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Addis Ababa Institute of Technology
School of Mechanical and Industrial Engineering



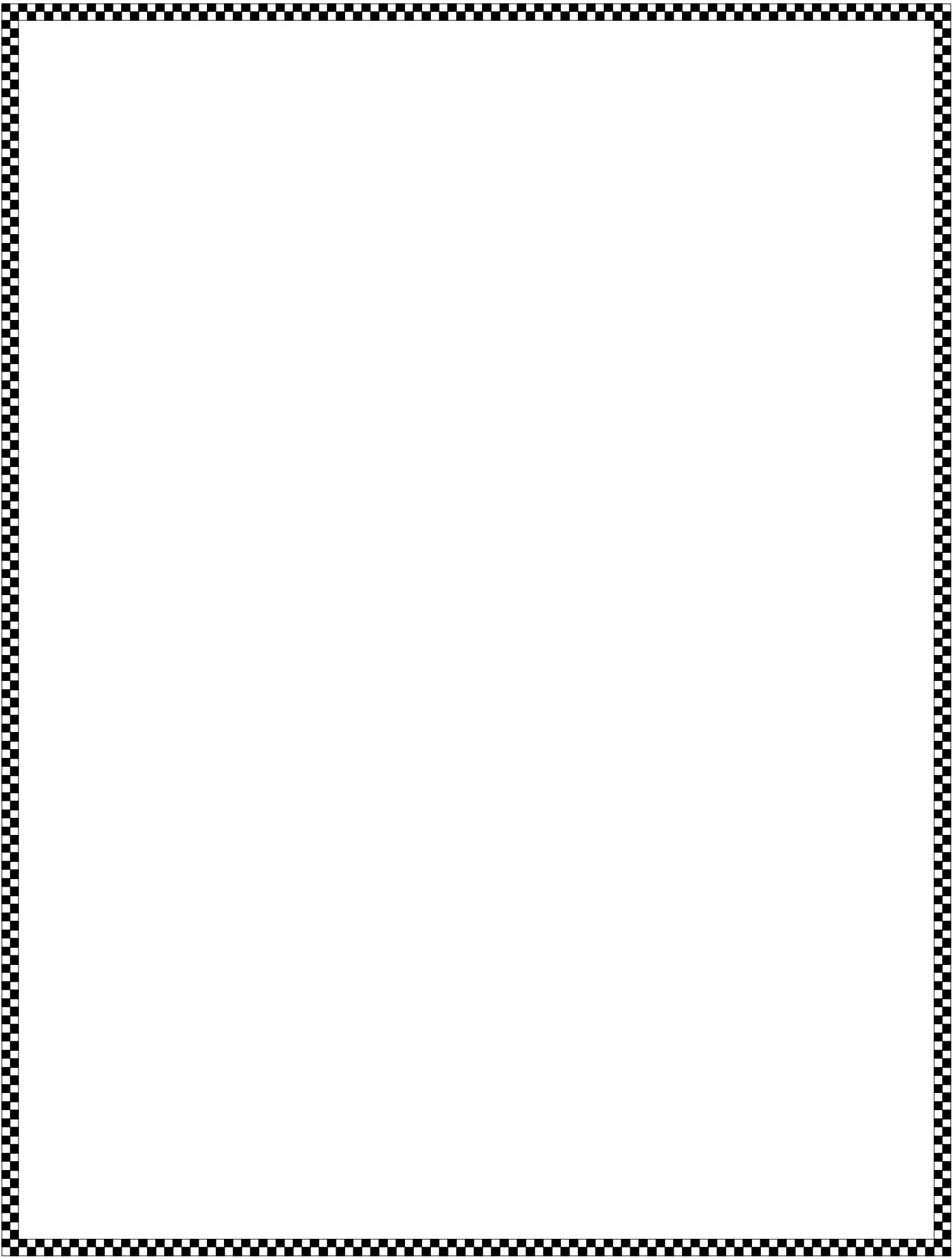
**Analyzing Dynamic Performance of Fright Rail
Wagon Using Multibody Simulation (SIMPACK)**

**A Thesis Submitted to the School of Graduate Studies of Addis Ababa
institute of Technology in Partial Fulfillment of the Requirement for the
Degree of Masters of Science in Mechanical Engineering ((Rolling stock
Engineering Stream))**

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June 2015
Addis Ababa, Ethiopia



Declaration

This is to certify that the thesis prepared by **Henok Chala** entitled: **Analyzing Dynamic Performance of Freight Rail Wagon Using Multi-Body Simulation (SIMPACK)** submitted in fulfillment of the requirements for the degree of masters of Science (Rolling stock engineering) compiles with the regulations of the university and meets the accepted standards with respect to originality and quality.

Approved by Board of Examiners:

Ato Habtamu Tikubet (M.Sc.)

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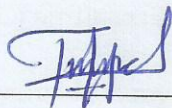
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Abstract

Complex nonlinear interaction phenomena between rail wagon and track significantly influence the dynamic performance. Thus to alleviate the drawback arise due to poor vehicle riding behavior, predicting parameter that quantify performance and certifying the wagon have inevitable significant. The objective of the study is to model and simulate dynamic performance of X4K container carrier freight wagon with 48 DOF on selected Ethio- Djibouti 1.8 km horizontal track i.e. continuous straight, transitional and circular track using Simpack 9.6 having considered critical operation maneuver i.e. 100Km/hr maximum wagon running speed and 93 ton total load. The obtained simulation result indicated; maximum derailment coefficient ($Y/Q=0.24098$), maximum lateral track shift force $\sum F_y=72,438.5\text{N}$, ride quality index $WZ(\text{vertical})=2.5769$ & $WZ(\text{lateral})=2.1786$ and maximum wear index $\Gamma_w=360.47\text{N}$ during curve negotiation have been found. Then conclusion drawn i.e. derailment coefficient obtained result and lateral track shift as it's compared UIC 518 validating standard, the propensity of wagon derailment validated safe range and the track shift force that induce lateral track irregularity, gauging validated in acceptable range. The ride quality index for both vertical and lateral direction that quantify the running performance of the vehicle validated as per CEN (2005) and the result found acceptable riding performance. While, the wear index resulted from simulation indicated beyond the allowable wear index limiting standard particularly during the wagon negotiate curved track and at intermediate & curved track junction. Consequently at operational speed maneuver (100Km/hr) the tendency of mild or severe wear occurrence between wheel/rail contacts validated high. To alleviate this problem optimization wear index have been conducted by consider operational speed reduction and a medium wear rate found in favor of reduced speed. Therefore the study recommended to use 70Km/hr during the wagon negotiates the curve and transition track.

Keywords:-Railway dynamic performance Safety, lateral track shift force, Derailment Coefficient ride quality index, wear index

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Notations

s_{XL}	Longitudinal creepage
V_x^w	Horizontal velocity of the wheel
V_x^r	Horizontal velocity of the rail
V	Mean velocity
ω	Wheel Angular rotation
b	Half distance b/n left and right wheel
S_{YL}	Lateral creepage on left wheel
S_{ZL}	Spin creepage on left wheel
S_{Zr}	Spin creepage on right wheel
V_y^w	Lateral wheel velocity
V_y^r	Lateral rail velocity
Ω_z^w	Wheel angular velocity
Ω_z^r	Rail angular velocity
γ_l	Angle of attack left wheel
γ_r	Angle of attack left wheel
F_{xr}	Longitudinal right wheel rail contact force
F_{xl}	Longitudinal left wheel rail contact force
F_{yl}	Lateral left wheel rail contact force
f_{22}, f_{23}, f_{11}	kalker coefficients
δ_{rr}	Right wheel rail contact angle

δ_{rl}	Left wheel rail contact angle
i	Number of wheelset
m_C	Mass of the frame the car
m_b	Mass of bogie
m_w	Mass of wheelset i
F_{yr}	Lateral reaction force right wheel rail contact
F_{yl}	Lateral reaction force left wheel rail contact
M_{xy}	Moment in x-y direction
M_{zx}	Moment in z-x direction
M_{zy}	Moment in z-y direction
M_z	External moment along z axis
F_{xl}	Wheel rail contact normal force at left wheel
F_{xr}	Wheel rail contact longitudinal force at right wheel
F_{fy}	Lateral frictional damper of adapter plus
N_l	Wheel rail contact normal force at left wheel
r_r	Right wheel radius
r_L	Left wheel radius
r_o	Mean radius wheel
R_O	Rail contact curvature
h_1	Vertical distance b/n CG of bogie primary suspension
k_y	Lateral stiffness constant adapter plus

I_{w1}	Wheelset moment of inertia
$\dot{z}_b$	First ode vertical bogie displacement (bogie vertical velocity)
z_b	Vertical displacement of bogie
$k_{y,cp}$	Lateral stiffness constant centre plate
$c_{y,cp}$	Lateral damping constant centre plate
Ψ_c	Yaw rotation of car
ϕ_o	Cant or tilt angle
WZ	(Wertungszahl), ride quality index
N_{MV}	Ride comfort index
ΣFy_{lim}	Limiting lateral track shift force
Γw	Wear index number

Nomenclatures

C.G	Centre of gravity
CEN	European Committee for Standardization
DOF	Degrees Of Freedom
DAE	Differential Algebraic Equations
EN	European Standard
ERC	Ethiopian Railway Corporation
LAE	linear algebraic equations
MBS	Multibody System
ODE	Ordinary differential Equations
RMS	Root Mean square
UIC	International Union Railway
MetEC	Metal and Engineering Corporation
Ptp	Point to point
Cmp	Compact

Chapter 1 Introduction

1.1 Background

The history of rail transport dates back nearly 500 years and includes systems with man or horse power and rail of wood or stone. In 1604, the first railway in Britain was built, but it was called a wagon way and it was made of wood. Modern rail transport systems first appeared in England in the 1820s. These systems, which made use of the steam locomotive, were the first practical forms of mechanized land transport, and they remained the primary form of mechanized land transport for the next 100 years. By 1900, the railway was mostly completed, and there were more than a hundred train companies in Great Britain. In 1904, an engine called 'The City of Truro' became the first to travel at more than 100 miles an hour. The electrification of the railways began in 1933. This means that the trains began to run on electricity instead of steam. Railways were originally intended to carry mostly goods rather than passengers but in the 1970s, the value of carrying passengers overtook goods for the first time [1].

The railway lines construction in Ethiopia was first started in October 1897 from Djibouti during the period of Emperor Menelek II. The first commercial service began in July 1901, from Djibouti to Dire Dawa. By 1915 the line reached Akaki, only 23 kilometers from the capital, and two years later came all the way to Addis Ababa [2]. Since the rail transportation neglected for decades it adversely affected the import and export of commodity.

To alleviate the existing problem the government of Ethiopia gave due attention in constructing railway infrastructure in different parts of the country. Among those the, 718 km railway infrastructure that link Ethiopian and port Djibouti [3] given higher priority in growth and transformation plan. Meanwhile, the transportation system to achieve the anticipated objective, it needs to have better performance and safety.

Considering the fact that, the long history of railway engineering challenged due to many practical examples of dynamical performance and safety related problem. Particularly inadequate guidance on track with poor vehicle system performance results in high dynamic forces between wheel and rail might lead to possibility of dynamic instability, high track shift force, excessive wear at wheel rail contact and derailment [4]. Therefore it is essential to predict the running behavior of the railway vehicles in order to obtain performance related parameter.

Furthermore, the large number of nonlinear components in a railway vehicle moving along a track creates a very complex mechanical system. In particular, the complex nonlinear interaction between wheel – rail, suspension and frictional element have a significant effect on vehicle dynamic performance and safety.

With the advent of personal computers and faster processors, the use of analytical modelling programs has become less complicated, and far more practical. This is true in the area of rail vehicle modelling as well. The low cost of computer modelling and simulation compared with real-world testing have unlimited significance. Such technique allows as testing a new vehicle design without having to build a prototype and tie up a track for testing, thereby increasing productivity through saving valuable time and manpower [3]. This cost and time savings advantage of railway vehicle simulation further magnifies the importance of its versatile applications.

One such use is “what if” analysis. Computer modelling allows the user to test out various situations without spending the time, money, and use of equipment to test them on a track. Further, modelling can provide the means for performance testing and validation to enable prediction of when a given car might derail or overturn. Modelling can predict at what speeds derailment will occur, or under what conditions it may be prevented. Directly related to derailment studies is stability analysis; one can model multiple suspensions and loading options and examine dynamic responses. [6, 29]

Another important aspect of railway dynamics is ride comfort analysis, or predicting what travellers and cargo may experience under various conditions. Modelling software can predict forces and accelerations at various positions throughout the vehicle to model ride characteristics, or to evaluate ideas for improving ride quality.

There modelling programs that have received wide acceptance in recent years include:

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

➤ GENSIS

Although the above programs have differing attributes, they were all developed specifically for rail dynamic modelling. Each program includes different solution methodology, wheel/rail models, analysis methods, and user interface [3].

In the present research, one of the most well-known software accepted by industrial and academic communities, SIMPACK 9.6 build 93 multi-body simulation software, developed at the German aerospace research center (DLR) together with INTEC GmbH, is used to perform assessment of the dynamic behaviour of the Ethiopian national freight rail wagon on a track that is recently going to be implemented for operation [20].

To analyze the dynamic behaviour of rail wagon running on newly constructed tracks; the complete vehicle–track system modelling can be split up into subsystems: vehicle, wheel–rail contact, suspension force and track. Finally by taking designed and actual working condition in to consideration the simulation conducted.

1.2 Statement of the problem

The complex dynamic forces developed in the wheel-rail interface, the characteristics of the suspensions, the mass of system elements and the geometry of the track play an important role on; track shift force, derailment safety risk, ride quality and ride index. As a result different questions need to be answer such as:

- How do we assess rail wagon running behavior whether it is good or poor?
- What is the running behavior of the rail vehicle regarding derailment safety dynamic stability, track shift force and riding quality on continuous curved, tangent curve and intermediate track? How do we evaluate the obtained virtual simulation result as per recognize standard?
- How do we assess the wear rate and correlate or validate with standard?
- What parameters influence the dynamic performance rail vehicle and how do the rail wagon & the track interacts?

Such and other kinds of questions divert the interest of most researchers to focus their attention on Predicting dynamic performance rail wagon using multi-body simulation software package.

1.3 Objective of the study

General objective

The main objective of the study is to develop container carrier rail wagon model X70 as per Ethiopian rail way corporation procurement data and analyze its dynamic performance along 1.8 km continuous track using Multi-body Simulation software package SIMPACK, and also validate or certify its performance in accordance with CEN (2005)/UIC 518 standard for ultimate operational maneuver i.e. maximum loading condition 93 ton and 100km/hr speed.

Specific objective

- ❖ Determine lateral to vertical force ratio or derailment coefficient (Y/Q) at wheel-rail contact that indicates derailment risk, and validates the result in accordance with UIC518/CEN (2005) allowable standard.
- ❖ Determine and validate of sum lateral track shift force in accordance to UIC 518/CEN (2005) standard.
- ❖ Analyze and validate of Ride quality index (WZ) in accordance with CEN (1999) standard.
- ❖ Analyze wear index that quantify the dissipation of energy at wheel- rail contact and validate the result as per recognize standard limit.

1.4 Significance of the Research

In general for assessing derailment risk, track shift force, wear rate and ride quality index usually numerous experiment need to be done, including complex measurement by sophisticated instrument in fully equipped laboratory. Since from our country context those facility are not available currently, the study have inevitable significant on providing dynamic performance prediction.

Besides local manufacturer like MetEC that involved in manufacturing the rail wagon body frame and assembly by incorporate the study output to their design phase or manufacturing stage they might re-engineer in order to improve dynamic performance. In addition this study will benefit Ethiopian Railway Corporation via minimizing both money and time in the costly process of new vehicles certification. Furthermore, this study has a considerable significant on operating conditions and environmental factors that difficult to integrate in physical tests can be applied on virtual simulation study.

This research study have specific significant on:

- Predicting safety using the simulation parameter that quantifies the risk of derailment(Y/Q), and validate the result by comparing with derailment limiting standard coefficient.
- Determination wheelset lateral shift forces on the track and validate the result by considering allowable standard limit that have a significance on predicting the lateral irregularity of the track caused due to extreme track shift force that leads to high maintenance cost.
- Validating the overall ride quality by determining ride quality index and comparing with admissible limit.
- Validating the newly implemented rail vehicle performance with regard to dynamic lateral stability based on SIMPACK simulation result.
- The research finding can be input for further study i.e. optimization rail wagon major rigid body component and suspension element.

1.5 Scope and organization

Under this study the following general element considered

- Modeling of the rail wagon in accordance to ERC procurement data i.e. two separate bogie and wagon platform having 48 DOF
- Modeling of wheel rail interaction
- Dynamic performance simulation of the rail wagon at 100km/hr. running speed and maximum loading condition using SIMPACK and validation according to CEN/UIC standard along continuous; tangent, intermediate and circular track.

This thesis is organized in six chapters. In **Chapter one**, background, general methodology, significance of this thesis work and the objectives to be achieved are discussed. **Chapter two** review a literature relevant to this thesis work, which has been investigated by different researchers Literature survey. **Chapter three** describes the main structure of the MBS tool was presented to model rail wagon by SIMPACK, including Pre-Processing:-Track definition, Body Definition, Joints Definition Force Elements Processing; and formulating analytical relations and equation of motion which deal with the interaction between vehicle and track. **Chapter four** describe dynamic performance literature theory and CEN/UIC standard limit regarding dynamic stability, permissible Track shift forces, Nadal's risk of derailment Criteria, Permissible maximum vertical loads and ride quality presented. **Chapter five** illustrates the simulation result and overall discussion and validation. Finally **Chapter six** gives conclusion and recommendation achieved from this thesis work and propose future work in this field of study.

1.6 Limitation

- The rail vehicle model used in the simulation study comprises a rigid car body, i.e. Car body flexibility was not considered in the current study but it does influence rail vehicle dynamics and it is another engineering problem which needs to be solved.
- Track irregularity and Track flexibility is not considered due to unavailability actual track measurement data, therefore the simulation is undertaken based on ideal track condition but they do influence rail vehicle dynamics; therefore, the effect of such track parameters

on rail vehicle dynamic behavior is another engineering problem which needs to be studied.

- External induced force, i.e. Effect of wind, aerodynamics and vertical elevation or gradient resistance force and braking are not taken in to account in the current study but they do influence rail vehicle dynamics so incorporating traction and braking in rail vehicle dynamics analysis results in better simulation results

1.7 Methodologies

To fulfill the objectives of the study the following are used.

- **Data Collection:** Data regarding the X4K rail wagon model are collected from ERC and rail wagon design standard.
- **CAD modeling:** Major rigid wagon body build based on acquired ERC procurement data and rail vehicle design standard.
 - CATIA V5R16 software is used to build the computer model of the major rigid body.
- **SIMPACK Multi-Body Modelling**
 - SIMPACK 9.6 is used to model wheel-rail three dimensional rigid bodies and wheel rail interaction in pre-preprocessor interface using acquired wheel /rail procurement and standard data.
 - Modelling rail Wagon multi-body by importing rigid body modelled in catia to SIMPACK interface and assembled with model wheel-rail by allocating proper joint and marker or coordinate space.
 - Modelling suspension force element and constrained relative motion.
- **SIMPACK Multi-Body Simulation:** corresponding dynamic simulation analysis performed by considering actual ERC operational maneuver using SIMPACK 9.6 software.
- **Validation:** The dynamic performance result obtained validated using CEN (1999) and CEN (2005)/UIC518 objective standard.

Chapter 2 Literature survey

2.1 Introduction

Engineering have such a various type of niche in the term of its dynamical system and railway vehicle dynamic is one of the most complex systems within it. There is much condition on it that must be count in and put into consideration, such as the contact between the wheel and rail that generate different forces in different kind of speed and the interaction between the wheel and rail that involve complex geometry of both side as we can see flange shape on its wheel.

2.2 Theory of Multi-body system dynamics

The first stage in setting up a computer model is to prepare a set of mathematical equations that represent the vehicle – track system. These are called the equations of motion and are usually second order differential equations that can be combined into a set of matrices. The equations of motion can be prepared automatically by the computer package; a user interface collects vehicle parameters, described in graphical form or by entering sets of coordinates along with other data describing all the important aspects of body, and suspension components.

The vehicle is represented by a network of bodies connected to each other by flexible, massless elements. This is called a multi-body system, and the complexity of the system can be varied to suit the simulation and the type of results required. Each of the rigid bodies can be considered to have a maximum of six degrees of freedom, three translational and three rotational. Physical constraints may mean that not all of these movements are possible, and the system can be simplified accordingly. Application of the constraint equations results in a set of equations of motion which are ordinary differential equations (ODE) or linear algebraic equations (LAE) and ODEs, depending on how the constraint equations are used [29,8,].

There are various coordinates and formalisms that lead to suitable descriptions of multi-body systems. In this work, the methods presented in SIMPACK are based on the use of Cartesian coordinates, which lead to a set of differential-algebraic equations that need to be solved. It is assumed that appropriate numerical procedures are used to integrate the type of equations of motion obtained with the use of Cartesian coordinates. It is also assumed that the various numerical issues that arise from the use of this type of coordinates, such as the existence of redundant constraints and the possibility of achieving singular positions, are also solved. A

typical multi-body model is defined as a collection of rigid or flexible bodies that have their relative motion constrained by kinematic joints that are acted upon by external forces. Let the multi-body system be made of N number bodies. The equations of motion for the system can be described as [29].

$$M\ddot{q} = g \dots \dots \dots 2.1$$

Where M is the mass matrix, which includes the masses and inertia of the individual bodies, q is the vector of generalized coordinates, and correspondingly $\ddot{q}$ is the acceleration vector, and g is the vector with applied forces and gyroscopic terms. The relative motions between the bodies of the system are constrained by kinematic joints, which are mathematically described by a set of n_c algebraic equations, written as

$$\Phi(q, t) = 0 \dots \dots \dots 2.2$$

The first and second time derivatives of equation (2.2) constitute velocity and acceleration constraint equations, respectively, written as n constrained bodies according to

$$\dot{\Phi}(q, t) = D\dot{q} = v$$

$$\ddot{\Phi}(q, \dot{q}, t) = D\ddot{q} = \gamma \dots \dots \dots 2.3$$

Where D is the Jacobian matrix for a system of constrained bodies, the effect of the kinematic joints can be included in equation (2.1) by adding to its right-hand side the equivalent joint reaction forces

$$g(c) = -D^T\lambda, \text{ leading to}$$

$$M\ddot{q} = g - D^T\lambda \dots \dots \dots 2.4$$

Where λ is a vector with n_c unknown Lagrange multipliers. Equation (2.4) has $n_b + n_c$ unknowns that must be solved together with the second time derivative of the constraint equations. The resulting system of differential-algebraic equations is:

$$\begin{bmatrix} M & D^T \\ D & 0 \end{bmatrix} \begin{bmatrix} \ddot{q} \\ \lambda \end{bmatrix} = \begin{bmatrix} g \\ \gamma \end{bmatrix} \dots \dots \dots 2.5$$

Note that the solution of equation (2.5) presents numerical difficulties resulting from the need to ensure that the kinematic constraints are not violated during the integration process. After grouping the force matrixes according to coordinate vectors, the equations of motion for the vehicle will have the following form

$$M\ddot{q}(t) + (H + \Omega_0 G)\dot{q}(t) + (k + \Omega_0^2 Z)q(t) = h(t) \dots \dots \dots 2.6$$

Where the vector $q(t)$ contains the generalized coordinates, matrix M is the mass matrix, matrix H is the damping matrix resulting from viscous coupling elements between the wheelset, the bogie frame, and the car body, and matrix K is the stiffness matrix resulting from the elastic coupling elements and the modal stiffness of the wheelsets. The matrices $\Omega_0 G$ and $\Omega_0^2 Z$ describe gyroscopic and centrifugal forces, respectively. The vector $h(t)$ represents generalized external forces resulting from wheel–rail contact, from gravitation, and from nonlinear yaw damping. Because of the symmetric structure of the vehicle, the equations of motion can be split up into two separate systems for symmetric and asymmetric motions.

Dynamic analysis of a multi-body system requires that the initial conditions of the system, i.e. the position vector q^0 and the velocity vector $\dot{q}^0$ are given. With this information, equation (2.6) is assembled and solved for the unknown accelerations, which are in turn integrated in time together with the velocities. The process, schematically shown in Figure 2.1, proceeds until a system response is obtained for the required period. Thus, the problem is reduced to solving a set of differential-algebraic equations (DAE).

Furthermore, all time-domain analyses are performed with the second order Runge-Kutta integration method. Once equations of motion can be compiled and integrated, they can provide us with the information about variation in response for different vehicle designs.

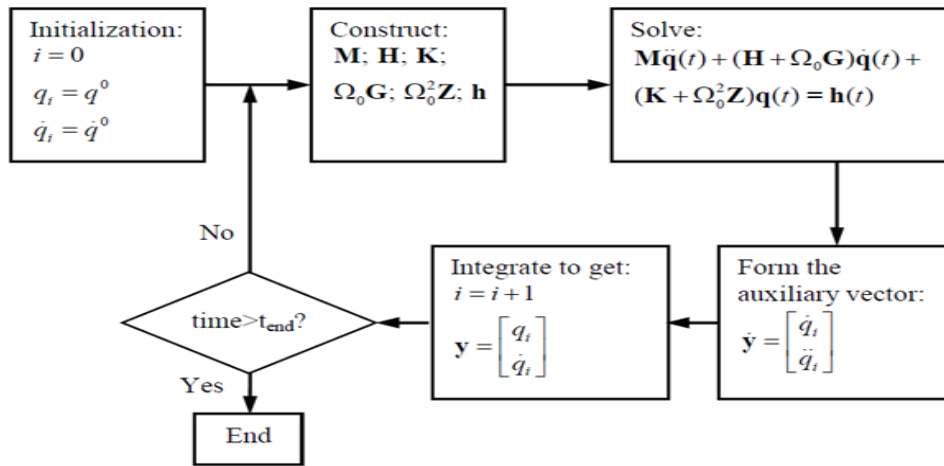


Fig 2.1 Flow chart MBS simulation algorism [29]

2.3 Modeling and simulation using SIMPACK

The SIMPACK software package is a multi-body system mechanical design tool which assists engineers to model, simulate, analyze and design complex mechanical systems; such as vehicles, robots, machines and mechanisms. The software also allows the inclusion of electrical, hydraulic and pneumatic elements. It is also able to analyze vibrational behavior, calculate forces and derailment, wear index, and ride quality as well as describe and predict the motion of multi-body systems. The basic concept of software is to create the equations of motion for mechanical and mechatronic systems and then from these equations various different mathematical procedures produce a solution (e.g. time integration). The model is built up using the modeling element and system equations automatically generate from this model. The equations of motion can be generated both symbolically and numerically (where the numeric form is the usual form). The software has a comprehensive range of modeling and calculation features together with a user interface well adapted to an engineer's needs. From the initial concept stage to the point where the results has present, there are six steps. The first three steps are performed outside SIMPACK interface. They are as follows:

I. Development of a mechanical model

- Mechanical structure is divided into bodies and joints, the interconnecting structures
- Constraints are then defined, which constrains the mobility of the elements by removing degrees of freedom
- Forces in between the ground and the bodies are defined

II. Provision of the physical parameters for the model

- The physical parameters for the model such as the mass, moments of inertia and center of mass for various different bodies are defined
- The geometry of the structure and how it fits together are defined; i.e. the distances in between coupling points
- The parameters for the coupling elements are defined, such as the force element values and constraints

III. Pre-processing

Input the data set, obtained from steps 1-3, with the help of the SIMPACK user interface:

- The physical model i.e. bodies and joints

- All the input functions for the model including the constraints, forces and excitation functions
- The associated 3D geometrical data for the graphical representation of the bodies
- The numerical calculation settings

IV. Processing

SIMPACK Calculations: Generation and solution of the motion governing differential equations. The differential equations are generated from the data entered in the previous steps and then solved within SIMPACK. The analyses can be based on time or frequency domain can be carried out.

5. Post-processing: Presentation of the results

User determined plots such as load indices or limiting values

- 2D line plots and Filtering
- 3D animation of the model
- Export to Microsoft Excel and MATLAB

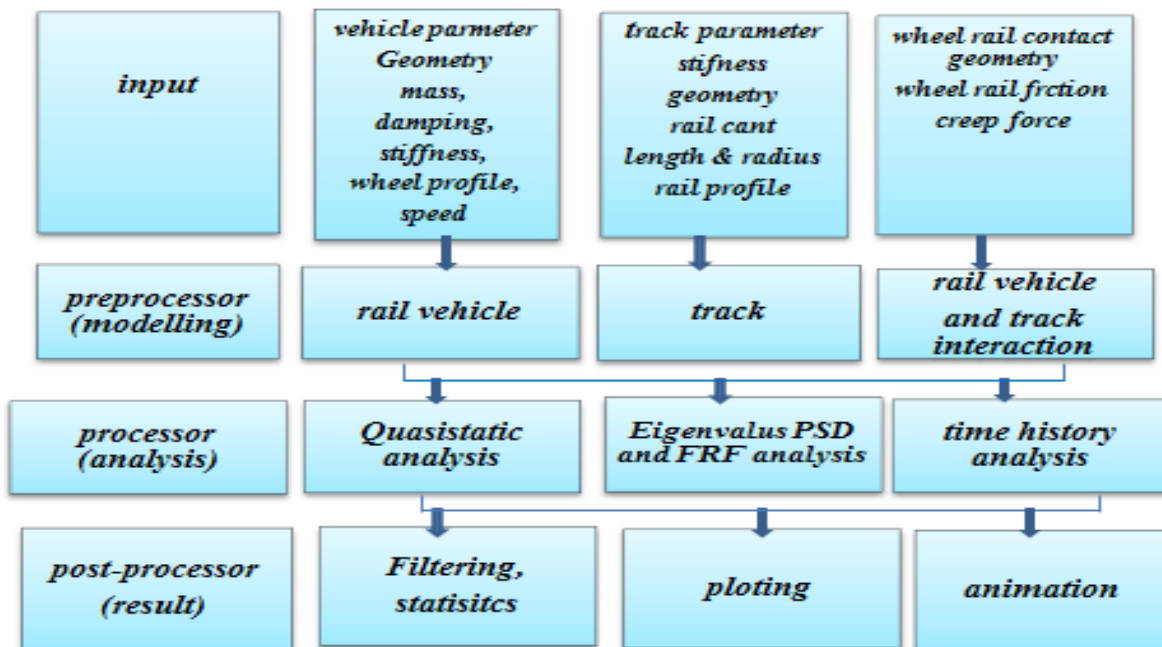


Figure 2.2 Rail vehicle modelling and simulation procedure schematics in SIMPACK

2.4 Review of researches related to the current study

The dynamic behavior of railway vehicles relates to the motion or vibration of all the parts of the vehicle and is influenced by the vehicle design, particularly the suspension and the track on which the vehicle runs. Due to this issue, several models of simulation schemes were developed in which all the factors affecting the dynamics of a railway vehicle were studied, such as the model developed by Iwinicki and Wickens (1998) [13]

Anneli Orvnäs(2007)[24].The purpose with their simulation study is to investigate, what measures to be taken in order to modify a bogie that guarantees a stable running behaviour at higher speeds, and generates acceptably low track forces and amount of wear in curves with smaller curve radii. Simulations are performed with a one-car Regina train modelled in the simulation tool SIMPACK. The simulations have been performed with track geometry cases with varying curve radius, cant and speed (i.e. varying cant deficiency and hence track plane acceleration). The assessment of track forces and ride comfort has been performed on straight track and incurves with radii between 300 and 4900 meters. In order to maintain a stable running behaviour at higher speed the wheelset guidance stiffness of the bogie, as well as the yaw damping, had to be increased, compared with the original design intended for 200 km/h. The purpose of the present study has been to investigate the difference between two different types of bogie configurations with soft and medium wheelset guidance. For comparison, original soft (original Regina forv200 km/h) and stiff wheelset guidance have been included in the evaluation. The results indicate that the difference between the two bogie types (“soft” and “medium”) is not that evident on straight track and in large-radius curves. However, in small-radius curves the soft bogie configuration generates lower lateral track forces and hence lower energy dissipation (related to wheel and rail wear) than the medium bogie configuration

Oldrich Polach, Andreas Böttcher [9].The paper presents investigations of the validation process, the criteria and the limits for the validation of multi-body system vehicle models in regard to simulations of on-track acceptance tests carried out in the Dyno-TRAIN project. These investigations represent unique work in regard to both simulations as well as measurements. The analyses are carried out using measurements with a test train with 4 types of vehicles and 10 force measuring wheelsets, running over 20 days through 4 European countries and being equipped with simultaneous recording of track irregularities and rail profiles. The simulations,

comparisons with measurement and evaluations were conducted using vehicle models built in two different simulation tools by several partners. The proposed criteria and validation limits are based on 12 quantities covering the quasi-static and dynamic wheel/rail force measurements and vertical as well as lateral vehicle body accelerations. For each quantity a set of at least 24 comparisons between simulations – measurement is evaluated using values based on EN 14363 from at least 12 sections, which represent all 4 zones from straight to curves with very small radius. It is intended to introduce these criteria in to the revision of EN 14363 and to gain experience with this method in future projects. The validation metric is analyzed, too, but does not provide better and more reliable assessment than subjective assessments. This can be explained by the identified drawbacks of validation metrics. Future investigations could remove these drawbacks by modification of validation metrics in regard to railway vehicle dynamic behavior.

Adrian Herrero (2013) [7] Studied a 50 DOFs high speed rail vehicle model through Multi-Body Simulation software SIMPACK to Improving the dynamics behavior suspension in terms of ride comfort ride, safety and wheel-rail wear objective functions. The dynamic analysis was conducted for five different curvature and speed respectively including with measured data as the track irregularities his finding implies track shift found from his result indicated relatively reduced from low curvature radius(zone 4) to high curvature radius with high speed (zone 1) and also at the tangent curve or straight track it was found the track shift force significantly lower and also similar result has been obtained for derailment coefficient, the wear index value relatively on straight track shows some increment as compared with the larger curvature radius with larger speed this might explain that at higher speed even if it run at the tangent curve the amount of energy dissipation due to poor lateral and vertical stability became high. Eventually study made further optimization made MATLAB/SIMULINK

D. Ramy Elsayed. (2013)[8] Investigated the dynamical performance analysis for TGV 001 locomotive vehicle using tool (VIA) and the results obtained are compared with SIMPACK simulation Package for the same vehicle model, at the same operating condition. The developed simulation tool VIA is applied to different case tests in order to validate the obtained results, as well as the determination of the suitability of the proposed methodology to achieve different type of analysis on railway systems. For this issue, the VIA simulation tool is used to analyze the

dynamic behaviour of the Manchester Benchmark vehicle number one, negotiating track case number one. The obtained results show a good agreement with the results obtained by the commercial packages used in the analysis of the Benchmark. A comparison has been made between the obtained dynamic results produce from the simulation tool VIA for TGV001 locomotive vehicle in different operating scenarios, and the results obtained during my exchange period realized in the Politecnico di Milano, obtained by SIMPACK program. The results demonstrate good agreement between both simulation tools at the same operating conditions.

Vivek Kumar (2009)[18], investigated vertical dynamic behavior of a typical Indian railway and Dynamic analysis for Indian passenger vehicle at different speed of 15m/s, 30m/s, 45m/s and 60m/s performed using SYMBOLS SHAKTI software, acceleration response of the car-body obtained Plots show that initially, the value of acceleration is nearly equal to -9.8m/s^2 , which is mainly the acceleration due to gravity. Finally it goes to zero, when the vibration of the car body ceases and it stabilizes. The acceleration is generally within acceptable range and does not show any instability. Ride comfort analysis has been performed for speeds ranging from 15m/s to 60m/s. Comfort index has been calculated through based on Sperling's ride comfort index and the result found meets standard limit. As a starting point to understand the behavior of a rail vehicle and the corresponding effects on the ride comfort, safety and wear it is necessary to investigate different linear and nonlinear system dynamic responses. The critical hunting speed as the origin of the instabilities of a rail vehicle as well as the effects of the wheel conicity, the wheel-rail contact and the track imperfections have been studied thoroughly in [Fan and Wu (2006)]. To overcome the negative effects of such parameters on the above mentioned objectives functions, several suspension systems and control strategies have been proposed.

M. Hoffmann. (2006)[11] investigated fundamental dynamic behavior of European two-axle railway freight wagons with special attention to their unwanted hunting motion on straight track and curvilinear track In his thesis A model of a single two-axle freight wagon running on a general curvilinear track is developed and described in detail. Essentially, the model is a system of ordinary differential equations. The non-smooth characteristics of the interacting forces are challenging both in the modelling phases as well as in the formulation of an appropriate numerical integration strategy to extract the solution from the system of differential equations. The model is appropriately divided into states, such that the no smoothness in the model is

defined by the switching boundaries between the states. These switching boundaries are located by the numerical integration procedure allowing one to find the solution efficiently and accurately. The model is analyzed for different parameters (wheel-base, suspension parameters, rail inclination etc.). The solution space is shown to have several attractors emphasizing the intricate dynamic properties of these wagons. The study summarize, if the wagon is running at high speed the wheelsets are attracted to a flange-to-flange motion. This violent motion is not transferred to the car-body if the lateral excitation frequency of the wheelsets is far from the yaw eigen frequency of the car-body. However, the two-axle freight wagon can experience resonance motion at low, medium or high speed when the previously mentioned frequencies coincide. Here, the wheelsets are not necessarily moving from flange to flange but the car-body has a severe lateral and yaw motion.

Escalona. (2008)[14] studied that the calculation of wheel-rail contact forces in the dynamic simulation of railroad vehicles involves the following steps: Location of the position of the contact points on the surfaces of the wheel and rail, Calculation of the normal contact forces And Calculation of the tangential (creep) forces and moments consideration which are the input for further dynamic simulation of the vehicle-track

Tadeusz Niezgoda.[28] The railway wagon with a low flat rotatable loading floor was analyzed. The model of a railway wagon consisted of standard carriages, undercarriages, and a rotatable loading floor was developed. The model was built of rigid solids. Between individual elements of the model, the appropriate joints and contact connections were created. The model was analyzed using MSC Adams code, which allows performance of the 3D kinematic – dynamic analyses of the entire model. The analyses were carried out with use of the model of the loaded semitrailer. Passage of the railway wagon with a trailer on horizontal rails, which are the smallest standard arc with a radius 250m, was simulated at different speed. Reactions occurring in couplings of the rotatable loading floor, which are the main element of connection between the rotatable loading floor and motionless undercarriages part of the frame of the wagon, were tested. The maximum velocity at which the railway wagon can move was analyzed as well. There was also examined the speed at which the wheels separate from the rails, what results in derailment of the whole wagon. Based on the realized simulations of the transit of the special wagon at different velocities on the standard section of the track, there was determined the

boundary – maximum velocity of the wagon at which it can move on the arc of 250 m radius. After conducting a number of analyses, it was found that at velocity of 120 km/h a wagon with a loaded semitrailer tilts to the side after entering the arc, what results in derailment of a whole structure (Fig. 14). The centre of gravity of the semitrailer is located halfway of its height, what causes that the structure is more susceptible to interaction of centrifugal force and loss of stability. Lowering of the gravity centre of the semitrailer, due to a different type of load (for example, the load with reduced volume occupying the semitrailer), enables transit at higher velocity, however, it is not allowed from the point of arc characteristic and railway standards. The allowed standard velocity for this type of transit is 80 km/h. The boundary – maximal velocity at which a wagon can move on this type of arc with a gravity centre located halfway of semitrailer height without danger of derailment is 115 km/h.

Chapter 3 Analysis and Simulation of Railway Freight Wagon

3.1 Freight Wagon Model Description

The flat wagon dedicated along Ethiopia- Djibouti root path with gauge 1435mm provide transportation facility for two 6.096 meter or one 12.192 meter container that have 70t loading capacity according to ERC technical procurement data[19].



Fig 3.1 Freight wagon model [30]

A. Bogie Description

Regarding the type bogie which the freight wagon is equipped with, the most common running gear with the history of 150 years the so called three-piece bogie has been used in this study. The three pieces come from the one bolster which sits on the two side frames. The three piece bogies are the cheapest to purchase and most economical to maintain. However they provide low level lateral stability and poor ride quality. This is mainly due to having only primary suspension creating a high unsprung mass. The unsprung mass is mass of component which is not dynamically isolated from the track by suspension element.

Regarding the configuration bogie component, side frames connected to the wheelset via rubber blanket and adapter plus. The primary suspension system is comprised of a group of vertical springs and frictional damper between the bolster wedge and side frame [6].

I. Adapter plus

Side frames and axles are connected via the adapter plus. Adapter plus is rubber element which provides a weak elastic coupling between the mentioned bodies and avoid metal to

metal contact of the bodies.

II. Secondary suspension

The carbody is connected to the bogie via the secondary suspension. One of the main tasks of the freight wagon's secondary suspension is to damp out the yaw motion of the carbody. Yaw motion is the rotation about an axis perpendicular to the track plane. It is a possibility of resonance with the wheelset hunting frequency and the carbody yaw Eigen frequency around 1 Hz. Therefore, the secondary suspension's damping plays an important role in stability of the vehicle. The secondary suspension together with the primary suspension filters out the vibrations frequencies coming up from the wheelset above 20 Hz. Thus, carbody is isolated by its suspension system. The main elements of the three-piece bogies' secondary suspension are the centre plate and the side bearers [6].

Centre plate: - placed right in the middle of the bolster and it bears 90 percent of the vertical load of the loaded wagon and 20 percent of the vertical load of the unloaded wagon. The outer part of the spherical centre plate has chamfer which let the bolster to have roll motion [6].

Side bearing: - placed at both ends of the bolster. They handle the 80 percent of the vertical load of the unloaded wagon and 10 percent of the vertical load of the loaded wagon. The bearing always needs to contact with the carbody and provide Damping frictional force excess yaw motion of the carbody [6].

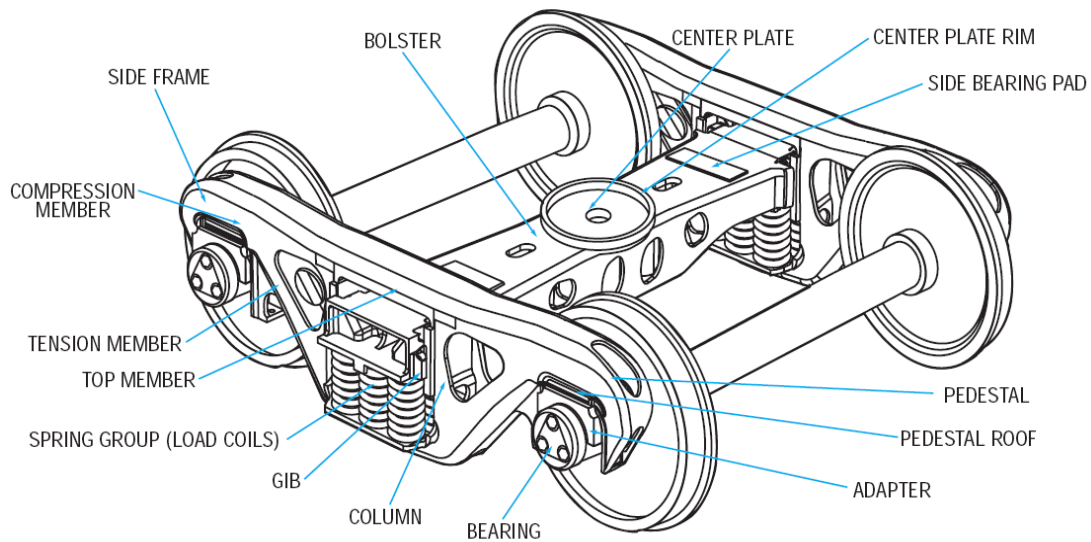


Fig. 3.2: A three-piece freight car truck [17]

B. Wagon Technical specification

Table 3.1 Technical specification of freight wagon [19]

No	The technical parameters of the wagon	Description
1	Model	X4K Wagon Flat Car
2	Bogie configuration	Bo-Bo
3	Wheel base distance	8715mm
4	Total weight	70 ton
5	Dead weight/ tare load	≤ 23 ton
6	Wagon under frame mass	13800kg
7	Height from Container Bearing Surface to Top of Rail (Empty Vehicle)	≤ 1200 mm
8	Container specification	Two Containers with Dimension 6.096 each or one container 12.192 (Total Weight of Single Container ≤ 35 t), or total 70 ton
9	Vehicle Length(Coupler Connection Point) (Coupler Connection Point)	≤ 13500 mm
10	Height from Container Bearing Surface to Top of Rail (Empty Vehicle)	1200mm
11	Side bearing distance	1500mm
12	Height from Center of Coupler to Top of Rail (Empty Vehicle)	880mm
13	Design Speed	100 km/hr.
	Technical parameters Bogie	
1	Bogie Type	3-piece American bogie
2	Fixed Wheel Base	1830mm
3	Side Bearing Center Distance	1520mm

4	Wheel Diameter	840mm
5	Height from Cored Disk to Rail Surface (Empty Vehicle)	680mm
6	Number of wheelset	4 in 2 bogie

3.2 Multi Body System Description

The freight rail wagon presented in Figure 3.1 consists of the three sub-assemblies: the wagon frame, the front bogie, and the rear bogie. Each bogie consists of the bogie frame, bolster, and two wheelsets as present in figure 3.3.

Under this study, the car body and bogie frames, bolster as well as the wheelsets are treated as rigid bodies; this means that the elasticity of each body and the shifting of their weight are neglected. Each rigid body has as illustrated in fig 3.4 have **6** DOF; (longitudinal x , lateral y , vertical z) to define the position of the center of gravity from inertial system and (roll ϕ , pitch and yaw angle ψ). On the other hand the bolster had three degree of freedom (lateral y , vertical z and roll ϕ). Totally the wagon model implemented under this study has 48 DOF.

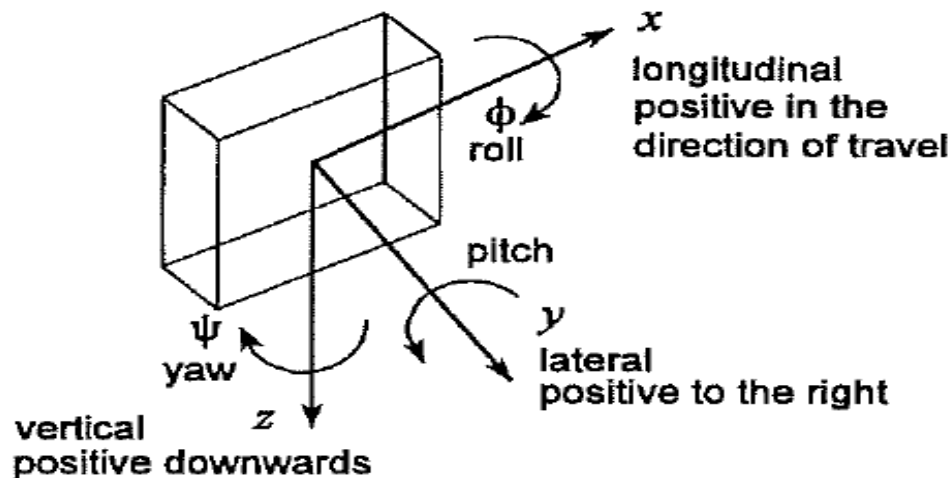


Fig 3.3 Rigid body degree of freedom

All rigid bodies are connected by suspensions element; which include torsional springs, non-linear damper and frictional dampers. Those suspensions are often used to support the car components and to provide vibration isolation.

Table 3.2 Summary of DOF each rigid body

Wagon component	Translation motion			Rotational motion		
	horizontal	lateral	bounce	yaw	roll	pitch
wagon frame	x_c	y_c	z_c	ψ_c	θ_c	ϕ_c
Bolster1	-	y_{bol1}	z_{bol1}	-	-	ϕ_{bol1}
Bolster2	-	y_{bol2}	z_{bol2}	-	-	ϕ_{bol2}
Wheelset 1	x_{w1}	y_{w1}	z_{w1}	ψ_{w1}	θ_{w1}	ϕ_{w1}
Wheelset 2	x_{w2}	y_{w2}	z_{w2}	ψ_{w2}	θ_{w2}	ϕ_{w2}
Wheelset 3	x_{w3}	y_{w3}	z_{w3}	ψ_{w3}	θ_{w3}	ϕ_{w3}
Wheelset 4	x_{w4}	y_{w4}	z_{w4}	ψ_{w4}	θ_{w4}	ϕ_{w4}
Bogie frame 1	x_{b1}	y_{b1}	z_{b1}	ψ_{b1}	θ_{b1}	ϕ_{b1}
Bogie frame 2	x_{b2}	y_{b2}	z_{b2}	ψ_{b2}	θ_{b2}	ϕ_{b2}

3.3 Analytical approach freight wagon multi-body dynamics

Analytical approach of dynamics includes:-

- A. Determination of creepage
- B. Determination of Creep force
- C. Determination of induced force and moment w/r contact
- D. Formulate equation of motion

A. Determination of creepage

The relative motion between two bodies i and j that are in contact can be the result of rolling and sliding motion. In the general case of rolling and sliding, the two bodies have different velocities. The different angular velocities Ω_i and Ω_j or relative angular velocity along the normal to the surfaces at the contact point creates spin. On the other hand velocities V_i and V_j at the contact point are not equal; the rolling motion is accompanied by longitudinal and lateral slide.

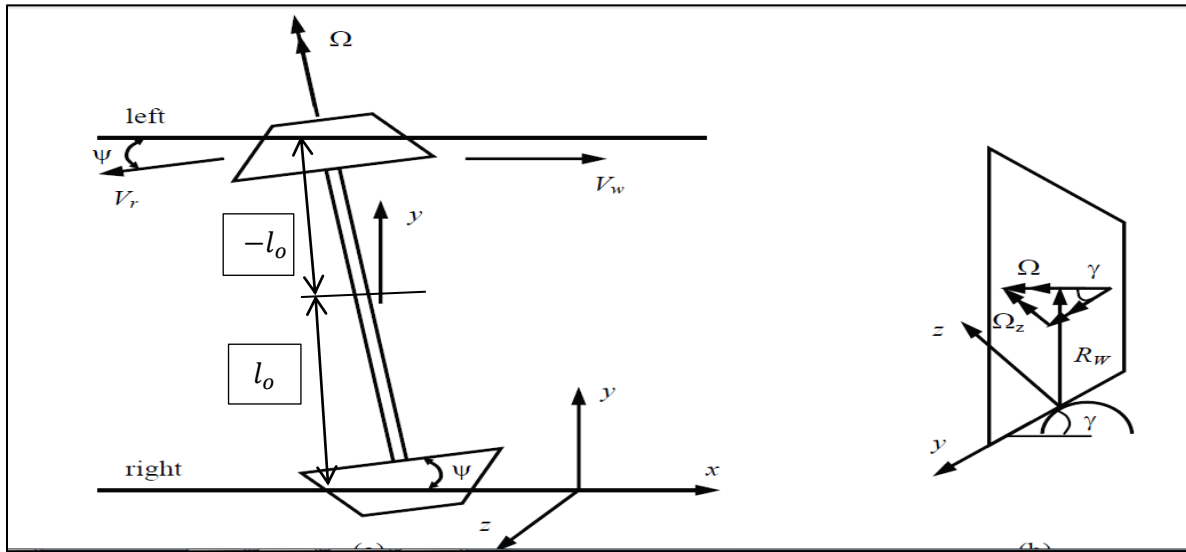


Figure 3.4 wheelset creepage [29]

Longitudinal creepage

$$S_{Xi} = \frac{V_x^{wi} - V_x^r}{V} \dots\dots\dots (3.1)$$

Where V_x^{wi} and V_x^r are longitudinal velocity for i-wheelset and rail respectively,

V Designate mean velocity

i imply (1,2,3 and 4 wheelset)

Equation 3.1 according to [27, 4] rewrite as:

$$S_{XLi} = 1 - \left(\frac{r_L}{r_o} \right) + \frac{l_o \cdot \dot{\Psi}_{wi}}{V} \dots \dots \dots (3.2)$$

$$S_{XRi} = 1 - \left(\frac{r_r}{r_o} \right)_i - \frac{l_o \cdot \dot{\Psi}_{wi}}{V} \dots \dots \dots (3.3)$$

Lateral creepage

$$S_{Yi} = \frac{V_y^{wi} - V_y^r}{V} \dots \dots \dots (3.4)$$

According to [27, 4] Equation 3.4 lateral creepage can be expressed as

$$S_{YRi} = \left[\frac{\dot{y}_{wi}}{V} - \Psi_{wi} \right] \left(\frac{r_r}{r_o} \right) + \frac{\dot{\Theta}_{wi}}{V} \dots \dots \dots (3.6)$$

$$S_{YLi} = \left[\frac{\dot{y}_{wi}}{V} - \Psi_{wi} \right] \left(\frac{r_L}{r_o} \right)_i + \frac{\dot{\Theta}_{wi}}{V} \dots \dots \dots (3.7)$$

Spin creepage

$$S_{ZLi} = \frac{\Omega_Z^{wi} - \Omega_Z^r}{V} \dots \dots \dots (3.9)$$

Where Ω_Z^{wi} and Ω_Z^r designate spin creepage of the wheelset and rail along z axis respectively

According to [27, 4] Equation 3.9 Spin creepage for the left and right i wheelset derived to

$$S_{ZLi} = \frac{\dot{\Psi}_{wi}}{V} - \left(\frac{\gamma_L}{r_o} \right)_i \dots \dots \dots (3.10)$$

$$S_{Zr} = \frac{\dot{\Psi}_{wi}}{V} + \left(\frac{\gamma_r}{r_o} \right)_i \dots \dots \dots (3.11)$$

B. Creep force determination

Kalker established a linear relationship between the developed creepages at the contact patch and the creep forces [4]. The maximum creep forces as determined by Kalker are as follows

$$\text{Longitudinal creep force right wheel} \rightarrow F_{xR} = -f_{11} \cdot S_{XRi}$$

Substitute longitudinal creepage equation (b.3) in the above equation gives

$$F_{xri} = f_{11} \cdot \left[1 - \left(\frac{r_r}{r_0} \right)_i - \frac{l_o \cdot \dot{\Psi}_{wi}}{V} \right] \dots \dots \dots (3.12)$$

Longitudinal creep force on left wheel $\rightarrow F_{xl} = -f_{11} \cdot S_{XL}$

$$F_{xli} = -f_{11} \cdot \left[1 - \left\{ \frac{r_L}{r_0} \right\}_i + \frac{l_o \cdot \dot{\Psi}_{wi}}{V} \right] \dots \dots \dots (3.13)$$

Lateral creep force on left wheel' $\rightarrow F_{yli} = -f_{22} \cdot S_{YLi} - f_{23} \cdot S_{ZLi}$

Substitute lateral creepage and spin creepage equation (b.7) and (b.8) in the above equation

Lateral creep force on the left wheel

$$F_{yli} = -f_{22} \cdot \left[\left[\frac{\dot{y}_{wi}}{V} - \Psi_{wi} \right] \left(\frac{r_L}{r_0} \right)_i + \frac{\dot{\theta}_{wi}}{V} \right] - f_{23} \cdot \left[\frac{\dot{\Psi}_{wi}}{V} - \left(\frac{y_L}{r_0} \right)_i \right] \dots \dots \dots (3.14)$$

Lateral creep force on the right wheel

$$F_{yri} = -f_{22} \cdot S_{YRi} - f_{23} \cdot S_{ZRi}$$

$$F_{yri} = -f_{22} \cdot \left[\left[\frac{\dot{y}_{wi}}{V} - \Psi_{wi} \right] \left(\frac{r_r}{r_0} \right)_i + \frac{\dot{\theta}_{wi}}{V} \right] - f_{23} \cdot \left[\frac{\dot{\Psi}_{wi}}{V} + \left(\frac{y_r}{r_0} \right)_i \right] \dots \dots \dots (3.15)$$

$$f_{11} = GabC_{11}, f_{22} = GabC_{22} \text{ and } f_{23} = G(ab)^{1.5}C_{23}$$

Where C_{11}, C_{22} and C_{23} Are creep coefficient tabulated by kalker, refer Appendix E

G Shear modulus of rigidity

a and b are the semi-axis of elliptical wheel-rail contact

C. Wheel Rail Contact force

Vehicle dynamic performance is strongly influenced by Wheel-Rail interacting force. For an applied load on a wheel-rail interface; normal contact forces and tangential (creep) force develop on the contact patch depending on the total vertical force applied. And also the contact angle of the wheel-rail contact formed as a result of the lateral displacement y of the wheelset during wagon motion.

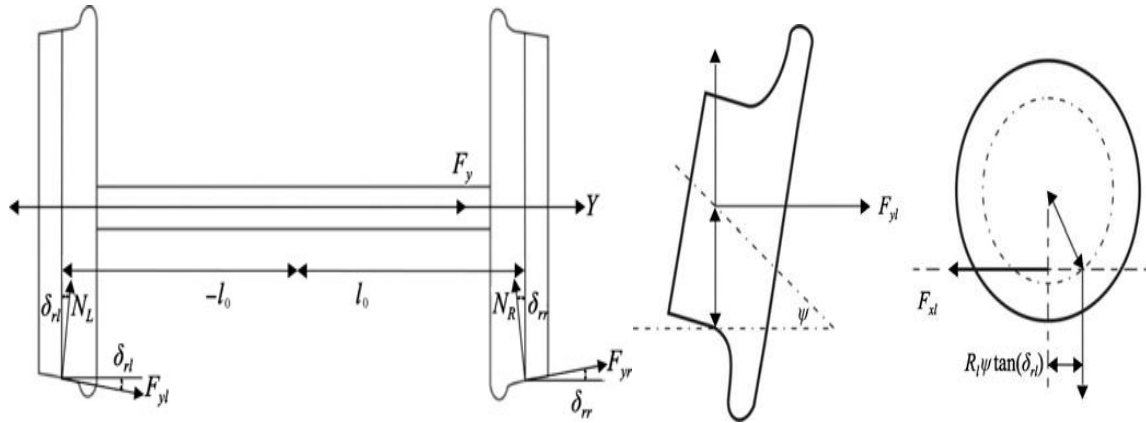


Fig 3.5 wheel-rail contact force [26]

The total sum of the forces developed on the wheel-rail contact for i number of wheelset as illustrated in Fig.3.6 can be expressed as follows;

Longitudinal force developed in w/r contact expressed as

$$F_{xi} = F_{xri} + F_{xli} \dots \dots \dots (3.1a)$$

Lateral force induced in the w/r contact by using sum of force lateral direction derived as

$$F_{yi} = -[F_{yri} \cos \delta_{ri} + F_{yli} \cos \delta_{li} - N_{ri} \sin \delta_{ri} + N_{li} \sin \delta_{li}] \dots \dots \dots (3.2a)$$

Vertical force induced in the w/r contact

Total reaction force developed in the vertical direction at wheel rail contact for i number wheelset can be derived

$$F_{zi} = (-F_{yli} \sin \delta_{rli} + F_{yri} \sin \delta_{rri} + N_{ri} \cos \delta_{rr} + N_{li} \cos \delta_{rl}) - \left(\frac{1}{4}m_c + \frac{1}{2}m_{bi} + m_{wi} \right) \cdot g$$

$m_a = \frac{1}{4}m_c + \frac{1}{2}m_b + m_{wi}$ Where m_a designate mass on the axle

$$F_{zi} = (-F_{yli} \sin \delta_{li} + F_{yri} \sin \delta_{ri} + N_{ri} \cos \delta_{ri} + N_{li} \cos \delta_{li}) - m_a \cdot g \dots \dots \dots (3.3a)$$

The total moment can be expressed based on illustrated in figure 3.6

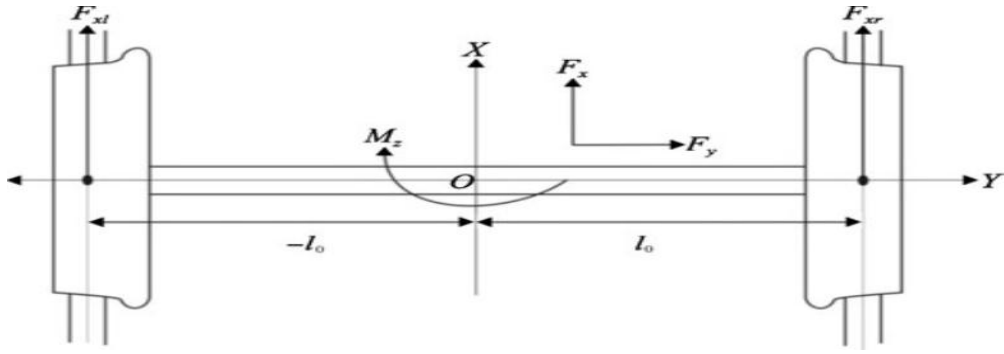


Fig 3.6 wheelset force and moment [26]

Moment along x-y direction can be determine as

$$M_{xyi} = F_{xli} \cdot l_o - F_{xri} \cdot l_o + M_{zi} \dots \dots \dots (3.4a)$$

Where M_z external moment along vertical direction

Similarly, the total moment in the z-y direction can be derived to

$$M_{zyi} = l_o(F_{yri} \sin \delta_{ri} + N_{ri} \cos \delta_{ri}) - l_o(F_{yli} \sin \delta_{li} + N_l \cos \delta_{li}) - r_r(F_{yri} \cos \delta_{ri} - N_{ri} \sin \delta_{ri}) - r_{Li}(F_{yli} \cos \delta_{li} + N_l \dots \dots \dots (3.5a)$$

The total moment in the z-x direction can be derived to

$$M_{zxi} = [F_{xri} + (F_{yri} \cos \delta_{ri} - N_r \sin \delta_{ri})\Psi_{wi}]R_{ri} - [(F_{yr} \sin \delta_{rr} + N_r \cos \delta_{rr})\Psi_{wi} \cdot \tan \delta_{ri}]R_{ri} + [F_{xli} + (F_{yli} \cos \delta_{li} + N_r \sin \delta_{li})\Psi_{wi}]R_l + [(-F_{yri} \sin \delta_{li} + N_{li} \cos \delta_{ri})\Psi_{wi} \dots \dots \dots (3.6a)$$

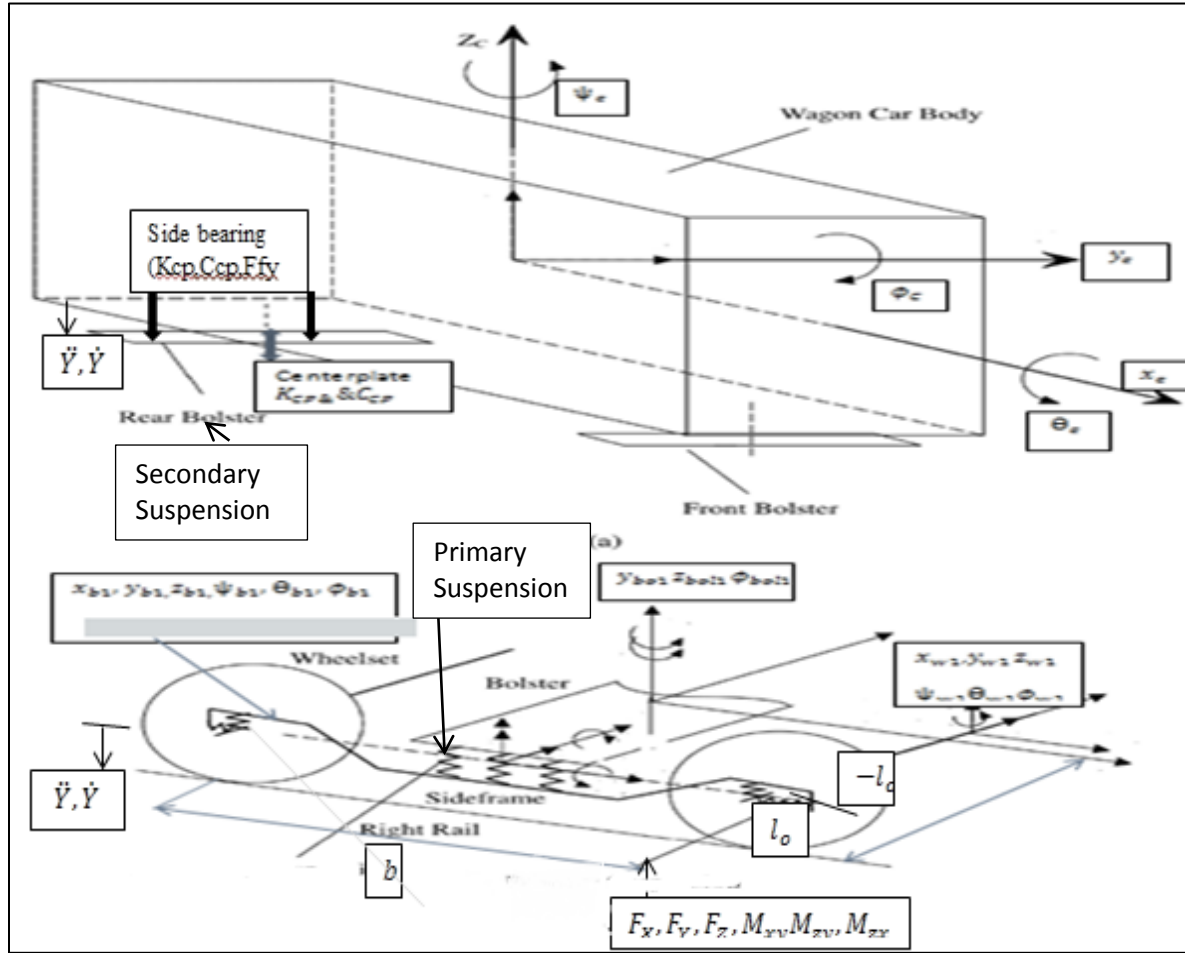
D. Equation of motion formulation

The wagon frame, two bogies and four wheelsets are considered as rigid bodies. Spring and damping elements representing the secondary suspension connect the car body with the two bogies as illustrate in figure 3.8. The total vehicle system model is represented by 48-DOF.

- Global coordinate taken at the wagon centre therefore coordinate transformation not consider
- The rail vehicle is symmetric along longitudinal plane.
- The rail vehicle is travelling at constant speed

- Adapter-plus analytically modeled as springs and damper parallel.
- Creep forces are assumed as linear and are determined using linear Kalker's theory.
- Car body is assumed to be rigid.
- The wheel and rail do not loose contact during motion.

NB. Since driving all system motion equation is tedious only major rigid body's equation of motion have been derived (i-number of wheelset and carbody).



3.7 simplified rail vehicle schematics

Wheelset (i number) lateral motion

$$m_{w1}\ddot{y}_{w1} + F_{ES} + F_{ED} = F_y$$

$$F_{\text{Elastic suspension stiffness}} = F_{ES} = 2k_y[(y_{w1} - y_{b1}) - b\Psi_{b1} - h_1\Theta_{b1}]$$

$$F_{\text{Elastic suspension damping}} = F_{ED} = 2k_y[(y_{w1} - y_{b1}) - b\dot{\Psi}_{b1} - h_1\dot{\Theta}_{b1}]$$

Where, b Represent bogie wheel base distance

h_1 Represent height from bogie C.G to primary suspension wheelset

F_y Imply Wheel rail reaction force determine in equation 3.2a

$$m_{w1}\ddot{y}_{w1} + 2k_y[(y_{w1} - y_{b1}) - b\dot{\Psi}_{b1} - h_1\dot{\Theta}_{b1}] + 2k_y[(y_{w1} - y_b) - b\dot{\Psi}_b - h_1\dot{\Theta}_b] = F_y$$

Wheelset vertical motion

$$m_{w1}\ddot{z}_{w1} + F_{z,ES} + F_{z,ED} = F_{Z1}$$

$$F_{Z,\text{vertical Elastic damping}} = 2c_{pz}(\dot{z}_{w1} - \dot{z}_b - \phi_{b1}b)$$

$$\text{and } F_{\text{Vertical elastic stiffness}} = 2k_{pz}(z_{w1} - z_{b1} - \phi_{b1}b)$$

$$m_{wi}\ddot{z}_{wi} + 2c_{pz}(\dot{z}_{wi} - \dot{z}_{bi} - \phi_{bi}b) + 2k_{pz}(z_{wi} - z_{bi} - \phi_{bi}b) = F_{Zi}$$

F_{Zi} Vertical Reaction force at wheel rail contact derived in equation 3.3a

Wheelset yaw motion

$$I_{z,wi}\ddot{\Psi}_{wi} + 2k_x l_o^2(\Psi_{wi} - \Psi_{bi}) = M_{xyi}$$

Where l_o distance between right and left wheel

M_{xyi} Represent moment along z axis derived in equation 3.4a

Wheelset roll motion

$$I_{y,wi}\ddot{\Theta}_{wi} + 2k_{pz}b^2(\Theta_{wi} - \Theta_{bi}) + 2k_{pz}b(z_{wi} - z_{bi}) + 2k_{pz}b^2(\dot{\Theta}_{wi} - \dot{\Theta}_{bi}) + 2c_{pz}b(\dot{z}_{wi} - \dot{z}_{b1}) = M_{zx,i}$$

$M_{zx,i}$ Represent moment along y-axis derived in equation 3.6a

Wheelset pitch motion

$$I_{x,wi}\ddot{\phi}_{wi} + 2c_{pz}l_o^2(\dot{\phi}_{wi} - \dot{\phi}_{bi}) + 2c_{pz}l_o(\dot{z}_{wi} - \dot{z}_{bi}) + 2k_{pz}l_o^2(\phi_{wi} - \phi_{bi}) + 2k_{pz}l_o(z_{wi} - z_{bi}) = M_{zy,i}$$

$M_{zy,i}$ Represent moment along x- axis derived in equation 3.5a

Derived motion equation for Car body

Lateral motion

$$M_c\ddot{y}_c + F_{\text{stiffness,cp}} + F_{y,\text{side bearing}} + F_{y,\text{damping,cp}} + F_{\text{lateral Centrifugal}} = 0$$

$$M_c\ddot{y}_c + 2k_{y,cp}[y_c - (y_{bol,1} + y_{bol,2}) + 2\Theta_ch_2] + 2c_{y,cp}[(\dot{y}_c - (\dot{y}_{bol,1} + \dot{y}_{bol,2})) - 2\dot{\Theta}_ch_2] + 2F_{fy} - m_c(\frac{v^2}{R}\cos\varphi_t - g\sin\varphi_t) = 0$$

Where F_{fy} Side bearing frictional damper

h_2 Height from bolster centre of gravity to centre plate suspension

Vertical motion

$$M_c\ddot{z}_c + F_{z,\text{stiffness,cp}} + F_{z,\text{stiffness,side bearing}} + F_{z,\text{damping,side bearing}} + F_{z,\text{damping,cp}} + F_{\text{vertical Centrifugal}} = 0$$

$$m_c\ddot{z}_c + 2c_{z,cp}(\dot{z}_c - (\dot{z}_{bol,1} + \dot{z}_{bol,2})) + c_{z,sb}(\dot{z}_c - (\dot{z}_{bol,1} + \dot{z}_{bol,2})) + k_{z,cp}(z_c - (z_{bol,1} + z_{bol,2})) + 2k_{z,sb}(z_c - (z_{bol,1} + z_{bol,2})) + -m_c(\frac{v^2}{R}\sin\varphi_t + g\cos\varphi_t) = 0$$

Yaw motion

$$I_{cz}\ddot{\Psi}_c + 2k_{x,cp}l_o[\Psi_c] + [2k_{y,cp}l_o(y_c - (y_{bol,1} + y_{bol,2}))] - I_z V \frac{d}{dt}(\frac{\cos\phi_t}{R_o}) = 0$$

$I_z V \frac{d}{dt}(\frac{\cos\phi_o}{R_o})$ Implies centrifugal force angular [4]

Since yaw motion bolster constrained, $\Psi_{bol,1} + \Psi_{bol,2} = 0$

Car body roll

$$I_{y,c}\ddot{\Theta}_c + 2k_{sb,z}l_o^2(\Theta_c - (\Theta_{bol,1} + \Theta_{bol,2})) + 2k_{sb,z}l_o(z_c - (z_{bol,1} + z_{bol,2})) \\ + 2k_{sb,z}l_o^2(\dot{\Theta}_c - (\dot{\Theta}_{bol,1} + \dot{\Theta}_{bol,2})) + 2c_{sb,z}l_o(\dot{z}_c - (\dot{z}_{bol,1} + \dot{z}_{bol,2})) = 0$$

N.B Centre plate suspension contribution on rolling motion is almost insignificant.

- The equations of motion of rail vehicle are obtained in the following form

$$[M]\{\ddot{Y}\} + [C]\{\dot{Y}\} + [K]\{Y\} = [F_r(w)]$$

- Where $[M]$, $[K]$ and $[C]$ are the 48×48 mass, stiffness and damping matrices respectively for vehicle $[F_r(w)]$ wheel rail contact force, centrifugal force, gyroscopic force including momentum for wagon.

$$M = \text{diag}(m_{wi}, m_{boli}, m_{bi}, m_c, I_{wxi}, I_{wyi}, I_{wzi}, I_{bxi}, I_{byi}, I_{bzi}, I_{xboli}, I_{yboli}, I_{zboli}, I_{cx}, I_{cy}, I_{cz})$$

$$Y = [x_{wi}, y_{wi}, z_{wi}, \Theta_{wi}, \Phi_{wi}, x_{bi}, y_{bi}, z_{bi}, \Psi_{bi}, \Theta_{bi}, \Phi_{bi}, y_{boli}, \psi_c, \Theta_c, \Phi_c]^T$$

3.4 Multi-body modelling and simulation in SIMPACK interface

3.4.1 Development of the vehicle model

The wagon model that consist the following major components, modelled in Simpack interface as rigid bodies:-

- Wheelsets (4 bodies)
- adapter plus Axle boxes (8 bodies)
- wagon platform (1 body)
- Bolster 2 (14 bodies)
- bogie frame (2 bodies)

I. 3D Modelling

The procedure used to model the wagon

- The 3D geometry of car body, side frames bolster constructed in Catia modelling interface as per Appendix D respectively, and saved as “stl” model format in SIMPACK database directories, or in the local working director.
- The files containing the geometry allocated in the SIMPACK database directories, or in the local working directory, using global search path.
- 3D primitive type 19 (cad interface) used to import the build wagon part from working directory.
- Once the complete three piece rear bogie built further duplication made for front bogie model using by loading the already completed bogie manually in to SIMPACK searches path directory database, then substructure will be used to import the model based precise allocation of marker which define wheel base distance.

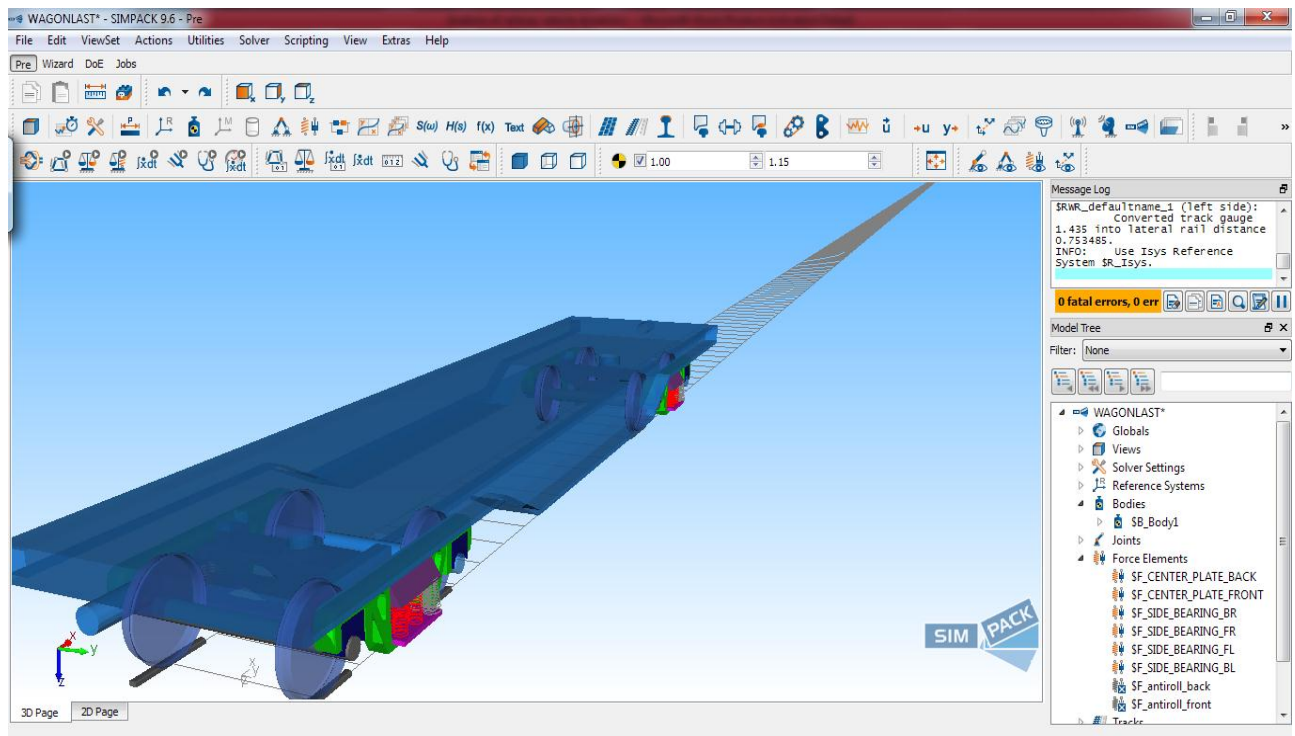


Fig 3.8 Fright Rail Wagon Model

II. Input

Each rigid body defined by their characteristics parameter i.e. mass, moments of inertia, and the position of the centre of gravity incorporated in preprocessor modelling as per the attached **appendix A.1** along with relative position of the bogies with respect to reference marker as illustrated in figure 3.9.

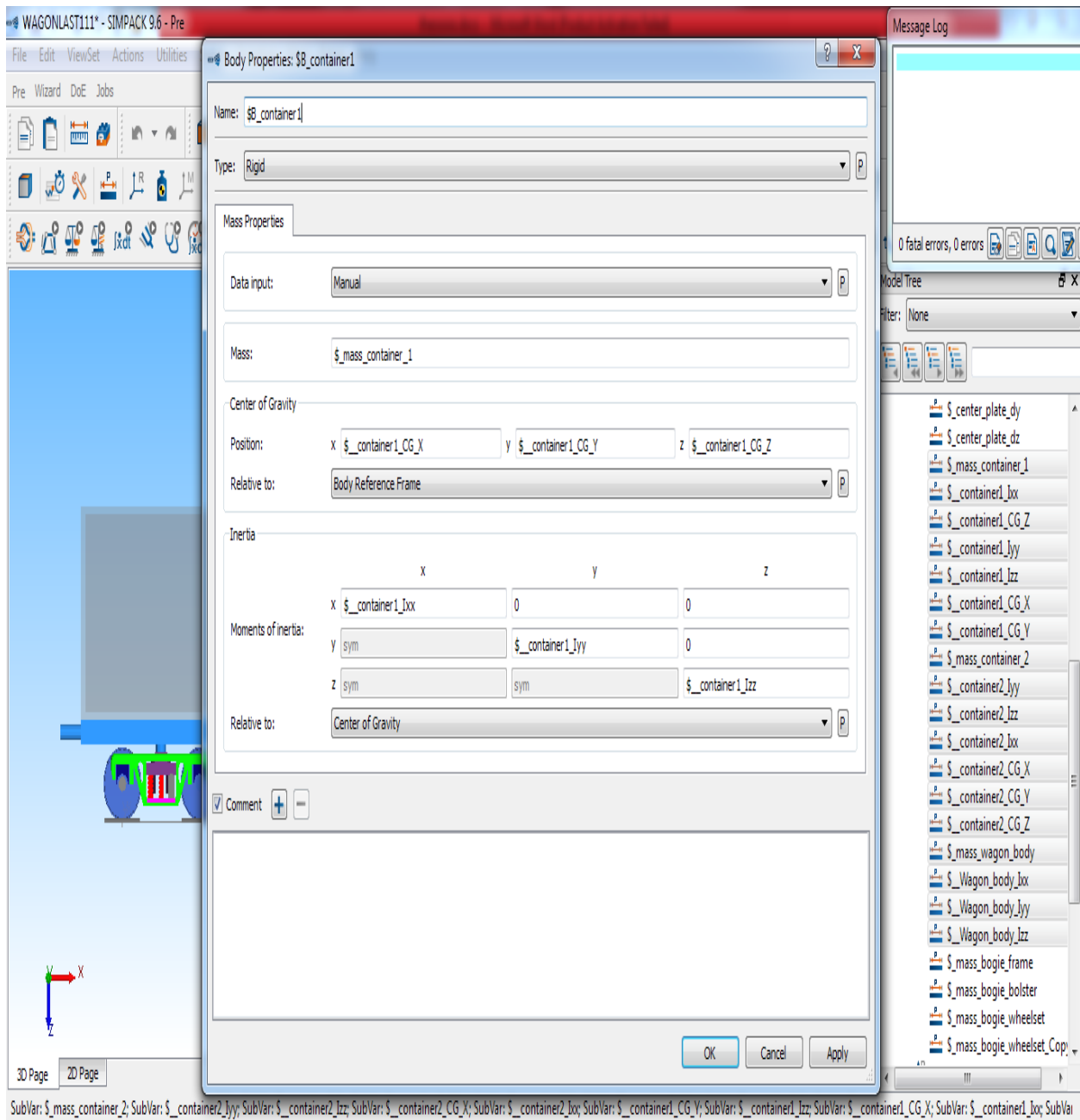


Fig 3.9 Rigid Body Input Data

III. Wheel rail contact modelling

- SIMPACK wheel and rail modelled using data as described in **appendix B.1 & C.1** i.e. type wheel & rail profile type, nominal wheel radius, wheel lateral distance, rail gauge, rail cant for both right/left wheel- rail and material property data as illustrate in figure 3.10.
- Wheel rail contact modelling regarding normal force i.e. “Hertez” half elastic theory recommended by Simpack has been used.
- To model wheel-rail contact considering tangential Forces (Creep force) “FASTSIM” simplified version of the nonlinear used.
- The friction coefficient parameter of the rail-to-wheel contact $\mu=0.4$ has been taken as input to determine the tangential or creep force.

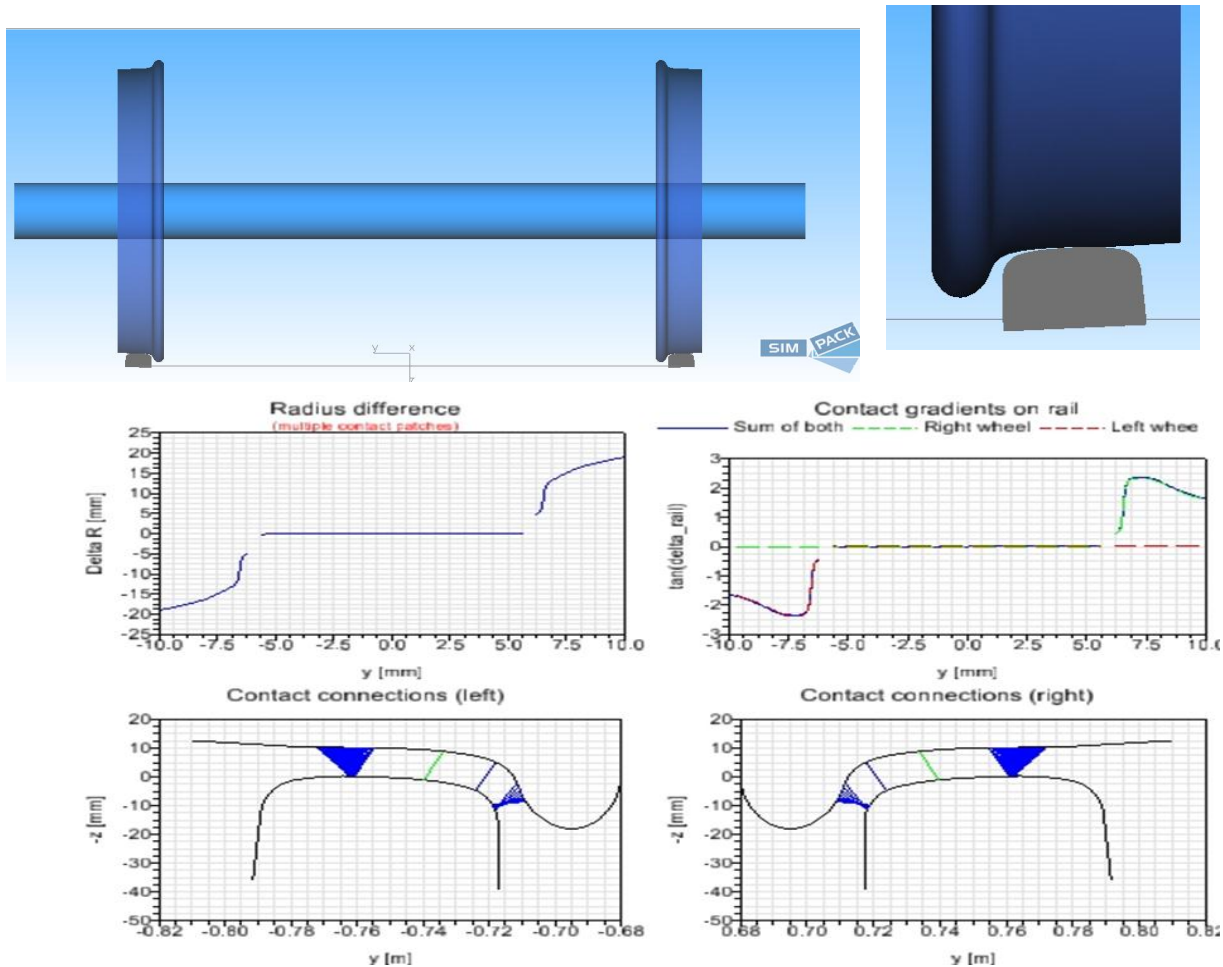


Fig 3.10 wheel rail contact

- During operation the wheelset will perform a typical sinusoidal movement relative to center of track, which is a sine-like oscillation around the track center. As describe figure 3.11 it is obvious that the effective linear oscillation characteristic looks quite different from the theoretical solution in the origin. The task of the quasi-linearization is to find most realistic linear characteristics of the three non-linear functions listed below, considering a given amplitude of the hunting oscillation
 - The difference of the actual left and right wheel radii at the contact points
 - The mean roll angle of the wheels about longitudinal track axis
 - The mean value of the left and right contact angles on the rail
(Both contact angles are measured about the longitudinal track axis.)
- SIMPACK provides two methods for an automatic quasi-linearization; “harmonic” and “equivalent”. Under this study a harmonic (sinusoidal) approach for the assumed cyclic hunting movement of the axle used to model rail-wheel contact [20].

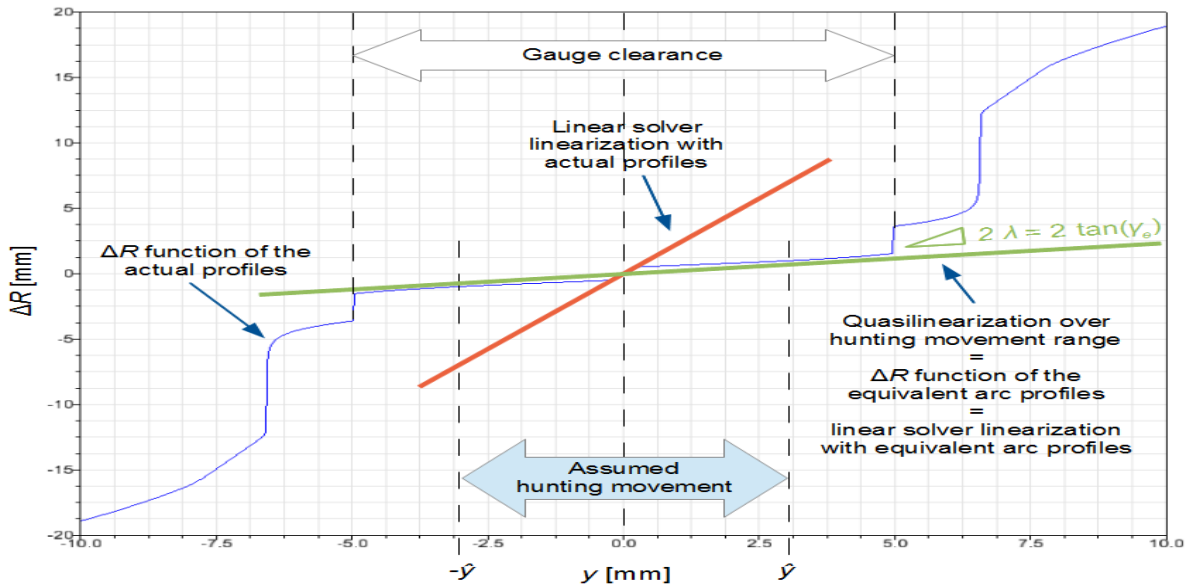


Fig 3.11 Difference between the normal linearization by the linear solvers and the quasi-linearization

IV. Joint

- The car body, bogie frame, the wheelsets and bolster maintained with general rail track by joint type 7 as illustrated in fig 3.14, this joint type describes the movement of the main components of a rail vehicle in space with six degrees of freedom The coordinates

used for the description of the movement are not related to a global Cartesian coordinate system but to the course of the currently active Track. This allows us to define and see the relative positions to the track directly in the joint states.

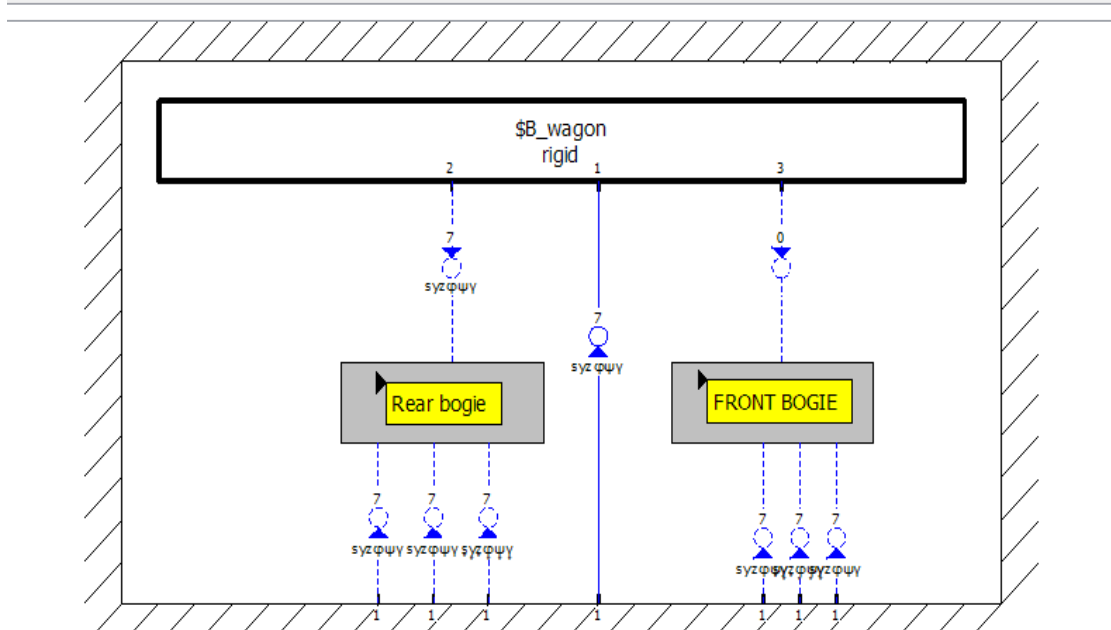


Fig 3.12 Rail wagon joint description

V. Suspension Force element

- Wheel rail contact modelled using wheel rail interface force element (type 78). This force element provides the rail –wheel contact force calculation.
- The side frames are maintained on the axle boxes through adapter, which can move longitudinally on the side frame in order to provide the wheel sets with a yaw degree of freedom; modelled by stick-slip 2D Compact force element (type 194). It enables to damp out excess wheelset yaw motion.
- Bolster wedge and frame modelled using stick-slip 2D Compact force element (type 194) as per appendix 1B suspension design data and the motion of bolster constrained along longitudinal(x), rotation in horizontal axis(pitch) and rotation in vertical direction(yaw) using user constrained (type 25).
- Primary springs connects bolster to bogie frame are modelled by using torsional spring (type 79) force element. The force element describes a typical coil spring (helical spring) with shear forces and bending torques coupled.

- Vertical Force element connects bolster and lower part of bogie frame modelled using spring damper serial point to point (type 6).
- Side bearing modelled using stick-slip 2D Compact force element (type 194) according suspension design data.
- The centre plate modelled as a nonlinear stiffness parallel with a damping element using SIMPACK force element 5 (spring-damper parallel compact) according **appendix 1B** suspension design data. This Force element applied for suspension forces between two coincident Markers in multiple axis directions, including optional clearances.

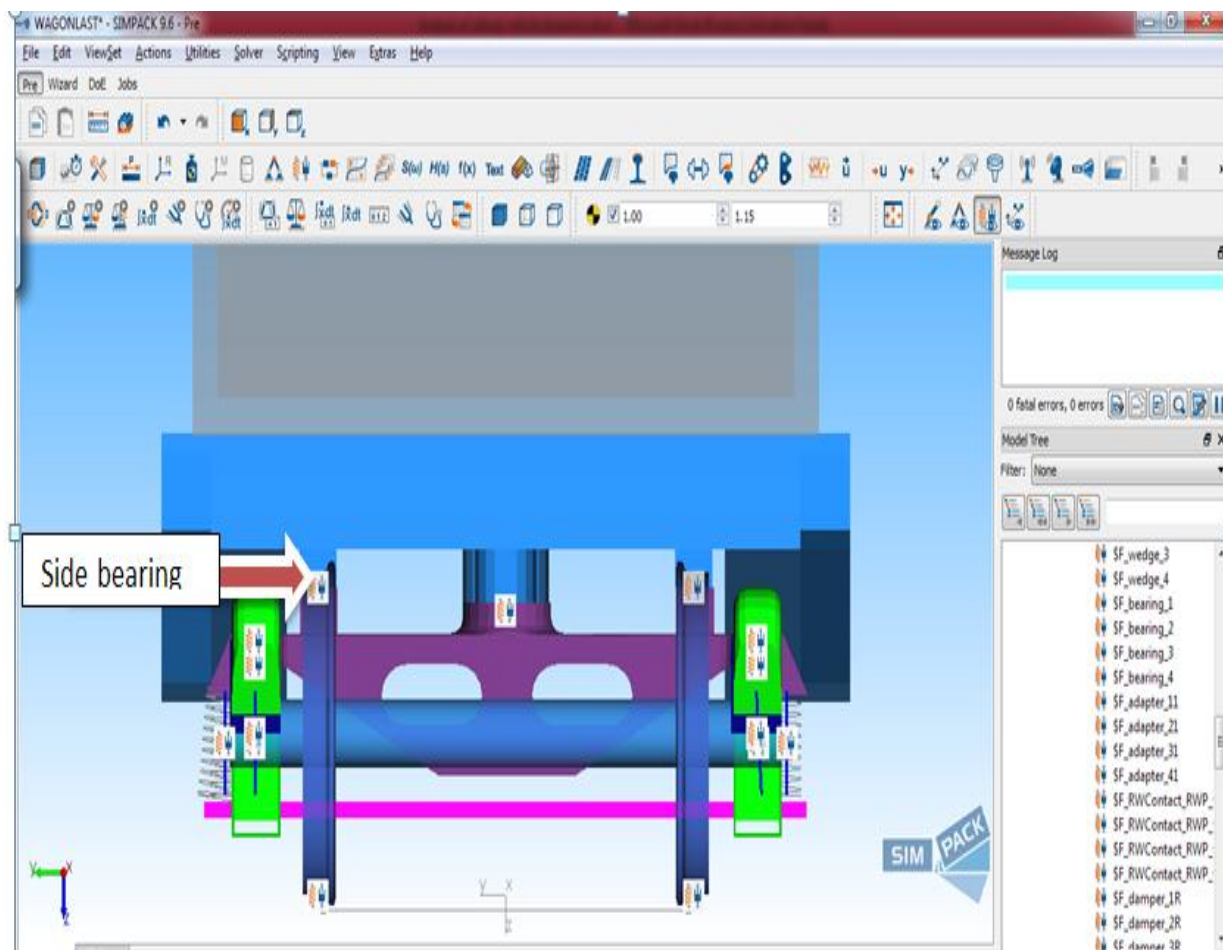


Fig 3.13 side view illustration suspension force

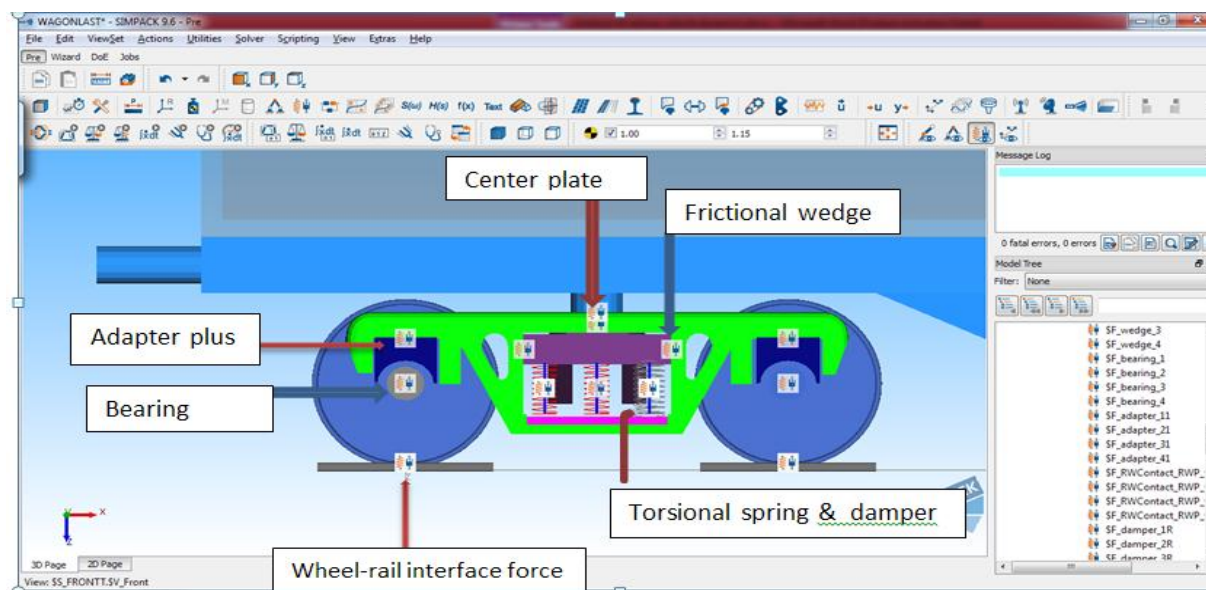


Fig 3.14 front view illustration suspension force

The force elemrnt can be summerized as follow:-

Table 3.3 suspension force summary

Force type	Element connect	Type of force
Wheel/rail interaction	between wheel and rail	Wheel/rail interface 78
Primary suspension		
Adapter plus	Wheelset axle box and side frame	Stick/slip friction 2d cmp
Wedge	Bolster and side frame	Stick/slip friction 2d cmp
Coil spring	Bolster and lower side of frame	Torsional spring type 79
Damper	Bolster and lower side of frame	Spring damper serial ptp
Secondary suspension		
Side bearing	Bolster and pivot car body	Stick/slip friction 2d cmp
Center plate	Bolster and wagon body side	Spring damper parallel cmp

3.4.2 Track model

One of the characteristic feature of railway vehicles is to support and act as a guidance of rail wagon. In SIMPACK modelling interface standard or catographic track type have been used. To consider actual operational scenario i.e. straight track(tangents), curve entries (transition curves) and constant radius curves have been taken. Table 3.4 track geometry parameter i.e. length, curvature radius track cant and rail inclination been used to model the standard track layout.

Table 3.4 track input data

No	Track parameter	Integration Time(sec)	Length(m)	Radius(1)m	Radius(2)m	Cant height(m)
1	Straight track	$0 \leq t \leq 17.7$	500	0	0	0
2	Intermediate	$17.7 \leq t \leq 32$	400	0	350	0.150
3	Circular track	$32 \leq t \leq 53.3$	600	350	350	0.150
4	Intermediate track	$53.3 \leq t \leq 64$	300	350	0	0.150
	Total track length		1800	Time integration= L/V = $1800\text{m}/27.7\text{m/s}$ = 64sec		

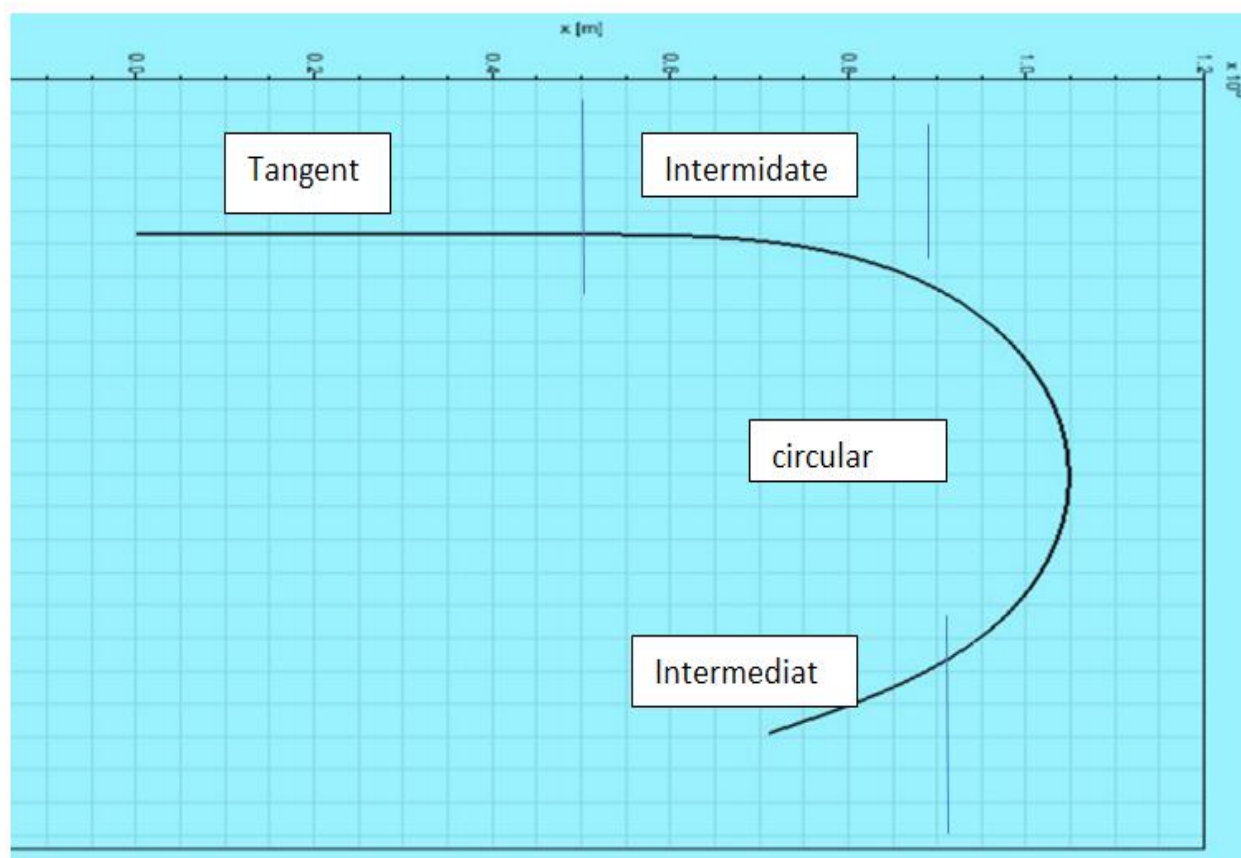
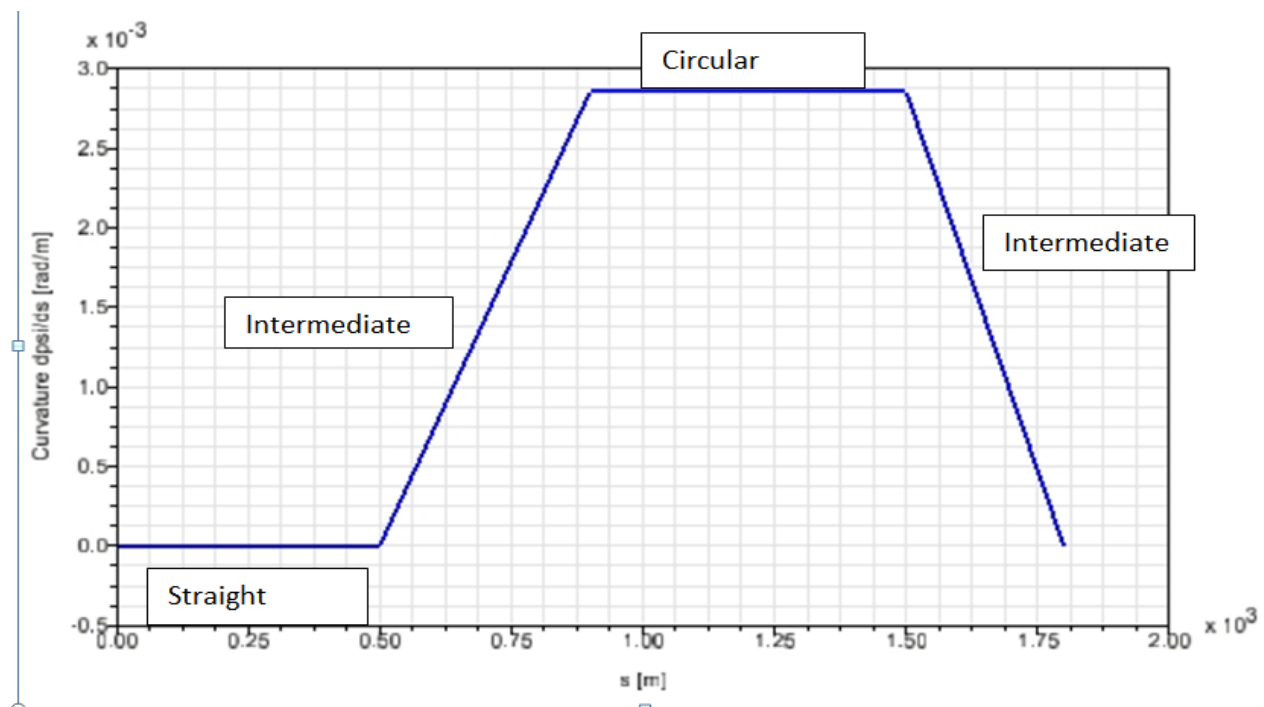


Fig 3.15 Track curvature and root path

3.4.4 Simulation scenario

Rail wagon performance simulation undertake based on condition outline i.e.

- Continuous straight, intermediate track condition according to specified design data table 3.5, under maximum loading condition (laden load + tare load=93 ton)
- Ultimate operation maneuver (speed=100km/hr or 27.7 m/s)
- constant Coefficient of friction between wheel-rail contact $\mu = 0.4$ taken throughout the vehicle run
- Integration time=
$$\frac{\text{track length}}{\text{speed}} = \frac{1800\text{m}}{27.7\text{m/s}} = 64 \text{ sec}$$
- Integration method SODASRT 2(Recommended by SIMPACK)
- Integration time step frequency 100Hz
- Integration number of point 401
- Time integration step size 0.1

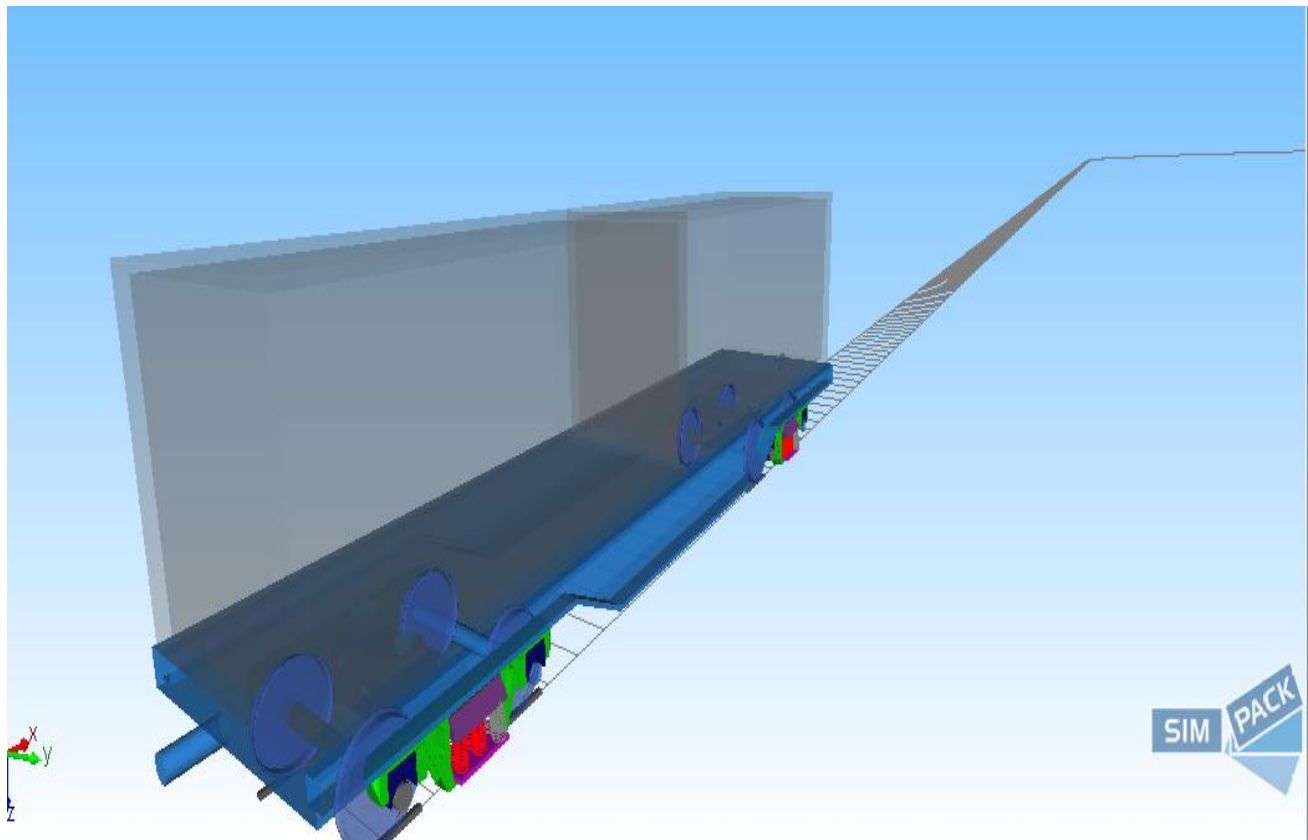


Fig 3.16 Freight wagon simulation model

Chapter 4 Validating Functions

4.1 Description validating standard

In the process of evaluating and approving a new rail vehicle there are many requirements that have to be fulfilled in order to get it qualified to run on track, without any risks of exceeding limit values for safety and running behaviour. According to the international standards specified in the UIC code 518 there are, above all, requirements on track forces and accelerations that have to be met when evaluating running behaviour and lateral dynamic stability. All quantities and limit values in this section refer to UIC 518 [29], EN 14363 [21, 22] and research based standard for wear index. The vehicle model used in the present simulation study has a total mass of 93 ton, equally distributed on four axles.

4.2 Ride Quality index

One of the most important aspects that any type of transportation must ensure is an acceptable level of comfort perceived by the passengers or freight. There are different methods for measuring and evaluating ride quality and comfort, e.g. Discomfort. A very common approach, which also is used in the present simulation work, is to use the W_z (Wertungszahl), which originates from German research.

W_z is evaluated from lateral and vertical accelerations in specific measuring points on the carbody floor. It is a frequency-weighted root mean square-value of accelerations, in logarithmic scale, which can be calculated according to [22, 21].

$$WZ = 4.42 \cdot (a^{wrms})^{0.3} \dots\dots\dots 4.1$$

a^{wrms} Makes reference to the root-mean-square value of the frequency-filtered accelerations.

Table 4.1 Ride quality classification [CEN (1999)].

condition	W_z (passenger)	W_z (fright)	W_z (locomotive)
Excellent	<2.5	<3.5	<2.75
Good	2.5-2.75	3.5-4.0	2.78-3.1
Limitation	2.75-3.0	4.0-4.25	3.1-3.45

4.3 Safety

Apart from the ride comfort index, another aspect that compromises the freight integrity is the level of the railway vehicle safety. Without any kind of hesitation, this parameter has to be scrupulously studied until acceptable levels are achieved. In this way and following the UIC 518 and EN-14363 standard [CEN (2005)] [27, 22], the safety of a rail vehicle is assessed by means of two quantities: the track shift forces and derailment coefficient.

4.3.1 Track shift forces

The first parameter that quantifies safety deals with the lateral forces created due to the wheel-rail contact as the vehicle runs over the track. This is particularly important because a high value of track shift forces leads to track irregularities (which increases the maintenance costs).

Equation (2.4) defines how to calculate this value for the leading wheelset [CEN (2005)] [27, 22]:

$$\Sigma Y \leq k_1 \left(10 + \frac{2Q_o}{3} \right) \text{ (KN)} \dots\dots\dots 4.2$$

Where k_1 a constant is factor and $2Q_o$ is mean axle load of the vehicle defined by the equation

$$2Q_o = \frac{m_{veh} \cdot g}{n} \dots\dots\dots 4.3$$

Where m_{veh} is the mass of the rail wagon, g the gravitational force and number of axle of the vehicle.

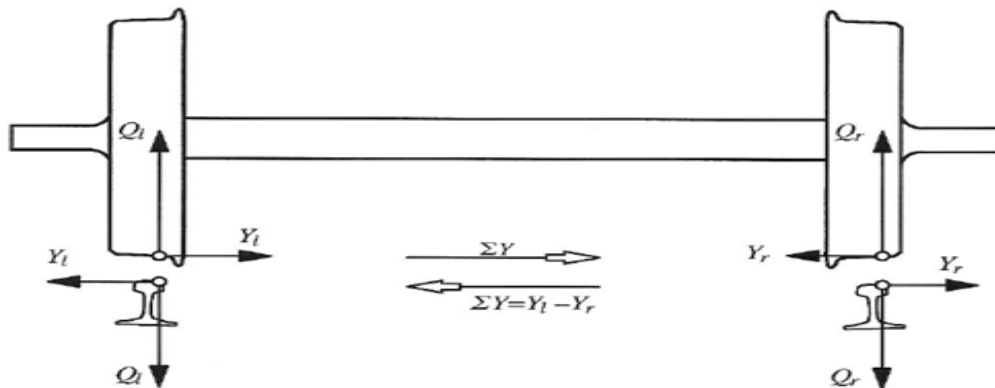


Figure 4.1 Resultant Track shift force on wheelset

The safety critical limit for track shifting according to prud'homme criterion and UIC standard is

$$|Y_{lim}| = \sum Fy_{lim} = k_1 \left(10 + \frac{2Q_0}{3} \right) \text{ (KN)} \dots\dots\dots 4.4$$

Where $k_1=1$, under this study $m_{wagon} = 93,000kg$

Using equation (4.3) $2Q_0 = 228.0825KN$

The limiting lateral track shift force under this study calculated as

$$\sum Fy_{lim} = 76037.5N$$

4.3.2 Derailment coefficient

Studies show that insufficient railway maintenance, train collisions, severe braking, passing over zigzag or curved routs, vertical and lateral rail irregularities are the main causes of train derailments. Various criteria have been used to predict the onset of derailment. One of these parameters is derailment coefficient which is defined as the ratio of lateral to vertical load at the wheel-rail contact point (y/Q). Nadal, put forward the first equation to calculate the critical derailment coefficient. Figure 4.2 shows the system of forces acting on the flange contact point.

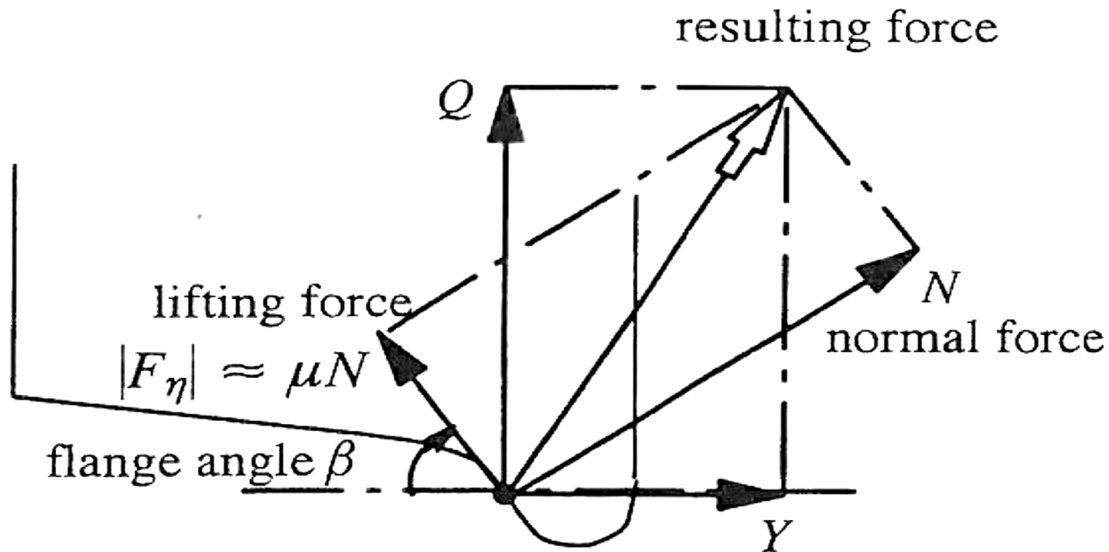


Figure 4.2 Forces acting on the wheel in wheel-rail flange contact.

This coefficient determines the minimum value of $\frac{y}{Q}$ at which flange may climb the rail,

$$Y = N \sin \beta - N \cdot \mu \cos \beta \dots\dots\dots 4.5$$

$$Q = N \cos \beta + N \cdot \mu \sin \beta \dots\dots\dots 4.6$$

$$\left[\frac{Y}{Q} \right]_{nadal} = \frac{\tan \beta - \mu}{1 + \mu \tan \beta} \dots\dots\dots 4.7$$

According to **CEN and UIC Code 518** [27, 22]

$$\left(\frac{Y}{Q} \right) = 0.8; \text{ For } R \geq 300 \text{ m}$$

Where, Y and Q represent the lateral and vertical forces on wheel–rail contact respectively.

And β designate the attack angle and μ coefficient of friction at wheel-rail contact.

4.4 Rail–wheel Wear

The last design parameter used to characterize the economic aspects of a railway vehicle considered in this work is the so called rail–wheel wear and is related to the change of geometry of both wheel and rail profiles due to the contact forces and corresponding wear present in the contact patch between both elements.

The contact formulation used here is governed by non-linear equations and the theory behind the contact forces, the corresponding creepages in the contact patch. Several approaches are present in the literature explaining with more or less accuracy the loss of material present in the above mentioned contact.

For the purpose of this project, a simple but widely accepted approach of the wear computation has been adopted [Orvnäs (2011)]] [Mousavi, M., Berbyuk, (2013)] [23, 25]. It is based on the assumption that the wear present in the rail-wheel contact is linearly related to the energy dissipated in the process.

The energy dissipated is defined by equation:

$$\bar{E} = F_X \cdot S_{XL} + F_Y \cdot S_{YL} + M_Z \cdot S_{ZL} \dots\dots\dots 4.8$$

Where F_x and F_y denote creep forces in the longitudinal and lateral directions, where as M_z stands for spin moment. Moreover S_{xL}, S_{yL} and S_{zL} are the corresponding creepage in longitudinal, lateral and angular. The longitudinal and lateral creepages are defined by equations. The RMS value of the wear number in the leading outer wheel is the parameter used to quantify the wear validating function is given by equation.

$$\Gamma_w = \sqrt{\frac{1}{t_f - t_o} \int_{t_o}^{t_f} (F_x \cdot S_{xL} + F_y \cdot S_{yL})^2 dt} \dots \dots \dots 4.9$$

According to [Pearce and Sherratt (1991)] [23] the objective function illustrated as classified as

Table 4.3 Categories for the Wear number.

Wear categories	description
Wear number < 100	Low
$100 \leq \text{wear number} \leq 200$	Medium
Wear Number ≥ 200	High

Chapter 5 Result, Discussion and Validation

5.1 result

5.1.1 Lateral, Vertical force and Derailment coefficient

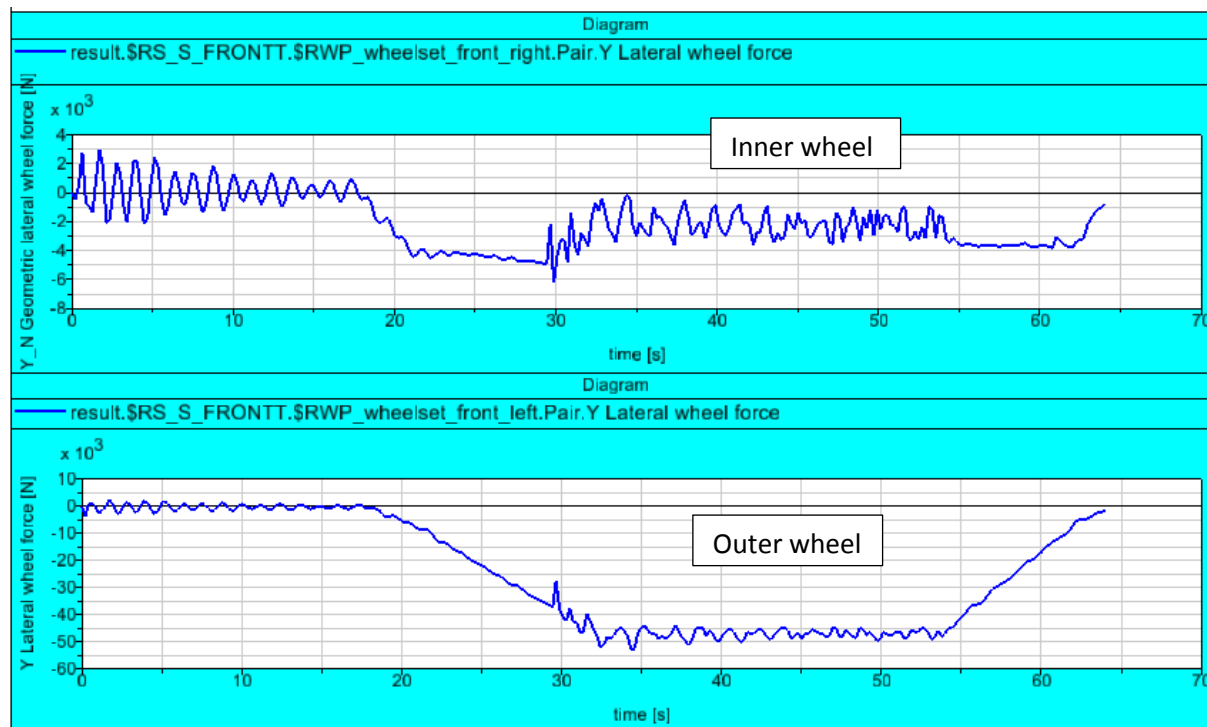


Figure 5.1 Lateral force distributions for inner wheel and outer wheel

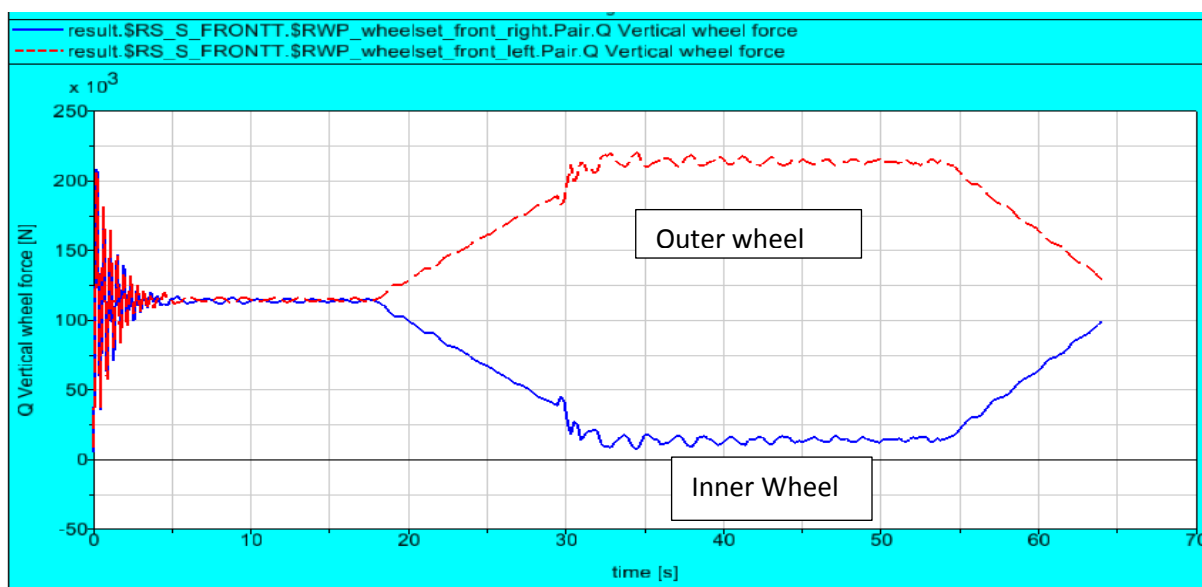


Figure 5.2 vertical force distributions for inner wheel and outer wheel

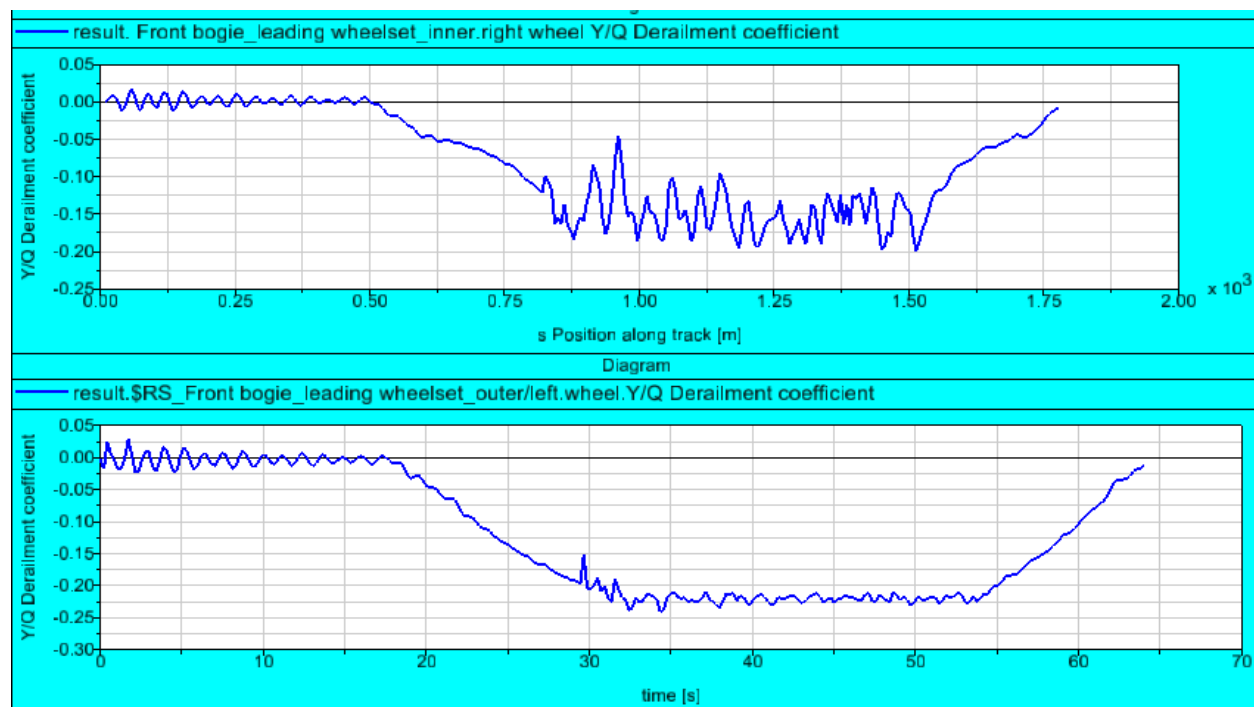


Fig 5.3 Derailment coefficient inner (top) and outer (bottom) front leading wheelset front bogie

5.1.2 Wear index result

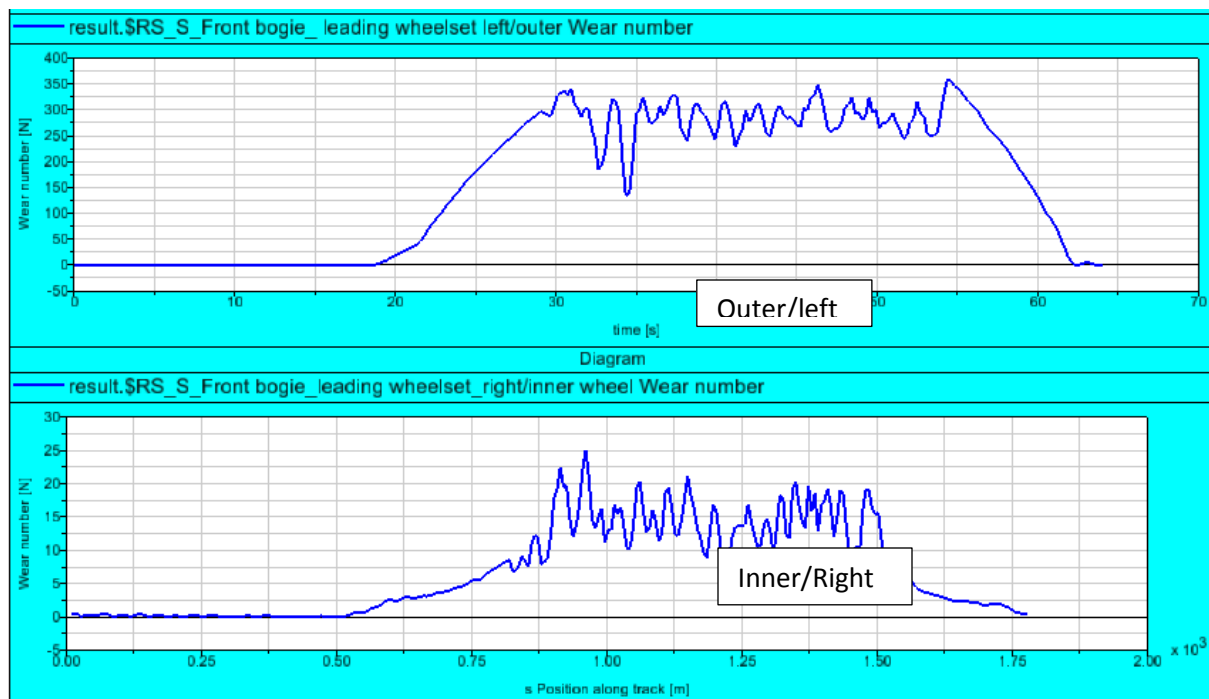


Fig 5.4 wear index inner and outer wheel, for leading wheelset, front bogie1

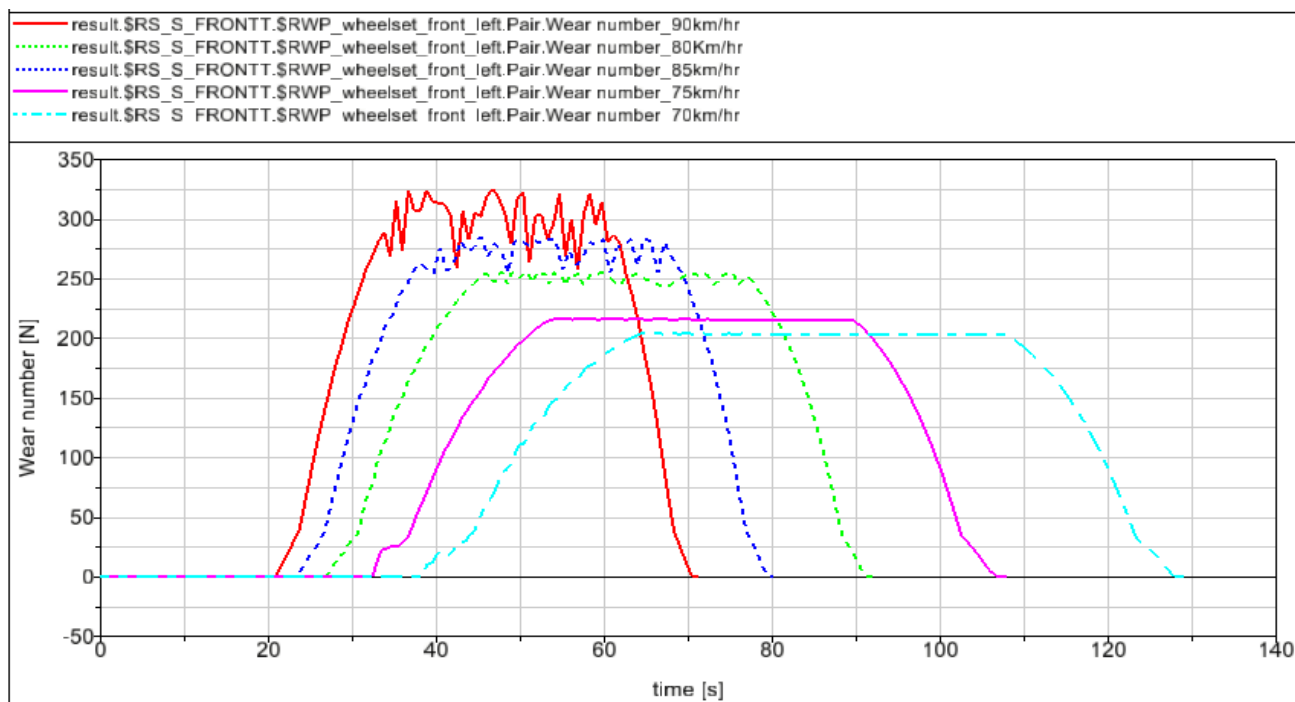


Fig 5.5 wear index for different speed

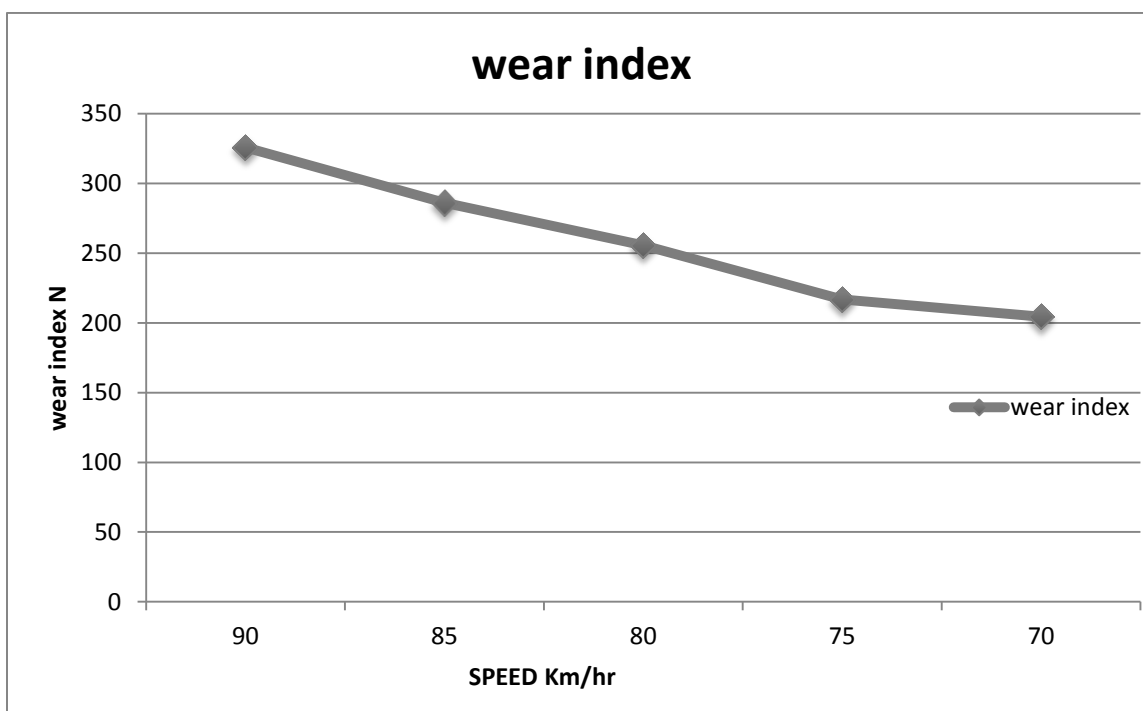


Fig 5.6 wear index for different speed

5.1.3 Lateral track shift force

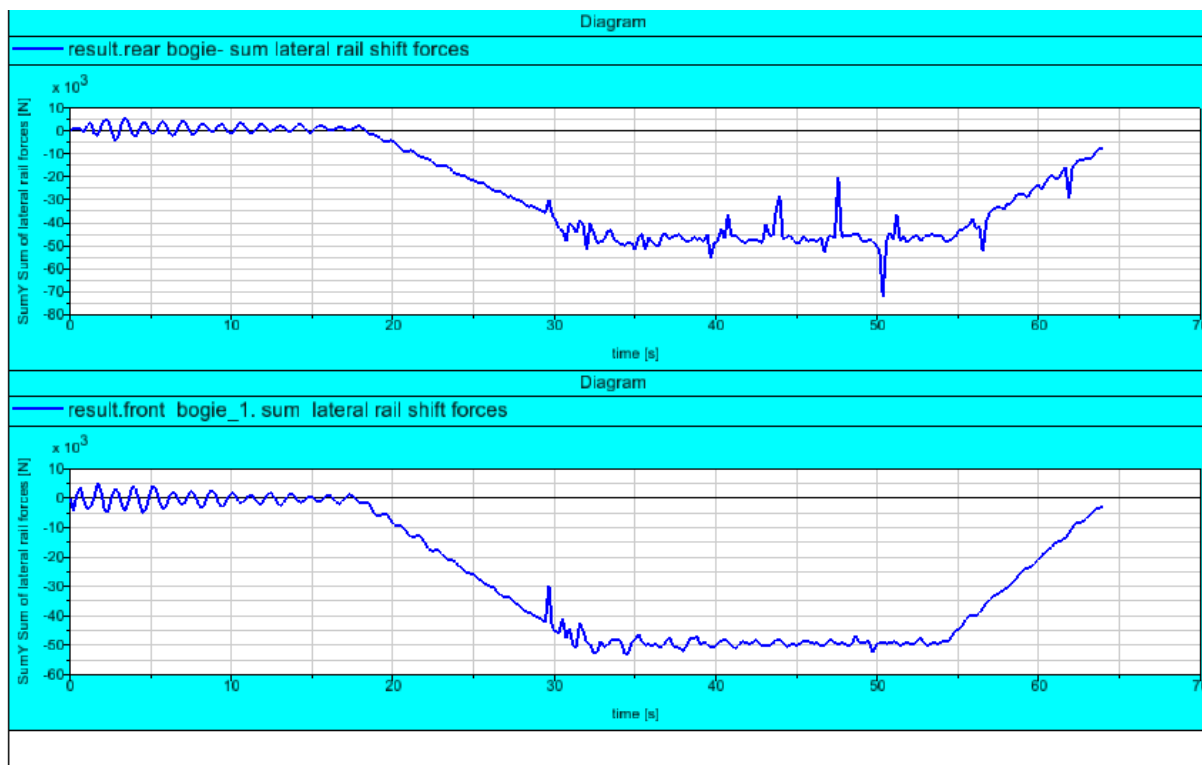


Fig 5.7 Lateral track shift force front bogie (top) and rear bogie (bottom)

5.1.4 Ride Quality index and Acceleration

a. Acceleration

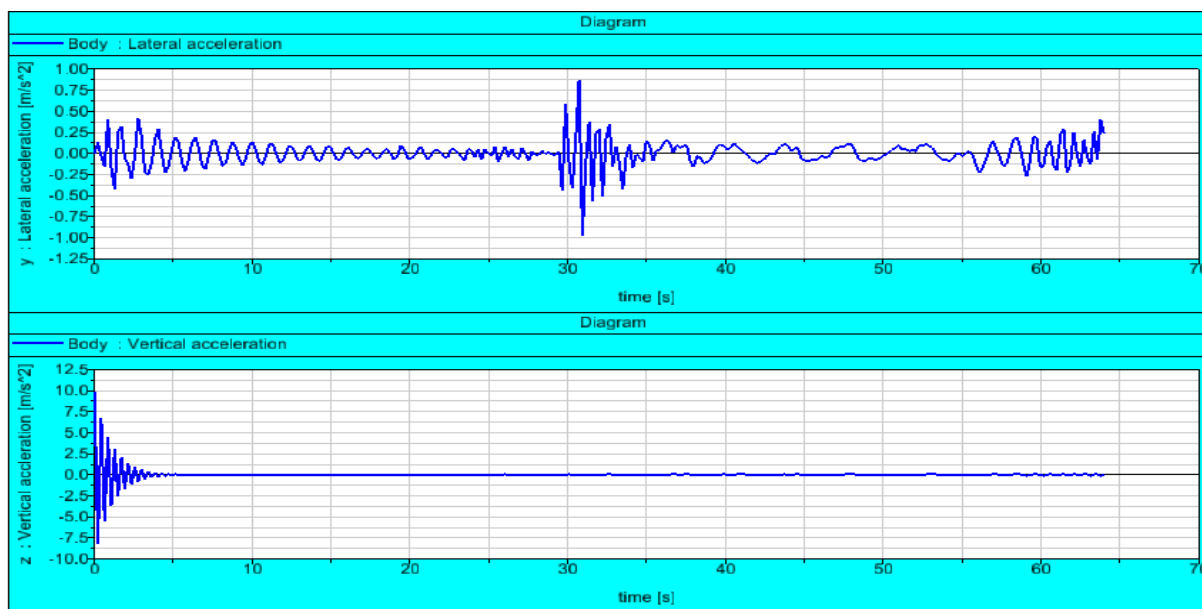


Figure 5.8 lateral acceleration (top) and vertical acceleration (bottom)

b. Ride quality index

Lateral and vertical acceleration after filtered by Wz (ride index) as per ISO 2631

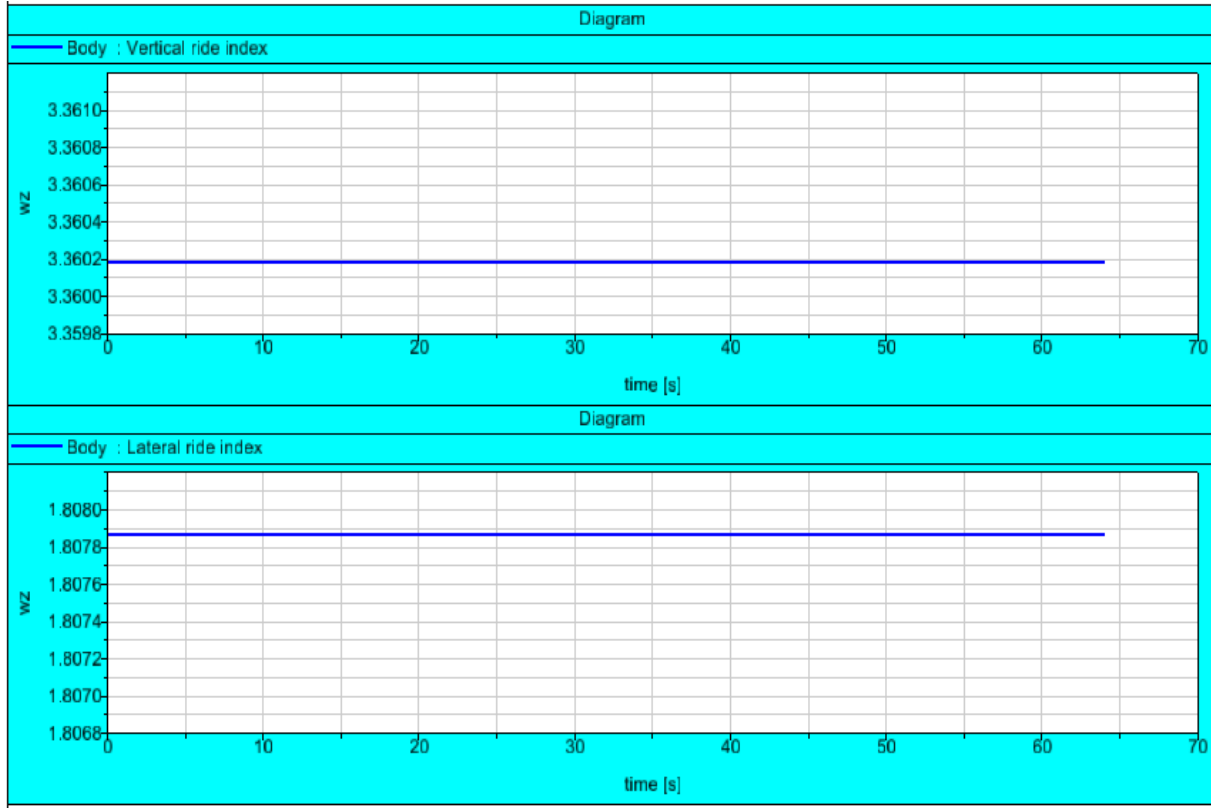


Fig 5.9 mean ride quality index for lateral and vertical motion

5.2 Validation and Discussion

Derailment

Wheelset derailment is one of the most dangerous occurrences affecting the safety of railway vehicle. Figure 5.3 shows front bogie leading wheelset and rear bogie wheelset derailment coefficient (Y/Q) for both left/outer wheel and right/inner wheel. On the straight track between $0 \leq t \leq 17.7s$ derailment indexes show insignificant. When the vehicle enter to intermediate arc ($17.7 \leq t \leq 32s$) time integrations, the derailment become rise because of the effect of quasi-static centrifugal force. And when it reach circular curve $32 \leq t \leq 53.3s$ derailment index reach maximum value and reduced to zero for the subsequent intermediate track between $53.3 \leq t \leq 64sec$.

The Nadal's criterion treats wheel-climb derailment for normal driving using lateral-to-vertical force limit, Nadal wheel-climbing or derailment occur in the situation where wheel experiences high lateral force combined with reduced vertical load, As illustrated in figure 5.2 during the negotiating curve due to quasi-static centrifugal force decrease normal force on inner rail and increase on the outer rail [34,12], the quasi-static force enhances the lateral (horizontal) pressure on outer rail well as shown in figure 5.2. These phenomena particularly in the front bogie leading wheelset created increase the propensity for the left hand wheel to climb over the rail.

Since the center of gravity loaded wagon located half way of its height the structure more reveal to centrifugal force and reduced stability to tilts wheel to certain attack angle β Note that both the wheel L/V ratio limit and distance limit will converge to a constant value as the wheelset angle of attack reaches a certain level [12].

From simulation maximum $(Y/Q) = 0.240889$ at outer wheel leading wheelset of rear bogie obtained. This obtained derailment coefficient as compared with the limiting value $Y/Q = 0.8$ set by UIC 518, the results validated in acceptable range. Which means the probability of wheel climb or derailment would be less likely to occur.

Lateral shift force

The value of the lateral track shift forces obtained from simulation illustrated in fig 5.7. On straight track between $17.7 \leq t \leq 32s$ time intervals the lateral force induced by the wagon somehow insignificant, when it negotiating intermediate track $17.7 \leq t \leq 32s$ and circular track $32 \leq t \leq 53.3s$ the track shift force become increased significantly and reached peak due to centrifugal pulling the vehicle outward. Finally the index returned to equilibrium for the next intermediate track $53.3 \leq t \leq 64s$. The maximum track shift obtained from the result $\sum F_y = 72438.5N$ on integration time = 52 sec, which is exactly at the end of curved track due high impacts load between wheel flange and rail.

Regarding the validation as per CEN/UIC calculated limiting track shift lateral force $\sum F_{y_{lim}} = 76037.5N$ Thus the Simpack result as it compare with this allowable limit, the track shift force that quantify the safety of rail wagon validated that in acceptable range.

Ride quality index

The vertical acceleration response of the car-body at Plots in figure 5.8 shows that, initially the value of acceleration is nearly $a_z = 9.81m/s^2$, which is mainly the acceleration due to gravity. Finally it goes to zero, when the vibration of the car body ceases and it became stable [18].

Lateral acceleration in accordance with figure 5.4 was noticed that during tangent track simulation, in which the model starts movement with insignificant acceleration, the contact forces are small. However it was found that at time integration 32 second, when the wagon enters the circular curve at the constant velocity, lateral acceleration increases reach peak value $a_y = 0.98m/s^2$. It caused by tilting of the structure resulting from the centrifugal force.

Comfort Index has been evaluated through filtering vertical and lateral acceleration by WZ ride 'quality index ISO-2631 in SIMPACK post processor.

- The result of maximum ride quality index based the acceleration value at C.G of body found that; Wz (*Wertungszahl*) =2.5769 for vertical and $WZ = 2.1786$ lateral direction.
- According to chapter four tables 4.1 the ride index obtained through filtering with ISO 2631 it is less than upper limit 3.75, therefore as per result found the wagon riding performance validated that it's in acceptable or good ride quality.

Wear index

Literally as described in equation 4.8, the wear index quantifies the energy dissipation or loss per unit meter of travelled at wheel-rail contact that results due to of creep force vector and creep vector in the contact patch. This parameter expresses the degree of wheel-rail wear under vertical and lateral dynamic forces. If the resulted number is high, shows that the rail wear is high, indirectly this number predict the future state of the track maintenance[25,29] Having consider this fact the diagram in figure 5.5 shows wears index result on outer/left and inner/right wheel for leading front bogie wheel set. The obtained simulation result discussed as, during the wagon ride on the 500meter straight track or between time integration intervals $0 \leq t \leq 17.7s$ the energy dissipation due to creep and creep force somehow negligible, therefore the wear index value obtained somehow small. But during the wagon enters the intermediate canted curve with

length of 400 meter or time integration $17.7 \leq t \leq 32$ s the index value became rise and reach pick value during curved 600 meter length track or between time integration $32 \leq t \leq 53.3$ sec. this due to reason that during curve negotiation the distance travelled by the left/outer relatively higher than inner/right wheel therefore to compensate this phenomena the wheel set slide relative to rail thus the so called creep age induced that had adverse effect on boosting wear index value.

Furthermore as illustrate the same figure 5.5, significant wear index difference between the outer and inner wheel recognized. This is explained by the fact that when wagon negotiate curve the wheel tread conicity is not sufficient to ensure the wheelset steering without flange contact . Therefore on the left wheel outer wheel it is clear that the; double contact at the wheel tread & flange and at the rail head & rail gauge corner occurrences [29] this result a high wear rate on left wheel compared to right single wheel rail contact.

Considering validation, the maximum wear index $\Gamma_w = 367.45$ found at time = 54.4015 sec for front bogie outer leading wheels. As per the limiting standard mentioned in chapter 4 section 4.4, $\Gamma_w \geq 200$ categorized as high [25]. Therefore the result found as compared validation standard, the tendency of leading outer wheel to become severely wear, and also occurrence mild wear is high.

To reduce this maximum wear index and maintain medium wear rate, a optimization study has been conducted i.e. as illustrate in figure 5.6 and 5.7 different speed interval has been considered and further evaluation and suggestion made on corresponding wear index. The obtained result particularly on curved track indicates a reduction speed had linearly correlated with wear index reduction. Since the main intention this comparative analysis is to obtain and suggests optimal operating speed that results a least possible wear rate. Therefore during the wagon run predominantly on curved track as per specified track data it has been suggested that to optimize the high wear rate rather than 100 km/hr as per ERC data, if we use 70km/hr wagon speed the wear index value can be significantly reduced from $\Gamma_w = 367.45$ Joule/m(N) to $\Gamma_w = 200$ Joule/m(N).

Chapter6. Conclusion, Recommendation and Future work

6.1 Conclusion

The paper presents dynamic performance simulation using SIMPACK and validation of Ethio-Djibouti container carrier freight wagon on 1.8 km length of continuous tangent, intermediate and circular track. The dynamics simulation conducted at 100Km/hr and 93 ton. The wagon dynamic performance regarding derailment factor, track shift force that quantify safety related parameter and performance parameter which regards to ride quality index and wear index had been analyzed. The result found from SIMPACK has been validated based on UIC 518 and CEN (2005) objective function.

- The maximum derailment ratio found from the simulation was exactly 0.2408 as it compare with maximum CEN derailment coefficient 0.8 the wagon safety regards to derailment accident validated in safe range.
- Regarding maximum lateral track shift force $\sum Fy = 72,438N$ result obtained from simulation, when it compared with admissible lateral track shift force limit by CEN; i.e. $\sum Fy_{limiting} = 76037.5N$ it is been found in safe range. Therefore it has been conclude the tendency induced wheel lateral shift force causing lateral irregularity closely below the limit.
- $\Gamma w = 367.45$ Joule/m (N) wear index value result obtained from simulation. The obtained result as it compare with standard limiting value it has been found high. This indicates when the wagon negating curved track the probability of two-point contact created high dissipation frictional energy.
- A reduction speed had almost linear correlation with wear index reduction. Therefore during the wagon run predominantly on curved it has been concluded that to optimize the high wear rate; rather than 100 km/hr as per ERC data, if we use 70km/hr wagon speed the wear index value can be significantly reduced from $\Gamma w = 367.45$ Joule/m(N) to $\Gamma w = 200$ Joule/m(N).
- It has been concluded that, the ride quality index obtained for specified operation condition validated in acceptable range.

6.2 Recommendation

The recommendation the study described as:

- Even though the result found for derailment risk and ride quality index below the limit at curved track, the wear index value that predict the wear rate found high excessive. Therefore to alleviate this problem optimized equilibrium wagon speed at curved and junction between intermediate & circular track shall be maintained.
- Additionally the lateral track shift force found from result almost near below allowable limit. Therefore after certain period of time this result might boost and therefore particularly at curved root path the track shall be stiffer to accommodate the induced track shift force.
- Predominantly at curved track with radius 350m according to track data tables 3.5 It has been recommended that to optimize the high wear rate; rather than 100 km/hr as per ERC data, if we use 70km/hr wagon speed the wear index value can be significantly reduced
- Interchanging of inner rail and outer rail can be considered as a means to lengthen the service life of the rail at horizontal curved track since the high side wear occur at outer rail.
- Curved rail geometry requires more attention than that of tangent track due to an additional centrifugal force.

6.3 Future work

The development of advanced railway vehicles is a complex research field that requires new ideas and novel design solutions. So the future work in the field of railway dynamics will not finish comparing with the large challenges can be faced by the research efforts in the analyzing and enhancing rail vehicle dynamic performance. But the future work proposed by the end of this work for the improvement of the vehicle models and enhancement methodologies, can be summarized in the following points

- The rail vehicle model used under this study comprises a rigid car body. However; in order to get more accurate simulation results using more advanced vehicle model with flexible car body instead of rigid body can be another point of interest.

- Track parameters such as track irregularity and vertical elevation are not considered in the current study therefore, by considering actual those parameter studying vehicle dynamic behavior is another engineering problem which needs to be study
- Finding optimization solutions to improve the wagon running performance and safety have inevitable significance to enhancing transport quality and reduce maintenance cost. Therefore by link simpack and matlab(Simulink) interface further optimization of performance dynamics, is another research area need to be emphasize.
- Analyzing wagon dynamic stability and finding a critical speed which hunting will occur is another point of research area need to be addressing in future study.

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Appendix A Wagon modeling Data [19, 20]

A.1 Rigid body design data

Table A.1 rigid body mass, CG and moment of inertia input

Rigid body	Mass(kg)	Center of gravity Relative to BRF	Area moments of inertia [Kg m ²] With respect to center of gravity		
			Ixx	Iyy	Izz
Wheel set m_w	1200	0,0,0	486.404	4.809	486.404
Bogie frame m_b	1200	1.2, -0.9, 0.2853	1010.8007	355.8432	1254.5419
Bolster m_{bol}	1000	0.9222,0,-0.58687	462.7725	38.7993	479.9125
Wagon frame m_{cf}	13800	5.2763,0,-1	8,371.1821	164,437.519	172,182.8201
Container 1	35000	2.05158,0, -2.4967	64,360.6945	172,262.5326	170,107.495
Container 2	35000	8.4935,0,-2.4967	64,360.6945	172,262.5326	170,107.495

A.2 Suspension design data [31, 20]

Table A.2 suspension primary and secondary force input

component	Stiffness type	Slip coefficie nt friction	Stick coefficient friction	Damping (Ns/m)			Stiffness (N/m)		
				dx	dy	dz	cx	cy	cz
Adapter plus	194 Stick- Slip 2D Cmp	0.2	0.3						
Bearing	5 Spring- Damper Parallel			1e4	1e4	2.5e4	8e5	8e5	6.25e6

	Cmp								
Primary stiffness									
Frictional wedge		0.2	0.3						
Torsional spring	79 Shear	cp	cr				1.6e	1.6e5	4.3e6
	Spring Cmp	1.05e5	1.05e5				5		
Vertical damper	6 Spring-Damper Serial PtP	-	-	Corresponding to fig 1.1			0	0	6e6
Secondary stiffness									
Center plate	5 Spring-Damper Parallel Cmp			1e4	1e4	5.5e4	8e5	8e5	6.25e6
Side bearing	194 Stick-Slip 2D Cmp	0.2	0.3	1e4	1e4	5.5e4	1e6	1e6	5e6
Track da									

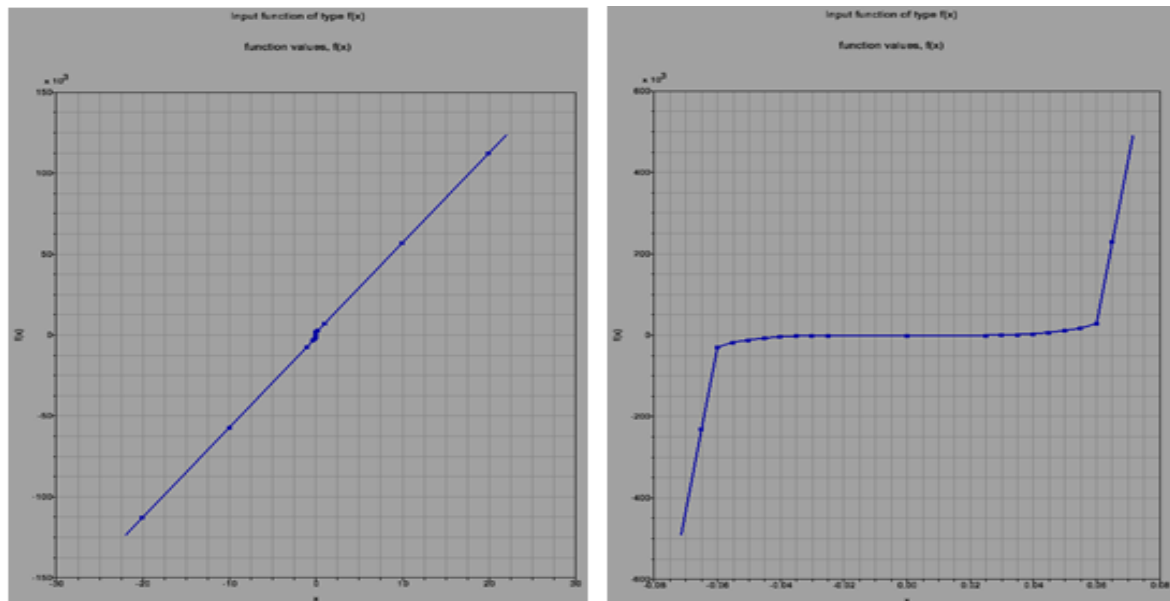


Fig A.1 nonlinear damper input

Appendix B

B.1 Geometry and modelling parameters of the wheel[19,33]

Table B.1 wheel technical data

Wheel parameter	description
Wheel type	S1002
Modulus of elasticity of wheel material(N/m ²)	2.10 e11
Poisson's ratio of wheel material	0.28
Nominal radius(m)	0.42
Semi distance(m)	0.75

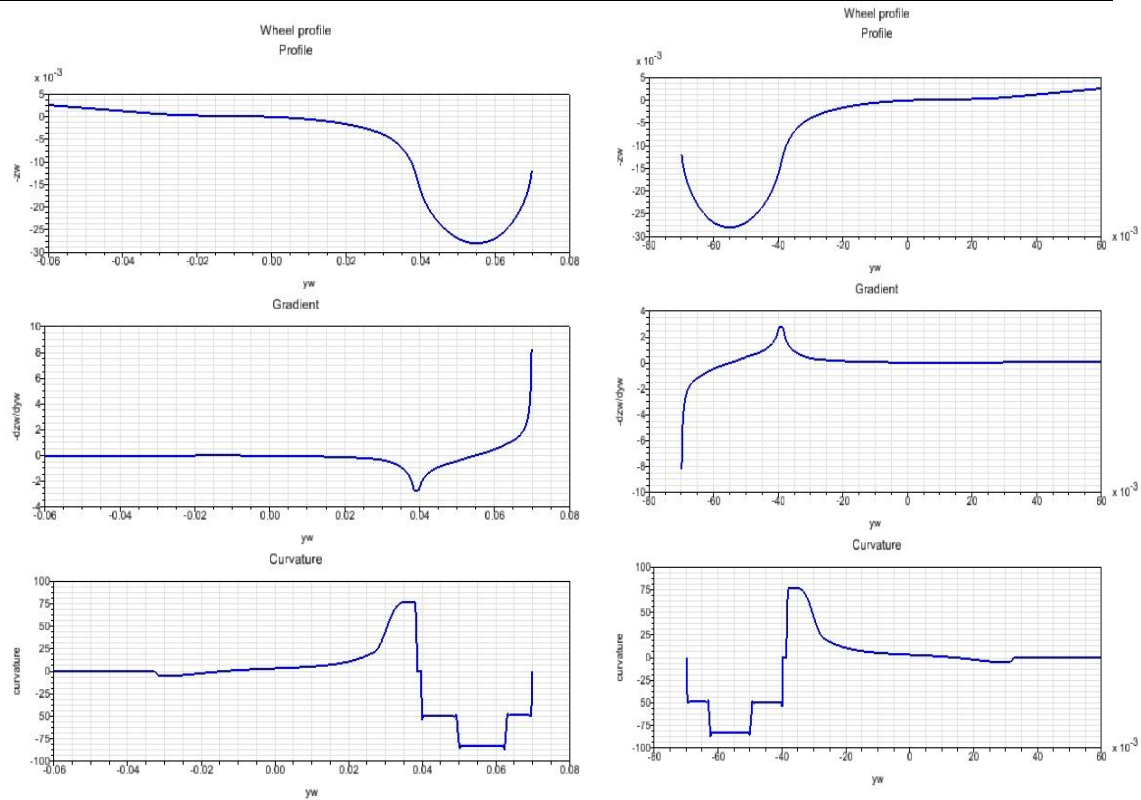


Fig B.1 wheel profile and curvature

Appendix C. Geometry profile and rail modelling parameter [19, 32]

Table C.1 rail modeling data

Rail parameter	description
Rail type	UIC 60
Modulus of elasticity of rail material(N/m ²)	2.10e11
Poisson's ratio of rail material	0.28
Rail cant	1: 20
Lateral rail distance(m)	0.75

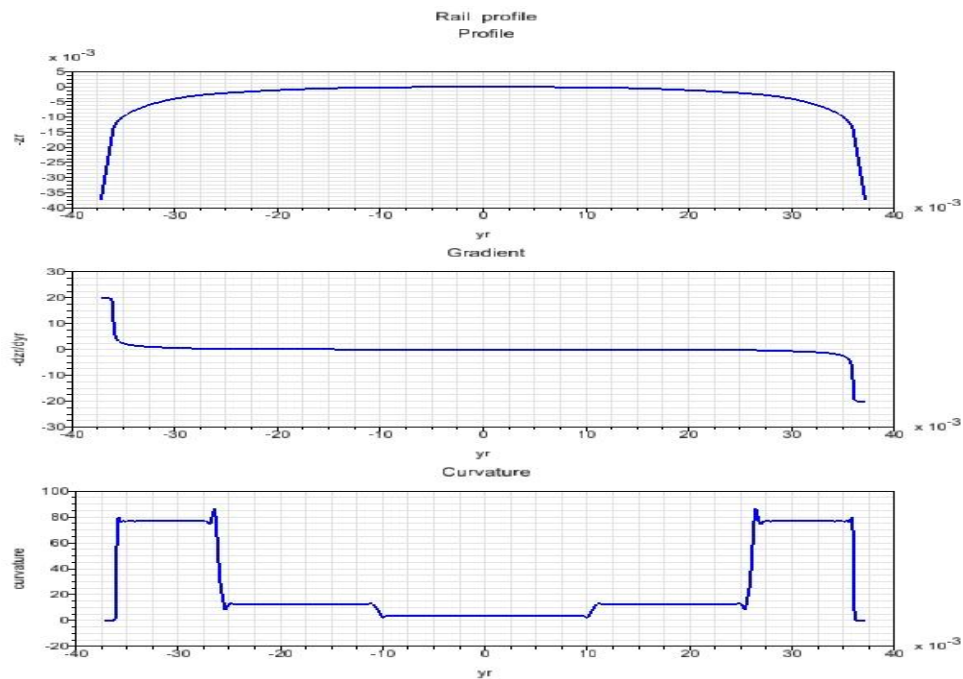


Fig C.1 Rail profile and curvature

Appendix D 3d modelling of wagon fright part in Catia interface

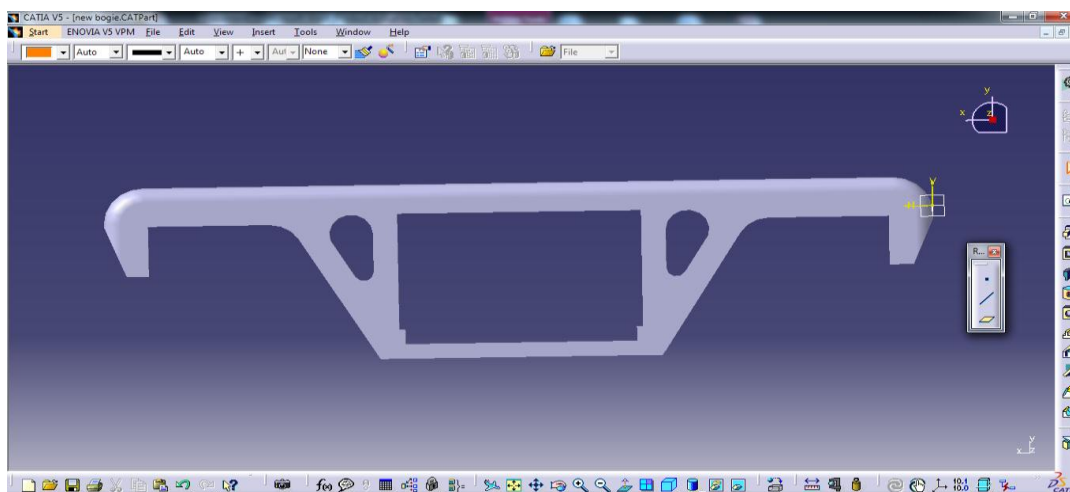


Fig D.1 bogie frame drawing

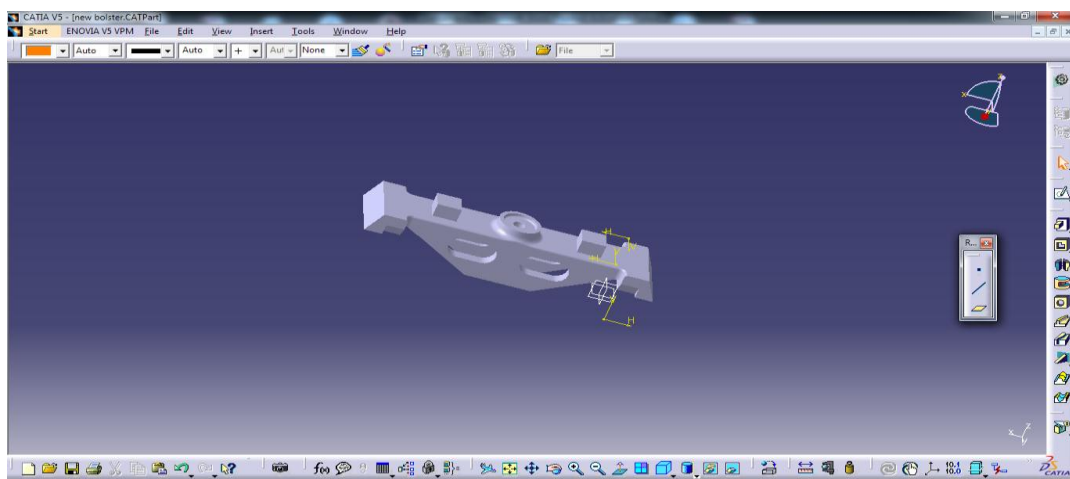


Fig D.2 bolster 3D drawing

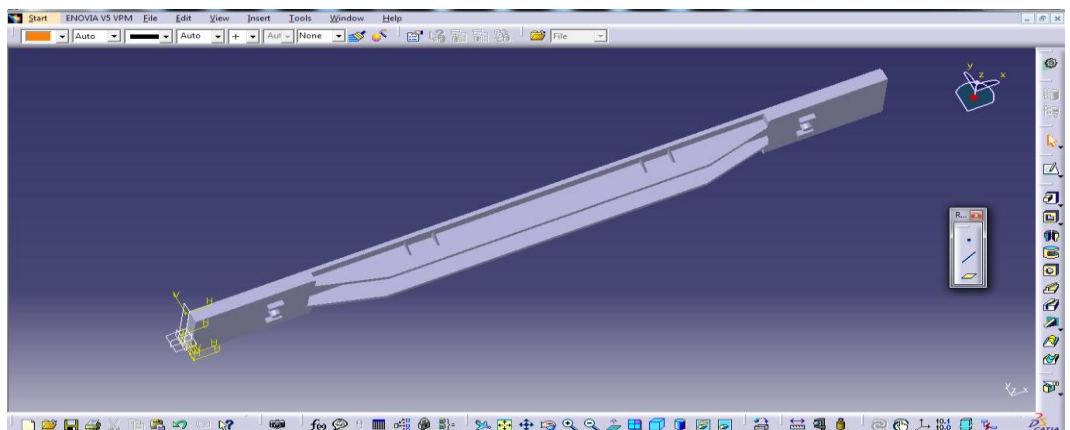


Fig D.3 wagon frame 3D drawing

Appendix E Kalker coefficient

Table E.1 Kalker coefficient [12]

g	c_{11}			c_{22}			$c_{23} = -c_{32}$			c_{33}		
	$\nu = 0$	0.25	0.5	$\nu = 0$	0.25	0.5	$\nu = 0$	0.25	0.5	$\nu = 0$	0.25	0.5
<i>(a/b)</i>												
0.1	2.51	3.31	4.85	2.51	2.52	2.53	0.334	0.473	0.731	6.42	8.28	11.7
0.2	2.59	3.37	4.81	2.59	2.63	2.66	0.483	0.603	0.809	3.46	4.27	5.66
0.3	2.68	3.44	4.8	2.68	2.75	2.81	0.607	0.715	0.889	2.49	2.96	3.72
0.4	2.78	3.53	4.82	2.78	2.88	2.98	0.720	0.823	0.977	2.02	2.32	2.77
0.5	2.88	3.62	4.83	2.88	3.01	3.14	0.827	0.929	1.07	1.74	1.93	2.22
0.6	2.98	6.72	4.91	2.98	3.14	33.1	0.930	1.03	1.18	1.56	1.68	1.86
0.7	3.09	3.81	4.97	3.09	3.28	3.48	1.03	1.14	1.29	1.43	1.50	1.60
0.8	3.19	3.91	5.05	3.19	3.41	3.65	1.13	1.25	1.40	1.34	1.37	1.42
0.9	3.29	4.01	5.12	3.29	3.54	3.82	1.23	1.36	1.51	1.27	1.27	1.27
<i>(b/a)</i>												
1.0	3.4	4.12	5.2	3.40	3.67	3.98	1.33	1.47	1.63	1.21	1.19	1.16
0.9	3.51	4.22	5.3	3.51	3.81	4.16	1.44	1.57	1.77	1.16	1.11	1.06
0.8	3.65	4.36	5.42	3.65	3.99	4.39	1.58	1.75	1.94	1.10	1.04	0.954
0.7	3.82	4.54	5.58	3.82	4.21	4.67	1.76	1.95	2.18	1.05	0.965	0.852
0.6	4.06	4.78	5.8	4.06	4.50	5.04	2.01	2.23	2.50	1.01	0.892	0.751
0.5	4.37	5.10	6.11	4.37	4.90	5.56	2.35	2.62	2.96	0.958	0.819	0.650
0.4	4.84	5.57	6.57	4.84	5.48	6.31	2.88	3.24	3.70	0.912	0.747	0.549
0.3	5.57	6.34	7.34	5.57	6.40	7.51	3.79	4.32	5.01	0.868	0.674	0.446
0.2	6.96	7.78	8.82	6.96	8.14	9.79	5.72	6.63	7.89	0.828	0.601	0.341
0.1	10.7	11.7	12.9	10.7	12.8	16.0	12.2	14.6	18.0	0.795	0.526	0.228

Note: $g = 0$, $c_{11} = \pi^2/4(1 - \nu)$; $c_{22} = \pi^2/4$; $c_{23} = -c_{32} = \pi\sqrt{g} \{1 + \nu(0.5\Lambda + \ln 4 - 5)\}/3(1 - \nu)$; $\Lambda = \ln(16/g^2)$; and $c_{33} = \pi^2/16(1 - \nu)g$.