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ADDIS ABABA INSTITUTE OF TECHNOLOGY

SCHOOL OF MULTIDECIPLINARY ENGINEERING

GRADUATE PROGRAM IN RAILWAY ENGINEERING

OPTIMIZING THE VERTICAL VIBRATION ON RAIL VEHICLE DYNAMIC FOR PASSENGER COMFORT

By

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A Thesis submitted to the School of Graduate Studies of Addis Ababa University in partial fulfillment of the requirements for the Degree of Masters of Science in Mechanical Engineering

(Railway Engineering)

Advisor:- Dr . Ing. Demiss Alemu

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Addis Ababa Institute of Technology (AAiT)

School of Mechanical & Industrial Engineering

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DECLARATION

I hereby declare that the work which is being presented in this thesis entitled "**OPTIMIZING THE VERTICAL VIBRATION ON RAIL VEHICLE DYNAMIC FOR PASSENGER COMFORT**" is original work of my own, has not been presented for a degree of any other university and all the resource materials used for this thesis have been duly acknowledged.

__________________________ _______________

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This is to certify that the above declaration made by the candidate is correct to the best of my knowledge.

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Dr. Ing. Demiss Alemu (Advisor) Date

ABSTRACT

Investigation of vibration is an important topic for the purposes of ride comfort in railway engineering. The vibration of rail vehicles becomes very complex because it is affected by the condition of vehicles, including suspensions and wheel profile, condition of track sections, including rail profile, rail irregularities, cant and curvature. This thesis aims at optimizing validated modelling methods to study its dynamics, how it influences on vertical vibration, how it excited on track and how it interacts with the passengers. The primary interest is ride comfort, considering the standard vibrations frequency range. In the standard frequency range, the structural flexibility of the carbody is concern. The models are intended for use in time-domain simulation, calling for small-sized models to reduce computational time and costs. Key parameters are proposed to select carbody eignmodes for inclusion in a flexible multibody model.

Numerical simulations can be used to evaluate the comfort index at the railway vehicles. To this purpose, a melty-body modelling system is used that takes into account a string of factors influencing the level of the vertical vibrations, thus allowing a correct evaluation of the dynamic behavior in the railway vehicles, mainly at high velocities. The influence of the vertical suspension features upon the vibrating comfort will be examined, as a function of velocity. The concept of the critical point of the rail vehicle vibration behavior is introduced, as the point where the comfort index is the highest. While assuming the idea to minimize the level of vibrations in the critical point, it is thus proven the possibility of establishing the best damping of the vertical suspension that leads to the best values of the comfort index.

Finally, the effects of different parameters and the optimized parameters are carried.

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NOMENCLATURE

A	Track excitation intensity coefficient
a_b, a_c	Axle bases of bogies and carbody
a^{wrms}	Frequency-weighted acceleration
$B(f)$	Factor of acceleration evaluation
c_b	Primary damping coefficient per axle
c_c	Secondary damping coefficient per bogie
E	Elasticity modulus
f	Null response frequency
$G_c(x,w)$	Shaping filter of the track vertical unevenness
$G_c(f)$	Power spectral density of acceleration
$H_c\ (x,\omega)$	Carbody frequency response
I	Beam moment of inertia
k_{zb}	Primary spring stiffness per axle
k_{zc}	Secondary spring stiffness per bogie
K	Stiffness matrix
l_1 , l_2	The positions where the secondary suspension forces act referred to the carbody left end
L	Carbody length
L_r & H	Track irregularity wavelength
m_c, m_b	Mass of carbody and bogie frame
M	Inertia matrix
n	Integer number
$qi(t)$	Modal coordinate
t	Time variable
V	Vehicle running speed
$Yi\ (x)$	Mode shape function
z_c , z_{b1} , z_{b2}	Carbody, bogie 1 and bogie 2 vertical displacement
η_1 , η_3 ,	Track vertical irregularities experienced

η_2 , η_4 ,	By wheelset respectively
$\beta 1$, $\beta 2$, $\beta 3$	Phase lags with respect to the first wheelset
$\delta(x)$	Dirac delta function
θ_c , θ_{b1} , θ_{b2}	Carbody, bogie 1 and bogie 2 pitch angle
μ	Internal damping coefficient
ξb , ξc	First and second bending mode damping ratio
ω	Angular frequency
$\omega 3$, $\omega 4$	First and second bending mode frequency
Ωc , Ωr	Cut-off frequencies

CHAPTER ONE

INTRODUCTION

1.1. Background

Rail transport is one of the major modes of transportation, so it must offer a high comfort level for passengers and crew. However, the comfort that passengers experience is usually perceived differently from one individual to another. A low vibration level is one of the important factors of a good ride comfort. The vibrations are mainly caused by track irregularities, from which they are transmitted via the bogies and the carbody to the passengers. The carbody is not rigid, but bends and twists from the excitation coming from the bogies. In some cases, this carbody structural flexibility accounts for half of the perceived vibrations, the rest being due to rigid body motions. The current trend towards lighter vehicles and higher speeds makes the issue of vehicle vibration crucial in the design and development of competitive vehicles. Also, the demand for a high comfort standard calls for a better understanding of the passenger-vehicle vibration interaction.

The first international standard on ride comfort and whole body vibrations was published in 1975 [18], but already in 1941 [18] “equivalent comfort contour” curves for seated persons vibrated in the frequency range of 1 to 12 Hz were produced. These curves were developed to become the “Wertungsziffer” (Wz), which, for a long time, was widely used by European railways. The current standard is a modified ISO-2631 from 1997 [6]. There are, however, differing opinions, in particular on the “weighting filters” prescribed by this standard and a new standard is proposed [18]. Traditionally, only frequencies up to 20 Hz have been considered, but current standards also take higher frequencies into account, up to 80 Hz [10].

Engineers have two tools to assess vibrations: measurements and simulations. Measurements are, however, not possible during the design phase of a new vehicle. Here, simulations based on rail vehicle dynamics models offer a possibility to predict ride comfort, but, in order to be reliable, the simulation models must be validated. There is, therefore, a need for validated methods in analyzing and modelling the structural dynamics of the carbody.

Along the years, many methods were established to assess the ride comfort, with a selective applicability in dependence on the specific of traffic either urban, suburban, or long distance, etc. These methods are included in a series of standards, such as ISO 2631 [8], BS 6841 [9], Index Sperling Ride [10], ENV-12999[12] and UIC 513[13]. All the methods involve the evaluation of the level of vibrations from the comfort perspective based on the acceleration in different directions, via certain weighting filters that trigger the influence of the differentiated sensitivity of the human subjects to vibrations, in dependence on direction and frequency. The method herein – the comfort index – has been adopted by most railway administrations in Europe. The reason is that it represents a high level and an instrument of skill that takes into account a number of specific issues of the railway vehicles vibration behavior, important as they influence the comfort state, but neglected by other methods.

The present work addresses the issue of carbody structural dynamics. It studies the role played by the structural flexibility of the carbody, and, in particular, it aims at developing validated modelling methods to analyses the dynamics of the vehicle vertical vibration and how it is optimized.

Additionally it deals with an issue not talked about much before, i.e. Optimization of the suspension damping upon ride comfort evaluated by the *Wz* index. The complete model of a passenger car has been considered, including the carbody bending vibrations. Upon implementing the technique of modal analysis, the equations of motion are processed differently, intuitively, which points out that the vehicle movement is symmetrical and excitation modes. The damping influence upon the vehicle frequency response is analyzed. And the impact of suspension damping upon the ride comfort is seen in terms of velocity, bogie axle base and carbody rigidity. The geometric filtering effect is under study, an effect due to the distance between wheelsets and its influence upon the comfort index, not dealt with before.

1.2. Statement of the problem

The ride comfort is a complex notion and represents an important criterion while examining the dynamics of the railway vehicles and needs to be taken into account for their modeling and behavior evaluation. Among other factors, the ride comfort strongly depends on the vibration behavior of the vehicle. The increase of the velocities will trigger the amplification of the vibration behavior at the car body level, with a negative impact on the rolling comfort. This condition are the most common problems in rail vehicle including Addis Ababa LRT. Hence, maintaining the vibration behavior at a level that will provide comfort to the passengers requires permanent adjustments in the vehicle construction. Therefore the problem areas that were studied in this paper are: primary suspension dumping parameter, Secondary suspension dumping parameter, wheel base geometry and carbody stiffness of A.A LRT. Accordingly this problem the paper studded stable solution for passenger comfort by optimizing of the suspension characteristics and vehicle construction.

1.3 Objective of the research

1.3.1 Main objective

The main objective of this thesis is optimizing the vertical vibration on rail vehicle dynamic for passenger comfort

1.3.2 Specific objectives

- Study multi-body system dynamics software, SIMPACK and presented a procedure for developing vertical vibration of railway vehicle.
- Determination the optimum solution for different type of vibration behaviors.
- Determining the optimum ride comfort index.
- Determining the optimum dimension and size of the vehicle parameter components using computer simulation.
- Investigation of the primary and secondary suspension damping ratio with the optimum comfort index.
- Determine the optimized wheel base dimension and carbody stiffness.

1.4 Research area (scope)

The overall purpose of our numerical modeling is to determine the optimum comfort for railway passenger by reducing vertical vibration. The vibration of rail vehicles can be occurred vertically or laterally. This vibration affected by suspensions (spring, damping), wheel profile, wheel base dimension, car body stiffness and the condition of track sections, including rail profile, rail irregularities, cant and curvature etc. However, in this thesis only improving passenger comfort (raid index) by optimizing the damping ratio, wheel base size and car body stiffness in the vertical direction will be studied.

1.5 Methodology

1.5.1 Introduction

The research begins by identifying the determining factors of vertical vibration on rail vehicle, raid comfort data were collected and the relationship between those factors have been established to set method to analyze railway raid index. Hence, the following methods were followed.

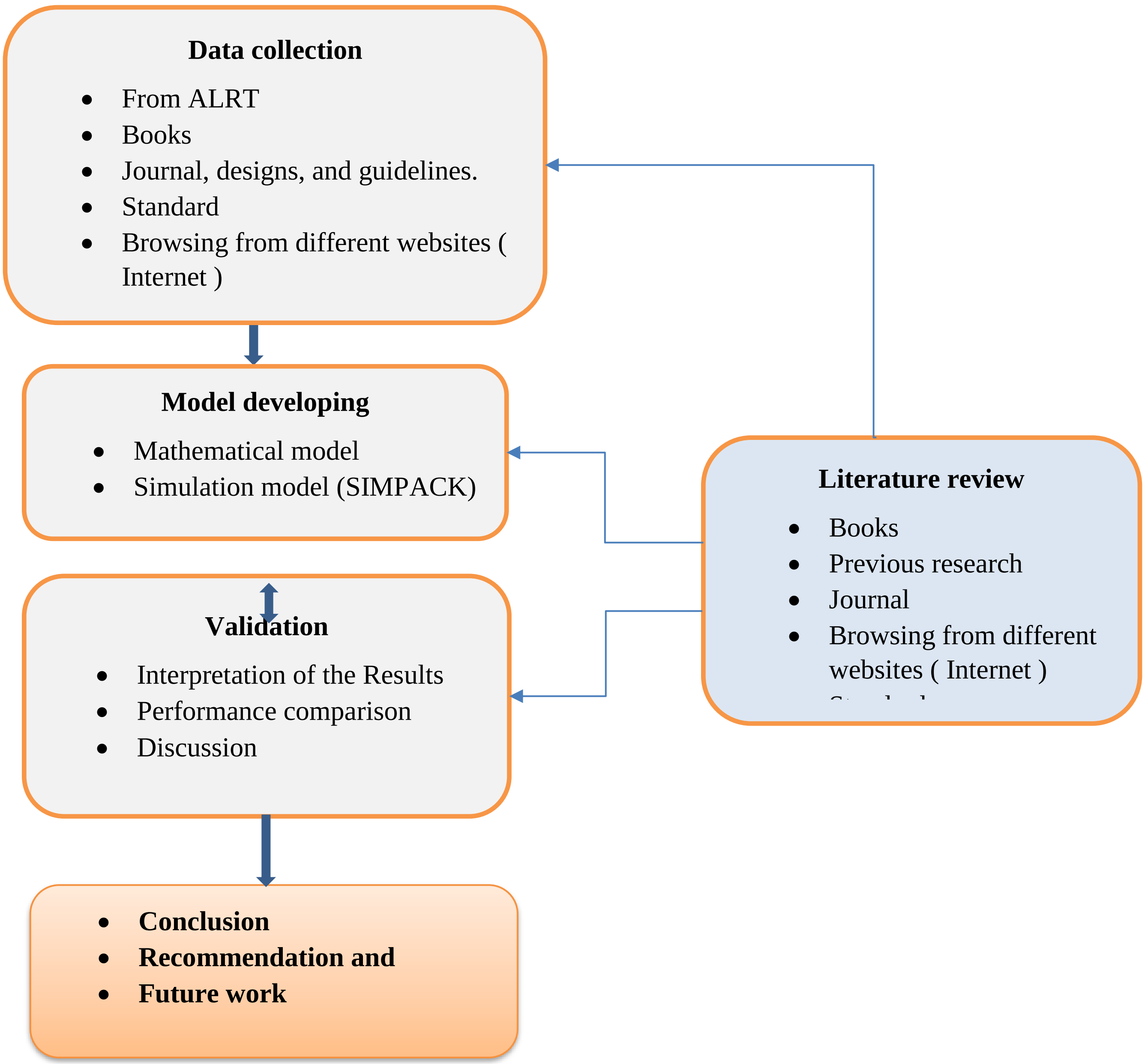

Figure 1: Methodology

1.5.2 Data collection

This paragraph will outline the methods that were used to achieve the objectives and Specific objectives and the primary research designed, implemented and analyzed. Therefore, the primary data collection will be discussed first, followed by the secondary data collection.

A) Primary data collection

The primary data is data which are collected from ERC based on the existing and reality condition. This data were conducted by questionnaire survey. The initial idea was to use structured interviews as research methodology to form a picture of the situation.

B) Secondary data collection

Every research project should include secondary data because secondary data gives on overview of what has been researched before in the same subject area, which will not only help to choose a research topic and place the research in context, but is also crucial for the decision on research design for the own research [32].

The secondary data is data which is collected to support the primary data and as comparison to other studies. The secondary data in this research are obtained from the institution or organizations associated with the research object such as books and proceeding about the vertical vibration and raid quality on rail vehicle . Other sources were articles and websites.

1.5.3 Data analysis

The collected data is analyzed and filtered according to the study needed. Here there are several requirements or factors which are useful to optimize the raid comfort in different researches which are not studied, therefore these are collected and arranged and put to this study.

The collected data will be analyzed and arranged accordingly to give the final outcome of the thesis.

1.5.4 Modeling

Modeling is the key section to optimize the raid comfort of AALRT. Then after analyzed the collected data the rail vehicle vertical vibration and comfort is formulated mathematically and all the constraints are presented following with detailed explanation of all the requirements. Once the data has been defined it is possible to simulation modeling to solve a rail vehicle raid index problem for satisfying all the passenger comfort.

CHAPTER TWO

LITERATURE REVIEW

In order to improve ride comfort, it is important to have knowledge about the different vibration modes of the carbody. The vibrations of a carbody are mainly caused by track irregularities that are transmitted up to the carbody via the bogies. In order to achieve good ride comfort. These vibrations have to be suppressed. Since ride comfort is negatively affected by high accelerations in the carbody. The vertical vehicle vibrations can be divided in two types of modes: rigid and flexible. The rigid body modes that affect vertical ride comfort are bounce, pitch and roll. The flexible modes are twisting and bending deformations of time carbody due to forces acting on it. Typical flexible modes that are interesting when focusing on vertical ride comfort are the first vertical bending mode, first torsion mode (torsional motion along the longitudinal axis) and the first so called breathing mode, resulting from cross-section shear. Normally, the frequencies for the first flexible modes lie in the range of 8-15 Hz [25]. However, depending the vehicle configuration and settings. The higher the Eigen frequency, the more complex time mode shapes. To avoid resonance between carbody and bogie, the lowest possible bending frequency of the carbody is determined by the Eigen frequencies of bounce and first bending mode of the bogie. The human body is most sensitive to vertical vibrations in the frequency range 4 -15 Hz[25]. Therefore, to secure good ride comfort, it is of importance not to let the Eigen frequencies of the carbody coincide with this sensitivity range.

2.1 Carbody Vibration

This topic describes previously investigated methods to reduce vertical vibrations of the carbody, and hence to improve vertical ride comfort. If the carbody is exposed to the same frequencies (or close to) as its natural frequencies, the corresponding eignmodes will be excited. Vertical carbody vibrations can be reduced either by focusing on the structural stiffness of the system or by optimizing the spring and damping components. When an optimum has been reached for a conventional passive damping system, active component can be a solution for further improvements.

Tilt control mentioned in this section is concerned with the roll mode of the carbody. Mainly to reduce lateral accelerations experienced by passengers. However, the vertical mode is to some extent connected to rolling and therefore it is treated in this study.

Carlbom [27,29,30] has performed a comprehensive study on the carbody structural dynamics of a rail vehicle. A Swedish commuter coach is used as case study, where both simulations and measurements are performed. The study examines the structural flexibility of the carbody and what role it plays on ride comfort. Vibrations up to 20 Hz are considered. This is the frequency range where structural flexibility of the carbody has the largest influence. In some cases, the carbody structural flexibility accounts for nearly half of the comfort weighted *rms* values of the vertical vibrations. The mode shapes of the eight lowest flexible modes can be seen in Figure 2. The dark shading corresponds to deformations with small amplitudes. The corresponding undamped eigenfrequencies are listed in Table 1.

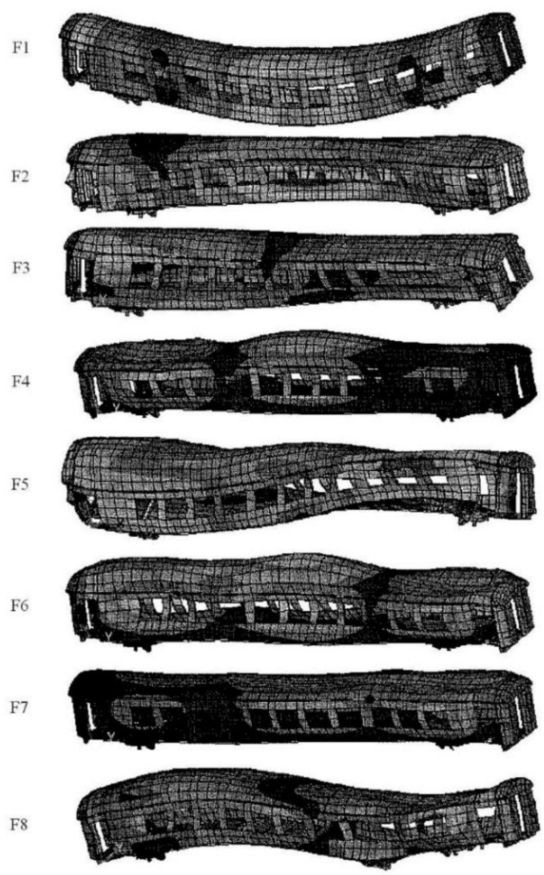

Figure 2: The eight flexible eigenmodes of the carbody [19].

Table 1: The eight mode shapes and undamped frequencies of the carbody

No.	*Mode shape*	*Frequency (Hz)*
F1	First vertical bending	9.1
F2	First lateral bending	12.2
F3	Torsion 1	12.8
F4	Breathing 1	13.4
F5	Torsion 2	13.9
F6	Breathing 2	14.3
F7	Breathing 3	15

Zhou, Goodall, Ren [19] and Zhang have studied how the vertical carbody flexibility influences vertical ride comfort. As mentioned before, the first vertical bending mode has the largest influence on vertical ride comfort. A theoretical vehicle model with rigid and flexible carbody modes is used. When the structural stiffness of the carbody is decreased, so are the bending frequencies, and there is a risk of significant vibrations in the carbody.

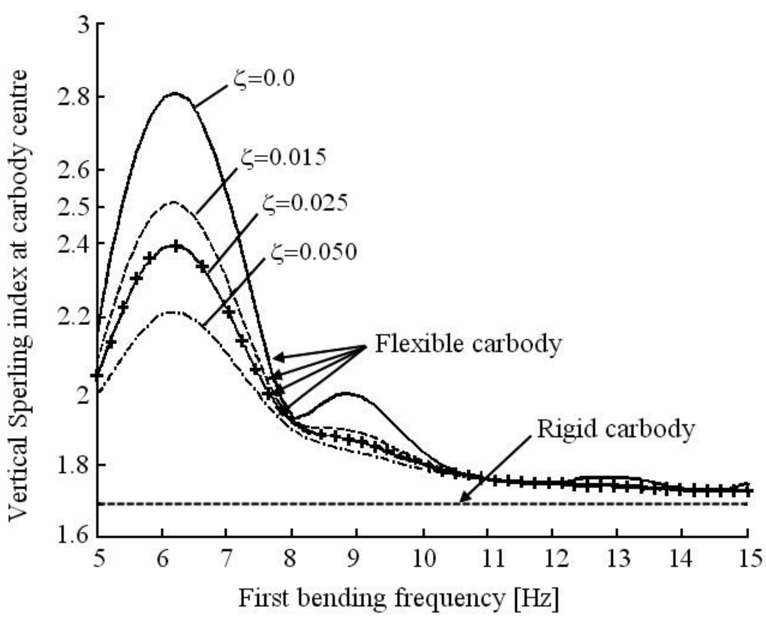

Figure 3: Ride comfort vs frequency, for different damping ratios at center of the carbody [19].

Simulation results show that when the first vertical bending frequency is less than 7 Hz the carbody vibrates violently; see Figure 3. However, increased damping ratio (ζ) of the first bending mode of the carbody may attenuate the resonant peak values to some extent. Further, it is shown that when the first bending frequency is greater than 10 Hz ride comfort of the flexible carbody is approaching the value for the rigid carbody, used as reference in the study.

However, the resonance peak around 6.2 Hz seen in Figure 3 depends on vehicle running speed. The bending frequency where ride comfort of the flexible carbody is approaching the one for the rigid carbody is increased with increased speed [19]. Hence, the higher the speed, the higher stiffness of the carbody is required.

2.2 Primary suspension

The primary suspension consists of spring and damper components between the bogie and the wheelset. The purpose with this suspension level is to secure a stable running behavior, but simultaneously to ensure low track forces. Low wear and good curving behavior. Depending on how it is designed, it can also be a measure to influence ride comfort in the carbody. The component in the primary suspension that if properly designed can have an influence on ride comfort is the vertical axle box damper. There are normally four axle box dampers in each bogie.

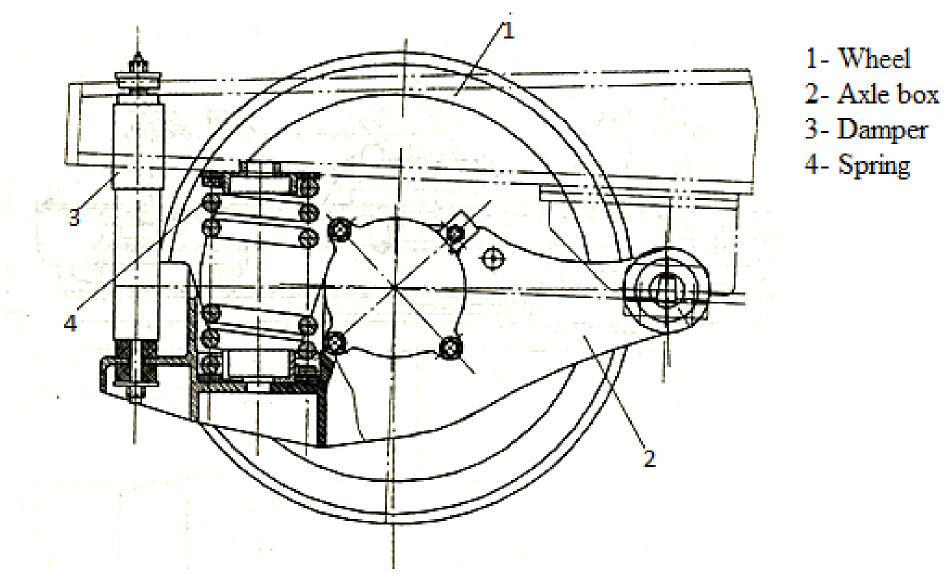

Figure 4: Primary suspension and axle box [7].

2.3 Secondary suspension

The secondary suspension interconnects the carbody and bogie, with the purpose of isolating the carbody from excitations transmitted from track irregularities via the wheelsets and bogie frames. The air spring is part of the secondary suspension of most modern passenger rail vehicles, placed between the carbody and bogie. Its main task is to reduce carbody accelerations in the lower frequency range.

The main benefits with an air spring compared to a conventional coil spring are [2]:

- ✓ Increased stiffness for increased preload,
- ✓ Height independent of preload due to level control,
- ✓ Significant horizontal stiffness,
- ✓ Low height,
- ✓ Good sound and vibration isolation.

Further, the secondary suspension of a rail vehicle normally consists of vertical dampers. These are often hydraulic dampers, with the purpose of reducing low- frequency vehicle vibrations. There are normally two vertical dampers connected to each bogie. The carbody and bogie can be interconnected by an anti-roll bar. Sometimes known as stabilizer, with the purpose of reducing the carbody roll. The anti-roll bar is normally transversely mounted on the bogie with vertical links connected to the carbody, and can be regarded as a tensional spring.

Moreover, the secondary suspension consists of emergency springs (Bumpstops) that limit the relative vertical displacement between carbody and bogies. There is an air gap, ensuring that the bumpstop is activated first when the displacement has become large enough. The bumpstop is a rubber component with significant stiffness. When bumpstop contact occurs it leads a negative impact on ride comforts.

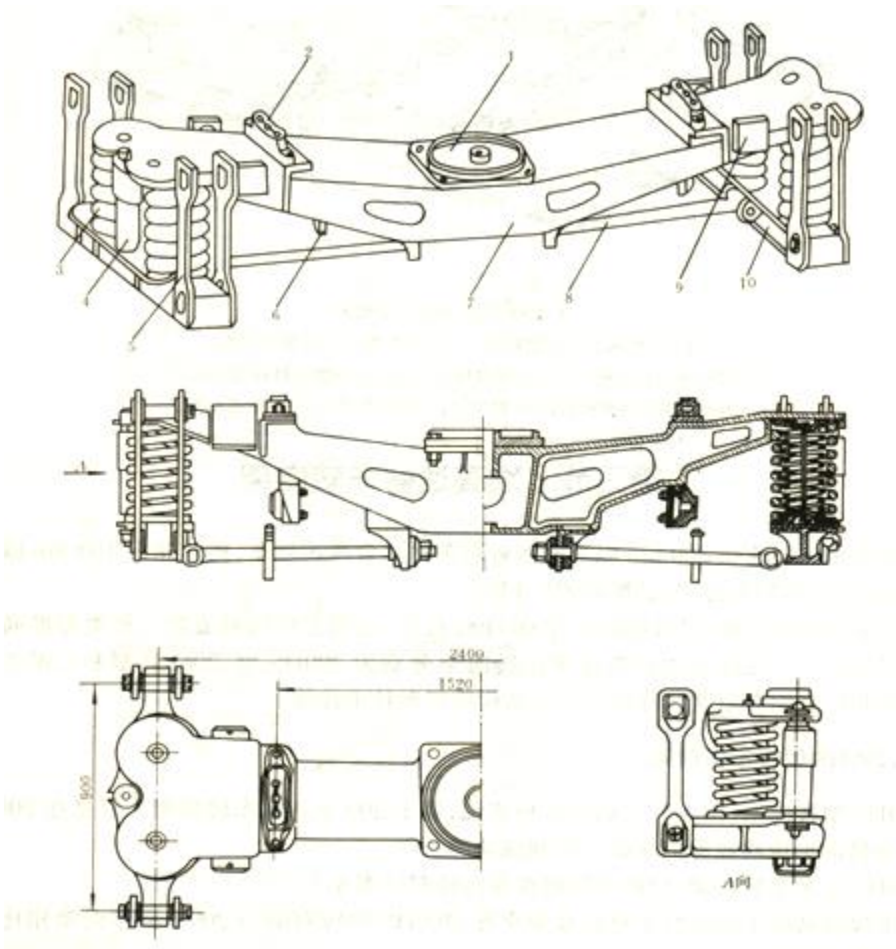

Figure 5: Secondary suspension [2]

2.4 Stiffness of the carbody

From a ride comfort point of view it is preferable to increase the structural stiffness of the carbody in order to reduce the risk of carbody oscillation resonance. The following measures can be taken to increase the carbody stiffness, and hence to increase the natural frequencies of the carbody [25]:

- Make the carbody as short as possible. This conflicts, however, with the desire to maximize the number of passengers in relation to e.g. the number of bogies.

- Make the carbody cross-section as large as possible. However, the cross-section must not interfere with the prescribed carbody gauge profile. Furthermore, crosswind stability problems may arise.
- Use stiffer carbody materials or a design resulting in high carbody stiffness. This is, however, normally contradictory with the requirements on lightweight carbodies for higher speeds.

In a study performed by Suzuki and Akutsu [31], it is investigated how vertical vibrations affect ride comfort (assessed by comfort level LT (dB)). If the structural stiffness is decreased, the natural frequency of the first bending mode is decreased as well, which gives poorer ride comfort, especially at higher speeds. To decrease flexural vibrations, higher structural stiffness is desirable.

2.5 Ride comfort

Good ride comfort is one important issue to aim for within rail vehicle development. Different conditions require various evaluation methods. Therefore, it is difficult to establish a universally applicable international standard on ride comfort of rail vehicles.

A study performed by Suzuki [31] gives a survey of different ride comfort evaluation approaches applied in Japan. The *rms* value of the carbody acceleration is the most common way to evaluate ride comfort. However, there are a number of other possible quantities. such as peak or peak-to-peak values of accelerations, stationary lateral acceleration in curves, crest factor (i.e. the degree of vibrational shock, derived from the peak value divided by the rims value), acceleration and deceleration in longitudinal direction, frequency of roll, yaw and pitch. For tilting trains, the problem with motion sickness is an important issue. Here carbody roll angular velocity and acceleration in transition curves are common indices to measure ride comfort. In addition to measuring physical parameters, the response of the human being must also be taken into consideration when evaluating ride comfort. This call be done by verbal reporting (answering questions about the comfort), physiological response (e.g. measuring the heart rate) and activity interference (e.g. tile ability to read and write).

2.6 Literature summery

From the above literature reviews, papers [6], [7] and [9] have discussed the effect of dumping and stiffness on the vertical vibration of rail vehicle for passenger comfort.

Therefore, in the present paper the simulation of vertical vibration of rail vehicle will be discussed to get the good raid comfort by optimizing primary dumping, secondary dumping, wheel base and stiffness of carbody.

The main target here is to develop the optimum raid comfort index for AALRT according to the existing data.

CHAPTER THREE

MATHEMATICAL MODELING

3.1 Mathematical model

The case of a two-floor suspension railway vehicle that travels on a constant speed *V* on a track with random irregularities is considered (Fig. 2). The vehicle carbody of a length *L* is modelled via an Euler- Bernoulli beam of constant section and uniformly distributed mass, with the bending module *EI*, mass on length unit *m* and damping coefficient μ. The displacement of a beam section is *w*(*x*, *t*), where *t* is time.

Here, only the carbody bending modes have been considered for both generality and simplicity. When a specific structure is studied and the carbody global and local mode shapes are taken into account, the FEM carbody model has to be used.

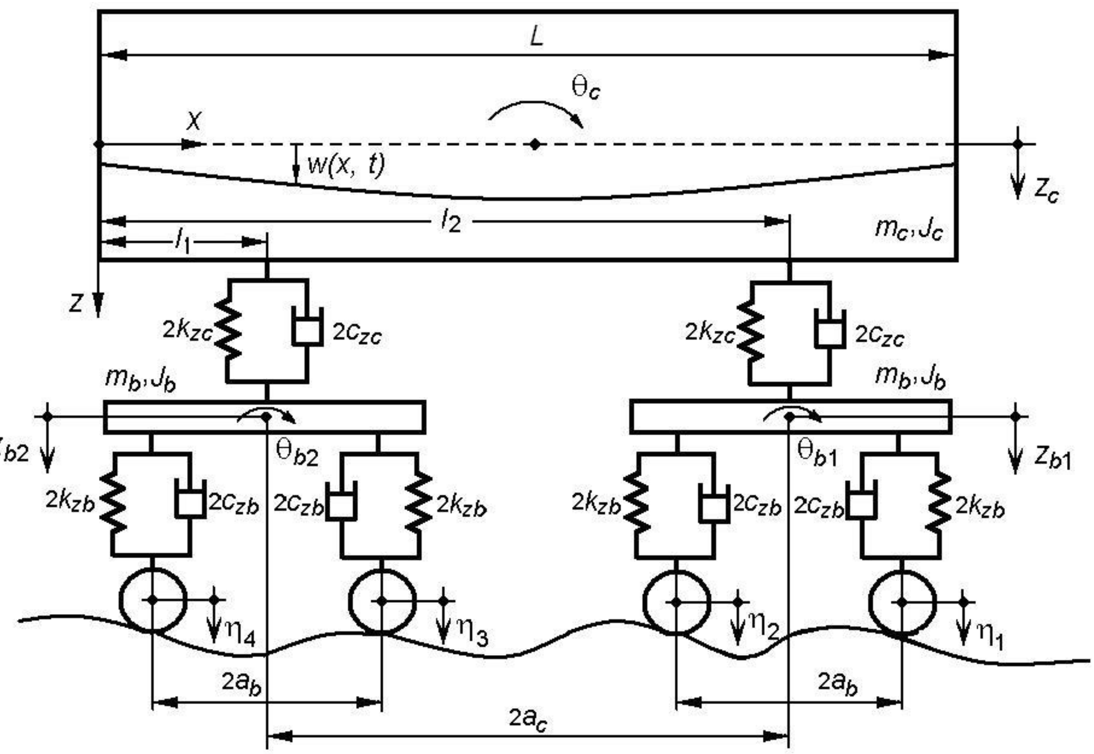

Figure 6: The vehicle mechanical model [35].

The model of the vehicle includes a body with parameters distributed for the carbody and a system of rigid bodies, respectively the wheelsets and the suspended masses of the two bogies. The carbody of the vehicle with length *L* and mass *mc* is modelled by an Euler-Bernoulli beam of a constant section and a uniformly distributed mass, with the bending module *EI*, where *E* is the longitudinal module of elasticity and *I* is the inertia moment of the beam transversal section.

The suspended masses of the bogies are considered rigid bodies on two degrees of freedom, with the following movements: bounce Z_{bi}, and pitch θ_{bi}, where $i = 1, 2$. The mass of a bogie is m_b, and its inertia momentum $J = m_b\, i_b^2$, with i_b – the bogie gyration radius.

The movement equations are: - for bending the carbody [35].

$$EI\,\frac{\partial^4 w(x,t)}{\partial x^4} + \mu I\,\frac{\partial^4 w(x,t)}{\partial x^4\,\partial t} + m\,\frac{\partial^4 w(x,t)}{\partial t^2} = \sum_{i=1}^{2} Fi\,\delta(x - li) \qquad \text{------------1}$$

Where δ(.) is Dirac’s delta function, and Fi represents the force due to the i secondary bogie suspension

$$F_i = -2c_{zc}\left(\frac{\partial w(l_i,t)}{\partial t} - \dot{z}_{bi}\right) - 2k_{zc}\left(w(l_i,t) - z_{bi}\right) \qquad \text{----------------------------2}$$

That acts at the distance *li* from the carbody end. - For bounce of bogies

$$m_b \ddot{z}_{b1,2} + 2c_{zb}(2\dot{z}_{b1,2} - \dot{\eta}_{1,3} - \dot{\eta}_{2,4}) + 2k_{zb}\left(2z_{b1,2} - 2\eta_{1,3} - 2\eta_{2,4}\right) +$$

$$2c_{zb}(z_{b1,2} - \frac{\partial w(l_{1,2},t)}{\partial t} + 2k_{zb}\left(z_{b1,2} - w(l_{1,2},t)\right) = 0 \qquad \text{-------------------------3}$$

That acts at the distance *li* from the carbody end. - For pitch of bogies

$$I_b \ddot{\theta}_{b1,2} + 2c_{zb}a_b(2a_b\dot{\theta}_{b1,2} - \dot{\eta}_{1,3} + \dot{\eta}_{2,4}) + 2k_{zb}a_b\left(a_b\theta_{b1,2} - 2\eta_{1,3} + 2\eta_{2,4}\right) = 0 \qquad \text{------------4}$$

where a_b, a_c are the axle bases of bogies and carbody and η_i with $i = 1,..,4$, the irregularities against the wheelset *i*.

It may be noticed that the pitch movements of bogies are decoupled from the other movements of the vehicle.

The movement equations with partial derivate may be turned into equations with ordinary derivate by implementing the method of modal analysis. For this, the rigid and bending modes of carbody are considered as

$$w(x,t) = z_c(t) + \left(\frac{L}{2} - x\right)\theta_c(t) + \sum_{i=2}^{\infty} X_i(x)T_i(t) \quad \text{----------------------5}$$

where $z_c(t)$ and $\theta_c(t)$ represent the carbody vibration rigid modes, namely the bounce and pitch. Ti(t) is time-dependent coordinate, and $X_i(x)$ is the eigenfunction of vibration mode *i* at bending

$$X_i(x) = \sin\beta_i x + \sinh\beta_i x - \frac{\sin\beta_i L - \sinh\beta_i L}{\cos\beta_i L - \cosh\beta_i L}(\cos\beta_i x - \cosh\beta_i x) \quad \text{---------------------6}$$

$\beta i = \sqrt{\omega_i{}^2 m/(EI)}$ and $cos\beta iLcosh\, \beta iL - 1 = 0$, where ωi is the natural angular frequency of the vibration mode *i*.

Looking at the first bending modes only, symmetrical the vehicle vibration is described by a set of equations.

$$m_c\ddot{q_1} + 4C_{zc}(\dot{q_1} + \varepsilon\dot{q_2} - \dot{q_3}) + 4k_{zc}(q_1 + \varepsilon q_2 - q_3) = 0 \quad \text{-----------------7}$$

$$m_{c2}\ddot{q_2} + C_{m2}\dot{q_2} + k_{m2}q_2 + 4C_{zc}\varepsilon(\dot{q_1} + \varepsilon\dot{q_2} - \dot{q_3}) + 4k_{zc}\varepsilon(q_1 + \varepsilon q_2 - q_3) = 0 \quad \text{-------8}$$

$$m_b\ddot{q_3} + 4C_{zb}(\dot{q_3} - \eta) + 4k_{zb}(q_3 - \eta) + 4C_{zc}(\dot{q_3} - \varepsilon\dot{q_2} - \dot{q_1}) + 4k_{zc}\varepsilon(q_3 - \varepsilon q_2 - q_1) = 0 \quad \text{---9}$$

Damping and stiffness of the second and third mode

$$m_{m2,3} = m\int_0^L X_{2,3}^2(x)\mathrm{d}x \quad \text{----------------------------10}$$

$$k_{m2,3} = EI\int_0^L \left(\frac{\mathrm{d}^2 X_{2,3}(x)}{\mathrm{d}x^2}\right)^2 \mathrm{d}x;\; c_{m2,3} = \mu I\int_0^L \left(\frac{\mathrm{d}^2 X_{2,3}(x)}{\mathrm{d}x^2}\right)^2 \quad \text{-----------------11}$$

The below equations is the excitation mode

$$4\eta = \eta_1 + \eta_2 + \eta_3 + \eta_4 \quad \text{-------------------------------12}$$

The parameters of the carbody bending vibration are the modal angular frequency and the damping ratio

$$\omega_{m2,3} = \sqrt{\frac{k_{m2,3}}{m_{m2,3}}},\; \zeta_{m2,3} = \frac{c_{m2,3}}{2\sqrt{k_{m2,3}m_{m2,3}}} \quad \text{--------------------13}$$

In order to facilitate the analysis of vibrations, the damping ratio of the suspension levels considered uncoupled, as below equation (14).

$$\zeta_{b,c} = \frac{4c_{b,c}}{2\sqrt{4k_{b,c}m_{b,c}}} \quad \text{---14}$$

The wheelsets of mass m_b have two degrees of freedom, thus generating a vertical movement of translation $Z_{oj,(j+1)}$ and a longitudinal movement of translation $X_{oj,(j+1)}$, where $j = 2i-1$, and $i = 1, 2$, where each bogie is equipped with the wheelsets j and $j+1$

The following movement equations will be used to evaluate the comfort of a passenger car. The vibration harmonic behavior where the track irregularities have a sinusoidal shape with the wavelength l and the amplitude η_0

$$\eta_{1,2} = \eta_0 \cos\frac{\omega}{V}(Vt \pm a_b + a_c)\,,\ \eta_{3,4} = \eta_0 \cos\frac{\omega}{V}(Vt \pm a_b - a_c) \quad \text{---------------------------15}$$

Where $\omega=2\pi V/l$ is the angular frequency.

The vibrations of carbody are excited by the track irregularities against the wheelsets, and the plan of bogie axles has both a bounce and pitch movement. Since the bogies pitch movement is decoupled from the vehicle vibration will be determined by the symmetrical axles bounce movements,

The frequency response in a point x of carbody is as below

$$\overline{H}^{+}(\omega) = \cos\frac{a_b\omega}{V}\cos\frac{a_c\omega}{V}\,;\quad \overline{H}^{-}(\omega) = \mathrm{i}\cos\frac{a_b\omega}{V}\sin\frac{a_c\omega}{V} \quad \text{-------------------16}$$

The excitation modes will bring a series of maximum and minimum values depending on the velocity and may attenuate or not the resonance peaks of the carbody frequency response. For instance, for the frequencies $f = (2n+1)V/4ab$ with n = 0, 1, ..., the wheelsets' bounce is zero and, thus, the carbody bounce and pitch are not excited at these frequencies – the geometric filtering effect of the axle base. The distance between bogies derives the geometric filtering effect for the symmetrical bounce of wheelsets at frequencies f=(2n+1)V/4ac.

The frequency response in a point x of carbody is as below

$$\overline{H}_c(x,\omega) = \overline{H}_{Z_c}(\omega) + \left(\frac{L}{2} - x\right)\overline{H}_{\theta_c}(\omega) + \sum_{i=2}^{3} X_i(x)\overline{H}_{T_i}(\omega) \quad \text{-------------------------17}$$

Where $\overline{H}_{z_c}(\omega)$, $\overline{H}_{\theta_c}(\omega)$, $\overline{H}_{T_{2,3}}(\omega)$ are the frequency response corresponding to the vibration modes z_c, θ_c and $T_{2,3}$.

3.2 Track Inputs to Rail Vehicle

The dynamic wheel loads generated by a moving train are mainly due to various wheel/track imperfections. These imperfections are considered as the primary source of dynamic track input to the railroad vehicles. Normally, the imperfections that exist in the rail-track structure are associated with the vertical track profile, cross level, rail joint, wheel flatness, wheel/rail surface corrugations and sometimes uneven support of the sleepers [3].

The following figure.7 shows the model of track irregularity. The shape of the irregularity is assumed to be similar on the left and the right rails

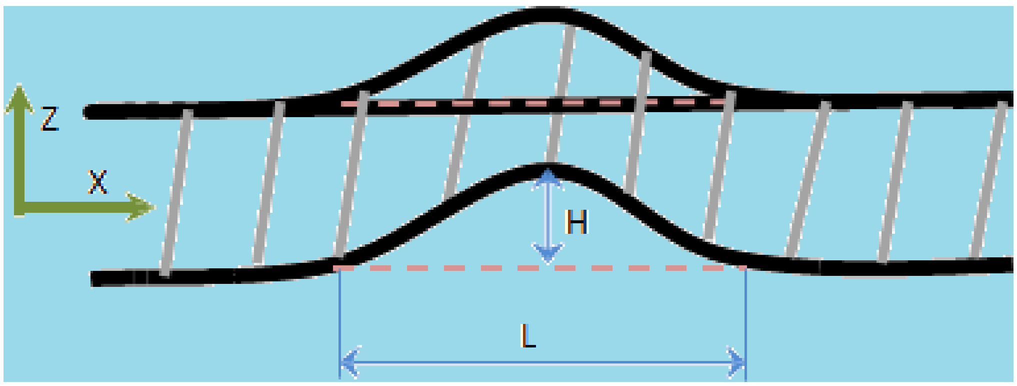

Figure 7: Bump excitations of the track

The bump excitations of the wheels (Figure. 9) of leading bogies are as follows:

$$Z_{ri} = \begin{cases} \frac{H}{2}\left[1-\cos\left(2\pi\frac{V}{L}\left(t-t_{di}\right)\right)\right] \\ 0 \end{cases} \quad \text{------------------------------18}$$

$$for \quad t_{di} \le t \le \frac{L}{V} + t_{di} \quad (i = 1..4) \quad \text{----------------------------19}$$

$$\text{Where } \left[t_{d1}, t_{d2}, t_{d3}, t_{d4}\right] = \left[0, \frac{2L_d}{V}, \frac{2L_b}{V}, \frac{2L_b + 2L_d}{V}\right] \quad \text{---------------------20}$$

3.3 Vertical vibrations and Comfort

3.3.1 Power Spectral Density (PSD)

A power spectral density (PSD) shows how much a point of the carbody vibrates at various frequencies when the vehicle is running on the track. The area under the PSD-curve corresponds to a mean square value. The peaks in the spectra are due to structural eigenmodes of the carbody structure, rigid-body eigenmodes of the complete vehicle or harmonics of the excitation induced by the track irregularities. In order to evaluate the comfort at vertical vibrations, there will be taken into account the track irregularities as being random and stationary. For the power spectral density PSD of vertical track irregularity as system inputs, which is described in spatial domain as.

$$S(\Omega) = \frac{A\Omega_c^2}{(\Omega^2 + \Omega_r^2)(\Omega^2 + \Omega_c^2)} \quad \text{---21}$$

Where Ω is the wave number, Ω*c* = 0.8246 rad/m, Ω*r* = 0.0206 rad/m, and *A* = 4.032·10-7 rad m or *A* = 1.080·10-6 rad m, depending on the track quality.The track irregularities become an excitation factor for a vehicle that travels at speed *V* and this is the reason why the power spectral density of the track irregularities need to be expressed as a function of the angle frequency *ω = VΩ*, and *G(ω) = S(ω/V) /V* respectively. As a result, the rel. will trigger

$$G_c(x,\omega) = \omega^4 G(\omega)\left|\overline{H}_c(x,\omega)\right|^2 \quad \text{-------------------------------------22}$$

Starting from the carbody frequency response *Hc (x,ω)* and the power spectral density of track irregularities, the acceleration power spectral density may be calculated at a point located at distance x from the carbody referential

$$G(\omega) = \frac{A\Omega_c^2 V^3}{[\omega^2 + (V\Omega_c)^2][\omega^2 + (V\Omega_r)^2]} \quad \text{----------------------------------23}$$

Further, based on the acceleration power spectral density, we will have the *Wz* comfort index, as seen below. An interest will be taken in the acceleration power spectral density in two points, namely at the center of carbody and above the rear bogie.

3.3.2 Ride Index (Wz)

Ride comfort evaluation according to ISO 2631 is well described in the European standard EN 12299 [10]. The *rms* values of frequency-weighted accelerations on the carbody floor are evaluated as:

$$a^{wrms} = \left[\frac{1}{T}\int_0^T \left[a^w(t)\right]^2 dt\right]^{0.5} \text{------------------------}24$$

Where α^w (t) is the frequency-weighted acceleration as a function of time *t*. *T = 5 s* is the duration of the measurement. The filter functions for the horizontal and vertical directions are illustrated in Figure 3.

According to ISO 2631 evaluation, human bodies are considered to be most sensitive to frequencies in the 0.5–2 Hz range in horizontal direction and in the 4–15 Hz range in vertical direction. The ride comfort levels for the individual lateral and vertical directions according to ISO 2631 [10] are described in Table 2. The levels are the same for the lateral as well as the vertical direction.

Table 2: Ride comfort levels (lateral and vertical) according to ISO 2631 [10].

α^{wrms}(m/s2)	Ride comfort
$\alpha^{wrms} < 0.2$	Very comfortable
$0.2 \le \alpha^{wrms} < 0.3$	Comfortable
$0.3 \le \alpha^{wrms} < 0.4$	Medium
$0.4 \le \alpha^{wrms}$	Less comfortable

A common comfort evaluation method applied, in most country is the comfort number *Wz*. It originates from German research in the 1940's and 1950's by Sperling and Betzhold [37, 38] and is a frequency-weighted *rms* value of the lateral or vertical accelerations on the carbody floor, normally evaluated over a one kilometer distance. *Wz* is defined as [10].

$$W_z = 4.42(a^{wrms})^{0.3} \quad \text{--}25$$

Where a^{wrms} is the *rms* value of the frequency-weighted acceleration.

According to W_z evaluation, human bodies are considered to be most sensitive to frequencies in the 4–15 Hz range, laterallyz as well as vertically vibration levels according to *Wz.* The levels are the same for the lateral as well as the vertical direction.

In order to calculate the total raid index *Wz* for a continuous spectrum, at a certain carbody point, we have the following formula [35].

$$Wz = \left(2\int_{0.4}^{30} G_c(f) B^2(f)\mathrm{d}f \right)^{(1/6.67)} \quad \text{-------------------------------}26$$

Where $G_c(f)$ is the power spectral density of acceleration in cm/s^2/Hz, *B(f)* is a factor of acceleration evaluation and f is the frequency of vibrations.

For the vertical comfort, it has to be [35].

$$B(f) = 0.588\sqrt{\frac{1.911f^2 + (0.25f^2)^2}{(1-0.277f^2)^2 + (1.563f - 0.0368f^3)^2}} \quad \text{-------------------------}27$$

The relation between the comfort index and the sensitivity to vibrations is assessed on a 1 to 4 scale, where *Wz* = 4 when the vibrations level has a harmful impact upon the human body at a prolonged exposure.

Table 3: Vibration levels (lateral and vertical) according to *Wz [35]*.

Wz	***Vibration level***
1	Just noticeable
2	Clearly noticeable
2.5	Pronounced, but not unpleasant
3	Strong, but tolerable
3.5	Very strong and unpleasant
4	Extremely strong and unpleasant

CHAPTER FOUR

MODELING AND SIMULATION

4.1 Multi-body System

To analyze the dynamic behavior of railway vehicles, usually the vehicle is represented as a ***Multi-body System***. A multibody system consists of rigid bodies, interconnected via mass less force elements and joints. Due to the relative motion of the system's bodies, the force elements generate applied forces and torques. Typical examples of such force elements are springs, dampers, and actuators combined in primary and secondary suspensions of railway vehicles.

The complete vehicle is modelled by using SIMPAC software. Simpack is generally purpose multibody simulation (MBS) software used for the dynamic analysis of any mechanical or mechatronic system. It enables engineers to generate and solve virtual 3D models in order to predict and visualize motion, coupling force and stress.

4.2 Assumptions

The assumptions made in formulating the model are as follows:

- Bogie and car body component masses are rigid.
- Axle box was assumed as a wheel set's integral part, hence its weight was combined with the wheel set's mass
- The springs and dampers of the suspension system elements have linear characteristics.
- Friction does not exist between axle and bearing.
- All wheel profiles are identical from left to right on a given axle and from axle to axle and all wheel remains in contact with the rails.
- The track irregularity mode in vertical direction is symmetrical for both left and right rail is considered.
- The dynamic simulation was made on straight track with no supper elevation and curve radius

4.3 Simulation using SIM-PACK

The procedure of the simulation analysis study in the SIMPACK software is shown in Figure 8 below: -

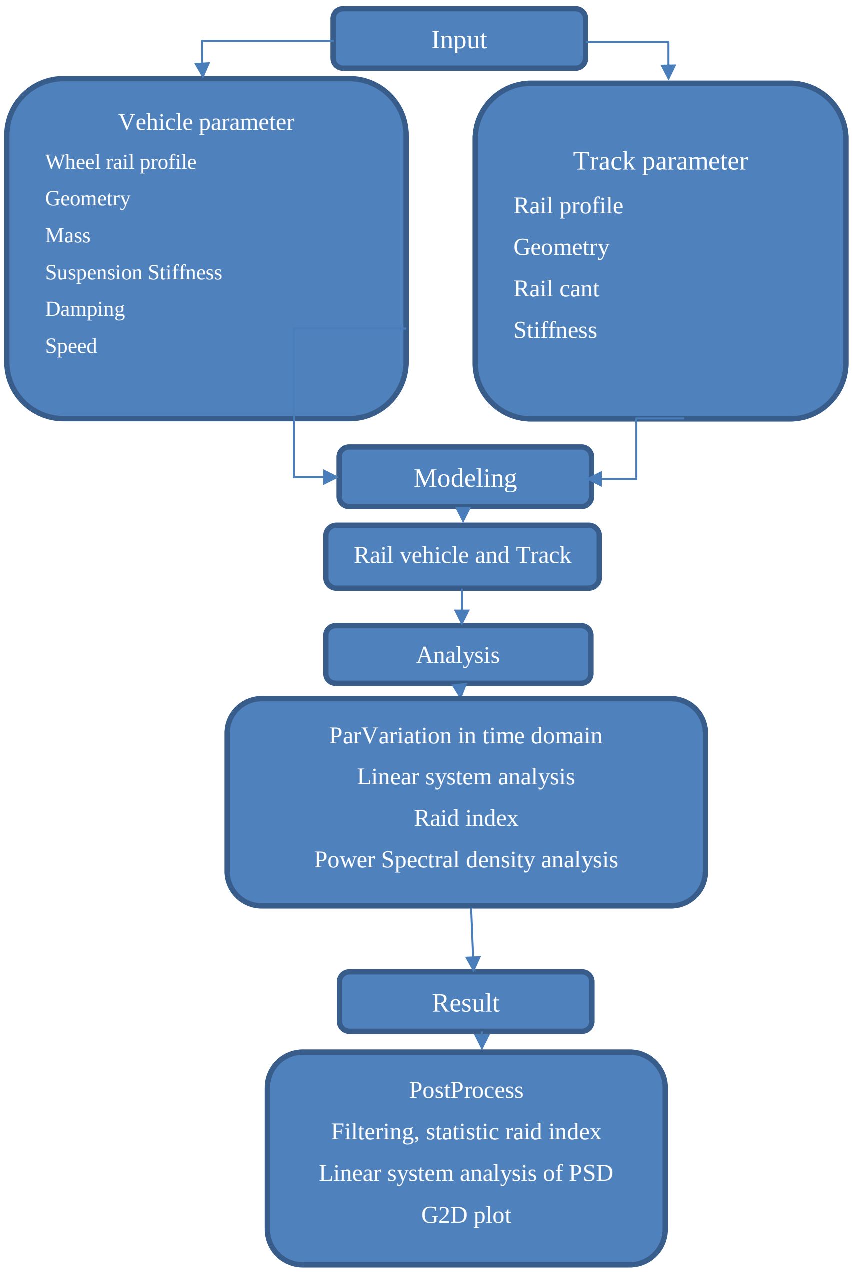

Figure 8: SIMPACK simulation procedure

4.3.1 Modeling of wheelset

To create the guided wheelset, I (the first wheelset) we can use two methods. The first one is select from General rail-track joint of existing Rail-wheelset module and then we can edit as needed or the second method is start from scratch. In this research, the first method was used.

Table 4: Wheelset parameter and value from ERC

Parameter	***Value***
Wheelset mass *(m_w)*	1000 Kg
Wheel diameter *(D_w)*	915mm
Wheelset shaft diameter*(d_s)*	0.915m
Wheel profile type	S1002
Rail profile	UIC60
Rail cant	0.069444444
Young's modulus	202GN/m^2
Wheel set base (W_b)	Variable
Friction coefficient	0.4
Vehicle center distance	18000mm

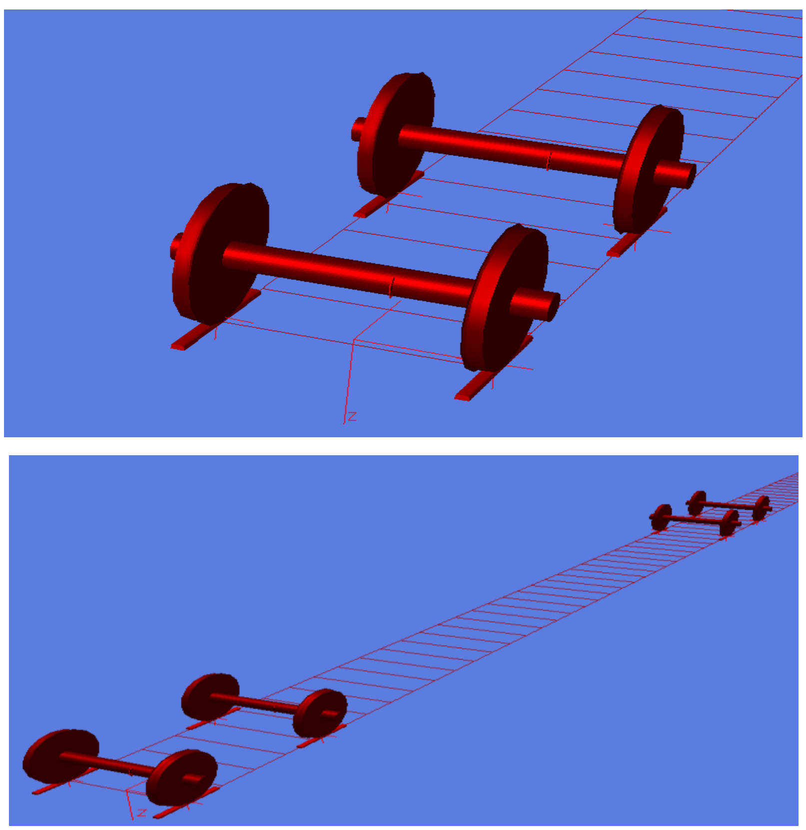

Figure 9: Modeling of Wheelset

4.3.2 Modeling of bogie frames

Developing the bogie frame is done by the following procedure: Go to *Bodies* toolbar and change the type of *3D geometry* of the body to *Wheel Rail bogie*. Then modify it according to ERC bogie type. The ERC Bogie parameters listed below table 5.

Table 5: Bogie frame parameter and value

Parameter	***Value***
Bogie frame mass (total)	3200 Kg
Axle base	2400mm
Bolster mass	220Kg
Bolster dimension (X-Y-Z)	0.7mx2.3mx0.12m

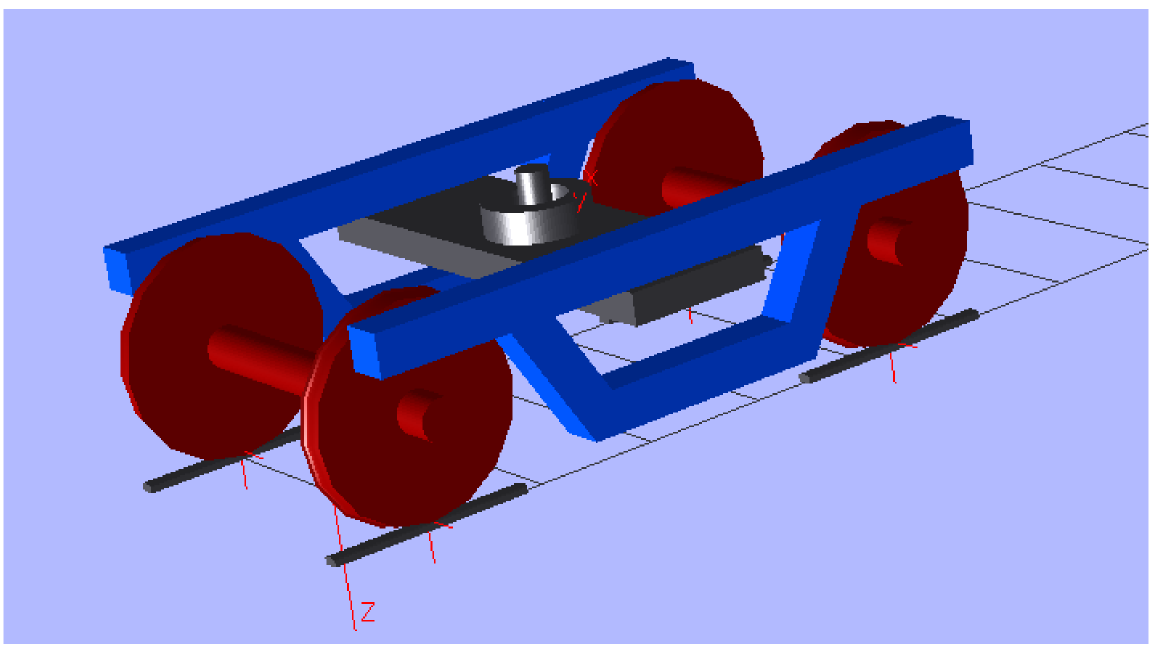

Figure 10: Modeling of bogie frames

4.3.3 Modeling of suspension elements

To create the primary spring suspension insert markers type 1: Identity Matrix on the frame and wheelset on the attachment points of the primary suspension. Insert force element type 86: Spring Damper Ser/Par cmp between the two markers. Again insert the same type of markers on the same places. Then insert force element type 6: Spring Damper Serial cmp between the two markers. To create the secondary suspension follow the same step using force types 79: Shear spring cmp and 6: Spring damper Serial ptp by inserting markers between carbody and bogie frame. Table 6 gives the data required for the development.

Table 6: Suspension element parameter

Parameter	***Value***
Longitudinal primary suspension spring stiffness	60 MN/m
Lateral primary suspension spring stiffness	7.5 MN/m
Vertical primary suspension spring stiffness	60 MN/m
Longitudinal primary damping coefficient	15000Ns/m
Lateral primary damping coefficient	2000Ns/m
Vertical primary damping coefficient	17000kNs/mm
Longitudinal secondary suspension spring stiffness	160000 N/m
Lateral secondary suspension spring stiffness	160000 N/m
Vertical secondary suspension spring stiffness	430000 N/m
Lateral secondary damping coefficient	17500 Ns/m
Roll bending stiffness for secondary suspension	10500 N/m
Pith bending stiffness for secondary suspension	10500 N/m
Torsional bending stiffness for secondary suspension	10500 N/m
Vertical Secondary damping coefficient	21000kNs/mm

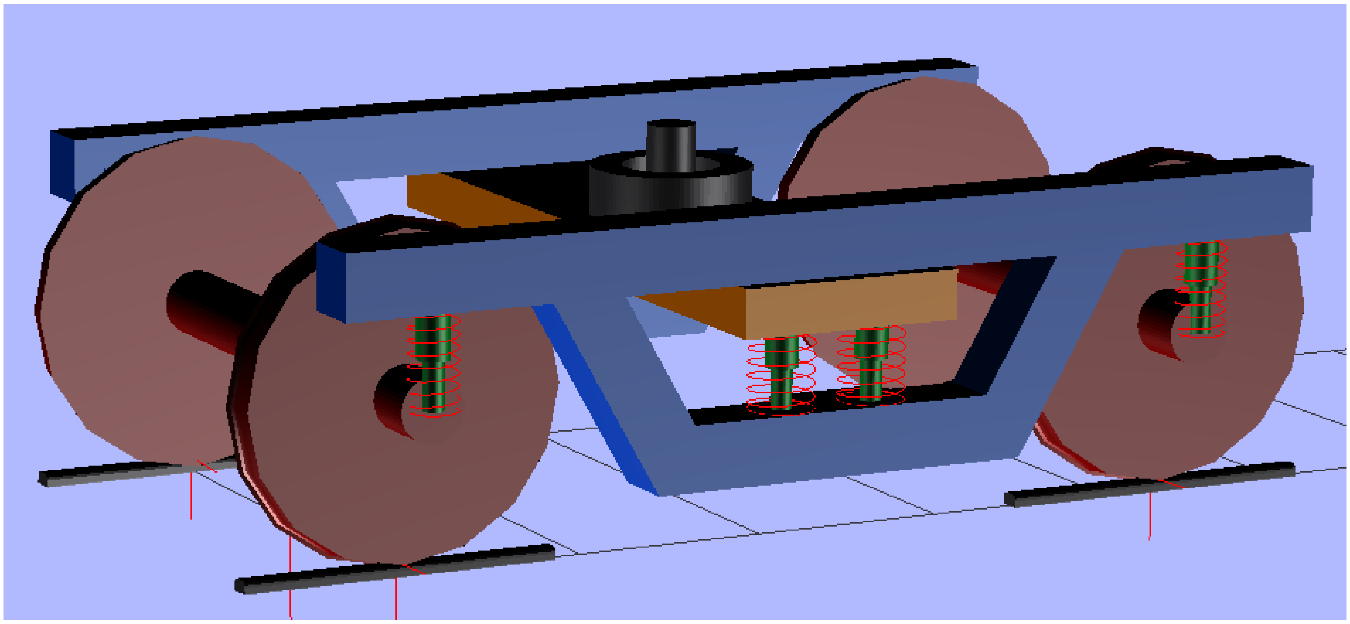

Figure 11: Modeling of suspension elements

4.3.4 Modeling of Car body

Developing the Car body in SIMPACK is a slight different than the others. It needs the SIMPACK documentation *3D-primitive: Wheel Rail Vehicle Cab*. This 3D primitive is representing a parameterized geometry for standard wheel rail vehicles. The car body model dimension parameter is referring the 3D-primitive and then modify it according to the ERC car body sizes.

Table 7: Carbody parameter and value

Parameter	***Value***
Car body mass	34320 kg
Bending module	3.2*109Nm^2
Carbody length	25500mm
Carbody width	3105mm
Car body axle base	18000mm

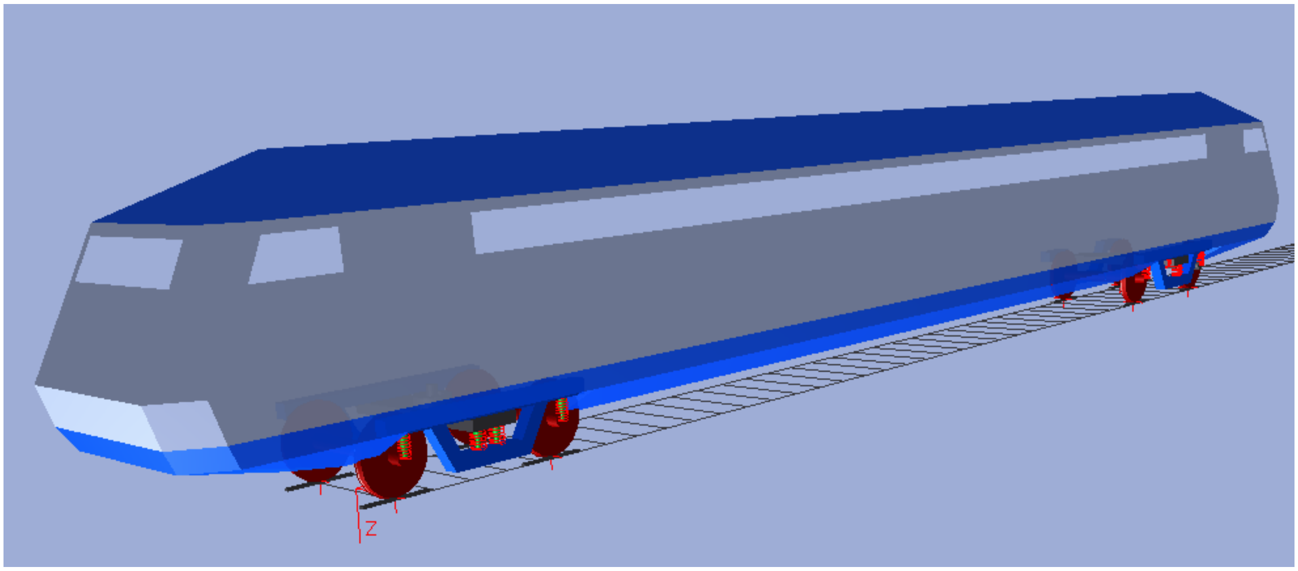

Figure 12: Modeling of Car body

4.3.5 Inserting track inputs and sensors

Finally create Input Function for the speed profile. Create track excitation of type 20: Speed Profile along Track and insert the speed profile function. Insert sensors where data recording is needed. Perform preload calculation in order to bring the model to initial equilibrium. The track irregularity model is shown in Figure 13.

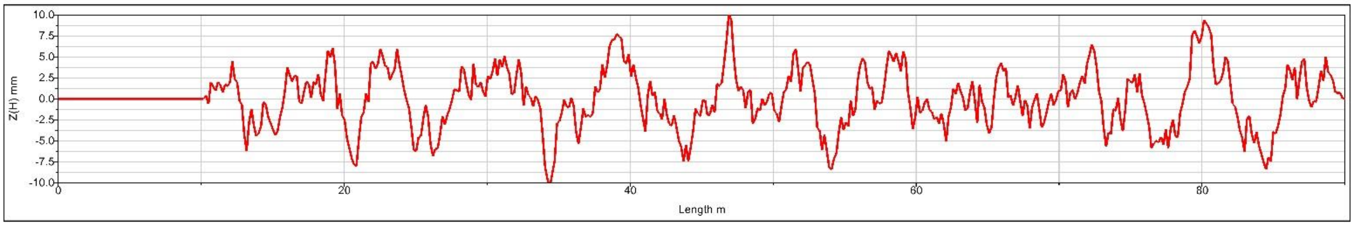

Figure 13: Modeling of track irregularity

CHAPTER FIVE

MODEL VALIDATION, RESULT AND DISCUSSION

In this section, the developed SIMPACK model will be validated and the output results will be presented and discussed. Since, the purpose of the study is to optimize the vertical vibration on rail vehicle dynamic for raid comfort. The model is developed with different parameters which are the stiffness and damping variability applied in the model by giving sinusoidal varying stiffness and damping inputs in addition to the vehicle speed. Furthermore, the model is developed by applied different vehicle axel base dimension and carbody stiffness.

5.1 Power spectral density

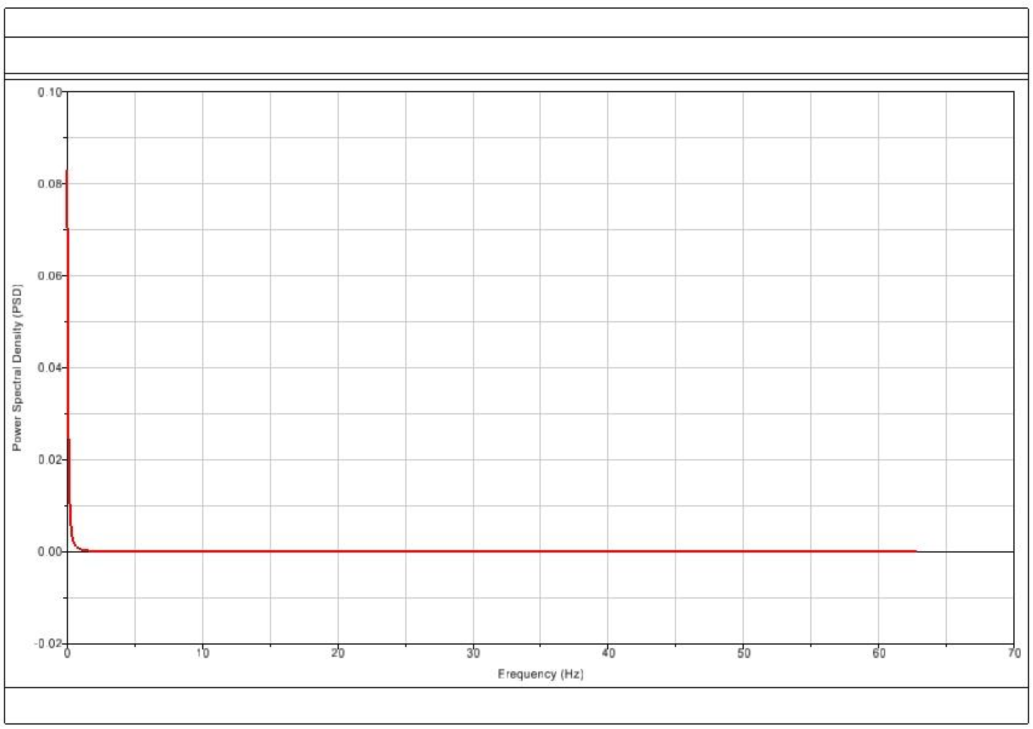

Figure 14: Power spectral density of acceleration with excitation frequency

5.2 Parametric Optimization

A parametric study was undertaken to find the optimum value of different suspension parameters on the Sperling index (Wz). The parameters considered for the analysis were primary damping, secondary damping, and stiffness of car body and axle base geometry. The results of the parametric study have been plotted in terms of performance index such as the Wz index versus vehicle parameter. Figures from 11 to 15 show the variation of Wz index as a function of primary and secondary vertical damping stiffness respectively. The increase in the primary suspension stiffness reduces the Wz index at the carbody marginally up to a speed of 60 m/s. The influence of secondary dumping has been found to be just the same as the primary dumping. The secondary damping has great influence on the ride index Wz for passenger comfort. An Increase of the secondary dumping value from the present value reduces the Wz index at the carbody at all the speeds. The variation of the primary damping is seen to have little influence on the Wz index at the carbody.
An increase in the primary damping reduces the Wz index at speeds greater than 20 m/s. It has also been observed that all the primary damping has very limited influence at low speeds.

5.2.1 Primary Suspension

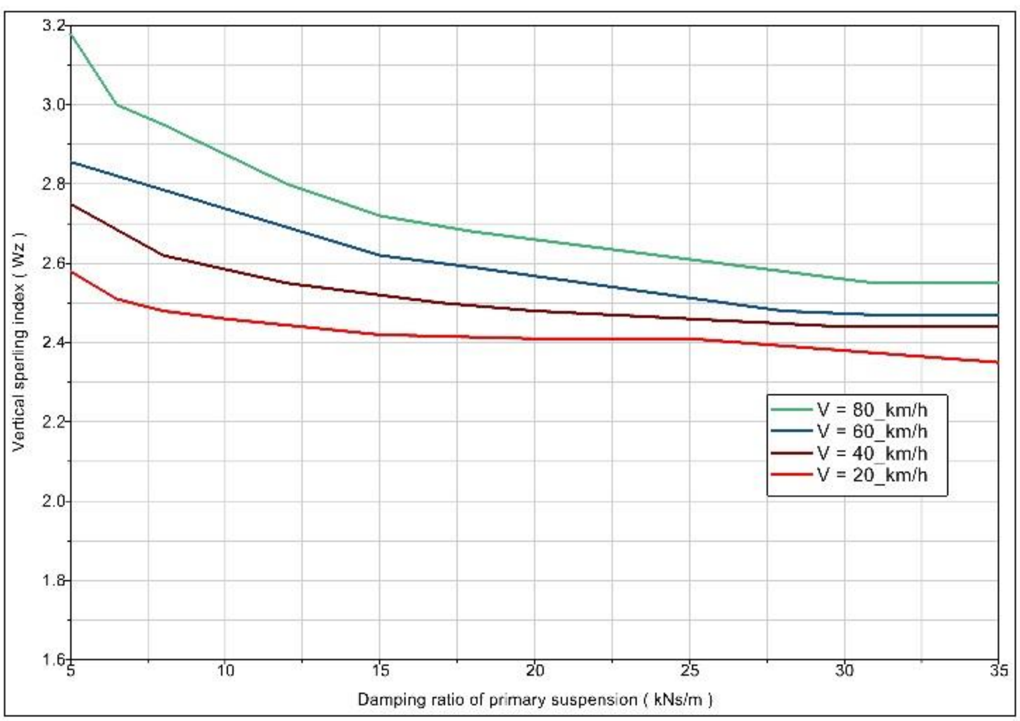

Figure 15: Primary suspension damping versus vertical sperling Index (Wz) at Bogie frame

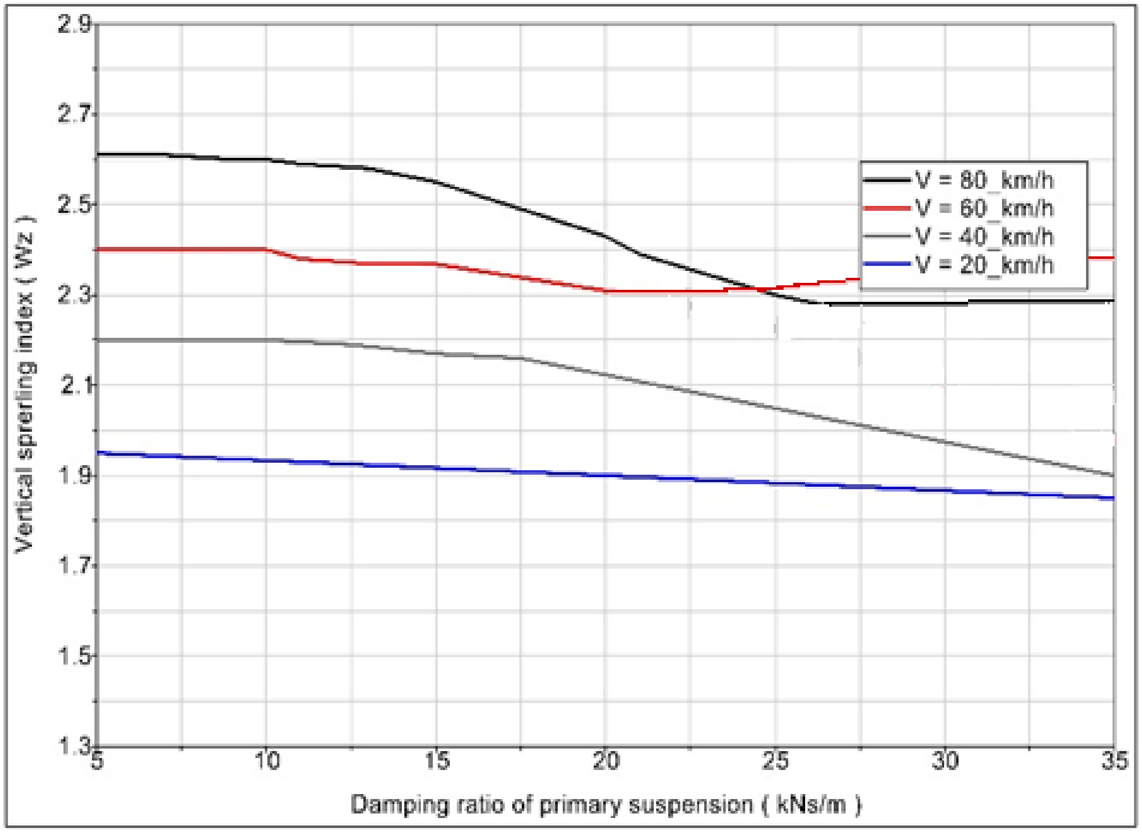

Figure 16: Primary suspension damping versus Wz at carbody center Secondary Suspension

In figure 15 & 16, the index Wz in calculated for 80, 60, 40 and 20 km/h. For the usual domain of primary suspension damping values, it may be noticed that an increased damping reduces the vibrations level and the index Wz, where this lowering is uniform in both points under study. It should be highlighted that there is also a damping value for this case that minimizes the carbody vibrations level in any point. For instance, at 60 km/h, the minimum value of Wz at carbody center is obtained for ζ_b = 20.62kNs/m, and above bogie for ζ_b = 35kNs/m.

5.2.2 Secondary Suspension

The secondary suspension damping greatly influences the ride comfort, as seen in figure 18. In fact, there is a value for damping that minimizes the carbody vibration level in any point of it. The explanation is that, on the one hand, this damping limits the vibration at resonance frequencies at small damping values and, on the other hand, when damping is high, the system dynamic rigidity rises, thus the vibrations behavior is more intense.

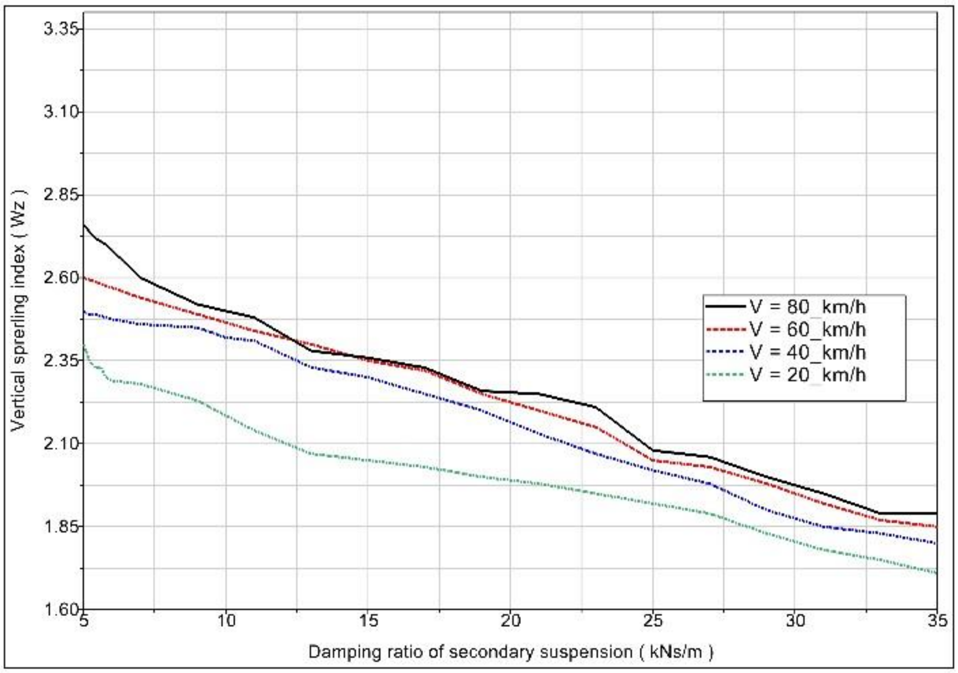

Figure 17: Secondary suspension damping versus vertical sperling Index (Wz) at Bogie frame

The simulation diagrams shows that index Wz has a minimum value corresponding to a certain damping value, which depends on the position along carbody center or bogie frame and on velocity. It may be noticed that comfort index Wz becomes minimum at carbody center for low vehicle speed. Increasing damping to a definite value means a general comfort improvement. Beyond this limit, damping has contrary effects, enhances comfort above bogie and worsens it at carbody center. The idea is to minimize the Wz value at carbody critical point (where vibrations level is higher) to obtain the best damping. Diagrams show that the critical point on car body center and at bogie frame. For instance, at 20 km/h, the index Wz that minimum in this point damping

value is = 35kNs/m, 35kNs/m at V = 40 km/h, 25.62kNs/m at 60km/h and 28.45kNs/m for 80 km/h at carbody center and the minimum damping value at bogie frame 28.45kNs/m at 80 km/h.
So far, the focus has been placed on how the secondary suspension damping ratio influences the vibrations behavior of a vehicle and, therefore, the comfort evaluation index. Even though the primary suspension damping ratio cannot be raised as much as desired due to the magnitude of wheel-rail contact dynamic forces, it is still interesting to analyze to what extent this parameter influences the carbody vibrations behavior.
Upon comparing the influence of damping of two-level suspension upon comfort, it is evident that primary suspension damping has a small influence. Indeed, the bogie mass comes between the primary suspension and carbody, and whose inertia reduces the damping efficiency of the primary suspension.

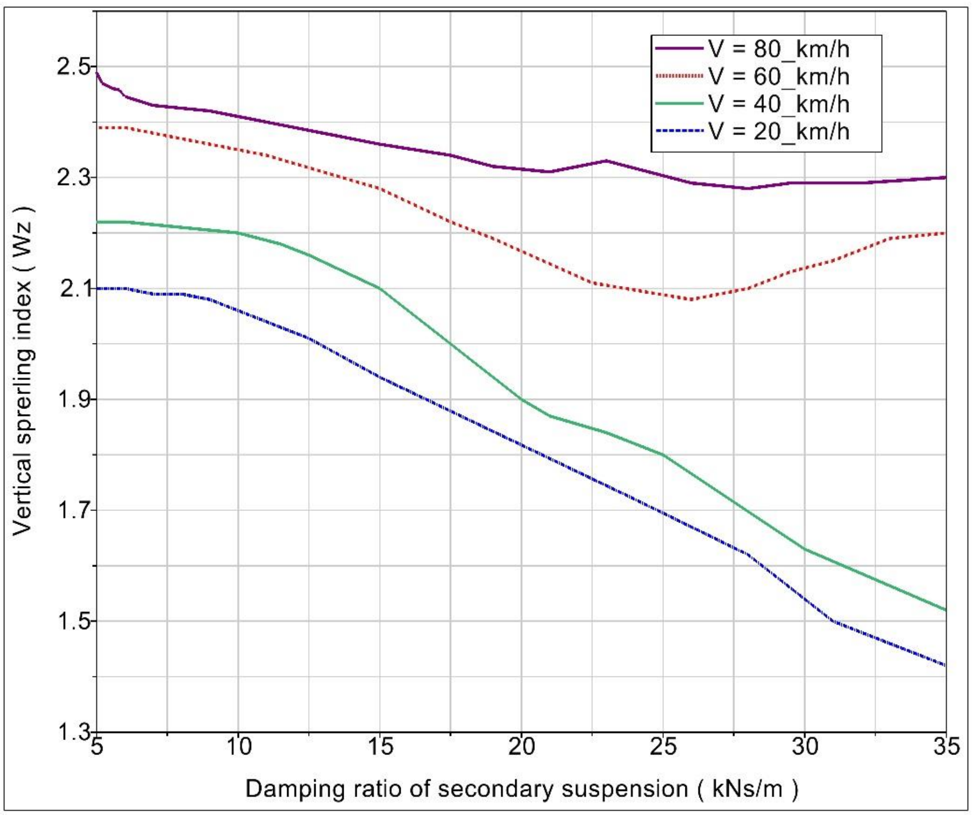

Figure 18: Secondary suspension damping versus (Wz) at carbody center

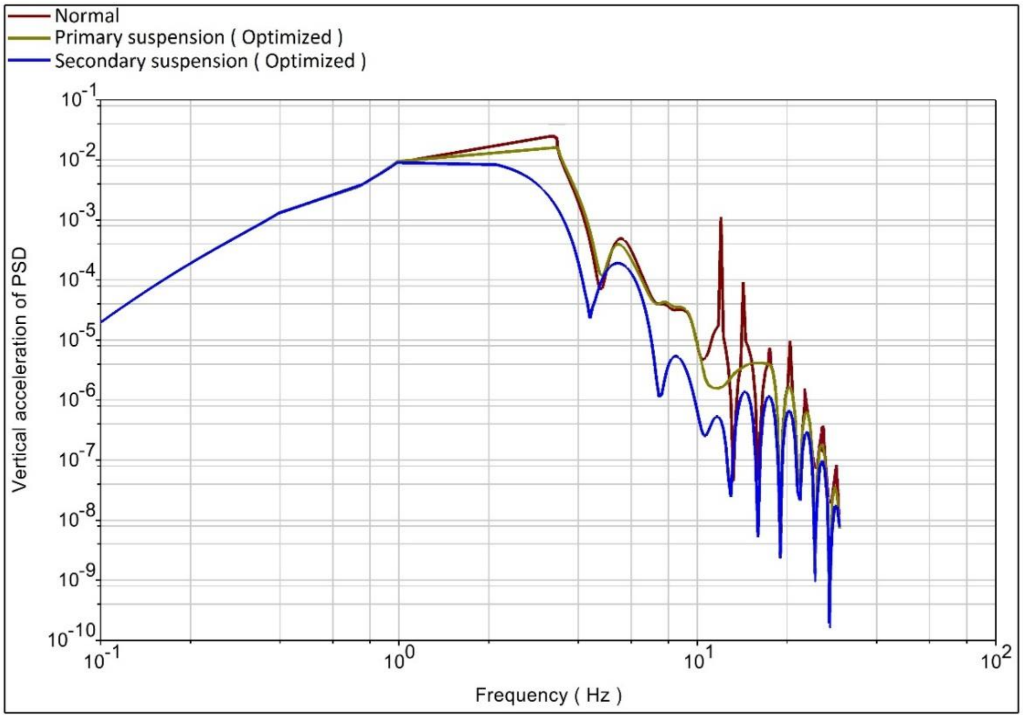

Figure 19: PSD diagrams of car body center

Higher Primary and secondary suspension vertical damping coefficient values from the existing values is preferred with respect to vertical acceleration PSD as the acceleration values is significantly reduced in all frequency regions of our interest.

5.2.3 Axle Base

The geometric filtering effect due to the vehicle axle base impacts the Wz index and acts selectively in dependence with velocity and point position along the carbody (see fig 20 & 21). The optimum value has been considered for the primary suspension vertical damping during the numerical simulations.

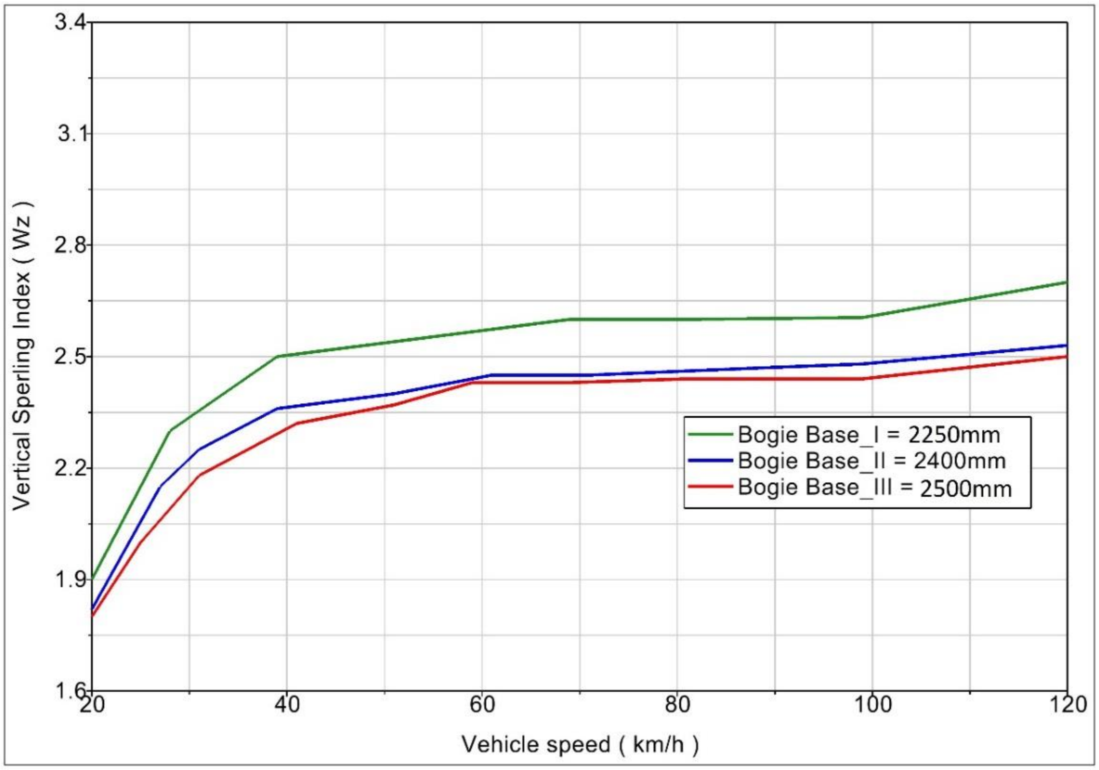

Figure 20: Influence of velocity and vehicle axel base upon Index *Wz* at bogie frame

The value of the Wz index changes along with velocity increase and its evolution is different compared to the position along carbody. Above bogie, we notice a tendency of Wz index going up with velocity – the carbody center is a different story, as a series of maximum and minimum values appears, due to the geometric filtering effect.

The maximum values apply to the condition where the geometric filtering effect is close to a natural frequency of the vehicle. Should the filtering effect frequency (the characteristic is zero) coincides with one of the vehicle natural frequencies (mostly the dominant one), then the Wz index is minimum.

The increase in the bogie axle base (fig. 21) generally triggers a lowering in vibrations level. At carbody center, where the increase of axle base has a contrary effect upon the Wz comfort index for velocities of 20 – 120 km/h. The velocity introduces a selective filtering similarly with the axle bases. For big bogie axle bases, filtering lobes height rises and filtering has a lower effect at the vehicle resonance frequencies. This fact explains that, the bogie axle base at bogie frame from 2.4 to 2.5 m will lead to a Wz from 2.46 to 2.4 and also from 2.4 to 2.25m will lead Wz from 2.48 to 2.62 On the contrary, the same change of axle base improves the vibratory comfort at high velocities. And also on carbody center the axle base from 2.4 to 2.5m will lead to a Wz from 2.278 to 2.2601 and from 2.4 to 2.25m will lead Wz from 2.3 to 2.32.

At 80 km/h, there is a lower comfort index Wz by 0.8%, at carbody center and above bogie by about 2.43%.

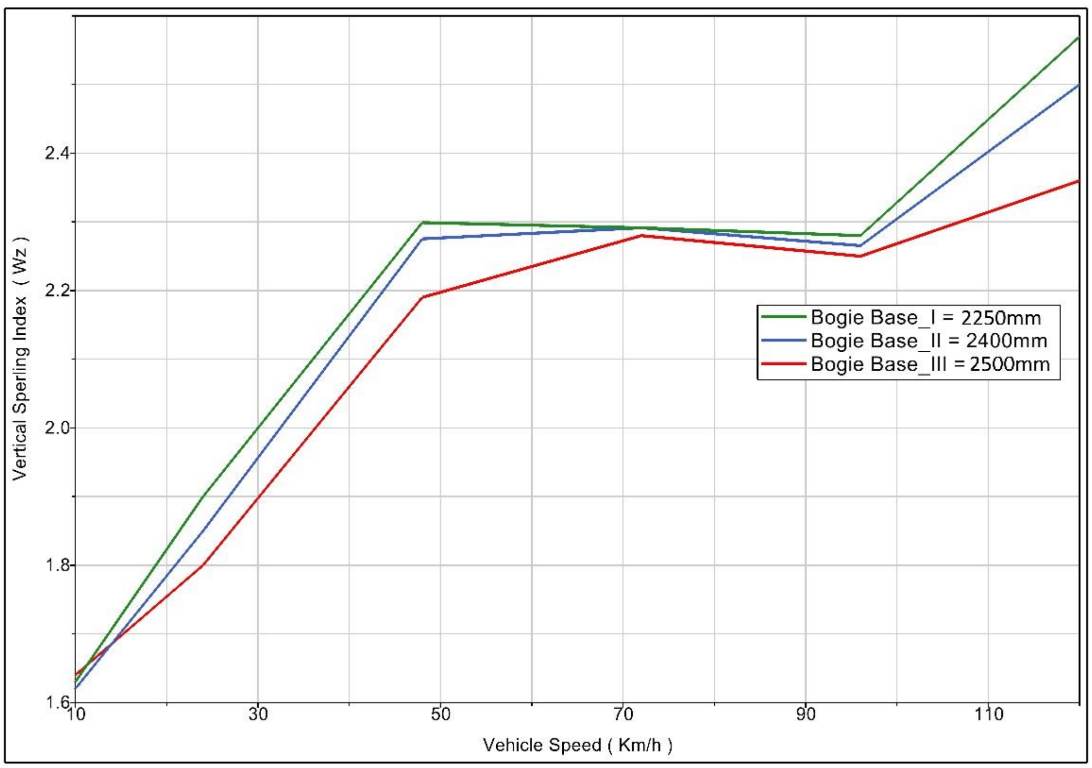

Figure 21: Influence of velocity and vehicle axel base upon Index *Wz* at car body center

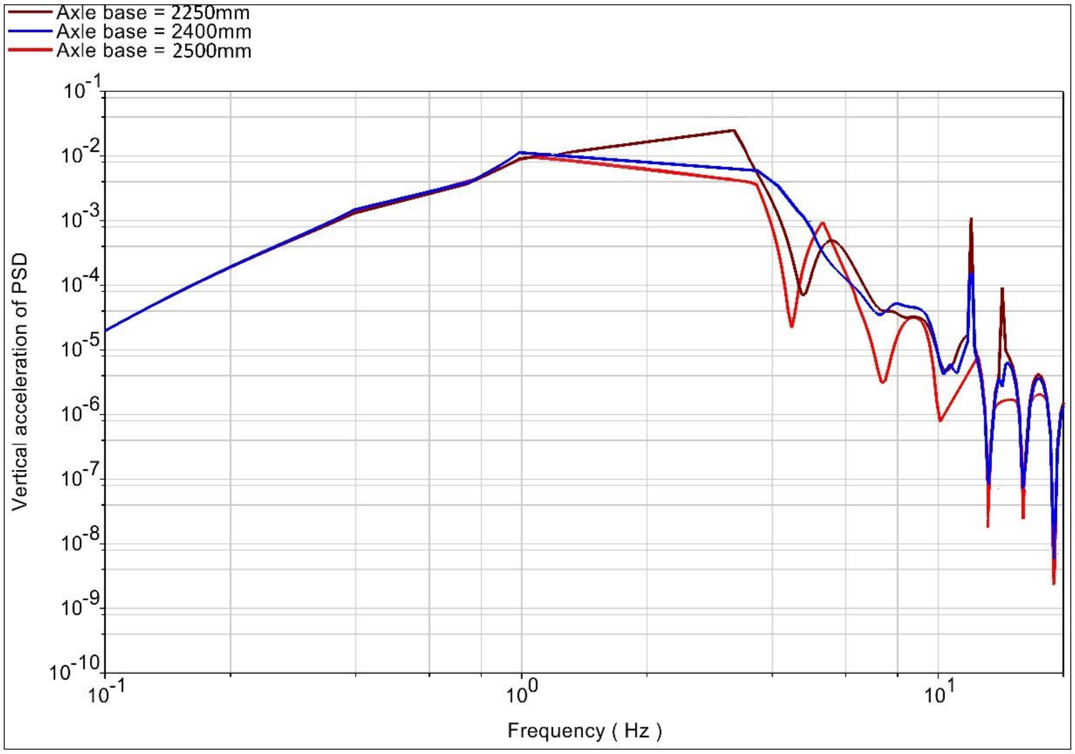

Figure 22: PSD diagrams of car body center

Fig 22 Show the relation between PSD of acceleration and frequency. The PSD of acceleration tends to decrease as frequency increase. For example when the axle base increase from 2.25m to 2.5m the PSD of acceleration decrease and hence the sperling comfort index become lower. As comfort index decrease the passenger comfort increase.

5.2.4 Carbody stiffness

From a ride comfort point of view it is preferable to increase the structural stiffness of the carbody in order to reduce the risk of carbody oscillation resonance. According to this concept the optimum solution is done by increasing the carbody cross-section from 3105mm to 3430 and by decreasing from 25500mm to 24250 the length of the carbody, as seen in figure 19.

When the frequency Hz value is from 0 to 1.5 Hz vertical acceleration of PSD is for both case are same but the frequency is above 1.6 the vertical acceleration PSD of high stiffness car body is decrease than the low stiffness car body.

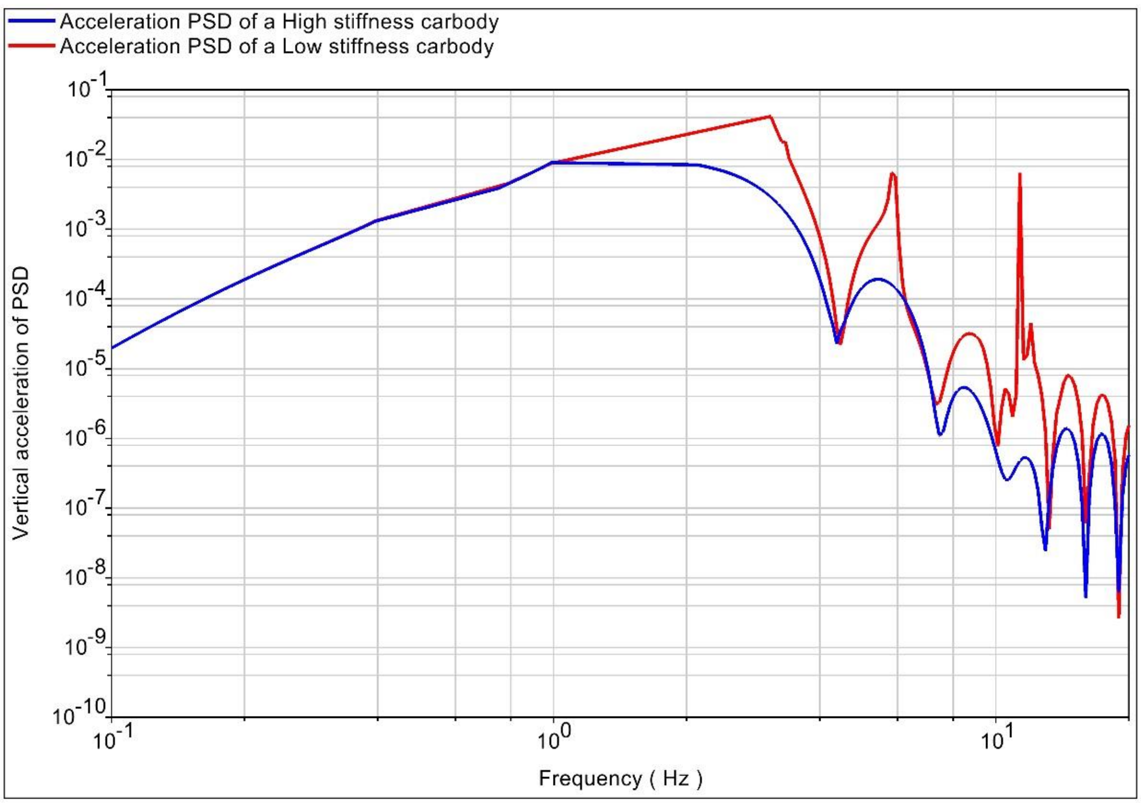

Figure 23: Carbody stiffness versus vertical acceleration of PSD at car body center

5.3 Comfort evaluation

As discussed in the above sections there are different methods to reduce vertical vibrations of the vehicle and hence to improve vertical ride comfort. The vehicle vertical vibrations is optimized by increasing spring and damping coefficient and by changing the structural stiffness and wheel base geometry as shown in the following figure 24,25 and table 8 .

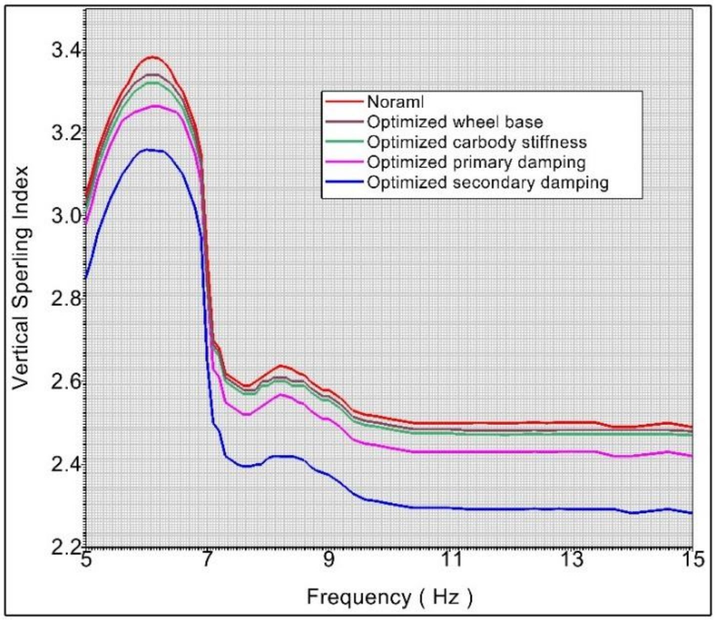

Figure 24: Vehicle Ride quality versus frequency at different parameters

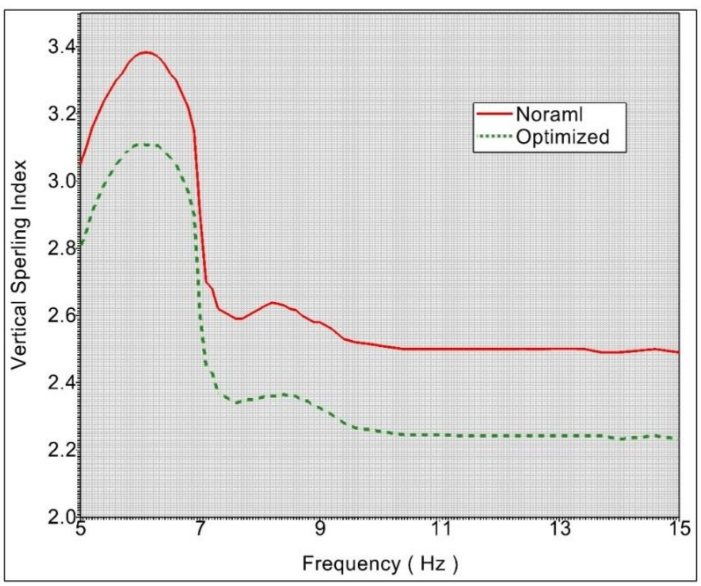

Figure 25: Vehicle Ride quality versus frequency

The optimum simulation results of ride index with structural stiffness and suspension parameters are shown in Figure 24. The optimum modifying suspension parameters and structural stiffness, passengers comfort is improved and the ride comfort index is optimized by the secondary damping about 8.72 %, primary damping about 3.4%, carbody stiffness about 1.24% and Wheel base (axle base) about 0.8%. Totally the raid comfort is reduced about 13.2 %.
Finally the optimum parameters result is describe in the following table 8.

Table 8: Normal and Optimized Parametric result.

Parameter	***Normal***	***Optimized***
Vertical damping coefficient of primary suspension	17.0kNs/m	20.625kNs/m
Vertical damping coefficient of secondary suspension	21.0kNs/m	28.45kNs/m
Wheel set (axle) base	2400mm	2500mm
Car body width	3105mm	3430mm
Carbody length	25500mm	24250mm

CHAPTER SIX

CONCLUSIONS, RECOMMENDATION AND FUTURE WORK

6.1 Conclusions, Recommendation

In this thesis the suspension system, wheel base and carbody stiffness for AALRT passenger trains was optimized in this paper. This was achieved by developing two multi-body dynamic models using SIMPACK software. The suspension and stiffness of one of the models was made constant according to AALRT data and the other was made variable. Both of the models allowed to move with the same velocity profile on a straight track. The suspension damping plays an essential role in terms of the carbody vibrations behavior and the ride comfort.

From the present work, which investigates these matters, several types of conclusions may be drawn as the following: -

- The analysis of the vibrations behavior due to the track random irregularities has proved that the level of vibrations increases along with the velocity and it is influenced by the geometric filtering effect given by the vehicle axle bases.
- A solution for improving the vibrations behavior based on the increase in the suspension floors damping has been considered limited.
- The selection of the vertical damping is made by starting with the idea to have the best comfort level at the carbody critical point.
- The optimum modifying suspension parameters and structural stiffness, passengers comfort is improved and the ride comfort index is optimized by the secondary damping about 8.72 %, primary damping about 3.4%, carbody stiffness about 1.24% and Wheel base (axle base) about 0.8%.
- The vehicle vertical vibrations (raid comfort) is can be optimized by increasing spring and damping coefficient and by changing the structural stiffness and wheel base geometry. In this thesis, the simulation results showed that the ride comfort index is improved about 13.2% which means it decreased from 2.5 to 2.17 with respect to the ISO standard.

6.2 Future work

The present study is an attempt to optimize the vertical vibration of rail vehicle by focusing primary and secondary damping, wheel base and carbody stiffness to get good passenger comfort.

Finally, some suggestions are listed below for future work as extension and continuity of this paper.

- The present study simulation done on a model developed by SIMPACK flexible bodies. To get more accurate result the flexible elements instead of rigid bodies could be another point of interest.
- Achieving good ride comfort is an important goal in the design of new competitive rail vehicles. So, this research can be extended to analyze the ride index and passenger comfort level by optimizing the spring stiffness.
- This research focused only the raid index optimized by primary and secondary damping ratio, wheel length and carbody stiffness. However, the model can be modified by optimizing the piezoelectric actuators and by using separated mass method.

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APPENDIX

Technical Description of Passenger Car

1. General Technical Specification

Car strength complies with TB/T1335-1996 《Railway vehicles Strength design and test evaluation specification》.

Vehicle dynamic performance complies with GB/T 5599-1985 《Railway Vehicle Dynamics Performance Evaluation and test identification norms》.

Vehicle gauge complies with GB146.1-198《Standard railway rolling stock gauge 》.

Service life: 30 years

Maintenance period:

- Maintenance back to factory: 2.4 (±0.6)M Kilometers or 10years.
- Maintenance in deport: 0.6(±20)M kilometers or 2.5years.

Conditions of Use:

- Maximum configuration: 20car per transit
- Ambient temperature: -40℃～+40℃
- Maximum relative humidity ≤95%

- Platform height, Suitable for platform height with 300mm 、 500mm and1250mm, distance between platform edge and railway center is 1750mm.
- Maximum line gradient ≤30‰

2. Main technical specification

- Railway gauge 1435mm
- Maximum operation speed 120km/h
- Emergency brake distance on straight line (with 30% overload and with initial speed 120km/h) ≤800m
- Minimum negotiable curve

  Single car 100m
- Coupling 145m
- Ride index W≤2.5
- Noise (120Km/h) ≤68dB（A）
- Body static heat transfer coefficient (K value) ≤1.16W/m2.k

Illumination complies with TB/T2917 － 1998 《Railway passenger coach illumination technical conditions》

- Axle load ≤17t

3. Main dimension

- Carbody length 25500mm

- Carbody width Approx 3105mm
- Height between rail top to car top(empty car) 4433mm
- Vehicle center distance 18000mm
- Height from rail top to coupler center (empty car) 880^{+10}_{-5} mm
- Height from rail top to inter car crossing pedal (empty car) 1333mm
- Height from rail top to floor surface (empty car) 1283mm
- Carbody center plate to rail top (empty car) 780mm

4. *Carbody steel structure*

Body steel structure will use overall carrying beam cylindrical structure, side wall will use flat plate. Thickness of sheets and profiles which ≤ 6mm will use nickel- chromium weathering steel, thickness which ≤ 2.5mm will use 05CuPCrNi, thickness which between 3 ~ 6mm will use 09CuPCrNi-B (Q295GNHL), plate thickness more than 6mm will use ordinary carbon steel. Air conditioning seat, toilet and wash room iron floor whichever easily corrosive will use stainless steel.

5. *Coupler and buffer device*

No.15 C-grade steel coupler, C-grade steel hook tail box and G1 buffer will be used.

6. *Gaugway and bridge plate*

Gaugway will use rubber material

7. *Bogie 209P bogie will be used*

- Construction speed: 140km/
- Axle load ≤17t, axle base: 2400mm
- Wheel is overall rolling steel, diameter is Φ915mm
- Proven hydraulic damper will be used.
- NJ3226X1, NJP3226X1 bearing will be used; Axle box is Metal labyrinth seals, and be installed with digital sensor for axle temperature alarming.

Bogie components will have good wear resistance, expect brake pads, hydraulic shock absorbers and wheel tread, the rest of the components will not required to replace or repair within 2 million kilometers operation.

Bogie safety performance

- Derail parameter ≤1.0
- Wheel load reduction rate ≤0.6
- Overturning coefficient ≤0.8

8 Drawings

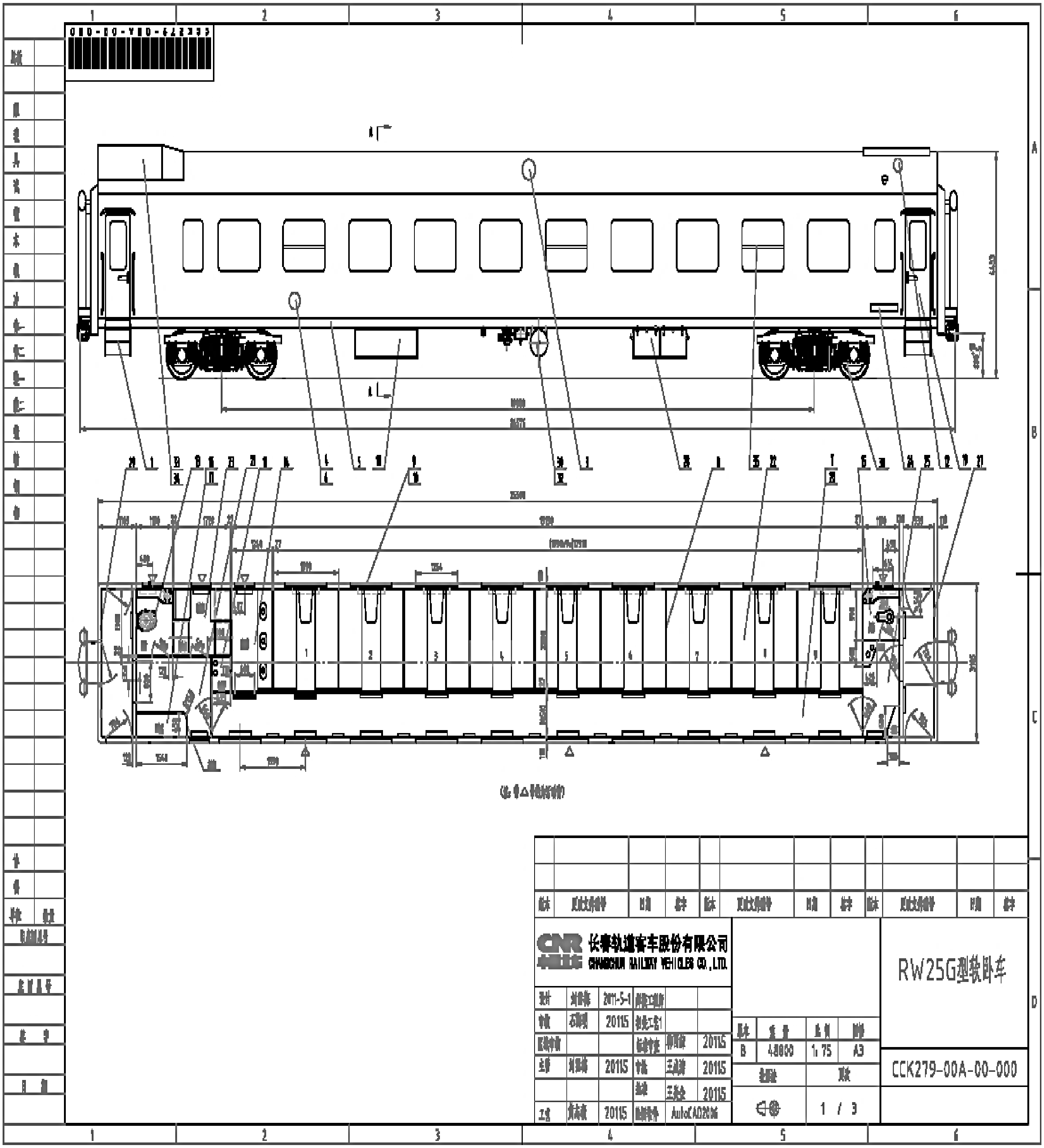

Figure 26: Interior layout drawing

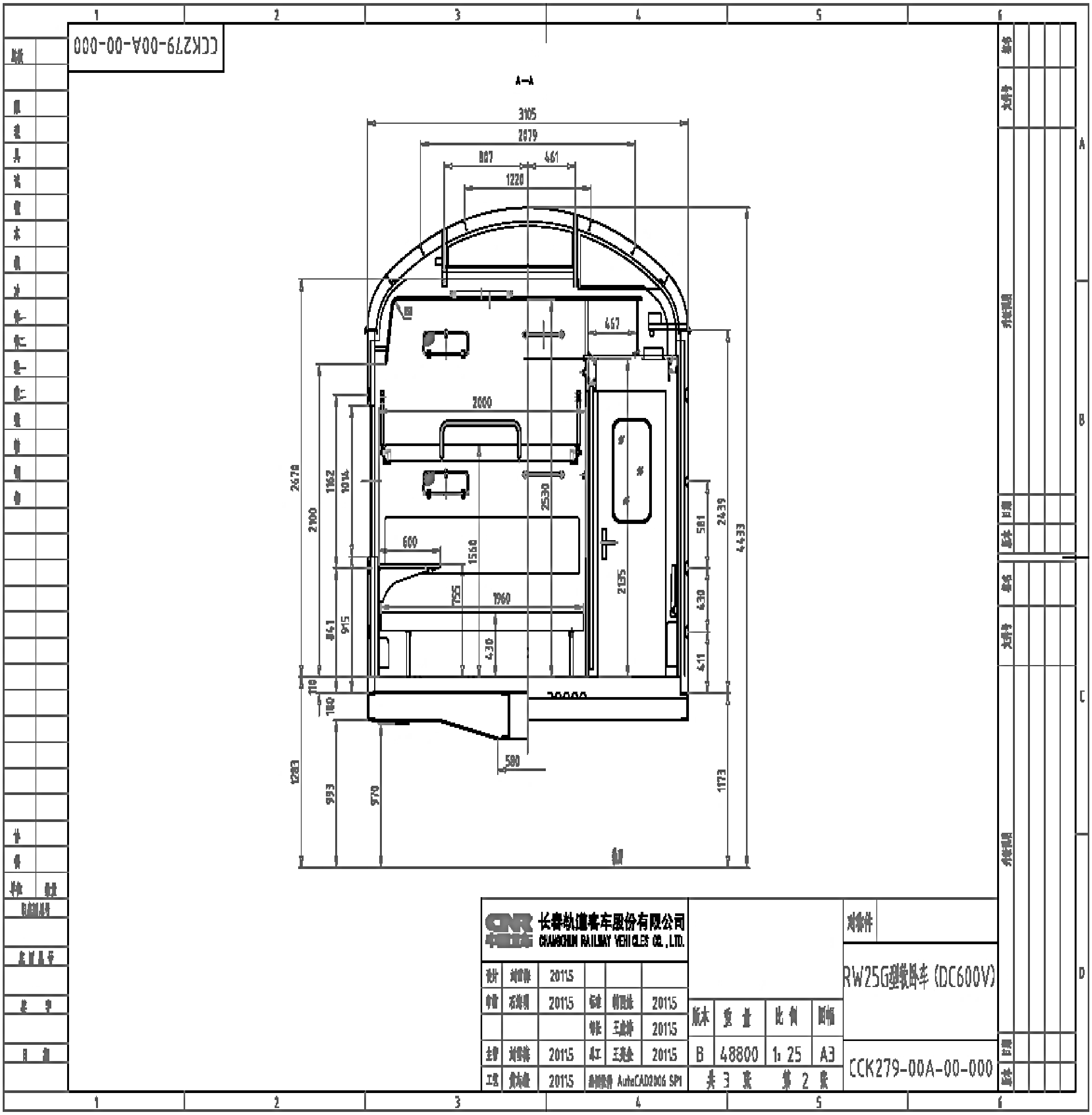

Figure 27: Car interior/exterior pictures

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