ADDIS ABABA UNIVERSITY

ADDIS ABABA INSTITUTE OF TECHNOLOGY

DEPARTMENT OF MECHANICAL & INDUSTRIAL ENGINEERING

DYNAMIC ANALYSIS WITH MULTI-BODY SIMULATION OF ARTICULATED PASSENGER CAR OF AALRT

A Thesis submitted in partial fulfillment of the Requirements for the Degree of Master of Science in Mechanical Engineering in Graduate Studies of Addis Ababa University

By: BELAYHUN EJIGUALE

Advisor: Dr.Ing. DemissAlemu

July 2017

Declaration

I hereby declare that the work, which is being presented in this thesis entitled "**Dynamic Analysis with Multi-Body Simulation of Articulated Passenger Car of AALRT**" is original work of my own and currently happen problem in AALRT which has not been presented for a degree of any other university and all the resource materials which mentioned in the reference have been duly acknowledged.

Belayhun Ejiguale ________________ __________

Signature June, 2017

Approval sheet

ADDIS ABABA UNIVERSITY

ADDIS ABEBA INSTITUTE OF TECHNOLOGY

SCHOOL OF MECHANICAL AND INDUSTRIAL ENGINEERING

Title:

Dynamic Analysis with Multi-Body Simulation of Articulated passenger car of AALRT

By

Belayhun Ejiguale

July, 2017

Approved by Board of Examiners:

Dr.Ing. DemissAlemu	______________	______________
Advisor	Signature	Date
____________________	______________	______________
Internal Examiner	Signature	Date
____________________	______________	______________
External Examiner	Signature	Date
____________________	______________	______________
Railway Center Head	Signature	Date

Acknowledgement

First of all I would like to express heartfelt sincere gratitude to my advisor D.r Ing. Demis Alemu next to God for his follow up, guide, invaluable advice and technical support in strengthening during the course of my paper work. In fact, this thesis would not have come in its present form without his proper follow up in reshaping and organizing of ideas.

Secondly, I would like to express heartfelt thanks for progress examiners such as ato Arya and ato Hiredin to give fruit full comments, sharing ideas to address the stated problem. Thirdly I would like to thank Ethiopian Railway Corporation (ERC) kality depot administration and technicians for their appreciable help and technical support and AAIT and all stuff of Department of Mechanical and Industrial Engineering because of their great support in completing this paper.

Fourthly, I would like to acknowledge, Addis Ababa University& Ethiopian Railway Corporation for giving to me the chance to study masters of degree and their kindness and help during those academic years.

Lastly but not the least I would like to thank my wife and my friends Tesfaye Atalay and Dula Fkadu for their encouragement, and support with different materials and ideas. Without their guidance, help and patience, I would have never been able to accomplish the work of this paper.

Abstract

The railway vehicle running along a track is one of the most complex dynamic systems in railway engineering and it has many degree of freedom, the interaction `between wheel and rail, suspension and articulation systems involves complex systems. However, thevehicle body lowering down to the rail at maximum load condition was critical issue for the corporation. To identify the stated problem, dynamic analysis with multi-body simulation of articulated passenger car is crucial by SIMACK MBS software package. The 3D modeling and the simulation process pass through three steps by using this multi-body simulation software package. The Simulation results were the predictions of vertical deformation of railway vehicle suspension and articulations on straight and curved track can be made. The primary suspension, secondary suspension and articulation systems vertical deformation were 5.25mm, 15.125mm, and 12.35mm in straight track and 17.125mm, 40mm and 22.125mm in curved track. In order to withstand the vehicle body lowering down problem, vertical primary and secondary suspension stiffness and damping value optimized. After optimization the primary and secondary suspension and articulation system vertical deformations were 15.975, 20.325 and 16.375mm. Therefore, the vertical suspension and articulation systems, maximum vehicle load, minimum track curve radius and wheel diameter variation have great effect for the current problem. To withstand this problem, the vertical suspension stiffness and damping value have to be optimized.

Key words: dynamic analysis, multi-body simulation, simpack, articulation, suspension, 3D model, simulation

Table of Contents

List of table

List of figure

Symbols and Abbreviation

MBS……………………………………………………….Multi-body-simulation

AALRT…………………………………………………Addis Ababa Light Rail Train

DOF……………………………………………………….Degree-of-freedom

3D………………………………………………………….Three dimensional

FMS…………………………………………………….. Flexible multi-body system

M_c, M_b, M_w ………………………………….……Mass of car body, bogie, wheel set

MDOF………………………………………………….. Multiple-Degree of-Freedom

X, Y, Z. …………………….…... Longitudinal, lateral, vertical directions

Ø, β, Ψ……...................................... Roll (x-rot), pitch (y-rot), and yaw (z-rot) positive, right hand rule

$\ddot{X}$, $\ddot{Y}$, $\hat{Z}$ …………………….……Acceleration along longitudinal, lateral, vertical direction

F_{yAB},F_{yBC}………………………….. Internal forces of car body articulations lateral direction

F_{zAB}, F_{zBC}…………………….…..Internal forces of car body articulations vertical direction

F_{yA}, F_{yB}, F_{yC}.................................... Mutual forces of car body along lateral direction

F_{zcA}, F_{zcB}, F_{zcC}……………….……. Mutual forces of car body along vertical direction

g……………………………………Gravitational acceleration

Φ_{seci}……………………………….... Car body deflection along lateral direction

F_{xccpi}, F_{ycpi}, F_{zccpi}…………………….External excitation forces

I_x, I_y, I_z………………………….Movements of inertia of car body

M_{yA}, M_{yB}, M_{yC}………….……. Mutual movements of car body lateral directions

M_{zA}, M_{zB}, M_{zC}……………….. Mutual movements of car body vertical directions

M_{yAB}, M_{yBC}, M_{zAB}, M_{zBC} ……..Internal movements of articulations along y and z directions

M_{xcpi}, M_{yccpi}, M_{zccpi}……………….……. External excitation movements of car body

I_{yb}, I_{zb}……………………………..… Movements of inertia bogie along x, y, z directions

F_{ys1}, F_{ys2}, F_{ys3}, F_{zbA}, F_{zbB}, F_{zbC}………Mutual forces of car body along y and z directions

F_{y1}, F_{y2}, F_{y3}, F_{y4}, F_{y5}, F_{y6}, F_{zw1}, F_{zw2}, F_{zw3}, F_{zw4}, F_{zw5}, F_{zw}.......Internal forces between bogies and wheel sets along y and z directions

M_{yb1}, M_{yb2}, M_{yb3}, M_{zbmf1}, M_{zbmf2},M_{zbmf3} ………Mutual movements of bogies along y, z directions

M_{yw1}, M_{yw2}, M_{yw3}, M_{yw4}, M_{yw5}, M_{yw6}, M_{zw1}, M_{zw2}, M_{zw3}, M_{zw4} ,M_{zw5},M_{zw6}..……Internal movements between bogies and wheel sets.

F_{xbcpi}, F_{ycbi}, F_{zbcpi}………………….... External excitation forces along y, z directions

M_{ybcpi}, M_{zbcpi} …………………..….. External excitation forces and movement along

I_{yw}, I_{zw}………………………………Movements of inertia along y, z directions

$Ø_w$, $β_w$, $Ψ_w$…………………………. Angular acceleration of wheel sets along y, z directions

F_{ywmi}, F_{zmwi}……………………..…..Mutual forces of wheel sets along y, z directions

M_{ywmfi}, M_{zwmfi}………….………… Mutual movements of wheel sets along y, z directions

F_{ywcfi}, F_{zcfwi}………….……………...Contact forces of wheel sets

F_{xwcpi}, F_{ywcpi}, F_{zwcpi}…………..…..... External excitation forces of wheel sets

M_{ywcfi}, M_{zwcfi}……………......…..Contact movements of wheel set

$M_{xwcpi,}$ M_{ywcpi}, M_{zwcpi}…………….…External excitation movements of wheel sets

C1, C2, K1 and K2 ……………...Damping and stiffness of primary and secondary suspension

CHAPTER ONE

1.1 General Introduction

Along the historical development of transportation system, rail transport dates back nearly 500 years and includes systems with man or horse power and rail of wood or stone [2]. These railroad transportations are the most safe, fast, mass transportations, high income generation, environmentally friendly and widely used methods of transporting passengers and goods in the world. Due to these and other advantages, Ethiopia Railway Corporation received 70% low floor rail vehicles with 3 body modules and 3 bogies (2 motor bogies and 1 trailer bogie) from Changchun Railway Vehicles co. Ltd [1] as shown in the following figure.

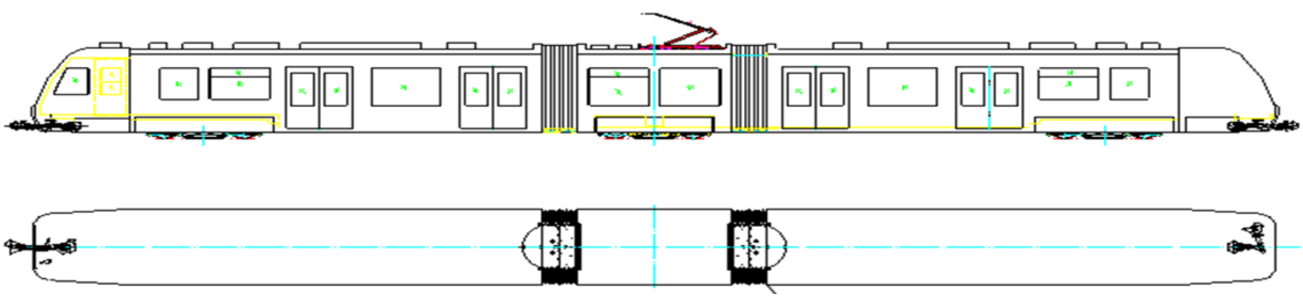


Figure 1. Side and top view reperesentation of passenger vehicles

This vehicle running along a track is one of the most complex dynamic systems in railway engineering and it has many degrees of freedom, the interaction between wheel and rail involves both complex geometry of wheel tread and rail head [2]. However, this vehicle body lowering down to the rail and damage the gangway and ballice were taken while the peak time of operation. The following figure shows currently occurred problem.

Figure 2. Gangway system folding bellows lower dawon to the rail.

This vehicle body lowering down problem is more related to vertical suspension system deformation, wheel diameter variation, maximum vehicle load effect and articulation folding bellow attachments. To solve the stated problem, they used 10mm washer for the secondary suspension systems to keep the acceptable height from top of rail but while increasing the number of washer to get the allowable height, additional problem occures for the secondary spring suspension suport.The previous research's, journals and text books studied two bogie railway vehicles. The stated problem occurs on three bogies with three car body modules by connecting articulation systems. So that, the three bogies and the three car body module articulated passenger vehicles at maximum vehicle load, wheel diameter difference, critical speed and vertical suspension stiffness values are not studied. The overall components, force elements, rigid and flexible bodies need multi-body simulation technique to address the objective of the research.

1.2. Hypothesis

The vehicle body lowering down problem may arise fromvertical deformation of primary and secondary suspension systems, flexiblity of articulation systems, vertical flexibility of connecting systems, variation of wheel set lateral movement, maximum passengers load, the loosen tight of the rope attachment of folleding belows and the rapid wear of wheel flangeto lead the lowering down of vehicle body to the rail. Increasing the number of washers lead for secondary suspension minimum suport may cause for high risk of deriliment, while the vehicle running along the curved track with maximum passenger load.Based on the fact that, the vertical suspension deformation, maximum load, wheel diameter variations and articulation systems take the great role for the stated problem. Therefore, dynamic analysis with multi-body simulation by using SIMPACK is crucial.

1.3. Problem Statement

Railway vehicles running along two guided rail is one of the most complex dynamic systems and it has many degrees of freedom and difficulties to solve dynamic problem. The non-conservative forces generated by relative motion in the contact area and there are many non-linearity's between wheel and rail, primary and secondary suspension on the car body and articulation systems. In addition to that, these vehicles running along a minimum curve radius also generate additional difficulty. Due to this and other reasons AALRT passenger rail vehicle body was lowering down to the rail and damage the gangway and ballice while the peak time of operation was taking place before reaching the acceptable limit of wheel diameter. For detail understanding of the problem, see the following figure 3.

Figure 3. Vehicle body Lowering down and damaging of balliec from sleeper and damaging itself

This figure collected from the problem faced vehicle. For the lower right on the figure is that, they used to minimize the lowering down problem.

1.4. Objectives

1.4.1 General objective

The general objective of this research paper is devoted to dynamic analysis with multi-body simulation of articulated passenger car of AALRT by SIMPACK software package.

1.4.2 Specific Objective

- Determine the vertical deformation of primary and secondary suspension at maximum vehicle load in strait and curved track condition
- Determine fixed lower front and rear articulation vertical deformation in straight and curved track condition.
- Determining the vehicle mass variation on vehicle body vertical deformation under parameter variation condition.
- Showing the effect of wheel radius variation on vehicle body vertical deformation.
- Determine the effect of lateral displacement of wheel set on vehicle body vertical deformation.
- To determine the car body and bogie vertical deformation while maximum load conditions.
- Determine the dynamic performance analysis on minimum horizontal curved track conditions

1.5. Methodology

In order to identify the stated problem, dynamic analysis and multi-body simulation of articulated passenger car of AALRT by using SIMPACK Multi-Body Simulation software package have been carried out. Therefore, the methodology to address the stated problem as follows.

- Stating the function of vertical primary and secondary suspension which supporting vehicle body.
- SIMPACK 3D model made based on six DOF but modeling the governing equation of motion were made for vertical, lateral and pitching dynamics based on three DOF.
- 3D modeling of the AALRT by using SIMPACK MBS software for each component and assembled the full car body module.
- Simulating this model under the following main parameter simulation technique for which affect the stated problem:
  - ✓ Primary and secondary vertical suspensions stiffness and damping value.
  - ✓ Mass of vehicle load for maximum conditions.
  - ✓ Wheel diameter from new wheel(660mm) to lower limit of (595mm)
  - ✓ Critical speed variation

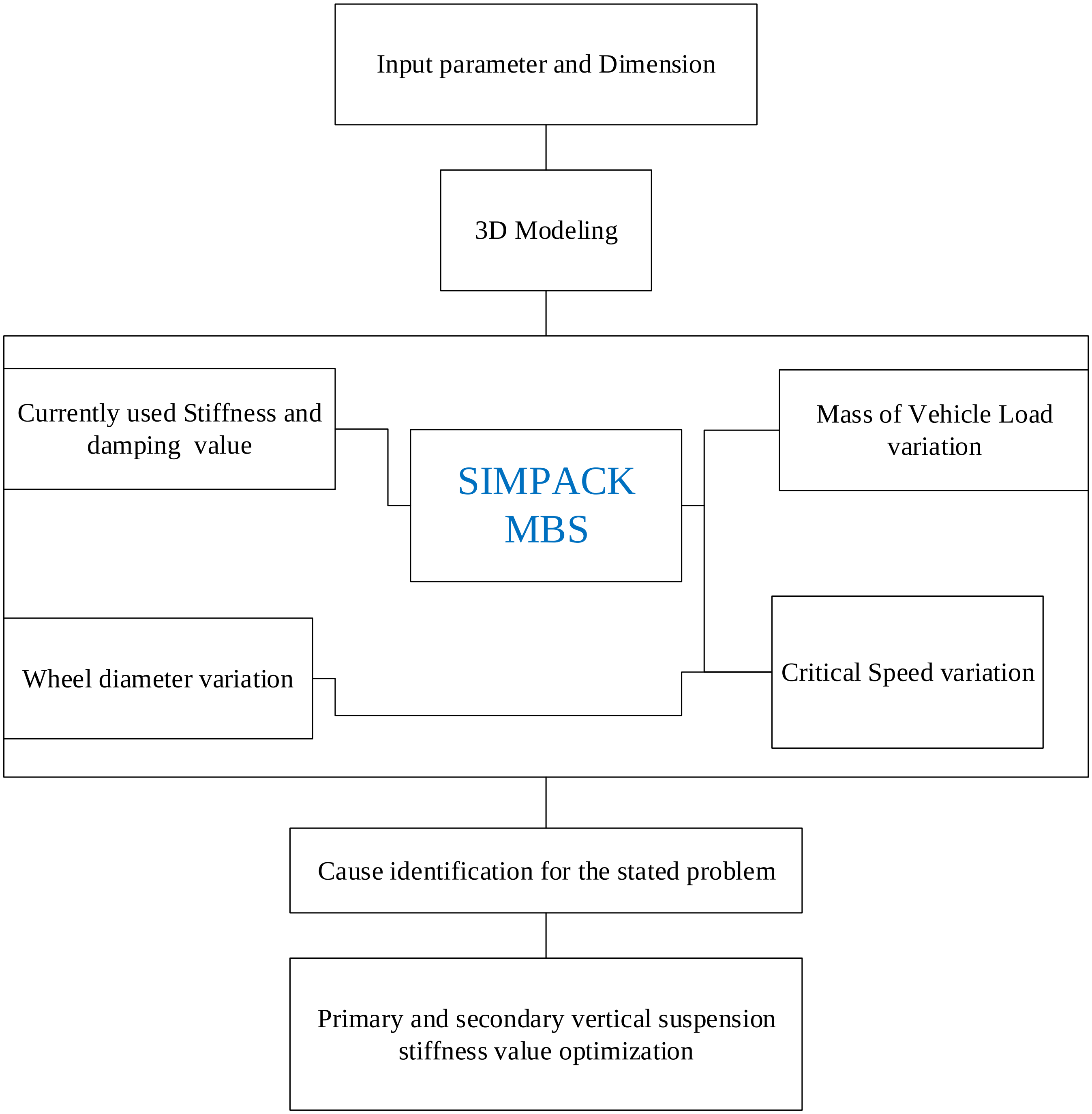


Figure 4. Methodology justification

This paper uses to achieve the objective of the dynamic analysis with multi-body simulation of passenger car of AALRT to identify the current problem by using AALRT data. The area of this paper mainly focuses on AALRT project to identify the lowering of the vehicle body and the articulation folding bellows towards the rail.

1.6. Software Validation

There are modeling and dynamic analyses programs that have received wide acceptance in recent years include [20]:

- NUCARS
- MEDYNA
- VAMPIRE
- SIMPACK
- ADAMS/RAIL
- GENSIS

Although the above programs have different attributes, they were all developed specifically for railway vehicle dynamic modeling and simulations [20]. Each program includes different solution methodology, wheel/rail models, analysis methods and user interface. Among those software, we had get training on SIMPACK MBS software and in addition to had got the training it has the following advantages. It is a general Multi-Body Simulation Software used to model the dynamic behavior of interconnected rigid or flexible bodies that are subjected to translational, rotational and displacements [20]. While single bodies or parts of a mechanical system are studied in detail with finite element methods, the behavior of the whole Multi-Body systems are studied with Multi-Body Systems and this enables engineers to generate and solve virtual 3D models in order to predict and visualize motion, coupling forces and stresses [20]. The main advantage of this software is that it is adapted to full transient non-linear analyses, even within the acoustic range, being particularly outstanding in transient high frequency analysis [20].Due to this and others important features, SIMPACK- MBS software package is used to achieve the objective of the research.

Building the passenger railway vehicle model by SIMPACK graphical user interface and realization of the simulation is achieved by following three main steps [20]:

A) Pre-Processing

B) Processing

C) Post-Processing

Based on this paper, the following process is explained separately:

A. The **pre-processing** module is used for building the MBS together with its corresponding3D geometry. Some of the available features from this menu are as follows; Open Model, Model set up, MBS Define body, MBS Define Joint, Define constraints, Force Elements, Define sensor, vehicle global, track definition, information, element menu and so on. In this MBS model window, each body mass, center of mass, movement of inertia, marker point, joint state, DOF, force element, force type, speed, track and wheel type, track irregularities and other important criteria's were justified.
B. **The processing** module mostly contains calculations and parameter variation process to allow the user to configure and perform different calculations and Para variation process. Assemble System, Nominal Force Parameters, Eigenvalues, Time Integration and Measurements, perform time domain, kinematic, Eigen frequency, critical
C. **The post-processing** step deals with the final result plotting and extracted data from the simulation. The SIMPACK program offers detailed results for each body forming the multi-body model in the railway vehicle. The output shows the Para variation time domain 2D plots, G2D plots and state plot.

SIMPACK allows the results of calculations to be animated in 3D in either the SIMPACK Post Processor or the 3D Model Setup window.

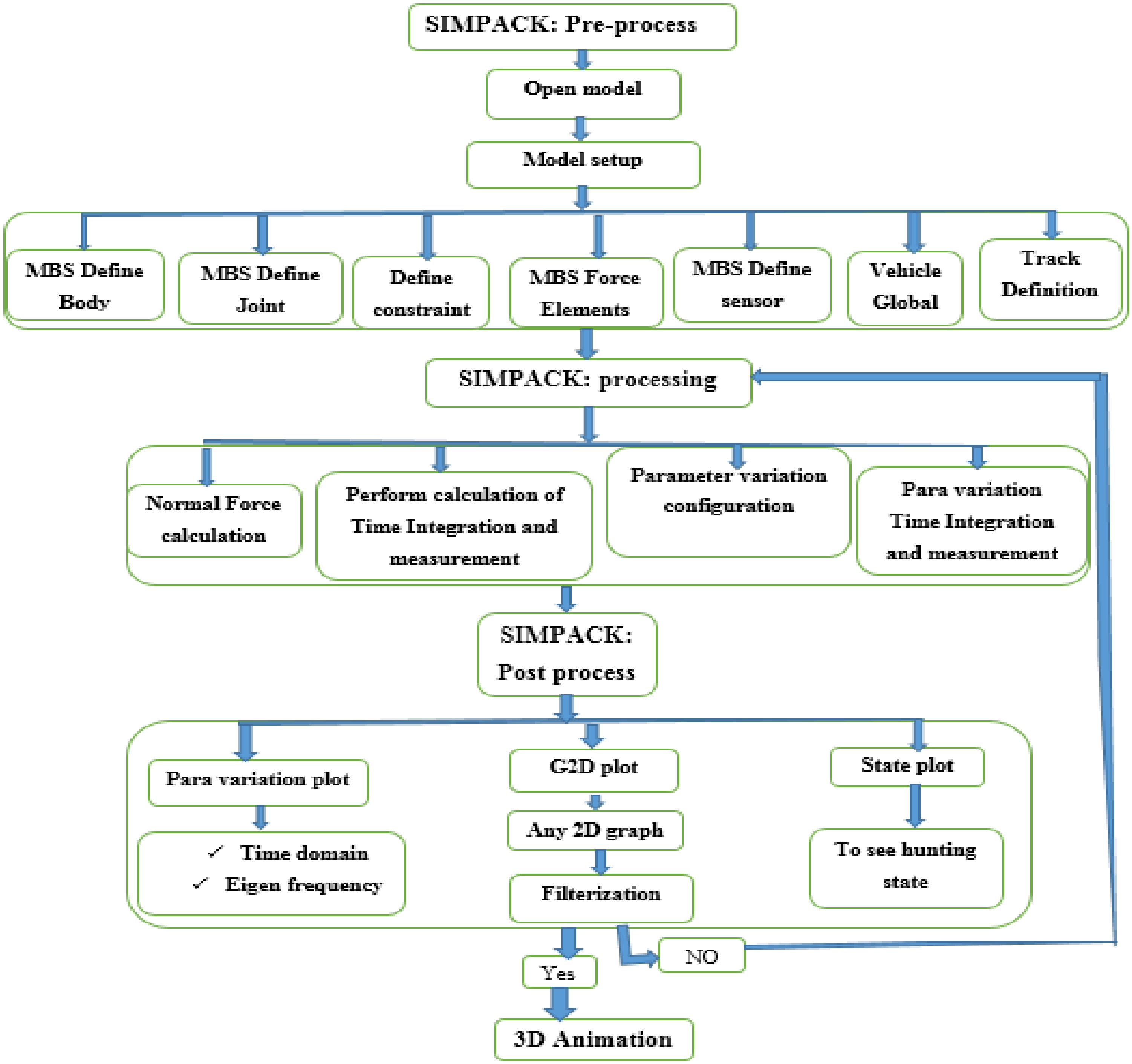


Figure 5. Simpack AALRT passenger vehicle model algorism

1.7. Scope or Delimitations

The passenger railway vehicles running along a track is one of the most complex dynamic systems in railway engineering. Therefore, this paper mainly focuses on the dynamic analysis with multi-body simulation of AALRT passenger car to identify the lowering down of the car body. The following points may be considered:

- The three car body module and the three bogie frame are considered as rigid body frame.
- Designing of folding bellows is not included
- Wire rope attachment is not included
- Simulation by using secondary suspension spring washer is not included
- Track flexibility and Track irregularity are considered for external excitation, because track flexibility and irregularity are not avoided. Forces which generated by centrifugal and centripetal effect while vehicle passes through curved tracks are considered.
- Track super elevation and curving effect are considered.
- Effect of wind, aerodynamics and braking are not taken in to account in the current study but they do influence on the rail vehicle vertical dynamics.

1.8. Significance of the Study

The major merits of the study energies the railway academy future research center to investigate the real cause of the vehicle body lowering down problem. Since the cause of the stated problem very vast, this study focus on the vertical deformation of primary and secondary suspension and articulation systems in straight and curved track condition by using fully assembled three bogie passenger car body modules by using SIMPACK MBS software package to get detail solution. This helps to identify the stated problem and make the right decision for further researchers.

1.9. Thesis organization

The chapter of this paper organized to guide the reader from theoretical background to final results. **Chapter two** reviews about the dynamic analysis of articulated passenger vehicles, lateral displacement of railway vehicles, vertical displacements, curve negotiation of railway vehicles and curved supper elevation track. **Chapter three** provide general concept about railway dynamics, articulated passenger car system components and joints, degrees of freedom, theoretical concept and mathematical modeling of lateral, vertical, pitching and vertical suspension systems. And also states the specification, standards and dimensions of passenger railway vehicles, 3D modeling of system components of vehicle body parts and fully assembled body. **Chapter four** provides the simulation of the 3D modeling according to the methodology and parameter variation technique carried out. The result of those simulations presented accordingly. **Chapter five** states the result discussion, validation, recommendation, future work and the conclusion of the whole results of the simulation.

CHAPTER TWO: LITERATURE REVIEW

2.1. General review of railway dynamics

The railway vehicle running along a track is one of the most complex dynamical systems in railway engineering [2, 3]. It has many degree of freedom, the interaction between wheel and rail involves both complex geometry of wheel tread, rail head and non-conservative forces generated by relative motion in the contact area and there are many non-linearity's [2, 3]. The force acting between the wheel and rail, the inertia forces and the forces exerted by the primary and secondary suspension and articulation systems for each adjacent car body. However, for the lowering down of vehicle body problem were not included in those references. This vehicle body lowering down to the rail and damage the gangway and ballice while the peak time of operation is more related to dynamic effect on the vertical and lateral suspension system, the articulation systems and the folding bellow attachments. Therefore, railway vehicle vertical and lateral Suspension systems and articulated systems play an important role when it comes to the dynamic behavior.

This vehicle body is supported by three bogies, lower and upper articulation, primary and secondary suspension components. The suspension systems for this articulated passenger railway vehicles are two-stage suspension systems such as: primary and secondary suspension systems. In the 3D vehicle model presented, the suspension components are treated as zero-mass elements and only force element but the articulation components have their mass.

2.2. Articulated passenger Vehicle on straight track

While the vehicle running along a straight track, the wheel set moves right to left due to wheel conicity effect. This leads rolling radius difference and this rolling radius difference makes vertical height difference between the two sides of car body. According to [2, 3] Klingel's, Carter's and Kalker's, theory of contact mechanism, the wheel set continuous oscillation in the lateral direction and to the center line continually while the operation is going on. This continuous lateral motion affects the vertical suspension due to weight transfer effects which leads to the lowering down of the passenger vehicle articulation folding bellows to the rail especially at minimum curve radius.

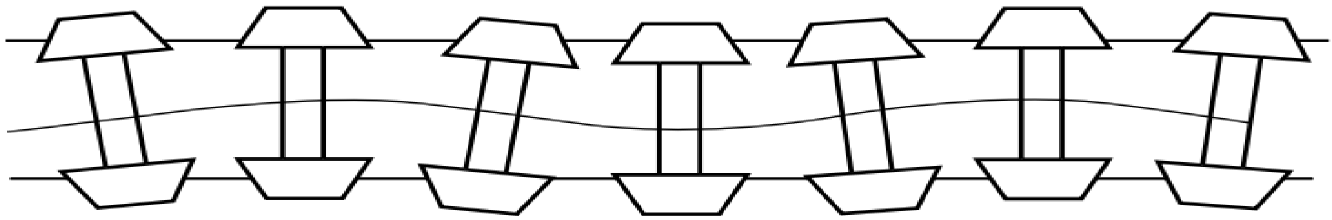


Figure 6. The kinematic oscillation of a wheel set [2, 3].

This conicity of wheels serve as a differential as an automobile which compensate the curve negotiation and lateral movement of wheel set. According to [3] the lateral motion caused by coning and the wheels are rigidly mounted on the axle, very slight errors in parallelism would induce large lateral displacements that would be limited by flange contact. This leads the maximum flange wear, which occure now in our railway project and lead to maximum lateral motion and vertical hight difference among the two sides and this occure continuously.

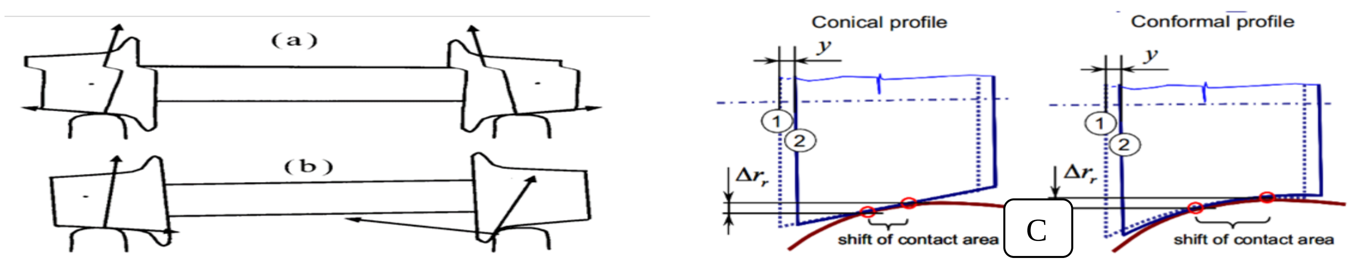


Figure 7. (a) In central position, (b) in laterally displaced position and flange rail contact [2] (c) Lateral and vertical motion of wheel set [7]

This vertical height difference occurs simultaneously while the vehicle running along the track. It has its own contribution for the lower down of the articulation folding bellows to the rail. To understand this, see figure.7.

2.3. Articulated passenger vehicles on Curved track

The passenger Rail vehicles during curve negotiation experience strong lateral influences depending on operational speed, curve layout, and track imperfections [3]. Even though the articulation systems maintain the effect of curving, the vertical suspension system deflects and lateral displacement of the car body relative to the track and weight transfer of the vehicle body towards the inner rail occurs.

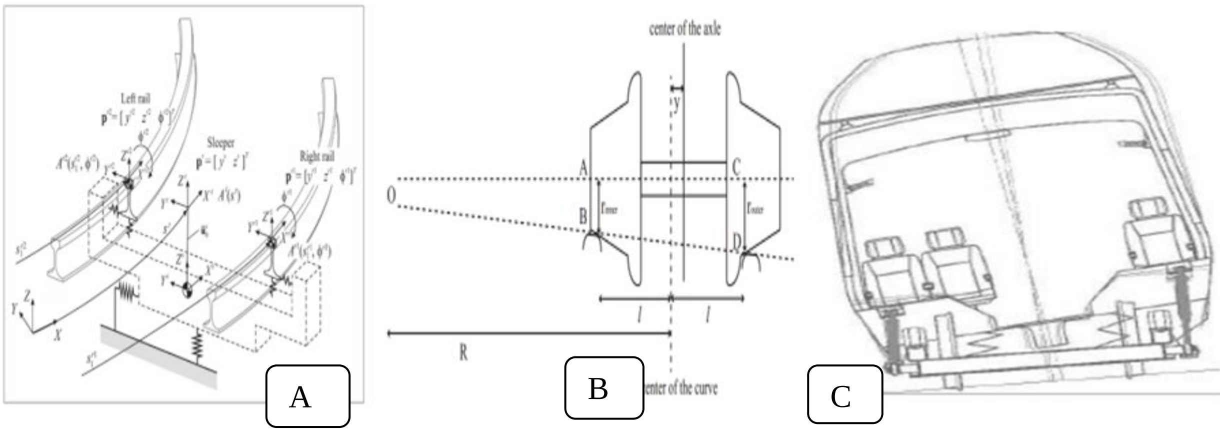


Figure 8. (A) Curved tack, (B) Geometry of a coned wheel set on a gentle curve (C) Vehicle curving effect on vertical suspensions [3, 7]

In curved track, the super elevation of the curved track alignments also characterized by the height difference between the tops of the rails, but can also be measured in terms of angle [3].

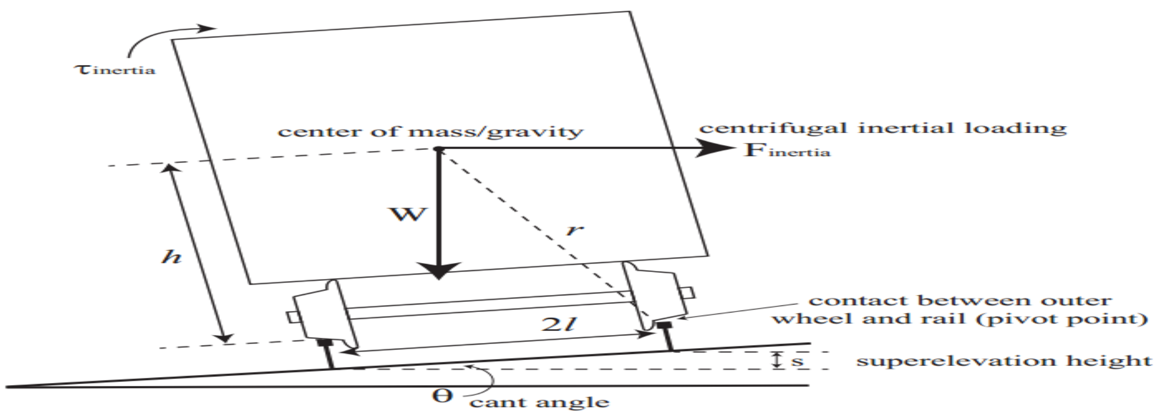


Figure 9. Railcar force diagram on a super elevated curve [3].

This reference mainly focus on critical speed analysis by considering primary suspension on curved and straight track condition but it did not consider secondary suspension and articulation systems. Due to the fact that the supper elevation curve cause maximum compression load on the primary and secondary suspension on one side and wheel unloading in other side. This is similar sense like the curved track but it has curving and super elevation effect on the vertical suspension to facilitate the lowering of vehicle body and articulation folding bellows to the rail.

2.4. Related literature review about the stated problem

[2] Studied the guidance and stability of rail way vehicles, modling equation of motions for two axle and thre axle bogie, discusing symetrical and unsymetrical vehicle model, hunting and creepage analysis and modeling of railway vehicle were made. This book used for general railway dynamic analysis and basic concept capturing.

[3] Studied Critical speed analysis of railcars and wheelsets on curved and straight track by considering primary longitudinal, lateral and vertical suspension stiffness and also focuce on Wheel set Geometry and the Role of the Conicity, Railcar Overturning and the Role of the Flange, On a Flat Curve, On a Super elevated Curve, Parametric Analysis for Speeds on Curved Track, Kinematic Analysis of the Wheel set on Straight Track, Lateral Displacement and Axle Yaw Oscillation, Critical Speed for the Onset of Hunting Oscillation, Flange Thickness and Conicity. Therefore, this reference did not consider secondary longitudinal, lateral and vertical suspension and articulation system.

According to [4] the Multibody approach for dynamic analysis of railway vehicles mainly focuse on Multi-body system analysis, wheel set dynamic analysis, wheel contact model by two bogie vehicle model by two steps. The first step is making a comparison between the results of the simulation tool and the results obtained by different simulation packages used to analyze the dynamic behavior of the Manchester Benchmark vehicle; the second step is making a comparison between the analysis and results obtained by the developed simulation tool in his work and the results obtained by the commercial simulation package SIMPACK for TGV001 locomotive vehicle in different operation scenarios. But the vertical suspension and articulation systems was not considered.

Reference [5] studied active suspension of rail way vehicles to replace the existing passive suspension system. That active suspension divided into four areas: which passive components that were replaced with active, where the active components were placed, with control strategy that is used, and which active components, such as sensors and actuators. In this reference not only the passive suspension but also articulation system was not considered.

[6] Studied on Active Vertical Secondary Suspension for Passenger Trains to improve vertical ride comfort by two different control strategies had been studied by sky-hook damping and the multi-variable control using first a quarter-car and then a full-scale vehicle model. Secondly, an active vertical secondary suspension system (AVS) was developed, using simulations. Dynamic control of the vertical and roll modes of the car body, together with quasi-static roll control of the car body, show significant

vertical ride comfort improvements and allow higher speeds in curves. But the study was not including passive primary and secondary suspension effect on three bogie passenger articulated vehicles. The Characteristic of parameters on nonlinear wheel/rail contact geometry was Studied [7] especially for wheel conicity effect on lateral displacement of wheel set to make vertical radius difference and also the effect of wheel/rail contact non-linearity on railway vehicle dynamics. But this reference did not consider vertical suspension and articulation systems.

The Analysis of Rail Vehicle Suspension Spring with Special Emphasis on Curving, Tracking and tractive Efforts was studied by [8] and also studied the lateral force on the track. But this reference was not cover the full three bogie analysis. [9] Focus on the Methods for Reducing Vertical Car body Vibrations of a Rail Vehicle, car body dynamic and ride comfort. A study performed by Suzuki gives a survey of different ride comfort evaluation approaches applied in Japan and includes secondary air suspension by two bogie railway vehicles. [10] Studied on Vertical Vehicle/Track Interaction, Lateral Vehicle/Track Interaction, and lateral and vertical track excitation under different consideration of railway vehicle parameters. But the suspension system and articulation system components were not studied. [11] The studied document in that report is to provide, to the extent practical, detailed engineering data on a representative sample of rail road passenger car trucks and car bodies recently being used, or likely to be used. And also the detailed definition railway vehicle was presented. In this reference [12] the Multi-body modelling of a multi-articulation tramway vehicle studied the influence of rail-excitations on forces acting in the yaw dampers simpack and mathlab. It also studied Railway Vehicles Dynamics, rail excitation, contact force and composition tram articulated railway vehicles. But the tram car vertical suspension and tram car articulation system did not cover in this reference. [13, 14, 15] Study on the Dynamic forces between the rails and the wheels of railway vehicle, detail analysis of the standards of UIC code 518 and EN 14363-2005. The lateral and vertical forces, between wheel and rail and vertical acceleration of bogie and car body which mentioned in table 4 were taken from those reference. The reference [16] studied the optimization of a high speed train bogie primary suspension, rail wheel pair, safety, derailment coefficient analysis and dynamic simulation by simpack MBS of two bogie vehicles. But in this reference secondary suspension did not included.

CHAPTER THREE

3.1 Dynamic analysis of passenger railway vehicles

The dynamic analysis of articulated railway vehicles consists in the study of their motion as a response to external applied forces and moments [2]. The modeling and simulations of the system in th field of railway dynamics is a complex interdisciplinary topic and the necessity for the enhancement of the performance of the passenger railway vehicles and obtaining more safety, reliability and comfort conditions. The vertical primary and secondary suspensions and articulation systems are also more complex discipline in the railway dynamics.

The mathematical modeling of this passengerrailway vehicles in three degrees of freedom such as the two translation and one rotation along lateral, vertical, bounceand vertical suspensions are the most principalobjectives of the stated problem. This analysis also provides a process to estimate vertical position of primary and secondary suspensions, the three car body modules, the three bogies and front and rear fixed lower articulation systems.Also external forces which generated as a consequence of the system interaction with the surrounding environment, such as contact and curving effects for the suspensions are considerd. This multi-body system consists of varities of system components which afect the dynamic systems of the railway vertical suspensions to lead the lowering down problem of the vehicles.This assemble multi-body system can be defined as an assembly of many rigid or flexible bodies joined together by kinematic joints and/or force elements. A joint in this multi-body system permits certain DOF of the relative motion and restricts or prevents others, based on the joint type and the marker element. The combination of each bodies and the motion with respect to each other forming the multi-body systems are controlled by the kinematic joint type. On the other hand, force elements such as primary and secondary suspension systems can be used to interconnect internal forces between wheel set and bogie and bogie and car body. The wheels connected with the axles by fixed joint mechanisms, the wheel sets connected the bogie system by primary suspension elements and prismatic joint systems. The bogies connected the car body by secondary suspensions and center bolts.For the detail describtions, see the following schematic figure 10.

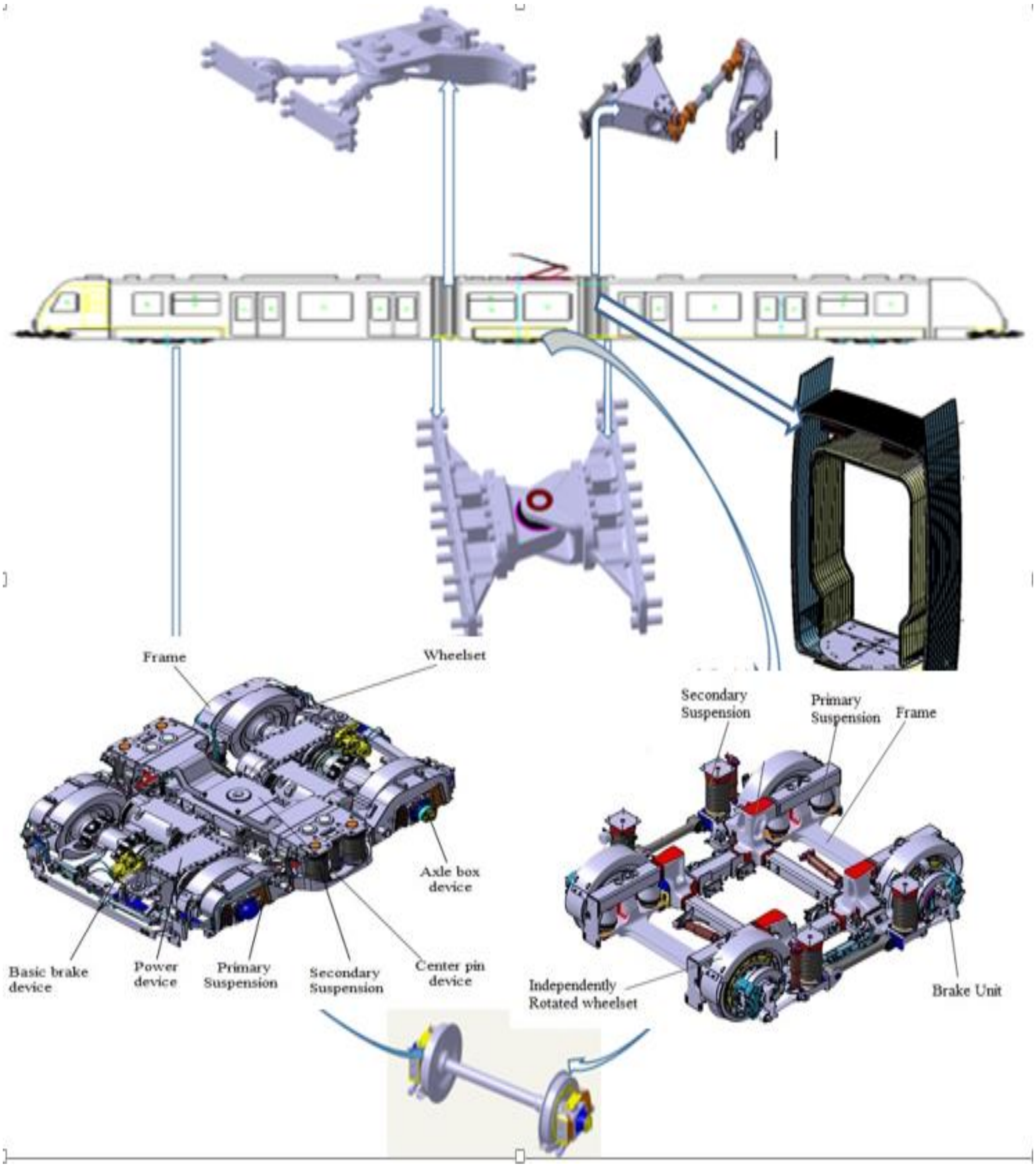


Figure 10. Main components of passenger vehicles system components

The three car body module connected by the articulation systems at lower and upper end of each adjecent car bodies. This Articulation device between each adjecent modules within vehicles should incorporate well-proven structure that consists of articulation unit and metallic spherical bearing. Metallic spherical bearing is motion node for articulation so that articulation can have more flexibility [1]. Damper systems are installed at both sides of lower articulation to cushion the vibration in longitudinal, vertical and lateral directions [1].

The lower hinge systems are also called as the fixed hinges, the core component is the middle rotating node module, and it is connected with the car body through two mounting seats. Various mounting seats are connected to the end surface of a module with bolts. The upper articulation of the free hinge includes the control console, connecting pipe, bearing receiver and installing parts. The center point of the upper hinge system must be aligned with the center point of the lower hinge system. For this reason, vertical fixing shall be conducted at the center point of the upper hinge system of the main body of the compartment with the appropriate equipment. The car bodies connected with the bogies at the center of the bogie by center bolts, traction rode and secondary suspension system. The bogies and the wheel sets are connected by the primary suspension in order to give ditail enfacices, let us plot the two dimensional plot of the three car body module with three bogies of the six wheel sets. Bogie one and bogie three have the same mass and dimantion. The primary and the secondary suspension stiffnes and suspension element dimentions are the same. See the following figure. But the mass of the middle bogie differ with the two and have the same primary and secondary suspension stiffness.

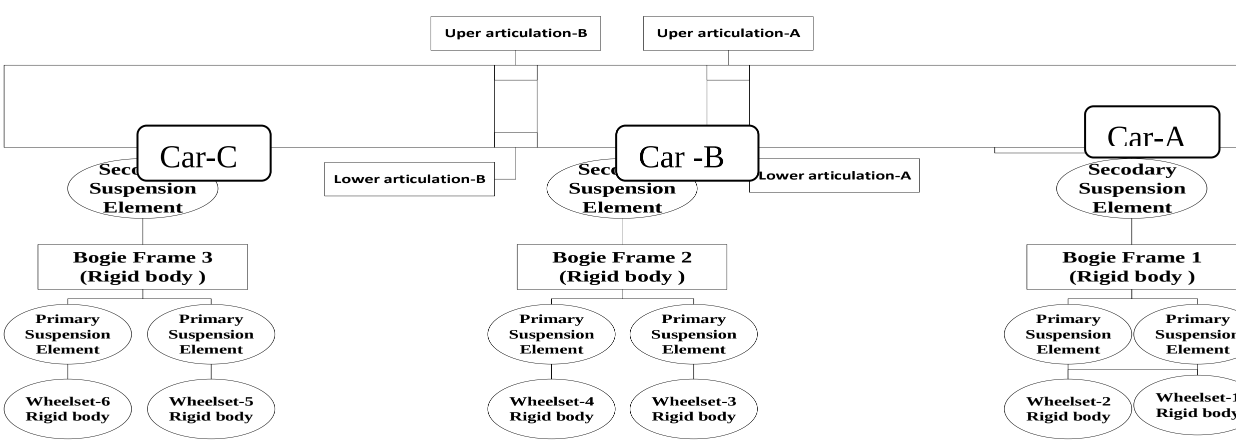


Figure 11. Schematic diagram of passenger rail vehicle

3.2. DOF for the three bogie passenger rail vehicle

Degrees of freedom are the ways in which the space configuration of a mechanical system may change, i.e. the independent movements the system can possibly undergo. Degrees of freedom are also independent displacements and/or rotations that specify the orientation of the body or system [2, 3, 4].

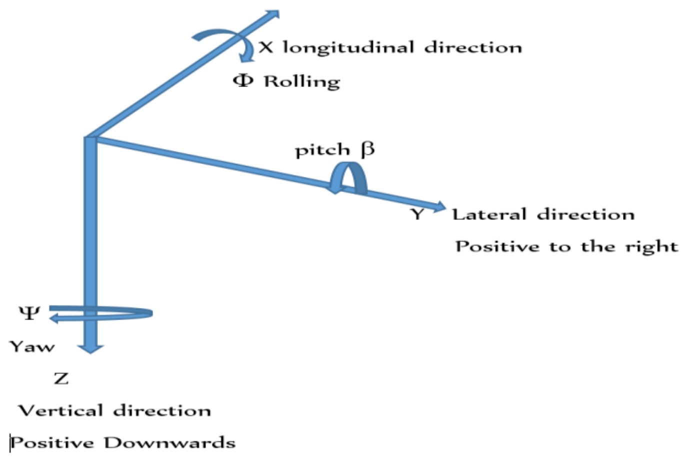
In reality, the passenger railway vehicles are not built of separate mass points, but consist of a continuous connected mass also called distributed mass. Such systems have an infinite number of degrees of freedom. However, it is virtually impossible to find dynamical solutions to any system. In general, it is necessary to discretize such systems, i.e. replace infinite-number-of-degrees-of-freedom systems with simplified models finite-number-of-degrees-of-freedom systems which are also called Multiple-Degree of-Freedom (MDOF) systems[4].Each car bodies, bogies, wheel sets and articulations components have three transelation and three rotation degrees of freedom, which are the longitudinal, lateral, vertical, rolling, pitching and yawing directions.

Table 1. Degree of freedom of articulated passenger rail vehicle

No	Items	Degrees of freedome	Quantity	Total DOF
1	Car body	6	3	18
2	Bogie	6	3	18
3	Wheel set	6	6	36
4	Uper flexible articulation B	6	1	6
5	Uper free articulation	6	1	6
Total				84

The number of degrees of freedom of a mechanical system is equal to the minimum number of independent coordinates required to define completely the position of all parts of the system (configuration of a mechanical system) at any instant in time.

Each mass with in the system has six degrees of freedom corresponding to three displacements (longitudinal, lateral and vertical) and three rotations (roll, pitch and yaw). For a conventional bogie vehicle, there are eighteen main masses (three vehicle bodies, three bogies, six wheelsets and two articulation) and therefore a total of 84degree of freedom.



Name	Direction	Positive
X	Longitudinal	For ward
Y	Lateral	Right
Z	Vertical	Downward
$\O_b$	Roll (X-rot)	Right hand rule
B	Pitch (Y-rot)	Right hand rule
Ψ	Yaw (Z-rot)	Right hand rule

(a) (b)

Figure 12. Degree of freedom (a) by figure and (b) by table

For the stated problem, vertical, lateral and bounce have high contribution for the problem. Therefore, this paper mainly focus for the three DOF especially for vertical suspension in related to articulated (gangway coupling mechanism) for the lowering problem.

3.3. Passenger railway vehicles Lateral dynamics

The lateral dynamics of passenger railway vehicles on which 3D coupled full vehicle–structure have its own influence for the vertical suspension and articulation systems. This three bogie with three car body (A, B, C) articulated passenger rail-vehicle systems lateral dynamics is strongly influenced by the interaction forces and moments which occurs in the wheel-rail contact regions, that affect the suspensions and articulation systems. when the wheel set moves lateraly in the Y direction, the rolling radius of the wheel changes due to wheel conicity effect. This leads hieght differencefrom top of rail to the car body with the two side of the car body which cause the suspensions highly compressed from the inner suspension to facilitate lowering down of the vehiclebody.These lateral movements affect the suspension system as well as the articulation structure which causes for lowring down of the vehicle. A fully nonlinear coupled model is proposed here for the lateral and vertical dynamic analysis of vehicles on the straight and sign change curved rail appearance. This articulated passenger rail Vehicles are considered as three-dimensional multi-body systems.

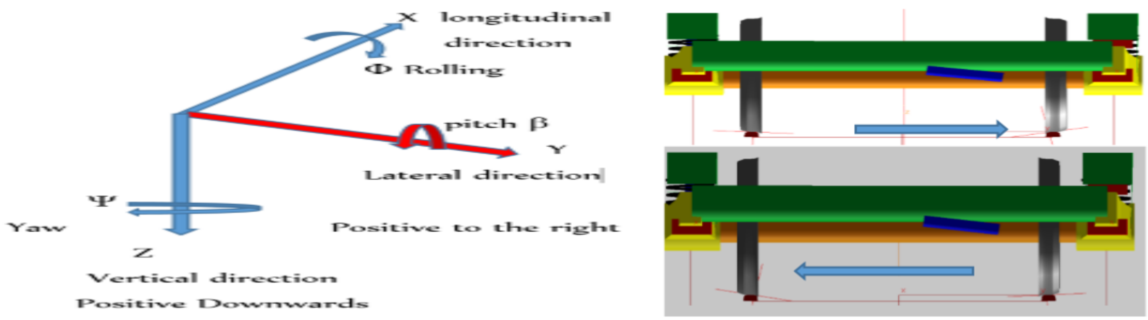


Figure 13. Degree of freedome along lateral direction

3.4. Passenger railway vehicle Vertical dynamics:

For the current problem of lowering down of the pessenger car body and articulation folding bellows, the vertical dynamics play an important roll. This articulated passenger car body separate components are rigid bodies' like wheels, axles, wheel set, bogie frame, articulation components and car bodies. The complete passenger vehicle model can be assembled by determining which masses and directions of motion are to be included, and then creating a corresponding set of assembled systems as well.

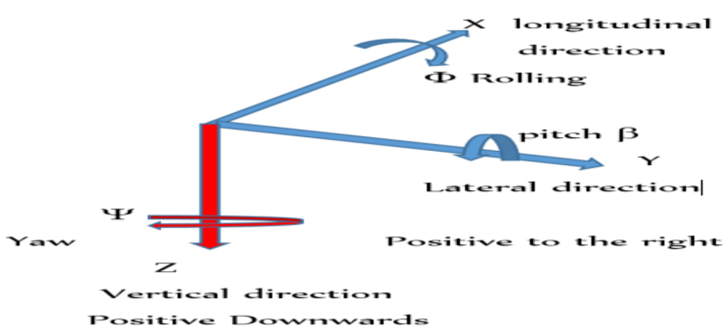


Figure 14. Vertical dynamic degree of freedome

While the vehicle running on a track, the axles make forced movements in a vertical plan, so that the plan of the axles in a bogie has a translation motion (bounce) and rotation motion (pitch).

3.4.1. Mathematical modeling of the equation of motion along lateral, vertical and pitch direction

The mathematical model of the lateral, vertical and pitch motion of articulated passenger vehicle have to be made for the car body, bogie and wheel set in a general manner based on Newton law of motion along lateral, vertical and pitching direction [2, 3, 4]. This vehicle passes through straight and curved track. Therefore, the following assumption has to be made. The mutual forces and movements, the inter component forces and movements, gravitational forces, contact forces and contact movements, external induced forces and movements due to centripetal and centrifugal effects while the vehicle negotiating curved track are considered. The effect of aerodynamics and braking are not considered.

Car body

$$\left.\begin{aligned} &M_{cy}\ddot{Y}y_c = F_{yA}+F_{yB}+F_{yC}-F_{yAB}-F_{yBC}+M_c g\,(\Phi_{sec1}+\Phi_{sec2}+\Phi_{sec3})+F_{ycp1}+F_{ycp2}+F_{ycp3}\\ &M_{cz}\hat{Z}_{cz} = -F_{zcA}-F_{zcB}-F_{zcC}-F_{zAB}-F_{zBC}+3M_c g_z+F_{zccp1}+F_{zccp2}+F_{zccp3}\\ &I_{yc}\beta_c = -M_{yA}-M_{yB}-M_{yC}+M_{yAB}+M_{yBC}+M_{yccp1+}M_{yccp2+}M_{yccp3} \end{aligned}\right\}\text{...........eq.1}$$

Bogie

$$\left.\begin{aligned} &M_{by}\ddot{Y}_{by}=F_{yw1}+F_{yw2}+F_{yw3}+F_{yw4}+F_{yw5}+F_{yw6}-F_{ys1}-F_{ys2}-F_{ys3}+M_b g(\Phi_{seb1}+\\ &\Phi_{seb2}+\Phi_{seb3})+F_{ycb1}+F_{ycb2}+F_{ycb3}\\ &M_{bz}\hat{Z}_{bz}=F_{zbA}+F_{zbB}+F_{zbC}-F_{zw1}-F_{zw2}-F_{zw3}-F_{zw4}-F_{zw5}-F_{zw6}+3M_b g_z+F_{zbcp1}+F_{zbcp2}+F_{zbcp3}\\ &I_{yb}\beta_{by} =-M_{yw1}-M_{yw2}-M_{yw3}-M_{yw4}-M_{yw5}-M_{yw6}+M_{yb1}+M_{yb2}+M_{yb3}+M_{ybcpi}+M_{ybcpi}+M_{ybcpi} \end{aligned}\right\}\text{......eq.2}$$

Wheelset

$$\left.\begin{aligned} &M_{wy}\ddot{Y}_{wyi} =-F_{ywmi}+F_{ywcfi}+M_{wg\Phi sewi}+F_{ywcpi}\\ &M_{wz}\hat{Z}_{wz}= F_{zmwi}-F_{zcfwi}+M_w g_{iz}+F_{zwcpi}\\ &I_{yw}\beta_{wi}=M_{ywmfi}-M_{ywcfi}+M_{ywcpi} \end{aligned}\right\}\text{...eq.3}$$

Where M_c , M_b , M_w are mas car body, bogie and wheel sets. I_{yc}, $I_{yb}$and$I_{yw}$are movement of inertia of car body, bogie and wheel sets. $\ddot{Y}_c$, $\hat{Z}_{cz}$, $\ddot{Y}_b$, $\hat{Z}_b$,$\ddot{Y}_{wi}$, $\hat{Z}_{wz}$ are accelelaration of car body, bogie and wheel set along lateral and vertical direction.β_c, β_{by}, $\beta_{wi}$are angular acceleration of car body, bogie and wheel set. F_{yA}, F_{yB}, F_{yC}, F_{zA}, F_{zB}, F_{Zc}, F_{ys1}, F_{ys2}, F_{ys3}, F_{zbA}, F_{zbB}, F_{zbC}, F_{ywmi} and $F_{zmwi}$are the mutual forces of car body, bogie, bogie and wheelset in lateral and vertical direction. F_{yAB}, $F_{yBC}$$F_{zAB}$, F_{zBC}, M_{yAB} and M_{yBC} are internal forces and movements of the articulation in lateral and vertical direction due to cause of adjacent vehicle inter connection components.F_{yw1}, F_{yw2}, F_{yw3}, F_{yw4}, F_{yw5}, F_{yw6}, F_{zw1}, F_{zw2}, F_{zw3}, F_{zw4}, F_{zw5}, F_{zw6} internal forces between bogie and wheelset in lateral and vertical directions. F_{ywcfi} and F_{zcfwi}contact forces of each wheelset between wheels and the rails in lateral and vertical directions. $M_c g$ $(\Phi_{sec1}+ \Phi_{sec2} +\Phi_{sec3})$,$3M_c g_z$, $M_b g(\Phi_{seb1}+ \Phi_{seb2} +\Phi_{seb3})$,$3M_b g_z$, $M_{wg\Phi sewi}$and$M_w g_{iz}$forces due to deflection of car bodies, bogies, and wheelset in lateral and vertical directions. F_{ycp1}, F_{ycp2}, F_{ycp3},F_{zccp1}, F_{zccp2}, F_{zccp3}, F_{ycb1}, F_{ycb2}, F_{ycb3}, F_{zbcp1}, F_{zbcp2}, F_{zbcp3}, F_{ywcpi} and $F_{zwcpi}$are external forces on car bodies, bogies, wheelsets in lateral and vertical direction due to the centripetal accelerations when the vehicles negotiating curved track.M_{yA}, M_{yB}, M_{yC}, M_{yb1}, M_{yb2}, M_{yb3} and $M_{ywmfi}$are mutual movements of car body, bogie and wheel sets. M_{yccp1}, M_{yccp2}, M_{yccp3}, M_{ybcpi}, M_{ybcpi}, $M_{ybcpi}$and $M_{ywcpi}$are external movements on car bodies, bogies, wheelsets in lateral and vertical direction due to the centripetal accelerations when the vehicles negotiating curved track.Φ_{seci} deflection angle of the systems. (i= 1, 2…)

3.4.2. Vertical suspension of the three bogie passenger vehicle vertical dynamics

As shown in the figure.15 for the vertical suspension systems the primary and secondary suspensions play an important role to maintain the lowering down of the vehicle body to the rail. For the equation of motion based on the suspension system can be formed. The two motered bogie have the same mass and dimention and paramameter. But the midle or the trailer bogie has different mass and the dimention and other parameter like springs and dampers have the same parameter to the motered bogie.

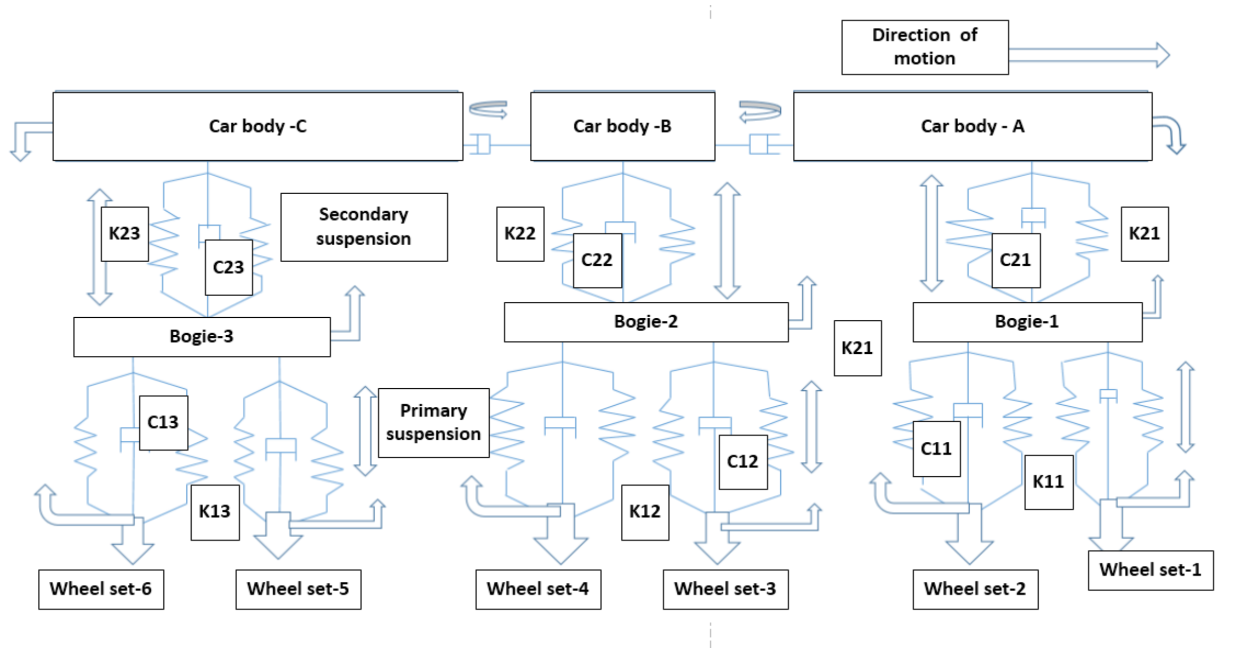


Figure 15. Vertical primary and secondary suspension[20]

Based on this skematic diagram of passenger car body primary and secondary suspension the equation of motion can be drived just for one bogie then the total equation of motion drived in matrix form. The stiffne and damping of the primary and secondary suspension parameters are constant but as the passenger increases the mass of car body increases. This simultanious increased mass of car body affects the suspension stiffnes.

$M_{cA}\, Z''_2 + C_2\, Z'_2 – C1\, Z_1 + 2k_2 Z_2 - 2k_2 Z_1 = 0$ 4

$M_{b1}\, Z''_1 + C_2\, Z'_1 - C_2\, Z'_2 + 2k_2 Z_1 - 2k_2 Z_2 + 4k_1 Z_1 - 4k_1 Z_o + 2C_1 Z_1 - 2C_1 Z_0 = 0$.......................................5

The equation 4and equation 5 represents the vertical system equation of motion of one bogie system. M_{CA}, M_{b1} mass of car body and bogie. Z''_2, Z''_1 acceleration of car body and bogie. C_2 and C_1 secondary and primary damping. K_2 and k_1 are secondary and primary siffnes springs. Z1 and Z2 are vertical displacement of car bogie and car body. This unrealistic displacement leads lowering down of car body to the rail.

The over all equation of vertical motion of the three bogie primary and secondary suspension represents in matrix form as follows.

$$\begin{bmatrix} M_{cA} & 0 & 0 & 0 & 0 & 0 \\ 0 & Mb1 & 0 & 0 & 0 & 0 \\ 0 & 0 & McB & 0 & 0 & 0 \\ 0 & 0 & 0 & Mb2 & 0 & 0 \\ 0 & 0 & 0 & 0 & McC & 0 \\ 0 & 0 & 0 & 0 & 0 & Mb3 \end{bmatrix} \begin{bmatrix} \dot{Z}_{21} \\ \dot{Z}_{11} \\ \dot{Z}_{22} \\ \dot{Z}_{12} \\ \dot{Z}_{23} \\ \dot{Z}_{13} \end{bmatrix} + \begin{bmatrix} C_{21} & C_{21} & 0 & 0 & 0 & 0 \\ -C_{21} & C_{21}+2C_{11} & 0 & 0 & 0 & 0 \\ 0 & 0 & C_{22} & -C_{22} & 0 & 0 \\ 0 & 0 & -C_{22} & C_{22}+2C12 & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{23} & -C_{23} \\ 0 & 0 & 0 & 0 & -C_{23} & C_{23}+2C13 \end{bmatrix} \begin{bmatrix} \dot{Z}21 \\ \dot{Z}_{11} \\ \dot{Z}_{22} \\ \dot{Z}_{12} \\ \dot{Z}_{23} \\ \dot{Z}13 \end{bmatrix}$$

$$+ \begin{bmatrix} 2K21 & -2K21 & 0 & 0 & 0 & 0 \\ -2K21 & 2K21+4K11 & 0 & 0 & 0 & 0 \\ 0 & 0 & 2K22 & -2K22 & 0 & 0 \\ 0 & 0 & -2K22 & 2K22+4K12 & 0 & 0 \\ 0 & 0 & 0 & 0 & 2K23 & -2K23 \\ 0 & 0 & 0 & 0 & -2K23 & 2K23+4K13 \end{bmatrix} \begin{bmatrix} \dot{Z}_{21} \\ \dot{Z}_{11} \\ \dot{Z}_{22} \\ \dot{Z}_{12} \\ \dot{Z}23 \\ \dot{Z}_{13} \end{bmatrix} = \begin{bmatrix} 0 \\ Fz1 \\ 0 \\ Fz2 \\ 0 \\ Fz3 \end{bmatrix}$$

The bogie mass, the primary and secondary suspension stiffness and damping values and car body mass are constant but while adding the passenger load the car body mass increases simultaneously until maximum limit. The other parameters like stiffness are constant. Due to this reason the vertical displacements are variables which depend on the passenger load. And also external induced forces vary based on passenger loads.

3.5. Data entry for the Multi-Body software

The data collected from the Ethipia railway corporation training manual and spesifications which is currently used. Based on this data, modeling the system by using SIMPACK multi-body simulation software package. Some dimensions gets from general dimensions. The center of mass and movement of inertia have got from calculations. See Table 5. Basic parameters are shown in appendix A.

3.5.1 Load parameter

The manufacturer stated that, the occupied area of standing passengers is 6 persons/m^2 for rated passenger capacity or 8 persons/m^2 for over-crowed capacity; the average weight of the passenger is 60kg/person [1]. See Table 4. Load parameter in appendix A. But the current situation is greater than 63 ton rather 64 or 65 tons.

3.6. 3D modeling of articulated passenger rail vehicle

Three-dimensional (3D) models represent a passenger rail vehicles by using a collection of points in 3D space, connected by various geometric entities such as wheels, wheel sets (axles), primary and secondary suspension component force elements,axle box, markers, joint mechanism and articulation components like; fixed hinge lower articulations to each adjecent car bodies, flexible and free hinng articulation at the top of each adjecent car bodies,weight balansses and supporting components and the three car body modules. The modeling process takes Ethiopia light railway transient project data. From starting wheels and wheel sets up to the whole component model, the force elements, joints, constraints and body markers also included. For the modeling of the passenger rail vehicles, the wheel profiles used s1002 and for rail profiles UIC 60 are used. According to this specification the maximum value of the flange angle $\gamma=70^0$ for the S1002/UIC60 wheel/rail profiles commonly used by the railway corporations in the European countries and in china railway industry.

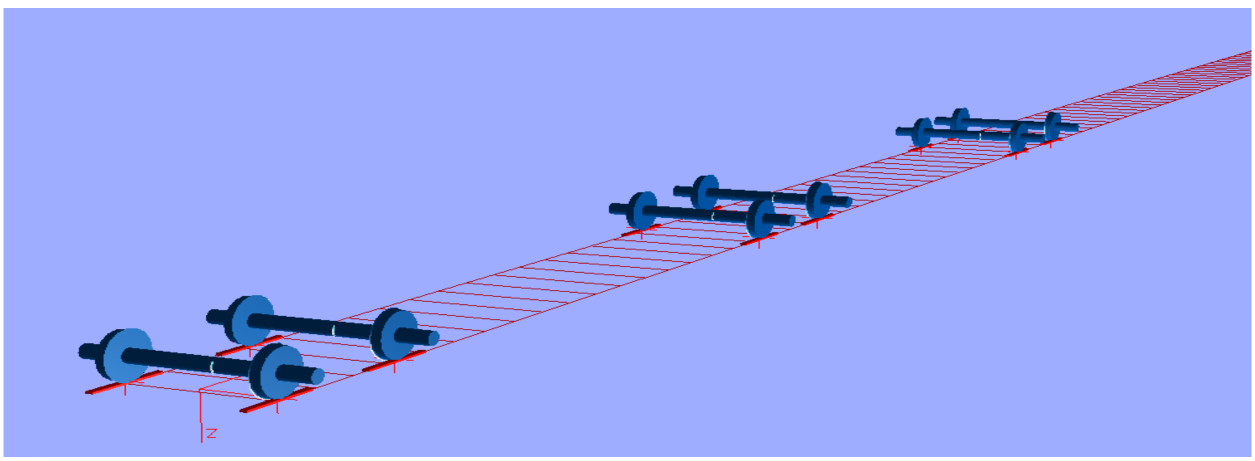


Figure 16. The six wheels including axles

After modeling wheels and axles, then, modeling the axle boxes, primary suspension components, and bogie frames by using graphical user interface window was made. The primary suspension force elements start from wheel set axle box to bogie frame and the force type is spring damper parallel component forces.

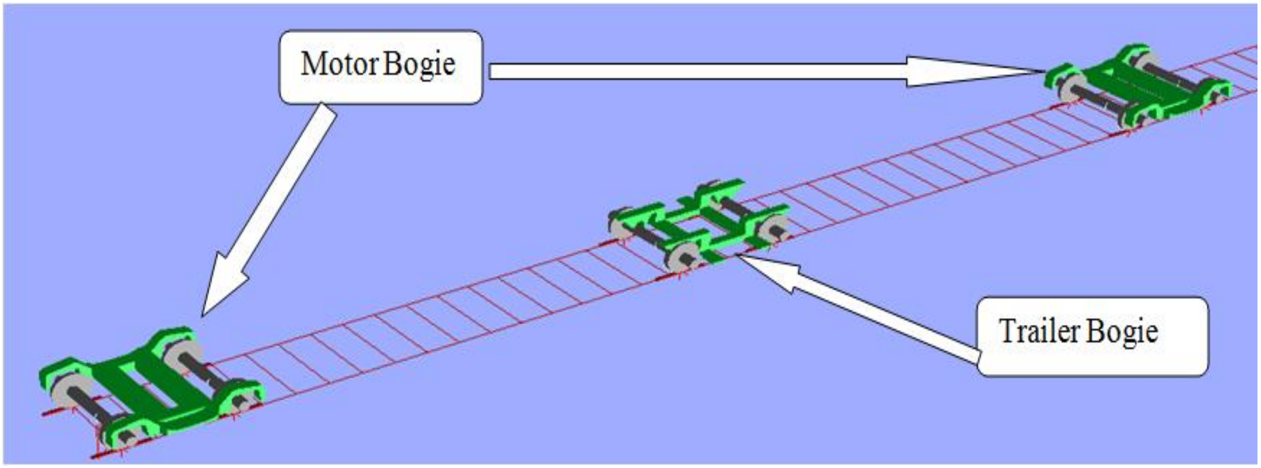


Figure 17. Wheel sets, axle boxes and bogie frames

The bogie frame is the intermediate component between wheel set and car body module. This system connects to the wheel set and car body by force element which means primary suspension connects bogie frame to wheel sets and secondary suspension connects car body to bogie frames. This frame has many marker points and force elements to connect different components like wheel set to bogie frame, bogie frame to car body in longitudinal, lateral and vertical point of contact, stopper, traction, weight balance and etc.

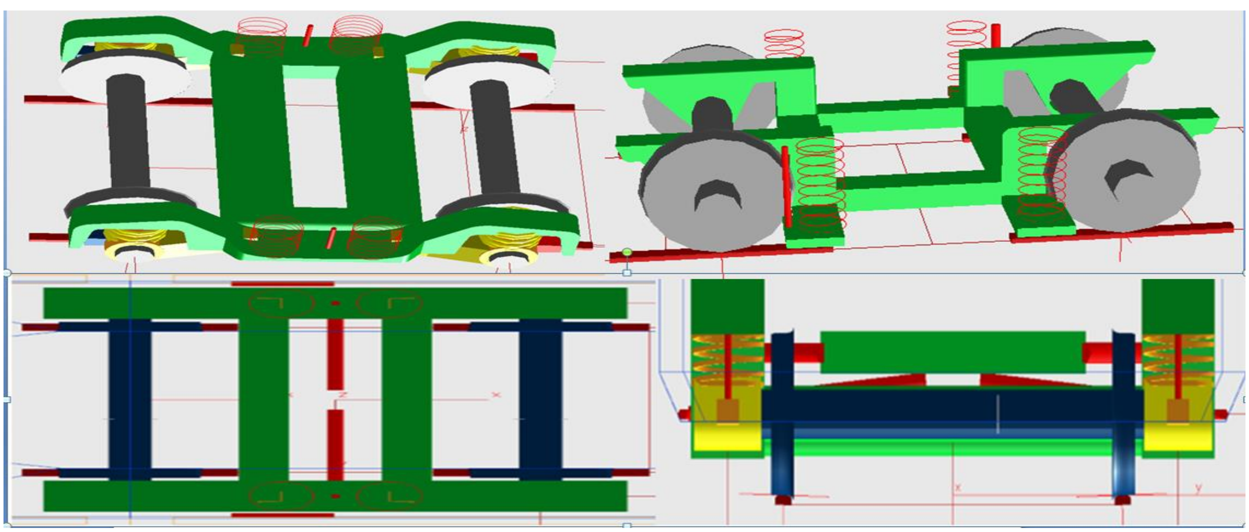


Figure 18. Motor and Trailer Bogie frame

3D modeling of both of Lower fixed articulations, upper flexible and free articulation components model by CATIA V5 and save in IGS format then open in solid work, save in solid work by pro engineer format, open this file by pro engineer software and save in slp format finally import to SIMPACK dynamic simulation software package, then give the markers points, joints and force elements to assemble with the car body module. The lower fixed articulation connects to the car body by fixed joint zero degree of freedom and between the two side components connected by revolute joint mechanism to rotate at the middle joint to compensate curve negotiation while the vehicles pass through curved track. The flexible upper articulation front side element connects the car body-B by fixed joint zero degree of freedom. The rear side back end components connects to the car body-C by fixed joint and connects the next point by revolute joint to compensate vertical supper elevated track and freely rotate at the middle pin joint point. The upper free articulation front component connect to car body-A by fixed joint mechanisms and freely rotate at the middle pin joint mechanisms and the rear joint also rigidly connect to car body-C and freely rotate at the middle joint with 540mm center arm. The following figure shows the lower and upper articulation components including fixed articulation dampers.

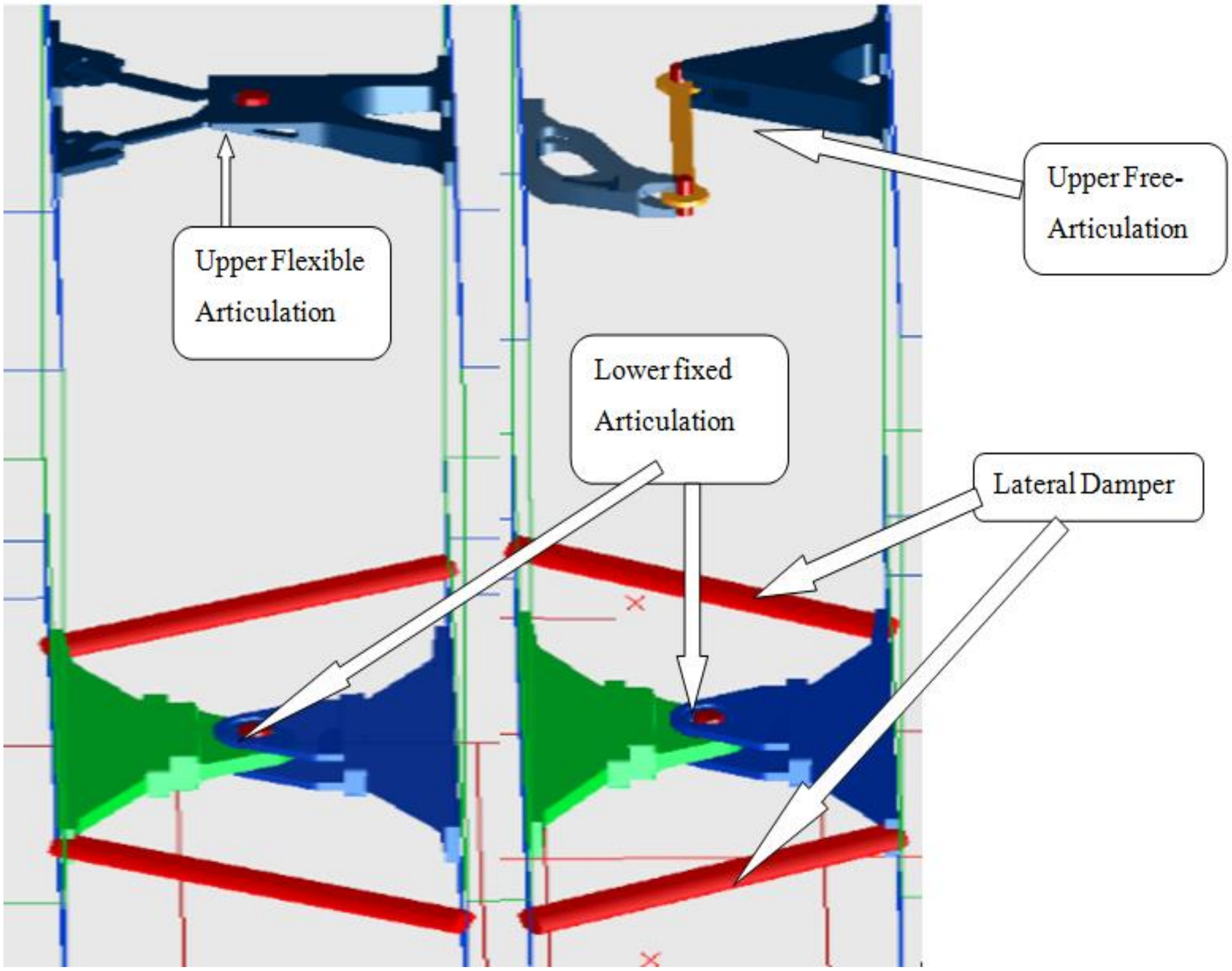


Figure 19. Fixed lower articulations, upper flexible and free articulations and dampers

The following figure shows that, the two side lower fixed articulations and upper flexible and free articulation including middle car body-B and middle bogie frame and also the lower articulation damping components.

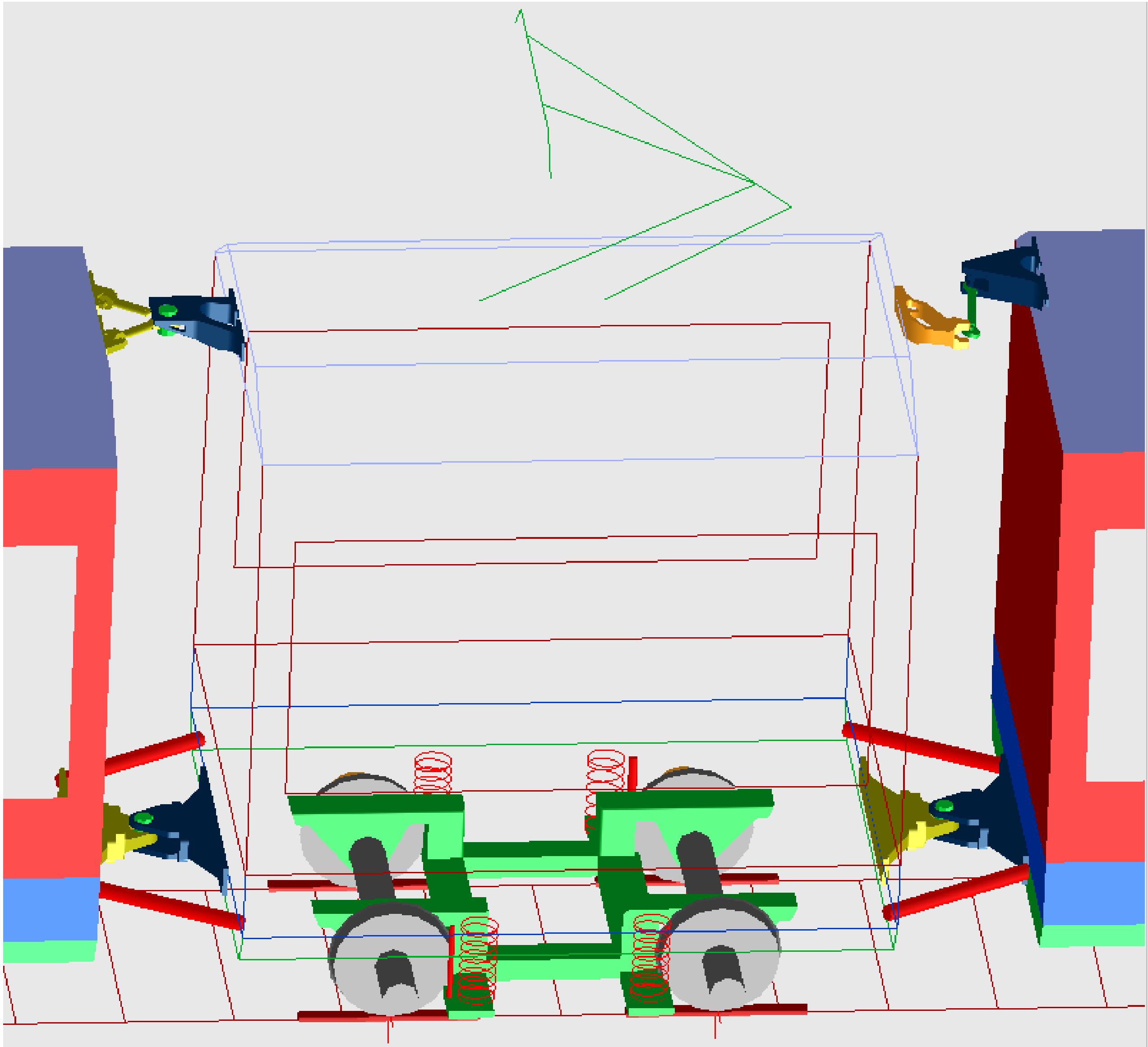


Figure 20. The two side lower and upper articulations and car body-B including middle bogie

The 3D assembled articulated passenger vehicle including some of marker points and corresponding force elements shown as follows in the following figure.

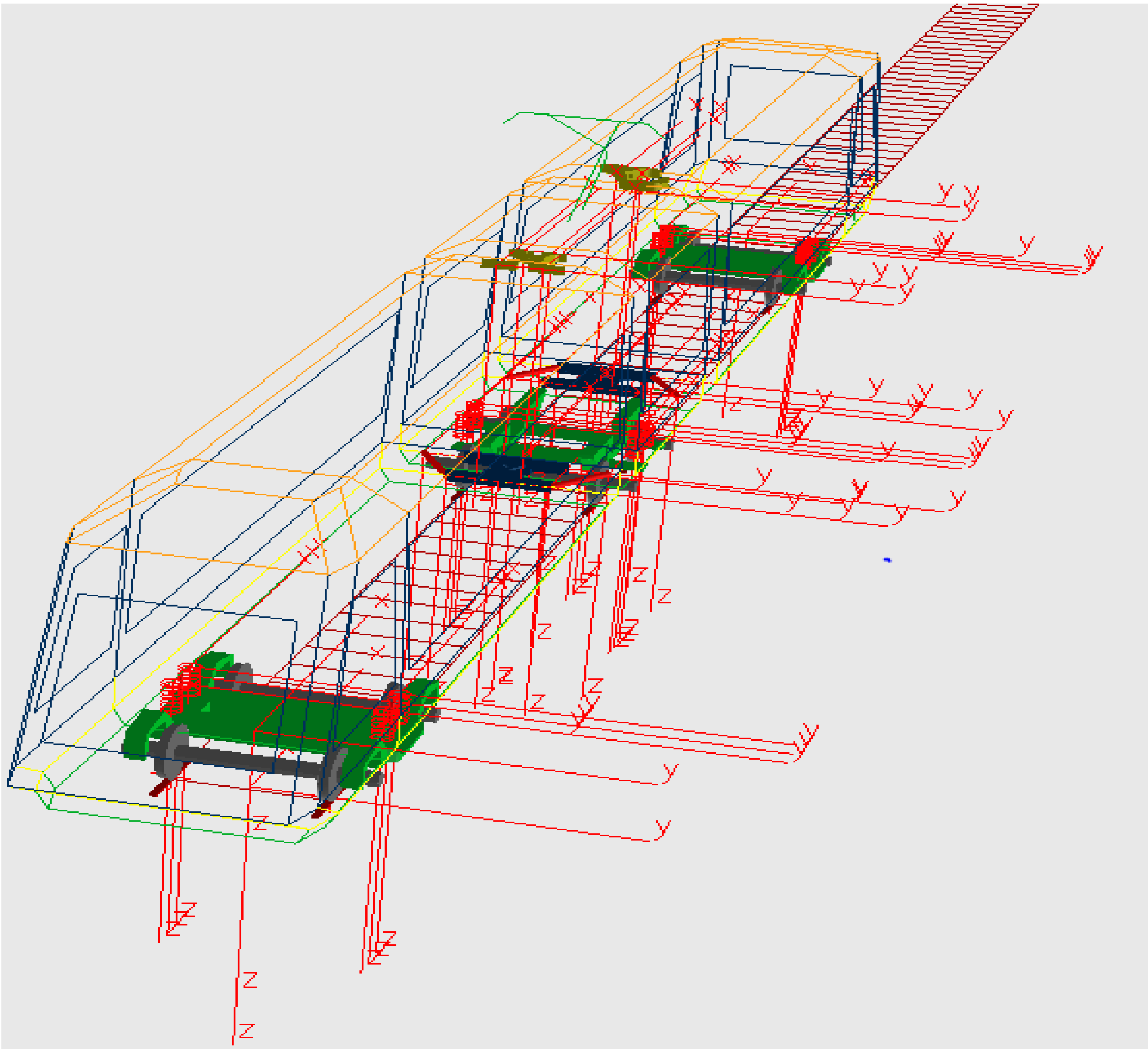


Figure 21. Model of car body including marker points

The assembled vehicle body contains all the necessary components, marker points, force elements, the wheel sets, the axle boxes, the three bogies, the three car body modules, the lower and upper articulation components, the primary and secondary longitudinal, lateral and vertical suspensions, dampers and other important components shown in the above figure in assembled form. To see all components, see the side view, front view andtop view in assembled form in the following figure.

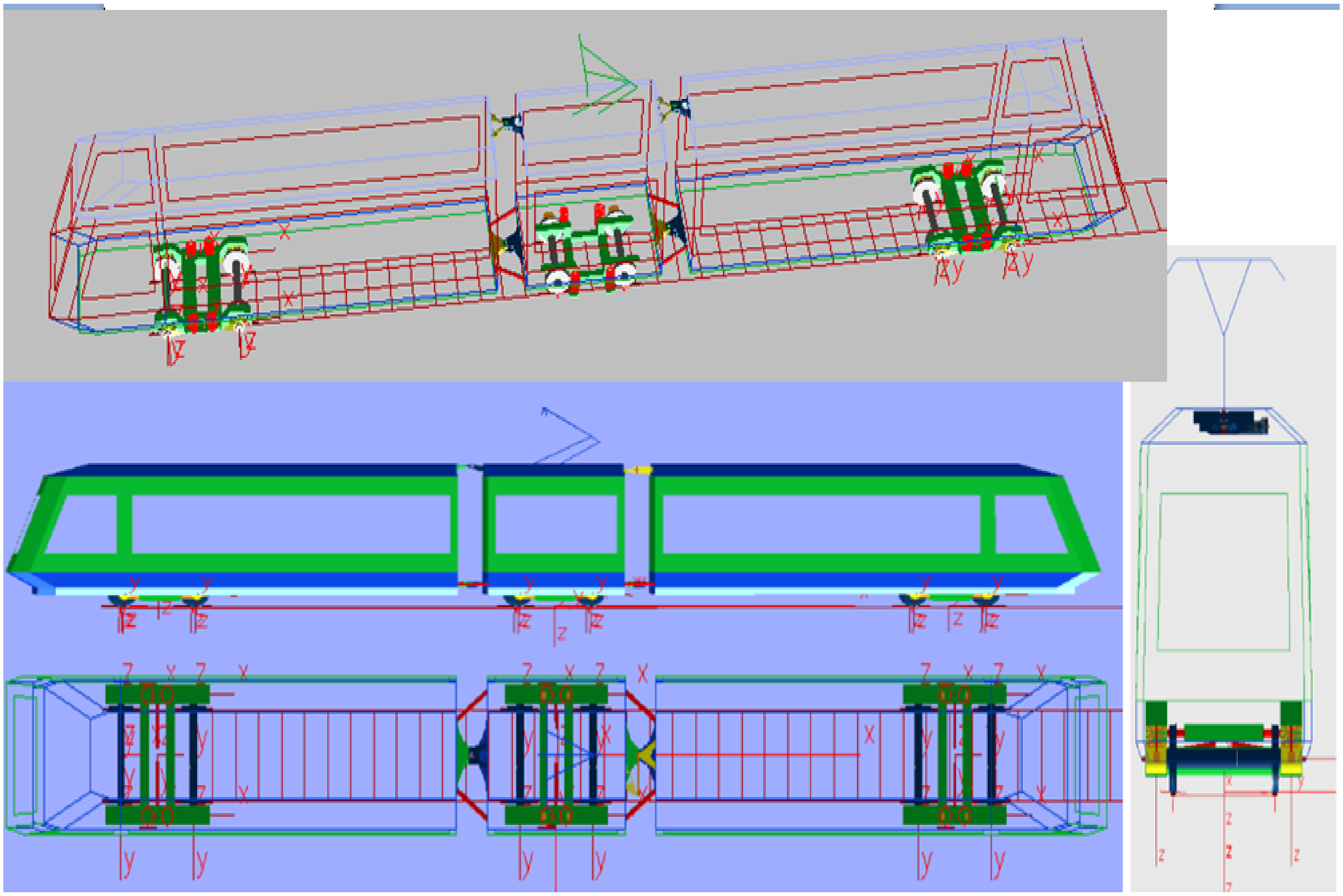


Figure 22. Front, side and top view of car body

According to track topology, there are different types of tracks such as straight track, constant curve, curve entry, curve passing, sign change curve and crossing. For this paper, straight track and sign change curve at super elevated track at different simulation parameter was studied. As shown the following figure at minimum radios curved track with supper elevated structure was studied.

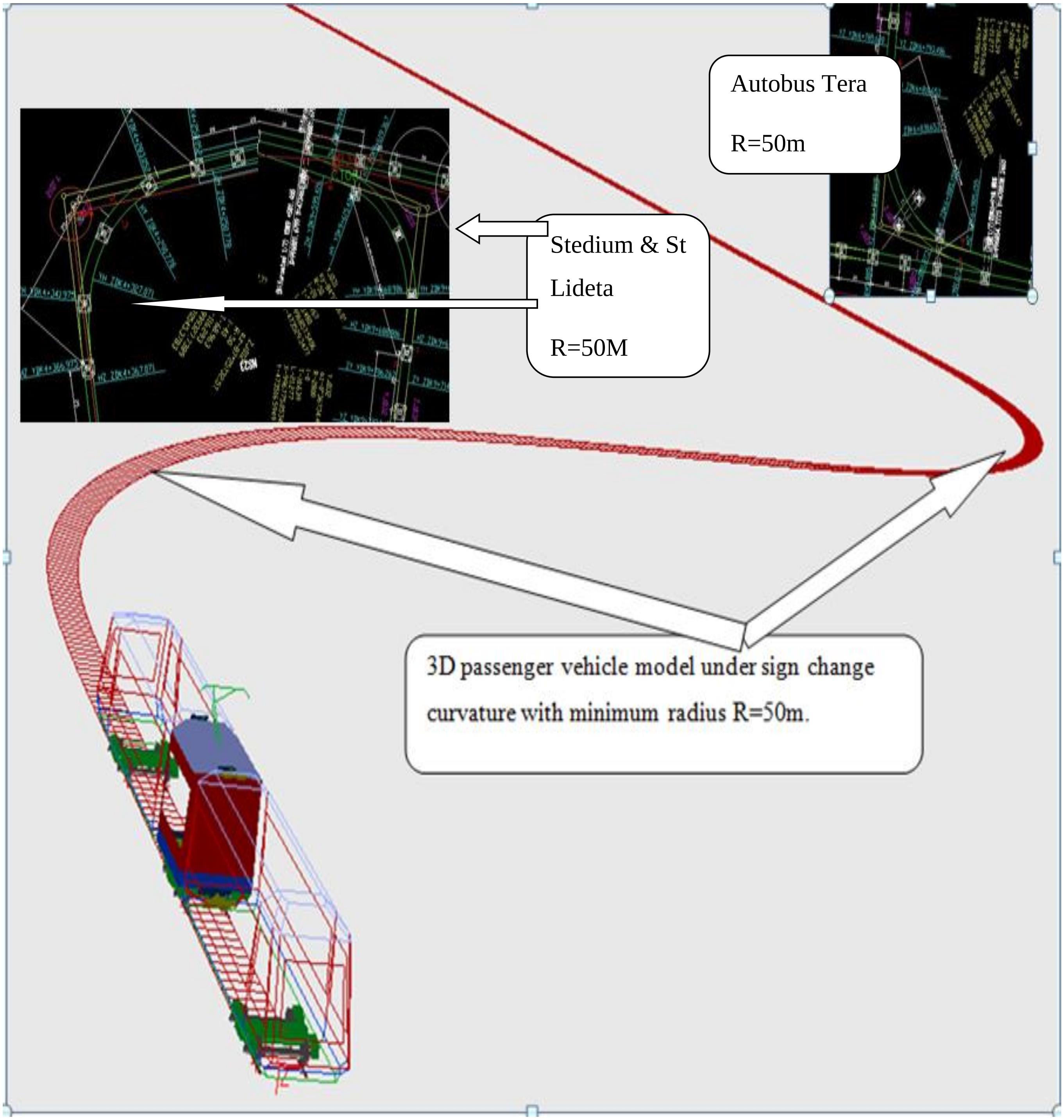


Figure 23. The 3D model of vehicle on curved track

CHAPTER FOUR

4.1. Multi-body simulation

A Multi-Body Simulation system is an attempt to model a real-life or hypothetical situation of the passenger rail vehicle in the computer so that it can be studied to see how the system works and when and which components in the failerity occurs. By changing variables in the simulation process, predictions of primary and secondary suspension vertical position and parameter variation on wheel diameter, vehicle load, suspension stiffness and speed can be made. The following table shows the pre-condition of multi-body simulations and expected simulation output values according to the stated problem and objectives.

Table 2. Dynamic simulation condition and Expected Dynamic simulation out put

Dynamic simulation condition	
Vehicle weight	Maximum pick hour load
Wheel type	S1002 wheel type with flange angle 70 degree, 21.21mm flange thickness
Rail type	UIC 60 rail type with track irregularity
Operating speed	Design speed (80km/h) for straight track
	35km/h for curved track with curve radius R50m
Operating condition	Under dry weather condition
Track condition	Straight track
	Curved track with minimum radius of curvature (50m)
Expected Dynamic simulation out put	
Primary suspension	Under maximum load condition with currently used stiffness and damping value simulating the vehicle in straight and curved track condition
Secondary suspension	Under maximum load condition with currently used stiffness and damping value simulating the vehicle in straight and curved track condition
Bogie	With the effect of those parameter to the bogie in vertical and lateral

	direction with the stated condition
Front and rear (AB and BC) car articulation	With currently used suspension stiffness value to show vertical displacement of articulation at maximum load condition
Car body	Car body vertical and lateral displacement under the stated conditions
Leading Wheel set lateral force	The leading wheel set lateral force must be under allowable limit according to rail way association standards
Leading wheel set vertical force	The vertical force of the leading wheel set must be under allowable limit
Ride index	Lateral and vertical ride index must be under the limit according to passenger comfort evaluation
Derailment coefficient	The lateral to vertical force quotient must be under the limit to get the railway vehicle operation in reliable and safe condition.

4.2. Hunting State Analysis

Hunting of railway vehicles refers to self-excited oscillation of truck and vehicle body [2]. This dynamic instability is due to the conicity of the wheels, the wheel/rail creep forces and the action of the suspensions [2]. The hunting state of wheel set, bogie frame and car body due to wheel conicity and track excitation effect on fully assemble passenger vehicle were occurred. At low speed and large curve radius the hunting effect were minimum. But under maximum critical speed and minimum curve radius the hunting state was maximum. Under minimum horizontal curve radius and the stated condition the following figure shows the wheel sets, the three bogies and the three car body modules hunting state result. Based on this hunting state, the straight and curved track condition dynamic symolation carried out accordingly.

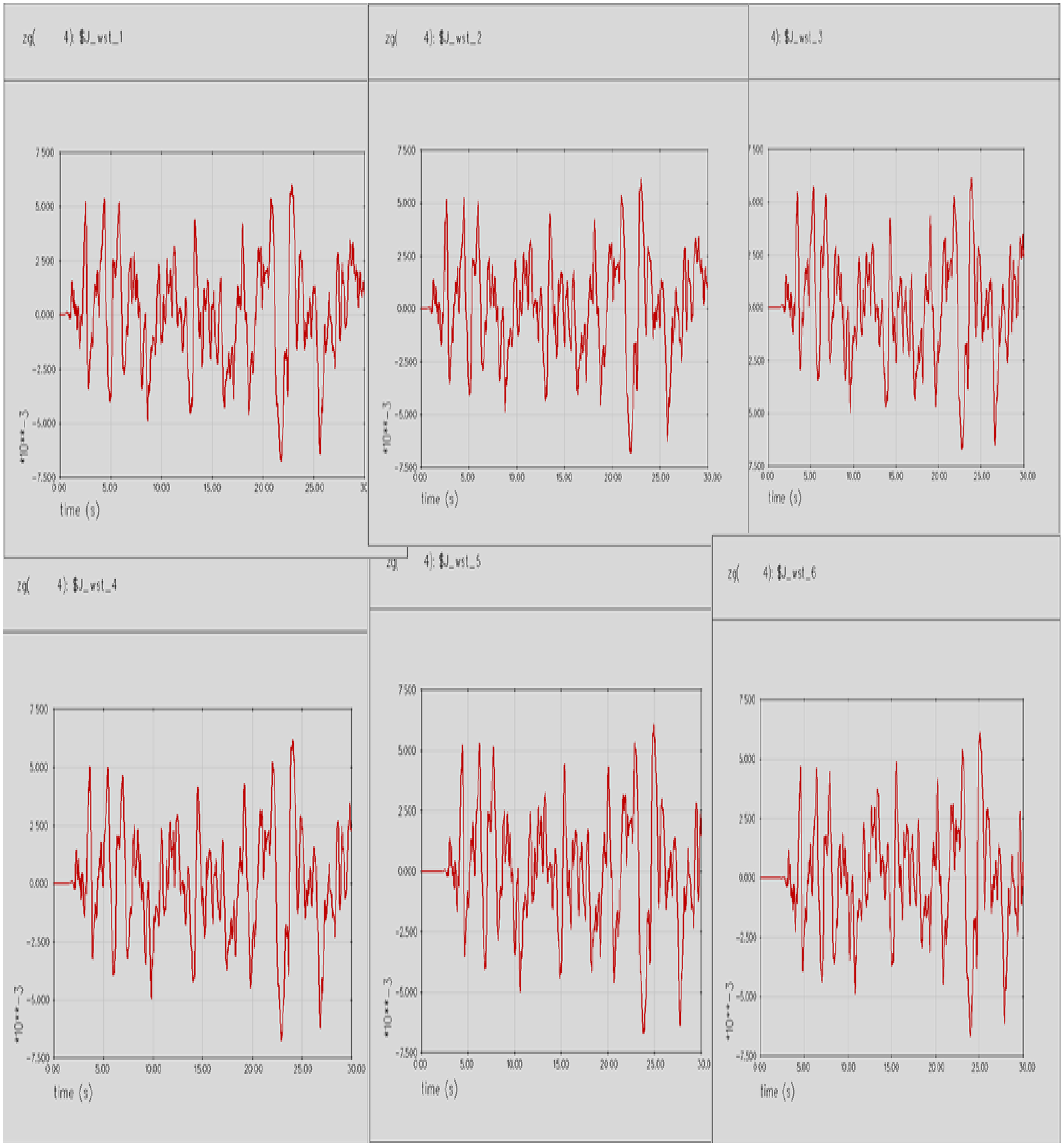


Figure 24. The six wheels set hunting state result under the stated condition

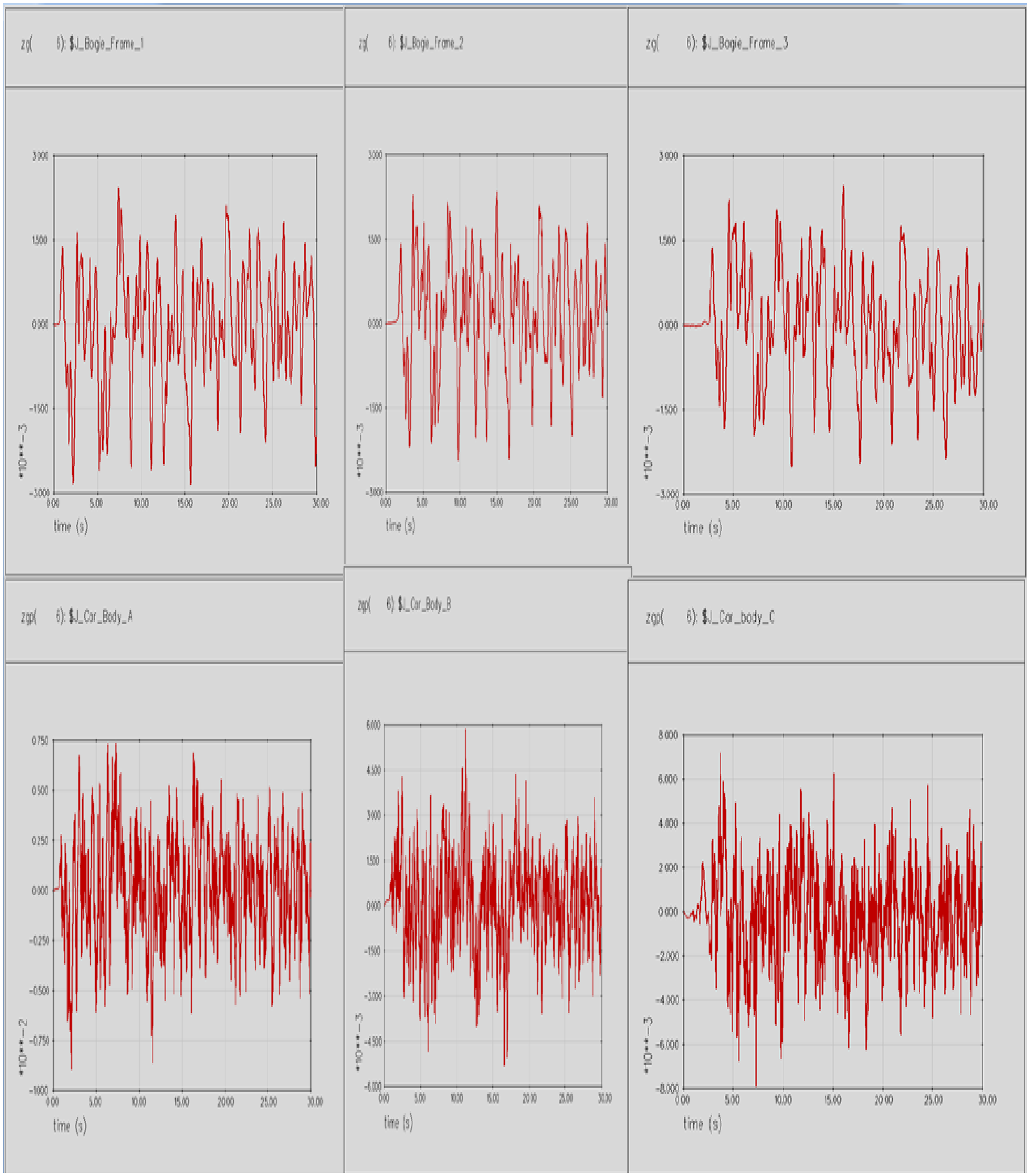


Figure 25. The three bogies and the three car body module hunting state result

4.3. Dynamic simulation on straight track

For this simulation the maximum vehicle load and permissible vertical and lateral suspension stiffness values to access the solution for stated problem were carried out. For the primary and secondary suspension, bogie frames, car body modules and articulation systems corresponding to vertical deflection shown in the following respective simulation results. Based on the above condition the dynamic simulation carried out in straight track.

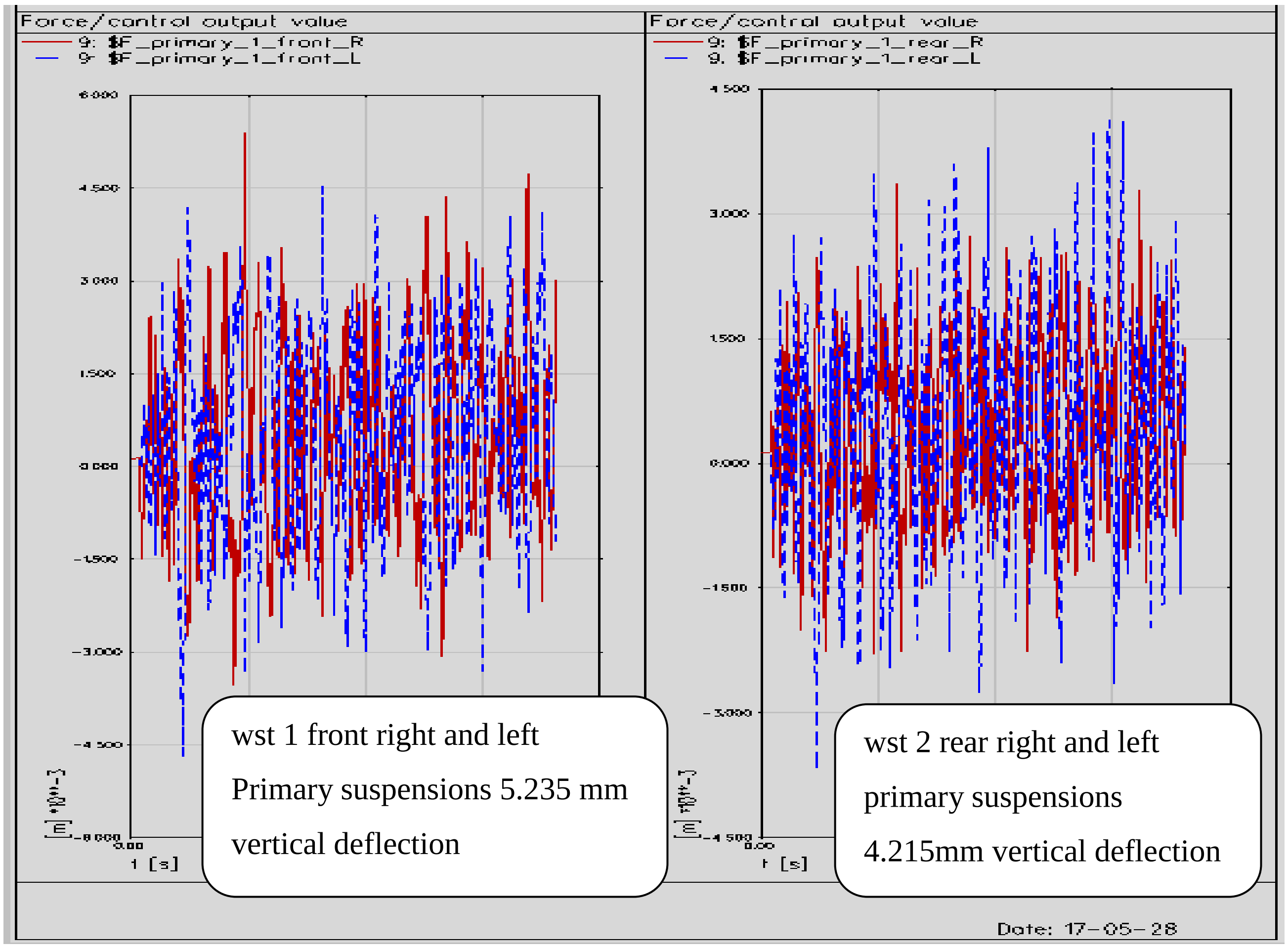


Figure 26. Bogie one Primary suspension on straight track in z-direction

Figure 26 shows bogie one front and rear right and left primary suspension vertical deflection while maximum load condition with maximum vertical deflection was 5.235mm.

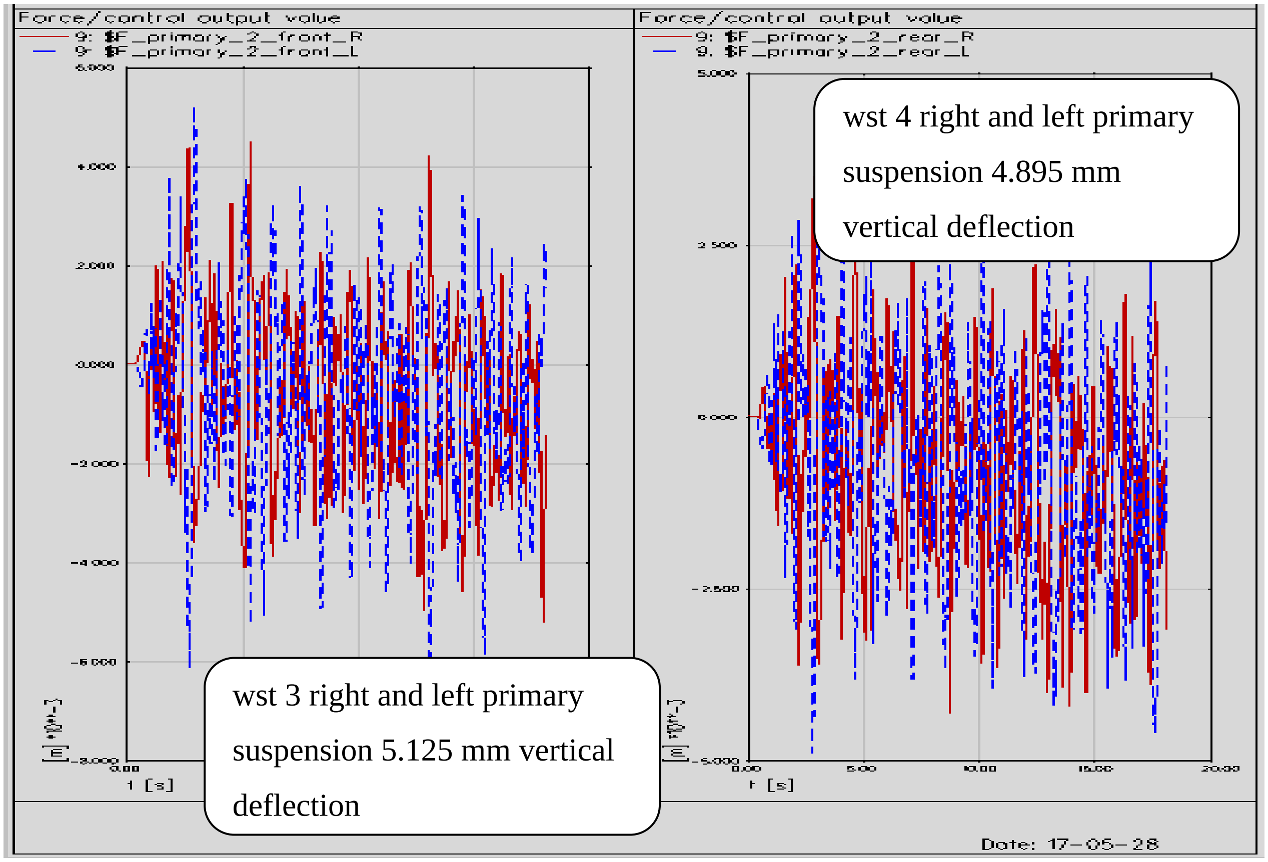


Figure 27. Middle bogie Primary suspensions on straight track in z-direction

This figure shows middle bogie primary suspensions while maximum load condition with maximum vertical deflection was 5.125mm. The third bogie primary suspension maximum vertical deflection was 5.25mm under the stated condition.

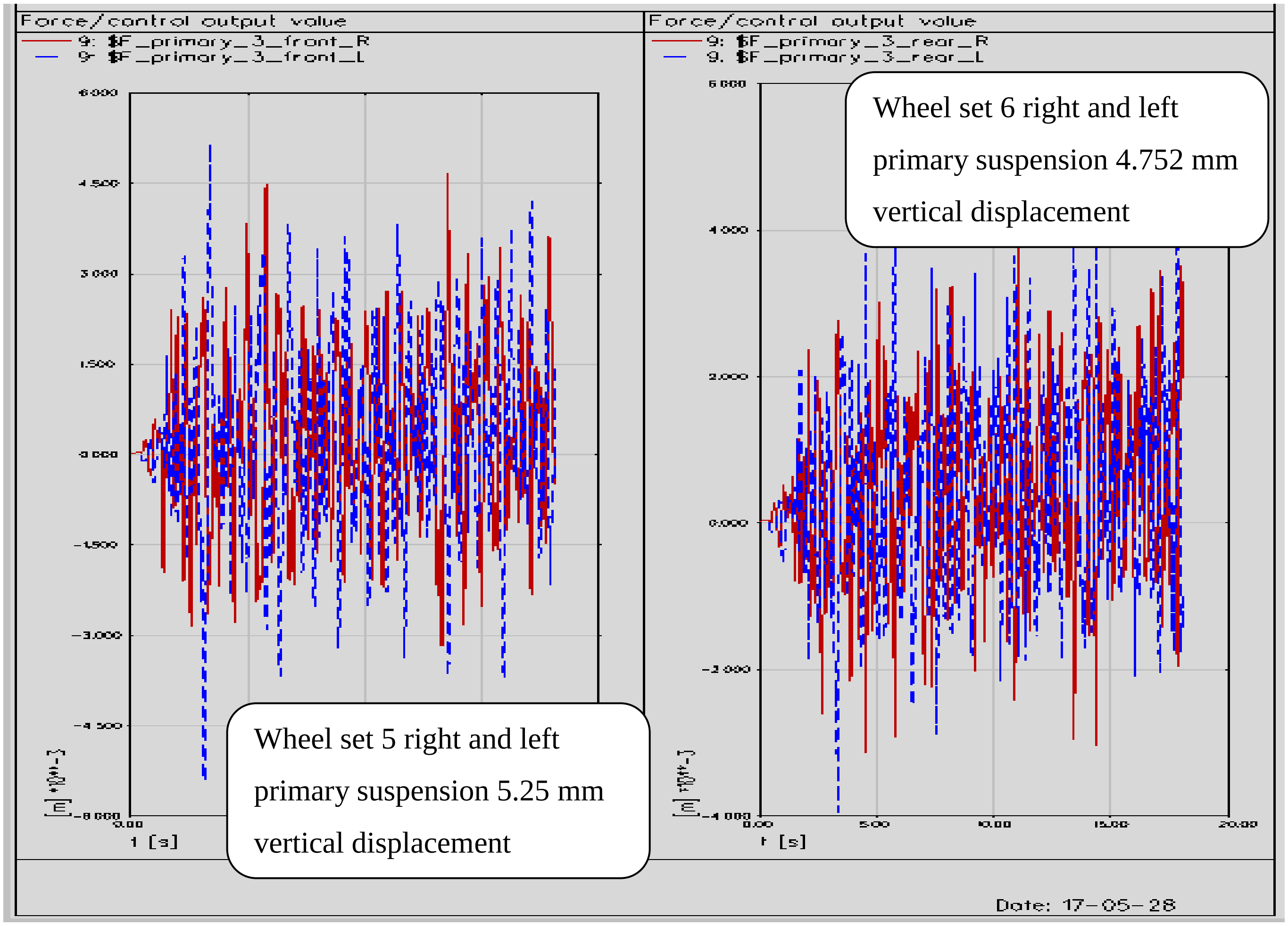


Figure 28. Third bogie primary suspension vertical flexibility

Each bogie frame primary suspension vertical compression shows that, each result below the standard limit value. Because the effects on the straight track are maximum load and kilngling effect due to wheel conicity.

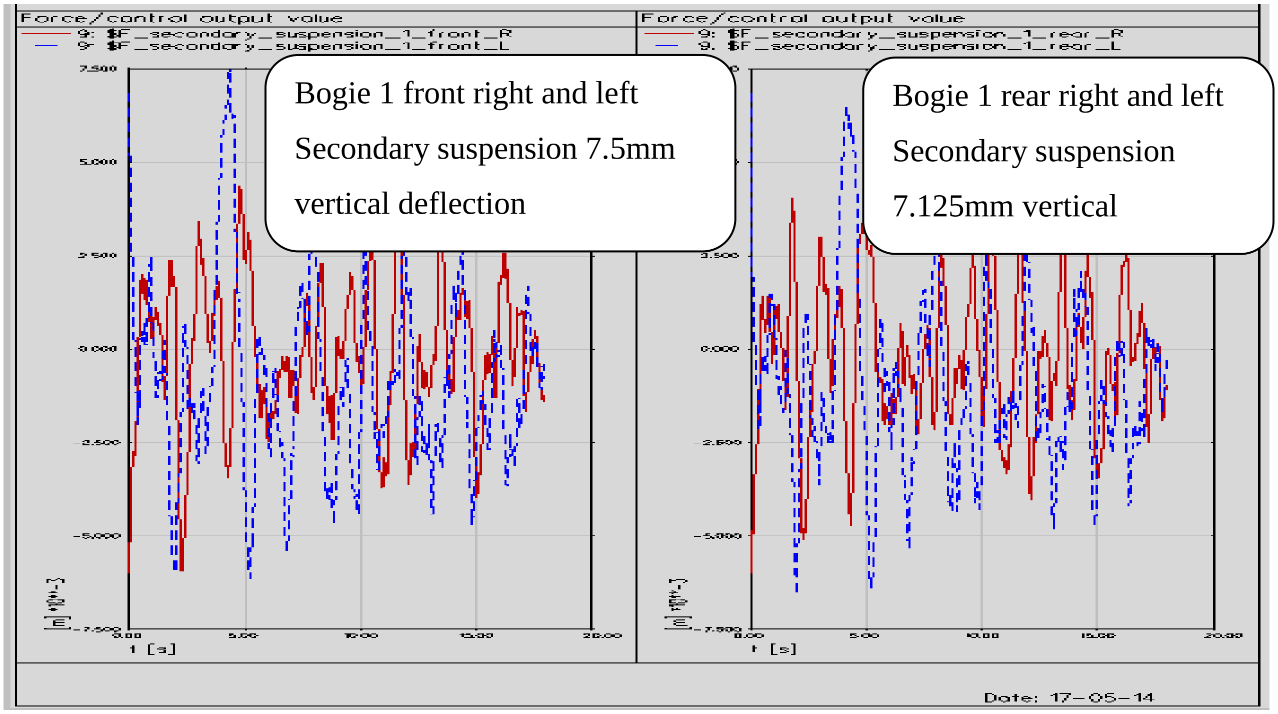


Figure 29. Bogie 1 secondary suspension vertical deflection in z-direction

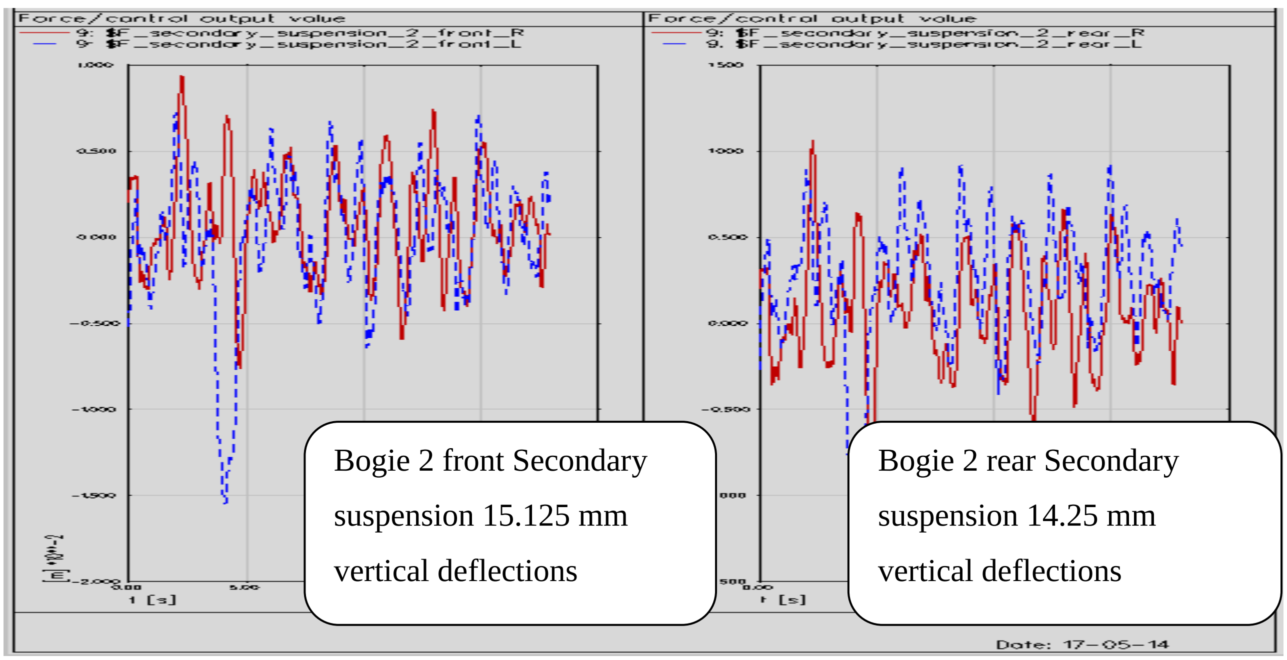


Figure 30. Bogie 2 front and rear secondary suspension in z-direction

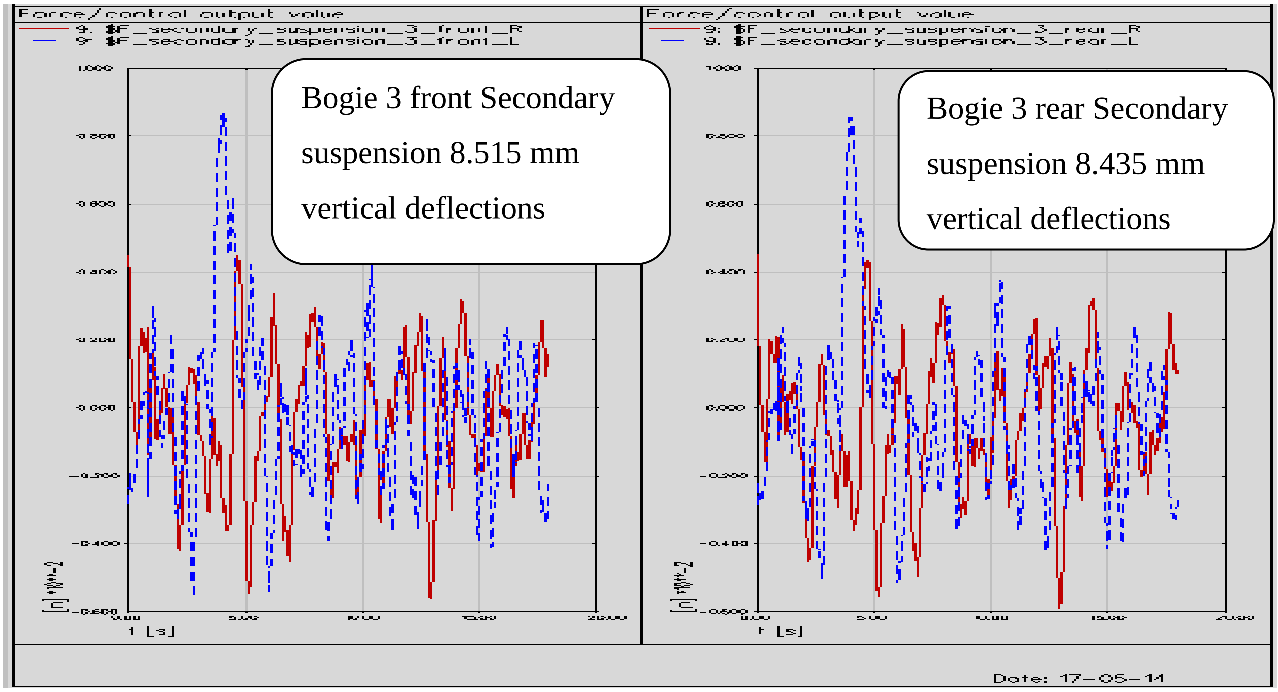


Figure 31. Bogie 3 front and rear secondary suspension in z-direction

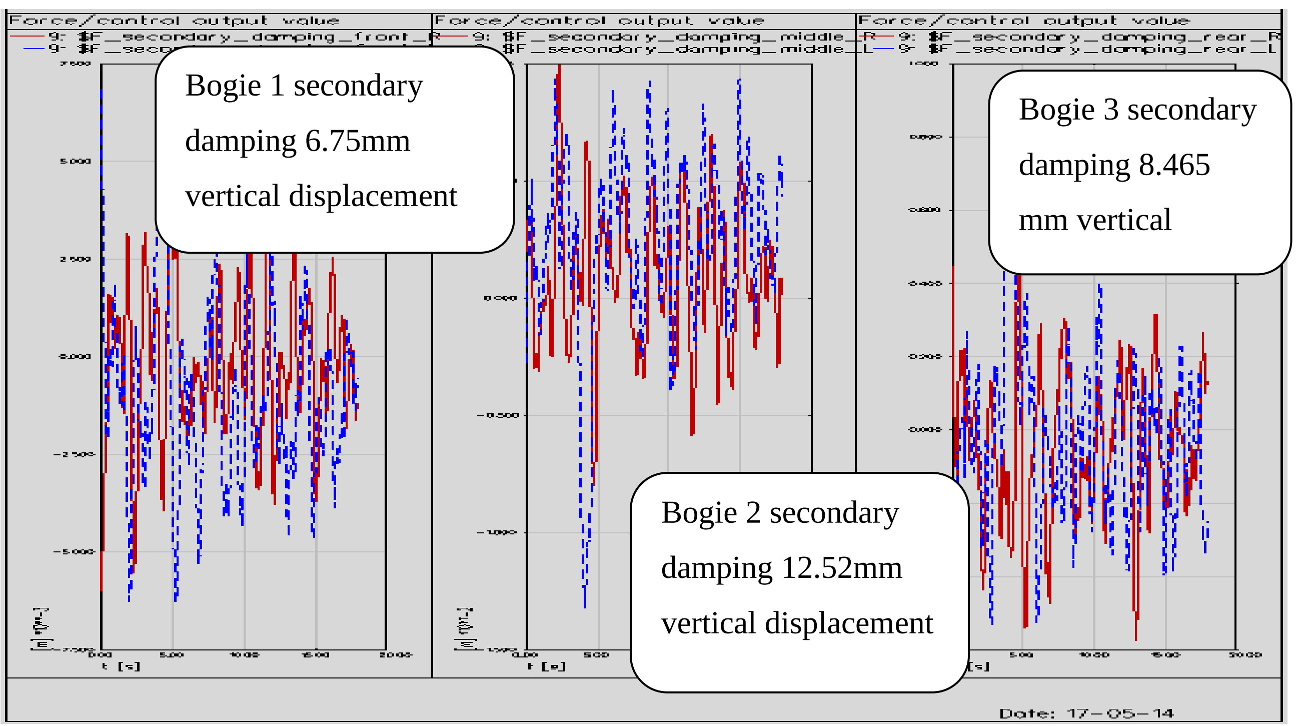


Figure 32. For each bogie secondary damping in z-direction

Figure 29, 30 and 31 shows bogie one, two and three secondary suspension maximum deflections due to maximum vehicle load in straight track. This maximum deflection of middle bogie secondary suspensions occurs due to sharing of load from the two side of the bogie, kilngling effect due to wheel conicity and maximum vehicle load. The articulation components found from the two side of car body B with bogie two. These maximum vertical deflections contribute to lead the current stated problem. The corresponding maximum vertical deflection values of secondary suspensions were 7.5mm, 15.125mm and 8.435mm respectively. The secondary damping vertical deflection also was 12.52mm in the middle bogie. All of straight track simulation was under the specification. Lower fixed articulation front and rear vertical displacement with maximum vehicle load with allowable vertical and lateral suspension and damping values for vertical displacement of articulation was 12.35mm.

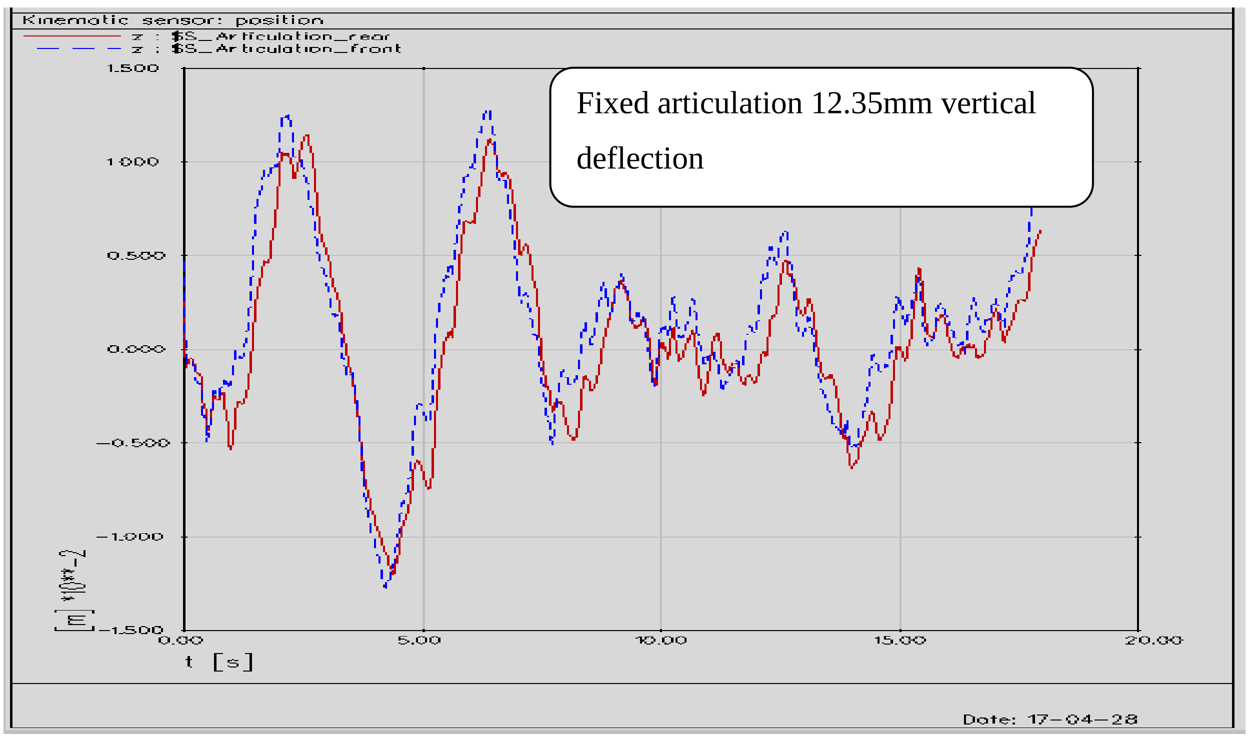


Figure 33. Fixed articulation vertical displacement in z-direction

Car body vertical displacement at maximum load condition on straight track with maximum vertical displacement was 18.25mm.

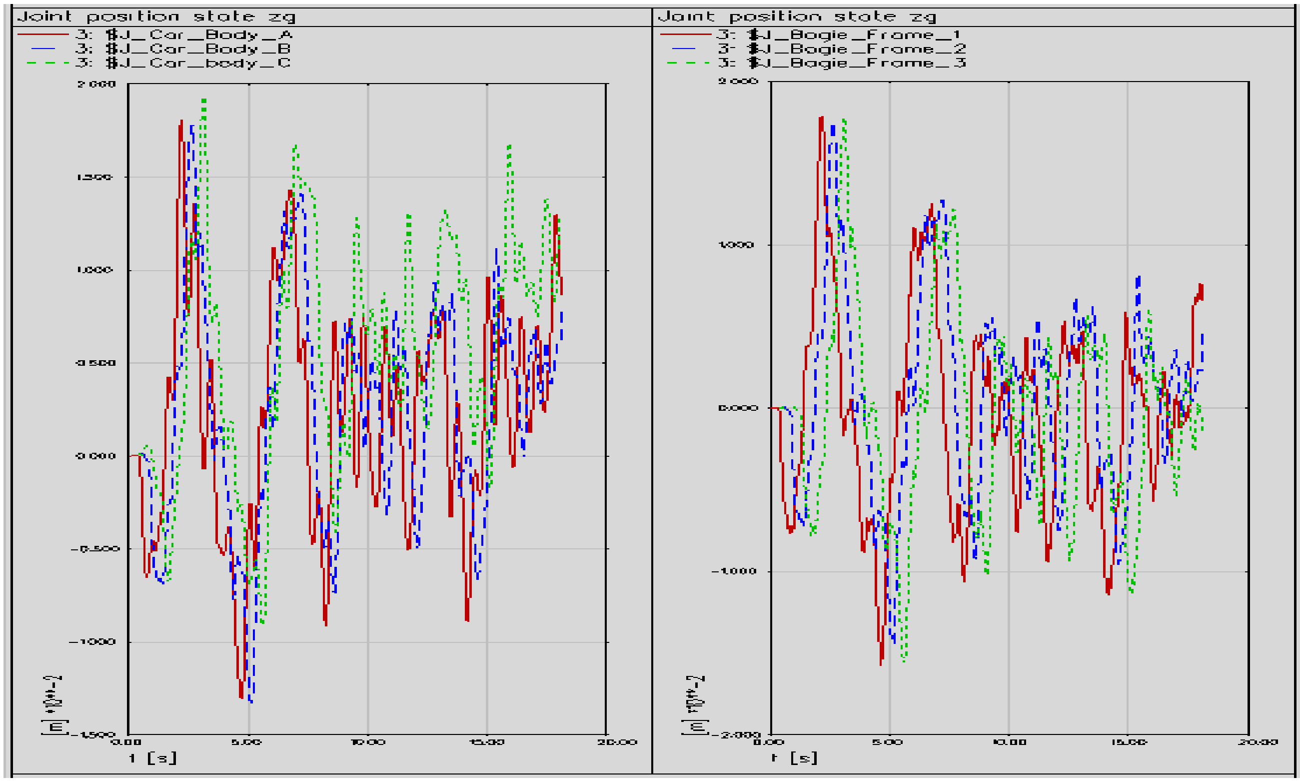


Figure 34. Car body and bogie vertical displacement

4.4. Dynamic simulation on curved track

As AALRT project shows that, the minimum horizontal curved track were found on North-south line. This minimum horizontal curved track with minimum radius R=50m were at stadium, St. Lideta and autobus Tera. The curving property at stadium and St. Lideta are the same. Therefore, this dynamic simulation on those curved track to show the effect of maximum load on the suspension system and on the articulation systems at those minimum radius R=50m was carried out. The multi-body simulation results of each value shows in the following respective graph.

Figure 35, 36 and 37 shows bogie one, bogie two and bogie three primary suspensions vertical displacement on those minimum curve radiuses were 14.765mm, 17.125mm and 16.126mm respectively. This shows that the track curvature have high effect on the suspension systems. Especially in the middle bogie maximum primary and secondary suspension vertical flexibility was occurs. Due to centripetal and centrifugal effect, weight transfer due to track super elevation effect educed this maximum compression on those vertical suspensions.

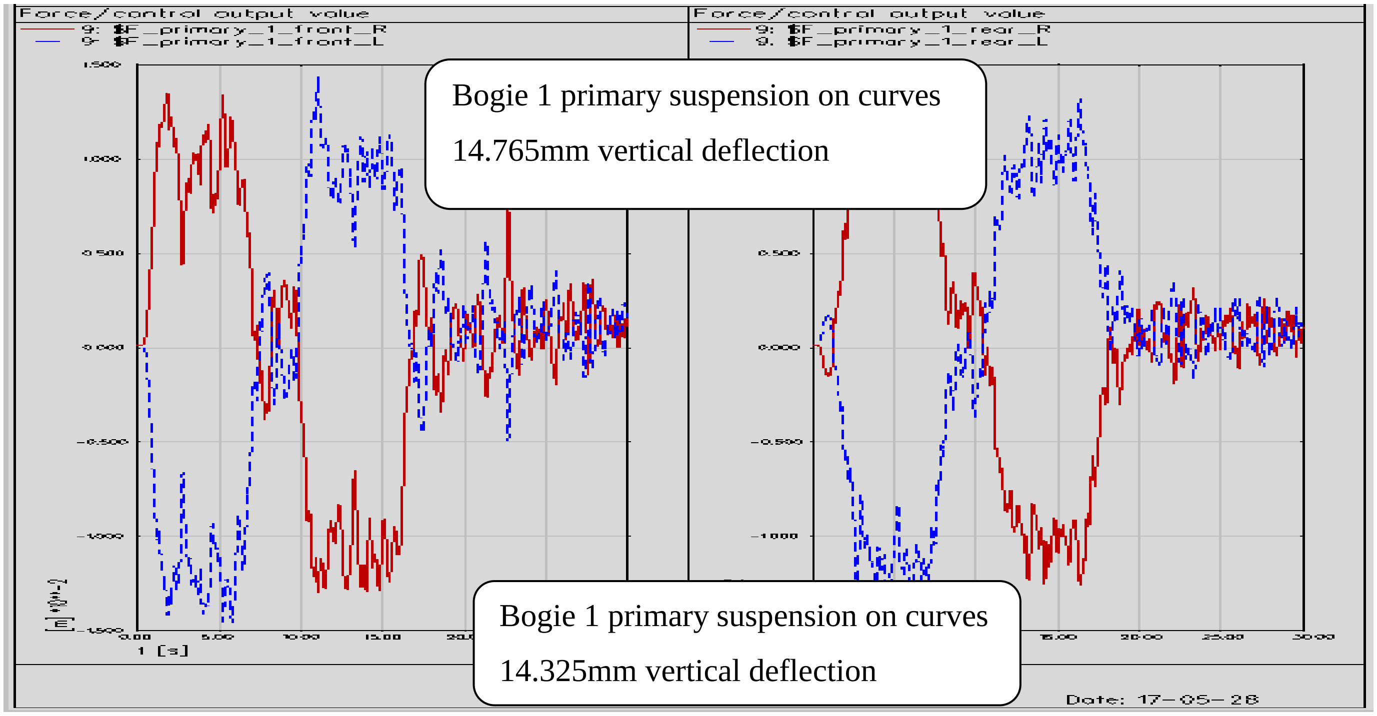


Figure 35. Bogie one primary suspension on curved track

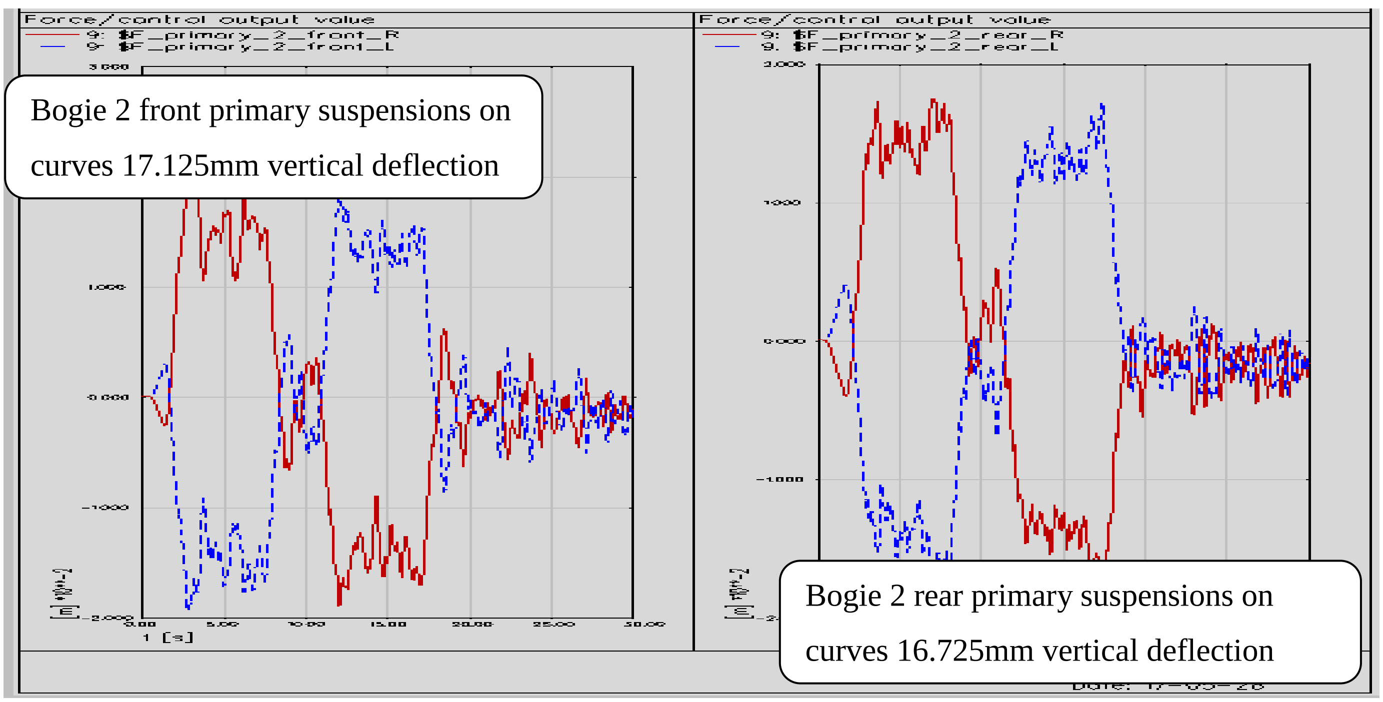


Figure 36. Bogie two primary suspensions on curved track

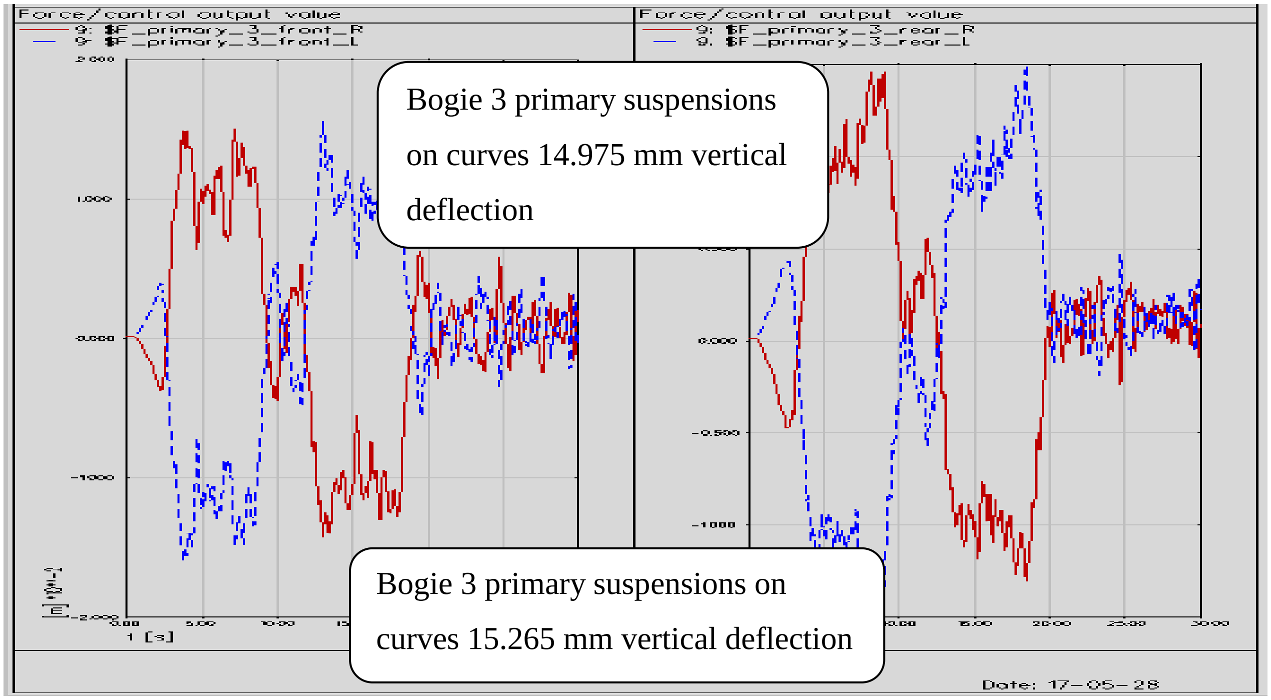


Figure 37. Bogie three primary suspensions on curved track

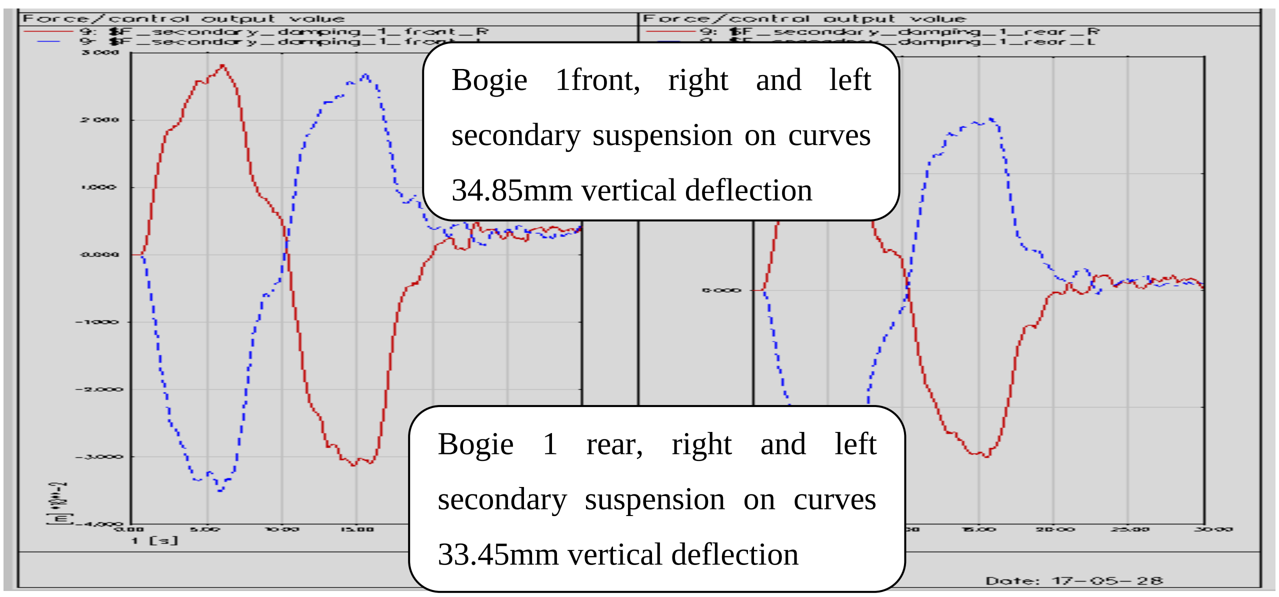


Figure 38. Bogie one and two secondary suspension in curved track

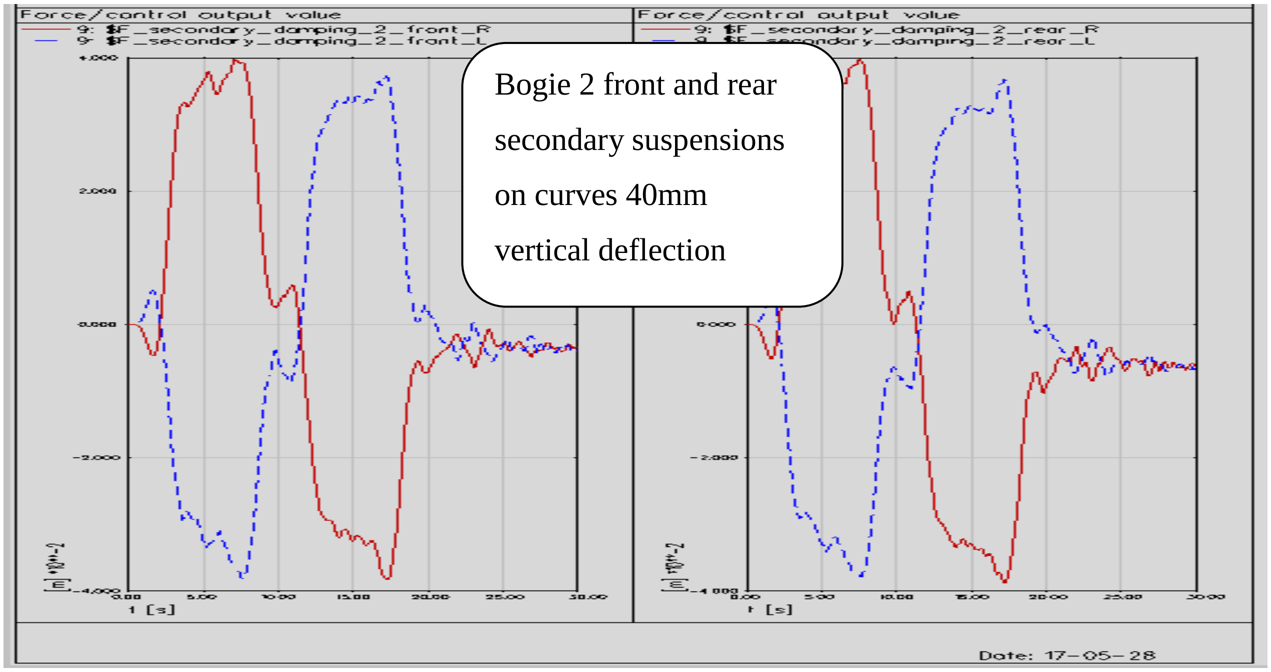


Figure 39. Bogie two secondary suspensions

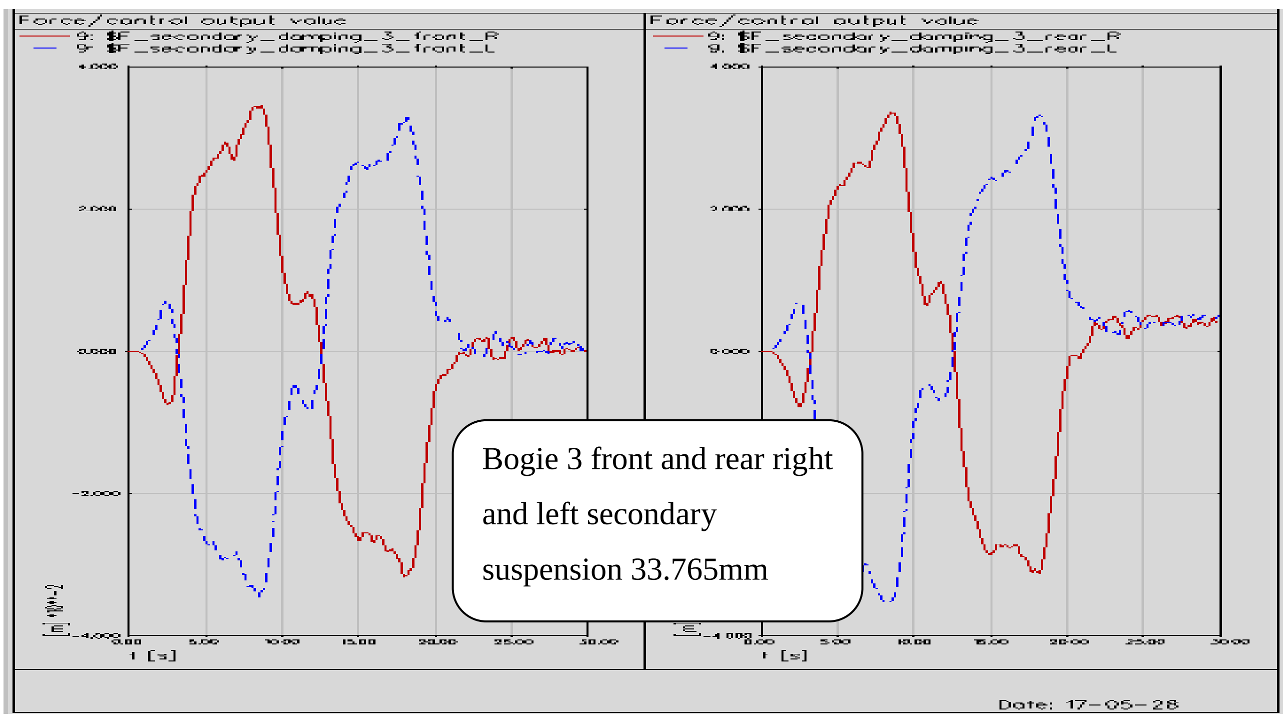


Figure 40. Bogie three secondary suspensions in curved track

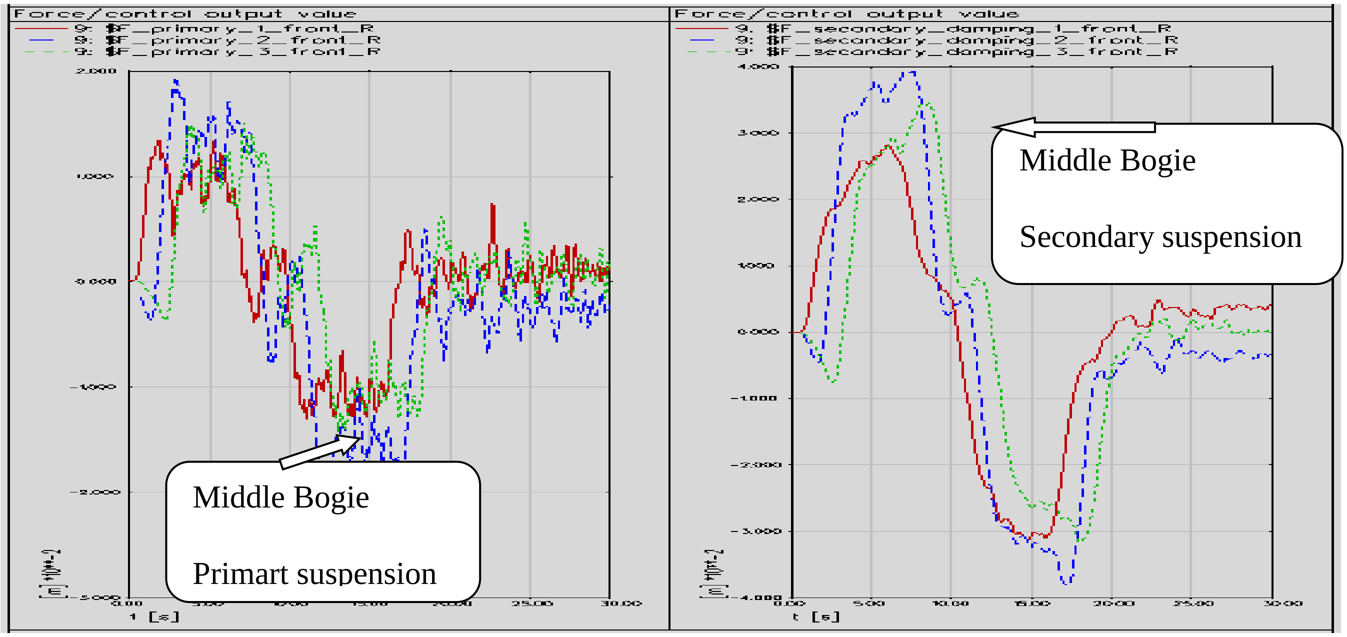


Figure 41. Bogie 1, 2 and 3 primary and secondary suspension vertical deflection deviation

Figure 38, 39 and 40 shows secondary vertical suspension at maximum load condition on minimum radius curved track condition with maximum vertical displacement of 34.85mm, 40mm and 33.765mm respectively. As discussed above the maximum deflection occurred in the middle bogie. The difference of the three bogie vertical suspension shows in figure 41. Therefore the middle bogie encountered by different dynamic condition such as weight transfer effect due to lateral displacement of wheel set, centripetal and centrifugal effect on the wheel set and aerodynamic effect on the car body.

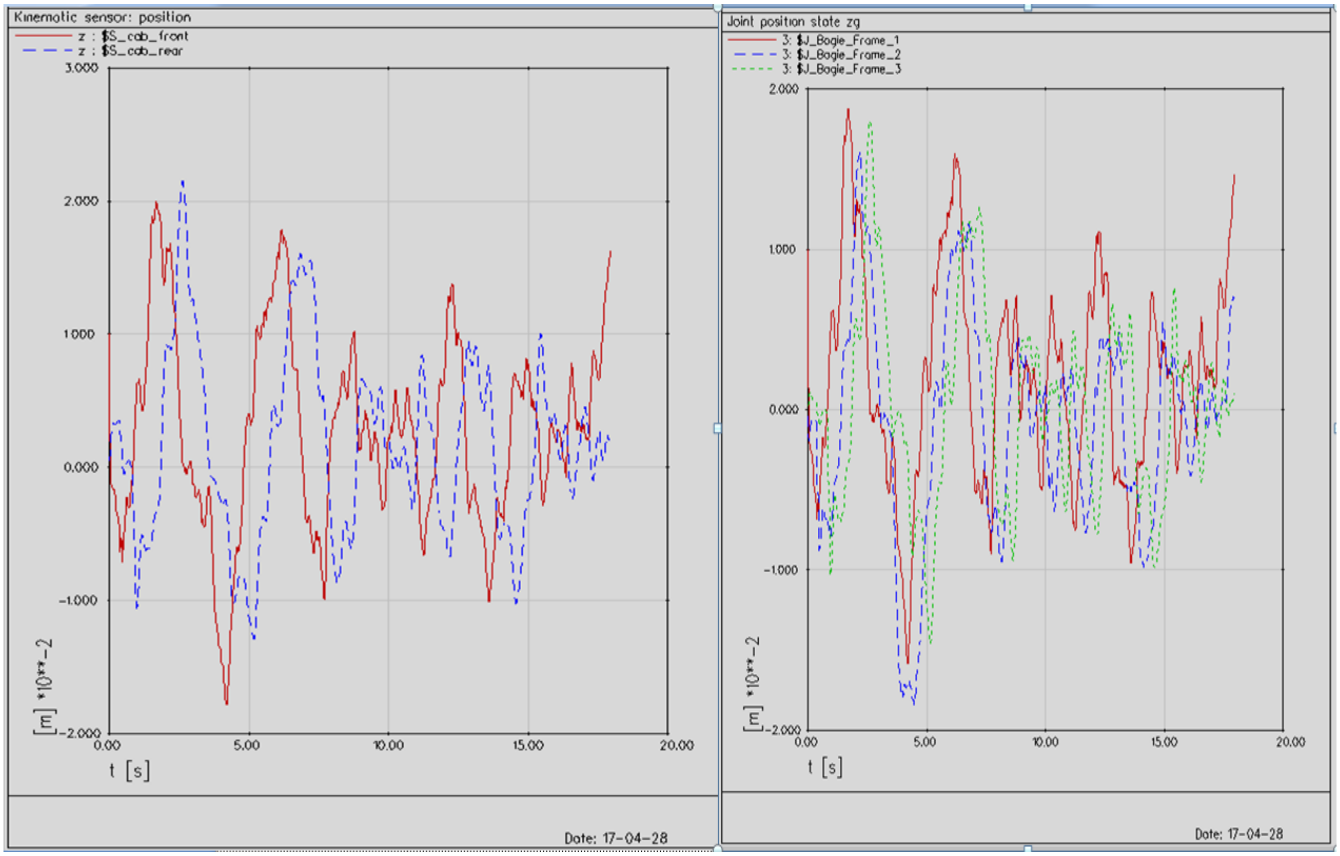


Figure 42. Each bogie frame and each car body module vertical deflection at the stated condition

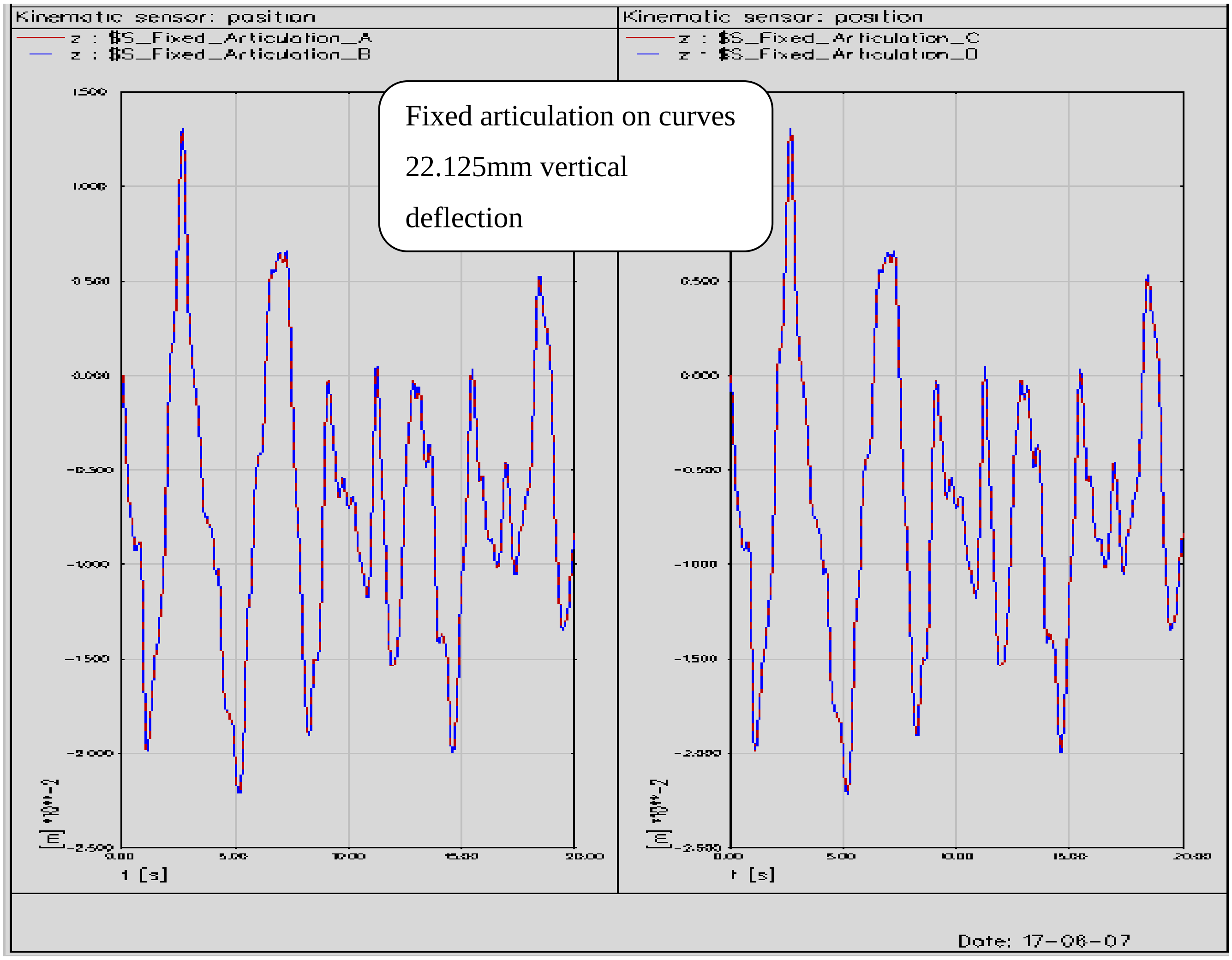


Figure 43. Front and rear articulation vertical displacement

For new vehicle, the articulation systems found 120mm from top of rail. At static measurement the specifications shows that the vehicle can be work until 85mm from top of rail. But due to maximum load and curving effect, the primary and secondary suspension vertical deflection the articulation system found 97.875mm from top of rail.

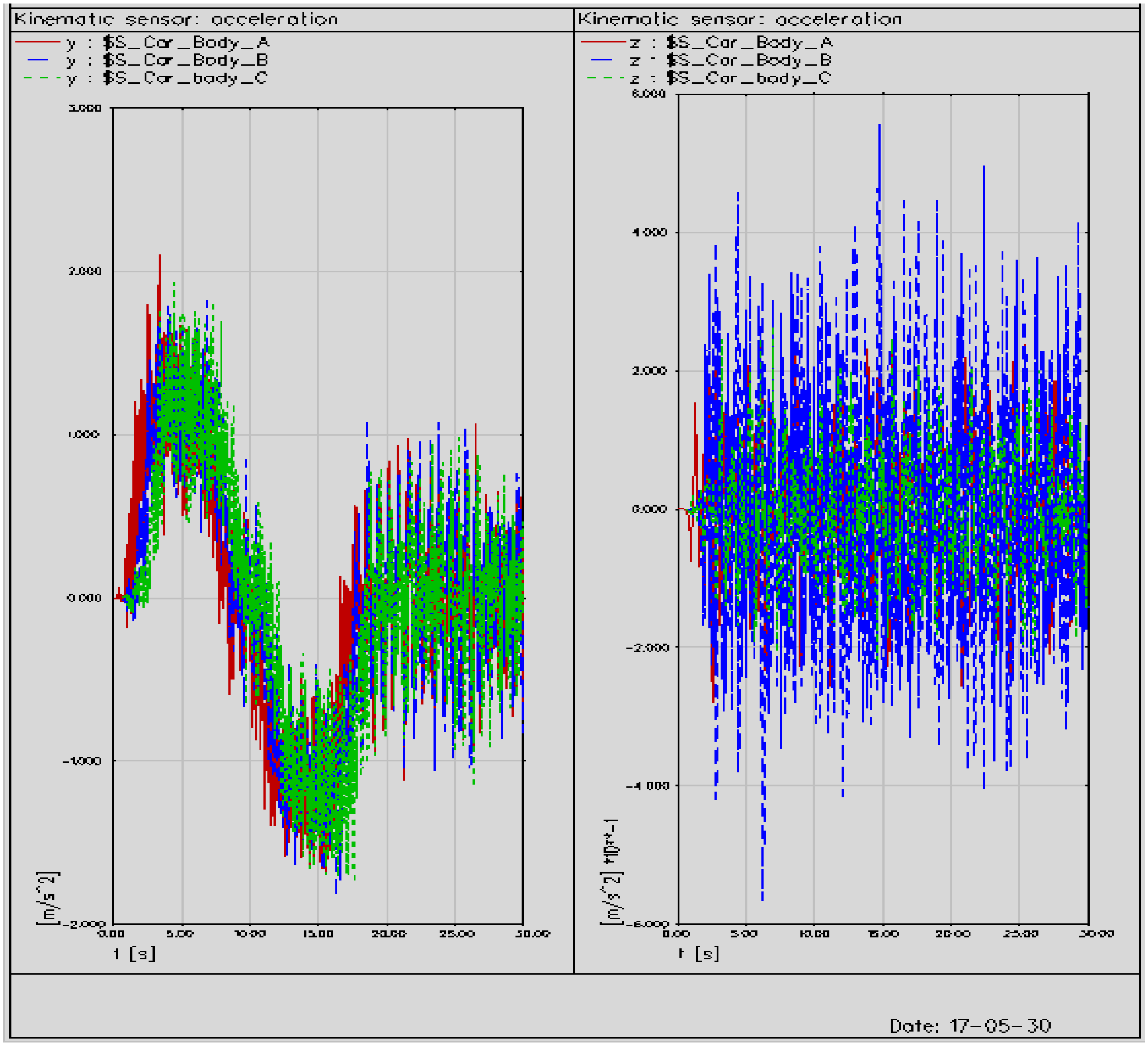


Figure 44. Car body lateral and vertical acceleration

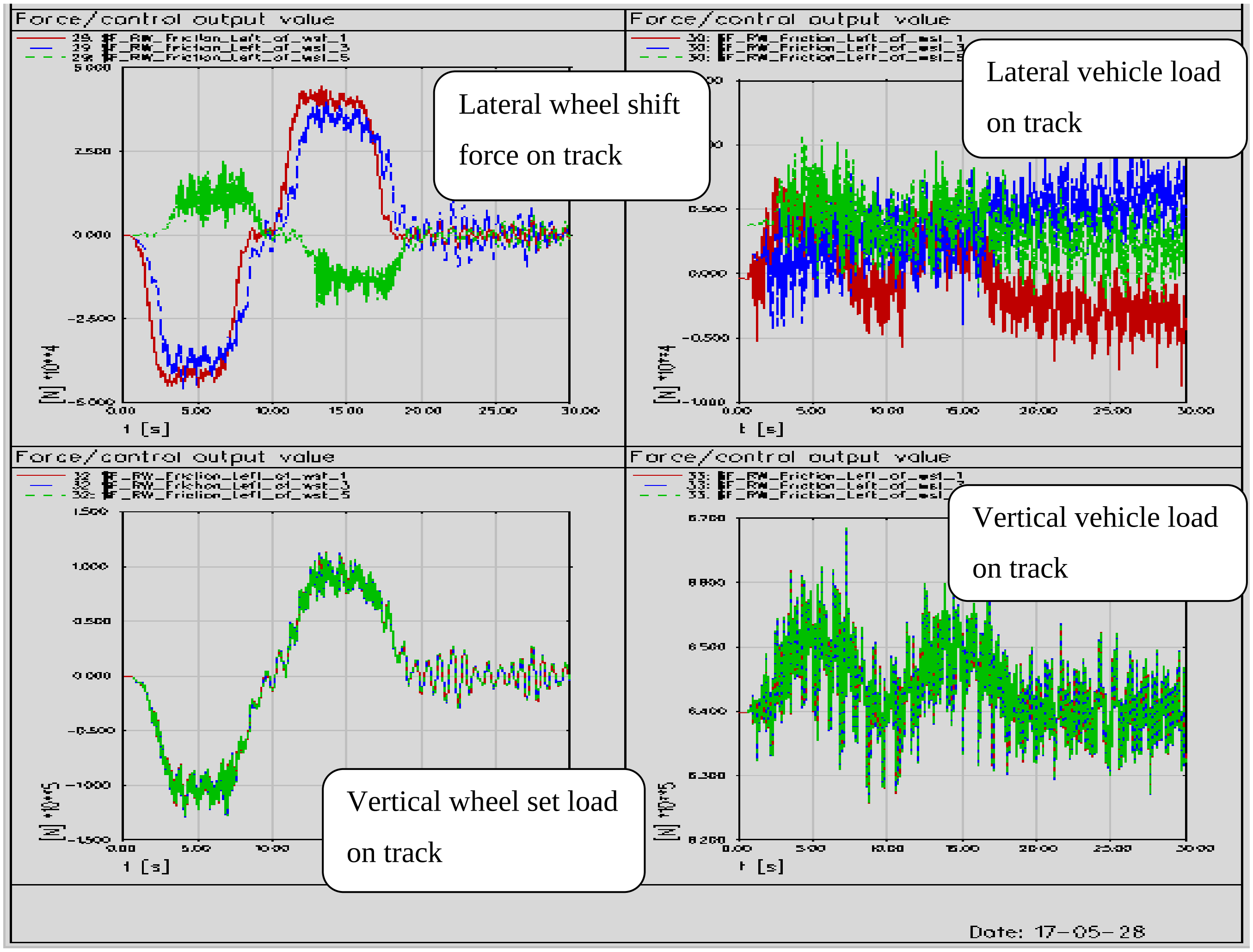


Figure 45. Lateral wheel shift force and Vertical force on leading wheel set

Generally, the primary and secondary suspension, bogie, car body and fixed articulation maximum vertical deflection were taken place on minimum radius horizontal curved track. This shows that, the rail curvature has many effects on the railway vehicles. Even though the result was below the limit, the middle bogie was maximum primary and secondary vertical deformation due to many dynamic effects on it. These maximum vertical deflections facilitate the lowering down of articulation folding bellows towards the rail. According to the dynamic simulation result shows that the mass of the vehicle, the minimum horizontal curve radius and suspension stiffness and damping values were the main cause for the stated problem. Among those causes the controllable variable was stiffness and damping value for the vertical primary and secondary suspension systems.

Therefore, the primary vertical suspension damping was optimized by 35% persent and secondary vertical suspension damping by 28% stiffness value were optimized to with stand the vehicle body lowering down conditions. The following graphs shows suspension system vertical deformation after stiffness and damping value optimized.

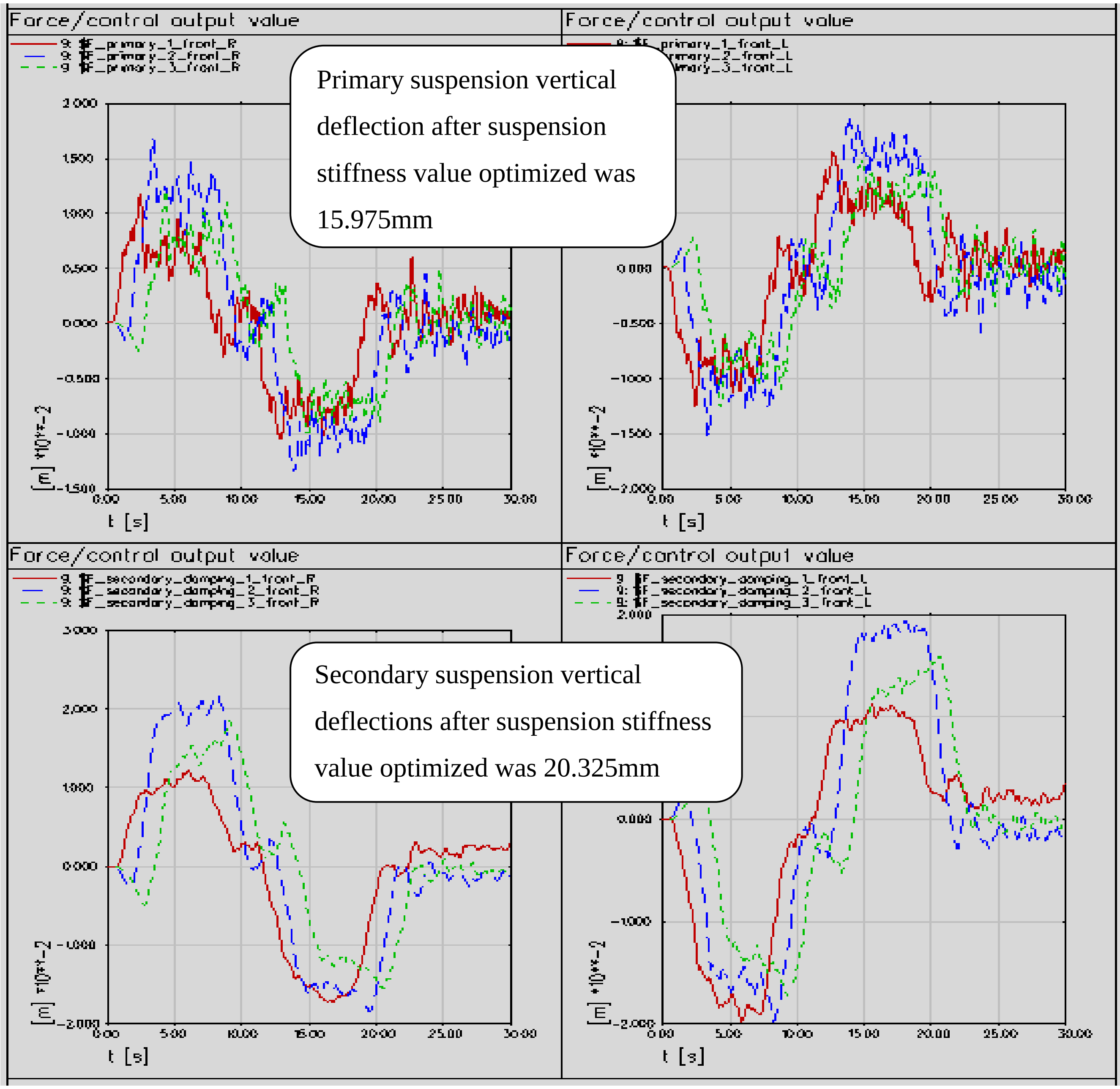


Figure 46. After optimization the primary and secondary suspension vertical deflections

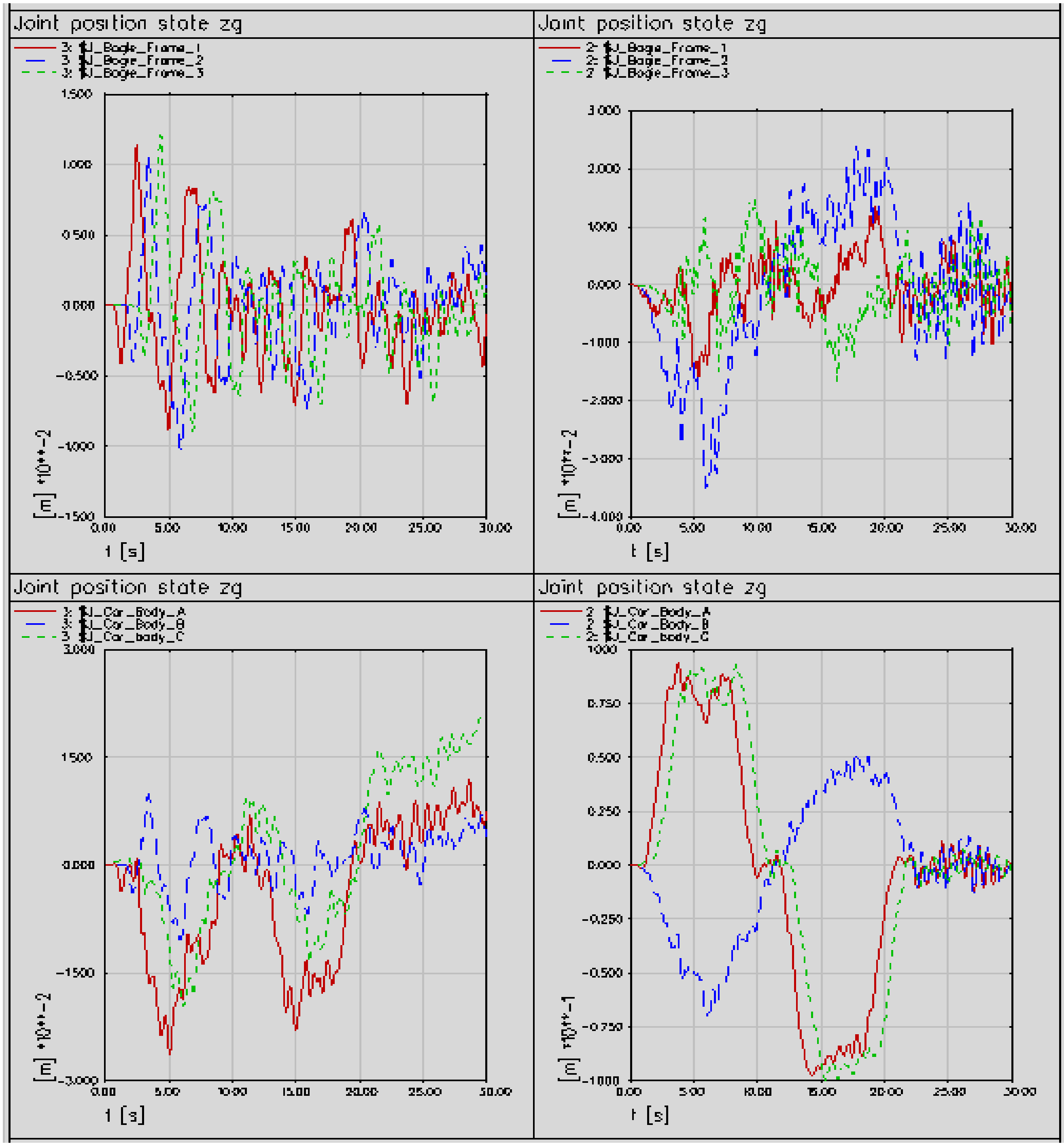


Figure 47. After suspension stiffness value optimization bogie and car body vertical deflections

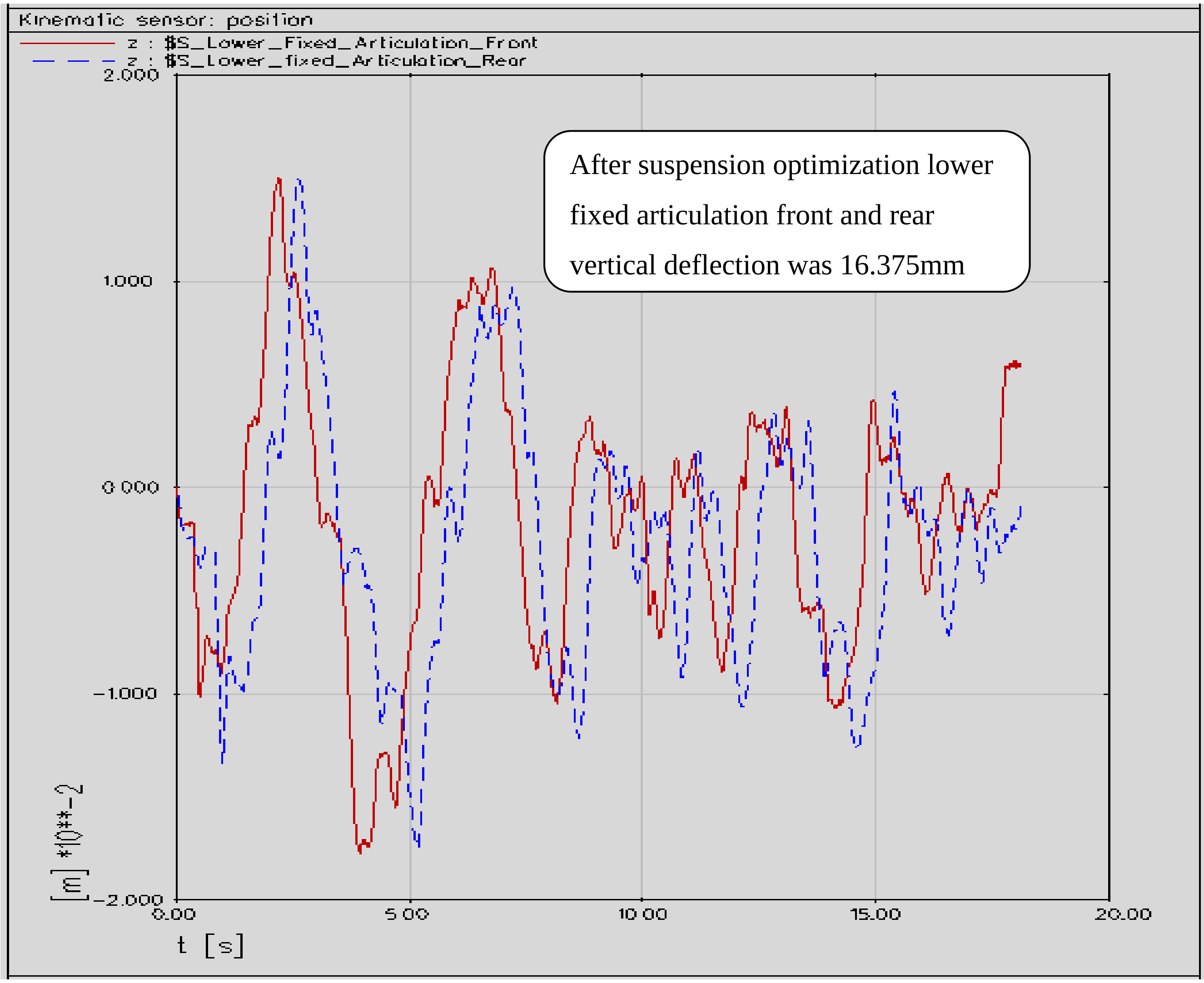


Figure 48. After suspension optimization lower fixed articulation front and rear vertical deflection

4.5. Parameter variation result

For the dynamic multi-body simulation, parameter variation were carried out in order to show the effect of vehicle mass, critical speed and wheel radius on vertical position of vehicle body.

The following consecutives graph shows the parameter variations on vehicle mass, wheel radius and critical speed.

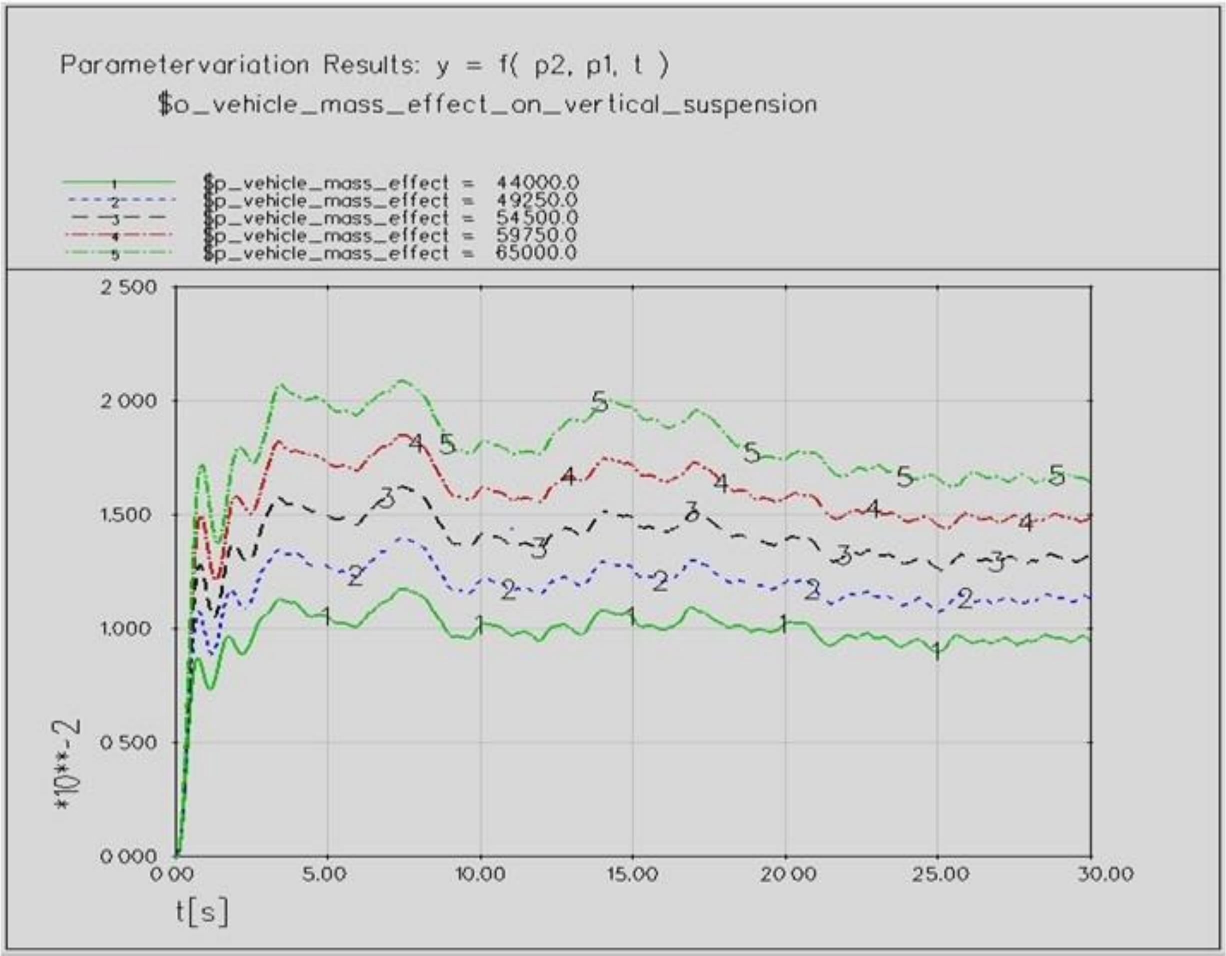


Figure 49. Vehicle mass effect on vertical suspension

Figure 49 shows that, while the mass of the vehicle increases from empty load to maximum load condition the vertical suspension continually compressed. This means that the mass of the vehicle have direct effect on the suspension system.

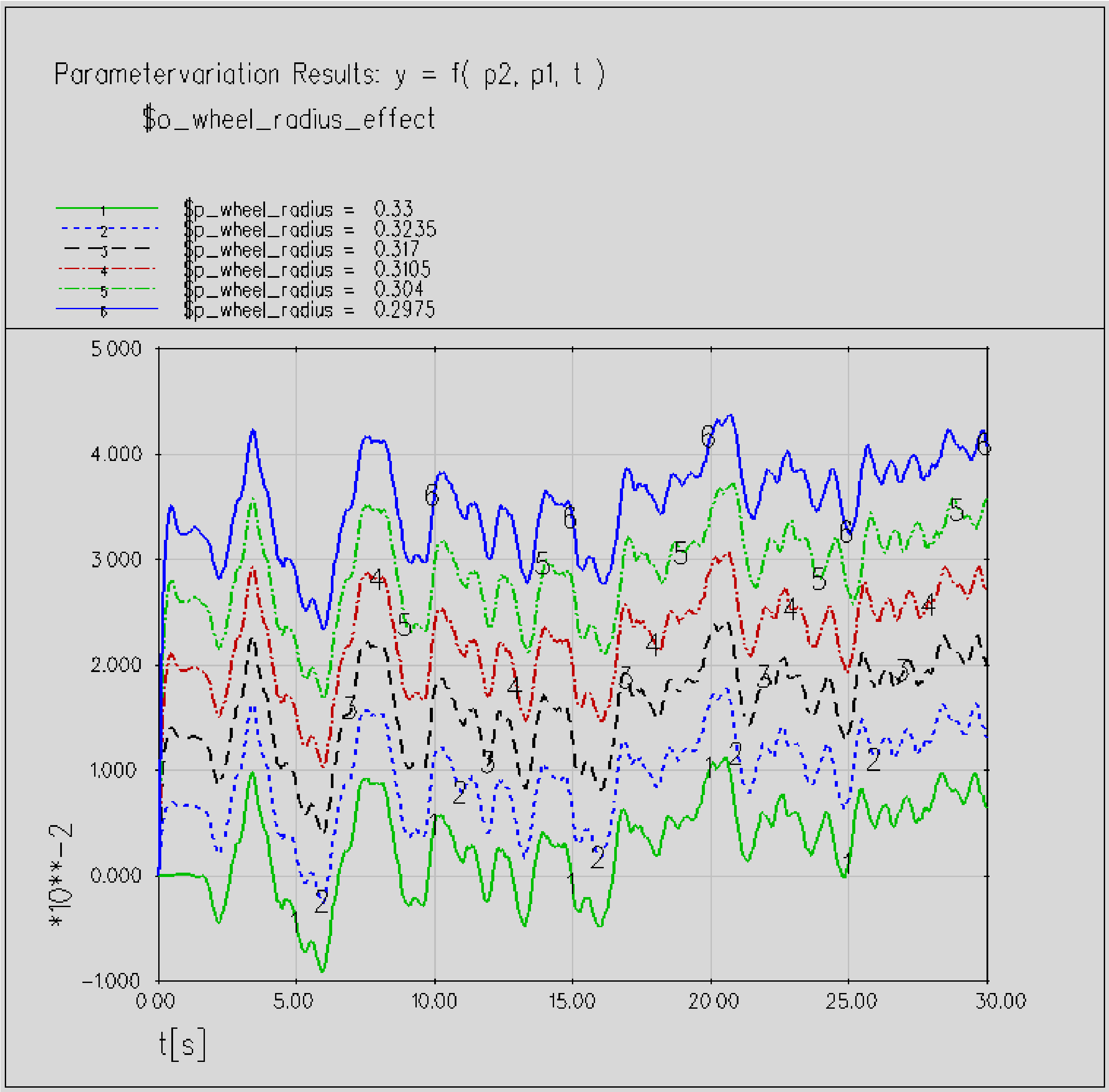


Figure 50. Wheel diameter variation effect on vehicle vertical position

As shown in this figure, the wheel diameter variation from new wheel (660 mm) to 590mm the suspension system continually was lowering down to the rail. In addition to lowering down effect, it makes running instability effect on the vehicles.

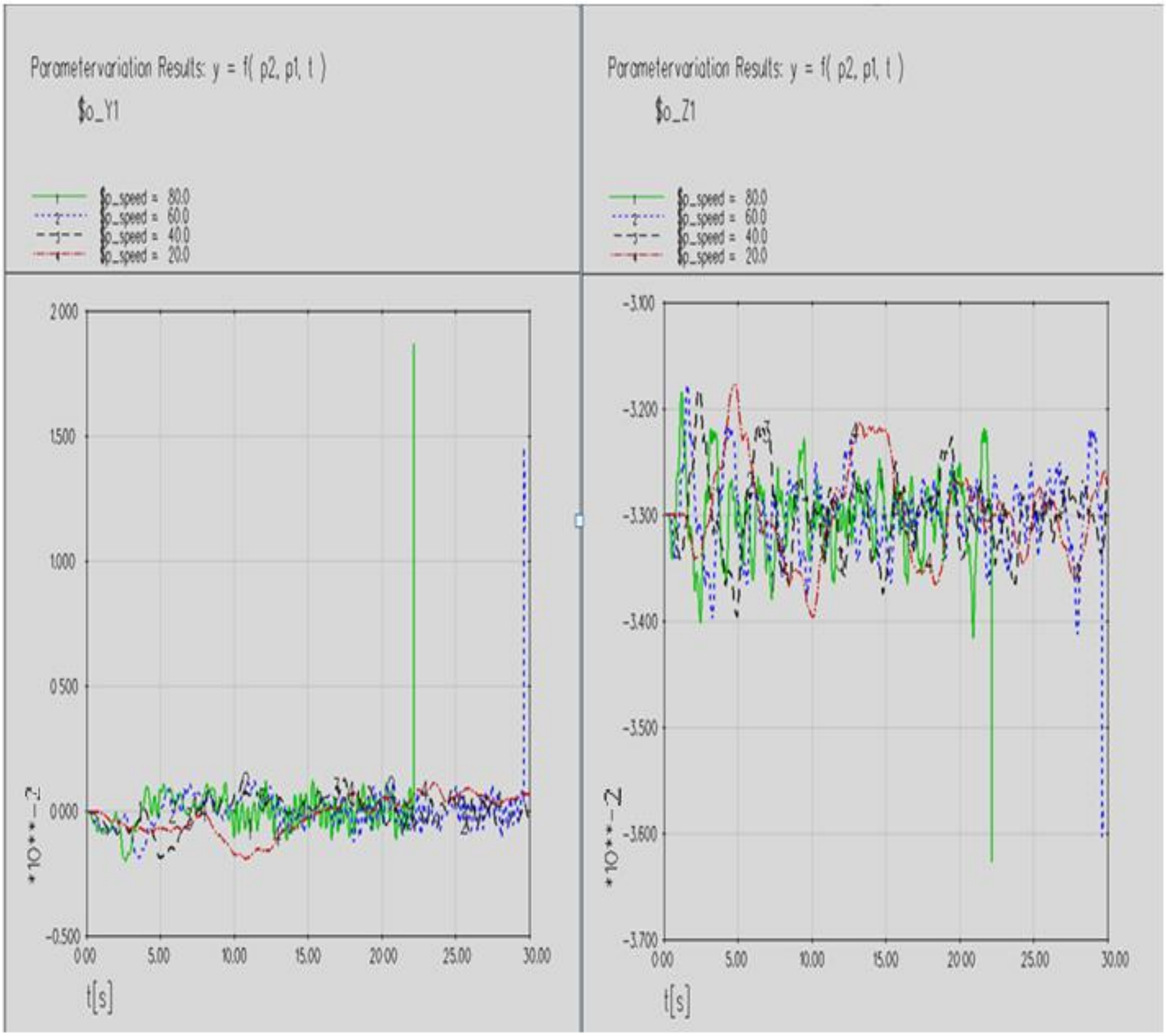


Figure 51. Critical speed variation on lateral and vertical position

While the critical speed increases, the lateral and vertical displacement increases. At minimum horizontal radius curved track critical speed is the basic parameter in dynamic analysis and multi-body simulation.

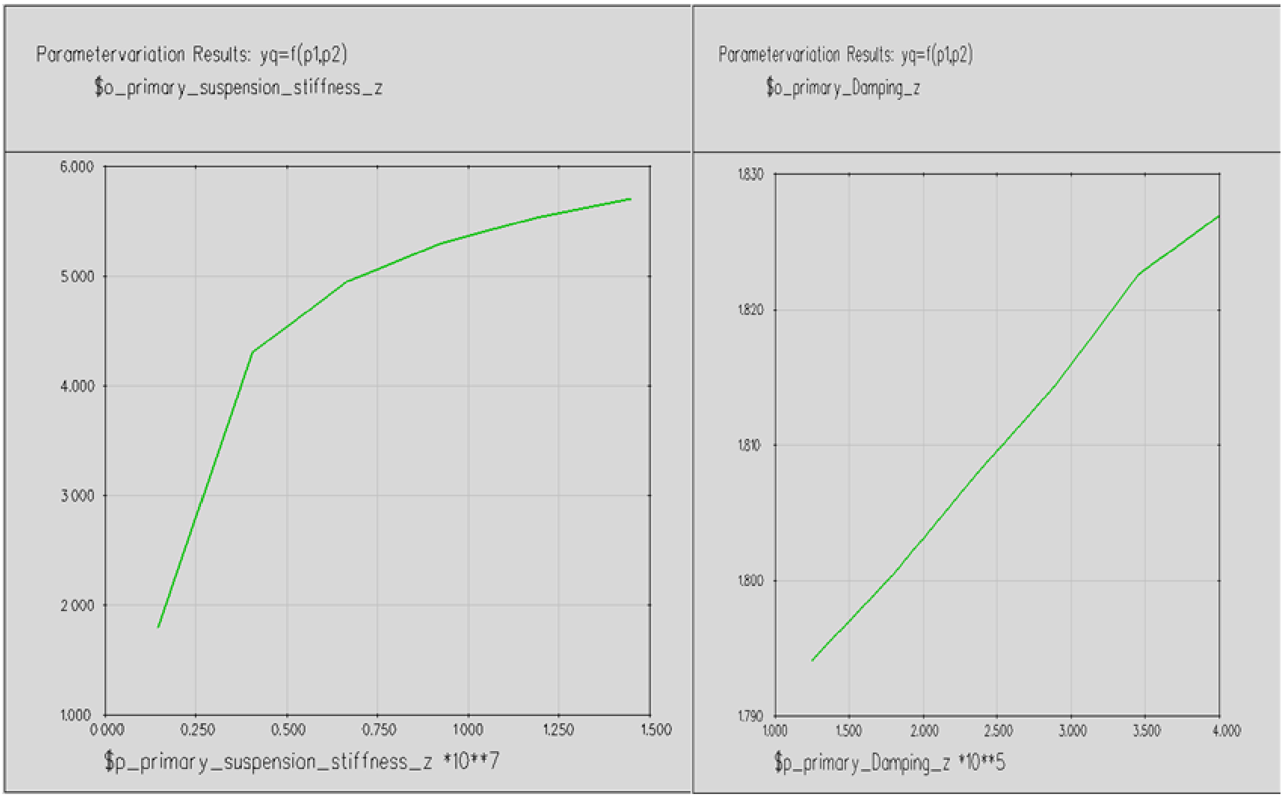


Figure 52. Primary suspension stiffness and damping value optimizations

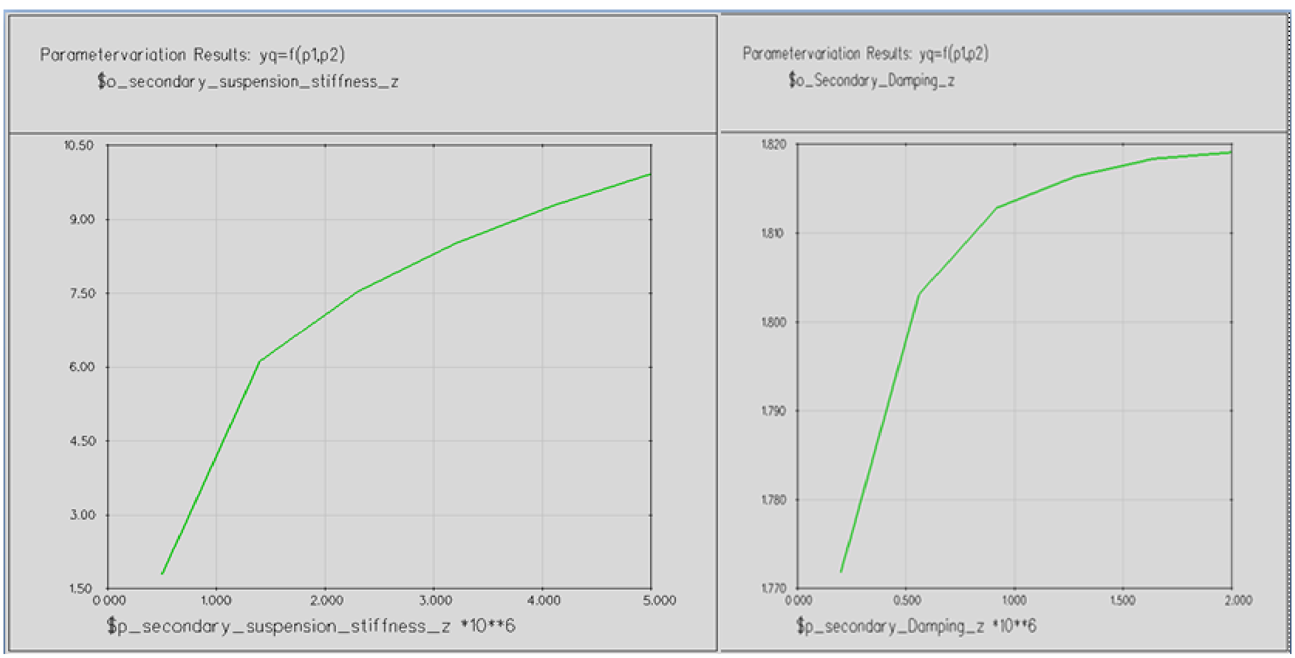


Figure 53. Secondary suspension stiffness and damping value optimizations

Table 3. Vertical primary and secondary suspension stiffness value

No	Items	Currently used	After optimization
1	Primary vertical suspension stiffness value (cz)	1450000	1957500
2	Primary vertical damping value	125000	168750
3	Secondary vertical suspension stiffness value (cz)	500000	640000
4	Secondary vertical damping value	400000	512000

4.6. Dynamic performance Evaluation

The performance of railway vehicles is often defined in terms of safety and productivity. Safety relating to the vehicle-track system is mainly concerned with derailment conditions. Derailment occurs for various reasons, including equipment or component failures, poor quality of the track structure, im-proper train-handling and train make up, as well as abnormal dynamic conditions arising from hunting and poor curve negotiability situations [18]. To avoid derailment and to ensure safety limits in terms of certain indexes are established. For the dynamic performance evaluation the following two points are the main conditions.

4.6.1 Ride index

The dynamic performance of a passenger railway vehicle as related to safety and comfort were evaluated in terms of specific performance such as ride index [13]. This ride quality will interpret as the capability of the passenger railway vehicle suspension to maintain the motion within the range of human comfort and Sperling's ride index which is a measure of the ride quality and ride comfort used by Ethiopia Railway Corporation. This Running safety and ride comfort are essential elements in the analysis of the passenger railway vehicle dynamic analysis and they have to be taken into account in modeling and assessment of the vehicle running behavior. The dynamic behavior of such vehicle also depends on the passenger load and the mechanical systems, such as primary and secondary suspension system and the rail irregularity etc, which interact with the wheels, the vehicle body and bogies. According to[13, 14], UIC 518 code, EN 12299 and ISO 2631 horizontal and vertical the result of this dynamic simulation was validated. The average horizontal and vertical ride index before optimization was 2.56 and 2.46 respectively and after optimization the average horizontal and vertical ride index were 2.6357 and 2.6954 respectively.

According to EN 12299 railway ride index standards these ride index were satisfactory. The following figure shows the three car body module lateral and vertical ride index before and after suspension stiffness and damping value optimization.

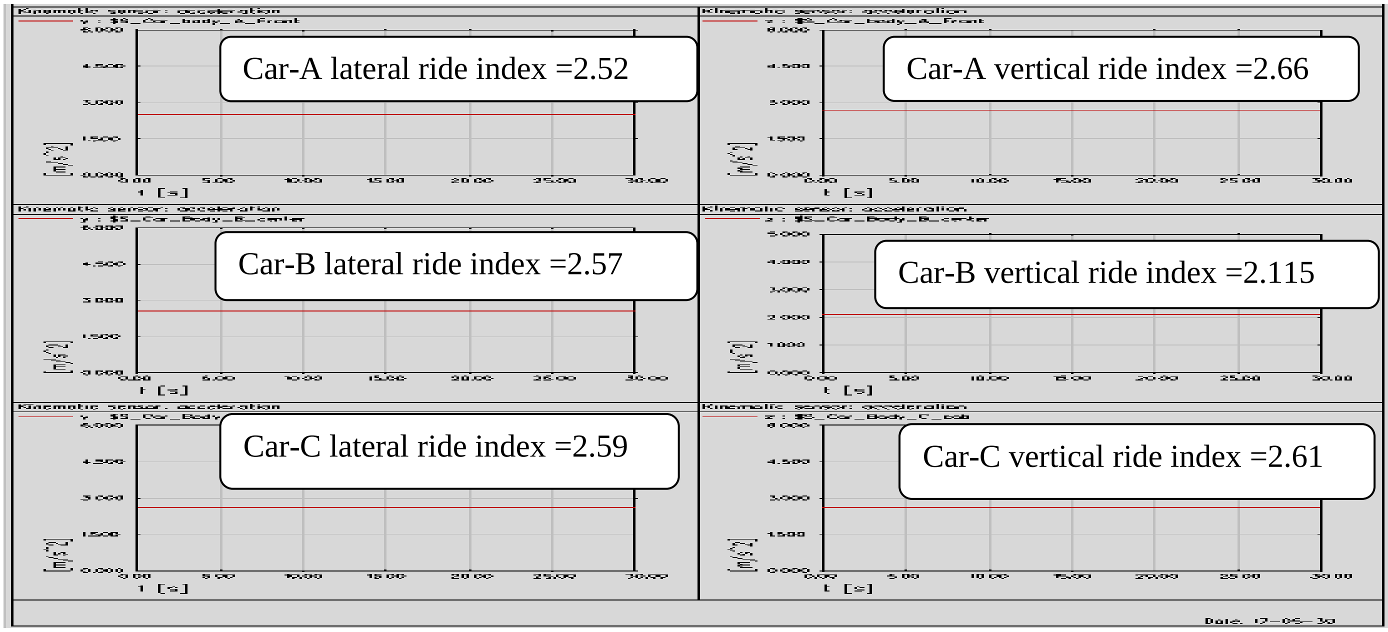


Figure 54. Ride index of each car body module

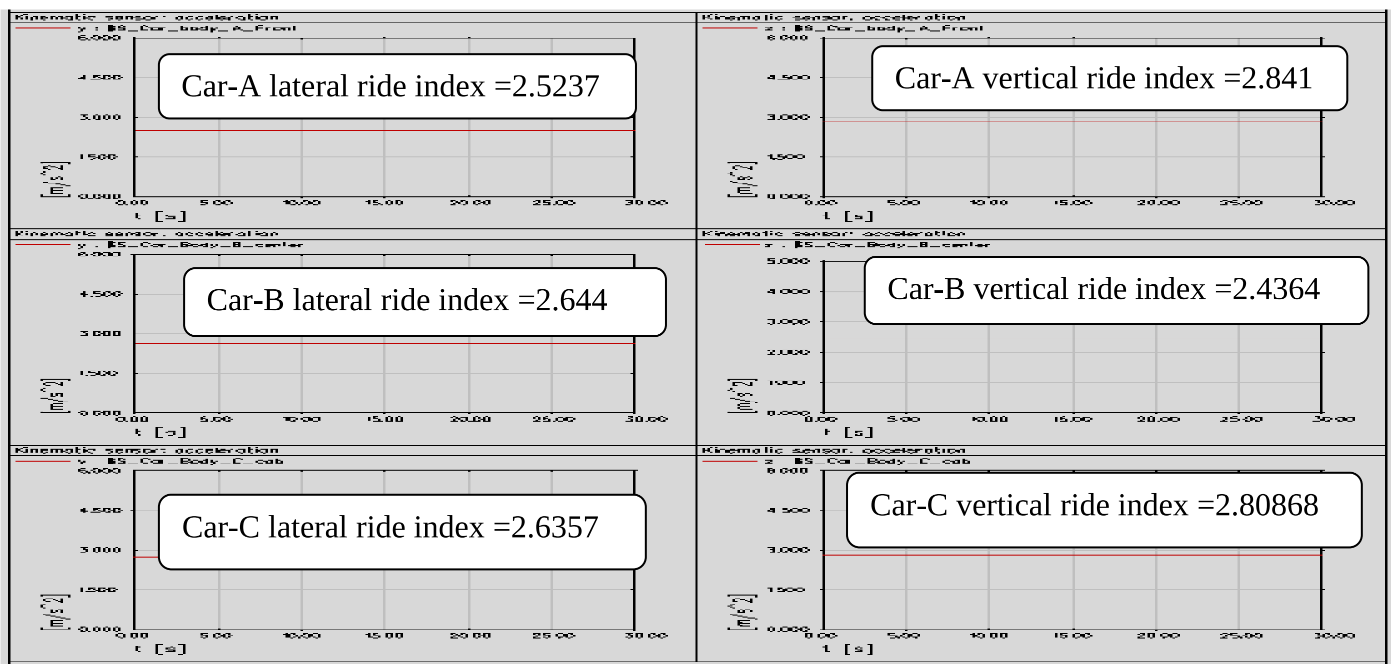


Figure 55. After suspension optimization each car body ride index

4.6.2 Derailment coefficient

One of the main scenarios of derailment is realized when during the vehicle motion a large lateral force acting on a wheel set leads to the wheel flange contact with the rail, as a result flange climbs up the rail rapidly (especially after the contact angle attains its maximum value) and the wheel set derails. According to [14, 18] the Nadals criterion, the occurrence flange climb is related to the ratio Y/Q of the lateral to vertical force components at the wheel-rail contact. The main objective of this performance analysis is that to set the running safety condition, such as ride comfort and derailment of the railway vehicle according to UIC 518 code and EN 12299 specifications. The derailment coefficient on the wheel rail contact before and after suspension optimization was 0.765 and 0.7875 respectively. It was below 0.8 and fulfills UIC 518:2005, and EN 12299 specification.

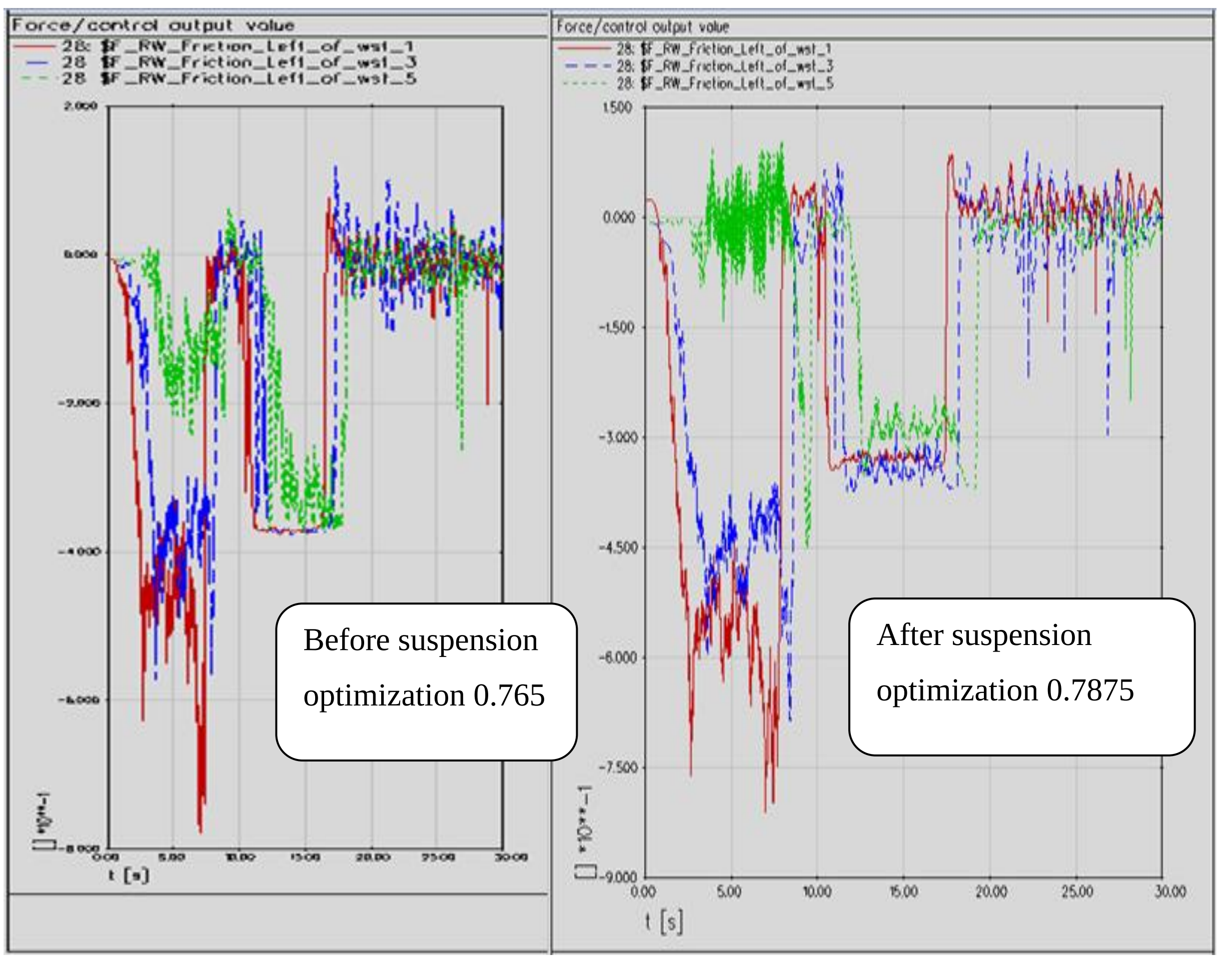


Figure 56. Derailment coefficient of leading wheel set

CHAPTER FIVE

5.1. Result Discussion

The simulations result as shown in chapter four have been performed after 3D modeling by SIMPACK MBS software package for each parameter according to the methodology. The primary suspension, secondary suspension, car body, front and rear fixed lower articulation in straight and curved track simulation results almost all fit with the hypothesis for the stated currently happen problem. The maximum vertical deformation for primary suspension, secondary suspension and front and rear fixed lower articulation were 5.235mm, 15.125mmand 12.35mm in straight track. All the results found in straight track were under the standards. For curved track maximum vertical deformation for primary suspension, secondary suspension and fixed lower articulation were 17.125mm, 40mm, and 22.125mm. Based on the results as shown, the middle bogie primary and secondary suspension, car body, front and rear fixed lower articulation highly compressed. These maximum deformation of middle bogie vertical suspension contribute for the vehicle body lowering down to the rail problem. To with stand the lowering down problem, primary and secondary vertical suspension stiffness and damping value was optimized. The vertical deformation of primary, secondary suspension and articulation system after suspension stiffness and damping value optimized were 15.975, 20.325and 16.375mm.According to railway standards such as UIC 518, UIC 519, s1002 and EN 14363:2005, the primary and secondary suspensions, the lateral shift force and vertical wheel loading for straight and curved track before and after optimization were satisfied the stated limit. The dynamic performance analysis conformed for ISO 2631 horizontal and ISO 2631 vertical. And also for the derailment coefficient Nadals criterion was satisfied. In the North South line, the vehicle runs 14/15 times per day, 420/450 time per month and 5040/5400 times per year. This repeated continuous motion at this minimum radius curved track have high effect on the middle bogie suspension systems to highly deformed due to maximum vehicle mass, minimum track curve radius, continuous lateral displacement, track super elevation, centripetal and centrifugal effect on the middle bogie are the main cause for the stated problem. Therefore, these continuous repetition motions on that minimum radius horizontal curved track facilitate the lowering of vehicle body to the rail. The optimized suspension and damping value can withstand this problem according to the simulation results.

Table 4. Simulation result and validation criteria

Value to be investigated	**Assessment criteria**	**Standards**	**Criteria limits**	**Simulation results**	**Status**
Primary suspension flexibility gap in straight track (mm)	Maximum vertical displacement	UIC 518	18 (17.5+0.5, -1.5)	5.235	✓
Primary suspension flexibility gap in curved track (mm)	Maximum vertical displacement	UIC 518	18 (17.5+0.5, -1.5mm)	17.125	✓
Secondary suspension flexibly gap in straight track (mm)	Maximum vertical displacement	UIC 518	44 (43±1 mm)	15.125	✓
Secondary suspension flexibility gap in curved track (mm)	Maximum vertical displacement	UIC 518	44 (43±1 mm)	40	✓
Articulation system in straight track from TOR (mm)	Maximum vertical displacement	UIC 518, UIC 519	≥85	12.35	✓
Articulation system in curved track TOR (mm)	Maximum vertical displacement	UIC 518, UIC 519	≥85	22.125	✓
Leading wheel set Lateral shift forces in straight track (KN)	Maximum wheel rail force in lateral direction	UIC 518, EN 14363:2005	60	1.075	✓
Leading wheel set Lateral shift forces in curved track(KN)	Maximum wheel rail force in lateral direction	UIC 518, EN 14363:2005	60	4.675	✓
Leading wheel set vertical forces in straight track(KN)	Maximum wheel rail force in vertical direction	UIC 518, 14363:2005	145	1.25	✓
Leading wheel set vertical	Maximum wheel rail	UIC 518,	145	6.4	✓

forces in curved track(KN)	force in vertical direction	EN 14363:2005			
Lateral car body acceleration (m/s^2)	Maximum acceleration in lateral direction	UIC 518	2	1.835	✓
vertical car body acceleration (m/s^2)	Maximum acceleration in vertical direction	UIC 518	3.2	0.573	✓
Horizontal ride index	Maximum value in lateral direction	ISO 2631 horizontal	2.5	2.362	✓
Vertical ride index	Maximum value in vertical direction	ISO 2631vertical	2.5	3.235	✓
Derailment coefficient	Maximum Ratio of lateral to vertical forces	UIC 518	0.8, 1.2	0.735	✓

To some up the MBS result as had had discussed above and Table.4. Simulation result and validation criteria show that, the primary suspension, secondary suspension, articulation systems vertical deformation have great rule for the stated problem at basic railway parameters, especially on middle bogie at curved track condition. Therefore, dynamic analysis with multi-body simulation of articulated passenger railway vehicles are very crucial to identify the main cause of the lowering down of vehicle body to the rail.

5.2. Conclusion

To summarize this work, the maximum deformation on vertical primary and secondary suspensions, the articulation systems, the vehicle body and bogie were found on minimum horizontal curved track at maximum load conditions. The main causes of the vehicle body lowering down problem were the vertical primary and secondary suspension maximum deformation, maximum vehicle load, wheel radius variation and minimum horizontal curved track. Especially for the middle bogie vertical primary and secondary suspension maximum deformation were facilitate for vehicle body lowering down problem. Therefore, the optimized suspension stiffness and damping value can withstand this vehicle body lowering down problem. ERC have to take the measure action on vehicle head time to minimize passenger load, wheel re-profile process to minimize vehicle running instability and the suspension system stiffness optimization to withstand the maximum load. This paper is not the completion of identifying the main cause of the stated problem. It has many futurities in relation to car body and bogie flexibility, rotational effect, standing people movement of inertia and other effects for the articulation folding bellows lowering down problem.

5.3. Recommendation

According to dynamic multi-body simulation of articulated passenger railway vehicles, shows that the mass of vehicle, the suspension system stiffness value, critical speed, curve radius and wheel diameter re-profile system have much contribution for the lowering of articulation folding bellows to the rail. At maximum vehicle load the suspension system of middle bogie highly compressed and affect the articulation folding bellows height from top of rail which lead the stated problem.

- Even though the result were found for straight track below the limit value, at minimum radius curved track much near to the limit value for the middle bogie. Therefore to alleviate this problem, primary and secondary suspension stiffness values have to be optimized to withstand the stated problem in curved track.
- Additionally, to reduce the maximum passenger load set vehicle hade time as constant as possible.
- The dynamic instability may arise after wheel re-profile process had been carried out, so that the re-profiling of wheel must be fine and exact process.
- Predominantly at curved track with radius 50m, primary and secondary suspension, car body module, and front and rear articulation systems highly compression and relaxations process. Therefore, at those continuous process, the articulation folding bellow losses its attachments.

5.4. Future Work

Dynamic analysis and multi-body simulation for the development of advanced passenger railway vehicles is a complex research field that requires new ideas and innovative design solutions. Therefore, the study of railway dynamics will not finish relating with the large challenges can be faced by the research efforts in the analyzing and improving passenger rail vehicle dynamic analysis and multi-body simulation. But the future work proposed by the end of this work for the improvement of the passenger vehicle articulation folding bellows lowering down to the rail and enhancement methodologies, can be summarized in the following points:

- The present study dynamic analysis and multi-body simulation done on a model developed by SIMPACK rigid car body modules, rigid bogie frames, axles and wheels are rigid body module. To get more accurate result finite element multi-body simulation technique including car body flexibility is other research area.
- Bogie and car body module rotational twisting effect for the articulations are not considered in the current study therefore, by considering those effect for the vehicle lower down is another engineering problem which needs to be study.
- The mass of passenger load and movement of inertia considered as rigid mass movement of inertia but around 81% of the passengers are standing, this have great effect on movement of inertia tensors to leads for instability of the vehicles. Therefore, this also researchable area.
- Perform the parametric variation analysis including effect of cross wind force effect on the suspension in straight and curved track also another research area.

Reference

[1].Training manual and mantainance procedure for 70% low-floor through passage of ethiopia light railway project project.

[2]. Wikense, A.H. Fundamental of Railway Vehicle Dynamics: Guidance and Stability/ Loughborough university, uk : swets and zeitlinger , 2003. Advance.

[3]. Critical speed analysis of railcars and wheelsets on curved and straight track. Moody, joanna charlotte. An honors thesis, s.l. : spring , 2014.

[4]. Multibody approach for dynamic analysis,department of mechanical and material engineering. Shaltout, ramy elsayed. Doctor of philosophy in mechanical engineering, valencia : s.n., marzo de 2013.

[5].Towards active car body suspension in railway vehicles, *department of signals and systems*, Chalmers University of technology, 2009.

[6]. Development and On-Track Tests of Active Vertical Secondary Suspension for Passenger Trains, Thesis Stockholm, Sweden 2014

[7]. Characteristic parameters nonlinear wheel/rail contact geometry, proceeding of 21st IAVSD symposium Stockholm, bombardier transportation, August 2009.

[8]. Analysis of Rail Vehicle Suspension Spring with Special Emphasis on Curving, Tracking and tractive Efforts, 2016 IOP Conf.

[9]. Methods for Reducing Vertical Car body Vibrations of a Rail Vehicle, Report in Railway Technology Stockholm, Sweden 2010

[10]. Dynamic analysis of railway vehicle/track interaction forces, Doctor of Philosophy of the Lough borough University of Technology June 1986

[11]. Design Data on Suspension Systems of Selected Rail Passenger Cars, U.S. Department of transportation Federal Railroad Administration, July 1996.

[12]. Multi-body modelling of a multi-articulationtramway in simpack v9 and research into the influence of rail-excitations on forces acting in the yaw dampers, Fernández, margarita mármol. Master thesis-nr.:816, october 28, 2014.

[13]. Railway applications -Testing for the acceptance of running characteristics of railway vehicles - Testing of running behavior and stationary tests, European Standard EN 14363:2005

[14]. Railway applications - Track - Track alignment design parameters - Track gauges 1435 mm and wider - Part 1: EN 13803-1June 2010.

[15]. Dynamic forces between the rails and the wheels of railway vehicle, Feb. 2, 2016.

[16].Towards optimization of a high speed train bogie primary suspension, Chalmers University of technology, master's thesis, Sweden 2013

[17]. Passive Components in Active Suspension System, Arun Arora, Master of Science Thesis, December 2012

[18].Bezin, Yann, Funfschilling, C., Kraft, S and Mazzola, L. (2015) Virtual testing environment tools for railway vehicle certification. Proceedings of the Institution of Mechanical Engineers, Part F: Journal of Rail and Rapid Transit, 229 (6). pp. 755769.ISSN 09544097

[19]. A new approach to define criteria for rail vehicle model validation, china August 2013.

[20]. SIMPACK Reference Guide SIMPACK Release 8.9, September 1, 2010/SIMDOC v8.904

[21]. Simulation of running dynamics for vehicle acceptance: application and validation, international journal of railway technology, August 2014.

[22]. Modelling of suspension components in a rail vehicle dynamics context. Brunia, stefano. Politecnico di milano : taylor & francis, 24 jun 2011.

[23]. The use of a linear half-vehicle model for the optimization of damping in the passive suspension system of a rail way car, Warsaw university of technology, faculty of transport, Warsaw, Poland, volume 39, issue 3, 2016.

[24]. Influence of the vertical suspension on the vibration behavior in the railway vehicles, politehnica university of Bucharest, (2011)

[25]. Modeling and control of railway vehicle suspensions, Loughborough University
Institutional Repository, 2007

[26]. Dynamic analysis of high speed railway bridge under articulated trains. Northen jiotong university : s.n., julay, 2003.

[27]. Integrated tilt and active lateral secondary suspension control in high speed railway vehicles, Loughborough University Institutional Repository, 2134/6964

[28]. Multibody dynamics, computational methods and applications. New york : springer science business media b.v, 2011, vol. Volume 23.

[29]. Analysis of the dynamic response in the railway vehicles to the track vertical irregularities. Part i: the theoretical model and the vehicle response functions. Dumitriu, m. S.l. : 2015 kavala institute of technology, 18 november 2015.

[30]. Parametric analysis of rail vehicle parameters influencing ride behavior. Mullana, india. S.l. : department of mechanical engineering maharishi markandeshwar university, vols. Vol. 3, no. 8.

[31]. Advanced multibody dynamics advanced multibody dynamics system. Ballew, brent s. Master thesis, april 28, 2008.

[32]. Multi-body simulation of influence of bogie interconnection on vehicle-track interaction. Pavel krulich, jan čapek. 2014.

[33]. DJ1-The Electric Freight Train locomotive for china

.

Appendix

Appendix. A. **some input data for the MBS software package**

Table 5. Load parameter

Working condition	Seating capacity	Standing capacity	Total capacity(person)	Total weight (t)
W_o	0	0	0	44
W_1	65	0	65	47.9
W_2	65	189	254	59.2
W_3	65	252	317	63

Table 6. Basic parameter

No	Basic parameter	Value
1	Track gauge	1.435m
2	Bogie base	10.4m
3	Wheel set base for motor bogie	1.9m
4	Wheel set base for trailer bogie	1.8m
5	Bogie mass for motor bogie	5120kg
6	Bogie mass for trailer bogie	3120kg
7	Car body A or B mass (front and rear car body)	17500kg
8	Car body movement of inertia along x direction	4375
9	Car body movement of inertia along y direction	8750
10	Car body movement of inertia along z direction	8750
11	Car body B mass	12325
12	Car B movement of inertia along x direction	3081.25
13	Car B movement of inertia along y direction	6162.5
14	Car B movement of inertia along z direction	6162.5
15	Bogie movement of inertia along x direction	1250
16	Bogie movement of inertia along y direction	1870
17	Bogie movement of inertia along z direction	2182

18	Front and rear car body length	11.800m
19	Middle car body length	3.600m
20	Height car body including pantograph	3.750m
21	Width of car body	2.650m
22	Vertical mass center of Car body	-0.9m
23	Vertical mass center of bogie	-0.528m
24	Primary spring stiffness along x/y/z	5.8*10^5/5.8*10^5/1.45* 10^6
25	Primary damping ration along z	1.25*10^5
26	Secondary spring stiffness value along x/y/z	3*10^5/3.5*10^5/5x 105
27	Secondary damper x/y/z	0/2.0 x 105/4*10^5
28	Minimum radius of vertical curve	1000m
29	Minimum radius of horizontal curve	50m

British Railways Ride Quality and Ride Index

Table 7. Standard values of ride index

No	Ride Quality	Ride Index
1	Very good	1
2	All most very good	1.5
3	Good	2
4	Almost good	2.5
5	Satisfactory	3
6	Just satisfactory	3.5
7	Tolerable	4
8	Intolerable	4.5
9	Dangerous	5

Appendix. B. Some MBS information

```
************************************
Coupling bodies
************************************

Body_i                                                 Body_j                                                 Force name
----------------------------------------------------------------------------------------------------------------------------
$B_Isys.........................................$B_wst_1......................................$F_RW_Friction_Right_of_wst_1
$B_Isys.........................................$B_wst_1......................................$F_RW_Friction_Left_of_wst_1
$B_Isys.........................................$B_wst_2......................................$F_RW_Friction_Right_of_wst_2
$B_Isys.........................................$B_wst_2......................................$F_RW_Friction_Left_of_wst_2
$B_Isys.........................................$B_wst_3......................................$F_RW_Friction_Right_of_wst_3
$B_Isys.........................................$B_wst_3......................................$F_RW_Friction_Left_of_wst_3
$B_Isys.........................................$B_wst_4......................................$F_RW_Friction_Right_of_wst_4
$B_Isys.........................................$B_wst_4......................................$F_RW_Friction_Left_of_wst_4
$B_Isys.........................................$B_wst_5......................................$F_RW_Friction_Right_of_wst_5
$B_Isys.........................................$B_wst_5......................................$F_RW_Friction_Left_of_wst_5
$B_Isys.........................................$B_wst_6......................................$F_RW_Friction_Right_of_wst_6
$B_Isys.........................................$B_wst_6......................................$F_RW_Friction_Left_of_wst_6
$B_Bogie_Frame_1................................$B_wst_1......................................$F_primary_1_R
$B_Bogie_Frame_1................................$B_wst_1......................................$F_primary_1_L
$B_Bogie_Frame_1................................$B_wst_2......................................$F_primary_2_R
$B_Bogie_Frame_1................................$B_wst_2......................................$F_primary_2_L
$B_Bogie_Frame_2................................$B_wst_3......................................$F_primary_3_R
$B_Bogie_Frame_2................................$B_wst_3......................................$F_primary_3_L
$B_Bogie_Frame_2................................$B_wst_4......................................$F_primary_4_R
$B_Bogie_Frame_2................................$B_wst_4......................................$F_primary_4_L
$B_Bogie_Frame_3................................$B_wst_5......................................$F_primary_5_R
$B_Bogie_Frame_3................................$B_wst_5......................................$F_primary_5_L
$B_Bogie_Frame_3................................$B_wst_6......................................$F_primary_6_R
$B_Bogie_Frame_3................................$B_wst_6......................................$F_primary_6_L
$B_Car_body.....................................$B_Bogie_Frame_1..............................$F_secondary_suspension_1_front_R
$B_Car_body.....................................$B_Bogie_Frame_1..............................$F_secondary_suspension_1_front_L
$B_Car_body.....................................$B_Bogie_Frame_1..............................$F_secondary_suspension_1_rear_R
$B_Car_body.....................................$B_Bogie_Frame_1..............................$F_secondary_suspension_1_rear_L
$B_Car_body.....................................$B_Bogie_Frame_2..............................$F_secondary_suspension_2_front_R
$B_Car_body.....................................$B_Bogie_Frame_2..............................$F_secondary_suspension_2_front_L
$B_Car_body.....................................$B_Bogie_Frame_2..............................$F_secondary_suspension_2_rear_R
$B_Car_body.....................................$B_Bogie_Frame_2..............................$F_secondary_suspension_2_rear_L
$B_Car_body.....................................$B_Bogie_Frame_3..............................$F_secondary_suspension_3_front_R
$B_Car_body.....................................$B_Bogie_Frame_3..............................$F_secondary_suspension_3_front_L
$B_Car_body.....................................$B_Bogie_Frame_3..............................$F_secondary_suspension_3_rear_R
$B_Car_body.....................................$B_Bogie_Frame_3..............................$F_secondary_suspension_3_rear_L
$B_Car_body.....................................$B_Bogie_Frame_1..............................$F_secondary_damping_front_R
$B_Car_body.....................................$B_Bogie_Frame_1..............................$F_secondary_damping_front_L
$B_Car_body.....................................$B_Bogie_Frame_2..............................$F_secondary_damping_middle_R
$B_Car_body.....................................$B_Bogie_Frame_2..............................$F_secondary_damping_middle_L
$B_Car_body.....................................$B_Bogie_Frame_3..............................$F_secondary_damping_rear_R
$B_Car_body.....................................$B_Bogie_Frame_3..............................$F_secondary_damping_rear_L
$B_Car_body.....................................$B_Car_body...................................$F_Articulation_Damere_Front_R
$B_Car_body.....................................$B_Car_body...................................$F_Articulation_Damere_Front_L
$B_Car_body.....................................$B_Car_body...................................$F_Articulation_Damere_rear_R
$B_Car_body.....................................$B_Car_body...................................$F_Articulation_Damere_rear_L
```

Figure 57. Vehicle body coupling system information

```
**********************************
Coupling Markers
**********************************

Marker_i                                        Marker_j                                        Force name
----------------------------------------------------------------------------------------------------------
$M_Rail_Contact_Right_of_wst_1..................$M_wst_1_Contact_Right..........................$F_RW_Friction_Right_of_wst_1
$M_Rail_Contact_Left_of_wst_1...................$M_wst_1_Contact_Left...........................$F_RW_Friction_Left_of_wst_1
$M_Rail_Contact_Right_of_wst_2..................$M_wst_2_Contact_Right..........................$F_RW_Friction_Right_of_wst_2
$M_Rail_Contact_Left_of_wst_2...................$M_wst_2_Contact_Left...........................$F_RW_Friction_Left_of_wst_2
$M_Rail_Contact_Right_of_wst_3..................$M_wst_3_Contact_Right..........................$F_RW_Friction_Right_of_wst_3
$M_Rail_Contact_Left_of_wst_3...................$M_wst_3_Contact_Left...........................$F_RW_Friction_Left_of_wst_3
$M_Rail_Contact_Right_of_wst_4..................$M_wst_4_Contact_Right..........................$F_RW_Friction_Right_of_wst_4
$M_Rail_Contact_Left_of_wst_4...................$M_wst_4_Contact_Left...........................$F_RW_Friction_Left_of_wst_4
$M_Rail_Contact_Right_of_wst_5..................$M_wst_5_Contact_Right..........................$F_RW_Friction_Right_of_wst_5
$M_Rail_Contact_Left_of_wst_5...................$M_wst_5_Contact_Left...........................$F_RW_Friction_Left_of_wst_5
$M_Rail_Contact_Right_of_wst_6..................$M_wst_6_Contact_Right..........................$F_RW_Friction_Right_of_wst_6
$M_Rail_Contact_Left_of_wst_6...................$M_wst_6_Contact_Left...........................$F_RW_Friction_Left_of_wst_6
$M_Bogie_Frame_1_pri_1_R........................$M_wst_1_pri_R..................................$F_primary_1_R
$M_Bogie_Frame_1_pri_1_L........................$M_wst_1_pri_L..................................$F_primary_1_L
$M_Bogie_Frame_1_pri_2_R........................$M_wst_2_pri_R..................................$F_primary_2_R
$M_Bogie_Frame_1_pri_2_L........................$M_wst_2_pri_L..................................$F_primary_2_L
$M_Bogie_Frame_2_pri_1_R........................$M_wst_3_pri_R..................................$F_primary_3_R
$M_Bogie_Frame_2_pri_1_L........................$M_wst_3_pri_L..................................$F_primary_3_L
$M_Bogie_Frame_2_pri_2_R........................$M_wst_4_pri_R..................................$F_primary_4_R
$M_Bogie_Frame_2_pri_2_L........................$M_wst_4_pri_L..................................$F_primary_4_L
$M_Bogie_Frame_3_pri_1_R........................$M_wst_5_pri_R..................................$F_primary_5_R
$M_Bogie_Frame_3_pri_1_L........................$M_wst_5_pri_L..................................$F_primary_5_L
$M_Bogie_Frame_3_pri_2_R........................$M_wst_6_pri_R..................................$F_primary_6_R
$M_Bogie_Frame_3_pri_2_L........................$M_wst_6_pri_L..................................$F_primary_6_L
$M_Car_body_A_sec_front_R.......................$M_Bogie_Frame_1_sec_1_front_R..................$F_secondary_suspension_1_front_R
$M_Car_body_A_sec_front_L.......................$M_Bogie_Frame_1_sec_1_front_L..................$F_secondary_suspension_1_front_L
$M_Car_body_A_sec_rear_R........................$M_Bogie_Frame_1_sec_1_rear_R...................$F_secondary_suspension_1_rear_R
$M_Car_body_A_sec_rear_L........................$M_Bogie_Frame_1_sec_1_rear_L...................$F_secondary_suspension_1_rear_L
$M_Car_body_B_sec_front_R.......................$M_Bogie_Frame_2_sec_1_front_R..................$F_secondary_suspension_2_front_R
$M_Car_body_B_sec_front_L.......................$M_Bogie_Frame_2_sec_1_front_L..................$F_secondary_suspension_2_front_L
$M_Car_body_B_sec_rear_R........................$M_Bogie_Frame_2_sec_1_rear_R...................$F_secondary_suspension_2_rear_R
$M_Car_body_B_sec_rear_L........................$M_Bogie_Frame_2_sec_1_rear_L...................$F_secondary_suspension_2_rear_L
$M_Car_body_C_sec_front_R.......................$M_Bogie_Frame_3_sec_1_front_R..................$F_secondary_suspension_3_front_R
$M_Car_body_C_sec_front_L.......................$M_Bogie_Frame_3_sec_1_front_L..................$F_secondary_suspension_3_front_L
$M_Car_body_C_sec_rear_R........................$M_Bogie_Frame_3_sec_1_rear_R...................$F_secondary_suspension_3_rear_R
$M_Car_body_C_sec_rear_L........................$M_Bogie_Frame_3_sec_1_rear_L...................$F_secondary_suspension_3_rear_L
$M_Car_body_A_sec_damping_R.....................$M_Bogie_Frame_1_sec_damping_R..................$F_secondary_damping_front_R
$M_Car_body_A_sec_damping_L.....................$M_Bogie_Frame_1_sec_damping_L..................$F_secondary_damping_front_L
$M_Car_body_B_sec_damping_R.....................$M_Bogie_Frame_2_sec_damping_R..................$F_secondary_damping_middle_R
$M_Car_body_B_sec_damping_L.....................$M_Bogie_Frame_2_sec_damping_L..................$F_secondary_damping_middle_L
$M_Car_body_C_sec_damping_R.....................$M_Bogie_Frame_3_sec_damping_R..................$F_secondary_damping_rear_R
$M_Car_body_C_sec_damping_L.....................$M_Bogie_Frame_3_sec_damping_L..................$F_secondary_damping_rear_L
$M_Car_body_A_to_car_body_B_damper_R............$M_Car_body_B_to_car_body_A_damper_R............$F_Articulation_Damere_Front_R
$M_Car_body_A_to_car_body_B_damper_L............$M_Car_body_B_to_car_body_A_damper_L............$F_Articulation_Damere_Front_L
$M_Car_body_B_to_car_body_C_damper_R............$M_Car_body_C_to_car_body_B_damper_R............$F_Articulation_Damere_rear_R
$M_Car_body_B_to_car_body_C_damper_L............$M_Car_body_C_to_car_body_B_damper_L............$F_Articulation_Damere_rear_L
```

Figure 58. Vehicle body marker point and system component

Body_i	Body_j	Joint name
$B_Isys	$B_wst_1	$J_wst_1
$B_Isys	$B_wst_2	$J_wst_2
$B_Isys	$B_wst_3	$J_wst_3
$B_Isys	$B_wst_4	$J_wst_4
$B_Isys	$B_wst_5	$J_wst_5
$B_Isys	$B_wst_6	$J_wst_6
$B_Isys	$B_Bogie_Frame_1	$J_Bogie_Frame_1
$B_Isys	$B_Bogie_Frame_2	$J_Bogie_Frame_2
$B_Isys	$B_Bogie_Frame_3	$J_Bogie_Frame_3
$B_Isys	$B_Car_body	$J_Car_body
$B_Car_body	$B_Fixed_Articulation_A	$J_Fixed_Articulation_A
$B_Car_body	$B_Fixed_Articulation_B	$J_Fixed_Articulation_B
$B_Car_body	$B_Fixed_Articulation_C	$J_Fixed_Articulation_C
$B_Car_body	$B_Fixed_Articulation_D	$J_Fixed_Articulation_D
$B_Car_body	$B_Flexible_Articulation_A	$J_Flexible_Articulation_A
$B_Car_body	$B_Flexible_Articulation_B	$J_Flexible_Articulation_B
$B_Car_body	$B_Free_Articulation_A_	$J_Free_Articulation_A_
$B_Car_body	$B_Free_Articulation_B	$J_Free_Articulation_B

****
Coupling markers
****

Marker_i	Marker_j	Joint name
$M_Isys	$M_wst_1	$J_wst_1
$M_Isys	$M_wst_2	$J_wst_2
$M_Isys	$M_wst_3	$J_wst_3
$M_Isys	$M_wst_4	$J_wst_4
$M_Isys	$M_wst_5	$J_wst_5
$M_Isys	$M_wst_6	$J_wst_6
$M_Isys	$M_Bogie_Frame_1	$J_Bogie_Frame_1
$M_Isys	$M_Bogie_Frame_2	$J_Bogie_Frame_2
$M_Isys	$M_Bogie_Frame_3	$J_Bogie_Frame_3
$M_Isys	$M_Car_body	$J_Car_body
$M_Car_body_A_to_Fixed_articulation_A	$M_Fixed_Articulation_A_to_car_body_A	$J_Fixed_Articulation_A
$M_Car_body_B_to_Fixed_articulation_B	$M_Fixed_Articulation_B_to_car_body_B	$J_Fixed_Articulation_B
$M_Car_body_B_to_Fixed_articulation_C	$M_Fixed_Articulation_C_to_car_body_B	$J_Fixed_Articulation_C
$M_Car_body_C_to_Fixed_articulation_D	$M_Fixed_Articulation_D_to_car_body_C	$J_Fixed_Articulation_D
$M_Car_body_B_to_Flexible_articulation_A_R	$M_Flexible_Articulation_A_to_car_body_B_R	$J_Flexible_Articulation_A
$M_Car_body_C_to_Flexible_articulation_B_L	$M_Flexible_Articulation_B_to_car_body_B_L	$J_Flexible_Articulation_B
$M_Car_body_B_to_Free_articulation_A	$M_Free_Articulation_A_to_car_body_B	$J_Free_Articulation_A_
$M_Car_body_A_to_Free_articulation_B_L	$M_Free_Articulation_B_to_car_body_A_R_L_	$J_Free_Articulation_B

Figure 59. Constraint point and system component

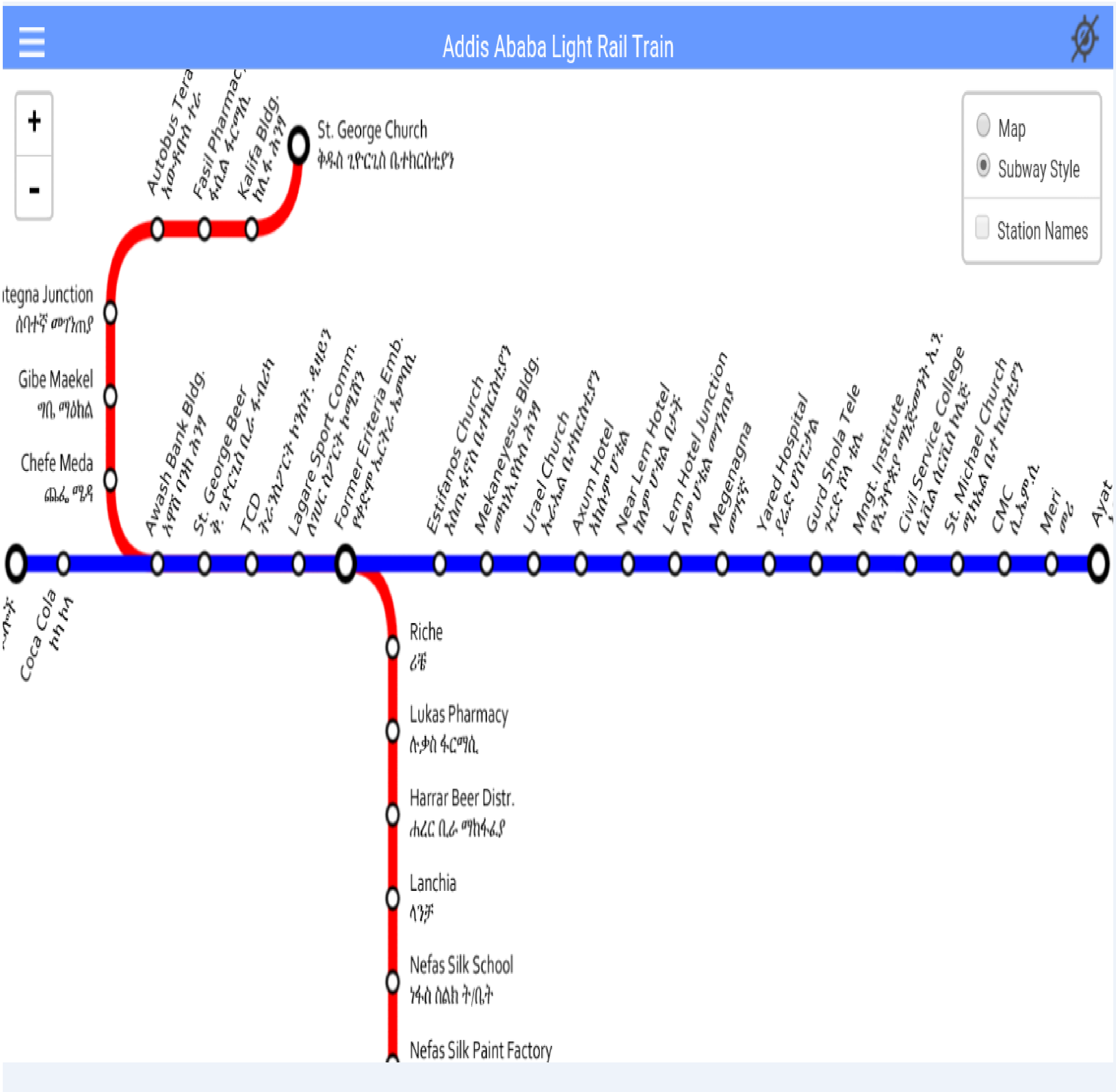


Figure 60. Addis Ababa Light Rail transient substations

Appendix. C. Some simulation results

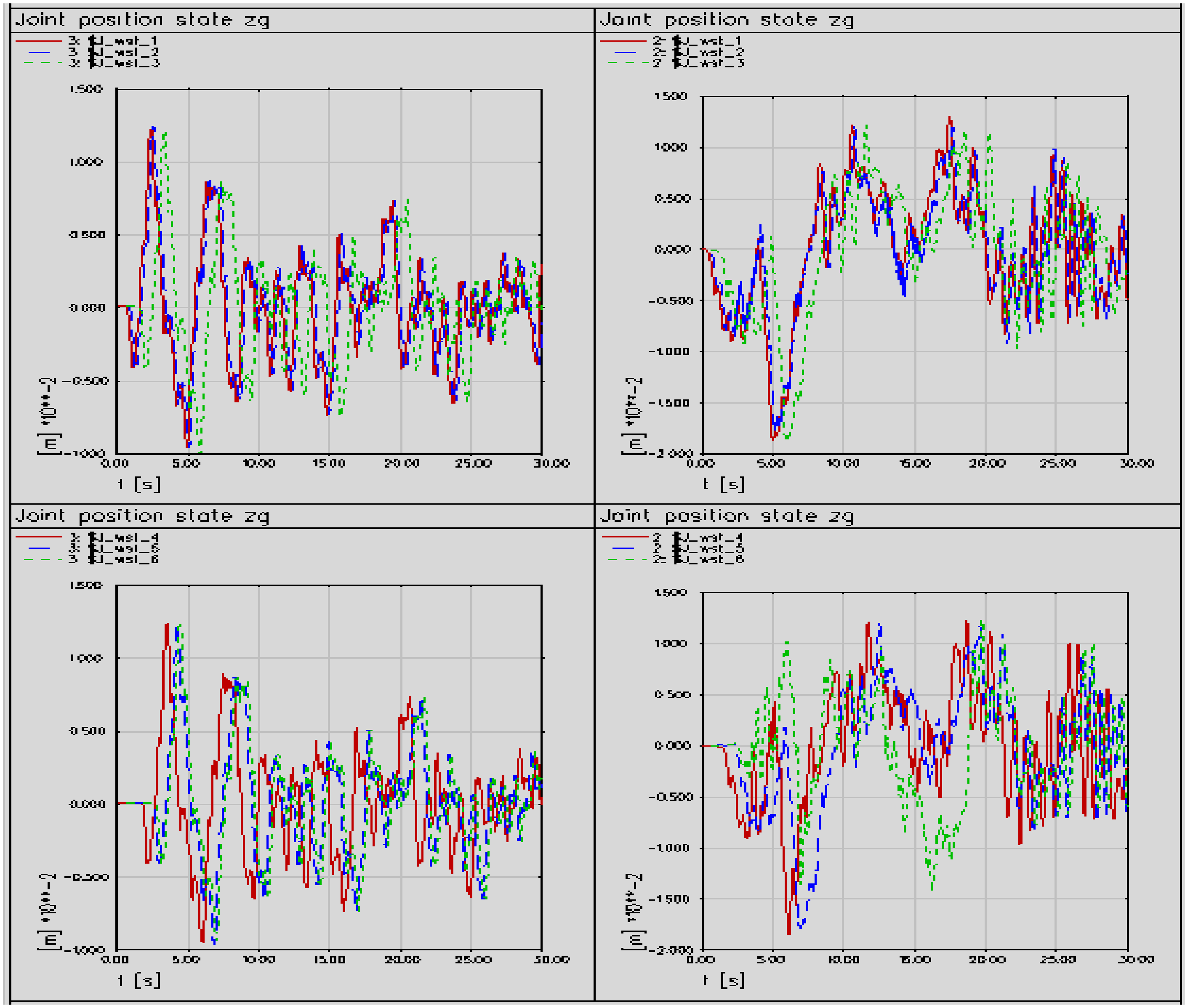


Figure 61. Wheel set lateral and vertical displacement

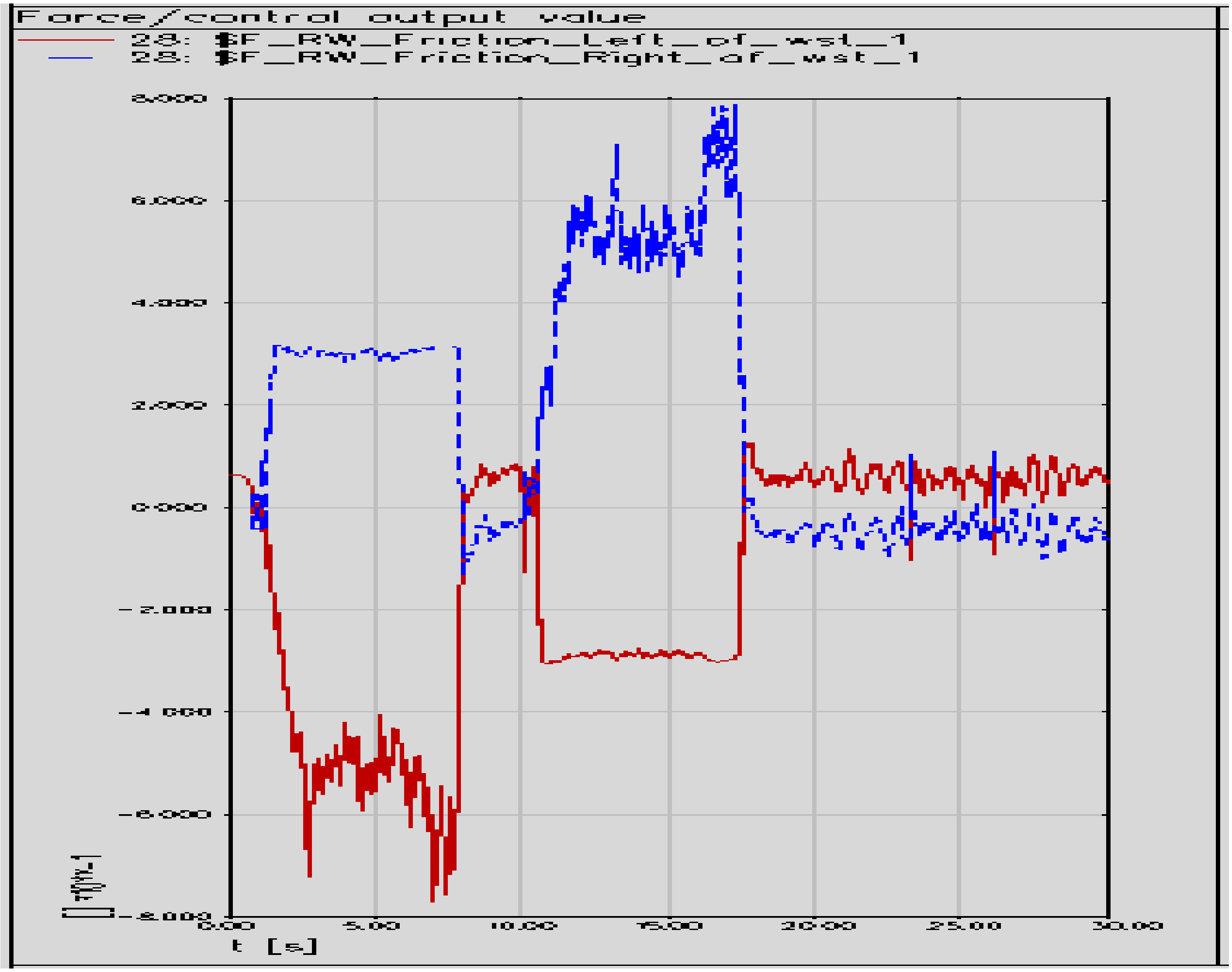


Figure 62. Derailment coefficient of wheel set left and right wheels

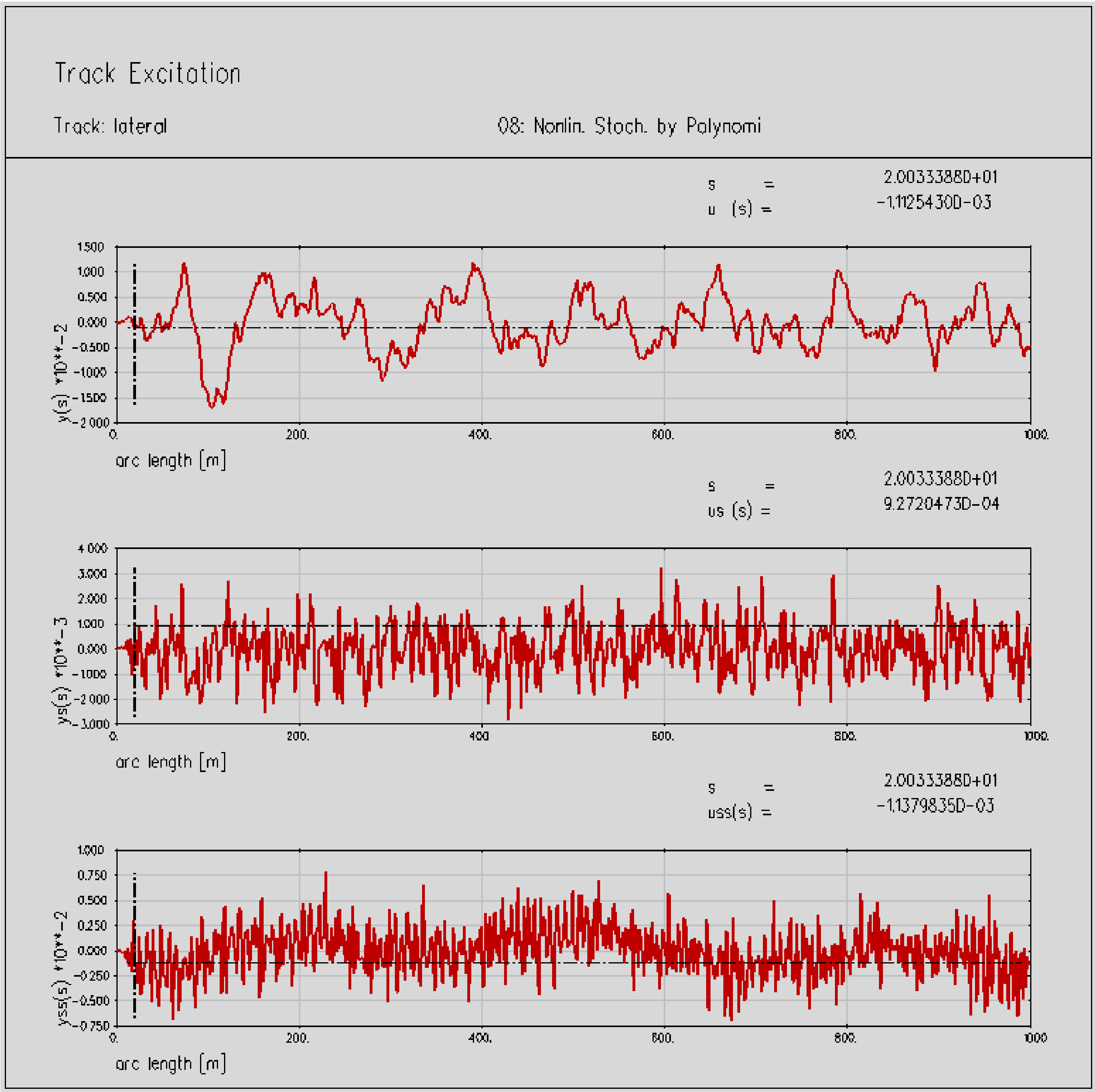


Figure 63. Lateral track excitation for each wheel set

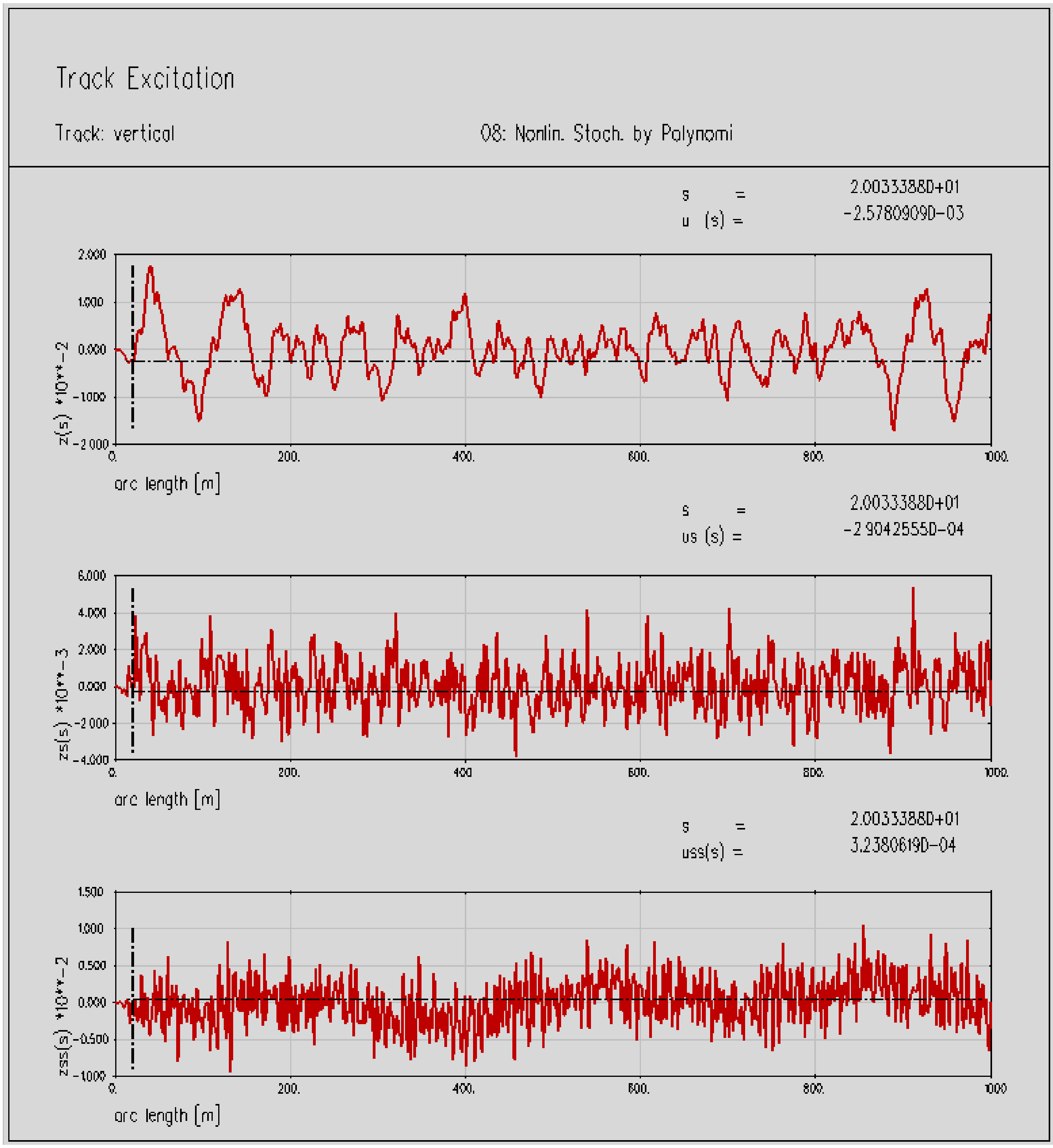


Figure 64. Vertical track excitation on each wheel set

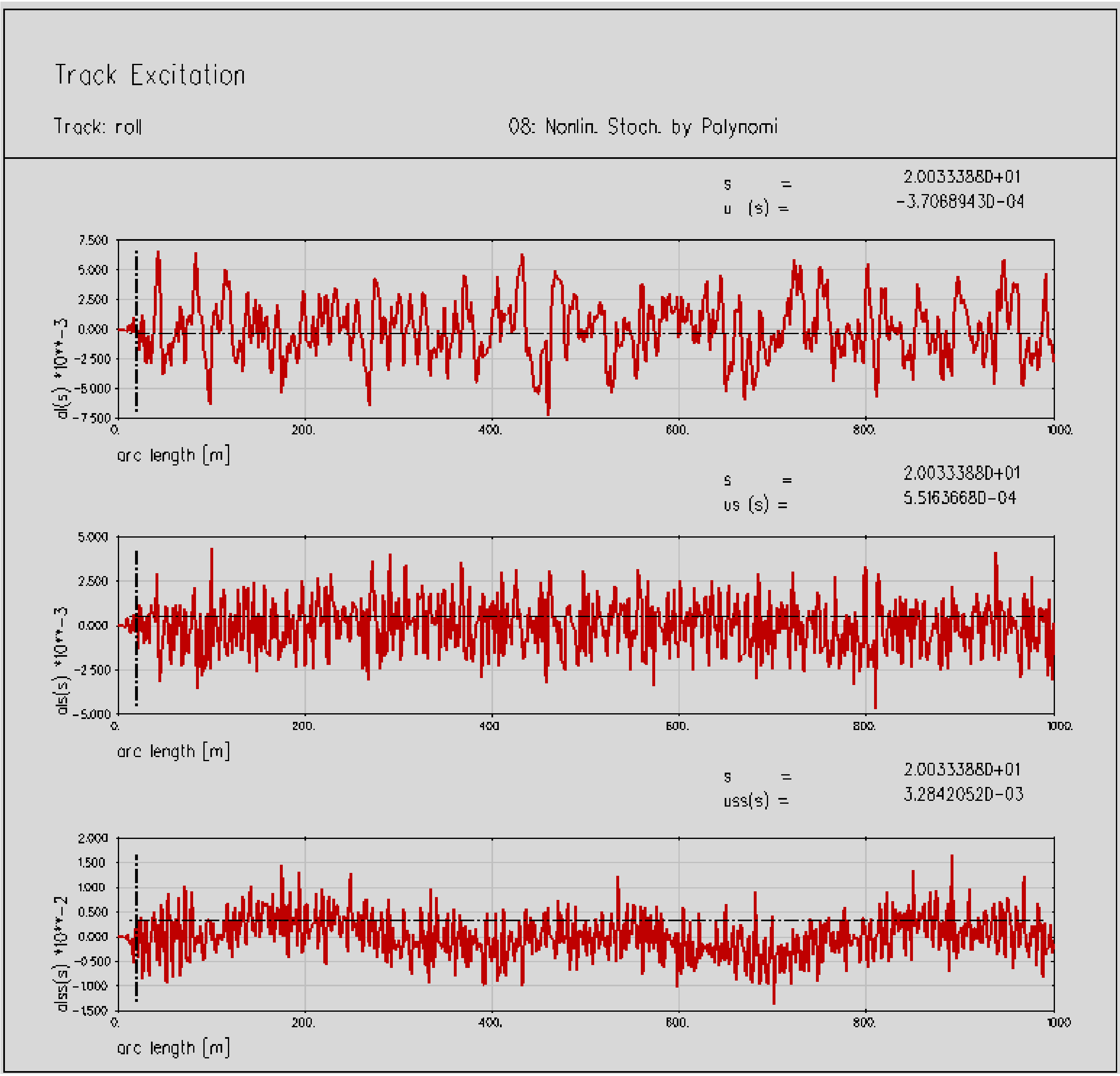


Figure 65. Track cross over excitation on each wheel set

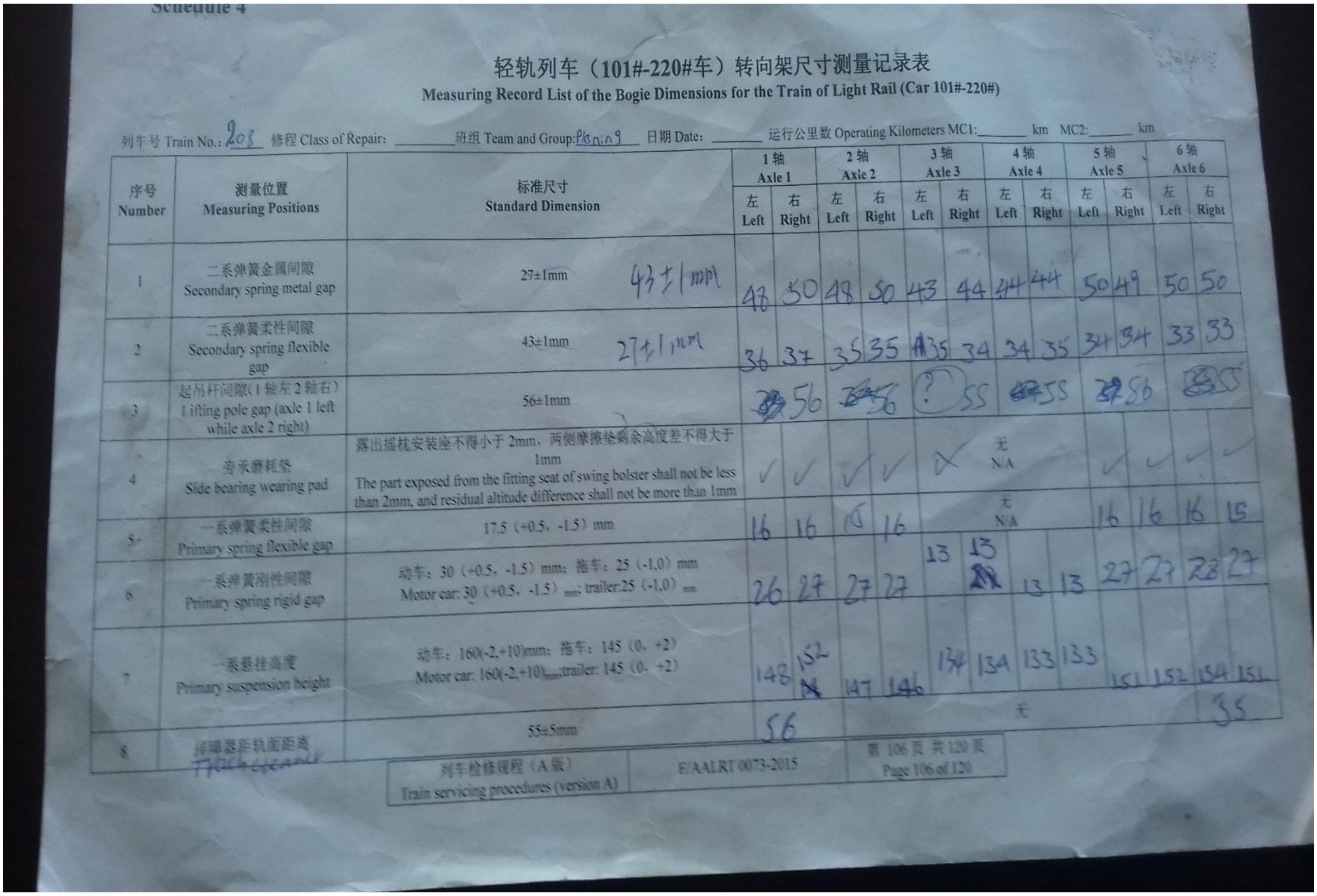
Design Parameters

Values to be investigated	Standards	Limits
Max. speed [km/h]		120
Max. testing speed [km/h]		135
Minimum curve Radius [m]		125
Net Axle Lateral Force [kN] for 92 to	Prud'Homme UIC 518	85.21
Net Axle Lateral Force [kN] for 100 to	Prud'Homme UIC 518	91.75
Derailment Safety Coefficient Dynamic L/V [-]	Contract	0.60
Derailment Safety Coefficient Static L/V [-]	ERRI B55	1.20
Vertical Wheel Load $V_{dyn.}$ [kN]	UIC 518	200
Lateral acceleration Car body [m/s²]	Contract	2.00
Vertical acceleration Car body [m/s²]	Contract	3.20
Driver´s Comfort Lateral Exposure Duration [h]	ISO 2631	< 8h Spectrum
Driver´s Comfort Vertical Exposure Duration [h]	ISO 2631	< 8h Spectrum
Lateral acceleration Bogie [m/s²]	UIC 515 Number of Oscillations exceeding 8 [m/s²]	< 6

FTS 3 Joohim

Transportation Systems   5 of 19 Pages

Figure 66. Design parameters for forces and accelerations [33]



Schedule 4

轻轨列车（101#-220#车）转向架尺寸测量记录表

Measuring Record List of the Bogie Dimensions for the Train of Light Rail (Car 101#-220#)

列车号 Train No.: 208 修程 Class of Repair: ________ 班组 Team and Group: Planing 日期 Date: ________ 运行公里数 Operating Kilometers MC1: ________ km MC2: ________ km

序号 Number	测量位置 Measuring Positions	标准尺寸 Standard Dimension	1轴 Axle 1		2轴 Axle 2		3轴 Axle 3		4轴 Axle 4		5轴 Axle 5		6轴 Axle 6	
			左 Left	右 Right	左 Left	右 Right	左 Left	右 Right	左 Left	右 Right	左 Left	右 Right	左 Left	右 Right
1	二系弹簧金属间隙 Secondary spring metal gap	27±1mm 43±1mm	48	50	48	50	43	44	44	44	50	49	50	50
2	二系弹簧柔性间隙 Secondary spring flexible gap	43±1mm 27±1mm	36	37	35	35	35	34	34	35	34	34	33	33
3	起吊杆间隙（1轴左2轴右）Lifting pole gap (axle 1 left while axle 2 right)	56±1mm	56		56		?	55	55		56		55	
4	旁承磨耗垫 Side bearing wearing pad	露出摇枕安装座不得小于2mm，两侧摩擦垫剩余高度差不得大于1mm The part exposed from the fitting seat of swing bolster shall not be less than 2mm, and residual altitude difference shall not be more than 1mm	✓	✓	✓	✓	✗	无 N/A			✓	✓	✓	✓
5	一系弹簧柔性间隙 Primary spring flexible gap	17.5（+0.5，-1.5）mm	16	16	15	16		无 N/A			16	16	16	15
6	一系弹簧刚性间隙 Primary spring rigid gap	动车：30（+0.5，-1.5）mm；拖车：25（-1.0）mm Motor car: 30（+0.5，-1.5）mm; trailer:25（-1.0）mm	26	27	27	27	13	13	13	13	27	27	28	27
7	一系悬挂高度 Primary suspension height	动车：160(-2,+10)mm；拖车：145（0，+2）Motor car: 160(-2,+10)mm;trailer: 145（0，+2）	148	152	147	146	134	134	133	133	151	152	154	151
8	排障器距轨面距离 Track clearance	55±5mm	56			无								35

列车检修规程（A版） Train servicing procedures (version A)	E/AALRT 0073-2015	第106页 共120页 Page 106 of 120

Figure 67. Measurement record list and standards