**Addis Ababa University**

**Addis Ababa Institute of Technology**

**African Railway Center of Excellence**

# Reliability and sensitivity analysis of multibody system due to uncertainty parameters. A case study of Addis Ababa Light Rail Transit

A Thesis Submitted to the School of Graduate Studies of
Addis Ababa University in Partial Fulfilment of the Requirements for the
Degree of Masters of Science in Railway Engineering
(Rolling Stock)

By:

Murungi Rodger

Advisor
Dr. Haileleoul Sahle HABTE

August 2023
Addis Ababa
Ethiopia

## Declaration

I hereby declare that the work being presented in this thesis titled "Reliability and sensitivity analysis of multibody system due to uncertainty parameters. A case study of Addis Ababa Light Rail Transit." is original work of my own, has not been presented for a Master's degree to any other university and all the materials used for this thesis have been duly acknowledged.

Student Name: Murungi Rodger

Signature: …………………………Date: …………………………….

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Approved by Examining Board:

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| Dr. Haileleoul Sahle HABTE<br>Advisor | Signature…………………….. | Date…………………… |
| Mr. Awel Mohammedseid<br>Internal Examiner | Signature…………………….. | Date…………………… |
| Dr. Araya Abera<br>External Examiner | Signature…………………….. | Date…………………… |
| Zewdie Moges (MSc)<br>A/Director of ARCE | Signature…………………….. | Date…………………… |
| Sosina Mengistu (PhD)<br>Associate Director of<br>Postgraduate Program | Signature…………………….. | Date…………………… |

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# Abstract

A multibody system is one of the complex mechatronic system whose dynamic performance is influenced by its connected components and the mechanical structure. However during the service life, the system is subjected to degradation of different components whose conditions may greatly impact the vehicles running characteristics like stability and reliability. The suspension system is one of the systems that ensure better and reliable running behaviors are achieved and hence any uncertain changes in its characteristics becomes of significant interest and must be treated with utmost attention. Hence this research aims carrying out reliability and sensitivity of multibody system due to uncertainties in the suspension system and provides a model which can be applied to evaluate the performance of the vehicle in terms of variation of suspension design parameters. The vehicle is modelled using SIMPACK dynamic software, incorporated with track irregularities from maintenance data and using statistical tools to vary suspension design parameters. The dynamic performance of the vehicle is evaluated by using dynamic indexes like derailment coefficient and vibration sensitivity to assess the level of reliability and sensitivity of the vehicle. The results showed that the running safety reliability is more sensitive at low values of damping and stiffness of the primary suspension and stability is achieved as they increase while vibration sensitivity increases with increase in damping and stiffness values of both primary and secondary suspension. Furthermore secondary suspension damping and stiffness greatly influence vibration sensitivity as opposed to primary suspension damping and stiffness which exhibits little influence on vibration sensitivity hence secondary suspension damping exhibits the greatest influence on vibration sensitivity while primary suspension stiffness exhibits the least influence on vibration sensitivity.


**Keywords:** *multibody*, *sensitivity, uncertainty, reliability, SIMPACK*

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## List of Abbreviations and Acronyms

AALRT………………………………………… Addis Ababa Light Rail Transit

MBS………………………………………….. Multibody System

HDMR…………………………………………High Dimensional Model Representation

M-DRM………………………………………..Multiplicative form of Dimensional Reduction

MDOF…………………………………………Multiple Degrees of Freedom

BF………………………………………………Bogie frame

CB………………………………………………Carbody

C,M, K………………………………………….Damping, Mass and stiffness of the system

Sim………………………………………………Simulation

$\ddot{y}$ , $\dot{y}$ $y$ ……………………………………………Acceleration, velocity and displacement of the vehicle

X, Y, Z………………………………………….Longitudinal, Lateral and Vertical directions

Φ, ψ, $\beta$ : Roll, Yaw and Pitch angles

$I_x, I_y, I_z$ ……………………………………………Moments of inertia in longitudinal, lateral and vertical directions

WS_1, WS_6……………………………………Wheelset 1, wheelset 6

$M_b$ $M_c$ $M_w$ ………………………………………..Mass of bogie, carbody, wheelset

$F_{yci}$ $F_{xci}$ $F_{zci}$ ………………………………………...Inter-vehicle forces between adjacent carbodies

$M_{yci}$ $M_{zci}$ $M_{xci}$ ……………………………………...Moments due to inter-vehicle connections between carbodies

$F_{zcc}$ $F_{ycc}$ ……………………………………………..Carbody external forces

$M_{wryi}, M_{wrxi}, M_{wrzi}$ …………………………………..Moments between wheels and rails

$F_{wryi}, F_{wrxi}, F_{wrzi}$ ……………………………………Contact forces between the rails and wheels,

# Chapter One

# Introduction

## 1.1 Background

The transport sector is one of the key sectors that plays an important role in achieving sustainable development in the world through facilitating other sectors. With an increasing world population, environmentally friendly, energy-saving, safety, and cost-effective, railway transport is seen as a solution to solving public transport system challenges. Railway transport takes 8% of world passenger traffic and 7% of total freight transport but this accounts for only 2% of energy consumption in the transport sector[1]. This has encouraged the growth of demand for railway transport and is estimated to continue growing. For instance in Europe, the demand for rail transport increased by 10.7% between the years 2015 and 2019 according to a Eurostat statistics report[2].

In Africa, the railway transport system is not fully developed and funding is required to support such projects. The railway transport coverage in Africa accounts for 5% of the global railway network and only 1% of the world's passenger traffic and 2% of world freight traffic[3][4]. However, due to railways being energy-saving, cost-effective, and environmentally friendly as it accounts for only 0.3% of direct carbon emissions released from the transport sector, railway transport is gaining prominence throughout the continent in a bid to boost their economies[1]. Railway transport is a heavy haul and safer mode of transport compared to transportation by road and this gives railway transport an advantage over road transport for development.

The railway system in Ethiopia dates back from the early days of 1897[5].This was a metre gauge railway line built to connect the capital city Addis Ababa to the red sea port of Djibouti where more than 95% of Ethiopia's trade passes and covered a distance of 780km[6].With the urge to modernize the railway system in Ethiopia under the National Railway Network of Ethiopia, this line was later replaced with an electrified one and three more constructed, the Awash–Weldiya Railway , the Weldiya–Mekelle Railway and Addis Ababa Light Rail Transit which runs in the capital Addis Ababa covering a total distance of 34.3km.It consists of mainly two lines where one runs from Ayat to Torhailoch covering a distance of 17.4km and the second one covers a distance of 16.9km running from Menelik II to Kality. Addis Ababa Light rail Transit consists of 70% low floor railway vehicles with each train comprising of three carbodies supplied by Changchun Railway Vehicles co. Ltd[7].

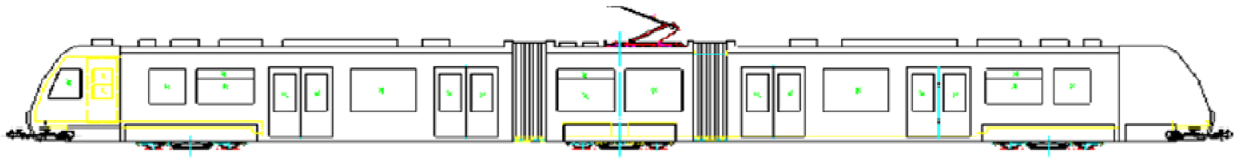

***Figure 1.1****: The side view of a passenger railway vehicle at AALRT*[7].

In an effort to solve the problem of public transport and congestion in Addis Ababa, the light rail transit was constructed to cater for the transport needs of people, and thus certain parameters had to be put into consideration to ensure that the service life of the trains is prolonged, comfortable passenger experience and maximize return on investments. These parameters include reliability and sensitivity.

In the process of manufacturing the rail vehicle, sensitivity and reliability of some of the vehicle components is considered within a certain minimal tolerance. The dynamics of the railway vehicle constitute of a variety of uncertainty sources during its service life which influence its reliability and sensitivity for instance passenger weight related where the actual loading of the carbody,wind force, curve negotiations, contact friction and the type of irregularities the system will be exposed to may not be clearly predicted at manufacturing stage. At AALRT, wear has always been a major challenge which necessitates continuous wheel reprofiling. This results into lowering of the carbody height distance from the track. This problem becomes more significant especially during the peak hours where maximum loading is expected and leads to damage of some components of the vehicle especially the gangway.

***Figure 1.2****: Damage to the gangway at AALRT.*

The problem is solved by the placing washers of approximately 10mm in the suspension system to raise the height however the concern for how the dynamic performance of the vehicle is affected by such modifications is not always catered for. This problem is directly related to the suspension system deformation and other contributing factors like changing wheel diameter, flexibility of the suspension system and loading of the carbody. It introduces uncertainty in dynamic performance of the vehicle due to such modifications in the suspension system. Several studies conducted by researchers have indicated that any change to the suspension can alter running characteristics leading to derailments, poor ride comfort and others. Sharma et al.,1986[8] investigated the sensitivity of the vehicle due to secondary suspension parameters, Chudzikiewicz & Opala,2008[9] conducted studies about reliability and running safety of the vehicle in connection to the derailment whereas Hamed et al., 2018 [10] assessed the influence of spring stiffness on the performance of the suspension.

***Figure 1.3**: Washers.*

Therefore it is imperative to investigate how these modifications and uncertain situations in the suspension system influence the dynamic performance of the vehicle.

## 1.2 Statement of the problem

During the manufacturing process of the railway vehicles, sensitivity and reliability of deterministic dynamic systems are carried out however this falls short in giving a comprehensive picture of the actual reality. Numerous sources of uncertainty are considered when designing and this implies that the results obtained only refer to single realisations which may never happen in reality hence exhibiting zero measure. The suspension system is one of fundamental systems that plays a vital role in supporting the carbody and the bogie, absorbs forces or vibrations generated from unevenness of the track, comfortable rides and ensuring that stable and safe movement of the railway vehicle is achieved however it is often modified to prevent the lowering of the carbody and damage to some components such as gangway[11]. Such modifications coupled with suspension system degradation tend to introduce uncertainty and temper with the performance characteristics of the vehicle, hence a need to quantify them such that safe limits are not exceeded. In addition, uncertainty contributes to maintenance problems through accelerating component degradation thus influencing the overall life of the vehicle.

## 1.3 Research questions

1. How does uncertainty in the suspension system affect the dynamic forces of the vehicle?
2. What is the influence of uncertainty on the carbody vehicle accelerations?
3. How is sensitivity and reliability of the vehicle influenced by the uncertainty in the suspension system?
4. How can the suspension system parameters be optimized to improve the dynamic performance?

## 1.4 Objectives

### 1.4.1 General objective.

The general objective of the study is to carry out an investigation on how uncertainty parameters affect the sensitivity and reliability of AALRT.

### 1.4.2 Specific Objectives

- To investigate the influence of uncertainty in suspension system on the dynamic forces of the vehicle.
- To determine the influence of uncertainty on carbody vehicle accelerations.
- To determine the dynamic performance reliability of vehicle due to uncertainty in the suspension.
- To investigate the sensitivity of vehicle to uncertainty in the suspension.
- To conduct optimization of the suspension system to improve on the dynamic performance.

## 1.5 Scope and limitation of the study

The research intends to investigate the dynamic response of the rail vehicle due to uncertainty parameters in the suspension system. These parameters will be used to formulate performance functions associated with the running safety of the vehicle. The intention is to determine the sensitivity of performance function with regards to random input suspension variables and inclusion of track of track irregularities which are frequently met during the running behavior of the vehicle. The study employed computer simulations with a rigid vehicle and no experimental tests were conducted.

## 1.6 Significance of the research

The design process requires consideration of the loads, dimensions of the geometry and material properties which are taken into account based on specific assumptions. However there could be a difference and such assumptions may not be met in the actual operating conditions. For every trip made by the train, the dynamic response of the train will defer due to either seasonal effects or degradation of the track. Consequently the designer may have prescribed certain inputs at the design stage but the actual realisation of the running system may be completely different. This contributes to errors between the actual events and analysed ones especially when the system has been in operation for a longer period of time. This study further seeks to extend the deterministic parameters of the suspension system and quantification of uncertainties to enable enhanced analysis which can be applied for risk assessment or identification of dangerous phenomena during the operation phases of the vehicle.

## 1.7 Organization of the report

The report constitutes of mainly five chapters namely with each chapter consisting of topics. Chapter one introduces and provide the background of the study. Chapter two covers MBS dynamics principles and literature related to sensitivity and reliability by other scholars. In chapter three, process of data collected, modelling of the vehicle in MBS and identification of suspension parameters for analysis are covered. Chapter four encompasses the result and discussion section in line with the specific objectives and chapter five is the last chapter that covers conclusion, recommendations and future works.

# Chapter Two

## Literature review

## 2.1 Introduction

This section presents a review of studies that have been conducted by other scholars and are related to this research. This will helps in breaking it down into a clear summary of the existing knowledge and identifying the missing links to be addressed. It will mainly cover

- Multibody system dynamics and simulation.
- Sensitivity and reliability analysis of railway vehicles.

### 2.2 Multibody system dynamics.

A multibody system dynamics is an integration of either rigid bodies, flexible bodies, or both bodies combined together by force elements and kinematic joints. The relative movement between bodies is constrained by kinematic joints whereas the internal forces due to relative movement are represented by force elements. The interaction between the surrounding and the system results into external forces for instance forces at the rail-wheel contact. Both internal and external forces are described using kinematic quantities like dampers and springs[12].A railway vehicle as a mechatronic system is a complex dynamic system whose component integration and mechanical structure play a vital role in determining its performance features. As the speed of the railway vehicle increases, extra consideration is given to the dynamic response in identifying key design factors that greatly affect the vehicle running characteristics like reliability, sensitivity, comfort, and safety. The rail vehicle is a multibody system dynamics that can be divided into individual component parts and its dynamic response to varying operating conditions studied. Railway vehicle movement involves an interaction between rail and wheel with a rail head and wheel tread complex geometry and multiple degrees of freedom[13]. The forces from the vehicle's car body are transmitted to the rail-wheel contact through secondary suspension, primary suspension, the bogie, and the wheelset. The figures show models of the railway

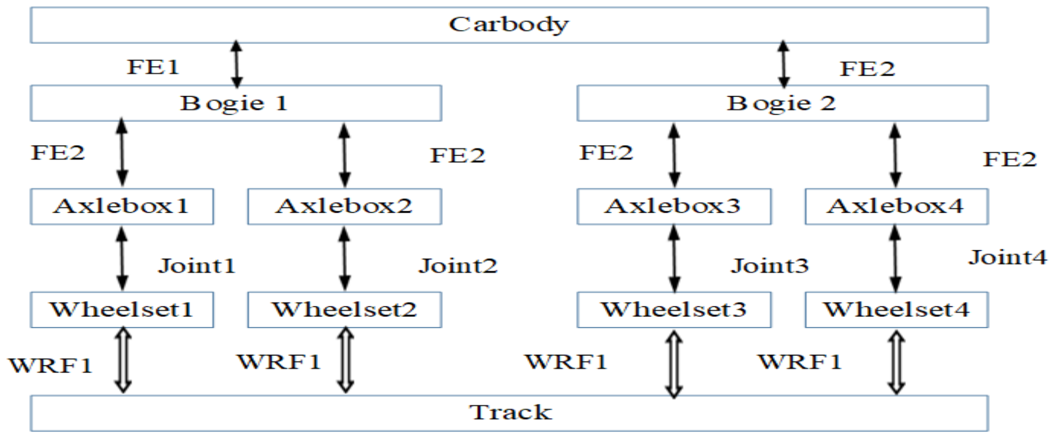


***Figure 2.1****: The topology of the railway vehicle*[14].

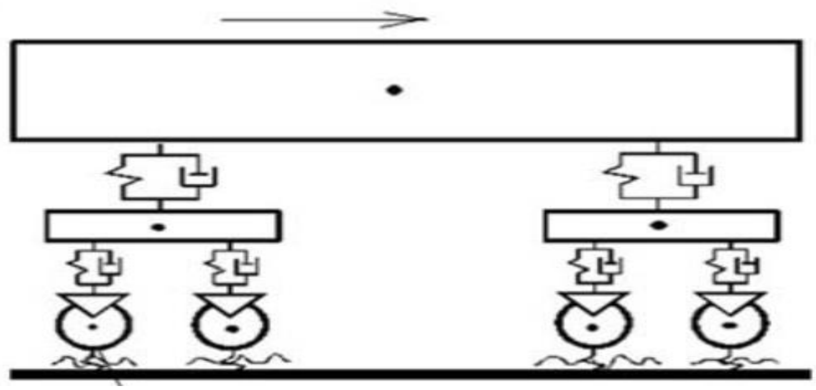

***Figure 2.2:*** *Physical model of the railway vehicle*[14].

- **Multibody system dynamics fundamentals.**

This concerned with setting up a mechanical model to enable carry out simulation and analysis of the desired design parameters. The model consists of mathematical expressions obtained from laws and principles. These equations of motion rely on Newton's laws of motion and others.

- **Equations of motion.**

The following equations are a representation of multibody system mathematically. Equations of motions are as a result of the use of constraint equations which can be linear algebraic equations or ordinary differential equation basing on how constraint equations are applied. However ordinary differential equations can be combined with linear algebraic equations to form differential algebraic equations[15].

Proportional differential forces of a multibody system results into ordinary multibody system and d'Alambert's principle is applies in Newton-Euler equation to obtain the equations of motion[15].Hence by applying d'Alambert's principle, the following equations of motion are obtained.

$$M(y,t)\ddot{y}+k(y,\dot{y},t)=q(y,\dot{y},t) \tag{2.1}$$

Thus number of equations is reduced from $6p\ to\ f, f \times f -$ inertia matrix, $M(y,t)$ and wholly symmetrized by $M(y,t)=\bar{J}^T\bar{\bar{M}}\bar{J}>0$ thus eliminating moments and constraint forces.

The equation in (1) holds true for unconstraint systems and it becomes $y = x$ and thus the Jacobian global matrix $\bar{J}$ becomes a quadratic equation $6p \times 6p -$ matrix. Furthermore Langrange equation obtained after adding implicit constraint, $\Phi(x,t)=0$ is

$$M(x,t)\ddot{x}+k(x,\dot{x},t)=q(x,\dot{x},t)-\Phi_x^T\lambda \tag{2.2}$$

Where $\lambda$ is the $q \times 1$- vector of langrangian multipliers. Equation 2 is $6p$ scalar equation which cannot be solved because of $6p + q$ unknowns in the $\lambda$ , $x$ vectors and thus $\Phi(x,t)=0$ is called.

$6p + q$ algebraic differential expression is maintained and hence resulting into multibody system equation:

$$\begin{pmatrix} M & \Phi_x^T \\ \Phi_x & 0 \end{pmatrix}\begin{pmatrix} \ddot{x} \\ \lambda \end{pmatrix}=\begin{pmatrix} q-k \\ -\Phi_t-\dot{\Phi}_x\dot{x} \end{pmatrix} \tag{2.3}$$

Force matrixes are grouped according to coordinate vectors and this results into equations of motions of the form:

$$M\ddot{x}(t)+(D+\Omega_0 G)\dot{x}(t)+(k+\Omega_0^2 Z)x(t)=f(t) \tag{2.4}$$

Where $x(t)$ vector has generalized coordinates, $D$ is the damping matrix from coupling elements of the bogie, carbody and the wheelset. $M$ is the mass matrix and $k$ is the stiffness matrix from elastic coupling.

Hence a set of algebraic differential equations are used to define multibody system dynamics and dynamic simulation is applied to combine nonlinear or linear differential equations in order to achieve accelerations, non-contact external loads and any other parameters[15]. In this study, obtaining forces and accelerations will be the main objective.

- **Multibody system dynamics simulations.**

This is a crucial part of analyzing computational dynamics which involves the dynamics of flexible and rigid systems and their kinematics. Numerical modelling combined with simulations are important techniques employed in carrying out studies about the dynamic behavior of multibody systems which constitutes multiple degrees of freedom. A number of software packages like SIMPACK, VAMPIRE, Mehrko¨rper-Dynamik (MEDYNA) and Adams are used to conduct simulations and analyse the sensitivity of the system[16][17].

SIMPACK is one the commonest packages provided for studies about nonlinear kinematics. It involves the creation of equations of motions implicitly or explicitly using relative coordinates. It is a package used carry out modelling, analysing and designing complicated mechanical systems. It is employed in conducting analysis of vehicle frequencies, accelerations, and forces due to certain operating conditions, predict multibody system motions and how to improve designs to meet the uncertainty conditions.

- **Multi-body dynamic model of the rail vehicle.**

For this study, to determine the dynamic response of the way vehicle, the geometric model of the railway vehicle is used to find the physical model. The topology of the railway vehicle is as shown in Figure 2.3.

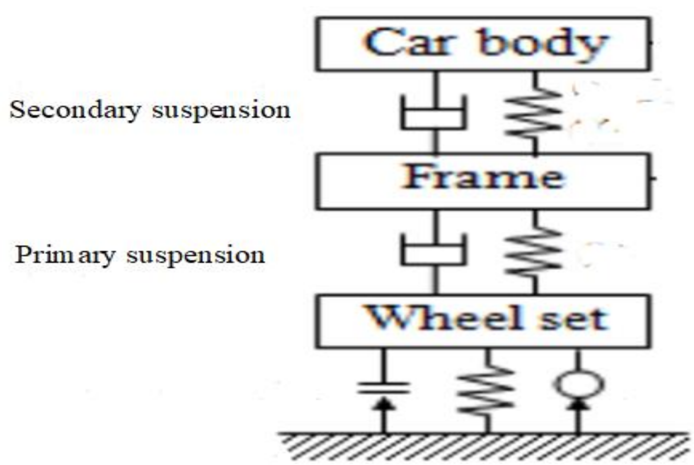


***Figure 2.3**: Rail vehicle topology*

An estimate of the vertical position of secondary and primary suspension, bogies and carbody systems was also required. The multibody system contains a number of components in the system which affect the dynamic response. Thus the system of multibody assembly constitutes of many rigid bodies and flexible bodies connected together by kinematic joints. These joints allow particular degrees of freedom of relative motion and limits others depending on the marker element and the type of joint. Also the force elements like the secondary and primary suspension systems may be used to in internal force interconnections between the carbody and the bogie and wheelset and the bogie. The wheels are connected to axles by fixed joint mechanism and the prismatic joint systems connect the bogie system through primary suspension element and the wheelset. The carbody and the bogie are connected through the secondary suspension with the help of center bolts. The schematic diagram below shows detailed descriptions.

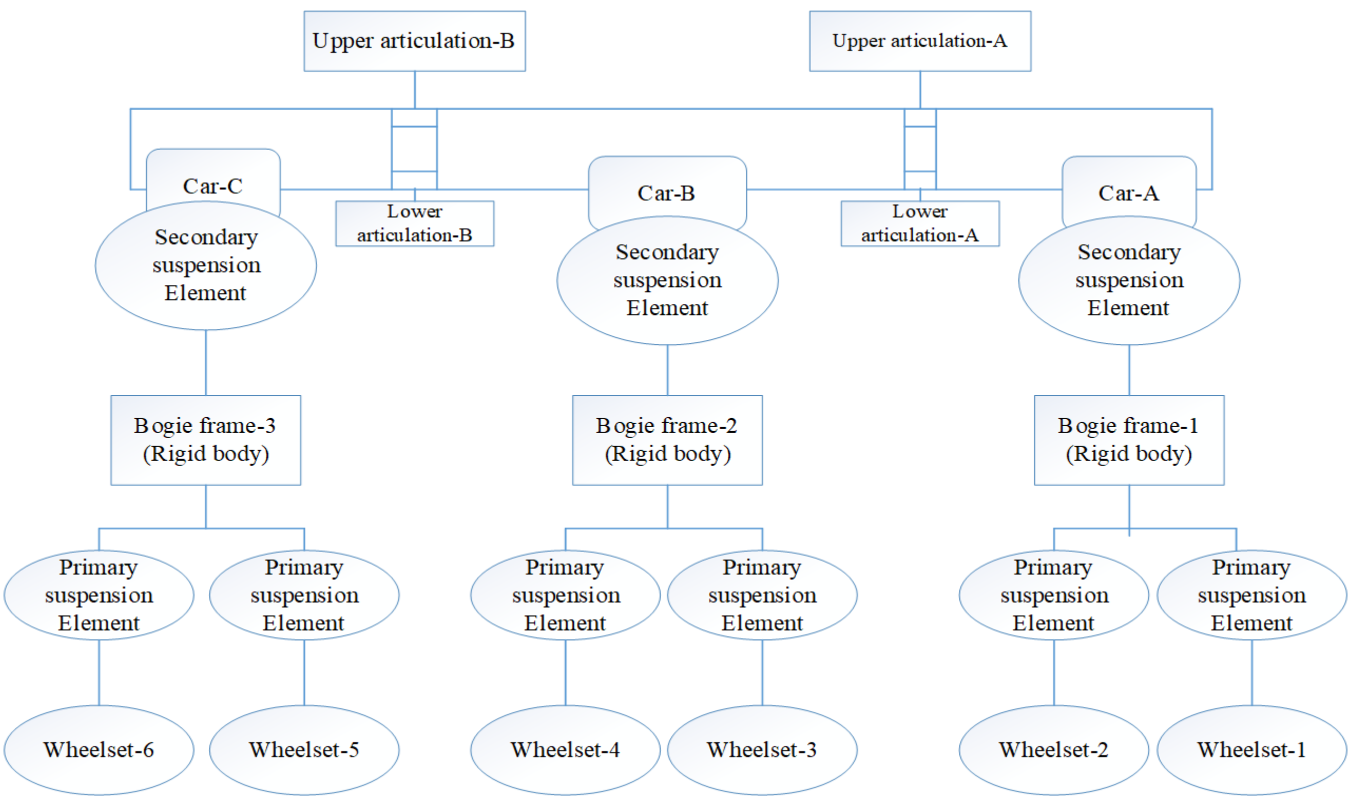


***Figure 2.4****: Schematic diagram of passenger railway vehicle*[18].

Articulation systems located at the lower and upper end of adjacent carbodies help to connect the three carbody modules. The adjacent modules which contain the articulation device must have a structure that is well proven and constitutes of the metallic spherical bearing and articulation unit.

A motion node which enables articulation to increase flexibility is called metallic spherical bearing. Vibrations in the lateral, vertical and longitudinal directions are controlled through the installation of dampers on lower articulation at both sides. Two mounting seats help to connect the carbody with the center rotating node module and fixed hinges which are lower hinge systems. Different mounting seats are connected by bolts to the end of surface of the module. The free hinge upper articulation consists of connecting pipe, installing parts, bearing receiver and control console. Alignment should be ensured between the upper hinge center point and the lower hinge center point system and in order to perform this, vertical fixing of the upper hinge center point system of the main body is conducted. Center bolts ensure connection at the bogie centers between carbodies, secondary suspension and traction rode whereas the wheelsets and the bogie are connected through the primary suspension.

### 2.3 Sensitivity analysis of random parameters.

This is employed to investigate the degree of effect of uncertainty parameters on the multibody system and determine how certain input parameters affect the output of the model through identifying the inputs with significant influence on the overall performance of the vehicle.

The following steps offer assistance during the analysis of sensitivities of input parameters.

- Determining the model input parameters, under this methods like deterministic variables and differentiated in probabilistic.
- Determining of input parameter distribution like value range and probability density functions.
- Propagation of distributions of inputs to the output parameters of the model using sampling methods.
- Quantifying of effect of input parameters on damage.

Different sensitivity methods and techniques are employed to conduct the studies from which the response of the vital system can be achieved.

- **Selection of methods for sensitivity analysis.**

Different methods can be used in conducting sensitivity analysis both quantitative and qualitative. They are further divided into global and local methods. Global sensitivity methods which include nonlinear and multilinear regression are variance based which take into account the effect of complete distribution ranges of all parameters whereas local methods only analyse the effect of a parameter at a particular working position while other parameters are fixed. They do not put into consideration the influence of coupling of input parameters on the outcome. Some of the methods employed include;

1. **Regression analysis.**

This method is based on Monte Carlo conducting sampling on a multi-linear regression analysis between several input parameters and thus the damage can be determined.

2. **One-Parameter-at-a-time method (OAT)**.

The simplest path to conduct a sensitivity analysis is to change each parameter independently while keeping other parameters at their nominal values even though the effect of one parameter may be dependent on the magnitude of other parameters. Under this, the influence of coupling is negligible.

3. **Variance-based methods.**

These are global sensitivity methods that investigate the effect of input parameter distribution on the distribution of the results. Under this, the effect of coupling is put into consideration and it allows for analyzing uncertainty propagation through the model. They call for the use of sampling techniques to estimate damage distribution. To differentiate between the effect of a single input parameter and the full influence of that particular input parameter putting into consideration coupling with other input parameters, the term total sensitivity index is used which refers to the sum of all indices related to the parameter[19][20]. To study the effect of a parameter $X_i$ on the variance, the value of the parameter is fixed and the distribution of $Y$ is measured using an appropriate number of samples and expected value $E(Y, X_i)$ is obtained.

Thus the sensitivity index $S_i$ is $\frac{V[E(Y|X_i)]}{V[Y]}$

4. **Variance-based sensitivity analysis with correlation of inputs.**

Indices of sensitivity in methods based on variance are based on assuming that the input parameters do not depend on each other. When the parameter correlation is near to each other, then the input parameter dimension vector is decreased and the parameters are taken as one input.

For dependent parameters$X_i$ $and$ $X_j$,

The sensitivity index $S_i$ is $\frac{V[E(Y|X_i, X_j)]}{V[Y]}$

**Classification of sensitivity analysis.**

Sensitivity analysis explains how the inputs of the model affect the output of the model. It is classified into two main methods which based on global or local definitions and our focuss will be on the global method.

- **Global sensitivity analysis.**

Under this, variance of the output of the model is focused on and how variability of the input affects the output variance. This helps to determine which output variance parts are as a result of various inputs. This can be done by using sobol indices and play a key role since they give a quantitative overview of how several inputs affect the output. The sobol indices can be estimated with the application of Monte Carlo methods and latin hypercube sampling[21].

I. Monte Carlo estimation.

This is employed in predicting the outcomes of an event that is uncertain. Monte Carlo method can be used to estimate the sobol sensitivity indices of any parameter. The estimation carries out a risk analysis by setting models of the probable results through the use of range of values known as probability distribution. It simulates the results repetitively each time applying a different set of random inputs and develops a mapping from input to analysis result. Depending on the number and the range of uncertainties simulations can involve many recalculations. This method can be used in different kinds of analysis such as crossings and switches and track degradation[22]. This is due to its accuracy and simplicity in calculating the variation in response to stochastic interaction systems[23].

II. Latin hypercube sampling.

This method was proposed by McKay et al., 2000[24] and since then has been developed further to be applied in diferent research.It is a type of stratified Monte Carlo method where the sapling area is divided into a particular manner and this enables easier generation of the samples for correlated components through the application of Gaussian distribution.The nethod is applied in computer simulaton models which are characterised by a lot of input variables and normally less of theses input variables are useful for a particular response.The response of the model is usually time dependant and multivariate and ensures that the whole range of a particulra input is fully covered[25].

## 2.4 Related literature about the research topic.

Numerous studies have been conducted about sensitivity of multibody system. Ding et al.,2014[14] conducted studies to investigate the effect of the longitudinal stiffness in the primary suspension using a prototype program, GVDs and sensitivity analytical equations. The results indicated that primary suspension longitudinal stiffness substantially affect derailment coefficient however the study didn't consider the rail/wheel contact effects.

A passenger rail vehicle with double deck having a bogie PW200 was studied by Zhou et al., 2014a[26] and an analysis of its characteristics conducted. Its objective was to investigate how secondary and primary suspension parameters affect the vehicle's dynamic response using SIMPACK platform. It was based on HXN5 locomotive model. It was found that primary suspension damping and vertical stiffness have the highest level of sensitiveness while vertical and longitudinal damping for secondary suspension and primary suspension lateral stiffness have the least effects on vibration.

Eom & Lee, 2010[27] studied the influence of speed, cant and curves on wheel unload ratios and derailment coefficient and further investigated fatigue variations of the rail vehicle on the track through varying damping and stiffness of the suspension using ADAMS/Rail software. The findings indicated that derailment coefficient value increases with increase in speed and reduction in curve radius which poses a danger to the movement of the vehicle. Derailment is one of the criteria used to evaluate the reliability of the railway vehicle for its safety. It can be triggered by the condition of the track, reduced fastening system stiffness, rail inclination and expansion of rail gauge as investigated by Konowrocki & Chojnacki, 2020[28] however the influence of track irregularities was not considered in these studies.

Hunt et al.,1986[29] conducted various studies of track –vehicle interaction in combination with track excitations under several parameters of the vehicle however the suspension system was not included. The effect of forces generated due to track excitations on the yaw dampers in tramway vehicle was studied using matlab and experimental studies and the findings indicated that stability of the vehicle reduces with increase in track excitations.

A method called high dimensional model representation (HDMR) breaks down the system's function response into a set of low dimensional functions and greatly reduces sampling and computational time[30]. This is of importance when solving the integral needed to evaluate global sensitivity indices in complex systems. This method also involves approximations like cut-HDMR where the function is evaluated along planes and lines. Using this knowledge, a multiplicative form of dimensional reduction method (M-DRM) was proposed for global sensitivity analysis which is more accurate, simple, and efficient. This method was proposed by Milad et al., 2016[31]. The results obtained using this method were as accurate as those using Monte Carlo with less computational time[32].A similar approach is used to investigate the global sensitivity analysis concerning the behavior of a one-car rail vehicle dynamically and the components of the suspension and also dynamic global sensitivity studies of the behavior of the bogie in relation to the suspension have shown that the longitudinal primary spring affects performance significantly in the rail vehicle[33]

Suspension parameters that influence dynamically the performance of the rail vehicle were studied using sensitivity analysis and this was based on the suspension neural network of high speed train and studies about the dynamic effect of suspension parameters on the performance through applying NUCARS software. Important suspension parameters were realised using orthogonal experiment method[32].

Yu et al., 2014 [34]carried out an analysis about the suspension parameter sensitivity on stability of the vehicle vertically using extended fourier amplitude sensitivity and the results showed that there is significant influence on the stability of the vehicle. Xie et al., 2016 [32]investigated about suspension parameter sensitivity expressions on dynamic indexes based on theoretical dynamic model of a multi-body described in second order linear differential equations. Vertical sperling index and rail vehicle vertical acceleration were established by Yang et al., 2014[35] using random vibration frequency domain method which specify the level of passenger ride comfort and the dynamic performance due to suspension parameter sensitivity using dynamic simulations was investigated by Zhou et al., 2014[36].

Park, Kim, and Bae 2009[37]evaluated the influence of contact mechanism suspension characteristics and mass properties on the stability and safety of a railway vehicle by comparing results using neural network method and response surface method and coding models using the relationship between the performance indexes and design variables. The results obtained using both methods were similar in view of main design variables and the sensitivity procedure proved to be satisfying for railway and other mechanical complex systems.

A combination of multibody dynamics software and Karhunen-Loeve expansion method has been used to conduct an investigation stochastic in nature for wheel/rail interface irregularities[38].

Similar studies conducted by Kassa & Nielsen, 2008[23] and Pacejka et al., 2008[39] involved dynamic track-train interactions in curved paths using two methods where better results were achieved using a rigid body.

Zhang et al., 2019[40] performed a dynamic reliability assessment track-vehicle coupled systems and rail-wheel parameters using SIMPACK and estimating the randomness of the parameters using probability. The results showed that increasing of friction coefficient enhances ride quality on straight tracks while worsening in curved tracks.

Mohammadzadeh et al., 2011[41] conducted an investigation about derailment probability using response surface method ,saturated design method and level 3 reliability method considering track geometry irregularities and realised that the derailment index sensitivity is very much correlated to alignment parameters like rails and their longitudinal level.

Cho et al., 2010[42] developed a method for assessing the reliability of complex structures with nonlinear time dependent behavior through improving response surface method. The design parameters in the study included uncertainties in moment of inertia of the vehicle passing over a bridge and stiffness and the results showed that reliability is also affected by the bridge-vehicle interaction in addition to uncertainties in the design parameters.

Konowrocki & Chojnacki, 2020 [28]investigated the railway vehicles operational reliability using Nadal's expression and also performed experimental tests taking into consideration the actual rail vehicle and track dynamic parameters and results helped estimate the conditions that can put operation of rail vehicle at risk using several evaluation criteria.

### 2.5 Summary of literature review

Several studies have been conducted on sensitivity and reliability of multibody system based global, local and various software packages focusing on uncertainty in multibody components including the suspension system, their interaction and response to the surrounding environment. However track irregularities or excitations combined with uncertainty in the suspension system have not been given much attention with respect to how the system dynamic behavior is affected and hence this study aims at conducting this investigation[14] [43][26][27]. A summary of literature review related to the topic is shown in Table 2.1.

***Table 2.1:*** *Summary of the literature review.*

| **Author** | **Objective** | **Methodology** | **Finding** | **Gap** |
|---|---|---|---|---|
| Park et al., 2009 | Conduct a sensitivity analysis of suspension characteristics for high speed train | -Response surface method<br>- Neural network mode | The values of ride comfort and stability obtained using both methods were close to each other. | Track-vehicle excitations were not considered. |
| Ding et al., 2014 | To identify the design factors that greatly influence the derailment coefficient of the vehicle. | -A prototype program named GVDS.<br>-Multibody system | At a value of < 3MN/m, the result show that longitudinal stiffness significantly influences the derailment coefficient | The influence of secondary suspension was not put into consideration |
| Kraft & Lüdicke, 2022 | Conduct a sensitivity analysis due to operating forces and reliability analysis in fatigue design. | -SIMPACK software.<br>-Finite Element method. | It was found out that bump stops and vertical air springs and their properties have a substantial impact on the dynamic performance forces. | Investigate the sensitivity on other components affected by operating loads like bogie frame. |
| Milad et al., 2016 | Conduct global sensitivity analysis of behavior of a rail vehicle dynamically and the components of the suspension. | -HDMR method<br>- M-DRM method | The longitudinal primary spring affects performance significantly in the rail vehicle. | The study focused on longitudinal springs only neglecting other suspensions parameters. |

| | | | | |
|---|---|---|---|---|
| Cho et al., 2010 | Assess reliability of time dependent complex structure like interaction of vehicle and bridge. | An improved response surface method | Reliability is influenced by bridge – vehicle interaction and uncertainties in other design parameters like moment of inertia | Track excitations and other external factors were not considered. |
| Eom & Lee, 2010 | Running safety assessment of rail vehicle using multibody system dynamics. | ADAMS/Rail 1 software package. | The analysis showed that running safety is affected by speed, rail cant and curve radii | Track excitations and influence of the primary and secondary suspension not considered. |
| Zhou et al., 2014a | Influence of secondary and primary suspension system on the vehicle dynamic behavior | SIMPACK dynamic platform | Primary suspension damping and vertical stiffness have the highest level of sensitiveness. Vertical and longitudinal damping for secondary have the least effects on vibration. | Track excitations and other surrounding conditions were not put into account |
| Hunt et al.,1986 | Dynamic analysis of track –vehicle interactions. | -Experimental studies. -Matlab | The outcome indicated that large dynamic loads are caused by lateral irregularities on the wheelset leading to lateral oscillations. | Conduct studies about the influence of the suspension system. |

# Chapter Three

## Method and Material

### 3.1 Introduction

An insight has been provided through reviewing of the literature in determining the sensitivity of the system. Upon keen understanding and evaluation of the literature, this chapter provides the methodology of the research, study location, collected data and the procedure for determination of reliability and sensitivity.

### 3.2 Location of study

Addis Ababa light rail transit was inaugurated in September 2015 with a capacity of 286 passengers and carrying 60000 passenger in four directions. It covers a total length of 31.6km having 39 stations. The line is divided into two with one running from east to west of Addis Ababa covering a distance of 17.4km from Ayat to Torhailoch through Megenagna, Meskel square, Leghar and Mexico. The second line runs from north to south covering a distance of 16.9km running through Menelik II square, Merkato, Lideta, Gotera and Kality as shown in figure 3.1.

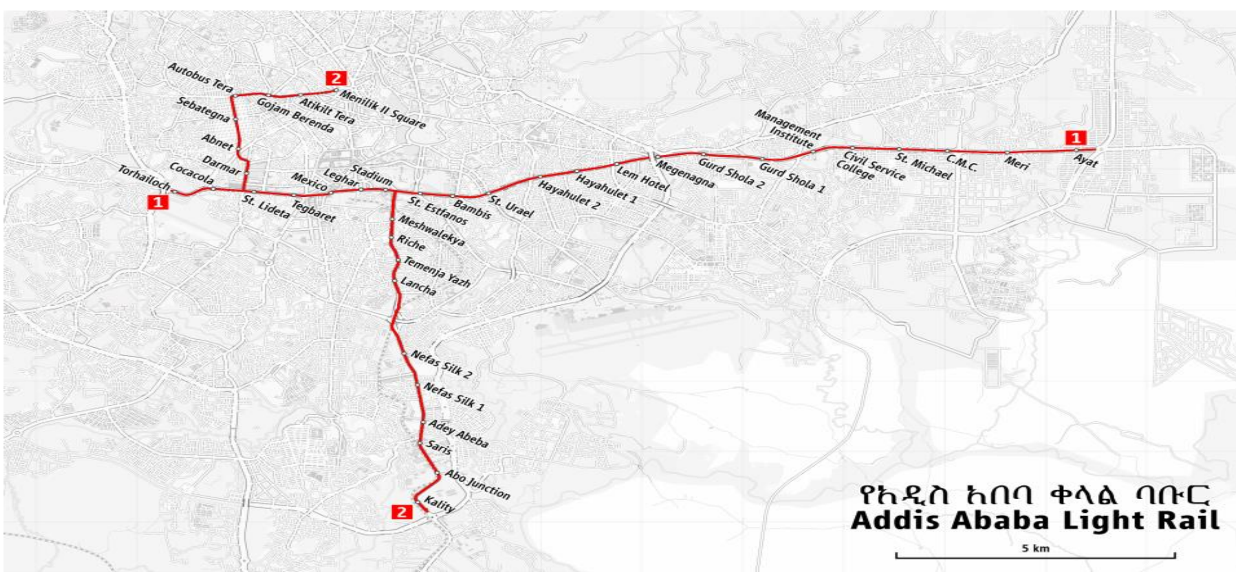


***Figure 3.1**: Map of AALRT.*

With a capacity to transport a large volume of passengers, ensuring that reliable and comfortable services are achieved is paramount. This can be attained through a careful study of the vehicles response to the operating environment and help develop mitigation procedures for proper services.

- **Flow chart of research methodology.**

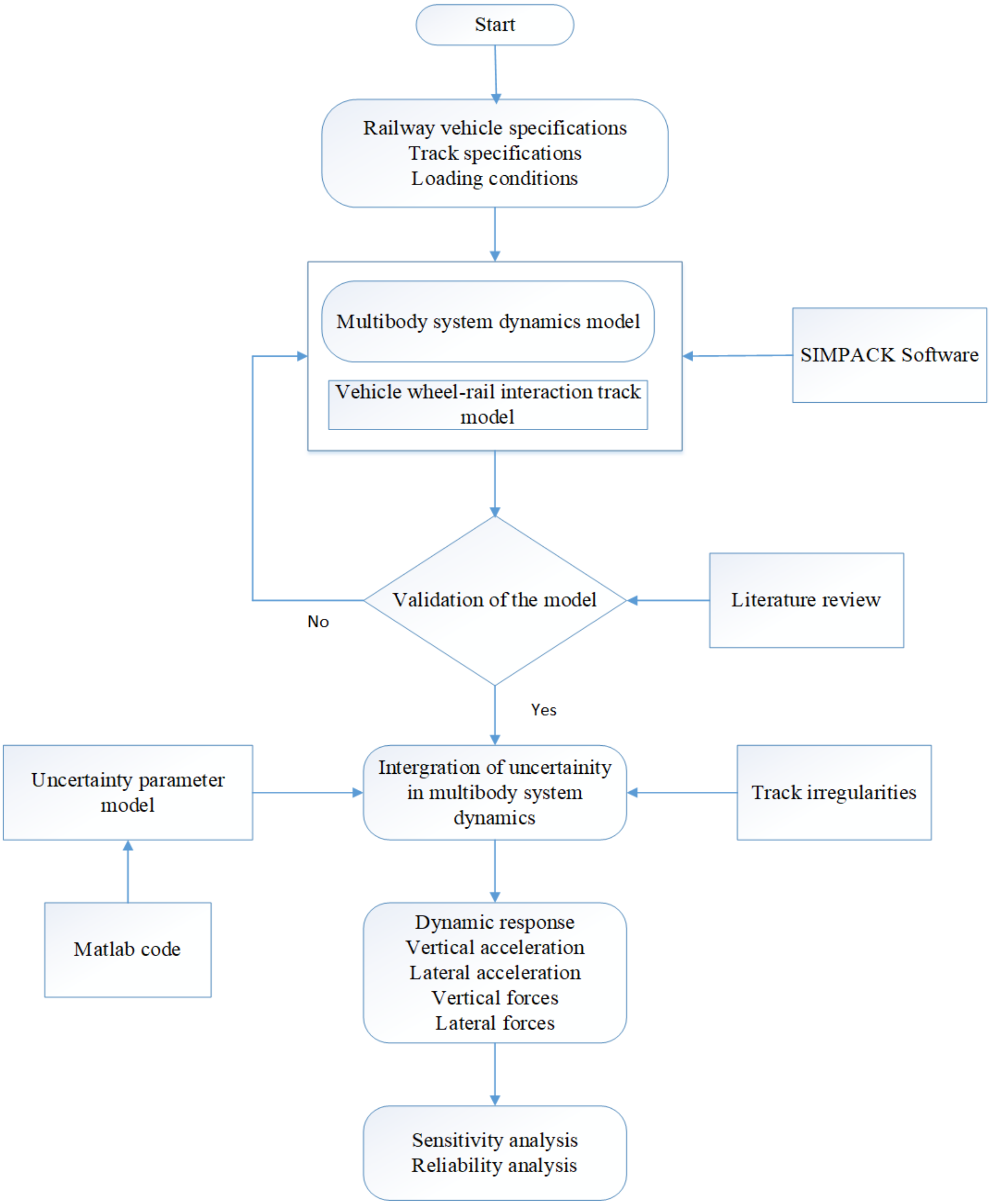


***Figure 3.2**: Sensitivity and reliability analysis flow chart.*

### 3.3 Data collection.

Data collection mainly focused on gathering data about track and vehicle specifications and maintenance data of the system.

#### 3.3.1 Railway vehicle.

***Table 3.1**: The design specifications of railway vehicle. (**Source**: AALRT passenger car specification sheet).*

| **Basic technical parameter** | **Parameters value** |
|---|---|
| Length | $28800mm$ |
| Width | $2650mm$ |
| Height | $3610mm$ |
| Total capacity | $317\ passengers$ |
| Axle load | $11.4ton$ |
| Speed | $70km/hr$ |
| Voltage range | $500V - 900V$ |
| Distance between wheels | $1600mm$ |
| Wheel base | $2750mm$ |

#### 3.3.2 Track specifications

The study was carried on a rigid and straight track consisting of the following design specifications.

***Table 3.2**: Basic parameters of the track*[44].

| **Parameter** | **Specification** |
|---|---|
| Profile | $S1002$ |
| Cant | 1/40 |
| Maximum gradient | 50% |
| Track gauge | $1.435m$ |
| Weight | $50kg/m$ |
| Friction coefficient | 0.4 |
| Maximum super elevation | $120mm$ |

### 3.3.3 Passenger Load conditions of the car body.

According to the design parameters of AALRT, each train is designed to carry a maximum of 317 passengers. In calculation, 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[45]. The number is broken down as shown in the table below.

***Table 3.3**: Loading conditions of AALRT*[45].

| Working condition | Seating capacity | Standing capacity | Total capacity | Total weight(t) |
|---|---|---|---|---|
| A0 | 0 | 0 | 0 | $\leq 44$ |
| A1 | 65 | 0 | 65 | $\leq 47.9$ |
| A2 | 65 | 189 | 254 | $\leq 59.2$ |
| A3 | 65 | 252 | 317 | $\leq 63$ |

### 3.4 Multi Body System description.

The rail vehicle constitutes of the carbody front and rear bogie with each bogie consisting of a bolster and two wheelsets. In this study, an assumption is made where the system is rigid and thus weight shifting and elasticity of the body is negligible. From figure 3.3, each rigid body consists of six degrees of freedom that is longitudinal 'x', lateral 'y' and vertical 'z'. The roll 'ϕ', yaw angle 'ψ' and pitch '$\beta$' help to locate the position of the center of gravity from the system of inertial.

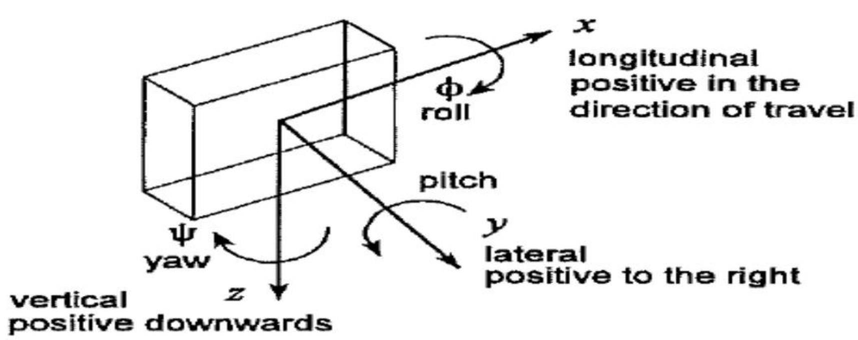


***Figure 3.3**: Rigid body degrees of freedom* [46].

The rigid bodies are connected to each other by suspension elements which range from frictional dampers, torsional springs and nonlinear dampers used absorb vibrations from track excitations.

- **Three bogie passenger rail vehicle DOF.**

The mechanical system may change in the space configuration and this change is described by degrees of freedom hence degrees of freedom can be independent rotations, displacements or movements that describe orientation of the system. The railway vehicle is not constructed of the individual mass points but constitutes of mass that is continuously connected referred to as distributed mass. The degrees of freedom of such systems are infinite. It is thus imperative to discretize these systems that is using simplified models to replace the vast quantity of degrees of freedom. These are referred to as Multiple-Degree of freedom (MDOF). Every bogie, wheelset and carbody have three rotational and translation degrees of freedom which include pitching, yawing, and rolling, vertical, lateral and longitudinal directions respectively.

***Table 3.4***: *Degrees of freedom of articulated passenger train.*

| Component | Quantity | Degree of freedom | Sum of DOF |
|---|---|---|---|
| Bogie | 3 | 6 | 18 |
| Wheelset | 6 | 6 | 36 |
| Carbody | 3 | 6 | 18 |
| Flexible articulation | 2 | 6 | 12 |
| **Total** | | | **84** |

The least quantity of independent coordinates needed to specify fully the locations of all system parts is equal to the quantity of degrees of freedom of a mechanical system at any particular time. Every particular mass in the system consists of 6 degrees of freedom relating to displacements i.e. longitudinal, vertical and lateral and three rotations i.e. roll, yaw and pitch.

### 3.4.1 Suspension system of passenger rail vehicle.

- **Primary suspension.**

The primary suspension seats between the wheelset and the frame of the bogie. The springs play a key role of stabilizing the movement of the vehicle along the track, minimizing the accelerations and dynamic forces resulting from track excitations and equalizing the vertical loads between the wheels. The assembly of primary suspension adopts chevron spring, and it is installed between frame and axle box. In addition, vertical rubber stop and primary lifting rod will be provided. The gap of primary suspension vertical rubber stop will be adjusted with shims installed on frame, and the primary suspension lifting rod will be installed at the bottom of frame side beam, to make sure the wheel set axle box will be lifted and transported together with bogie.

***Figure 3.4**: Primary suspension of chevron spring type found in AALRT.*

- **Secondary suspension.**

The secondary suspension assembly adopts side bearing loading type, and it is composed of bolster, secondary suspension spring, hydraulic damper, side bearing, lateral stop and secondary suspension lifting rod. The bolster adopts steel plate welding structure, and forging parts will be adopted on spring seat at both sides. The secondary suspension assembly adopts spiral steel spring, and rubber spring will be provided on the top, which is applicable to improve vehicle ride comfort. The secondary vertical rubber stop will be provided inside of spiral steel spring, when the spiral steel spring is compressed to a certain extent, the secondary suspension vertical rubber stop will be loaded simultaneously. Two lateral hydraulic dampers and two vertical hydraulic dampers will be provided between bolster and frame, and it will provide damping and vibration reduction for secondary suspension assembly.

The secondary suspension connects the carbody to the bogie. It helps in damping out of the carbody’s yaw motion. Yaw motion means the rotation about an axis normal to the plane of the track and thus plays a key role in ensuring the stability of the vehicle. A combination of primary and secondary suspension ensures filtering out of vibration frequencies from the wheelset and the suspension system isolates the carbody[47].

### 3.4.2 Mathematical modeling of the equation of motion.

The general expression of equation for the motion of railway vehicle is

$$[M]\{\ddot{y}\}+[C]\{\dot{y}\}+[K]\{y\}=\left[F_r(\omega)\right] \tag{3.1}$$

Where $[C]$, $[M]$ and $[K]$ are stiffness, mass and damping respectively.

The law of motion according to Newton along vertical, lateral and pitching motion for the carbody, wheelset and bogie in a general sense are formulated. The vehicle passes through a straight track. The equations of motion below are for the carbody, wheelset and the bogie.

- **Subsystem of the vehicle.**

The coordinate system considered is the Cartesian xyz where z is the vertical direction, y is the lateral and x is the longitudinal direction. Taking into consideration the degrees of freedom i.e longitudinal x, vertical z and lateral displacements and yaw, $\Psi$, pitch, $\beta$ and roll, $\phi$ angles.

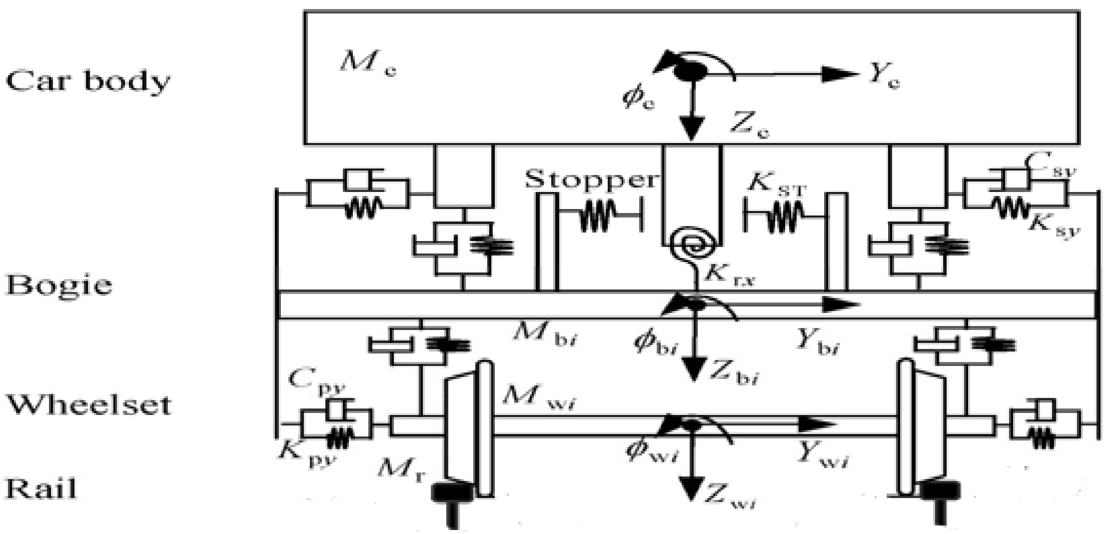


***Figure 3.5**: Free body diagram of the vehicle*[48].

- **Equations of motion of the carbody.**

The equations of motion below are for the carbody in the longitudinal, lateral, vertical, rolling, pitching and yawing directions.

$$
\begin{aligned}
&M_c\ddot{X}_c = -F_{xs1} - F_{xs2} - F_{xcf} - F_{xcb},\\
&M_c\ddot{Y}_c = F_{ys1} + F_{ys2} - F_{ycf} - F_{ycb} + M_c g\phi_{\sec} + F_{ycc},\\
&M_c\ddot{Z}_c = -F_{zc1} - F_{zs2} - F_{zcf} - F_{zcb} + M_c g + F_{zcc},\\
&I_{cx}\ddot{\phi}_c = -M_{xs1} - M_{xs2} + M_{xcf} + M_{xcb} + M_{xcc},\\
&I_{cy}\ddot{\beta}_c = -M_{ys1} - M_{ys2} + M_{ycf} + M_{ycb},\\
&I_{cz}\ddot{\Psi}_c = -M_{zs1} - M_{ZS2} + M_{zcf} + M_{zcb} + M_{zcc},
\end{aligned}
\tag{3.2}
$$

Where $M_c$ is the carbody mass, $I_{cx}$, $I_{cz}$, $I_{cy}$ are moments of inertia for rolling, yawing and pitching respectively. $M_{ysi}$ $M_{zsi}$ $M_{xsi}$ are moments between the frame of the bogie and the carbody and $F_{ysi}$ $F_{xsi}$ $F_{zsi}$ are mutual forces between the carbody and the bogie frame, $M_{yci}$ $M_{zci}$ $M_{xci}$ are moments due to inter-vehicle connections between carbodies, $F_{yci}$ $F_{xci}$ $F_{zci}$ denote inter-vehicle forces between adjacent carbodies. $M_{zcc}$ $M_{xcc}$ $F_{zcc}$ $F_{ycc}$ denote carbody external forces due to centripetal acceleration. $b$ and $f$ denote end and front of each car.

- **Equations of motion of the bogie frame.**

$$
\begin{aligned}
&M_b\ddot{X}_{bi} = F_{xsi} - F_{xf(2i-1)} - F_{xf(2i)},\\
&M_b\ddot{Y}_{bi} = F_{yf(2i-1)} + F_{yf(2i)} - F_{ysi} + M_b g\phi_{sebi} + F_{ycbi},\\
&M_b\ddot{Z}_{bi} = F_{zsi} - F_{zf(2i-1)} - F_{zf(2i)} + M_b g + F_{zcbi},\\
&I_{bx}\ddot{\phi}_{bi} = -M_{xf(2i-1)} - M_{xf(2i)} + M_{xsi} + M_{xcbi},\\
&I_{by}\ddot{\beta}_{bi} = -M_{yf(2i-1)} - M_{yf(2i)} + M_{ysi},\\
&I_{bz}\ddot{\Psi}_{bi} = -M_{zf(2i-1)} - M_{zf(2i)} + M_{zsi} + M_{zcbi},
\end{aligned}
\tag{3.3}
$$

$M_b$ is the bogie mass, $I_{by}$, $I_{bz}$, $I_{bx}$ are bogie moments of inertia, $M_{yfi}$, $M_{zfi}$, $M_{xfi}$ and $F_{zfi}$, $F_{yfi}$, $F_{xfi}$ are moments and mutual forces between wheelset and bogie. $M_{zcbi}$, $M_{xcbi}$ and $F_{zcbi}$, $F_{ycbi}$ are bogie external forces due to centripetal acceleration.

- **Equations of motion of Wheelset**

$$
\begin{aligned}
&M_w \ddot{X}_{wi} = F_{xfi} + F_{wrxi}, \\
&M_w \ddot{Y}_{wi} = -F_{yfi} + F_{wryi} + M_w g \phi_{sewi} + F_{ycwi}, \\
&M_w \ddot{Z}_{wi} = F_{zfi} - F_{wrzi} + M_w g + F_{zcwi}, \\
&I_{wx} \ddot{\phi}_{wi} = M_{xfi} - M_{wrxi} + M_{xcwi}, \\
&I_{wy} \ddot{\beta}_{wi} = M_{wryi} + M_{tbi}, \\
&I_{wz} \ddot{\Psi}_{wi} = M_{zfi} + M_{wrzi} + M_{zcwi},
\end{aligned}
\tag{3.4}
$$

$M_w$ is the wheelset mass, $I_{wy}, I_{wx}$ and $I_{wz}$ are wheelset moment of inertia, $M_{wryi}, M_{wrxi}, M_{wrzi}$ and $F_{wryi}, F_{wrxi}, F_{wrzi}$ are moments and contact forces between the rails and wheels, $M_{zcwi}, M_{xcwi}$ and $F_{zcwi}, F_{ycwi}$ denote wheelset external force due to centripetal acceleration, $M_{tbi}$ is braking or traction moment acting on the wheelset.

### 3.5 Simulation tools.

The study was carried out using SIMPACK dynamic software on a straight track of distance $3000m$ at a speed of $60km/hr$. $60km/hr$ is the maximum operating speed of the line while the straight line from St Lideta to Stadium which covers a total distance of approximately $3000m$ was considered. The following assumptions were made during the simulation process.

- The force of friction exists only between the rail and the wheel whereas it is negligible for other components of the railway vehicle.
- The masses of the carbody and the bogie were considered to be rigid.
- The axle box mass was assumed to be an integral part of the wheelset.
- The suspension system for dampers and springs exhibit linear characteristics.

- **Simulation of multibody system using SIMPACK software.**

This focusses on attempting to model the system's hypothetical situation in a real-life using computers in order to evaluate how the system would perform in actual conditions. The process involves setting up of an appropriate mechanical model to carry out the required simulation tasks which constitutes of the equation of motion for mathematical description[15]. Through variation of different input parameters like track irregularities, stiffness of the suspension and damping the changes in the behavior and performance of the vehicle was analysed. The performance and behavior of the vehicle at various cases of simulation will be based on identifying the vertical and lateral forces and accelerations, derailment coefficient and ride comfort index.

**3.5.1 Modelling of the rail vehicle.**

The 3D model of the vehicle was conducted using points connected using several entities like the suspension system, force elements, bogie frame, wheelsets and others. The wheelset is set up first and the wheel and rail profile specifications are as shown in Table 3.2. During modelling, defining of the reference system, xyz was done with z in downwards in the vertical direction,y is the lateral and x in the longitudinal directions. The vehicle components were modelled as rigid bodies and connected to each other by dampers and springs that make up secondary and primary suspension. The process of modelling starts with wheelset and track and further continuing to the entire vehicle.

- **Front wheelset.**

This first step involved creation of an axle shaft and renaming it $B_WS_1.The property dialog box is filled for mass, inertia as assigned in the subvar section. The axle shaft specifications are as

Primitive name- '$P_WS_1axle', type –cylinder, axle length-2.2m and diameter -0.66m having 25 as the number of planes. The same process is repeated upto $B_WS_6 while specifying the longitudinal distance of each wheelset. Wheelset joint, the wheelset movement is limited to only six degrees of freedom through rising the vertical distance to 0.43m above the track.

- **Rail-wheel pairs.**

The wheels on each side of the shaft are added by selecting $B_WS_1 in the rail-wheel property dialog box ,marker as $M_WS_1_BRF,setting the rotary joint to $J_WS_1 and joint state as pitch angle. The lateral distance was 0.75m.

Rail parameters are also specified where its lateral distance is also 0.75m, cant 1:40 and coefficient of friction is 0.4.The fastsim algorithm is employed to help in specifying the tangential forces that create creep forces. Parametrizing of the wheel and rail is completed. The wheel on each side is now create by clicking apply and create elements and OK. The process is repeated to model to other five remaining wheelsets.

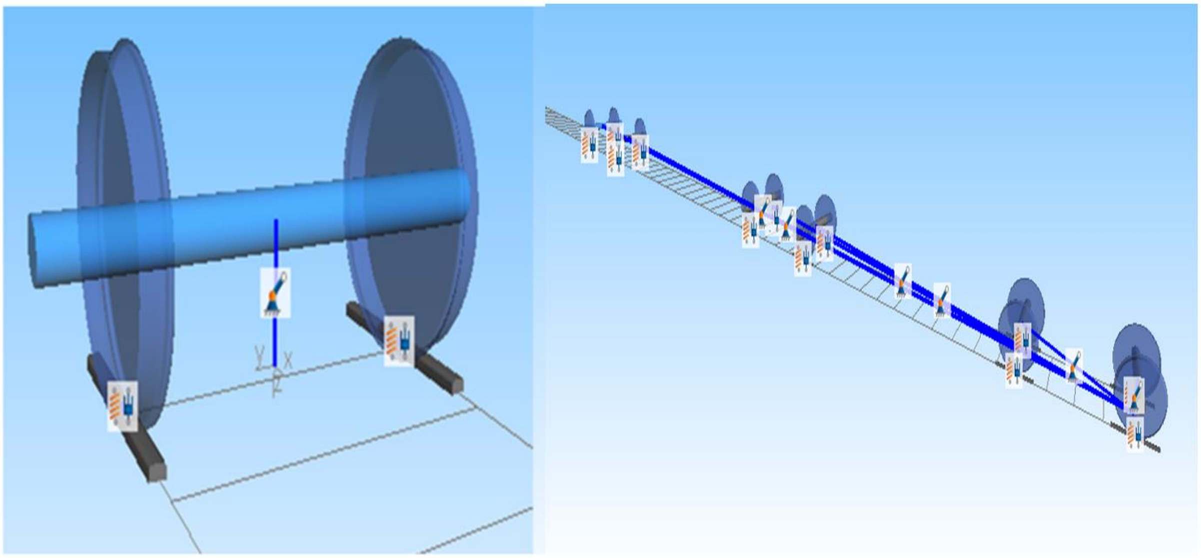

***Figure 3.6****: Modelled wheelsets of the carbody.*

- **Bogie frame.**

The bogie frame is created by adding it as body and renamed as $B_BF_1 and adding the inertia and mass properties in the dialog box as indicated in appendix A.2. The detailed dimensions of the bogie frame are entered through the geometry primitive properties dialog box as in figure 10.

This process is repeated for all bogie frames at proper longitudinal distances.

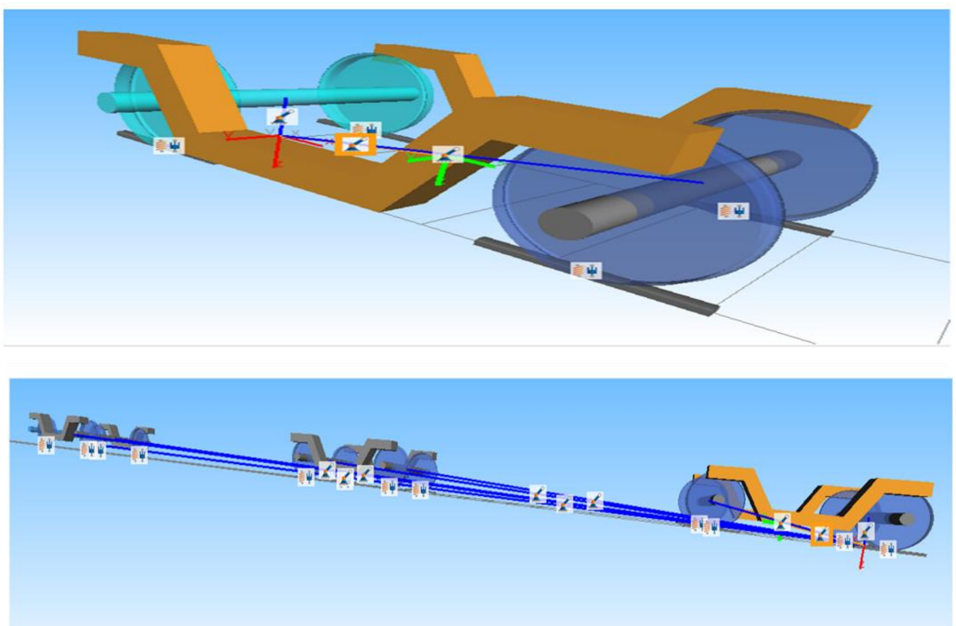

***Figure 3.7****: Bogie frame assembly.*

- **.Creation of the bolster frame.**

The bolster frame acts as a connection between the bogie frame and the carbody through, supports the secondary suspension system and its mass is negligible during analysis. It created through the bodies and naming it and specifying the inertia and mass properties, using the primitive properties to specify the actual dimensions. The mass of the mass were very small such that they tend towards zero and are neglected during analysis. The joint is fixed with no movement 0:0 degree of freedom to ensure effective connection of the carbody and the bogie frame.

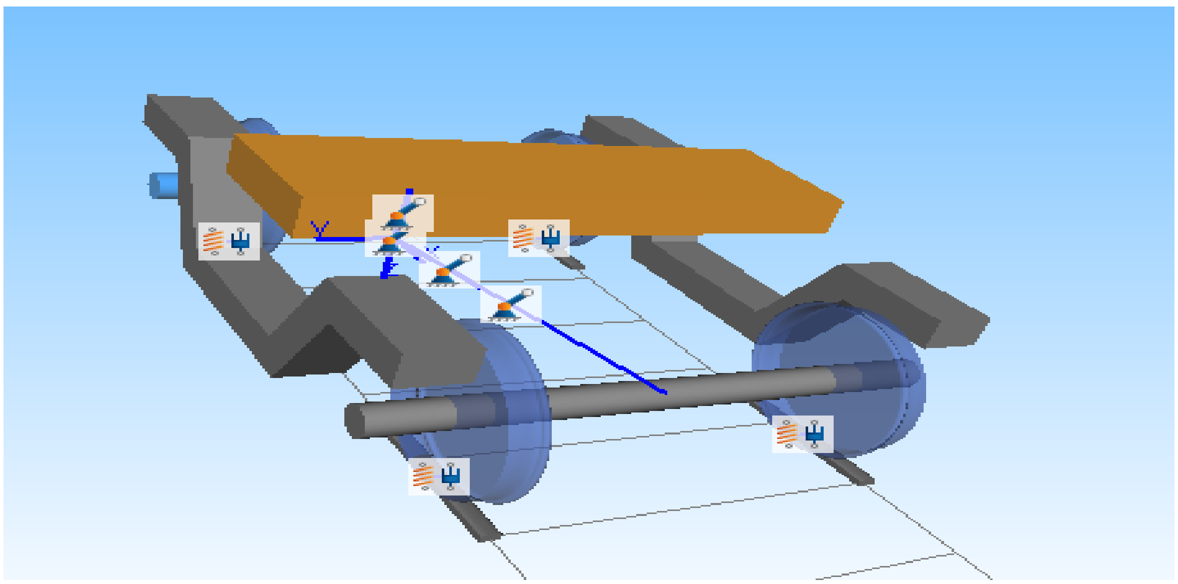

***Figure 3.8****: Dummy bolster integrated onto the bogie frame.*

- **Primary suspension.**

This acts as a linkage between the bogie frame and the wheelset and is modelled with the help of force elements. The first step involves setting of markers o the wheelsets and the bogie frame to indicate where the force elements are to be positioned. In the force elements properties, type 5, Spring-Damper parallel cmp which applies damper and spring forces multiple directions was chosen and various stiffness and damping values of the suspension input. Also from and to positions of the force elements are specified to clearly position them in the right location and then okay.

- **Secondary suspension.**

Under this markers are created between the dummy bolster and the center of the side frame of the bogie. The same process is repeated as for primary suspension.

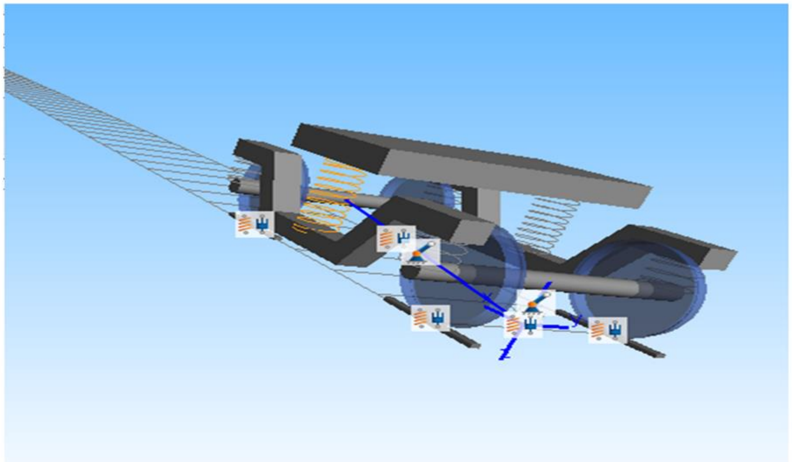

***Figure 3.9****: Suspension system integrated in the model.*

- **Railway vehicle Carbody.**

The vehicles are created as bodies and renamed $B_CB which enables specification of the inertia and mass properties. The actual dimensions of each carbody are individually outlined in the geometry primitive property. Markers are also created to locate where the dummy bolster will be connected to the carbody to create and assembly. The joint of the carbody is type 5- general rail track joint.

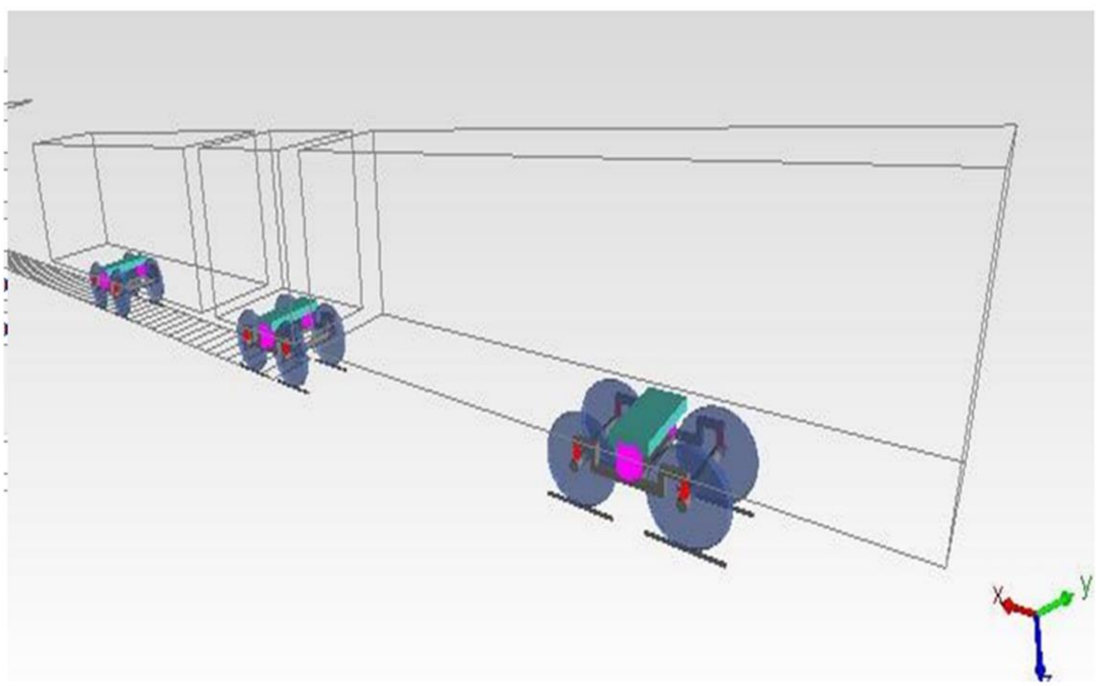


***Figure 3.10****: Assembly of the carbody.*

- **Track irregularities.**

This was conducted with the help of input function. From the maintenance data, the track irregularities under consideration were rail related irregularities which include vertical left and right and lateral left and right. The graphs below indicate the irregularities under consideration.

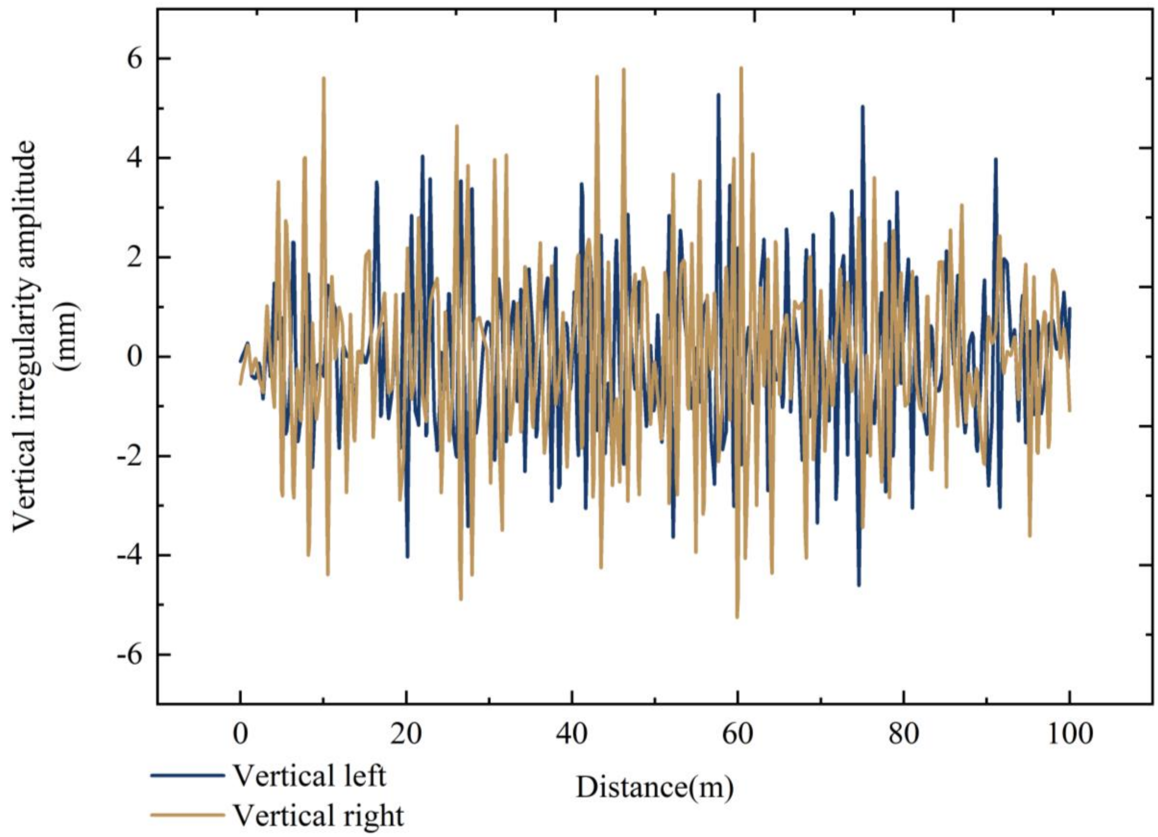


***Figure 3.11****: A graph showing vertical irregularities applied to the system.*

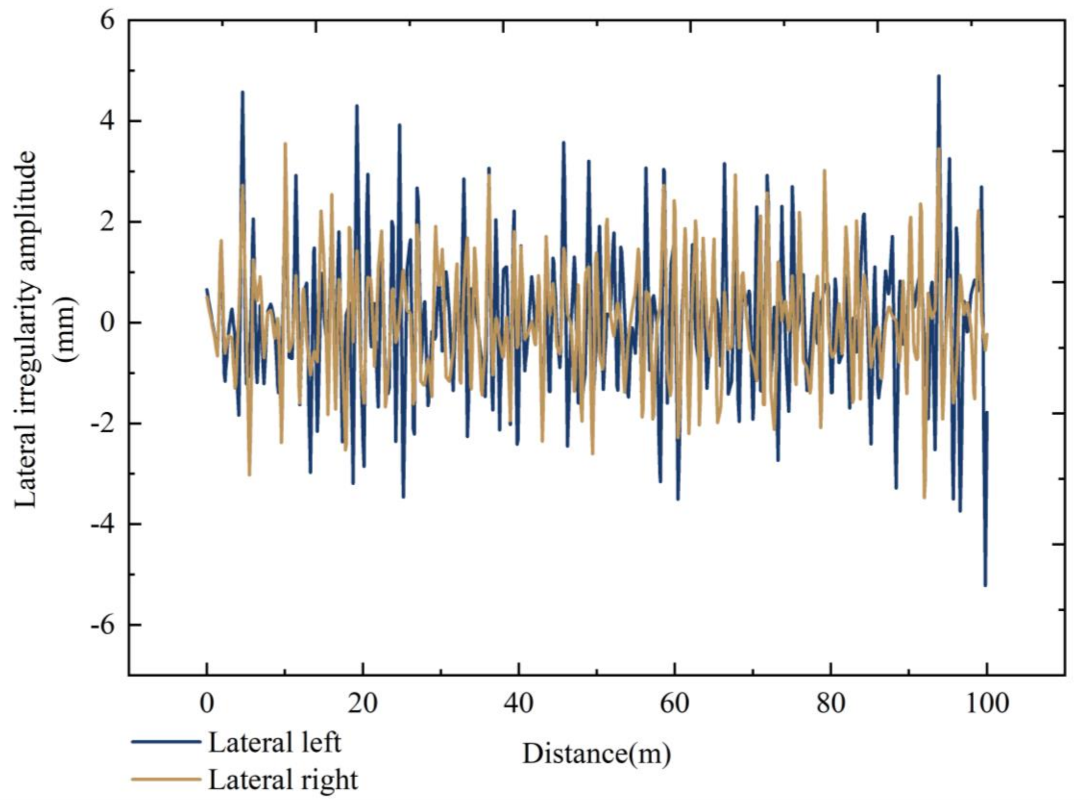


***Figure 3.12****: A graph showing lateral irregularities applied to the system.*

The data was arranged and input into Simpack as one set using input function set. This is followed by adding excitations of the different types of irregularities. The type of excitation is 109-from input function (adv.). The track is updated by positioning the irregularities in their respective positions on the track and specifying the actual length of the irregularities.

### 3.5.2 Validation of the model.

This was conducted through static equilibrium and preload calculation in SIMPACK. The preload load solver helps to calculate the vertical force at the wheel-rail contact area and should be the same for all the wheels. The only factor considered is gravity of different components of the vehicle. The preload load solver determines the preloads such that the system is in equilibrium i.e. the residual accelerations (joint body acceleration) are zero or are extremely small that they can be approximately to zero. The vehicle was run at $60km/hr$ for a distance of $3000m$.The peak dynamic vertical force obtained was $50.766kN$which was very close to $50.766kN$ static force under preload conditions whereas the joint body acceleration was $3.9027707x.10^{-8}m/s^2$ which is very small $(< 0.01m/s^2)$and tending towards zero[49]. Figures are shown in appendix c.

Furthermore the vertical and lateral accelerations realised were $2.468 m/s^2$ and $2.0247 m/s^2$ and compared against UIC 518 standard where the maximum allowable accelerations are $2.5 m/s^2$ for both lateral and vertical accelerations [50].

### 3.5.3 Vehicle response.

As the vehicle runs on the track, it constantly reacts to several imperfections such as track irregularities. The reaction of the vehicle can be obtained through simulation by using the measured imperfections as a section of the input data. Several parameters can be measured however the most commonly evaluated are accelerations and rail-wheel forces. Accelerations and suspension system movement in the system influence ride comfort and safety of operations.

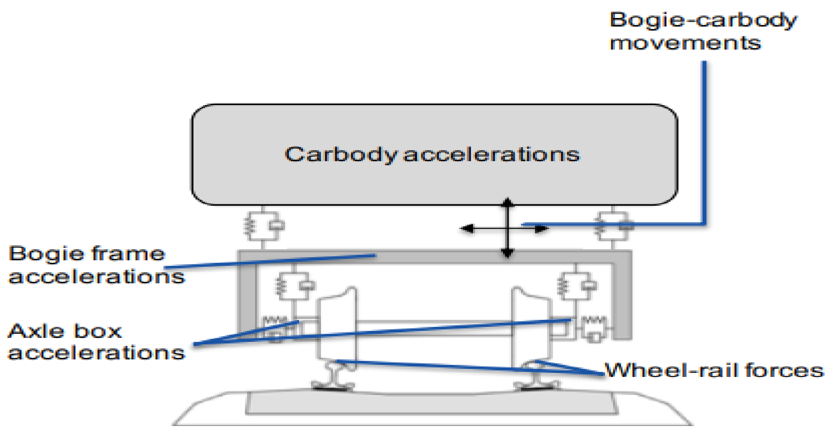


***Figure 3.13**: Schematic view of some responses of the vehicle [42].*

In order to investigate the influence of suspension parameters on the various dynamic indexes, all the primary and secondary suspension parameters of the railway vehicle were chosen as design variables and the uncertainty quantified through the application of standard deviation and error to get upper and lower limits of the parameters. Motion stability and ride comfort indexes are the major indicators of the vehicle that will be used and are based on GB/T 5599-1985[51] and EN12299.These include ride index, and derailment coefficient.

- **Ride comfort index.**

The method was used in standard railway applications EN12299[52].Under this, ride index method is used to evaluate the ride comfort and quality of vehicles by establishing a relationship between sensitivity to vibrations and comfort index. During the movement of the vehicle, it encounters excitations which generate vibrations. Mean comfort to vibrations are quantified by ride comfort index and help to provide information about the dynamic characteristics of the system. The assessment is usually based on a scale of 1 to 4 where vibrations at level 4 are harmful to the passengers when exposed for longer periods.

The perception of vibrations by the passengers which occurs during the movement of the vehicle is based on comfort index, Wz. Table 3.5 shows the range of values of the index and their significance[53].

***Table 3.5**: Range of values for ride comfort index*[53].

| Ride Comfort Index, Wz | Vibration sensitivity |
|---|---|
| 1.0 | Just noticeable |
| 2.0 | Clearly noticeable |
| 2.5 | More pronounced but not unpleasant |
| 3.0 | Strong, irregular but still tolerable |
| 3.25 | Very irregular |
| 3.5 | Extremely irregular, unpleasant, annoying, prolonged exposure intolerable |
| 4.0 | Extremely unpleasant, prolonged exposure harmful |

- **Derailment coefficient.**

Derailment coefficient is one of the primary procedure parameters used to evaluate safety reliability of the running vehicle. This occurs when greater lateral forces act on the wheels and results into the flange climbing over the rail leading to derailments. The derailment criterion was studied by Nadal given by the ratio of lateral force to vertical force and should not exceed 0.8[54].

### 3.6 Sensitivity analysis.

This was employed to describe how the output of the model is dependent on the input parameters.

The suspension system was identified as the one of vital systems that play a key role in the movement of the vehicle and thus was considered as the centre of the study. The uncertainty in the suspension system was quantified through generation of random numbers which are close to the specifications (mean values).

### 3.6.1 Uncertainty analysis.

This helps in conducting an assessment of the effects of inputs on the model outputs. It involves several sampling methods such as Latin hypercube sampling and Monte Carlo.

- **Uncertainty quantification.**

The random vector $Z$ is specified in the probability space $(\Omega, f, \mu_Z)$ where $f$ the borel set is built on $\mu_Z$ as probability measure and $\Omega$ .The interest is in computing the density function of the output and the first moments like variance and mean.

$$\mu_u(t) = E\left[u(t,Z)\right]_{\rho z} = \int_{\Omega^d} u(t,z) dF_Z(z), \tag{3.5}$$

$$\sigma_u^2(t) = \text{var}\left[u(t,Z)\right]_{\rho z} = \int_{\Omega^d} (u(t,z) - \mu_u(t))^2 dF_z(z), \tag{3.6}$$

- **Probabilistic assessment.**

The assessment was based on analysis of uncertainty of the model output and also apportioning the various sources of uncertainty to the inputs of the model. Standard deviation of 15% was applied to describe the uncertainty in the input suspension values of the model and involved the following major steps:

1. Identification of uncertainty parameters and generation of $N$ samples from each probability distribution to get matrix $Y$ and may involve scatter plots.

Consider a model in the form;

$$Y = f(Z_1, Z_2, \ldots\ldots., Z_r) \tag{3.7}$$

Where $Z_r$ ($i = 1, \ldots\ldots.., r$) is the input of the model and the output is $Y$ . The model input statistical distribution has an $N$ probable combination of samples to generate the matrix.

$$M = \begin{bmatrix} Z_1^{(1)} & Z_2^{(1)} & \ldots & Z_r^{(1)} \\ Z_1^{(2)} & Z_2^{(2)} & \ldots & Z_r^{(2)} \\ \ldots & \ldots & \ldots & \ldots \\ Z_1^{(N)} & Z_2^{(N)} & \ldots & Z_r^{(N)} \end{bmatrix} \tag{3.8}$$

$Y$ for each column is computed to come up with the outputs of the model vector.

2. Perform a single model simulation of each row vector giving $N$ values of the output of the model. The output of the model is assumed to be a scalar.

$$\boldsymbol{Y} = \begin{bmatrix} y^{(1)} \\ y^{(2)} \\ \cdots \\ y^{(N)} \end{bmatrix} \tag{3.9}$$

The $Y$ elements are used to calculate the dynamic response indexes of the vehicle.

- **Realization of the sensitivity analysis.**

The output and all input parameters have to be defined. The input parameters include the mean values of primary and secondary suspension system, track irregularities, carbody specifications and track specifications.

**-Operating conditions;** The effect of track irregularities on the dynamic performance of the multibody system were studied.

**-Influence of the suspension system;** The effect of both the primary and secondary suspension system is studied for different value putting into consideration the standard deviation during analysis.

The parameters identified are shown in the table below.

***Table 3.6****: Suspension system parameters for AALRT under consideration (****Source****: AALRT passenger car specification sheet).*

| **Parameters** | **Symbols** | **Unit** | **Mean values.** |
|---|---|---|---|
| Lateral primary suspension stiffness | $k_{py}$ | $kN/m$ | 750 |
| Vertical primary suspension stiffness | $k_{pz}$ | $kN/m$ | 600 |
| Longitudinal primary suspension stiffness | $k_{px}$ | $kN/m$ | 600 |
| Lateral primary suspension damping | $c_{py}$ | $kNs/m$ | 200 |
| Vertical primary suspension damping | $c_{pz}$ | $kNs/m$ | 170 |

| | | | |
|---|---|---|---|
| Longitudinal primary suspension damping | $c_{px}$ | $kNs/m$ | 120 |
| Lateral secondary suspension stiffness | $k_{sy}$ | $kN/m$ | 350 |
| Vertical secondary suspension stiffness | $k_{sz}$ | $kN/m$ | 170 |
| Longitudinal secondary suspension stiffness | $k_{sx}$ | $kN/m$ | 300 |
| Lateral secondary suspension damping | $c_{sy}$ | $kNs/m$ | 200 |
| Vertical secondary suspension damping | $c_{sz}$ | $kNs/m$ | 400 |
| Longitudinal secondary suspension damping | $c_{sx}$ | $kNs/m$ | 200 |

The parameters identified were based on those that play avital role in the operation of the train and the most common ones. Uncertainty in the suspension system was predicted by generating six sets of random numbers using standard deviation of 15% as shown in appendix A.1 using matlab which acted as inputs to the system thus six simulations were conducted to evaluate the influence of this uncertainty. Furthermore the influence of vertical stiffness and damping for primary and secondary suspension were independently analysed to identify the parameters with the greatest and least sensitivity. The results will focus on vertical and lateral forces, accelerations, and ride index and derailment coefficients.

**Optimization of the suspension system**

The suspension parameters taken as design variables were arranged in multiple groups of suspension parameter samples based on the generated random variables in appendix A.1.The ride index for each sampling point was determined from SIMPACK software.The sampling groups applied during optimization process are shown in appendix Table A.3. The parsing and replacment technique was used through substituting several suspension parameters and determining the ride index at different combinations of the suspension parameters.

# Chapter Four

## Results and Discussion

### 4.1 Influence of uncertainty in suspension system

The effect of uncertainty in the suspension system was carried out through determining the dynamic vertical and lateral forces at rail/wheel contact for various values of damping and stiffness of the suspension system. The wheels for front wheelset are put into account because they experienced the maximum forces compared to other wheels. Figure 4.1 shows a general trend of the vertical and lateral forces experienced by the vehicle for sim 1 and sim 6.

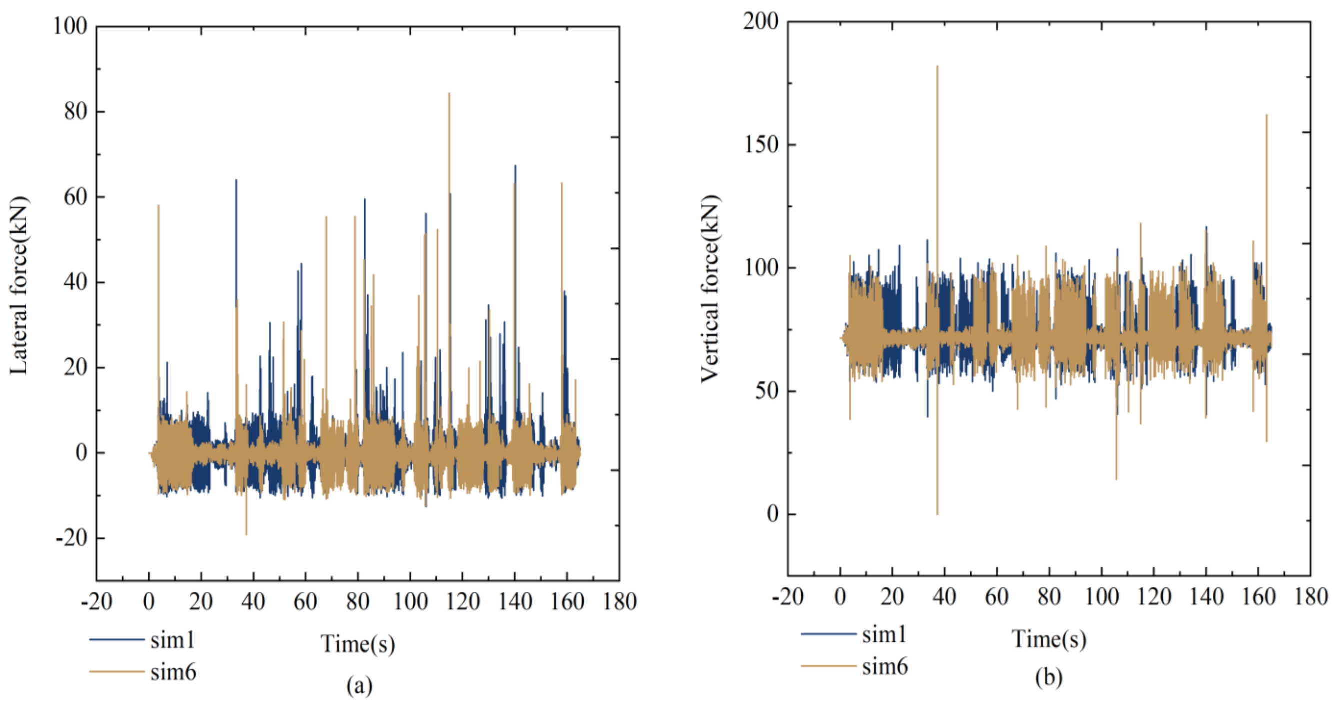


***Figure 4.1:*** *Lateral and vertical dynamic forces.*

It is observed in Figure 4.1 that the vertical forces at the rail-wheel contact become greater with increase in damping and stiffness values of both secondary and primary suspension system from $116.78kN$ at simulation 1 to $181.86kN$ at simulation 6. This is because a stiffer suspension system increases rigidity and the force transmission of dynamic impact loads through the system i. e from the carbody to the wheels and vice versa. The forces at the rail-wheel contact area are further amplified by the presence of track irregularities. The lateral forces in Figure 4.3 have a very small increment from $67.34kN$ at simulation 1 to $84.36kN$ at simulation 6 compared to the vertical forces. This is because lateral forces are not very significant on straight tracks and their effect is much felt in curved tracks. Since analysis was conducted on a straight track, these lateral forces were only influenced by the amplitude of the irregularities on the track. A graph of maximum vertical and lateral forces against time is shown in Figure 4.3.

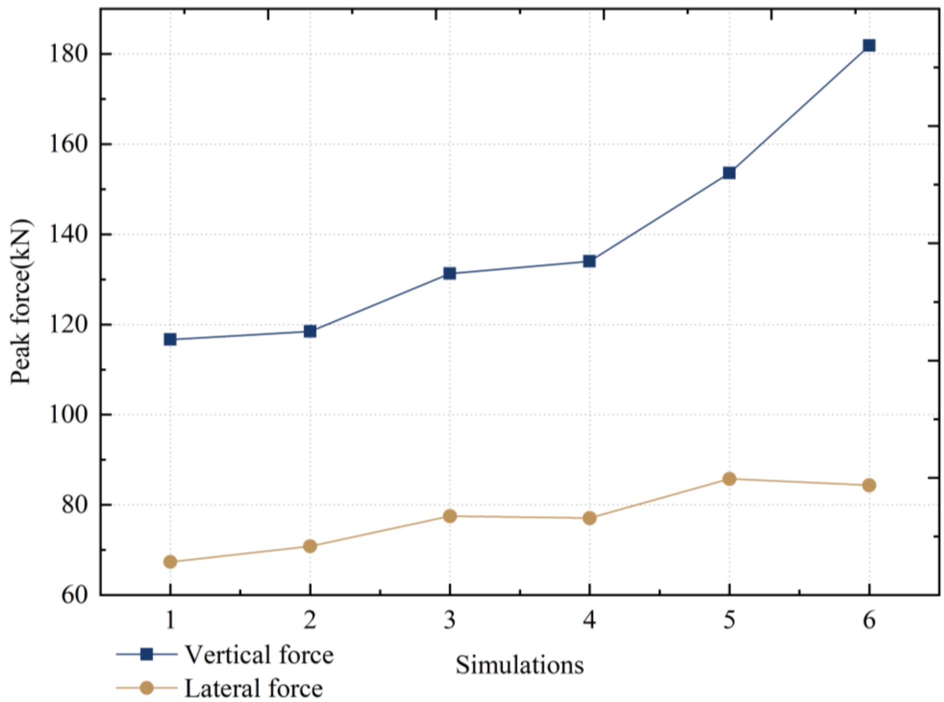


***Figure 4.2***: *Peak dynamic forces at different simulations.*

### 4.2 Influence of uncertainty on carbody vehicle accelerations

Table 4.1 shows values of lateral and vertical accelerations obtained from SIMPACK and shows how accelerations of the railway vehicle are affected by uncertainty at different parameter values of the suspension system.

***Table 4.1***: *Peak vertical and lateral acceleration values*

| **Simulations** | **Maximum lateral acceleration** $(m/s^2)$ | **Maximum vertical acceleration** $(m/s^2)$ |
|---|---|---|
| 1 | 2.407 | 2.278 |
| 2 | 2.471 | 3.167 |
| 3 | 3.039 | 4.023 |
| 4 | 3.041 | 4.326 |
| 5 | 3.079 | 4.823 |
| 6 | 3.223 | 7.407 |

A graph of peak vertical and lateral accelerations against time is plotted in Figure 4.4 to give an account of how the suspension system and track irregularities affect them. It is shown in the graph that both vertical and lateral accelerations have a general trend of increasing with increase in the values of the suspension however it is gradual for lateral accelerations. Vertical accelerations increase from $2.278m/s^2$ at sim 1 to $7.407m/s^2$ at sim 6 due to increase in damping and stiffness value of the suspension system coupled with track excitations. Lateral acceleration experience a gradual increment from $2.407m/s^2$ at sim 1 to $3.223m/s^2$ at sim 6.This is because lateral accelerations have substantial influence when cornering (curves) and therefore have less effect on a straight track. The magnitude of lateral accelerations experienced by the railway vehicle is mainly due to the amplitude of track irregularities. A graph of maximum values of lateral and vertical accelerations at different simulations is shown in Figure 4.4.

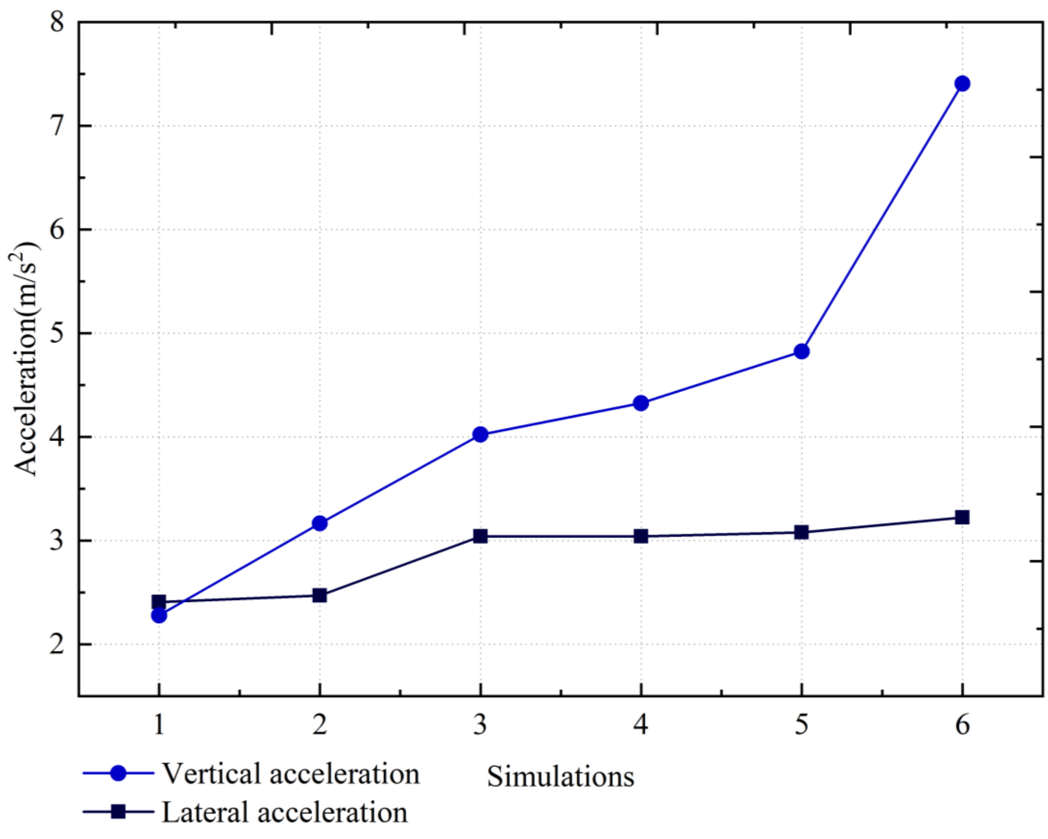


***Figure 4.3****: Dynamic vertical and lateral accelerations at different simulations.*

It is therefore evident the lower values of suspension stiffness and damping result into lower values of accelerations and consequently less rail-wheel contact forces and wear.

### 4.3 Dynamic performance reliability of vehicle

Running safety against derailment of the vehicle is used to evaluate the reliability of the railway vehicle[28]. Graph 4.5 shows the values obtained in each case considered for the suspension system. It is given by the ratio of lateral force to vertical force according to Nadal's expression.

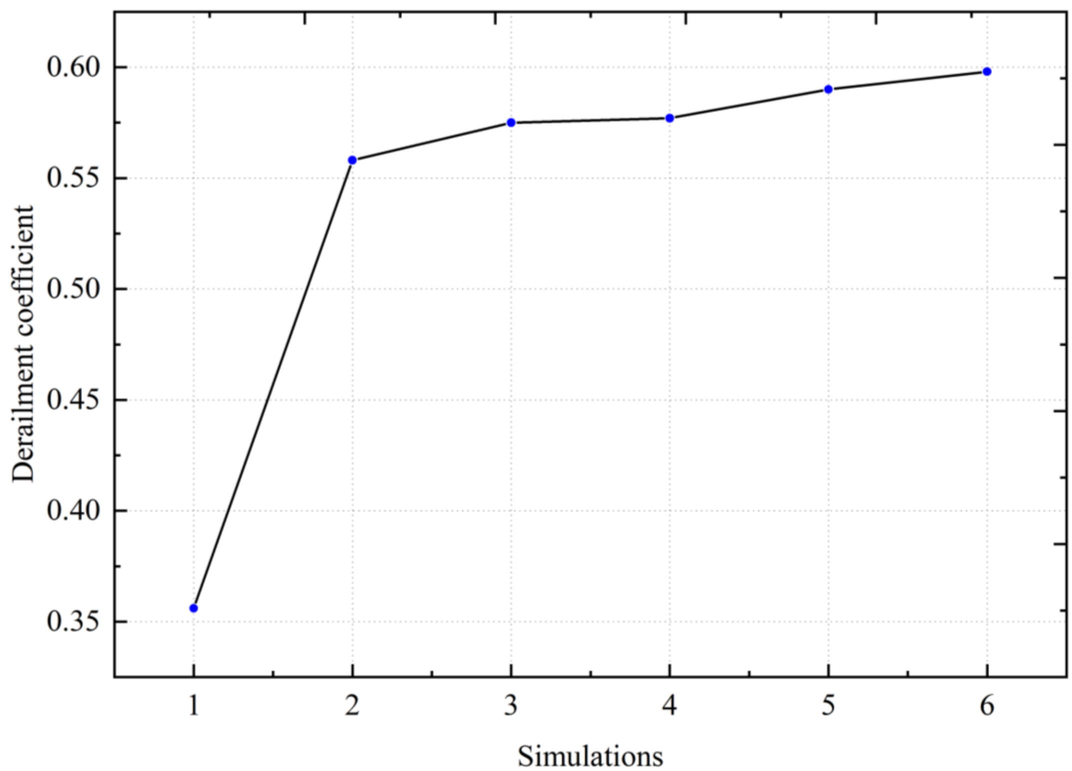


***Figure 4.4*** *: Derailment coefficient values.*

It can be observed that at low values of stiffness and damping in the suspension system, the coefficient is highly sensitive and increases rapidly however as the values increase further, the coefficient approaches almost a constant level implying that stability has been achieved in the system and the influence is reduced. The coefficient values realised are within the safe working limits of $\leq 0.8$ according to Nadal's expression. This shows that the suspension system plays a role in ensuring that running safety and operational reliability of the vehicle is achieved. It is also observed that at 15% less values of the system from the standard values, the lateral forces experienced increase by approximately 5% of the standard force because softer primary suspension system rarely aligns perfectly well with regards to the railway carbody contributing to hunting motion and thus has a negative impact on the stability of the vehicle.

## 4.4 Sensitivity of vehicle to uncertainty in suspension.

The dynamic performance of the vehicle with respect to the ride comfort index is evaluated according to vibration sensitivity and ride index values. This helps in interpretation of the ability of the vehicle suspension to maintain a certain level of ride index and human comfort. These two are crucial in the modelling and evaluation process of the running characteristics of the vehicle. The running characteristics are greatly affected by the secondary and primary suspension system and track irregularities which interact with the multibody system components like carbody and wheels. The ride index values are compared against the standard EN12299 to establish the degree of influence. Table 4.2 shows ride index values obtained at different values of the suspension system as indicated in appendix A.1 from simulation 1 to simulation 6 and the values are compared against those in Table 3.5 as per the standard.

***Table 4.2**: Ride index values*

| **Simulations** | **1** | **2** | **3** | **4** | **5** | **6** |
|---|---|---|---|---|---|---|
| Ride index, vertical | 2.52 | 2.53 | 2.54 | 2.55 | 2.55 | 2.57 |
| Ride index, lateral | 2.0 | 1.857 | 1.924 | 1.93 | 1.832 | 1.937 |

It is indicated in the Table 4.2 that softer stiffness and damping values of the suspension system contribute to low values of ride index and continues to increase with increase in damping and stiffness values. The ride index values increase from 2.52 to 2.57 implying that there is a general deterioration from ‘clearly noticeable’ to ‘more pronounced but not unpleasant’. This implies that lower values of damping and stiffness result into relatively better ride comfort of the vehicle. The maximum lateral ride index achieved is 2.0 falling in ‘clearly noticeable region’ and comfortable enough for passenger ride. This is because lateral forces and accelerations are more significant in curved tracks. The analysis was conducted on a straight track thus only track irregularities affected the lateral ride index. For better, comfortable and smooth passenger ride experience, the ride index should at least be nearing to 2.0 hence creating a need for optimization of the suspension system to achieve better results.

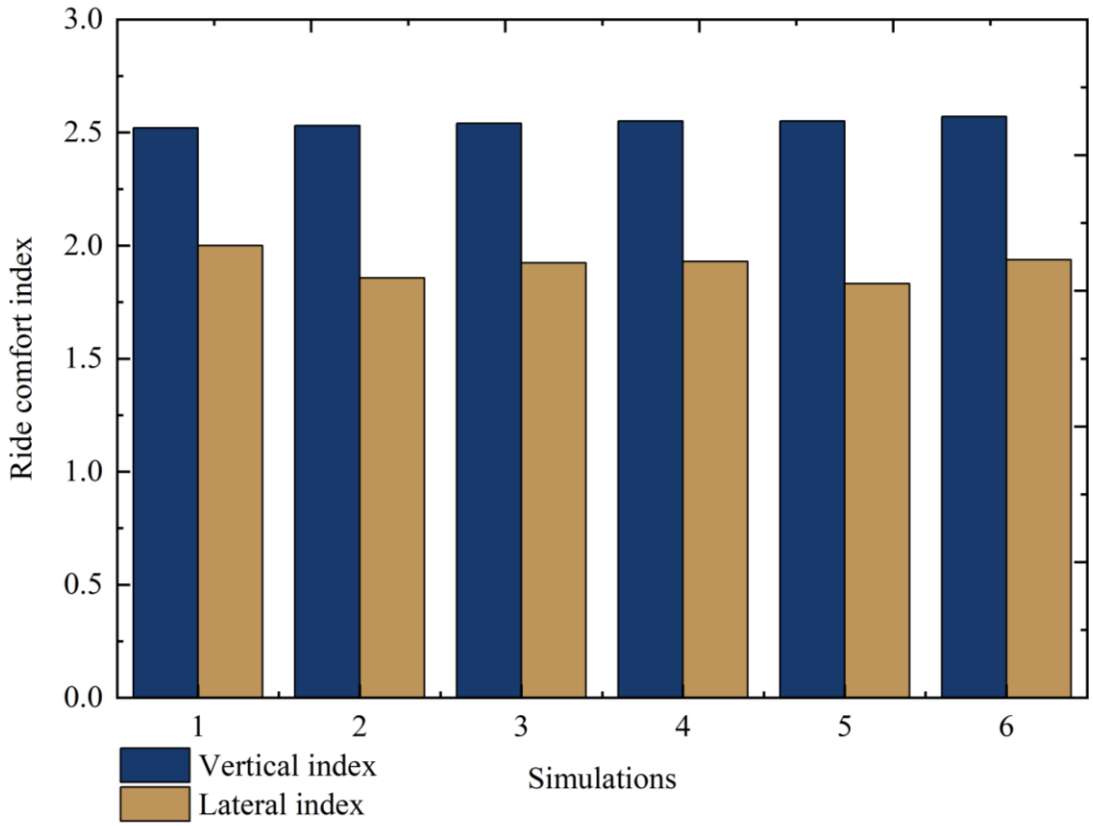


***Figure 4.5**: Vertical and lateral ride index values.*

### 4.4.1 Influence of primary suspension vertical damping on the sensitivity of the vehicle.

To conduct this study,the vertical stiffness and damping of the primary suspension system was considered because it plays a key role influencing the ride comfort index on a straight track. One parameter at a time(OAT) method was applied to investigate the influence of each parameter.

The values of damping were varied from $126.356\, kNsm$ to $204.943 kNsm$ as indicated in appendix A.1 and their corresponding ride index values obtained to determine their effect while maitaining all other parameters constant at design parameters in Table 3.6.

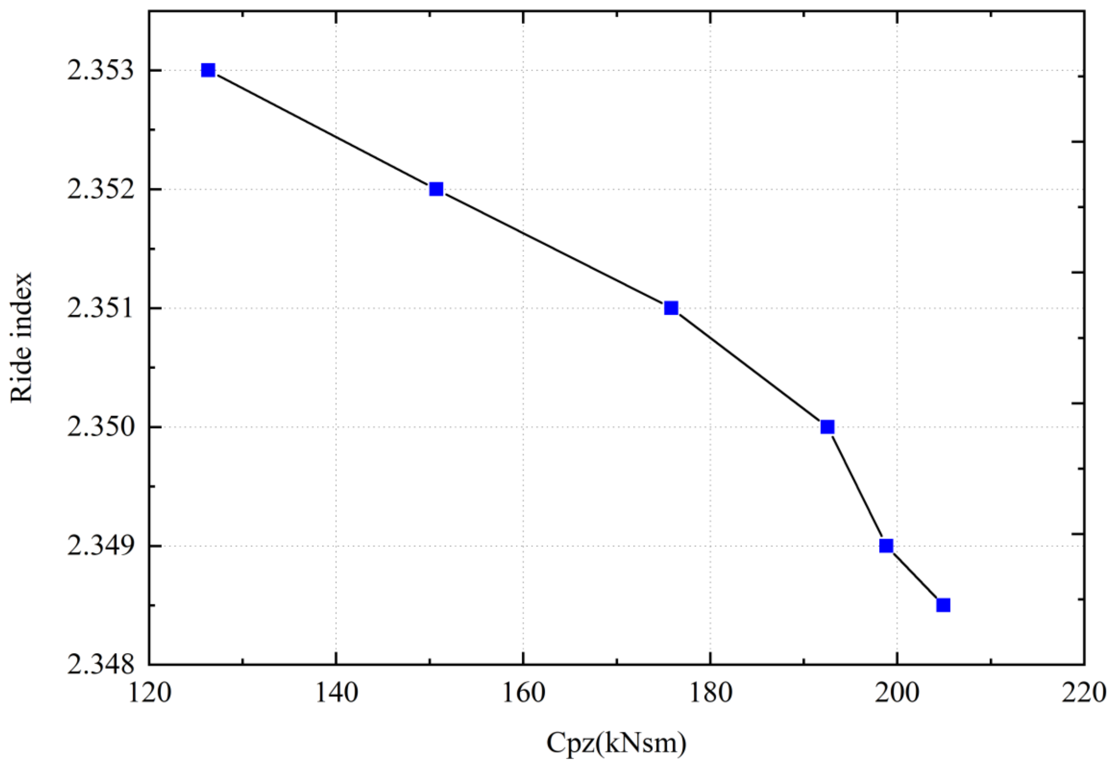


***Figure 4.6****: Effect of primary suspension damping on ride index.*

It is clearly indictaed in Figure 4.7 that damping in the primary suspenison has an extremely small or almost negligible influence on the ride comfort of the vehicle.With increase in the damping values,the ride index is observed to change from 2.353 to 2.348 which is almost no change hence doesnot greatly influence the vibration sensitivity of the vehicle.This is partly due to the effect of the bogie mass situated between the carbody and the primary suspension which tends to decrease the efficiency of primary suspension damping.It is also important to note that it is not possible to increase the damping of the primary suspension to as much as desired because of the magnitude of rail-wheel dynamic forces.

### 4.4.2 Influence of secondary suspension vertical damping on the sensitivity of the vehicle.

The effect of the damping ofsecondary suspenison on the vibration sensitivity of the vehicle was determined by varrying values from $38.7702\ kNsm$ to $489.382\ kNsm$ and the corresponding ridex index values obtained as shown in Figure 4.8 while keeping all other parameters constant at design parameters in Table 3.6.

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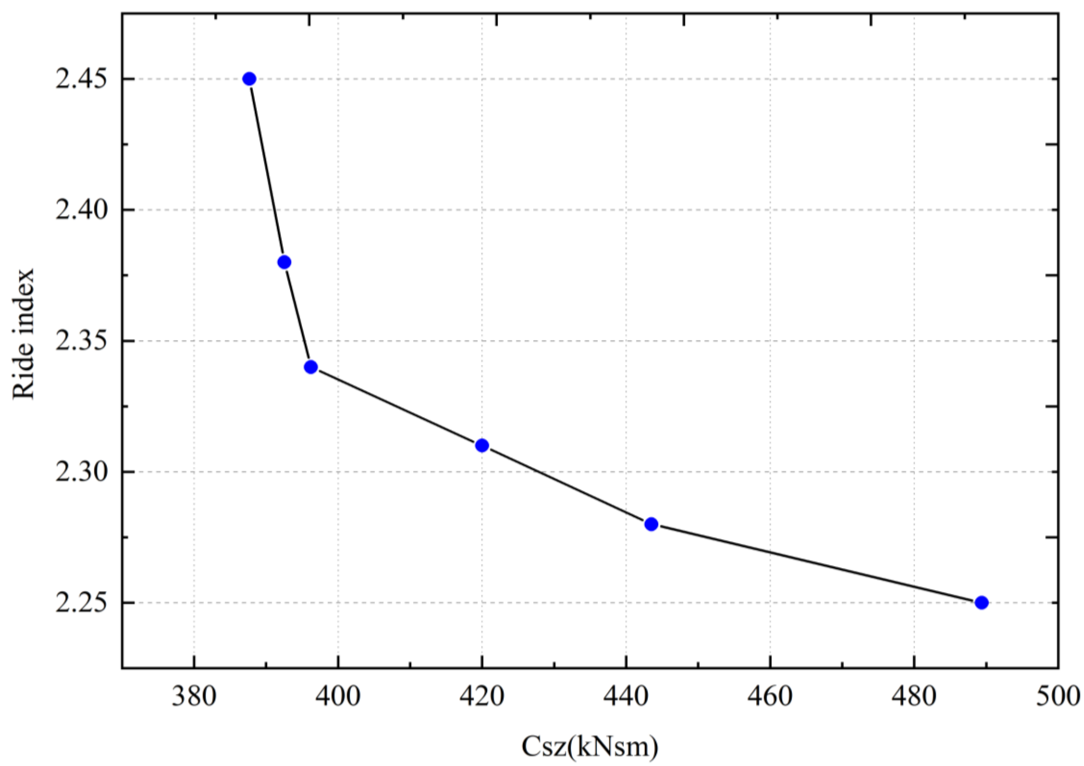


***Figure 4.7**: Effect of secondary suspension damping on ride index.*

It evident that secondary suspension damping greatly influences ride comfort index. With increase in the values of damping, the ride comfort index reduces from 2.45 to 2.25 i.e moving away from 'more pronounced but pleasant' to closer to 'clearly noticeable region' implying that there is a general improvement in minimizing vibrations due to track excitations. The implies that increasing the damping value can help to improve the ride comfort however this should be done carefully to avoid initiating contrary effects. This is because excessive damping results into increased rigidity of the system and intensifying the vibration behavior of the vehicle leading to poor ride comfort index.

### 4.4.3 Influence of primary suspension vertical stiffness on the sensitivity of the vehicle.

The effect of stiffness of primary suspenison on the vibration sensitivity of the vehicle was determined by varrying values from $532.235kNm$ to $655.017\ kNm$ and the corresponding ride index values obtained as shown in Figure 4.9 while maitaining all other parameters constant at design parameters in Table 3.6.

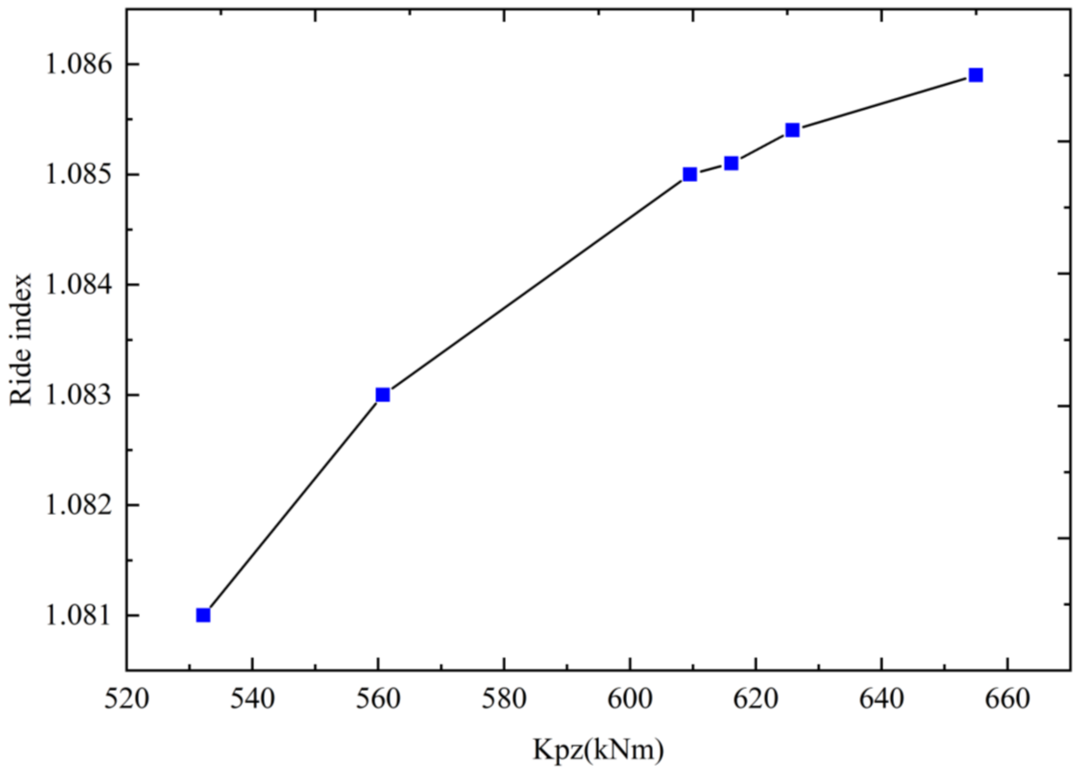


***Figure 4.8****: Effect of primary suspension stiffness on ride index.*

It is noted from Figure 4.9 that increase in stiffness values results into almost no change in the ride index values from $1.081$ to $1.0859$ and has an extremely small effect on the acceleration of the vehicle. This implies that changes in primary suspension stiffness have little effect on the vibration sensitivity of the vehicle and is maintained within the 'just noticeable', a region that is favorable for passenger ride comfort.

### 4.4.4 Influence of secondary suspension vertical stiffness on the sensitivity of the vehicle.

The effect of stiffness of primary suspenison on the vibration sensitivty of the vehicel was determined by varrying values from $126.356kNm$ to $204.943kNm$ and the corresponding ride index values obtained as shown in Figure 4.10 while maitaining all other parameters constant at design parameters in Table 3.6.

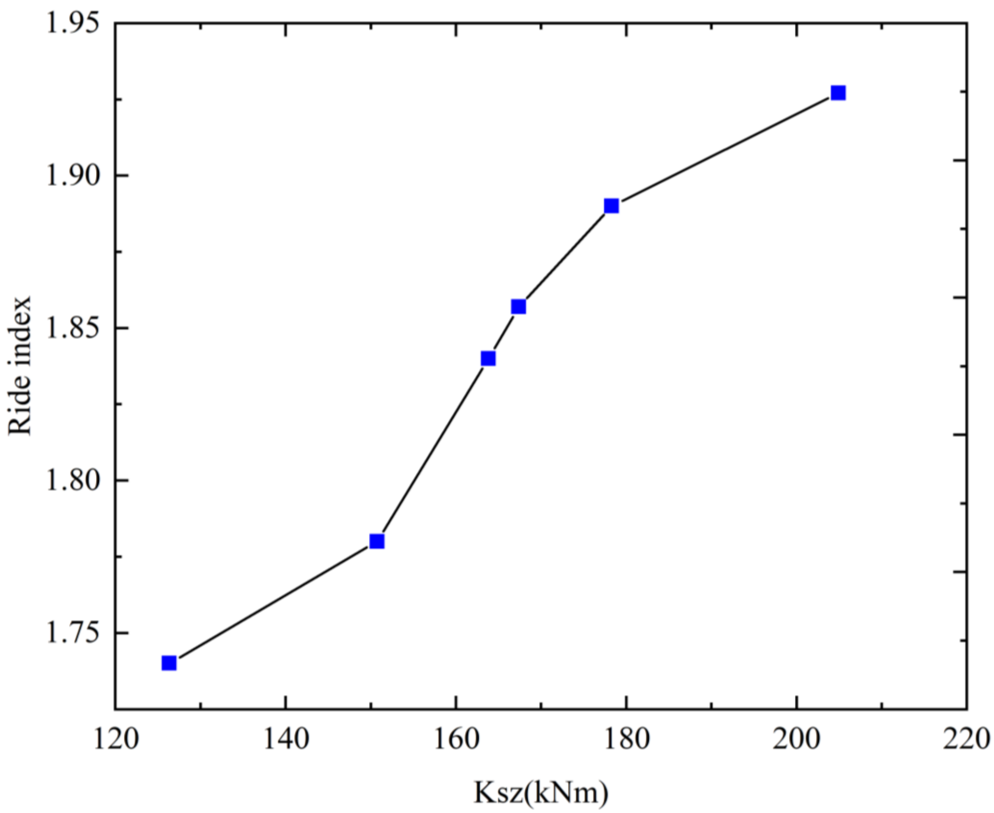


***Figure 4.9****: Effect of secondary suspension stiffness on ride index.*

It is evident in Figure 4.10 that increase in stiffness values lead to increase in the ride comfort index from 1.74 to 1.927 and hence there is a direct proportionality between the two and greatly influences the acceleration values of the vehicle. According to the results, the ride comfort index falls within the 'just noticeable region' thus exhibiting favorable passenger comfort however it is clear that further increase in the stiffness values can lead to deterioration of the ride index due to increase in the rigidity of the system.

### 4.5 Optimization of the suspension system

In order to achieve better and lower values of ride index, an optimization process was conducted on the suspension system. This process was carried out by simulating multiple groups of suspension parameter values in order to identify a combination with the best vertical and lateral ride index values. The maximum ride index achieved before optimization was 2.57 and 1.937 for vertical and lateral index respectively. Upon optimization, 2.51 and 1.83 ride index for vertical and lateral index were obtained. This shows an improved of approximately 2.3% and 3.7% for vertical and lateral ride index as shown in Table 4.3.

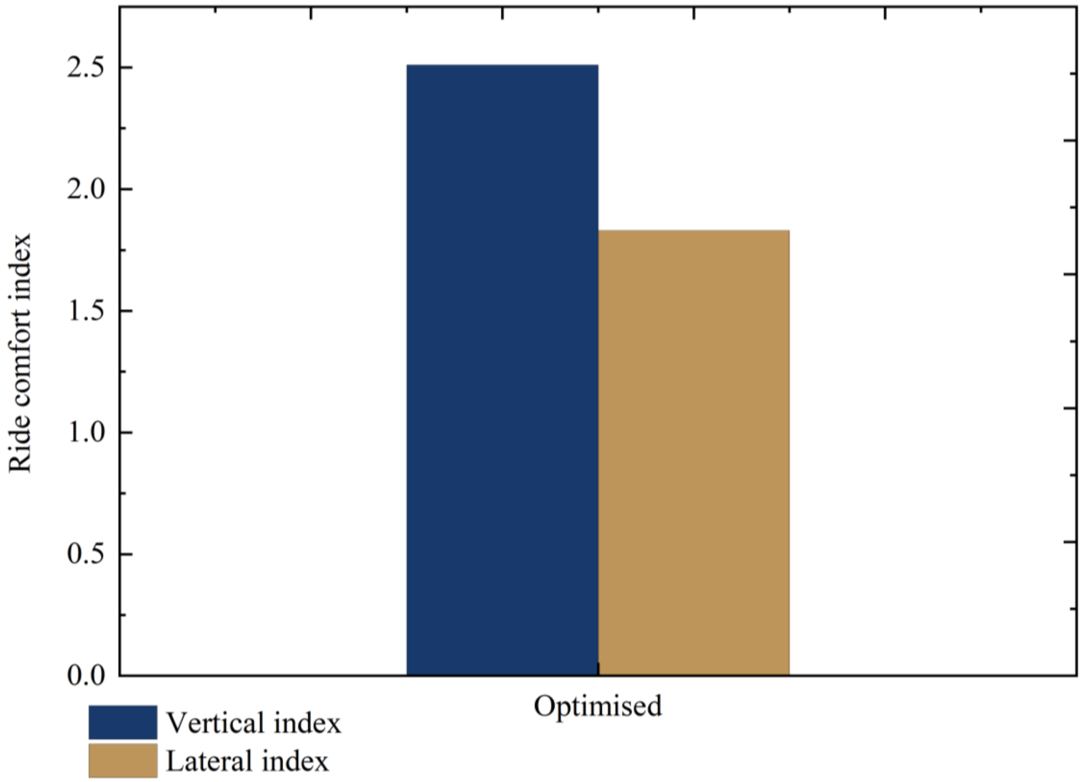


***Figure 4.10***: *Optimized ride index.*

Table 4.3 shows the parameter combinations that were used to conduct optimization and improve the ride index.

***Table 4.3***: *Suspension parameter values of maximum and optimized ride index.*

| **Symbols** | **Non optimized** | **Optimized** | **Percentage modification** |
|---|---|---|---|
| $k_{py}$ | 768340 $N/m$ | 758069 $N/m$ | 1.3 |
| $k_{pz}$ | 655017 $N/m$ | 600000 $N/m$ | 8.4 |
| $k_{px}$ | 655017 $N/m$ | 600000 $N/m$ | 8.4 |
| $c_{py}$ | 231041 $Ns/m$ | 208816 $Ns/m$ | 9.6 |

| | | | |
|---|---|---|---|
| $c_{pz}$ | 204943 $Ns/m$ | 178292 $Ns/m$ | 13 |
| $c_{px}$ | 145891 $Ns/m$ | 135991 $Ns/m$ | 6.8 |
| $k_{sy}$ | 408237 $N/m$ | 350000 $N/m$ | 14.2 |
| $k_{sz}$ | 186007 $N/m$ | 177978 $N/m$ | 4.3 |
| $k_{sx}$ | 382525 $N/m$ | 338798 $N/m$ | 11.4 |
| $c_{sy}$ | 231041 $Ns/m$ | 208816 | 9.6 |
| $c_{sz}$ | 489382 $Ns/m$ | 442885 $Ns/m$ | 9.5 |
| $c_{sx}$ | 231041 $Ns/m$ | 208816 $Ns/m$ | 9.6 |
| Wz, vertical | 2.57 | 2.51 | 2.3 |
| Wz, lateral | 1.9 | 1.83 | 3.7 |

### 4.6 Summary of discussion of results.

The assessment of the results achieved shows that the performance characteristics of a railway vehicle running along a track with excitation strongly depends on the suspension system for safer and reliable performances.

The simulations have been conducted using SIMPACK dynamic software. It is shown that primary and secondary suspension parameters play a vital in improving reliability and sensitivity of the vehicle. With a general increase in stiffness and damping values of the system, the vertical and lateral forces are observed to increase from $116.68kN$ to $181.86kN$ and $67.34kN$ to $84.36kN$ respectively as shown in Figure 4.3. The intensity of vertical force is strongly amplified by the presence of track irregularities resulting into increased accelerations thus influencing the ride comfort of the passengers and values of derailment coefficient. At higher values, the rigidity of the suspension system is increased leading to transmission of the impact loads directly to the passengers thus increasing the vibration sensitivity which can be harmful to passengers when exposed to them for a long period of time. It is also noted that the stiffness and damping values of secondary suspension system have a great influence on the ride comfort index of the vehicle as shown in Figures 4.8 and 4.10 whereas damping and stiffness values of primary suspension system have little effect on the ride comfort index of the vehicle as shown in Figures 4.7 and 4.9. According to the results in Figure 4.5, it is also shown that the safety reliability of the vehicle is more sensitive at low values of stiffness and damping in the region between 0.35 and 0.55.This is because the suspension system may not be rigid enough to withstand the dynamic impact loads leading to hunting motion with greater lateral forces that may result into damaging of the track, train and derailment in worst cases. However as the stiffness and damping values increase, a plateau form of the graph is achieved which indicates a more stable condition of running characteristics. The lateral forces obtained have been relatively small compared to the vertical forces because simulations were conducted on a straight track hence exhibiting minimum effects and their magnitudes were majorly influenced by the amplitude of track irregularities and consequently led to low ride index values between 1.832 and 2.0.This range of values is not harmful to passengers when exposed to them for a long time however the vertical ride index values increased with greater vertical forces from 2.52 to 2.57 which may not be very pleasant to passengers hence a need to perform an optimization on the system. This was performed by simulation of different combinations of the primary and secondary suspension stiffness and damping to identify a combination with better ride index. As shown in Table 4.3, the combination of those parameters was able to reduce the ride index from 2.57 to 2.51 which improves the passenger ride experience and comfort.

# Chapter Five

## 5.1 Conclusion.

Being able to estimate the behavior of the railway vehicles both at design stage and during service plays a vital role in ensuring that safe, reliable and comfort operations are achieved. The suspension system plays a crucial role in making sure that better passenger ride comfort is attained while the vehicle moves smoothly without any disturbances. Through conducting numerical simulations, the influence of track excitations and suspension system on the dynamic behavior of the vehicle has been evaluated using multibody system dynamics and the following conclusions have been.

- On a straight track, lateral forces do not have significant effect on the running characteristics of the vehicle and their magnitude is only depends on the amplitude of track excitations. The vertical forces experienced at the rail-wheel contact area strongly depend on the stiffness and damping values of the suspension system and track irregularities, since there was an increase from $116.68kN$ at sim 1 to $181.86kN$ at sim 6.Therefore a careful consideration should be taken into account when performing any modification to the system.
- The carbody accelerations increase with increase in the values of the suspension parameter and track irregularities and consequently lead to deterioration of the passenger ride comfort.
- Passenger ride comfort is strongly dependent on the suspension system. One of the major roles of the system is to absorb dynamic impact loads. It can be observed that the ride index increased from $2.52$ at a softer suspension to $2.57$ at a stiffer suspension. Furthermore, it is evident that damping and stiffness values of secondary suspension exhibit greater influence on the ride comfort index compared to damping and stiffness values of primary suspension which exhibit little influence on the ride comfort index. Therefore specifications of the system should be chosen to facilitate better ride experience by the passengers.
- Running safety reliability in terms of derailment (0.35-0.598) is observed to be more sensitive at low value of damping and stiffness which may result into accidents however stability is achieved when the values increase.

- Optimization process has been carried out and it is observed that the ride index was improved by 2.3% and 3.7% for vertical and lateral respectively.

## 5.2 Recommendations.

- Variation of suspension system parameters of the railway focused on analysing the key components necessary for smooth running of the vehicle like derailment coefficient, forces and ride index, even though the sensitivity and reliability results were found to be under limits required for safe and reliable running of the vehicle, track excitations were the only external factor considered, therefore there is a need to investigate their influence on wheel wear and consider other external factors that may have an effect on the running characteristics of the vehicle such as wind force and friction.

## 5.3 Future work.

- Investigate the influence of the suspension system on the wear rate of the wheels.
- Consider other external factors that may influence the dynamics characteristics of the vehicle like speed, friction, passenger weight and wind.
- Investigate the behavior of the railway vehicle in curved tracks with irregularities.

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# Appendices

## Appendix A

## Input data

***Table A.1**: Generated random numbers of the suspension system.*

| | $k_{pz}$ | $k_{py}$ | $k_{px}$ | $c_{py}$ | $c_{px}$ | $c_{pz}$ | $k_{sz}$ | $k_{sy}$ |
|---|---|---|---|---|---|---|---|---|
| | **600000** $(N/m)$ | **750000** $(N/m)$ | **600000** $(N/m)$ | **200000** $(Ns/m)$ | **120000** $(Ns/m)$ | **170000** $(Ns/m)$ | **170000** $(N/m)$ | **350000** $(N/m)$ |
| Simulations | | | | | | | | |
| S1 | 532235 | 736416 | 532235 | 176382 | 67002 | 126356 | 126356 | 286258 |
| S2 | 560769 | 757554 | 560769 | 190897 | 99352 | 150749 | 150749 | 291541 |
| S3 | 609563 | 758069 | 609563 | 208816 | 100760 | 163843 | 163843 | 304658 |
| S4 | 616130 | 765852 | 616130 | 214667 | 105429 | 167393 | 167393 | 354061 |
| S5 | 625865 | 765943 | 625865 | 221807 | 135991 | 178292 | 178292 | 407396 |
| S6 | 655017 | 768340 | 655017 | 231041 | 145890 | 204943 | 204943 | 408236 |

| | $k_{sx}$ | $c_{sz}$ | $c_{sy}$ | $c_{sx}$ |
|---|---|---|---|---|
| | **300000** $(N/m)$ | **400000** $(Ns/m)$ | **200000** $(Ns/m)$ | **200000** $(Ns/m)$ |
| Simulations | | | | |
| S1 | 198352 | 387702 | 176382 | 176382 |
| S2 | 241154 | 392551 | 190897 | 190897 |
| S3 | 314344 | 396217 | 208816 | 208816 |
| S4 | 324195 | 442885 | 214667 | 214667 |
| S5 | 338798 | 443524 | 221807 | 221807 |
| S6 | 382525 | 489382 | 231041 | 231041 |

***Table A.2**: Railway vehicle design specifications.*

| **Parameter** | **Value** | **Units** |
|---|---|---|
| **Bogie frame** | | |
| Motor bogie weight | <= 6000 | kg |
| Trailer bogie weight | 3200(<= 4000) | kg |
| Bogie height | 700 | mm |
| Bogie width | 2151 | mm |
| Bogie length | 3670 | mm |
| Distance between bogies | 10400 | mm |
| Bogie frame mass | 900 | kg |
| Moment of inertia, $I_{xx}, I_{yy}, I_{zz}$ | 1250, 1870, 2182 | Kg/m2 |
| Speed | 70 | Km/hr |

| | | |
|---|---|---|
| **Car body mass** | 44000 | kg |
| Length | 28800 | mm |
| Width | 2650 | mm |
| Height | 3610 | mm |
| Maximum load of the vehicle | 64000 | kg |
| Moment of inertia, $I_{xx}, I_{yy}, I_{zz}$ | $4375, 8750, 8750$ | Kg/m2 |
| **Wheelset** | | |
| Axle weight | 16000 | kg |
| Wheel weight | 985 | kg |
| Wheel diameter | 660 | mm |
| Axle diameter | 155 | mm |
| Distance between wheels | 1600 | mm |
| Moment of inertia, $I_{xx}, I_{yy}, I_{zz}$ | $176, 76, 176$ | Kg/m2 |

***Table A.3**: Sampling groups of the suspension system for optimization.*

| Symbols | Lower limit | 2 | 3 | 4 | 5 | Upper limit |
|---|---|---|---|---|---|---|
| $k_{py}$ | 736416 | 757554 | 758069 | 765852 | 765943 | 768340 |
| $k_{pz}$ | 532235 | 560769 | 600000 | 616130 | 625865 | 655017 |
| $k_{px}$ | 532235 | 560769 | 600000 | 616130 | 625865 | 655017 |
| $c_{py}$ | 176382 | 190897 | 208816 | 214667 | 221807 | 231041 |
| $c_{pz}$ | 126356 | 150749 | 163843 | 167393 | 178292 | 204943 |
| $c_{px}$ | 67003 | 99353 | 100760 | 105429 | 625865 | 145891 |
| $k_{sy}$ | 286259 | 291541 | 350000 | 354061 | 407396 | 408237 |
| $k_{sz}$ | 147946 | 165796 | 170000 | 177978 | 178140 | 186007 |
| $k_{sx}$ | 198352 | 241154 | 300000 | 324195 | 338798 | 382525 |
| $c_{sy}$ | 176382 | 190897 | 208816 | 214667 | 221807 | 231041 |

| | | | | | | |
|---|---|---|---|---|---|---|
| $c_{sz}$ | 387702 | 392551 | 396217 | 442885 | 443524 | 489382 |
| $c_{sx}$ | 176382 | 190897 | 208816 | 214667 | 221807 | 231041 |
| Wz, vertical | 2.52 | 2.53 | 2.51 | 2.547 | 2.5 | 2.57 |
| Wz, lateral | 2.0 | 1.86 | 1.83 | 1.89 | 1.85 | 1.9 |

**Appendix B**

**The vehicle weight and carrying capacity**

1. The maximum loading capacity of the vehicle ≤64ton
2. The total weight of the vehicle without any load or empty load: ≤44 ton
3. The average load of axle
   - Motor car≤10.5ton
   - Trailer car≤11.5ton

Main Parameters of the track lines

- Track gauge: 1435mm
- Minimum radius of horizontal curve; mainlines between sections 50m
- Minimum radius of vertical curve:1000m
- Maximum gradient; 5.5%
- Type of rails for mainlines and depot:50kg/m
  - Maximum super elevation: 120mm
- The vehicles adopt right-side running rules.
- Platform parameters

**Materials:**

- Frame: steel UNI EN 10025 S355J2G3C
- Central transom: Aluminum AIMg4.5Mn EN AW-5083 W28
- Axle box: EN-AC-AI-Si-7Mg0.6
- The suspension is made of stainless steel

**Power supply**

Range of voltage variation: - 500□900DC which is collected due to catenary

- ✓ Speed

  Maximum operation speed: 70Km/hr.

  Average travelling speed: 50km/hr. and dwelling time of train at station is 30sec

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Addis Ababa Light Rail Transit Service

**Parameter of the vehicle**

| S/N | parameters | value |
|---|---|---|
| 1 | Motor car | 11340mm |
| 2 | Trailer car | 3600mm |
| 3 | Maximum width of car body | 2650mm |
| 4 | Distance Between Wheels (Wheel Gauge) | 1600mm |
| 5 | Wheel diameter | 840mm |
| 6 | Track Gauge | 1435mm |
| 7 | Axle diameter | 155mm |
| 8 | Load of the Wheel | 985kg |
| 9 | Load or weight of axle | 16000kg |
| 10 | wheel base | 2750mm |
| 11 | Distance between wheels | 1600mm |
| 12 | Bogie height | 700mm |
| 13 | Bogie width | 2151mm |
| 14 | Bogie length | 3670mm |
| 15 | Bogie weight for trailer | 3200kg |

# Appendix C
## Multibody system dynamic information.

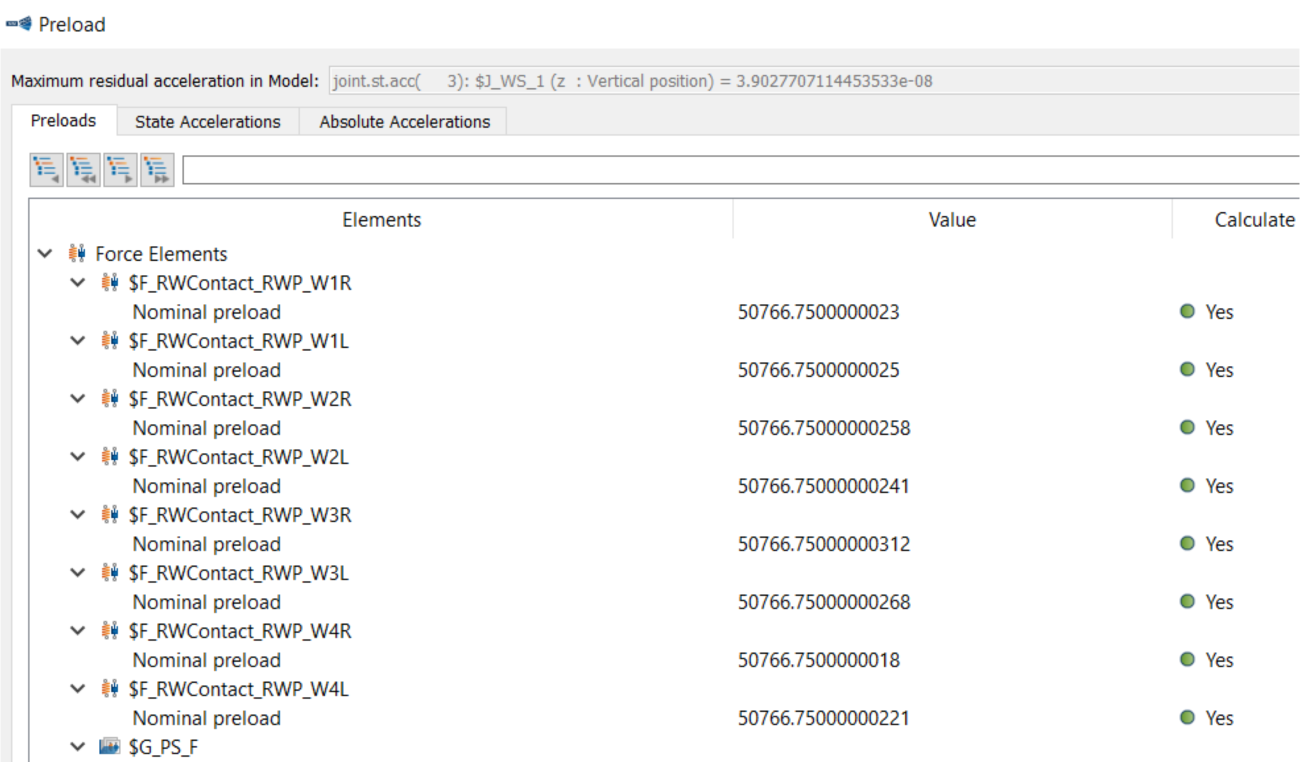


***Figure C.1**: Preload static force.*

**Peak vertical dynamic load without irregularities.**


***Figure C.2****: Peak vertical dynamic force without irregularity.*

**Track irregularities incoproration**

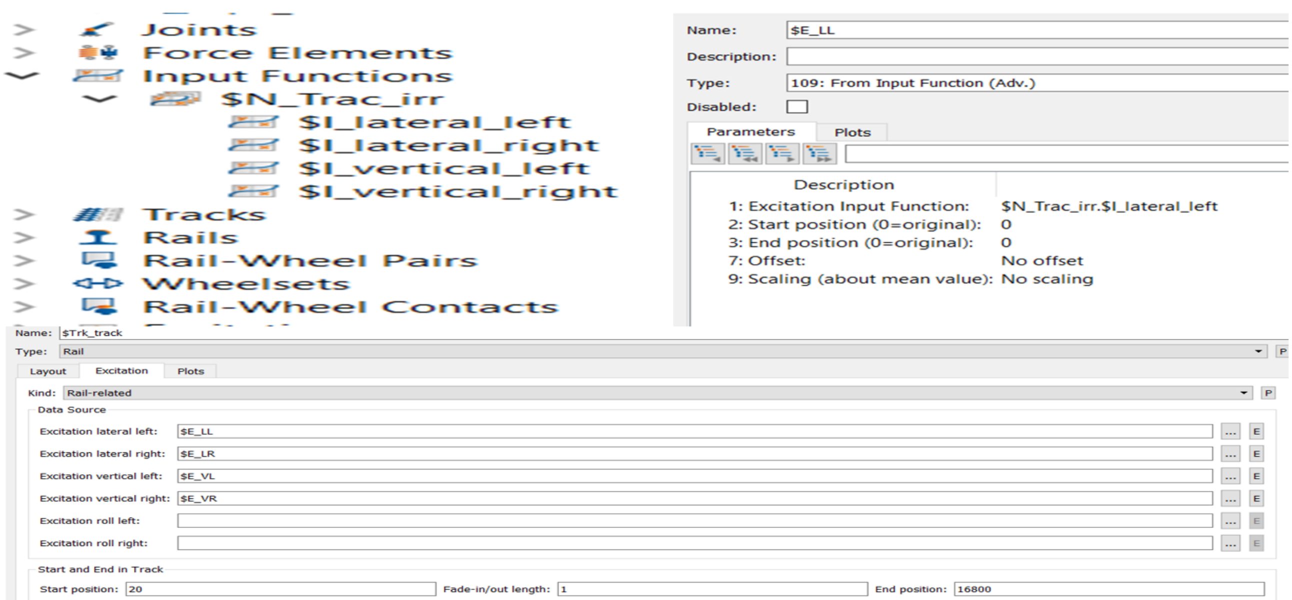


***Figure C.3****: Procedure for track irregularities into SIMPACK.*