B. TECH. PROJECT REPORT

On

Study of Suspension Elements of Railway Vehicle

BY

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DISCIPLINE OF MECHANICAL ENGINEERING

INDIAN INSTITUTE OF TECHNOLOGY INDORE

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Study of Suspension Elements of Railway Vehicle

A PROJECT REPORT

Submitted in partial fulfillment of the requirements for the award of the degrees

of
BACHELOR OF TECHNOLOGY
in

MECHANICAL ENGINEERING

Submitted by:
Vijay Singh Meena
Shailesh Kumar Meena

Guided by:
Dr. Pavan Kumar Kankar

INDIAN INSTITUTE OF TECHNOLOGY INDORE
December 2019

CANDIDATE’S DECLARATION

We hereby declare that the project entitled **“Study of Suspension Elements of Railway Vehicle”** submitted in partial fulfillment for the award of the degree of Bachelor of Technology in ‘Mechanical Engineering’ completed under the supervision of **Dr. Pavan Kumar Kankar (Associate Professor),** IIT Indore is an authentic work.

Further, we declare that we have not submitted this work for the award of any other degree elsewhere.

Vijay Singh Meena **Shailesh Kumar Meena**

CERTIFICATE by BTP Guide

It is certified that the above statement made by the students is correct to the best of my knowledge.

Dr. Pavan Kumar Kankar

Associate Professor, Discipline of Mechanical Engineering

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Preface

This report on “Study of Suspension Elements of Railway Vehicle" is prepared under the guidance of Dr. Pavan Kumar Kankar.

Through this report, we have tried to give a detailed design of suspension systems currently working in Indian railways with freight vehicles being of prime focus.

We have tried to the best of our abilities and knowledge to explain the content in a lucid manner. We have also added figures and experimental data to make it more illustrative.

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We wish to thank Dr. Pavan Kumar Kankar for his kind support and valuable guidance. We would also like to acknowledge Dr. Sanjay Shukla (RDSO) for providing his sincere cooperation and valuable guidance to carry out this project.

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Without their support, this report would not have been possible.

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Abstract

Better ride quality is one of the most important requirements for damage-free transportation of goods and services. Ride index is measured to get an indication of ride quality; the lower ride index indicates better ride quality, which improves with the effectiveness of a suspension system. The dynamic behavior of a vehicle depends on its suspension system; mainly the primary and secondary suspension system. In the present project, a parametric analysis of suspension elements of a freight vehicle has been performed to analyze the ride index of the vehicle. The secondary suspension system mainly consists of 3 springs: inner, outer and snubber springs. The values of spring stiffness and damping coefficients have been obtained from RDSO (The Research Design & Standards Organizations). These parameters have been used to generate a Simulink model on Matlab taking track irregularities as the main input. These parameters are then sequentially varied to study their effects on vertical acceleration and ride index. A new set of parameters has been suggested based on the results of the parametric analysis to get a better ride quality.

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Chapter 1 Introduction

1.1 Introduction to Basic Suspension System

Suspension systems have evolved significantly over the last few generations from horse-driven buggies to self-powered automobiles with the new technologies playing a big role in it, but the basic needs being the same. The basic requirements from today's suspension systems are safe handling and maximum traction while not compromising with passenger comfort. To achieve these goals, modern suspension systems depend upon various types of springs, shock absorbers, and other components.

All the components in a suspension system must work together to produce proper ride quality and handling characteristics as expected for safer and damage-free transportation and passenger comfort. Each component of the suspension system plays an important role in the overall system. If one part of the system fails to fulfill its role, it can lead to heavy and faster damages to other components as well and ultimately the whole suspension system comes apart. Hence, leading to serious damages to the vehicle and compromising passenger comfort. Therefore, a very illustrative understanding of each component of the suspension system and how it functions as part of the whole system is critical.

❖ Functions of Suspension System

All the suspension systems have some basic functions to perform, regardless of the type of suspension system or the vehicle they are used in.

- The suspensions system should allow the springs and dampers to absorb the energy of a bump or irregularities on the track for a smooth, comfortable and safe ride while not allowing uncontrolled movements of the tires.
- The suspension must handle movements produced because of vehicle acceleration, braking, and cornering.
- The springs used must be able to safely carry the weight of the vehicle. Its failure results in a very tough ride and may put extra loads and stresses on other components of the system as well.

- One of the most important roles of the suspension system is to keep the alignment of the tires as correct as possible under the worst scenarios as well, so that maximum contact is maintained between the tires and the road.

Therefore weight carrying, ride control, comfort, and handling must be balanced while designing a vehicle.

1.2 Objective and Motivation Behind Project

The main motive for designing a suspension system is to improve the ride comfort by absorbing the shocks due to the irregularities of the road. These complex vibrations need to be contained or it might in serious damages to the vehicle affecting both life and stability of the vehicle. Therefore we need an effective suspension system to successfully contain the vibrations to improve the ride quality and ensuring a long life for the vehicle.

To contain the vibrations produced due to track irregularities, we want the suspension system to firstly limit the vertical acceleration experienced by the carbody. Higher values of vertical acceleration for carbody means the goods and services transported through freight vehicles will be bouncing inside the carriage, which we do not want. Hence, limiting the vertical acceleration is one of the prime focuses of this project.

Ride quality refers to the effectiveness of a vehicle's suspension system in minimizing the vibrations so as to provide comfort for the people. Hence, we want to find the ride index of the vehicle which accounts for the ride quality of the vehicle.

For further improvements in ride quality, we want to sequentially vary the parameters to check their effect on ride index and suggest a new set of parameters that gives us the best ride quality for safely transporting goods and services.

Chapter 2 Indian Railway Freight Vehicle

2.1 Introduction

Indian railways are a vital sector for the economic development of our country. Movement of freight vehicles that carry goods and services contribute big sums to revenue generation. Hence, these vehicles need the best suspension systems. Cast steel CASNUB 22HS (three-piece bogie) is the commonly used bogie in Indian railway freight vehicles.

2.2 Three Piece Bogie

Three-piece bogie vehicles find maximum use in Indian railways freight stocks.

Three-piece bogie mainly consists of 3 main components-

- Bolster
- side frames
- wheel-set

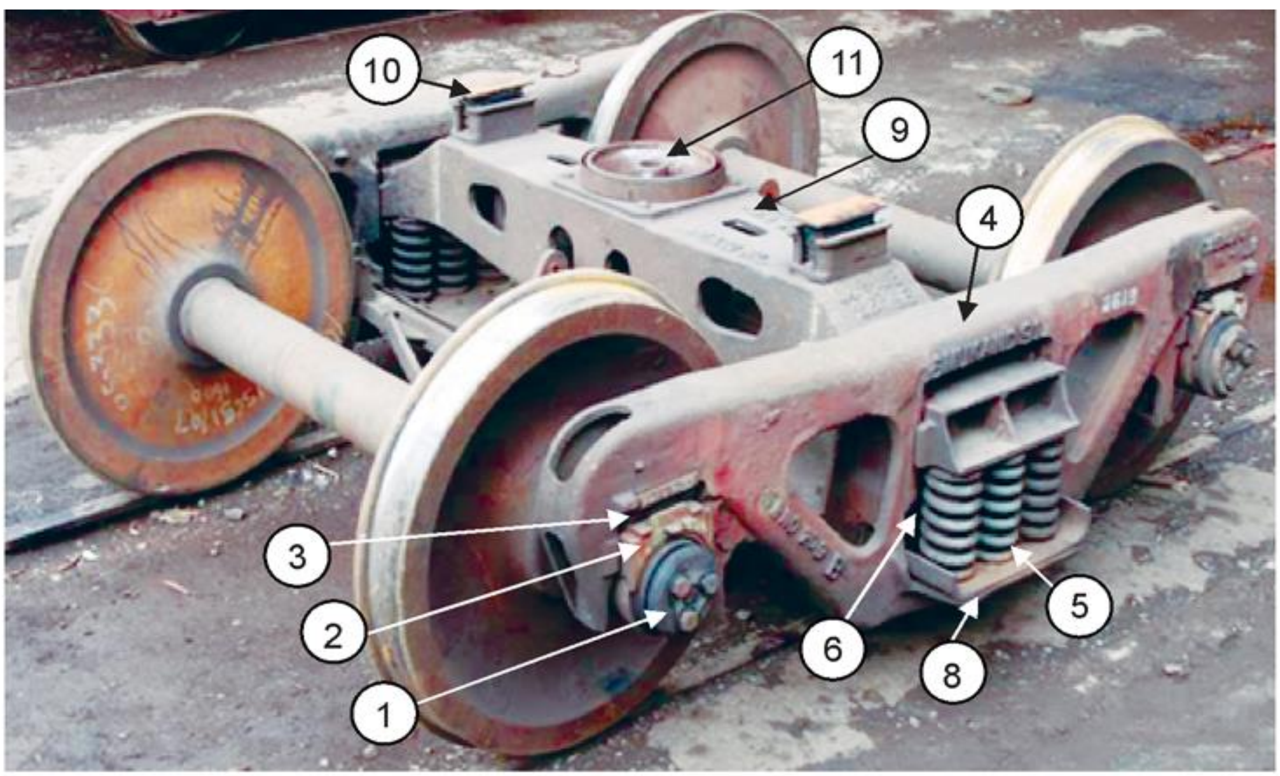


Fig 2.1 Casnub bogie [5]

Different parts labeled in figure 2.1 of casnub bogie are listed in table 2.1 -

Table 2.1 Bogie components

Number	Bogie component
1	Bearing
2	Bearing adapter
3	Elastomeric pad
4	Side frame
5	Outer springs
6	Snubber springs
8	Spring plank
9	Bolster
10	Side bearer
11	Centre pivot

In the Casnub bogie shown in figure 2.1, the bolster is coupled to 2 side frames via secondary suspensions. Side frames are connected to the wheel-set through primary suspension. Side bearers and center pivots can be seen over bolster where wagons are loaded. Outer springs of secondary suspensions are visible, where inner springs are covered by outer springs.

Functions of bogie components-

- **Bolster:** It is a component in bogie which is a part of the interface between bogie and carbody. It is rested on secondary suspension springs.
- **Centre pivot:** Load from the wagon is transported to bolster through central pivot which is provided at the center of the bolster.
- **Side bearers:** It accounts for countering oscillations developed during eccentric loading due to vibrations produced.
- **Side frames:** It covers a major part of freight wagon bogies, it transmits load on the axle through axle box.
- **Snubber springs:** It counters the oscillations produced due to random vibrations.

2.3 Sketch of Indian Railway Freight Vehicle

Front view **Side view**

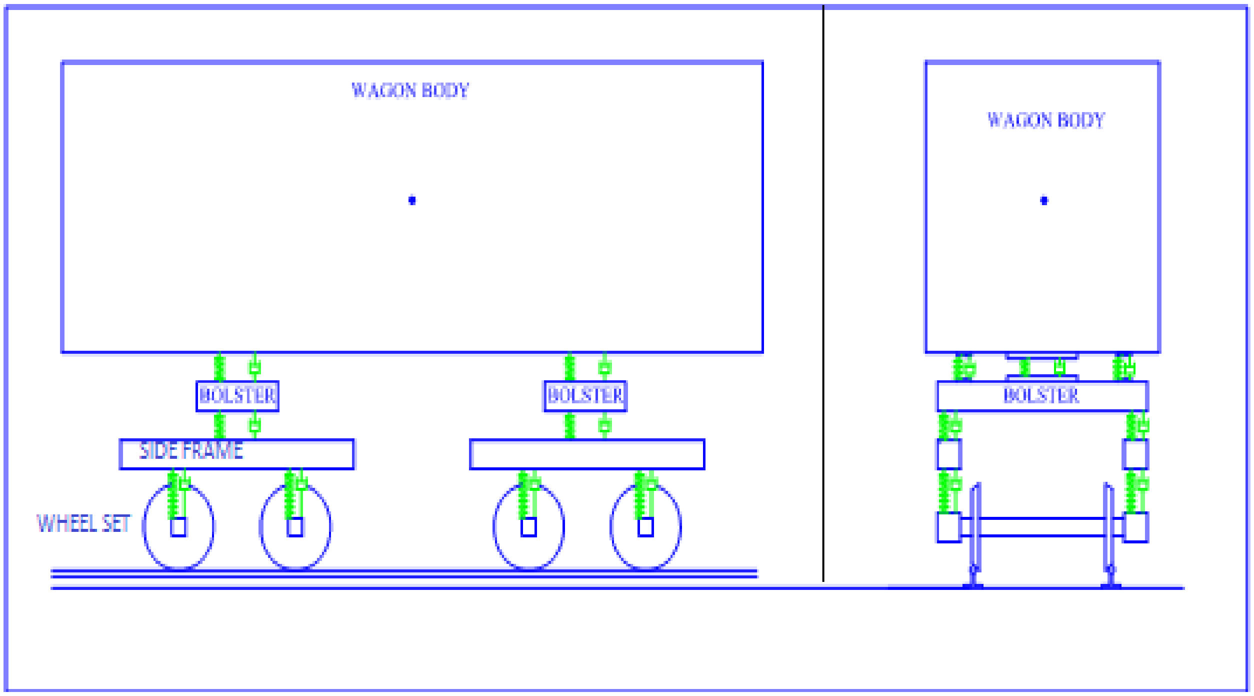


Fig 2.2. Indian railway freight vehicle

Indian railway freight vehicle shown in figure 2.2 includes both front and side views. It shows the different bogie components attached to each other with springs and dampers. Primary suspension springs and dampers can be seen between wheel-set and side frames. Secondary suspension springs are between side frames and bolster. The wagon is loaded at center pivot where side bearings help in eccentric loading, they can be seen between bolster and wagon.

Chapter 3 Suspension System for Freight Vehicle

3.1 Introduction

The track irregularities which basically include rough and uneven roads, stones and other unwanted things whose interaction with the wheel of the vehicle generates random vibrations of varying amplitude and frequency. These unwanted vibrations may cause damage to the vehicle and also affect the stability of the vehicle. Nowadays, all vehicles whether cars, trucks, planes, and trains include suspension systems consisting of springs, dampers and other components assembled to the vehicle parts such that it reduces the effect of vibrations generated through irregularities of the track.

Different vehicles use different types of suspension systems, which have evolved significantly over the last few generations due to advancements in technology. Hence, in this project, the suspension system of an Indian railway freight vehicle has been analyzed. The suspension elements used are briefly explained in chapters 3.2 and 3.3 along with figures to get a better understanding of how these suspension elements are assembled.

The suspension system of an Indian railway freight vehicle mainly consists of primary and secondary suspension systems.

3.2 Primary Suspension System

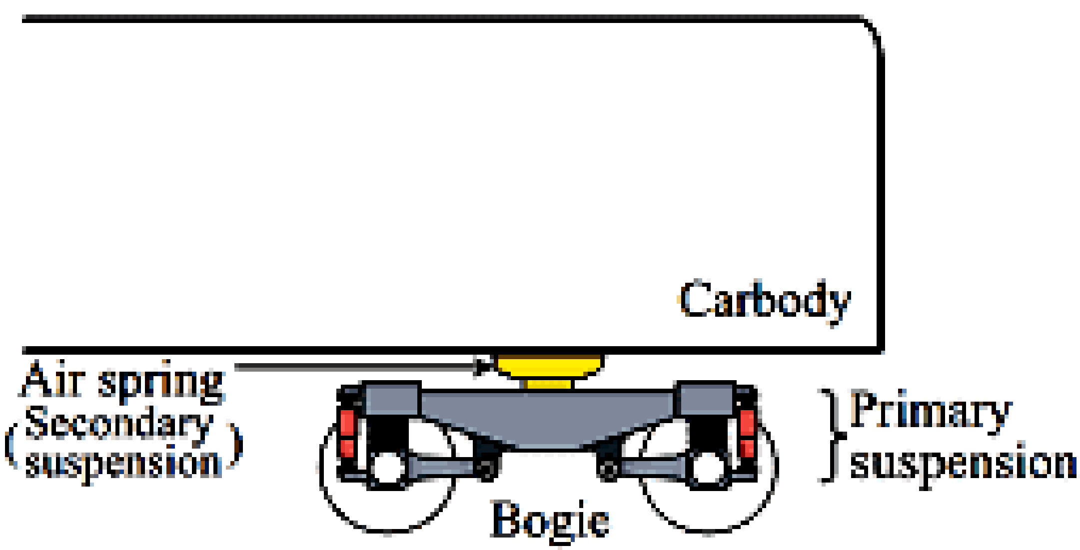


Fig 3.1 Suspension system [9]

The primary suspension in an Indian railway freight vehicle is present between the two side frames and wheel-set as shown in the figure in figure 3.1. The vertical vibrational forces experienced by wheels are transferred to the side frames through the primary suspension system. As these vibrations are directly transferred from the wheel-set, its magnitude will be high and needs to be controlled by the next suspension elements used before reaching the wagon.

Fig 3.2. Elastomeric pad [5]

The elastomeric pad is generally used as a primary suspension element as shown in figure 3.2. Its major role is to withstand the longitudinal, compressive and shear vibrational forces experienced by the bogie, help in cushioning and provide damping effect to the whole wagon.

3.3 Secondary Suspension System

The Secondary suspension system in an Indian railway freight vehicle is present between the side frames and bolster as shown in figure 3.1. The major suspension components in the secondary suspension system are:

- Outer spring
- Inner spring
- Snubber spring
- Wedge block

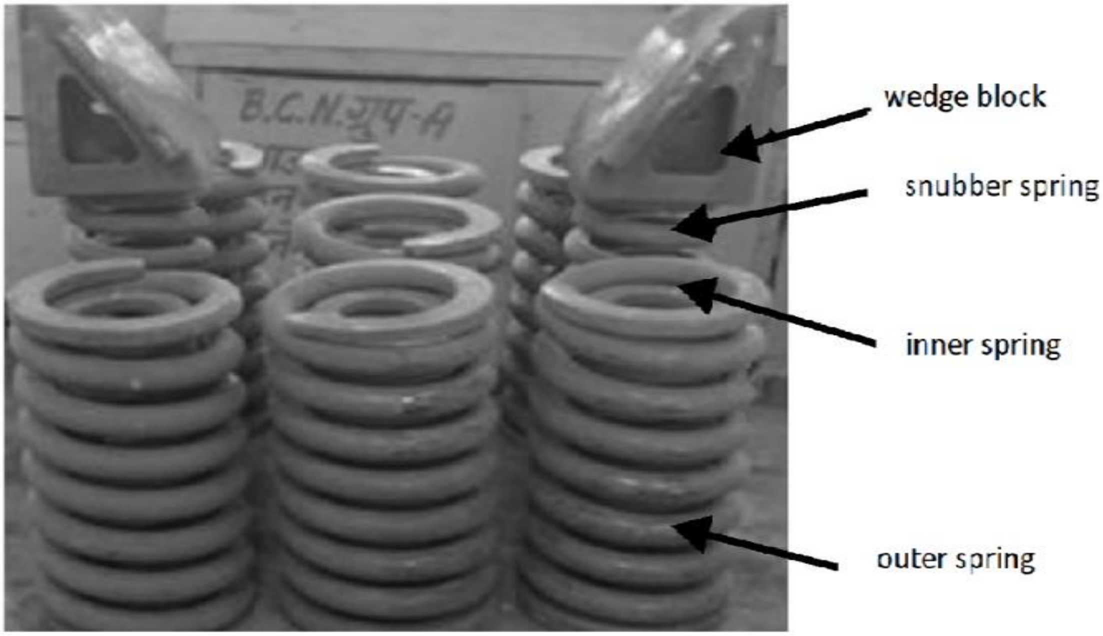


Fig 3.3. Secondary suspension springs arrangement [2]

A secondary suspension system is a group of nested springs in parallel such that each nest consists of 3 types of springs. The arrangement of springs in the nest, as shown in figure 3.2, is such that the outer spring covers the inner spring. Snubber springs account for oscillations produced due to random vibrations. A wedge block placed over two snubber spring is used as a coulomb friction damping element.

Fig 3.4 Side frame and spring plank [5]

The two nests of springs are rested over the left and right ends of the spring plank, where each nest includes 7 inner, 7outer and 2 snubber springs. The spring nests also provide vertical support between the side frames and bolster.

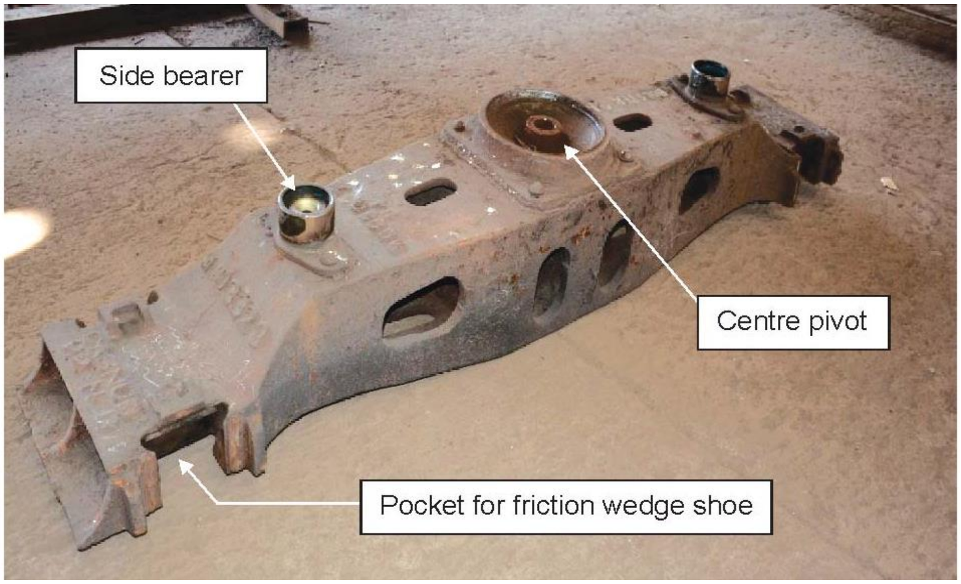


Fig 3.5 Bolster [5]

The slant surface of the wedge block used as Coulomb friction damping element is in contact with *pocket for friction shoe* at bolster and the vertical side is in contact with side frame as shown in figure 3.4. The wagon load is applied at the center pivot and the two side bearers account for eccentric loading due to random vibrations.

The weight of the bolster activates the snubber springs which counters oscillations. When the wagon is empty, only outer springs are handling the load. But, when the wagon is loaded, both inner and outer springs share the load.

Chapter 4 Analytical Modeling of Railway Suspension System

4.1 Introduction

Analytical modeling involves mathematical modeling of a system to study its dynamic response. A set of free body diagrams and governing equations are created to generate the response of the system. These equations, with the help of graphical block diagraming, are converted into a model where necessary inputs are provided to generate the response of the system.

4.2 Mechanical System and Equations

The entire suspension system of an Indian railway freight vehicle is converted into a mechanical system consisting of different bogie components attached with springs and dampers. The inputs for vibrations are applied at the wheel-set and its effect on the whole system is studied.

In the present work, two cases of vibrational inputs are considered:

I. **Harmonic vibrations:** Vibrational inputs are given in the form of sinusoidal waves, where a fixed frequency of vibrations are considered. The amplitude of the input vibrations is also fixed. These vibrations are no are not practical, as no track can be perfectly manufactured such that it will induce vibrations of the same amplitude and frequency. This vibrations are not seen anywhere and hence not practical. This case is considered to only study the dynamic response of the suspension system under the simplest of vibrations.

Therefore to analyze such systems including harmonic vibrations, a set of free body diagrams and equations are formed for the mechanical system as shown in figure 4.1. The mechanical system includes springs and dampers and other bogie components:

Carbody	M1
Bolster	M2
Side frame	M3
Wheel set	M4
Side bearers	(K1,C1) and (K3,C3)
Center pivot	(K2, C2)
Secondary suspension	(K4,C4) and (K5,C5)

Secondary suspension	(K6,C6) and (K7,C7)
Hertzian springs	(K8, K9)

The mechanical system for harmonic vibrations:

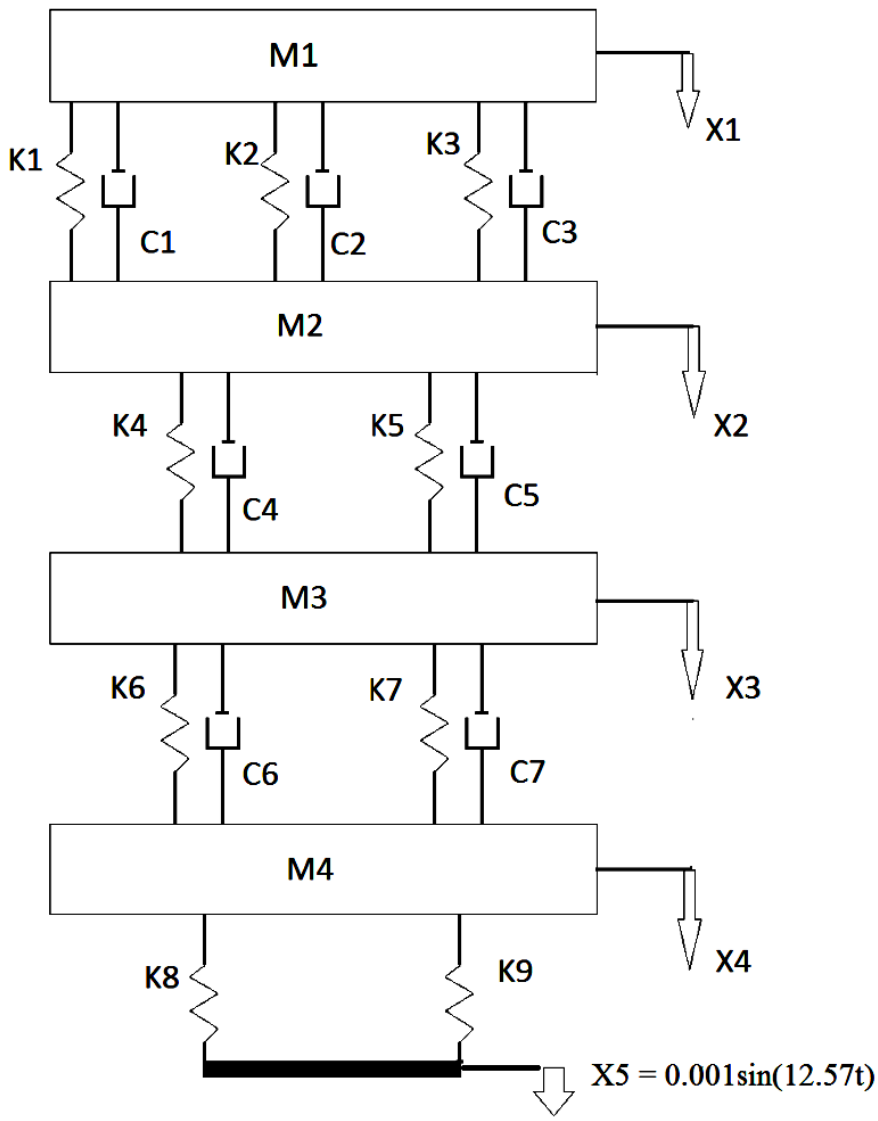


Fig 4.1 mechanical suspension system (Case I)

Figure 4.1 is the mechanical system for the suspension system of an Indian railway freight vehicle for harmonic vibrations. The figure includes carbody M1 loaded on bogie at center pivot (K2, C2) while side bearers (K1, C1 & K2, C2) counters eccentric lading. Center pivot and side bearers are located on bolster M2. Secondary suspension spring nest (K4, K5) provides vertical support between bolster and side frame M3. The elastomeric pad is used as a primary suspension between the side frame and wheel-set M4. Hertzian springs (K8, K9) are used to represent the stresses developed between two surfaces of different radii. The input signal in the form of sin wave is given at the wheel-set, which is in direct contact with the track irregularities.

❖ Equations for Vertical Acceleration (CASE I)

Free body diagrams (F.B.D) of all the bogies are constructed and equations for vertical acceleration are found.

1. CARBODY

Free body diagram of carbody (M1) is shown in figure 4.2. The suspension elements in contact with the carbody are center pivot and side bearers. Hence, spring and damping coefficients are of center pivot and side bearers appear in equation 4.1. The displacement (x1), velocity ($x1^*$) and acceleration ($x1^{**}$) of the carbody are also included in the equation along with its mass M1.

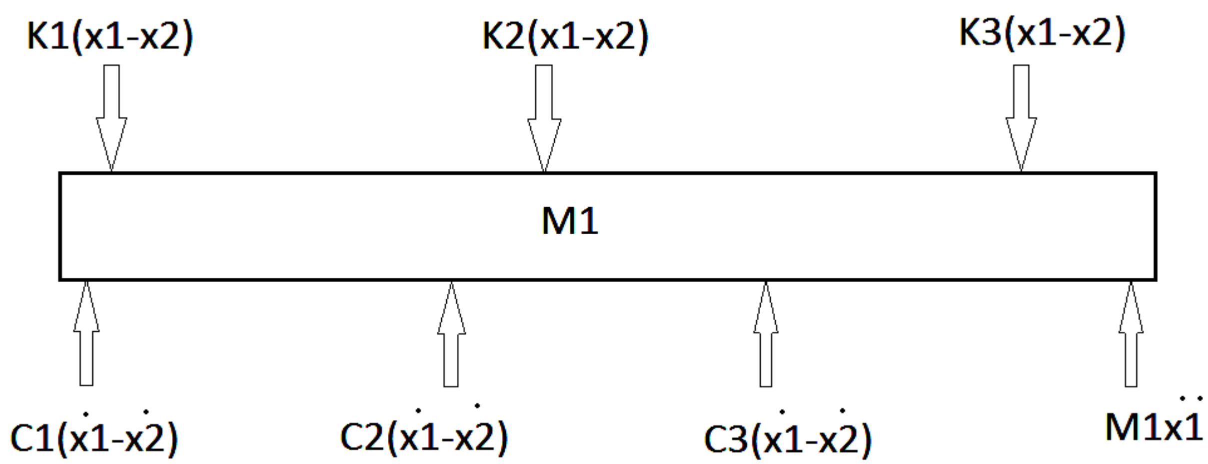


Fig 4.2 Free body diagram for carbody (M1)

The equation for vertical acceleration is:-

$$\ddot{x}1 = \frac{1}{M1}\left\{\begin{matrix} -K1(x1-x2) - C1(\dot{x}1-\dot{x}2) - K2(x1-x2) - C2(\dot{x}1-\dot{x}2) - K3(x1-x2) \\ -C3(\dot{x}1-\dot{x}2) \end{matrix}\right\} \qquad (4.1)$$

2. BOLSTER

Free body diagram for bolster (M2) is shown in figure 4.3. The suspension elements in contact with bolster are the center pivot, side bearers and secondary suspension, hence their spring and damping coefficients appear in equation 4.2. The displacement for bolster is given by (x2), velocity by ($x2^{*}$) and acceleration by ($x2^{**}$).

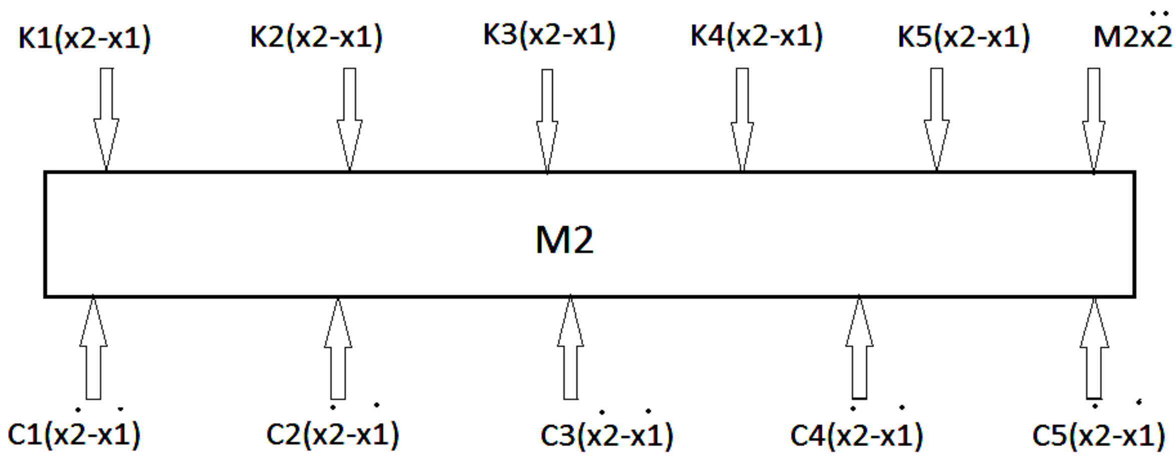


Fig 4.3 Free body diagram for bolster (M2)

The equation for vertical acceleration is:-

$$\ddot{x}2 = \frac{1}{M2}\begin{Bmatrix} -K1(x2-x1) - C1(\dot{x}2-\dot{x}1) - K2(x2-x1) - C2(\dot{x}2-\dot{x}1) - K3(x2-x1) \\ -C3(\dot{x}2-\dot{x}1) - K4(x2-x3) - C4(\dot{x}2-\dot{x}3) - K5(x2-x3) - C5(\dot{x}2-\dot{x}3) \end{Bmatrix} \quad (4.2)$$

3. SIDE FRAME

Free body diagram for side frames (M3) is shown in figure 4.4. The suspension elements in contact are secondary and primary suspension system, hence their spring and damping coefficients appear in equation 4.2. The displacement for side frames is given by (x3).

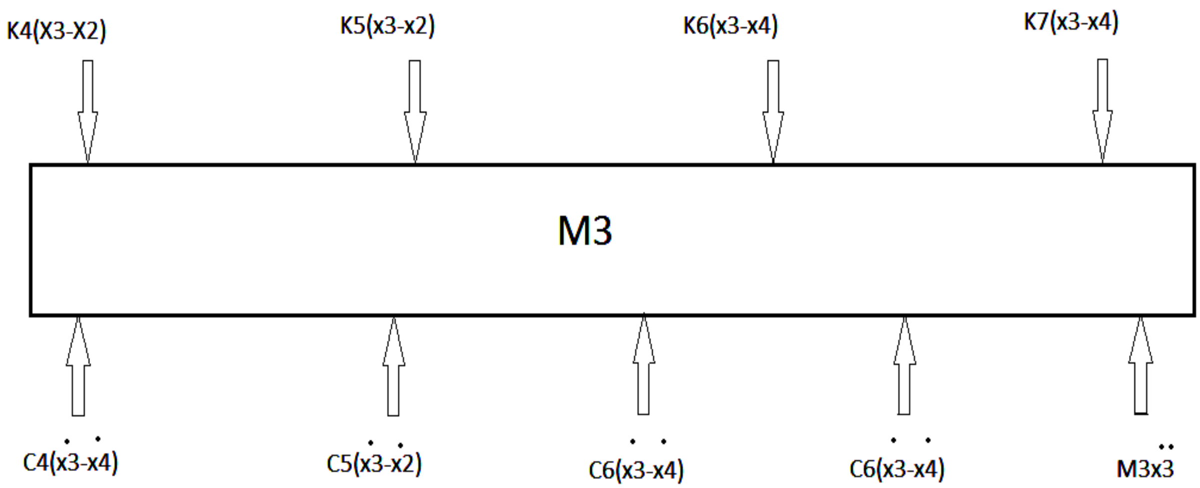


Fig 4.4 Free body diagram of side frame (M3)

The equation for vertical acceleration is:-

$$\ddot{x3} = \frac{1}{M3}\begin{Bmatrix} -K4(x3-x2) - C4(\dot{x3}-\dot{x2}) - K5(x3-x2) - C5(\dot{x3}-\dot{x2}) - K6(x3-x4) \\ -C6(\dot{x3}-\dot{x4}) - K7(x3-x4) - C7(\dot{x3}-\dot{x4}) \end{Bmatrix} \quad (4.3)$$

4. WHEEL

Free body diagram for wheel-set (M4) is shown in figure 4.5. The suspension elements in contact with the wheel-set are primary suspension and track. The contact between wheel-set and track is represented by hertzian springs, hence their spring and damping coefficients appear in equation 4.2. The displacement for wheel-set is given by (x4), velocity by ($x4^{*}$) and acceleration by ($x4^{**}$).

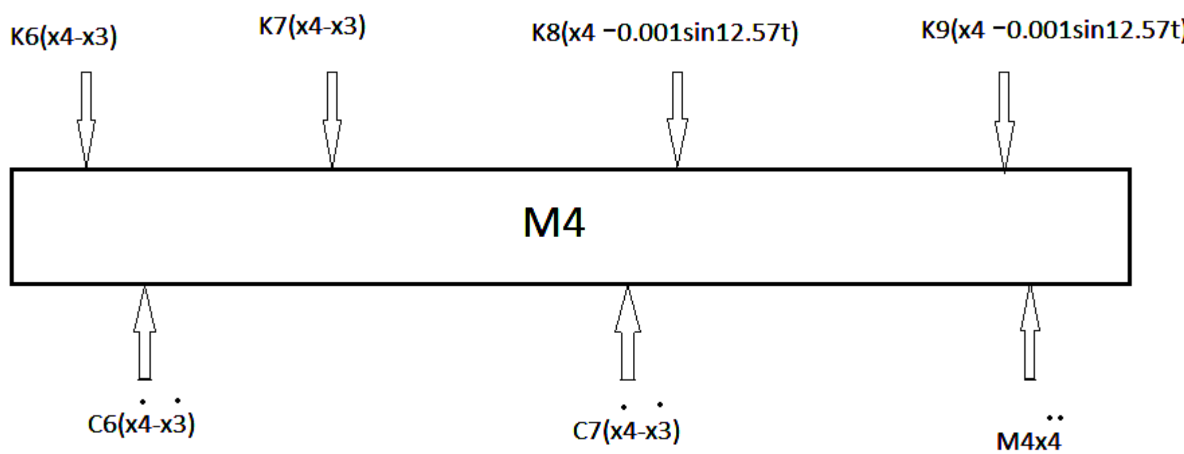


Fig 4.5 Free body diagram of wheel (M4)

The equation for vertical acceleration is:-

$$\ddot{x4} = \frac{1}{M4}\begin{Bmatrix} -K6(x4-x3) - C6(\dot{x4}-\dot{x3}) - K7(x4-x3) - C7(\dot{x4}-\dot{x3}) \\ -K8(x4 - \text{left rail}) - K9(x4 - \text{right rail}) \end{Bmatrix} \quad (4.4)$$

II. Random vibrations: In this case, the actual vibrations that are seen practically are considered. These random vibrations arise due to irregularities of the track, which includes roughness, scratches, uneven track, etc. These vibrations occur at varying amplitudes and frequencies, making it hard for the suspension system to limit it to safety. This happens because rail tracks are always in open, hence develop scratches, irregularities which leads to random vibrations in practical life.

Therefore to analyze such systems including random vibrations, a set of free body diagrams and equations are formed for the mechanical system as shown in figure 4.6. The mechanical system includes springs and dampers and other bogie components with input signals being the actual track irregularities, which are discussed in chapter 4.3.

The mechanical system for random vibrations case is:

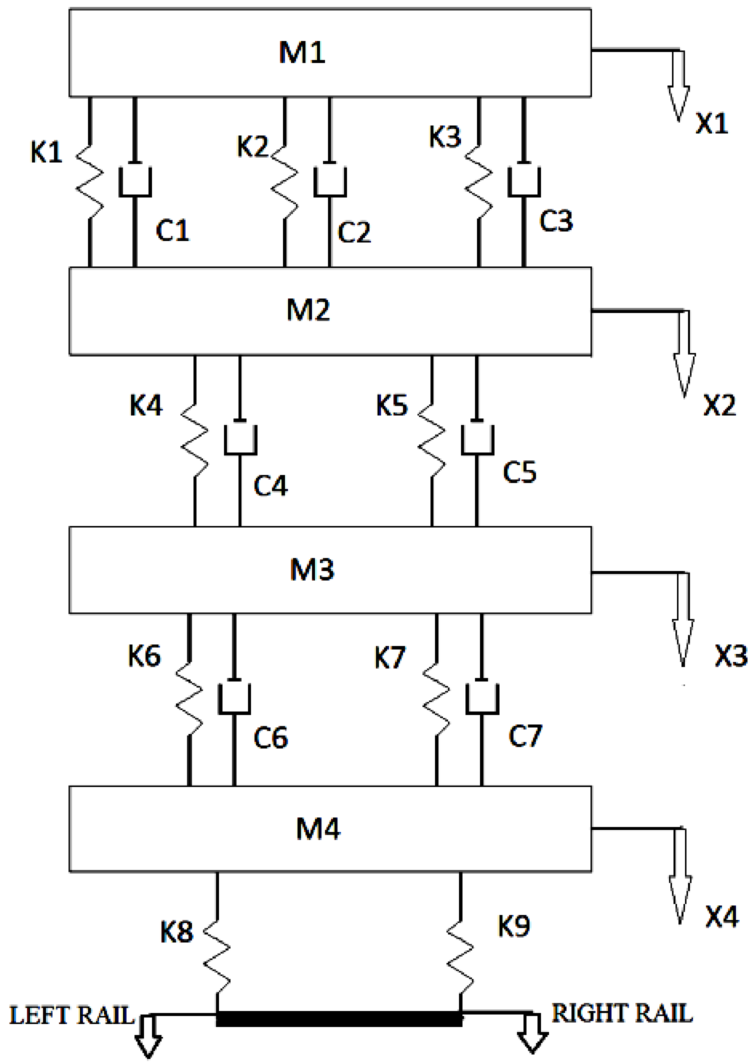


Fig 4.6 Mechanical suspension system (Case II)

Figure 4.6 is the mechanical system for the suspension system of an Indian railway freight vehicle for random vibrations. The assembly of bogie components and suspension elements remain exactly the same as figure 6.1 (mechanical suspension system for harmonic vibrations). Only the input signals which are applied at the wheel-set are changed from sinusoidal in case I to random vibrations in case II. These random vibrations due to track irregularities are applied separately for the left and right rails as shown in figure 4.7.

As the arrangement of suspension springs and bogie components is same, the equations for vertical acceleration of the bogie components are same except wheel-set. The free body diagram as shown in figure 4.7 of wheel-set will also include actual track irregularities for left and right rails as input data instead of a sinusoidal wave for harmonic vibrations case.

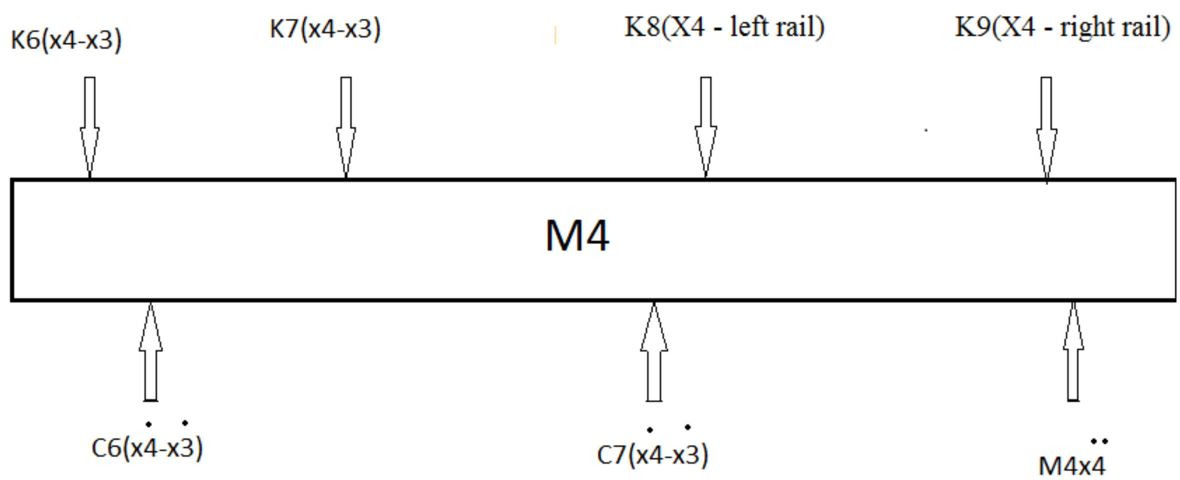


Fig 4.7 Free body diagram for wheel (M4) (Case II)

4.3 Track Irregularities(for case II)

In practical life, every vehicle that runs on a surface experiences random vibrations due to irregularities of the surface. This irregularity maybe because of the unevenness of the surface, scratches or roughness. Hence, the vehicle experiences vertical excitations due to these irregularities, which we call random vibrations as the excitations are of random amplitude and varying frequency. Higher the roughness or irregularities of the surface, larger the amplitude of vertical excitations. Therefore these vibrations need to be controlled to maintain vehicle stability and ensure good ride quality or passenger comfort.

Different vehicles use different types of suspension systems depending on the surface they run on and the load they carry. Therefore, suspension requirements for a train and a car are different as they run on different surfaces and carry different loads. A train usually carries the load in tones and runs on the rail track.

The actual track data for left and right rail:

- RIGHT RAIL:

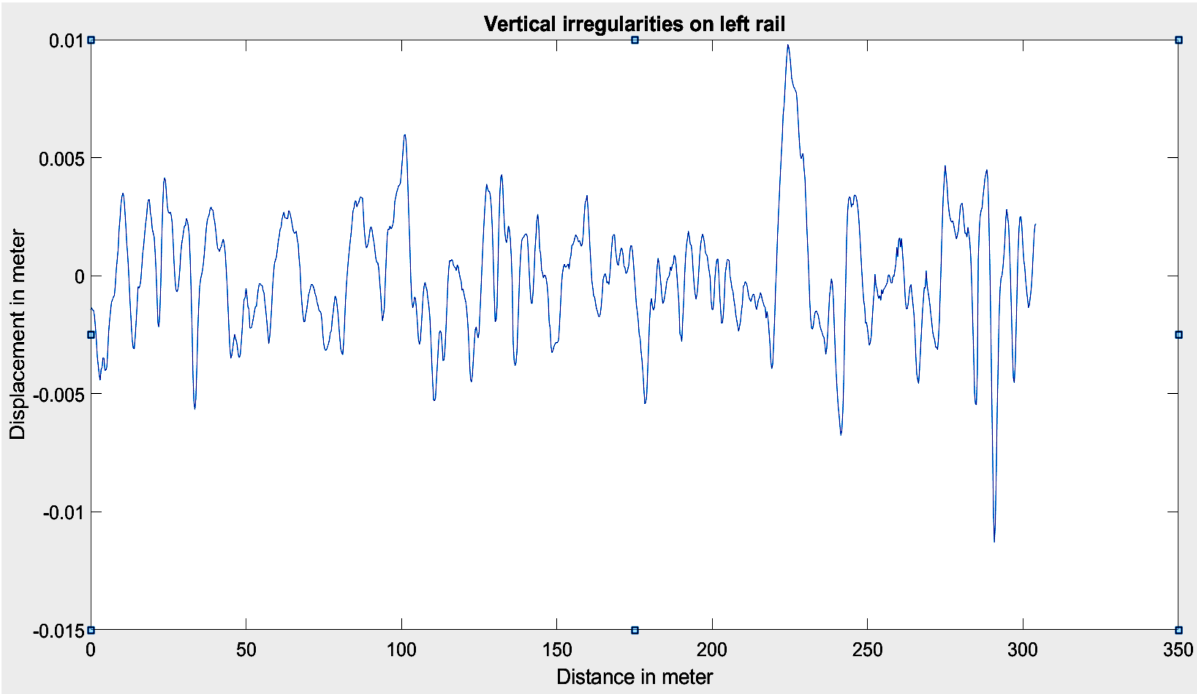


Fig 4.8. Track irregularities for the right rail

Figure 4.8 represents irregularities on the right rail of the track. The X-axis represents track length in meters whereas Y-axis represents the unevenness or irregularities present on the right rail. This are actual irregularities that cause vibrations in trains. As it can be seen in figure 4.8, the irregularities are random and hence it causes random vibrations of varying amplitude and different frequencies. Higher the irregularities, larger the amplitude of vibrations experienced by the vehicle. This nature of the track is given as input to the Simulink model to generate a response of the system in the form of acceleration of the bogie components. As the irregularities increases, it also increases the amplitude of vertical acceleration of the bogie components

- LEFT RAIL:-

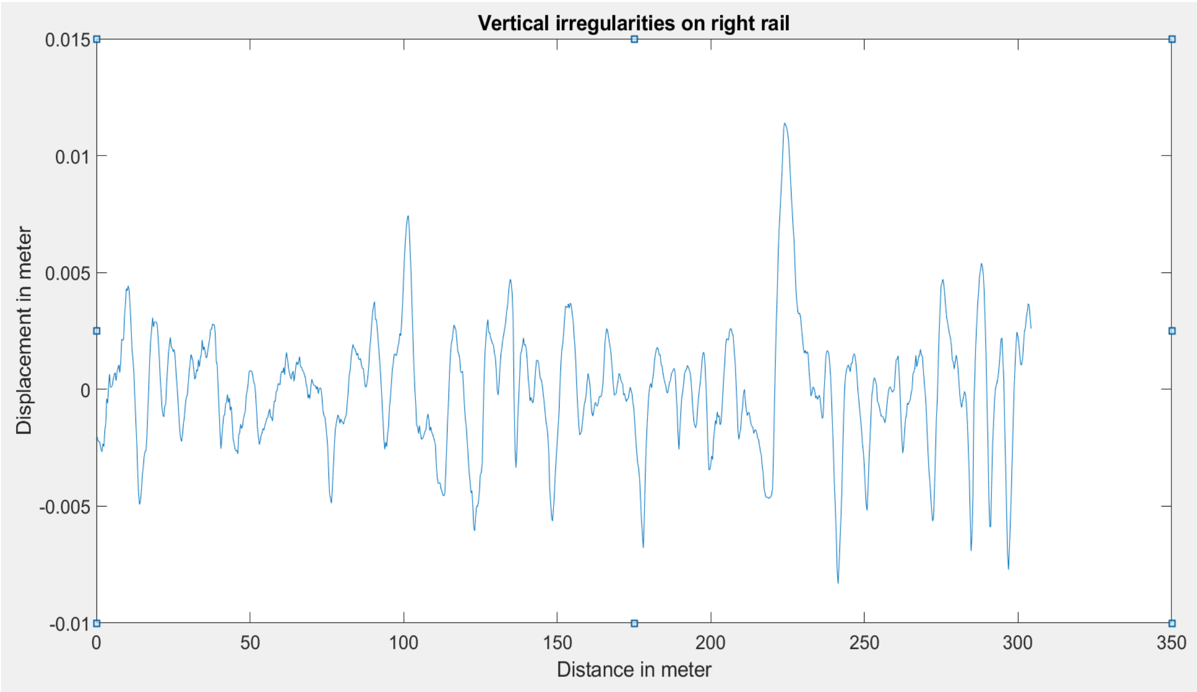


Fig 4.9 Track irregularities for left rail

Figure 4.9 represents irregularities on the left rail of the track. The X-axis represents track length in meters whereas Y-axis represents the unevenness or irregularities present on the right rail. Due to the randomness of the irregularities, the vertical excitations experienced by the bogie components are also. Hence, these vertical excitations cause vibrations of varying amplitude and frequency. These vibrations as input to the Simulink model gives a response in terms of vertical acceleration of the bogie components. As the irregularities increases, it also increases the amplitude of vertical acceleration for all the bogie components**.**

Chapter 5 Simulink Modeling for Suspension System

5.1 Introduction to Basic Simulink Model

Simulink is a software developed for simulation and model-based design of dynamic systems in Matlab. Simulink, which is also developed by Math works, is a graphical programing language tool for modeling, simulating and analyzing dynamic systems. Basically, Simulink is a tool that uses graphical block diagramming to generate models for complex looking systems. The data flow in the form of graphical programing makes even the complex systems look very easy. Simulink is also provided with a Simulink library browser which includes some predefined blocks that can be directly used in a model.

Simulink also allows a user to use Matlab algorithms in a model as well as the simulation can be exported into Matlab for further analysis. For a graphical response of a model developed in Simulink, all the data points can be easily obtained by exporting the same graph to Matlab workspace in the form of an array.

Therefore, in Simulink, any type of complex system can be easily solved through graphical block diagramming and simulated at any point of time

Simulink also supports –

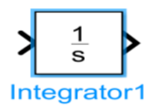
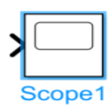
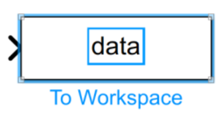
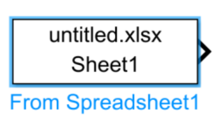
- System-level design
- Simulation
- Automatic code generation
- Testing and verification of embedded systems

5.2 Simulating Suspension System in Simulink

The suspension system for Indian railway freight vehicle consists of primary and secondary suspensions which basically includes springs and dampers. The governing equations for the vertical acceleration of carbody, bolster, side frame and wheels are to be simulated in Simulink through graphical block diagramming.

The blocks used in the model from the Simulink library browser are –

Table 5.1 Simulink blocks

Block	Symbol	Function
1. Add		To add inputs
2. Gain		To multiply the inputs by a constant
3. Integrator		To integrate a signal
4. Scope		To display signals generated upon simulation
5. Mux		To combine multiple input signals into a single output
6. To Workspace		To transfer data to Matlab workspace in the form of an array
7. From Spreadsheet		It takes input of track irregularities in the form of excel file

The track irregularities that cause vibrations are to be given as an input signal to the model to get the vertical acceleration of the carbody as an output of the simulation, run over a specific period of time.

The Simulink model generated is a combination of the pre-defined blocks, listed in table5.1, from the Simulink library. GAIN block is used to multiply a constant to the signal and ADD block is used to add multiple signals. These signals are then passed through INTEGRATOR 1 to get velocity and INTEGRATOR 2 to get the displacement of the bogie components. These displacement signals for all the bogie components are converted into a single output by MUX and the output in form of vertical accelerations for bogie components is displayed through SCOPE, whose data is then transferred to Matlab workspace in form of an array to calculate frequency spectrum for ride index calculation.

- **Case I: Harmonic Vibrations**

The input signals in form of track irregularities for harmonic vibrations are given as sinusoidal wave at the wheel-set. Hence, a model is created to simulate the suspension system at different speeds and the response of the system is calculated in the form of vertical acceleration for bogie components. Hence, to provide input signal of track irregularities, SINE WAVE block is used from the Simulink library browser as shown in figure 5.1, and the fixed amplitude and frequency are provided from equation 5.1.

Input sine wave $=0.001 \sin 12.57t$ (5.1)

where, the amplitude of 1mm and 2 Hz frequency is given as input vibration.

- **Case II: Random Vibrations**

The input signals in the form of track irregularities for actual vibrations are given at the wheel-set. Therefore, an input signal of random vibrations is provided through FROM SPREADSHEET block in Simulink library browser as shown in figure 5.2. It takes input in the form of an excel file, including vertical track irregularities at different track lengths. A Simulink model is generated for the same to get output in of vertical acceleration of all the bogie components. The amplitude and frequencies are not fixed here, as the track irregularities are random.

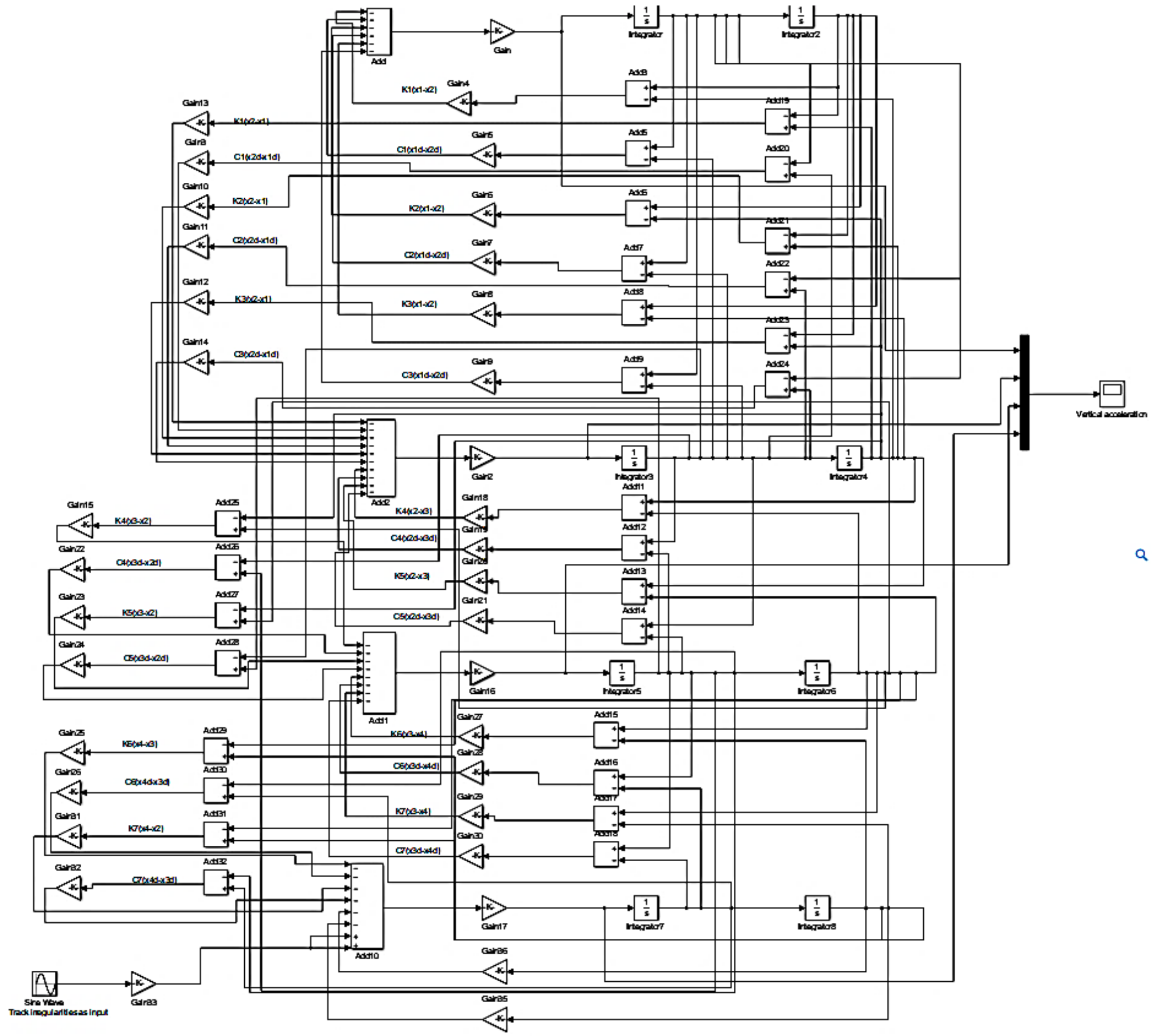


Fig 5.1 Simulink model for CASE I(harmonic vibrations)

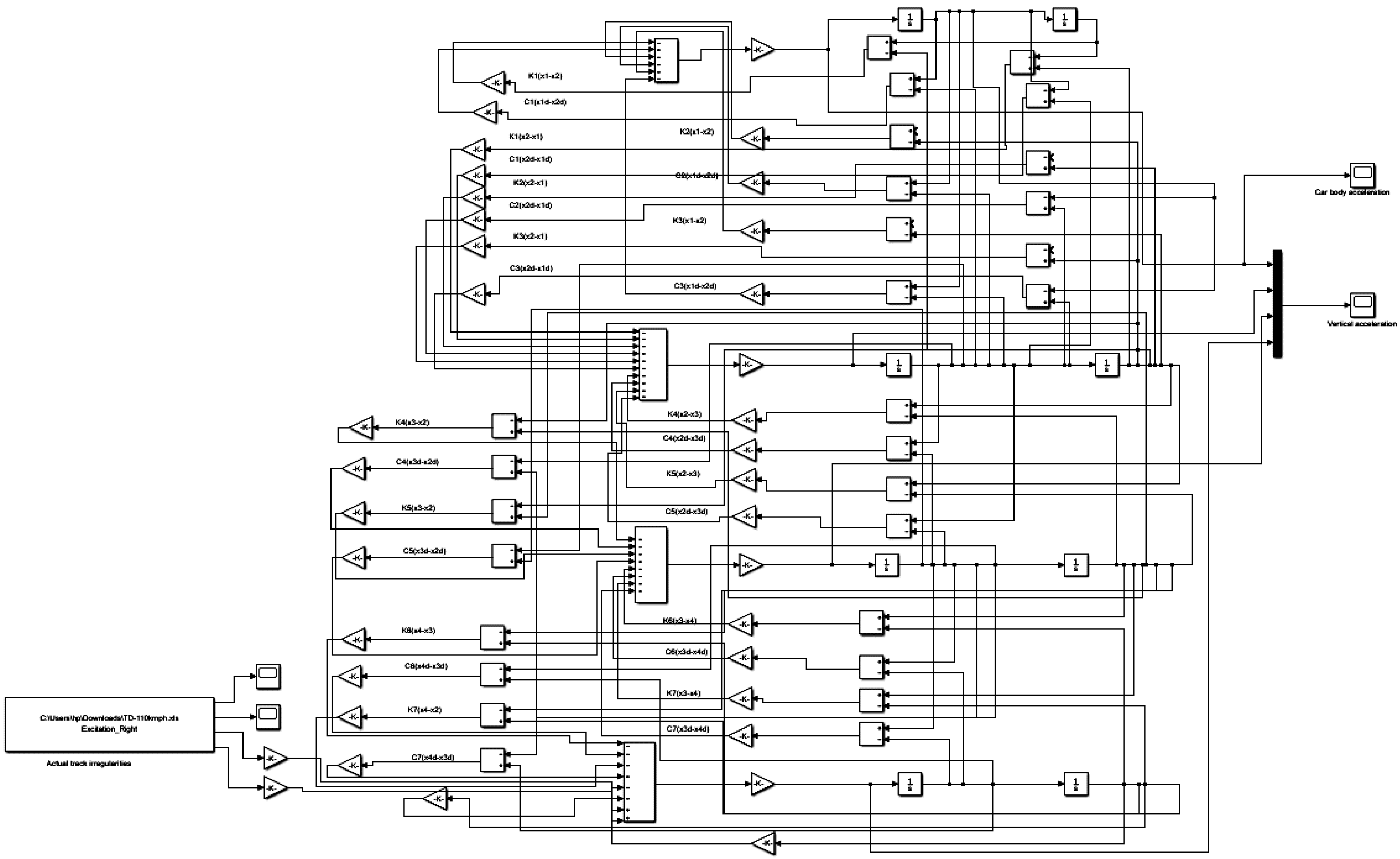


Fig 5.2 Simulink model for CASE II(random vobrations)

Chapter 6 Rise Index Analysis

6.1 Introduction to Ride Index

Ride index is a unit-less parameter that gives us an indication of ride quality as shown in table 6.1.

Table 6.1 Ride index appreciation

Appreciation	Ride index
Very good	1
Good	2
Satisfactory	3
Able to run	4
dangerous	5

Hence, better ride quality means a lower ride index. The ride quality of a vehicle depends on displacement, acceleration, rate of change of acceleration and other factors.

6.2 Ride Index Formula

- **Sperling's ride index (Wz)**

It is the ride index used by Indian railways to evaluate ride quality and ride comfort. Ride comfort implies that the vehicle is being assessed according to the effect of the mechanical vibrations on the human body, whereas ride quality implies that the vehicle is itself being judged.

Sperling's ride index is originally defined as:

For ride quality as, $W_z = 0.896(a^3/f)^{\frac{1}{10}}$ (6.1)

where 'a' denotes amplitude of the vertical acceleration and 'f' denotes frequency.

But, in practical situations vibration does not occur at the same frequency (harmonic vibrations), instead, they are random vibrations occurring at random frequencies. Hence, ride index is to be calculated for each of those frequencies using the formula as shown in equation 6.2,

$$W_z = (a^3 B(f)^3)^{1/10} \quad (6.2)$$

where, B (f) is a weighting factor, a frequency-dependent factor that expresses human vibration sensitivity.

$$B(f) = k\left[\frac{[(1-0.56f^2)^2 + (0.645f)^2(3.55f^2)]}{[(1-0.252f^2)^2 + (1.547f - 0.00444f^3)^2](1-3.55f^2)}\right]^{1/2} \tag{6.3}$$

The vehicle body vibrates not at a single frequency but over a whole spectrum of frequencies. So ride index needs to be calculated for the whole spectrum using the below formula,

$$W_z = \sqrt[10]{\sum_{i=1}^{n_f} {W_{z_i}}^{10}} \tag{6.4}$$

6.3 Vertical Acceleration and Frequency Spectrum

The two most vital requirements for calculating the ride index are peak vertical acceleration of the carbody and frequency of the vibrations. Hence, the values of peak vertical acceleration for carbody are obtained from acceleration plots of carbody for both cases, upon simulation of the model.

- **Case I: Harmonic Vibrations** The nature of vertical acceleration for carbody as obtained from the output of the Simulink model upon simulation is:

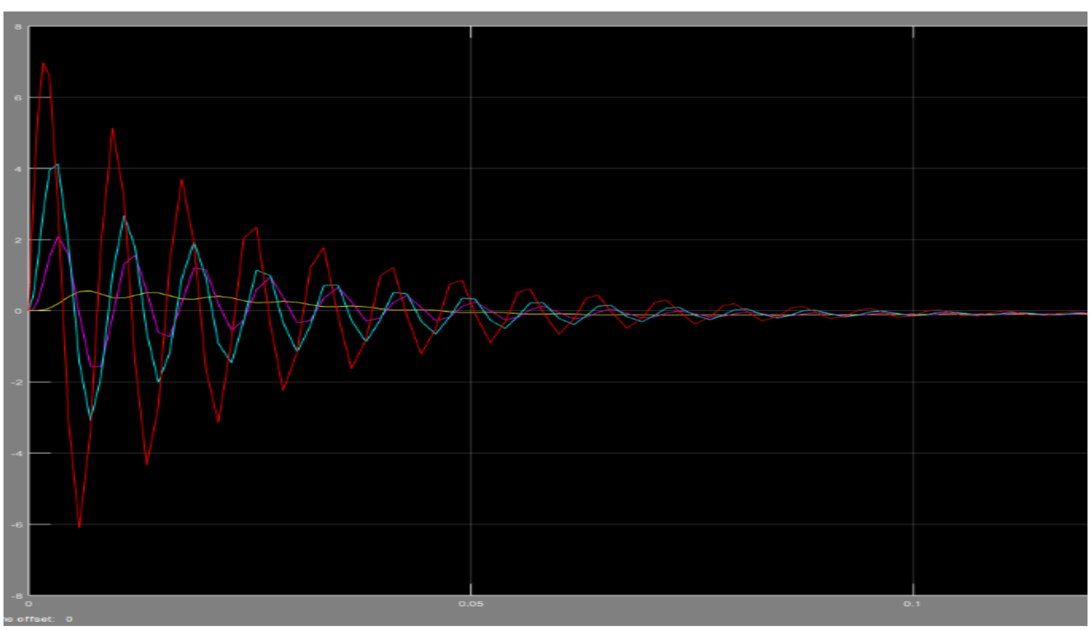


Fig 6.1 Vertical acceleration of bogie components (Case I)

Figure 6.1 shows the variation of vertical acceleration of all bogie components. The lines representing the vertical acceleration of bogie components are:

Wheel-set: red

Side frame: blue

Bolster: pink

Carbody: yellow

The simulation is carried out at 3 different fixed frequencies over a time period. The values of peak vertical acceleration recorded for all bogie components at different fixed frequencies are:

Table 6.2 vertical acceleration for bogie components (Case I)

Bogie component	Peak vertical acceleration (m/sec^2)		
	1 Hz	1.5 Hz	2Hz
Wheel-set	4.75	6.97	9.25
Side frame	2.85	4.13	5.61
bolster	1.42	2.06	2.75
carbody	0.37	0.56	0.75

Table 6.1 shows the values of peak vertical acceleration for all bogie components at 3 different fixed frequencies. It can be seen from figure 6.1 and table 6.1, that values of peak vertical accelerations for a given bogie component increases as we increase frequency. Also, at a fixed frequency, the values of peak vertical acceleration for carbody are least whereas for the wheel-set are highest. Hence, the values of vertical acceleration for carbody shows that most of the vibrations have been contained by the suspension system.

- **Case II: Random vibrations**

The nature of vertical acceleration for carbody as obtained from the output of the Simulink model upon simulation is**:**

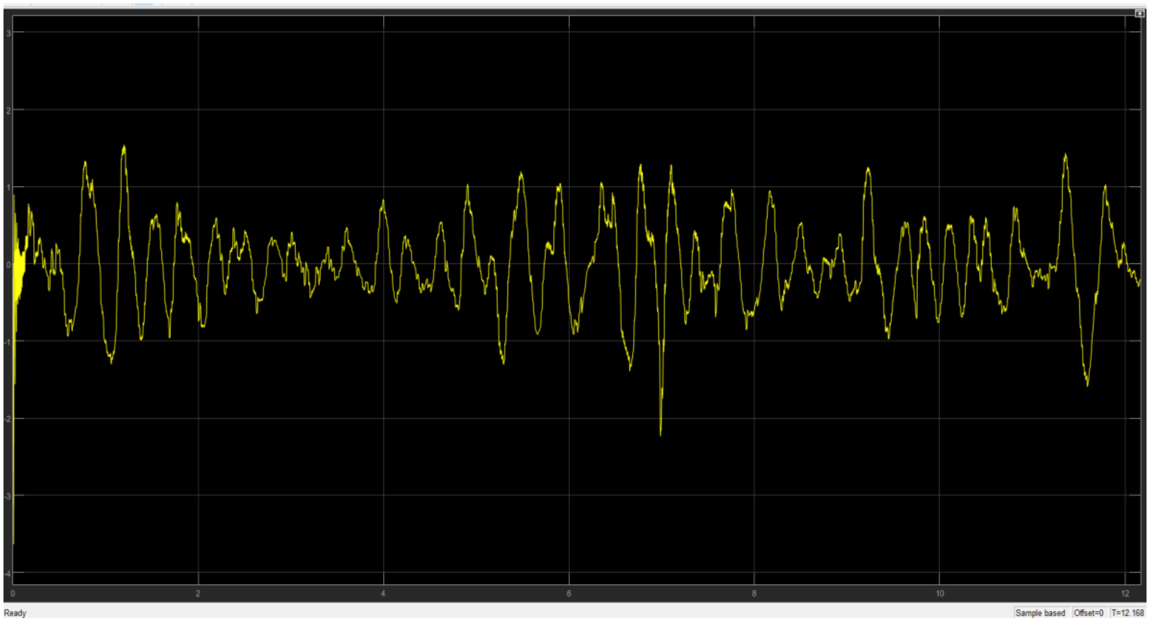


Fig 6.2 Peak vertical acceleration for carbody (Case II)

Figure 6.2 shows that the vertical acceleration of carbody in case of random vibrations also varies with frequency. Random values of amplitude for vertical acceleration are obtained. Only the peak value of vertical acceleration for carbody will be considered to calculate the ride index.

- **Frequency spectrum (Case II)**

The frequency spectrum is obtained by transferring time-domain data from the output of the Simulink model to Matlab workspace to convert it into frequency domain data. Only those frequencies are considered for ride index calculations that are powerful enough to affect the ride quality. Frequencies that are less dominant are neglected for ride index calculation.

Figure 6.3 shows the frequency spectrum for vibrations occurring at random frequencies. A large number of frequencies can be seen in the figure, but for ride index calculation, only those frequencies whose signals are powerful enough are considered for ride index calculation.

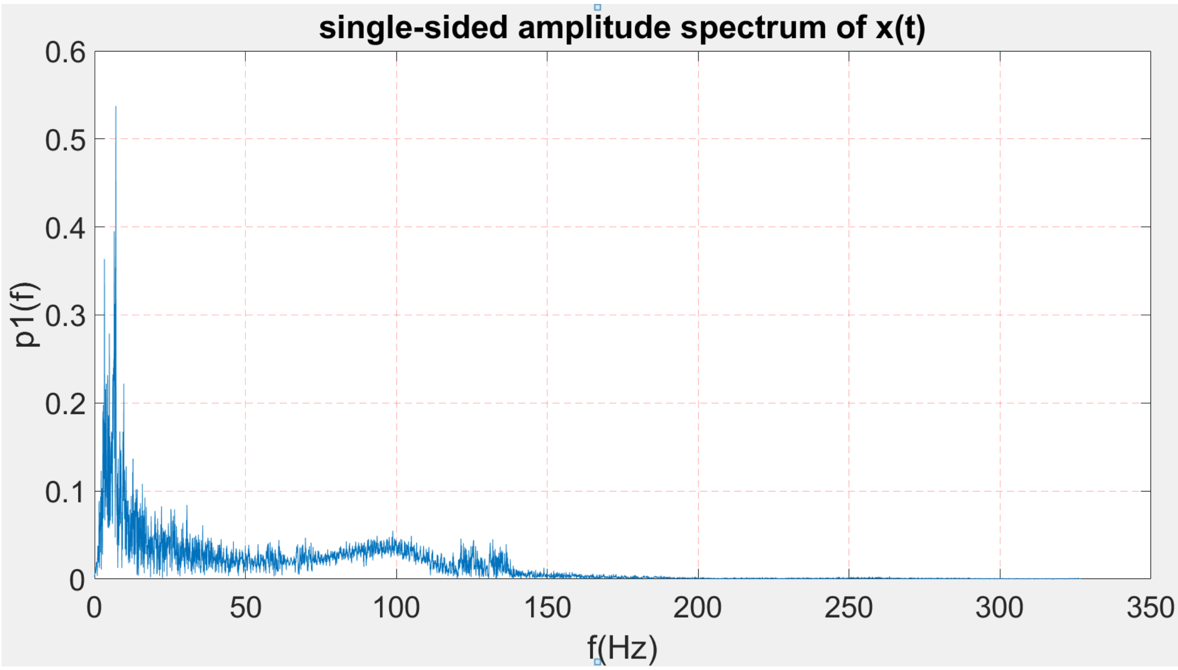


Fig 6.3 Frequency spectrum (Case II)

6.4 Ride Index Analysis for Original Parameters (Case II)

The values of peak vertical acceleration and frequencies recorded at different speeds for carbody are listed in table 6.2. Hence, the recorded values of ride index at various speeds are

Table 6.3 Ride index analysis for original data

Speed ($^{km}/_{h}$)	Peak vertical acceleration (m/sec^2)	Ride index
70	3.62	3.82
80	3.68	3.84
90	3.86	3.88
100	4.18	3.96
100	4.54	4.07

Hence, the variation of peak vertical acceleration and ride index with speed is plotted as shown in figure 6.4 and 6.5:

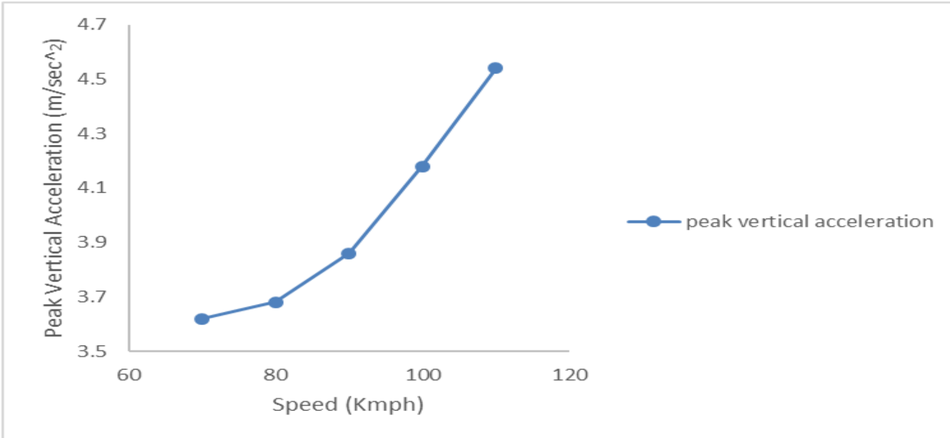


Fig 6.4 Variation of speed vs. acceleration

Figure 6.4 shows that the value of peak vertical acceleration for carbody increase with an increase in speed. This means that there will be vibrations of larger amplitude with increasing speed.

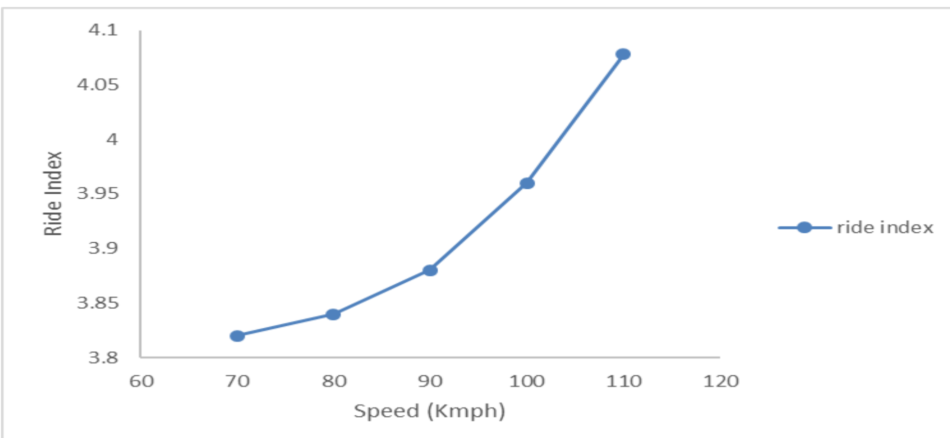


Fig 6.5 Variation of speed vs. ride index

Figure 6.5 shows that ride index for a vehicle increases as we increase speed because the values of peak vertical accelerations are increasing, the frequencies of vibrations will also increase. Ultimately, the ride index increases.

Chapter 7 Parametric Analysis

7.1 Introduction

A parametric analysis is the study of the influence of different geometric or physical parameters or both on the solution of the problem. The physical parameters here are the different parts of the suspension system.

Hence, in this chapter, we aim to vary our original parameters and see how the vertical acceleration and ride index varies with variations in suspension parameters. Suspension system basically consists of primary and secondary suspensions, hence varying their values each at a time, their effect on the final ride index can be calculated.

Based on the results of this parametric analysis, we can suggest a new set of parameters that gives better ride quality.

7.2 Primary Suspension Analysis

The primary suspension consists of elastomeric pad stiffness and damping. Hence, varying each parameter at a time while keeping others constant at different speeds, their effect is analyzed.

- **Elastomeric pad stiffness**

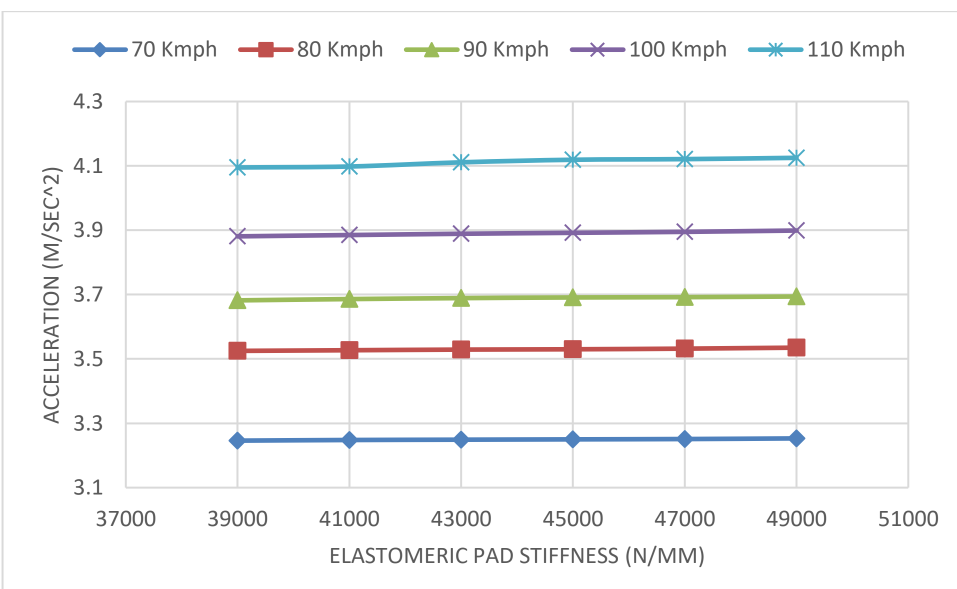


Fig 7.1 Sensitivity of elastomeric pad stiffness on acceleration

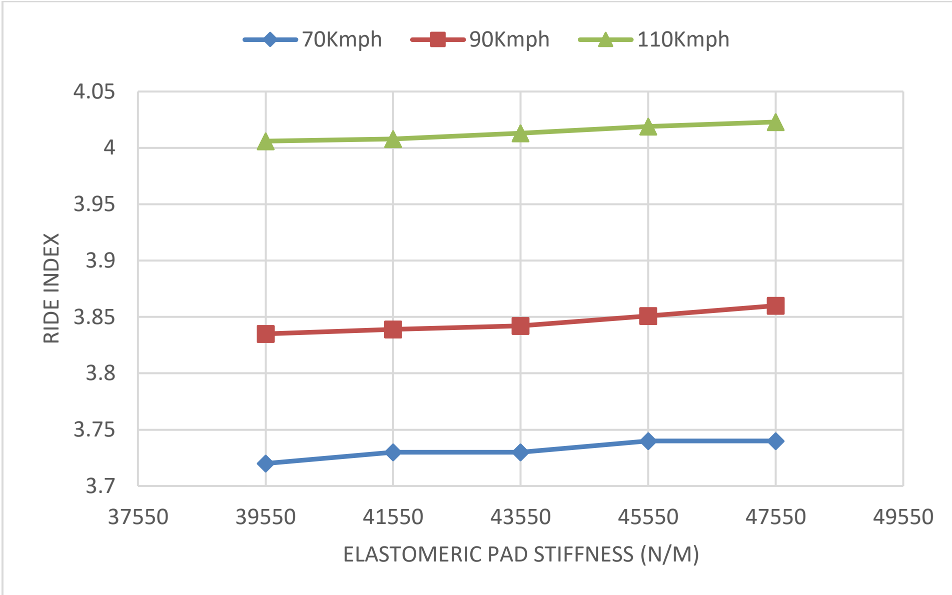


Fig 7.2 Sensitivity of elastomeric pad stiffness on ride index

Hence, a variation in elastomeric pad stiffness values does not produce significant changes in acceleration and ride index.

- **Elastomeric Pad Damping Coefficient**

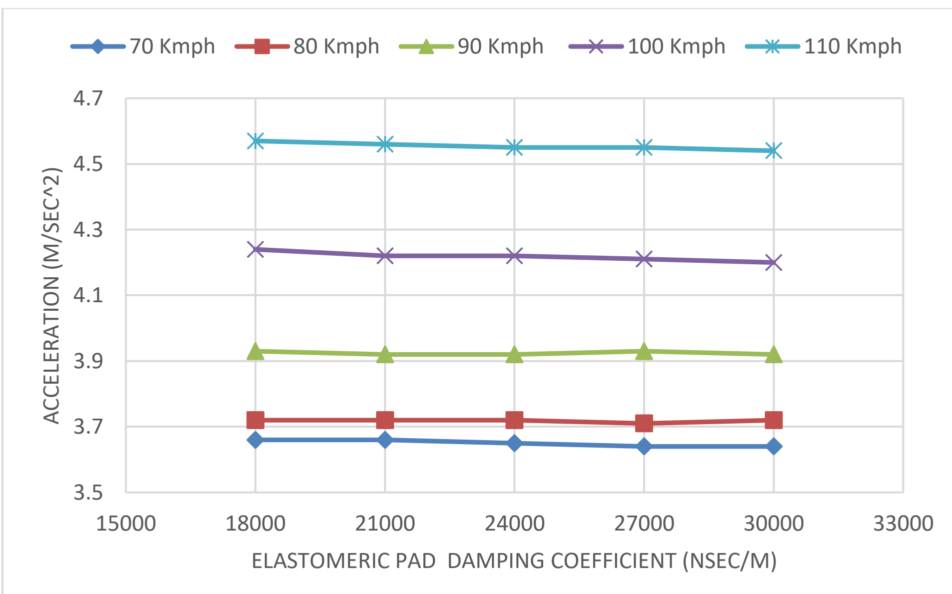


Fig 7.3 Sensitivity of elastomeric pad damping coefficient on acceleration

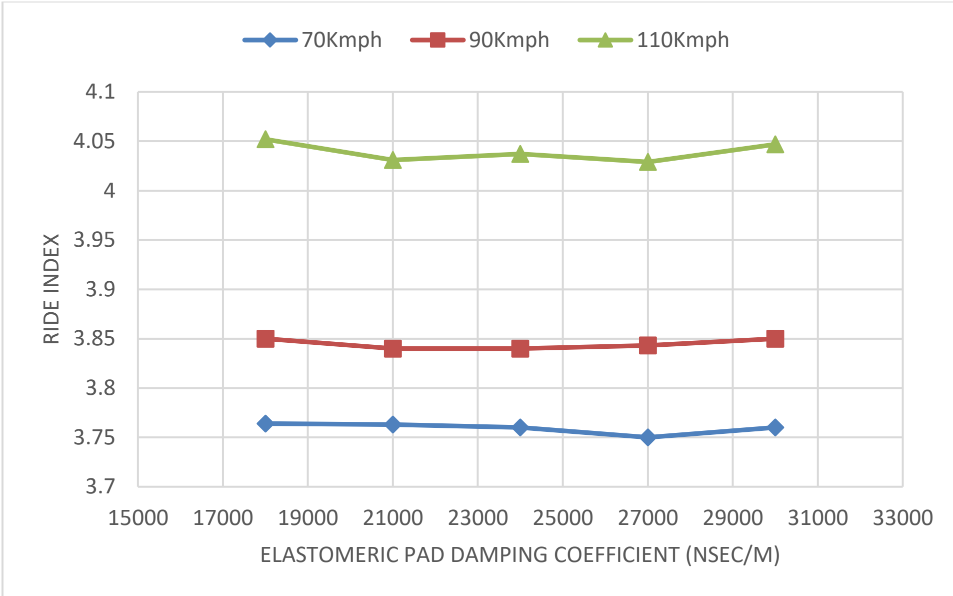


Fig 7.4 Sensitivity of elastomeric pad damping coefficient on ride index

Hence, the variations in the values of the elastomeric pad damping coefficient do not bring significant changes to the vertical acceleration and ride index.

7.3 Secondary suspension

The secondary suspension system consists of 3 main springs: outer, inner and snubber springs. Therefore, their values are sequentially varied keeping others constant and their effect on the final ride index is studied.

- **Outer spring**

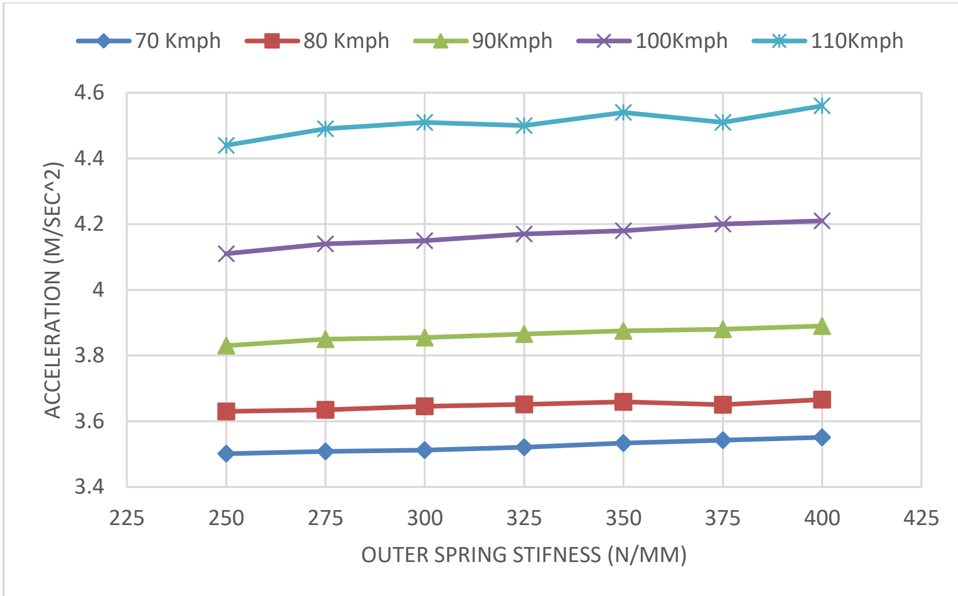


Fig 7.5 Sensitivity of outer spring stiffness on acceleration

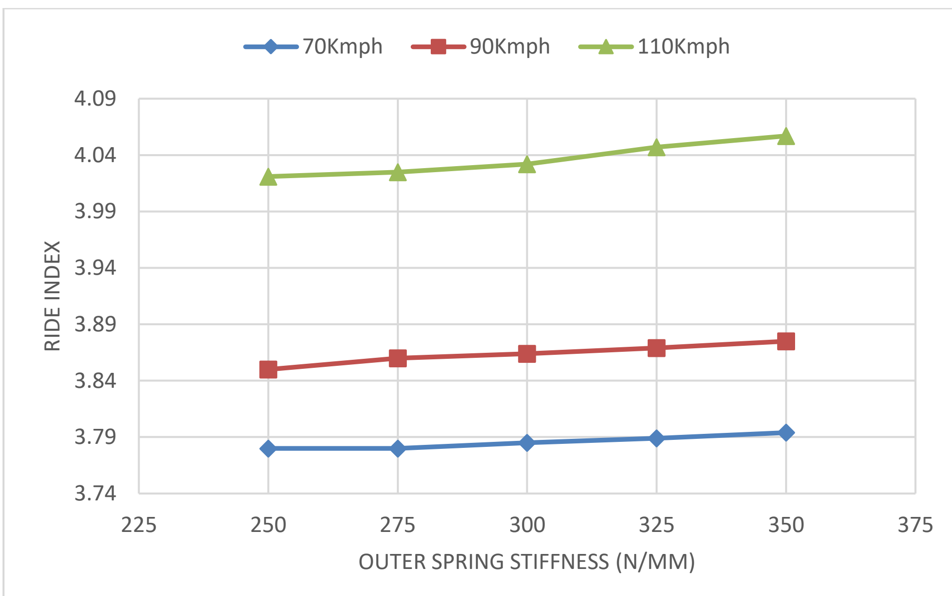


Fig 7.6 Sensitivity of outer spring stiffness on ride index

Hence, it can be seen that there are noticeable changes in acceleration and ride upon varying values of the outer spring.

- **Inner spring**

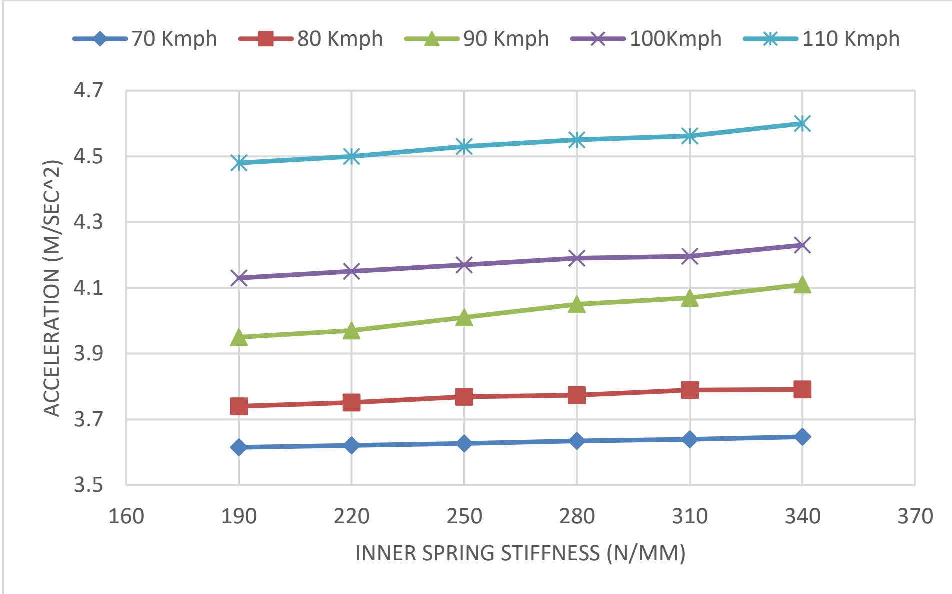


Fig 7.7 Sensitivity of inner spring stiffness on acceleration

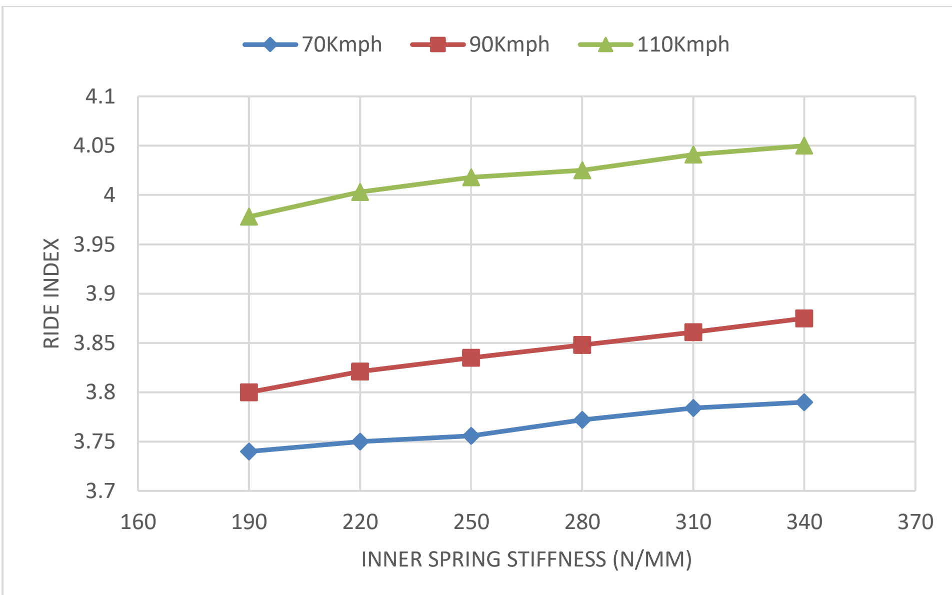


Fig 7.8 Sensitivity of inner spring stiffness on ride index

Hence, it can be clearly seen from the graphs that the variation in inner spring values affects the vertical acceleration and ride index.

- **Snubber springs**

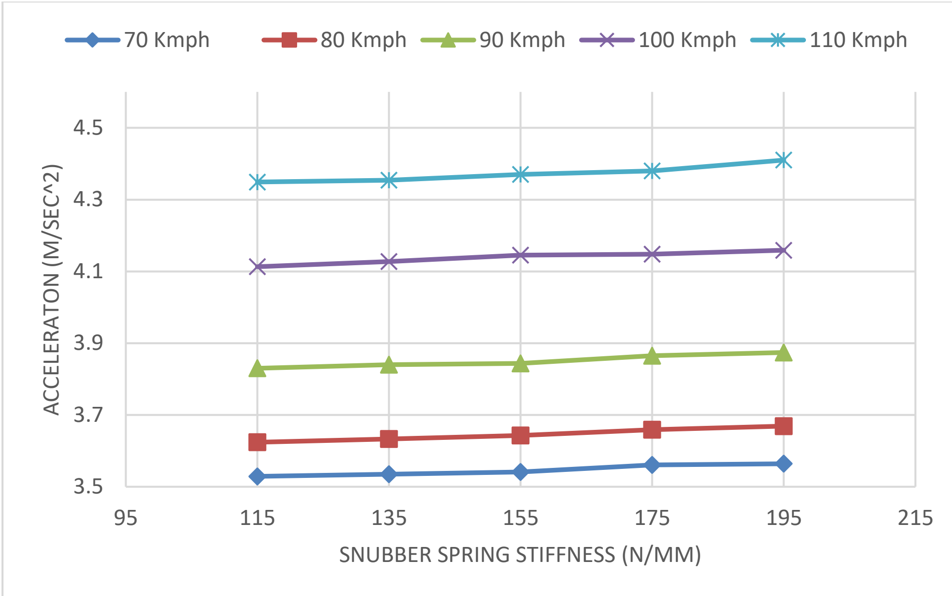


Fig 7.9 Stiffness of snubber spring stiffness on acceleration

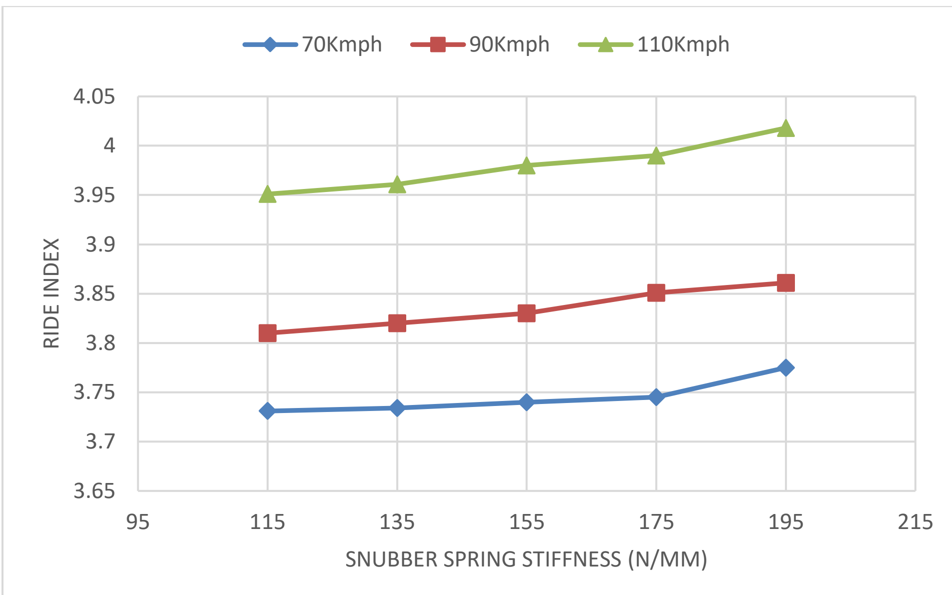


Fig 7.10 Sensitivity of snubber spring stiffness on ride index

Hence, it can be seen that variation in snubber springs results in significantly larger variations on acceleration and ride index.

Chapter 8 Results and Discussion

The results obtained so far in the project are:-

- For Case I: Harmonic vibrations
- The nature of peak vertical acceleration of all bogie components is plotted.
- The values of peak vertical accelerations for all bogie components at different fixed frequencies are recorded.
- The peak vertical acceleration starts damping after each cycle and reaches zero.

- For original parameters:-
- The peak vertical acceleration and frequency required for ride index calculation are found through the Simulink model and their graphs have been plotted
- Peak vertical acceleration and ride index has been calculated at speeds 70-110 km/h and their graphs have been plotted

- For parametric analysis:-
- The primary suspension system does not have a significant effect on the ride index.
- Secondary suspension elements have been varied sequentially and their graphs have been plotted.
- A new set of parameters is suggested based on the best values of secondary suspension elements:-
  - 190 N/mm(**I**) ,275 N/mm(**O**) ,155 N/mm(**S**)

- Comparison:-

  A new ride index is calculated for the new set of parameters at different speeds. A comparison between the ride index calculated from the original and new set of parameters is done. The same is plotted on a graph.

Table 8.1 Comparison of ride index

Speed (km/h)	Ride Index (original)	Ride Index (new)
70	3.82	3.74
90	3.88	3.83
110	4.07	3.98

Hence, the new set of parameters obtained through parametric analysis gives a lower ride index than the original parameters. Therefore, ride quality has been improved.

The graphical representation between original and new ride index is given below

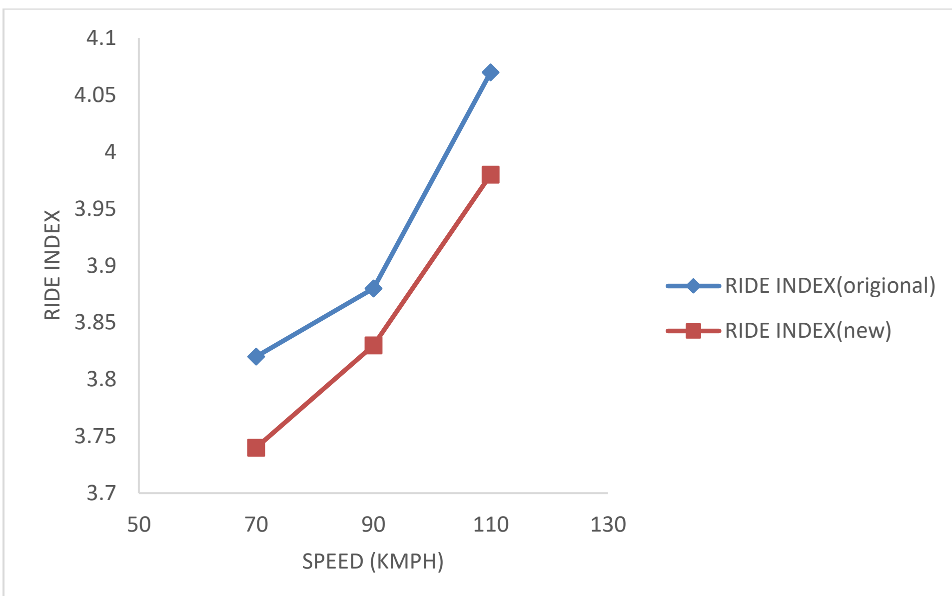


Fig 8.1 Comparison of ride index

Chapter 9 Conclusions and Future Work

❖ **Conclusions:**

The following conclusions can be drawn from the results obtained in this project-

- The amplitude of vertical acceleration obtained for carbody are well inside the limiting values of gravity 'g'. This is the basic requirement of any suspension system that the vehicle part which carries passengers or goods and services must not be bouncing, for which its vertical acceleration must be less than 'g'.
- In the case of harmonic vibrations, the value of peak vertical acceleration recorded is maximum for wheel-set because it is in direct contact with track irregularities, hence it experiences high vertical excitations. Whereas, carbody records least peak vertical acceleration at all frequencies as most of the vibrations are contained by the suspension system.
- For harmonic vibrations, the values of peak vertical acceleration for all bogie components increase with an increase in frequency, because by increasing frequency ultimately the speed increases.
- The ride index and Peak Vertical Acceleration for carbody remains practically unchanged with increment in the values of the Primary Suspension element (Elastomeric pad stiffness & damping coefficient). Hence, the primary suspension system provides a low contribution to the response of the vehicle suspension in the vertical direction, due to its high stiffness.
- The Ride Index and Peak Vertical Acceleration for carbody rise with an increase in the values of Secondary Suspension elements (Outer, Inner & Snubber springs).
- Hence, through parametric analysis of both primary and secondary suspension systems, we have derived a new set of parameters that gives better ride quality than original parameters.

❖ **Future work**:

In the present project, we have only varied spring and damping parameters to see how its variation affects vertical acceleration and ride index.

For carrying the same project further to improve the ride quality, we can study how the wire diameter of the suspension springs affects the ride index of the vehicle.

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Appendices

The values of original Suspension parameters are listed Table 1

Table 1. Original parameters

Parameters	**Values (CASNUB 22HS Bogie with BOXNHL Wagon)**
Mass of car body (M1)	40.68 T
Mass of bolster (M2)	565 $Kg/bogie$
Mass of side frame (M3)	430 $Kg/bogie$
Mass of wheel set (M4)	3 $T/bogie$
Spring stiffness of side bearings (K1=K3)	53.5 Kg/mm
Spring stiffness of center pivot (K2)	10000 Kg/mm
Spring stiffness of secondary suspension springs (K4=K5) (Kg/mm)	26.65(I), 34.68(O), 19.8(S)
Spring stiffness of primary suspension system springs (K6=K7)	49000 N/mm
Damping coefficients of side bearings(C1=C3)	Damping coefficient = 100000 $N.sec/m$
Damping coefficients of center pivot (C2)	Damping coefficient = 50000 $N.sec/m$
Damping coefficient of primary suspension system (C6=C7)	Damping coefficient = 28749.5 $N.sec/m$
Damping coefficients of secondary suspension system (C4=C5)	Damping coefficient = 200 $N.sec/mm$
Hertzian springs (K8=K9)	1000000000 N/m