of active damping. The peak roll angle when steady state is reached is reduced by 50% as shown in figure 4.23 with the use of active suspension compared to the passive suspension. Also, when the ARB is removed from the suspension setup, the peak roll angle is still lesser than the passive baseline suspension as shown in figure 4.24.

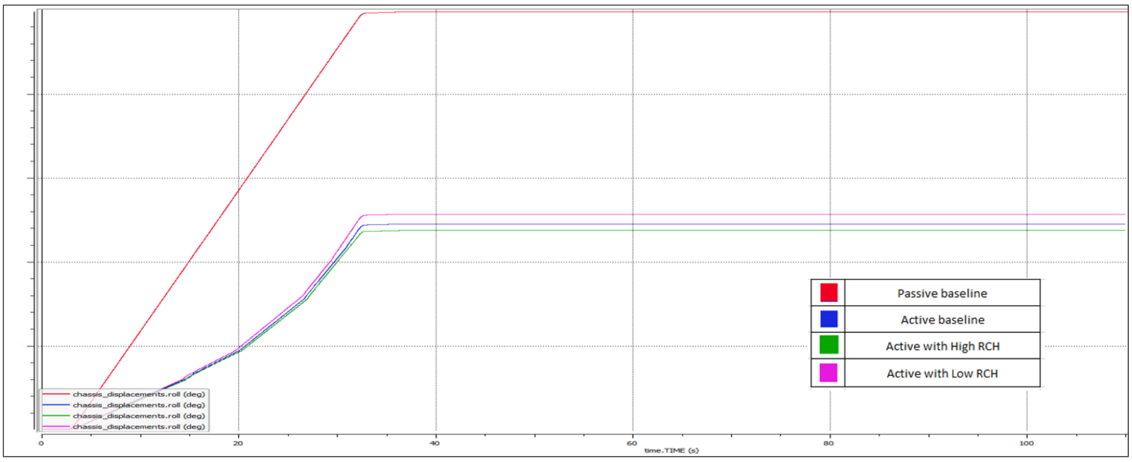

Figure 4.23: Body roll angle (deg) vs time (s) for baseline suspension - Constant Radius Cornering

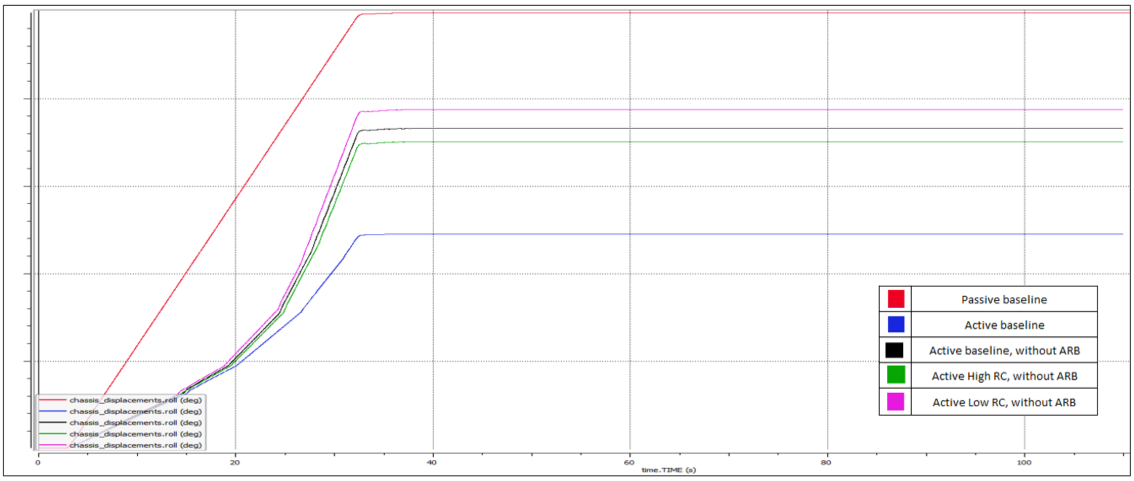

Figure 4.24: Body roll angle (deg) vs time (s) with and without ARB - Constant Radius Cornering

The results obtained in the simulated track events strengthen the finding that the ARB can be eliminated from the suspension in presence of active damping forces, considering the passive suspension performance as the reference.

5

Case Study: Effect of active forces on vehicle steering demands

The methodology established in the previous sections to assess suspension performance, is applied to the analysis of steering demands required in the presence of active damping forces. The investigation involved modelling several test tracks in driving scenarios explained in later sections and also developing vehicle models with varied suspension kinematics and suspension parameters. The findings obtained in this case study are summarised and explained in the results section.

5.1 Motivation

The earlier analysis of suspension performance with variation in kinematic parameters showed a positive impact on several vehicle performance attributes such as body roll, vehicle pitch and steering wheel angles. Bumpsteer reduction had a significant positive impact on the vehicle steering demands. But it would be interesting to investigate if any other suspension kinematic parameter affected steering demands as much as bumpsteer variation did. This formed the basis for undertaking a separate case study of analysing the steering corrections required in the presence of active damping forces and variation in several kinematic parameters.

5.2 Methodology

The methodology for this investigation remains the same as earlier, wherein the suspension performance is compared between passive baseline, active baseline and active with a varied suspension kinematic parameter. Active suspension models with a variation in kinematic parameter is developed by varying one suspension parameter at a time making it simpler to analyse the results and draw conclusions. This approach provides a better platform for more in depth analysis of the effect of a single suspension parameter.

5.2.1 Test tracks

It is imperative to choose the correct type of road profile for a case study involving steering demands analysis. The intention is to model a road with alternate out of phase bumps on the left and right side of the road, inducing continuous body roll when the car is driven over it. Several iterations of test tracks were conducted which can be seen from figures A.2 to A.4 in the appendix section. The selected test track was modelled with alternate bumps over a length of 200m. The road was modelled with one sided irregularity to induce roll in the car at a given instance with a length of 5m and a height of 40mm. The road profile used in the full vehicle simulations for this case study is as shown in the figure 5.1.

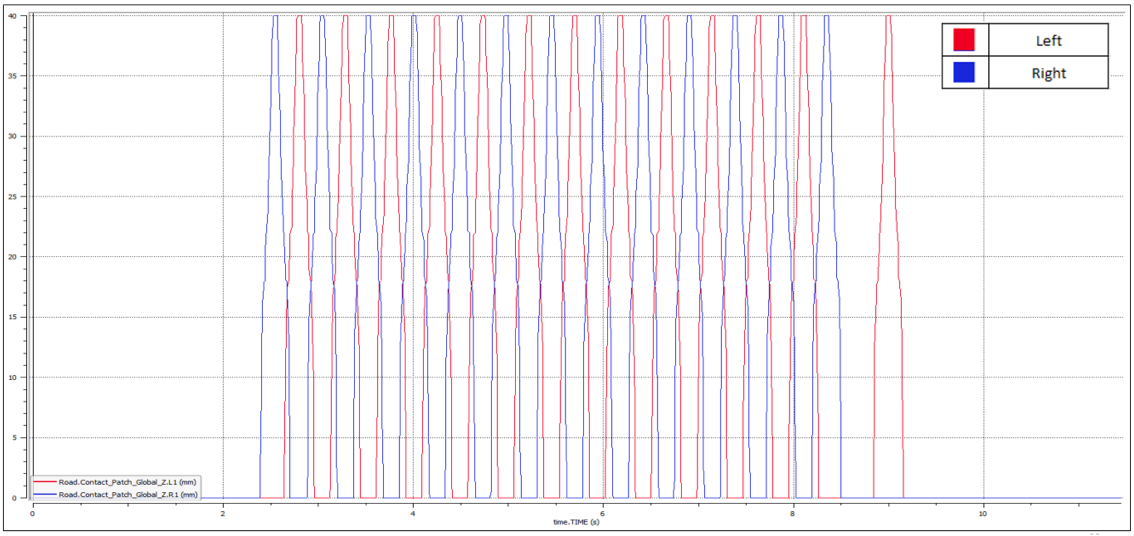

Figure 5.1: Road profile used for steering case study

5.2.2 Vehicle models with varied suspension kinematics

Vehicle models used in the case study are modelled in VI-CarRealtime software with modifications done to the front and rear suspensions, keeping the other vehicle subsystems the same for a better comparison. A number of suspension models were developed to test the variation in suspension kinematic parameters. The variation in the suspension parameter is achieved by tuning the suspension hardpoints in the VI-SuspensionGen utility of VI-CarRealtime and further these suspension models are used to create new full vehicle models. The kinematic parameters selected for this case study and the corresponding variation in their value are listed in the table 5.1.

Table 5.1: List of kinematic parameters and the corresponding variation for the steering demands case study

Suspension kinematic parameter	Unit	Variation in the value for front and rear suspension
Static toe	deg	Zero
Static camber	deg	50% reduction
Bumpsteer	deg/m	50% reduction
Camber gain	deg/m	50% reduction
Spring Stiffness	N/mm	20% and 40%reduction
ARB Stiffness	N/mm	Zero (No ARB)

5.2.3 Driving trials in the simulator

The vehicle model and the co-simulation model are built in the VI-CarRealTime platform which is compatible with the Volvo Driving Simulator. This provided an opportunity to run the models in the simulator and a possibility of comparing the results from simulations and simulator. Also one can experience the vehicle suspension behaviour in the simulated maneuvers and judge the performance subjectively along with the objective data obtained from the simulator.

5.3 Results

This section details regarding the influence of the varied suspension parameters on the steering wheel demands and the toe angles of the car. The aim is to objectively assess the steering demands required in each simulated setup with a certain variant of suspension kinematics. The results are categorised for each variant of kinematic parameter starting with the baseline simulation results. As mentioned in the earlier sections of the report, the focus is more on the trends in the suspension performance than on the magnitude of the attributes assessed. Also, the results from the simulator trials are summarised and presented in the corresponding section.

5.3.1 Baseline results

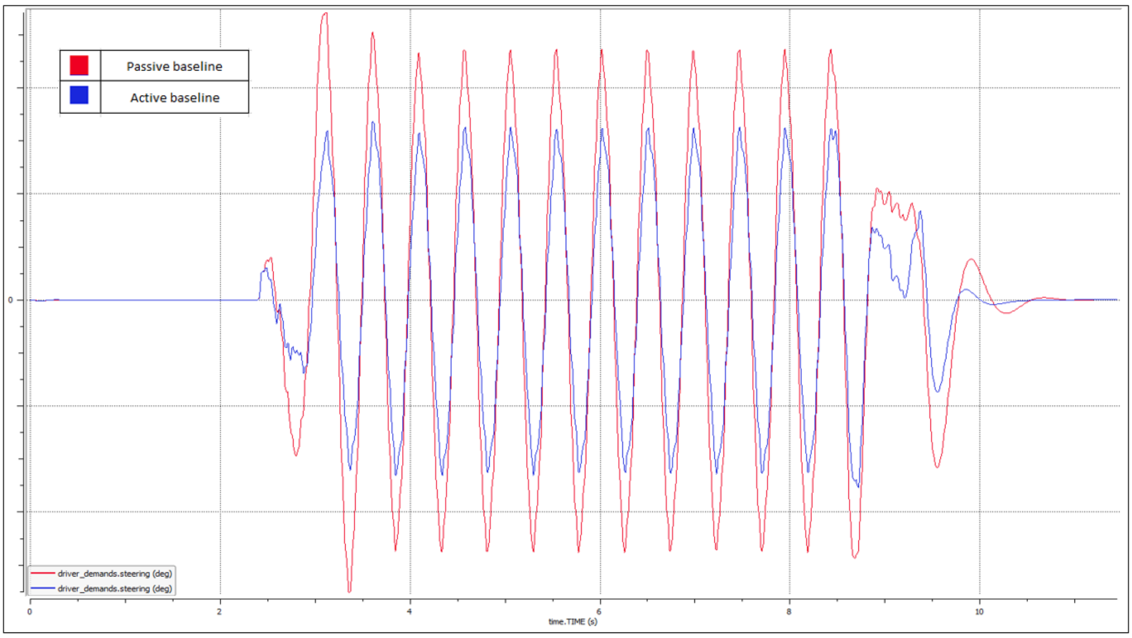

Figure 5.2: Steering wheel angle (deg) vs time (s) - Steering case study baseline simulation

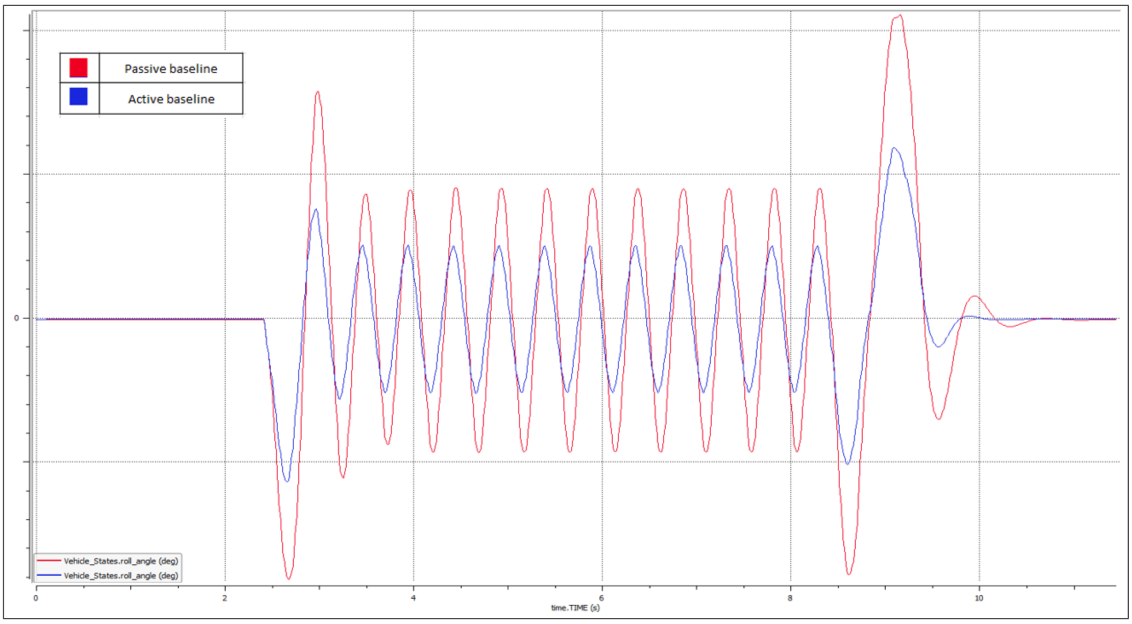

Figure 5.3: Body roll angle (deg) vs time (s) - Steering case study baseline simulation

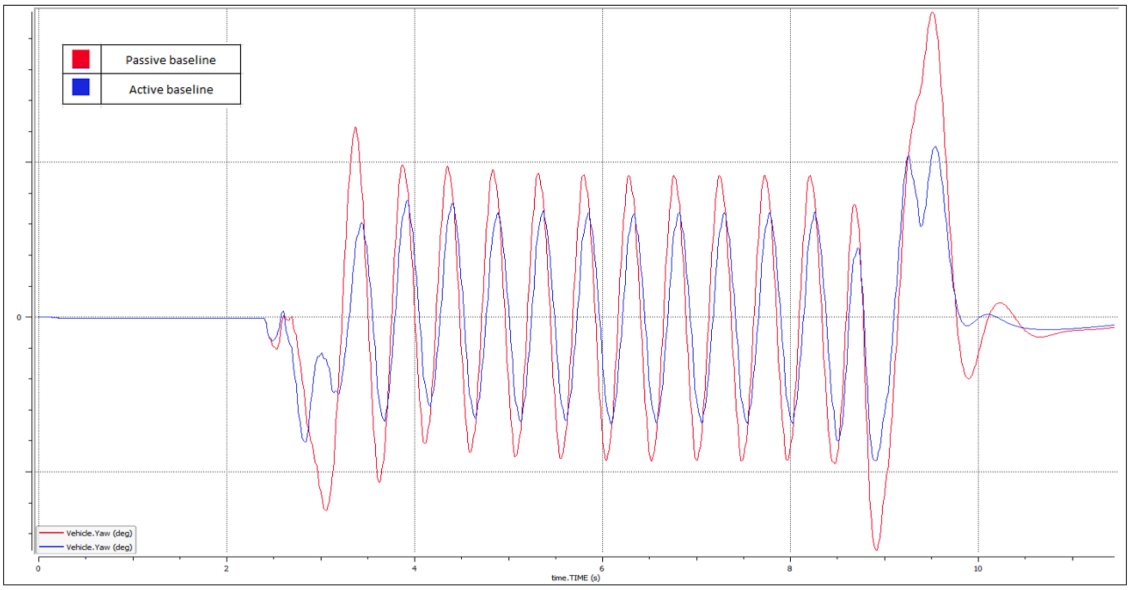

Figure 5.4: Vehicle yaw angle (deg) vs time (s) - Steering case study baseline simulation

5.3.2 Static camber and Toe variation

The static camber and toe variation did not have a significant impact on the steering characteristics of the car with active suspension as shown in the figures 5.5 and 5.6. It is generally a design judgment to have a little toe on the front axle (positive to help with cornering and negative to improve straight line stability), thereby improving the suspension performance.

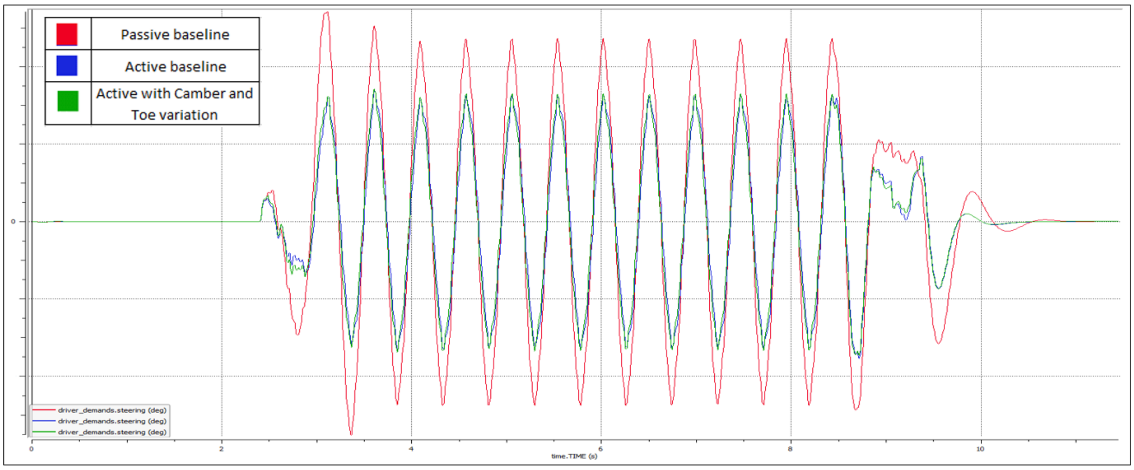

Figure 5.5: Steering wheel angle (deg) vs time (s) with camber and toe variation

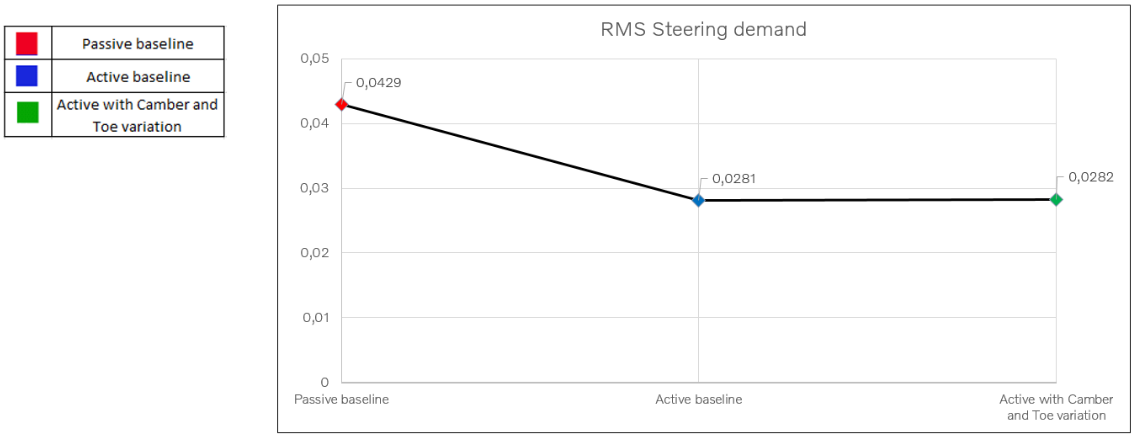

Figure 5.6: RMS value of steering wheel angle with camber and toe variation

5.3.3 Bumpsteer variation

As seen from the earlier investigations, bumpsteer variation has a signficant impact on the vehicle steering demands. This finding is further substantiated by figure 5.7 which shows the reduction in steering wheel angle with a reduced bumpsteer value. Root Mean Square (RMS) plots serve as a good measure of comparison to assess a parameter such as Steering wheel angle. Figure 5.8 shows a steady decline in the RMS values of steering wheel angle from passive baseline to active suspension setup with half the bumpsteer value compared to passive.

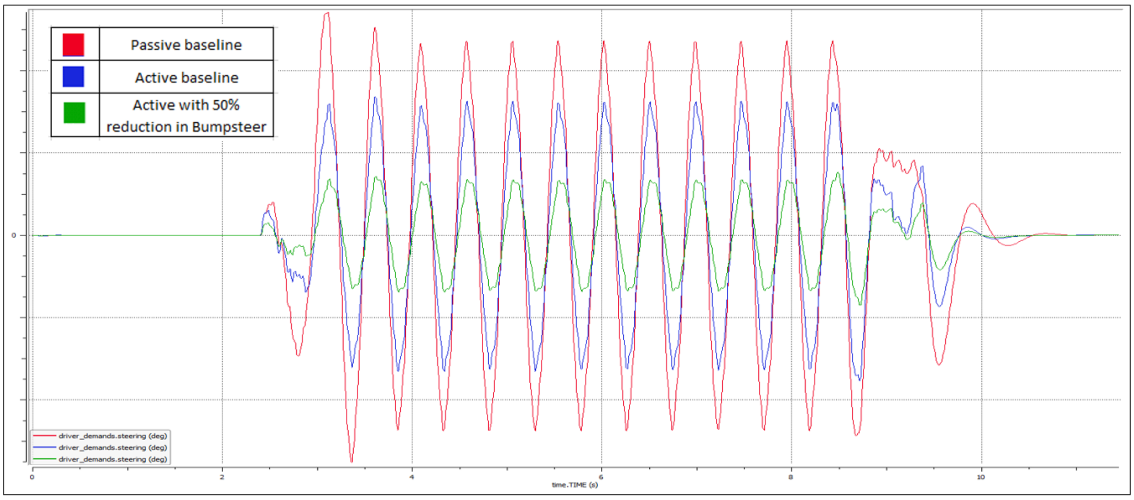

Figure 5.7: Steering wheel angle (deg) vs time (s) with bumpsteer variation

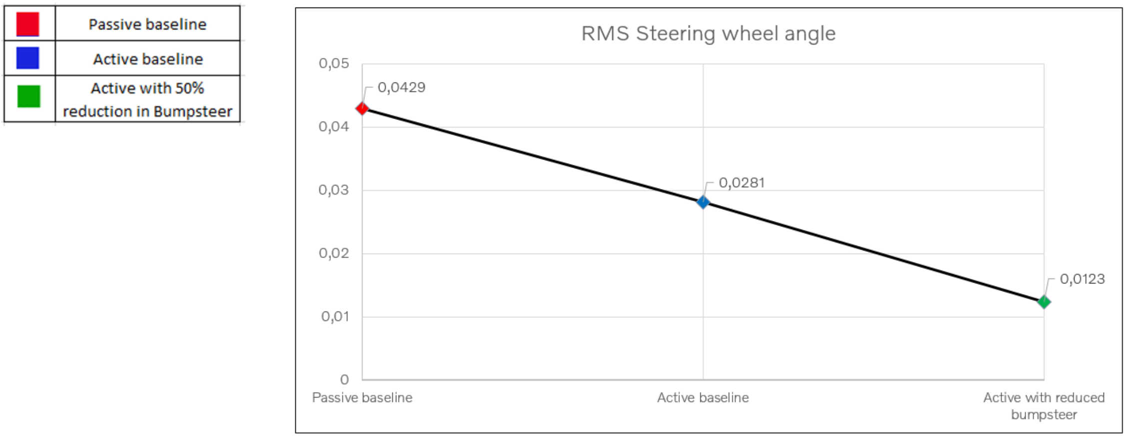

Figure 5.8: RMS value of steering wheel angle with bumpsteer variation

Another characteristic about the modelled test track is that the bumps are of low frequency. Thus, the above results are analysed in the frequency spectrum of 0 - 5 Hz i.e., assessment of primary ride. The corresponding results are shown in figure 5.9 where bumpsteer variation yields lower peaks in the frequency spectrum.

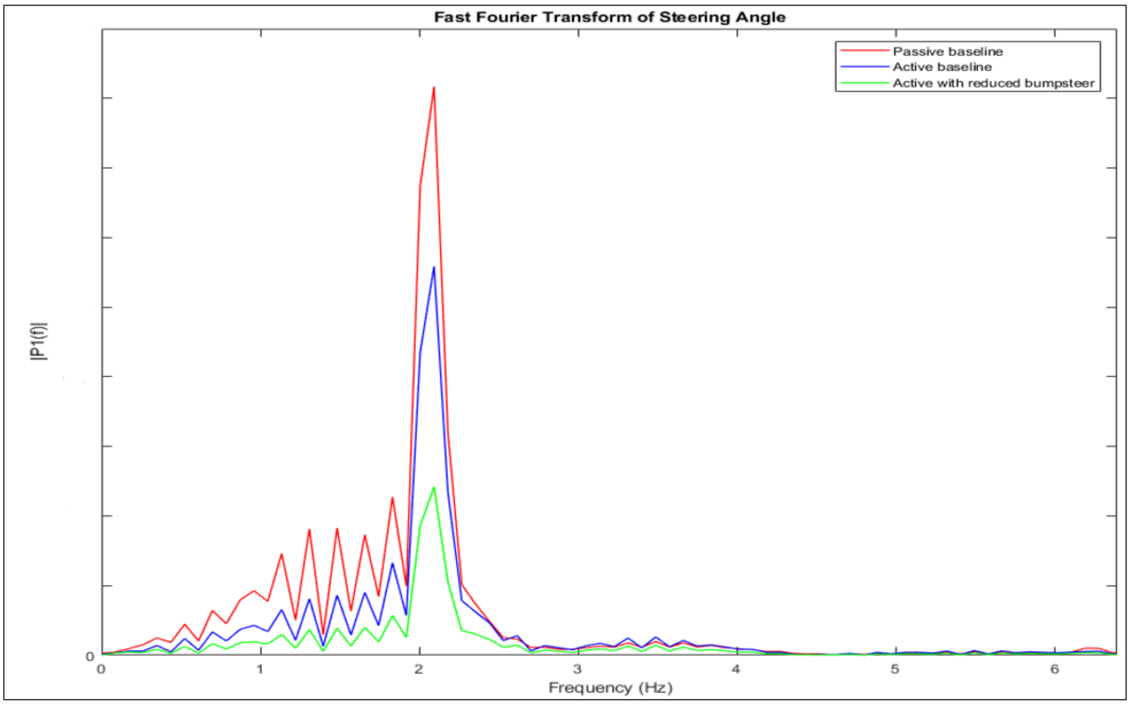

Figure 5.9: FFT Analysis of the steering demands with reduced bumpsteer

It can be further observed from figure 5.10 that the vehicle yaw is reduced in case of tuned active suspension setup. This finding can be correlated from the results obtained in the simulator trials as shown in the figures 5.11 and 5.12. In this case, the driver experiences less yaw and hence lesser steering corrections are required to keep the car straight.

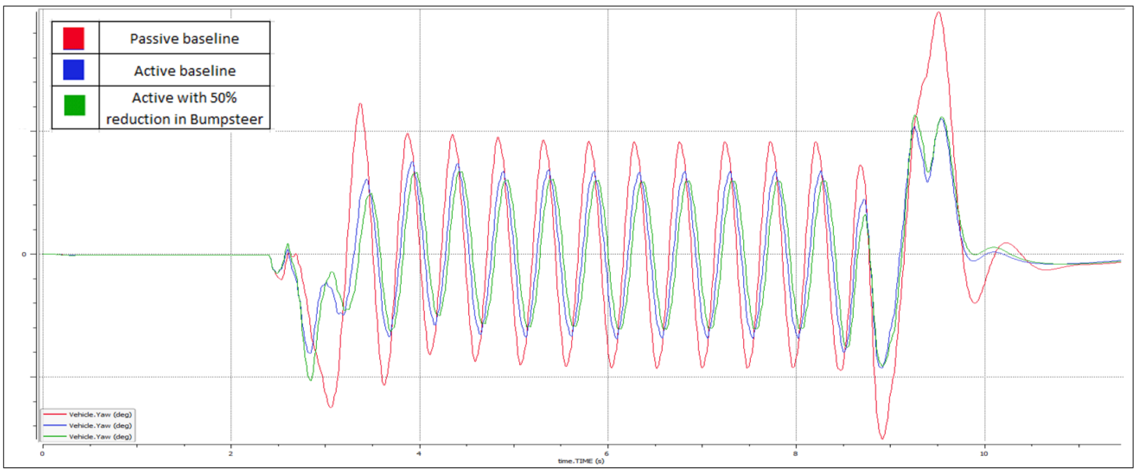

Figure 5.10: Vehicle yaw angle (deg) vs time (s) with bumpsteer variation

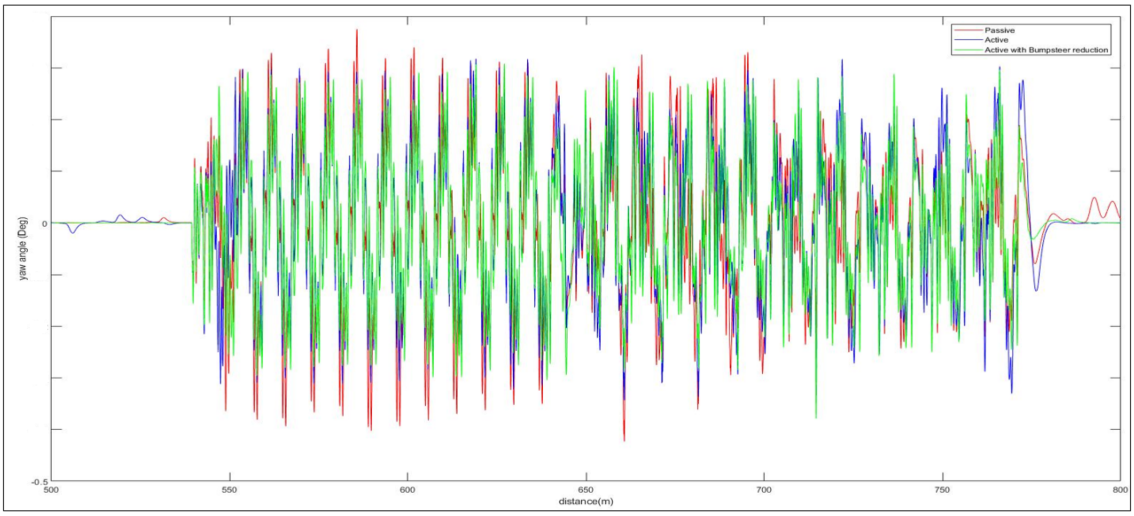

Figure 5.11: Vehicle yaw angle vs time with bumpsteer variation - from Driving Simulator results

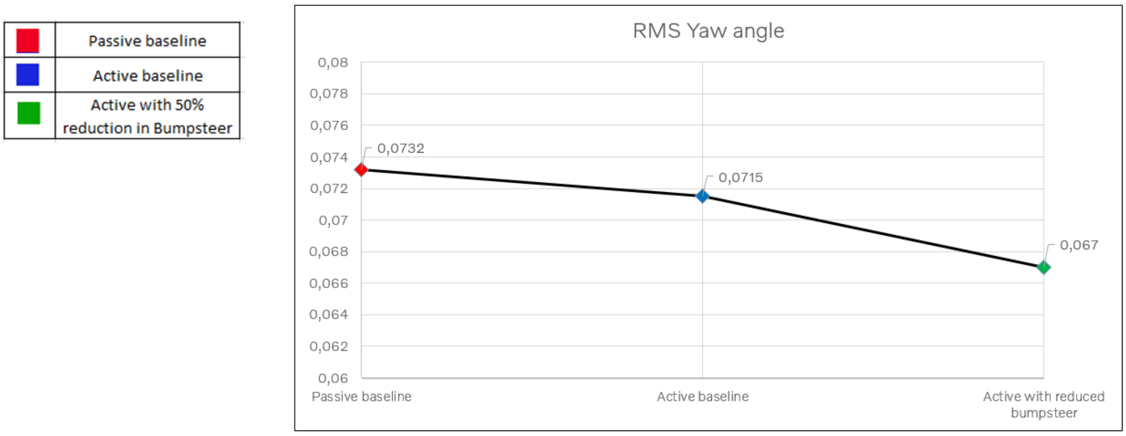

Figure 5.12: RMS value of vehicle yaw angle with bumpsteer variation -from Driving Simulator results

5.3.4 Spring stiffness variation

The variation in spring stiffness affects the lateral load distribution when roll is induced in the car. Thus, changing the stiffness of the spring affects the steering performance of the car when driven over a disturbance patch. The effect of reduction in spring stiffness is as shown in figure 5.18. The plot implies that the steering demands are reduced with a reduction in spring stiffness.

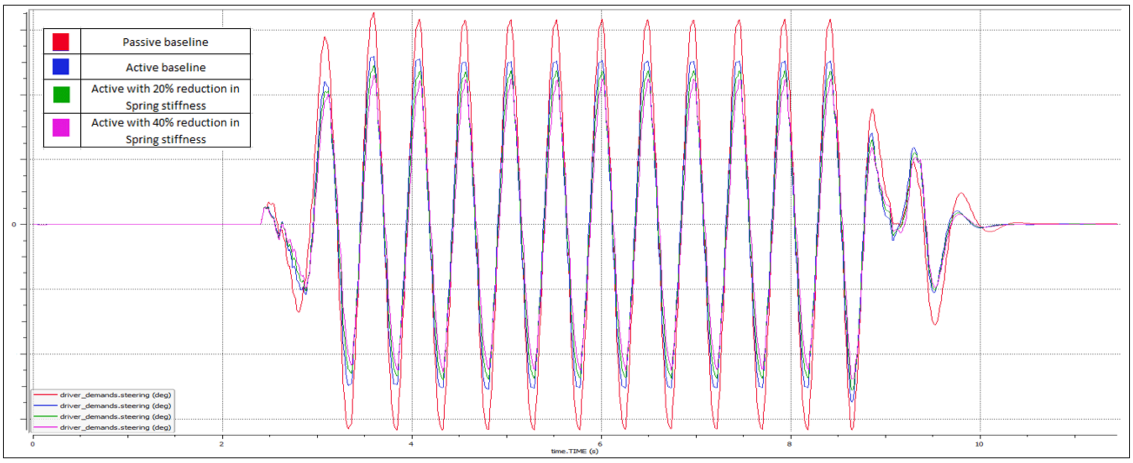

Figure 5.13: Steering wheel angle (deg) vs time (s) with spring stiffness variation

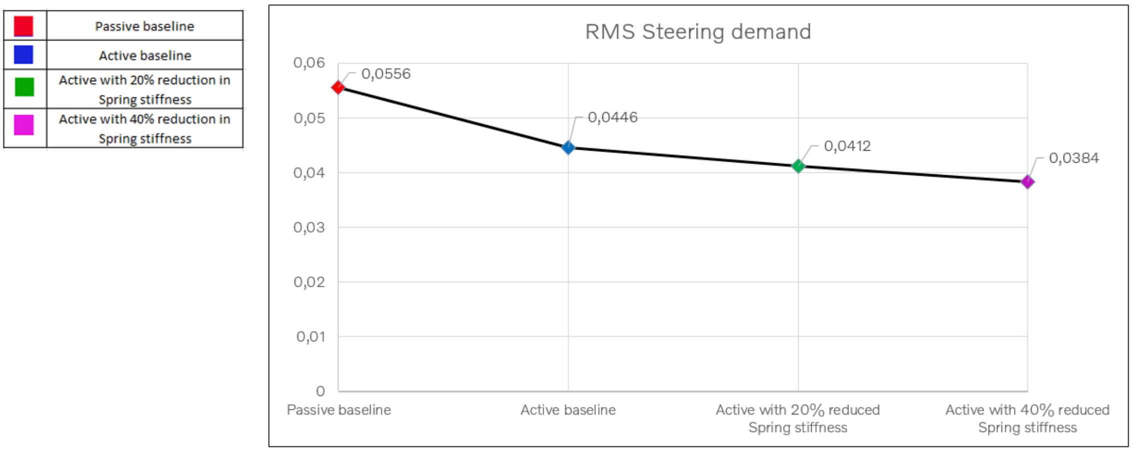

Figure 5.14: RMS value of steering wheel angle with spring stiffness variation

The same trend can be observed in the frequency spectrum of 0-5 Hz. The figure shows the effect of spring stiffness in the frequency spectrum. Also, an decrease in spring stiffness improves the vertical acceleration and thereby ride comfort, since active dampers has better control owing to lower spring rates.

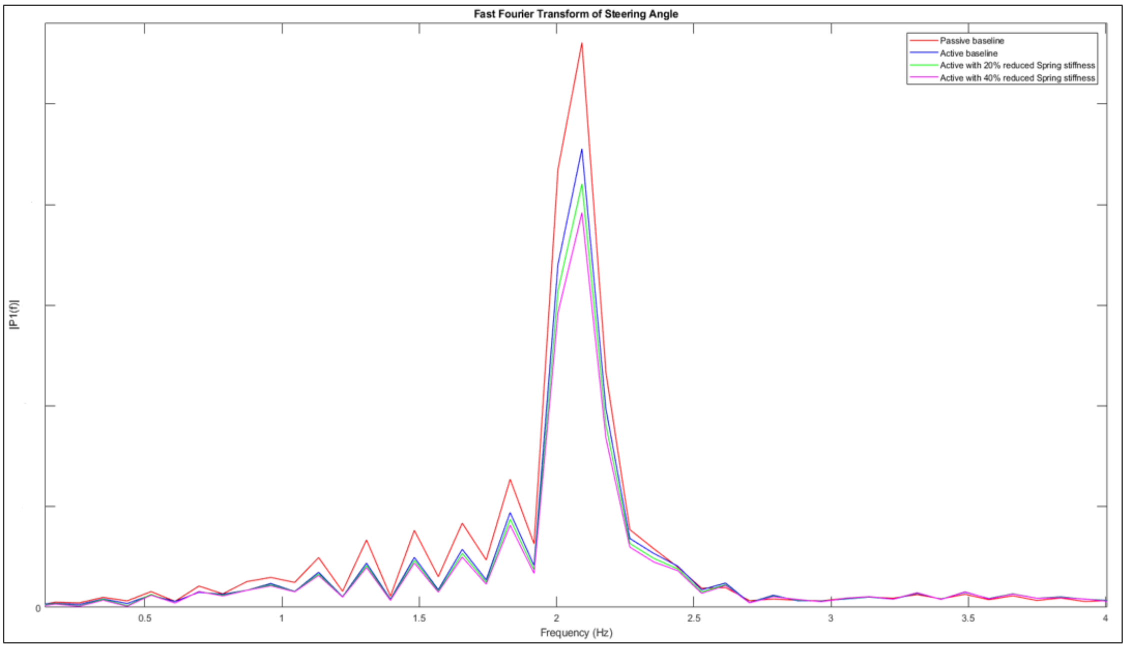

Figure 5.15: FFT Analysis of steering demands due to variation in spring stiffness

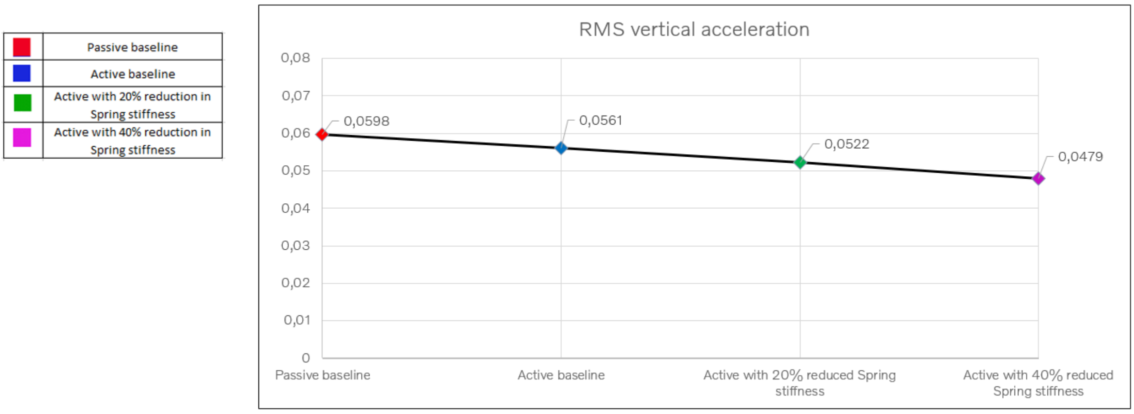

Figure 5.16: RMS value of vertical acceleration with variation in spring stiffness

The results from these full vehicle simulations can be further correlated with the simulator trials which shows lesser vehicle yaw with decreased spring stiffness and hence lesser steering corrections required to keep the car straight.

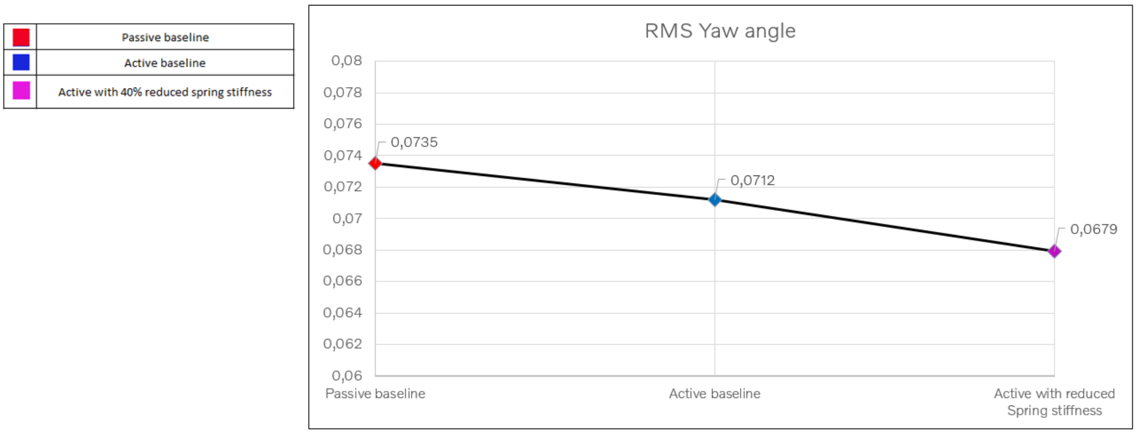

Figure 5.17: RMS value of vehicle yaw angle with spring stiffness variation -from Driving Simulator results

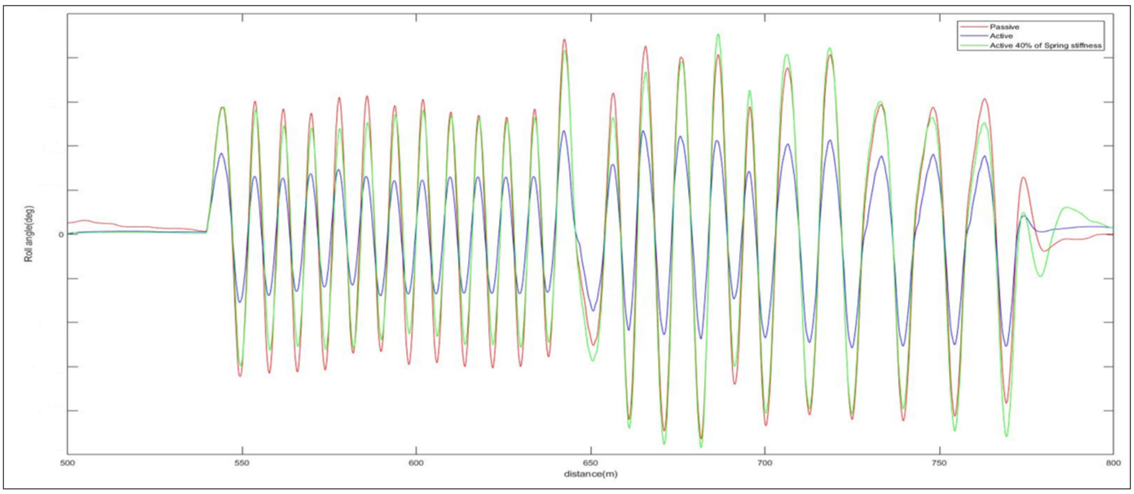

Figure 5.18: Body roll angle (deg) vs time (s) with spring stiffness variation - from Driving Simulator results

5.3.5 Driving without ARB

Investigating the possibility of suspension design simplification is one of the major objectives of this thesis work. Analysing the body roll in several driving scenarios will help in drawing conclusions regarding the possibility of ARB removal from the wheel suspension design.

Figure 5.19 shows the body roll angle vs time for different suspension variants. It is interesting to note that the body roll angle when ARB is removed is lesser than the active baseline setup which shows the potential of active damping in reducing body roll. Another finding to be noted here is that when the actuator force limit is increased, the peak values of body roll is reduced when compared to the case with lesser actuator force limit.

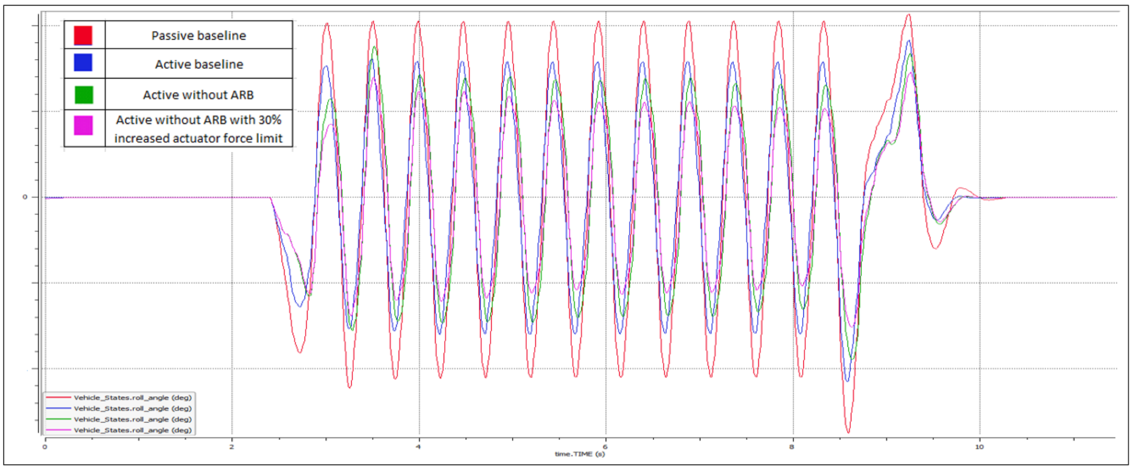

Figure 5.19: Body roll angle (deg) vs time (s) - Driving without ARB

However, driving without ARB affects the vehicle steering demands negatively in the simulated setup (refer figures 5.20 and 5.21). The steering demands increases when driving without ARB and this can be substantiated by higher toe variation as seen in the figure 5.22. This might be because of higher wheel travel resulting from inadequate active damping. Further, when the active damping is increased by 30%, it is observed that the steering corrections decrease, indicating that the actuator force limit is a crucial parameter of interest.

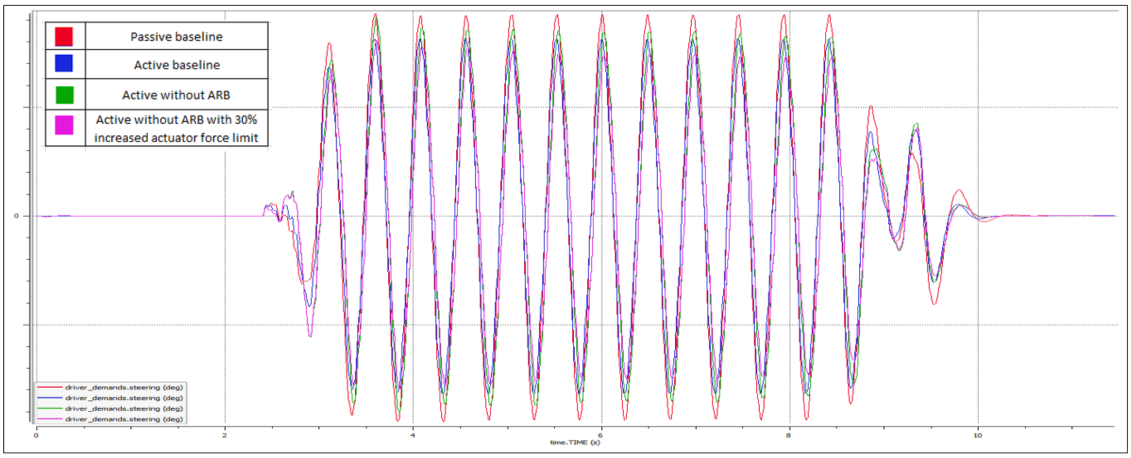

Figure 5.20: Steering wheel angle (deg) vs time (s) - Driving without ARB

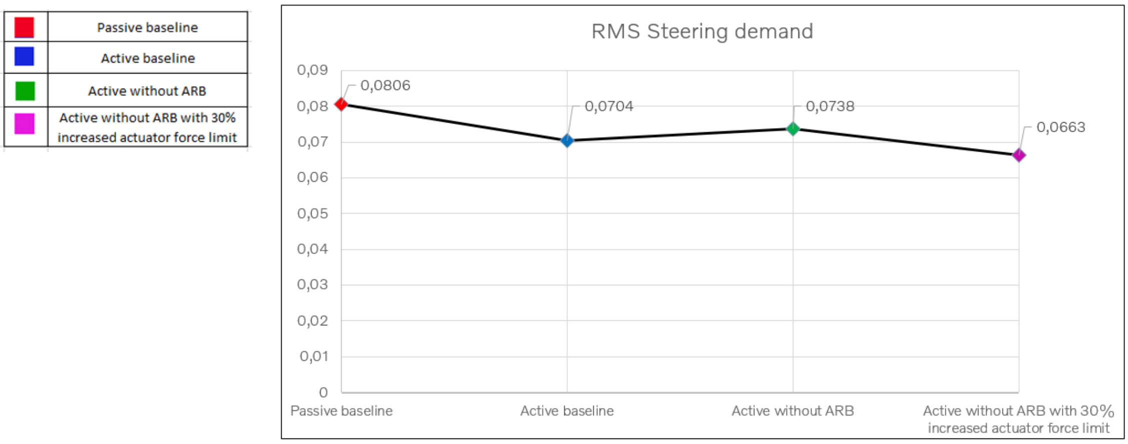

Figure 5.21: RMS values of steering wheel angle - Driving without ARB

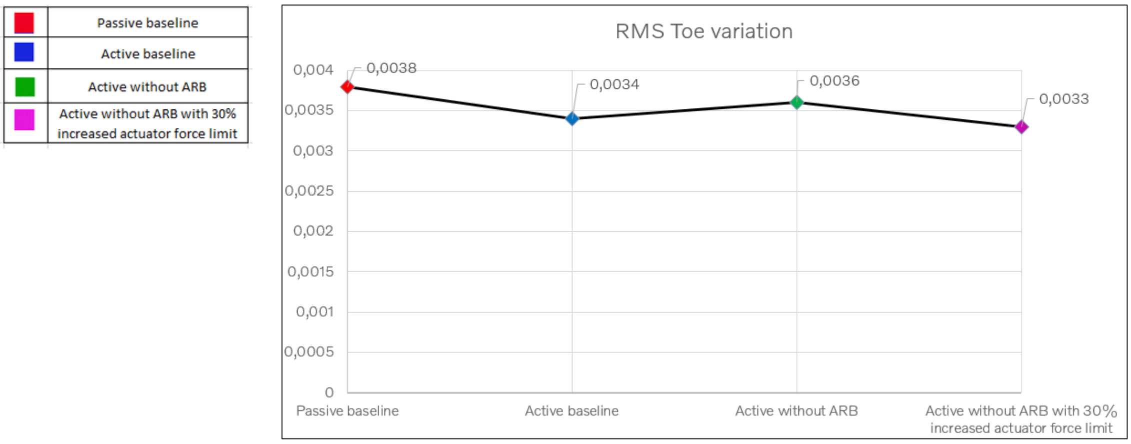

Figure 5.22: RMS values of front toe angle variation - Driving without ARB

6
Conclusions

The driving events for this thesis work are set up as per the ISO standards and suspension models are generated with modifications to the kinematic behaviour of the suspension. Important factors such as compliance analysis, bushing stiffness among others, are not considered. So, the exact values of the assessed parameters resulting in the simulations cannot be stressed upon. However, the behavioral trends of the suspension in the simulated scenarios are well observed. Thus, based on the methods adopted for the thesis work and the results obtained, following conclusions can be drawn.

- Active suspension improves several vehicle dynamics parameters/attributes and ride comfort parameters relative to passive suspension and shows potential to reduce the conventional trade-offs for active suspension, which might be unavoidable when designing its passive counterpart.

- There exists a scope to eliminate ARB in the presence of active dampers and still match the performance to that of passive suspension in handling maneuvers like Double Lane Change and Constant Radius Cornering. Data obtained from the full vehicle simulations shows promising results in this regard. Driving without an ARB on a road with stochastically placed bumps resulted in lesser body roll than the case with ARB. Thus, this can eliminate one physical component in the car by using the active dampers and can be cost effective while gaining similar or even better performance compared to passive suspension by increasing the actuator force limit.

  However, the body roll is higher when active suspension without ARB is compared with active "baseline" suspension. One has to think about reducing this difference in the performance by introducing high Roll center and increased actuator force limit which further results in increased cost and energy consumption by the active system.

- The variation in "anti" effects does not show significant effect in the constant speed maneuvers. However, the dynamic maneuvers like acceleration and braking show that the anti effects can be eliminated from the geometry and still match performance of active suspension to the passive baseline. Also, the elimination of anti effects from the suspension geometry enables active dampers to better control the wheel movement. However, the effects on zero anti-geometry on secondary ride assessment needs further evaluation. Only primary ride assessment was considered in the thesis work because of the simpler tire models and limitations on controller tuning.

- For some cases, steering demands increase when active forces are introduced in the suspension as seen from the constant speed maneuvers with unveven road irregularities. This, however can be reduced by reducing the bumpsteer (or jounce steer) value of the suspension. Bumpsteer plays a significant role in the steering behaviour of the car as indicated by most of the results obtained in this thesis work.

- Steering case study yielded interesting results with simulations performed for a variety of suspension kinematic changes. To summarise, bumpsteer reduction resulted in lesser steering demands. For certain driving scenarios, one might have to consider increasing the actuator force limit as this results in improved suspension behaviour and thereby better handling of the vehicle. Further, reducing the spring stiffness showed improved ride comfort.

7
Future Work

- To limit the scope of the thesis towards study of suspension kinematic behaviour, controller tuning was not undertaken. Active suspension controller plays a significant role in realizing the full potential of the system and to yield better results in the full vehicle simulations. The controller used in this thesis work is developed on skyhook principle and other control strategies can be explored and evaluated. The same controller was integrated with the Driving simulator interface which showed potential for further tuning.

- Compliance analysis plays a huge role in suspension design. It will be interesting to consider compliance analysis in a study like this, to note the variation in results. Introducing analyses involving bushing stiffness, weight distribution, load transfer, etc., will certainly help in improving the results and motivates to look at the magnitude of the assessed parameter and not just the behavioral trends.

- It is difficult to vary a single suspension kinematic design parameter and not affect other parameters since most parameters are interconnected in vehicle suspension design. For example, a modification to bumpsteer by changing the tierod hardpoints will vary the Instantaneous Center (IC) of the suspension and this most certainly affects other parameters as well. It will be interesting to study the combined effects of variation in more than one parameter as compared to an isolated sensitivity study of a single parameter.

- Case studies can be conducted which involves different variations of wheel travel. Wheel travel was a constant parameter in all the simulation setups in this thesis work. Previous studies have shown potential of reducing wheel travel in the presence of active damping forces. However, this also requires modifications to bumpstop behaviour to prevent bottoming out of suspension.

- Few simulations were performed in the driving simulator which enabled to obtain subjective feedback regarding the suspension performance along with the objective data logged by the simulator. This is a cost effective alternative to driving a test car with different suspension setups. Although one must be experienced with simulator driving to subjectively judge the suspension performance, it does help with tuning the controller and also to correlate the results from simulator with the results from full vehicle simulation in the CAE tool. This area should definitely be explored.

- In-depth investigations involving a case study can be performed with high fidelity models to improve the results. For example : Case study of removing ARB from the suspension can be conducted with a Multi Body simulation model with elasto-kinematic simulations.

- In this thesis work, variations in kinematic targets were achieved by modification to hardpoints. There was freedom to move the hardpoints in space since wheel assembly packaging study was out of scope. However in practice, packaging and wheel envelope studies need to be conducted to assess the feasibility of the desired kinematic targets generated as a function of new hardpoints. This area can be further investigated for the final set of hardpoints obtained after several iterations.

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A

Appendix

A.1 Additional result from full vehicle simulations

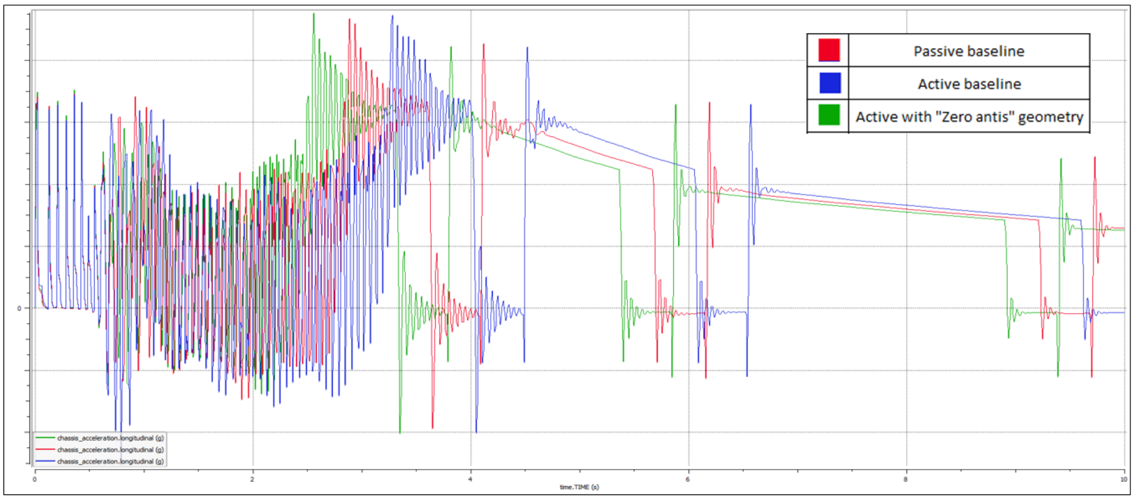

Figure A.1: Longitudinal acceleration (g) vs time - Acceleration event

A.2 Iteration of road profiles for steering case study

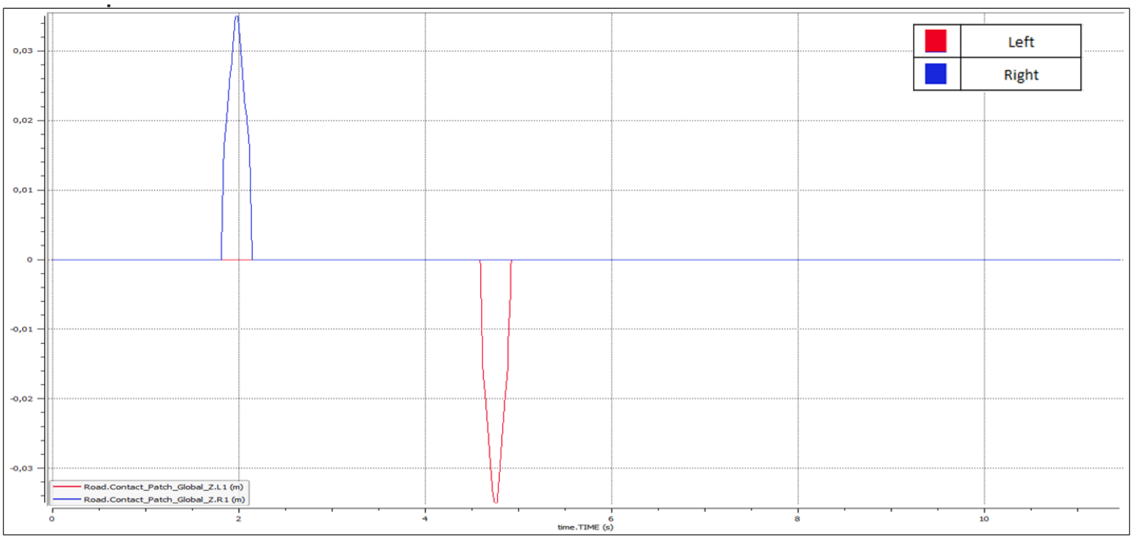

Figure A.2: Road profile iteration-1 for steering demands case study

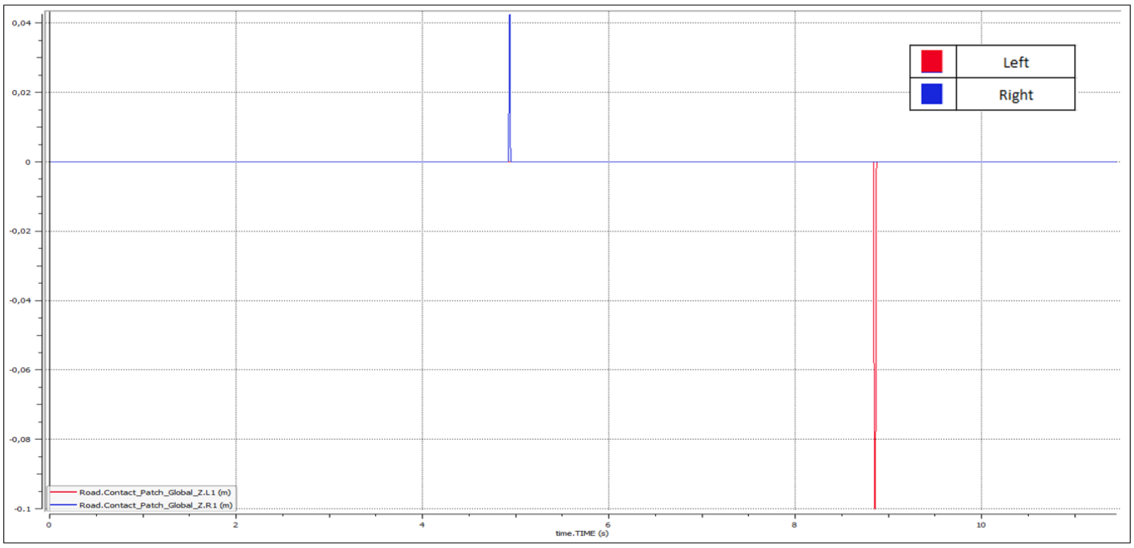

Figure A.3: Road profile iteration-2 for steering demands case study

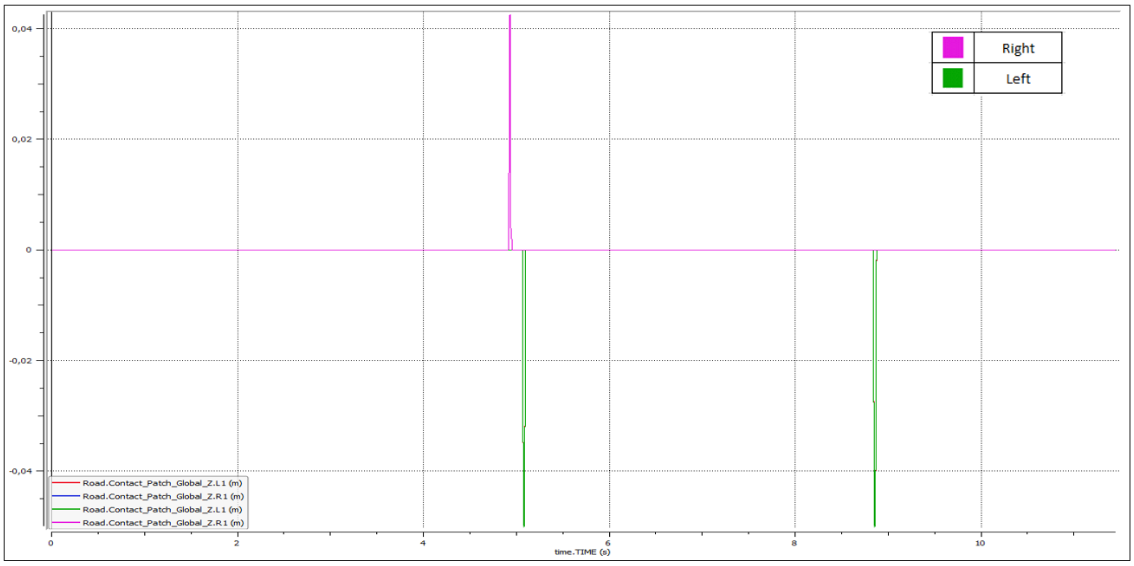

Figure A.4: Road profile iteration-3 for steering demands case study

A.3 Event files from VI-EventBuilder for different driving scenarios

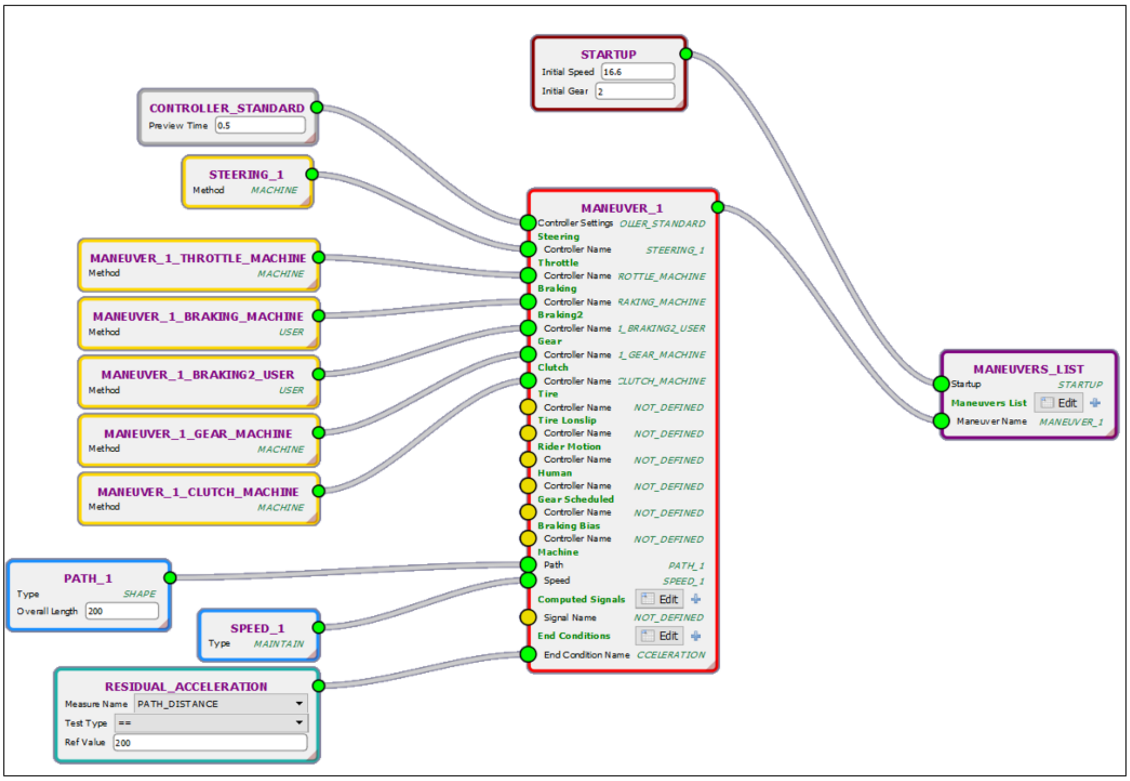

Figure A.5: Event file for driving straight at 60kmph - VI-Event Builder

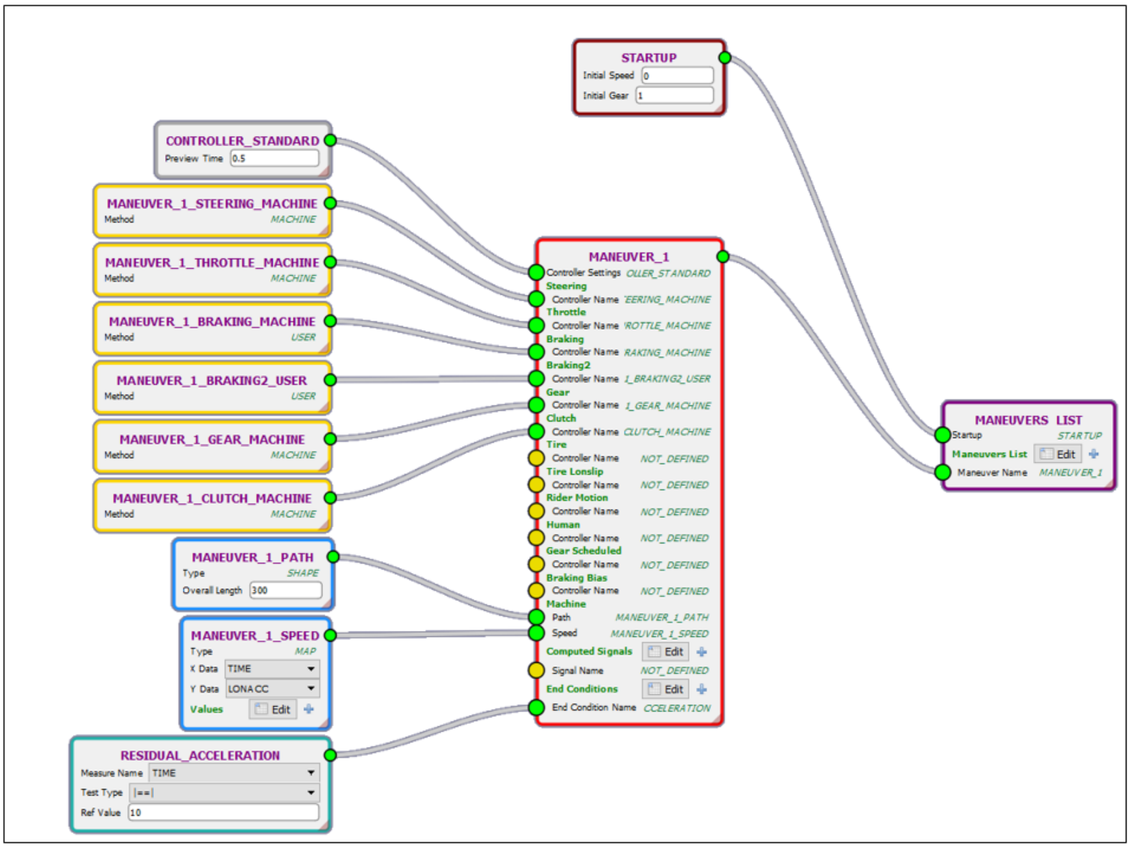

Figure A.6: Event file for Acceleration setup - VI-Event Builder

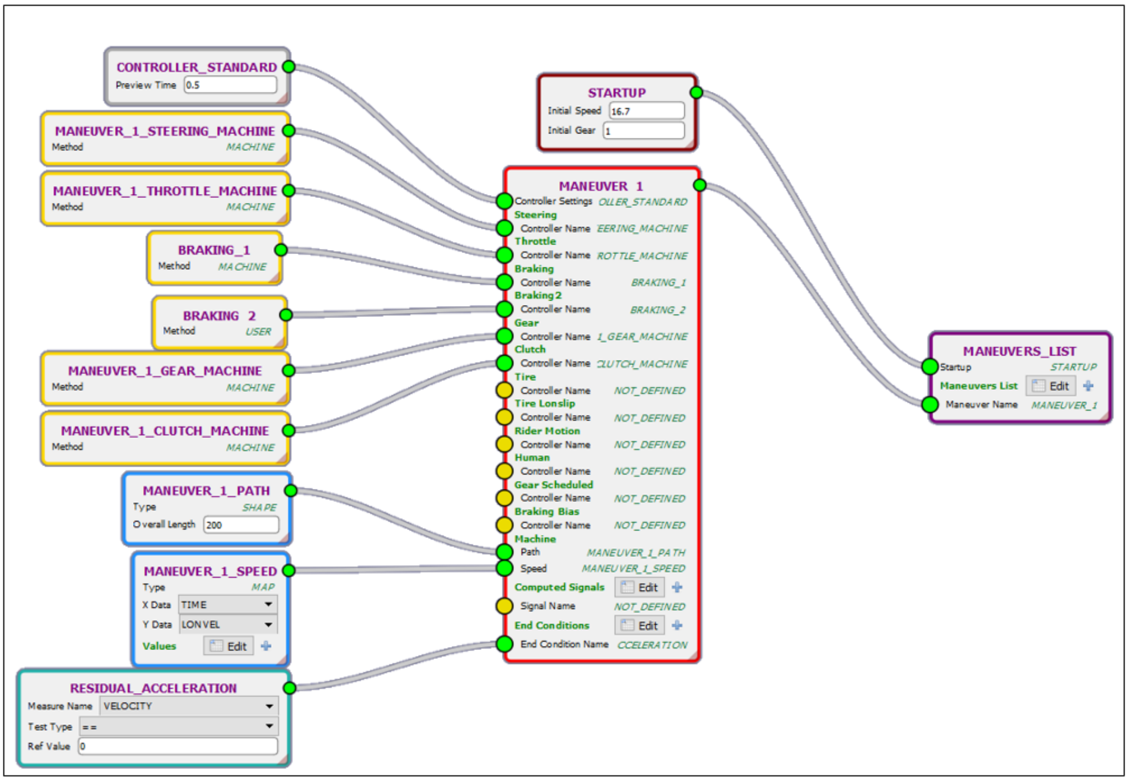

Figure A.7: Event file for Braking setup - VI-Event Builder

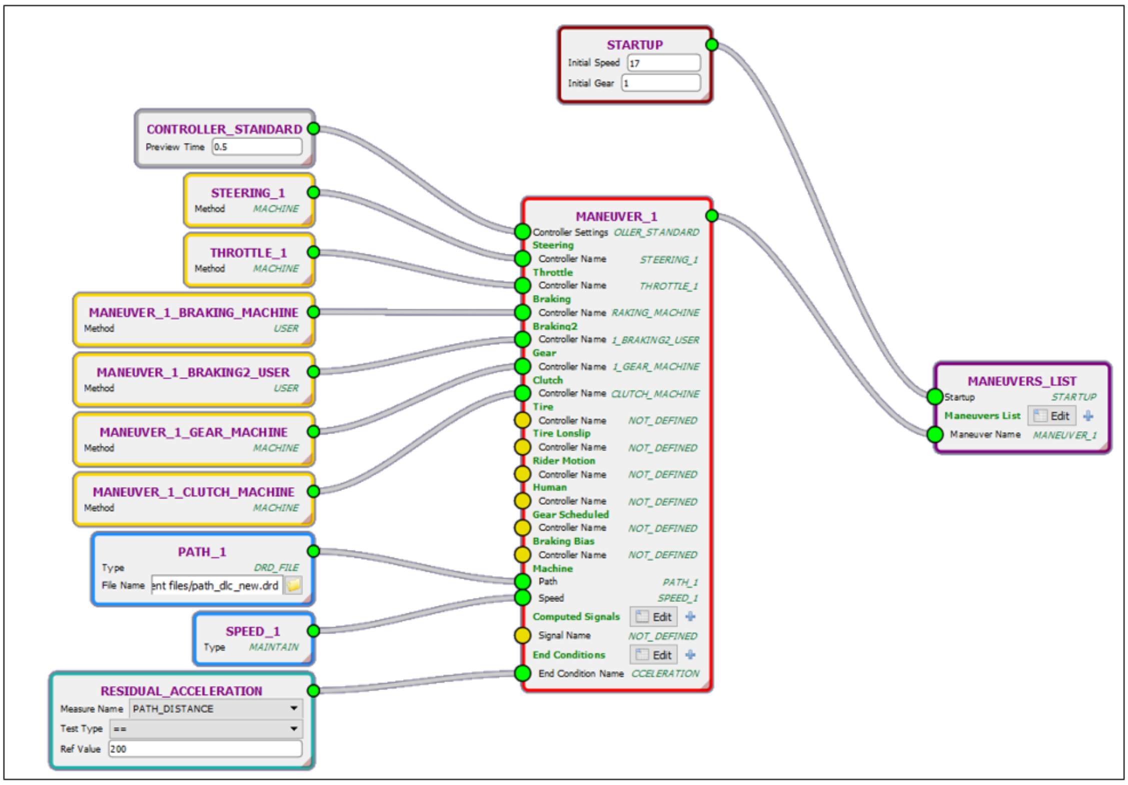

Figure A.8: Event file for Double Lane Change maneuver - VI-Event Builder

VI-CarRealTime Simulations - Planning Spreadsheet

				Type of suspension		Type of road			Speed (km/h)				Manoeuvers						Parameter varied					Assessment points/parameters that will be studied				
Sl No.	Name	Status	Comments	Passive suspension	Active suspension	Flat road	Uneven road - Class D	Stochastic bumps	40	60	80	Other	Drive straight	Double Lane Change	Acceleration	Braking	Steady State Cornering	Other	Roll Center Height	Anti-lift and Anti-dive	ARB Removed	Bumpsteer	Other	Pitch	Body roll	Damper force request	Steering demands	Vertical acceleration
1	Sim-1	Completed	Baseline simulation	■			■		■				■											■	■	■	■	■
2	Sim-2	Completed	Baseline simulation		■		■		■				■											■	■	■	■	■
3	Sim-3	Completed	Baseline simulation	■	■								■											■	■	■	■	■
4	Sim-4	Completed	Bumps	■	■			■	■				■											■	■	■	■	■
5	Sim-5	Completed	Bumps	■	■			■		■			■											■	■	■	■	■
6	Sim-6	Completed	Bumps	■	■			■			■		■											■	■	■	■	■
7	Sim-7	Completed	Bumps, RCH variation	■	■			■	■				■						■					■	■	■	■	■
8	Sim-8	Completed	Uneven road, Bumpsteer variation	■	■		■		■				■									■				■	■	
9	Sim-9	Completed	Bumps, Bumpsteer variation	■	■			■	■				■									■				■	■	
10	Sim-10	Completed	Uneven road, Antis removed	■	■		■		■				■							■				■				■
11	Sim-11	Completed	Bumps, Antis removed	■	■			■	■				■							■				■				■
12	Sim-12	Completed	DLC Baseline	■	■	■				■				■											■	■	■	
13	Sim-13	Completed	DLC Baseline	■	■	■					■			■											■	■	■	
14	Sim-14	Completed	DLC, RCH variation	■	■	■				■				■					■						■	■	■	
15	Sim-15	Completed	DLC, RCH variation	■	■	■					■			■					■						■	■	■	
16	Sim-16	Completed	DLC, ARB removed		■	■				■				■							■				■	■	■	
17	Sim-17	Completed	DLC, ARB removed		■	■					■			■							■				■	■	■	
18	Sim-18	Completed	DLC, RCH variation, ARB removed		■	■				■				■					■		■				■	■	■	
19	Sim-19	Completed	DLC, RCH variation, ARB removed		■	■					■			■					■		■				■	■	■	
20	Sim-20	Completed	Acc baseline	■	■	■						■			■									■		■		■
21	Sim-21	Completed	Acc - Antis removed	■	■	■						■			■					■				■		■		■
22	Sim-22	Completed	Braking baseline	■	■	■						■				■								■				■
23	Sim-23	Completed	Braking - Antis removed	■	■	■						■				■				■				■				■
24	Sim-24	Completed	SSC, Baseline simulation	■	■	■						■					■								■			
25	Sim-25	Completed	SSC, ARB removed	■	■	■						■					■				■				■			
26	Sim-26	Completed	SSC, RCH Variation	■	■	■						■					■		■						■			
27	Sim-27	Completed	SSC, AR removed, RCH variation	■	■	■						■					■		■		■				■			

Figure A.9: Simulations planning spreadsheet