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Parametric Sensitivity Analysis of Under-carriage Components of a Railway Vehicle and their Correlation to Derailment Safety and Ride Comfort

Debraj Bhattacharjee* Bidhan Malakar* Prashant Singh* Sushanta Neog* and B.K. Roy*

Abstract : This paper presents a simulation based analysis to find the correlation between derailment safety, ride comfort of passengers in Indian railway vehicle and its undercarriage components. The primary and secondary suspensions and dampers of the railway vehicle are considered in their healthy and different levels of faulty condition. ADAMS powered VI-Rail Multi Body simulation software is used for the simulation. The results confirm various reported correlations available in the literature. However, when the analyses are extended for the simultaneous variation of primary and secondary suspensions, it reveals that the derailment coefficient is more sensitive to the change in primary suspension while the ride comfort is more sensitive to the change in secondary suspension.

Keywords : Multi body simulation, VI-Rail, Derailment safety, Ride Comfort, Sensitivity analysis.

1. INTRODUCTION

The dynamic stability of a railway vehicle has always been an intriguing area of research. It is a representation of the balance among several forces: forces due to gravity and inertia, suspension system and other undercarriage components, and also the wheel-rail contact forces [1]. As is well established, a railway vehicle consists of several undercarriage components which range from passive ones such as coil springs to several active or semi-active ones. The link between the safety of the railway vehicle in running condition and the reliability of the undercarriage components is inherent. There have been consistent concerns over safety in the railways in a plethora of domains. These include several cases of derailments of the rail vehicle from the railway track as well as numerous cases of severe discomfort to travelling passengers. The origin of these issues can be attributed to the dereliction of the vehicle and track components. So, an effective metric for characterizing the stability of the railway vehicle in its running condition is the need of the hour. Several measures exist for both quantitative and qualitative measurement of the dynamic stability of the railway vehicle. Two of these measures are: derailment safety and ride comfort.

The determination of factors that influence the variation of the above measures is imperative. A parametric sensitivity can, thus, provide assistance in this regard. It is important to note that users have very little control over parameters related to the wheel-rail interface. On the other hand, parameters related to the vehicle configuration and that of the track can be varied over a certain range to find out their correlation to derailment safety and ride comfort.

Derailments can cause major losses to both life and property, and, therein derailment safety has a very crucial role in determining the stability of the railway vehicle. One of first instances of studies of the

* National Institute of Technology, Silchar-788010, Assam, India *E-Mail: bkr_nits@yahoo.co.in*

stability problem in railway vehicles was the study of DePater [2]. In this, the author addressed the problem of the determination of hunting movement in railway vehicles. Since then, rapid developments in this field have led to a variety of works related to parametric analyses. In [3], the author examines the sensitivity of the critical velocity of the vehicle to the changes in primary stiffness and damping parameters. The author attempts to characterise the change in lateral stability of railway bogies with changes in the suspension parameters in [4]. In [5], the lateral stability of a vehicle is discussed. An investigation is carried out for the determination of the nonlinear critical speed of a railway vehicle using bifurcation analysis. The authors assert that the suspension parameters have very little influence on critical speeds. The author goes on to state that critical speeds, rather, have more sensitivity to wheel rail contact characteristics.

Although, stability of the railway vehicle is of primary concern during its running condition, the quality of ride is also of significant importance. One of the essential measures to estimate passenger comfort and damage to cargo in railway vehicles during the running condition is the ride comfort. Among a host of other factors, ride comfort is primarily related to vibrations in the railway vehicle. In [6], the author has attempted to optimise the curving performance of a 21-DOF rail vehicle model using genetic algorithms and multi body dynamics. In [7], Nejlaoui et al. deal with ride comfort and rail vehicle safety on curved tracks with of radius. A quasi static study of a quarter car is derived from the study of the system behavior. The author provides a detailed analysis of the effect of various parameters on running safety and ride comfort in [8]. Several international standards are taken into consideration while determining the same.

While, in all of the above literature an extensive list of parameters has been considered to determine their correlation to derailment safety and ride comfort, it is evident that in most of the works, only one vehicle parameter (including the velocity of the rail vehicle) has been varied at a time to find out the variation in the above two measures.

However, in the present work, two parameters have been altered at a time and then the variations in the above measures have been found out. The reason for doing the same is that, it would help in establishing the relative rank of the undercarriage components when finding out the sensitivity of derailment safety and ride comfort to the changes in these parametric combinations.

An Integral Coach Factory (ICF) based Indian railway vehicle has been considered for this work. This vehicle has been modelled using ADAMS powered VI-Rail Multi Body Simulation (MBS) software. The nominal values for the vehicle parameters have been obtained through that provided by Research, Design Standards Organization (RDSO). The objectives of this paper are:

- To determine the sensitivity of the derailment coefficient to the changes in the undercarriage components in the railway vehicle.
- To determine the sensitivity of the ride comfort to the changes in the undercarriage components in the railway vehicle.
- To correlate the changes in both of these metrics and determine the relative order of the undercarriage components in terms of their effects on derailment safety and ride comfort.

The organisation of the paper is given here. After introduction in Section 1, Section 2 deals with the modelling of the railway vehicle while Section 3 provides a brief introduction to metrics related to derailment safety and ride comfort. Section 4, then, deals with the parametric sensitivity analysis. The results and discussion of the simulation are presented in Section 5. Finally, conclusions are provided in Section 6.

2. MATHEMATICAL MODELLING OF RAILWAY VEHICLE

The railway vehicle, considered in this work, mainly comprises of three parts: Car-body, Bogie, Wheelset. The equations of motion of these components are based on [10] and are given below:

2.1. Car-Body Dynamics

The Car-body dynamics is described here by using the lateral (y_{cb}), yaw (ψ) and roll (θ) motion in terms of the following equations:

$$m_{cb}\ddot{y}_{cb} = -(F_{sec\_ymr} + F_{sec\_yml} + F_{sec\_ynr} + F_{sec\_ynl}) - m_{cb}\, g\left(\frac{v^2}{gR} - \frac{h}{2a}\right) \quad (1)$$

$$I_{cbz}\ddot{\psi}_{cb} = (F_{sec\_xmr} + F_{sec\_xnr})d_{sec} - (F_{sec\_xml} + F_{sec\_xnl})d_{sec} - (F_{sec\_ymr} + F_{sec\_yml})l_b + (F_{sec\_ynr} + F_{sec\_ynl})l_b \quad (2)$$

$$I_{cbx}\ddot{\theta}_{cb} = (F_{sec\_ymr} + F_{sec\_yml} + F_{sec\_ynr} + F_{sec\_ynl})h_{cbsec} - (F_{sec\_zmr} + F_{sec\_znr})d_{sec} - (F_{sec\_zml} + F_{sec\_znl})d_{sec} \quad (3)$$

2.2. Bogie Dynamics

The railway vehicle model consists of two bogies: a front bogie and a rear bogie. The bogie dynamics can be delineated by the lateral (y_{bg1}, y_{bg2}), yaw (ψ_{bg1}, ψ_{bg2}) and roll (θ_{bgr}, θ_{bgl}) motions. The subscripts bg 1and bg 2, here, refer to the front and rear bogies, respectively.

$$m_{bg}\ddot{y}_{bg1} = (F_{sec\_ymr} + F_{sec\_yml}) - (F_{pri\_y1r} + F_{pri\_y2r} + F_{pri\_y1l} + F_{pri\_y2l}) \quad (4)$$

$$\begin{aligned} I_{bgz}\ddot{\psi}_{bg1} = {} & (-F_{sec\_xmr} + F_{sec\_xml})d_{sec} + (F_{pri\_x1r} + F_{pri\_x2r})d_{pri} - (F_{pri\_x1l} + F_{pri\_x2l})d_{pri} - (F_{pri\_y1r} + F_{pri\_y1l})w_b \\ & + (F_{pri\_y2r} + F_{pri\_y2l})w_b \end{aligned} \quad (5)$$

$$\begin{aligned} I_{bgx}\ddot{\theta}_{bg1} = {} & (F_{sec\_ymr} + F_{sec\_yml})h_{bgsec} + (F_{sec\_zmr} - F_{sec\_zml})d_{sec} + (F_{pri\_y1r} + F_{pri\_y2r} + F_{pri\_y1l} + F_{pri\_y2l})h_{bgpri} \\ & - (F_{pri\_zfr} + F_{pri\_z2r})d_{pri} + (F_{pri\_z1l} + F_{pri\_z2l})d_{pri} \end{aligned} \quad (6)$$

$$m_{bg}\ddot{y}_{bg2} = (F_{sec\_ynr} + F_{sec\_ynl}) - (F_{pri\_y3r} + F_{pri\_y3l} + F_{pri\_y4r} + F_{pri\_y4l}) \quad (7)$$

$$\begin{aligned} I_{bgz}\ddot{\psi}_{bg2} = {} & (-F_{sec\_xnr} + F_{sec\_xnl})d_{sec} + (F_{pri\_x3r} + F_{pri\_x4r})d_{pri} - (F_{pri\_x3l} + F_{pri\_x4l})d_{pri} - (F_{pri\_y3r} + F_{pri\_y3l})w_b \\ & + (F_{pri\_y4r} + F_{pri\_y4l})w_b \end{aligned} \quad (8)$$

$$\begin{aligned} I_{bgx}\ddot{\theta}_{bg2} = {} & (F_{sec\_ynr} + F_{sec\_ynl})h_{bgsec} + (F_{sec\_znr} - F_{sec\_znl})d_{sec} + (F_{pri\_y3r} + F_{pri\_y4r} + F_{pri\_y3l} + F_{pri\_y4l})h_{bgpri} \\ & - (F_{pri\_z3r} + F_{pri\_z4r})d_{pri} + (F_{pri\_z3l} + F_{pri\_z4l})d_{pri} \end{aligned} \quad (9)$$

2.3. Wheelset Dynamics

The model comprises of four wheelsets: two of those are assembled in front bogie and remaining are assembled in the rear bogie. It is assumed that wheel and rail always remain in contact with each other. This eliminates the vertical motion of the wheels. The dynamics of wheelset is, therefore, described by the lateral (y_{wst}) and yaw (ψ_{wst}) motions only. These motions, in terms of the forces in the suspension system, are presented by following equations:

$$m_{wst}\ddot{y}_{wst1} = (F_{pri\_y1r} + F_{pri\_y1l}) + (F_{y1r} + F_{y1l}) - F_{g1} \quad (10)$$

$$m_{wst}\ddot{y}_{wst2} = (F_{pri\_y2r} + F_{pri\_r2l}) + (F_{y2r} + F_{y2l}) - F_{g2} \quad (11)$$

$$m_{wst}\ddot{y}_{wst3} = (F_{pri\_y3r} + F_{pri\_y3l}) + (F_{y3r} + F_{y3l}) - F_{g3} \quad (12)$$

$$m_{wst}\ddot{y}_{wst4} = (F_{pri\_y4r} + F_{pri\_y4l}) + (F_{y4r} + F_{y4l}) - F_{g4} \quad (13)$$

$$I_{wstz}\ddot{\psi}_{wst1} = (-F_{pri\_x1r} + F_{pri\_x1l})d_{pri} + K_{z1} + K_{g1} \quad (14)$$

$$I_{wstz}\ddot{\psi}_{wst2} = (-F_{pri\_x2r} + F_{pri\_x2l})d_{pri} + K_{z2} + K_{g2} \tag{15}$$

$$I_{wstz}\ddot{\psi}_{wst3} = (-F_{pri\_x3r} + F_{pri\_x3l})d_{pri} + K_{z3} + K_{g3} \tag{16}$$

$$I_{wstz}\ddot{\psi}_{wst4} = (-F_{pri\_x4r} + F_{pri\_x4l})d_{pri} + K_{z4} + K_{g4} \tag{17}$$

In the Eqns. (1) – (17), F represents the forces on the different components of the vehicle assembly and v is the speed of the railway vehicle. The subscript pri and sec represent the primary and secondary suspension system, respectively. The subscripts x, y, z represent the respective direction of motion. The subscripts m and n represent the leading and trailing bogies, respectively. The subscripts 1,2,3,4 represent the respective wheelset under consideration and the subscripts l and r represent the left and right side of the suspension system being taken under consideration, respectively. Additional terms are described in Appendix A. A schematic diagram of the front view of the railway vehicle assembly is shown in Fig. 1.

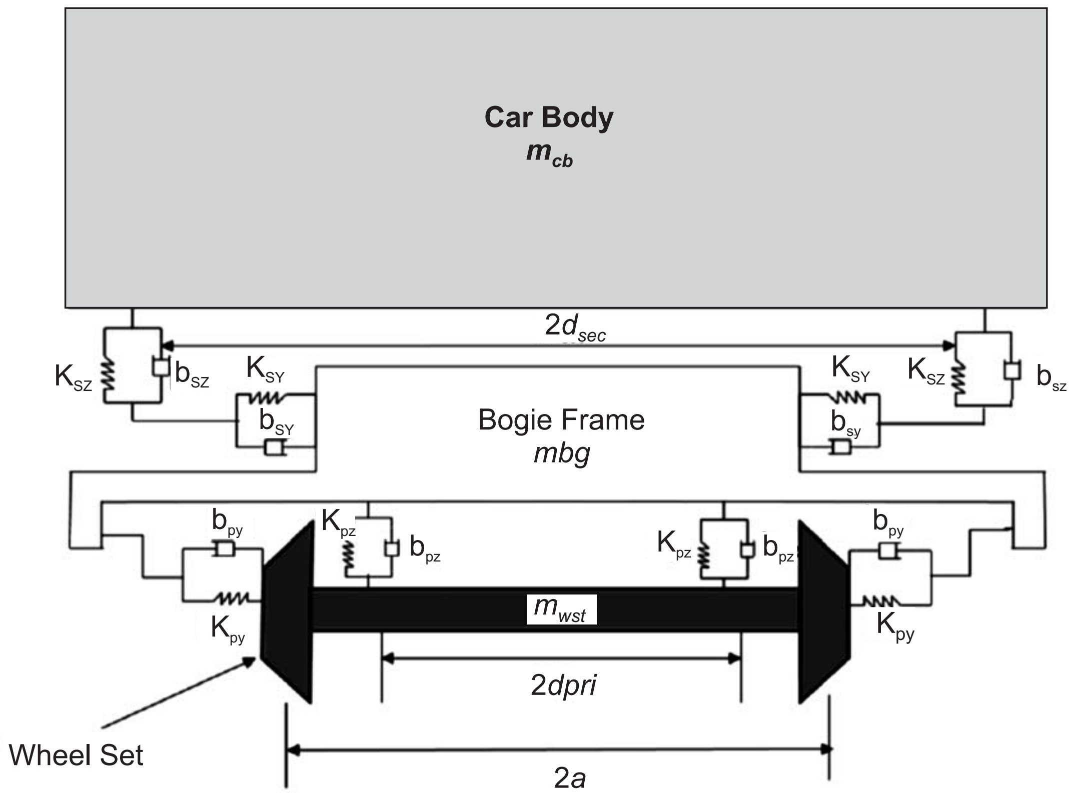


Figure 1: Schematic diagram of front view of the railway vehicle assembly

3. METRICS FOR DERAILMENT SAFETY AND RIDE COMFORT

3.1. Derailment Safety and Nadal's Formula

The occurrences of derailments can have fatal consequences on human life besides tremendous losses to property. The necessity to find out the causes of derailments is, therefore, of paramount importance. A representation of the wheel rail contact in a railway vehicle is given in Fig. 2 (*a*). The ratio of the lateral force on the wheel flange to the vertical load on the wheel *i.e*. L/V ratio plays a major role in the determination of the derailment safety of a railway vehicle. To avoid flange climb, this ratio must perennially be within a critical limit [9][10].

The critical limit of the L/V ratio is calculated using the Nadal's formula:

$$\frac{L}{V} = \frac{\tan\beta - \mu}{1 + \mu\tan\beta} \tag{18}$$

where μ is the coefficient of friction

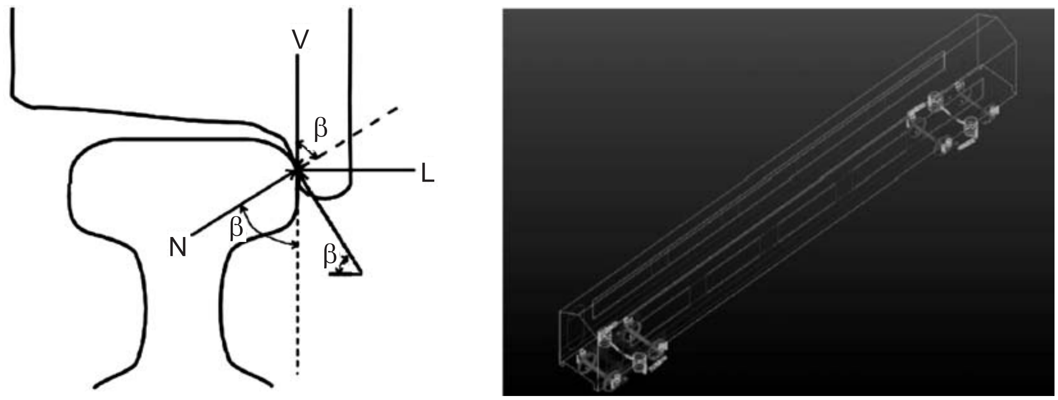
β is the flange angle

L is the lateral flange force

V is the wheel load

The 99.85th percentile of the curve, *i.e*. L/V values measured over a specified time interval, is known as the derailment coefficient. The derailment coefficient must be less than the critical limit as established by Nadal's formula.

The calculations for L *i.e*. the lateral force on the flange and V *i.e*. the vertical wheel load are based on Eqns (1)-(17).



(*a*) (*b*)

Figure 2: (*a*) Free Body diagram of the wheel-rail contact, (*b*) Model of single-wagon railway vehicle created in VI-Rail

3.2. Ride Comfort and Sperling Ride Index

The principal sources of vibrations in a railway vehicle are acceleration signals. Ride comfort, therefore, is a function of the amplitude, frequency and direction of the acceleration signals.

Sperling Ride index is an evaluation criterion related to ride comfort. It is a function of amplitude and frequency of vehicle acceleration [11]. The expression for ride comfort is given by:

$$W = \left(\int A^3 B^3 df\right)^{1/10} \tag{19}$$

$$B = 1.14\left[\frac{[(1-0.056f^2)^2 + (0.0645)^2\ (3.35f^2)]}{[(1-0.056f^2)^2 + (1.547f - 0.00444f^3)^2](1+3.35f^2)}\right]^{1/2} \tag{20}$$

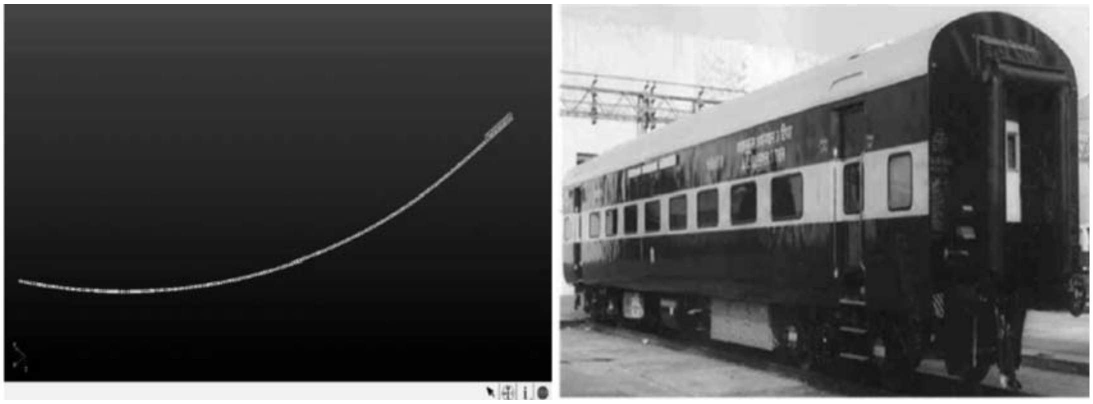
where A is the amplitude of acceleration, B is the acceleration weighting factor and f is the frequency of vibration. It is to be noted that the ride quality is inversely proportional to the ride index [10].

4. PARAMETRIC SENSITIVITY ANALYSES FOR THE EVALUATION OF DERAILMENT SAFETY AND RIDE COMFORT

In this section a discussion is presented on the process of determining the sensitivity of the derailment coefficient and ride comfort to the changes in the parameters of the undercarriage components of a railway vehicle based on simulation in VI-Rail Multibody simulation software.

In order to do the same, ADAMS powered VI-Rail is used to create models the railway vehicles having different parametric combinations of undercarriage components [11]. A parametric combination consisting of two parameters is varied at one time and the corresponding changes on derailment safety and ride comfort are observed. A model of single wagon railway vehicle created in VI-Rail is shown in Fig. 3 (*b*). The model of the curved track created in VI-Rail is shown in Fig. 3 (*a*).

It is to be noted that all the vehicle parameters are varied in a single-wagon model which is run on a curved track of 400 m. The nominal value of the undercarriage components are obtained from those provided in the literature for Integral Coach Factory (ICF) based Indian railway vehicles [12]. A sample Integral Coach Factory (ICF) based passenger railway coach is depicted in Fig. 3 (*b*).



(*a*) (*b*)

Figure 3: (*a*) Curved track created using VI-Rail, (*b*) ICF based railway coach [12]

4.1. Parametric Analysis for Evaluation of Derailment Safety

The following parametric combinations are considered and varied to determine the sensitivity of the derailment coefficient.

- Velocity and stiffness of primary suspension
- Velocity and stiffness of secondary suspension
- Velocity and nonlinear stiffness of primary damper
- Velocity and nonlinear stiffness of secondary damper
- Stiffness of primary suspension and stiffness of secondary suspension

In all of the above cases, the derailment coefficient is considered as the dependent parameter and its values are plotted against each of the parametric combinations.

4.2. Parametric Analysis for Evaluation of Ride Comfort

The following parametric combinations are considered and varied to determine the sensitivity of the ride comfort.

- Velocity and stiffness of primary suspension
- Velocity and stiffness of secondary suspension
- Velocity and nonlinear stiffness of primary damper
- Velocity and nonlinear stiffness of secondary damper
- Stiffness of primary suspension and stiffness of secondary suspension

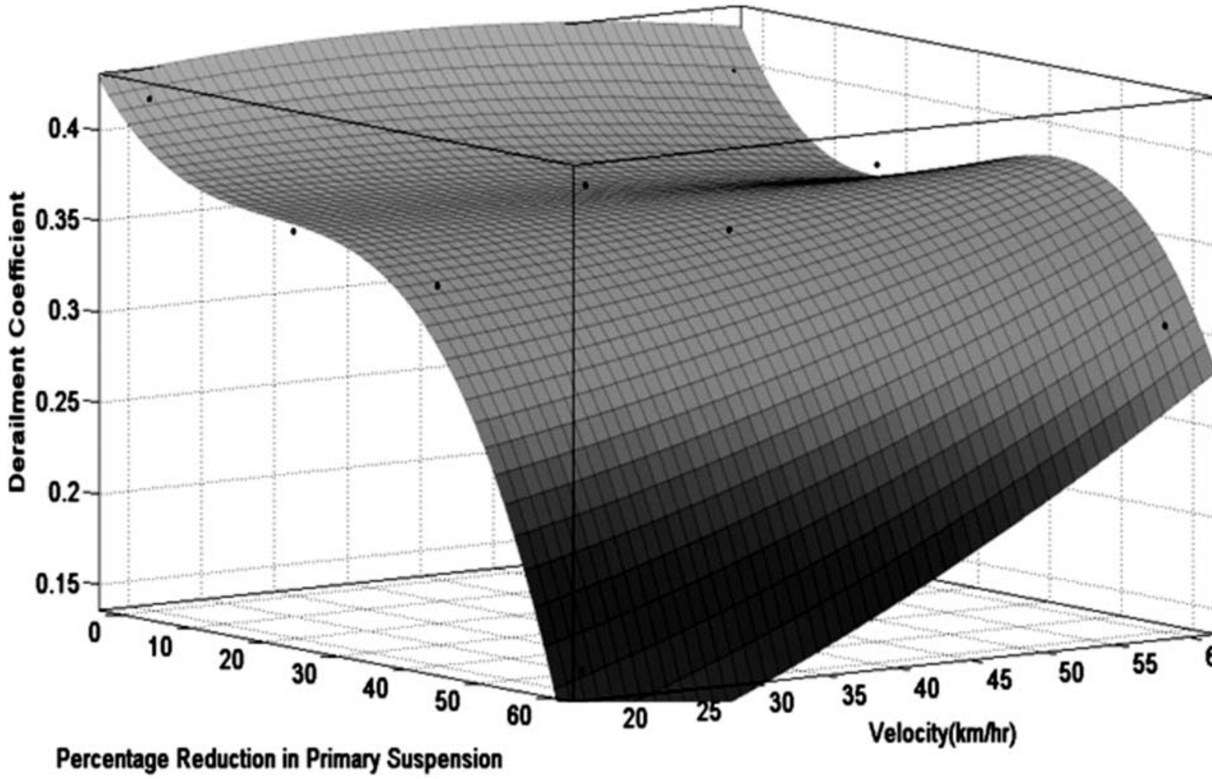
In all of the above cases, ride comfort is considered as the dependent parameter and its values are plotted against each of the parametric combinations.

5. RESULTS AND DISCUSSION

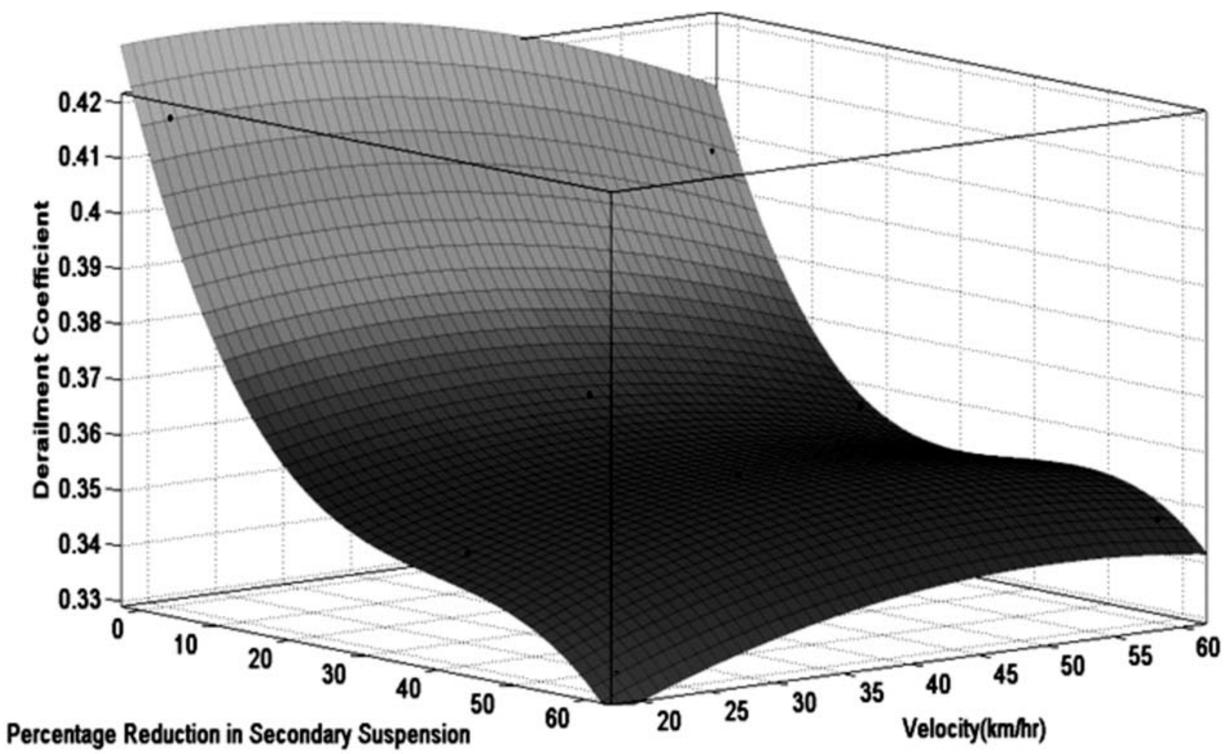
The simulation results are generated by varying the velocity from 10 km/hr to 55 km/hr along with the variations in stiffness of primary and secondary suspension and damper. These variations are considered in percentage variation of their nominal values. Percentage reduction in step up to 60% is assumed.

5.1. Variation of Derailment Coefficient with Changes in Parametric Combination of Undercarriage Components

The variation of derailment coefficient is plotted for changes in parametric combination of two undercarriage components at a time. The variation of derailment coefficient with the variation of velocity and percentage reduction in primary suspension is shown in Fig. 4 (*a*). Similar results for percentage reduction in secondary suspension, primary damper and secondary damper are shown in Figs. 4 (*b*), 5 (*a*) and 5(*b*), respectively. The variation in derailment coefficient with variation in both the primary and secondary suspension is shown in Fig. 6.



(a)



(b)

Figure 4: (*a*) Variation of derailment coefficient with variation in velocity and stiffness of primary suspension, (*b*) Variation of derailment coefficient with variation in velocity and stiffness of secondary suspension.

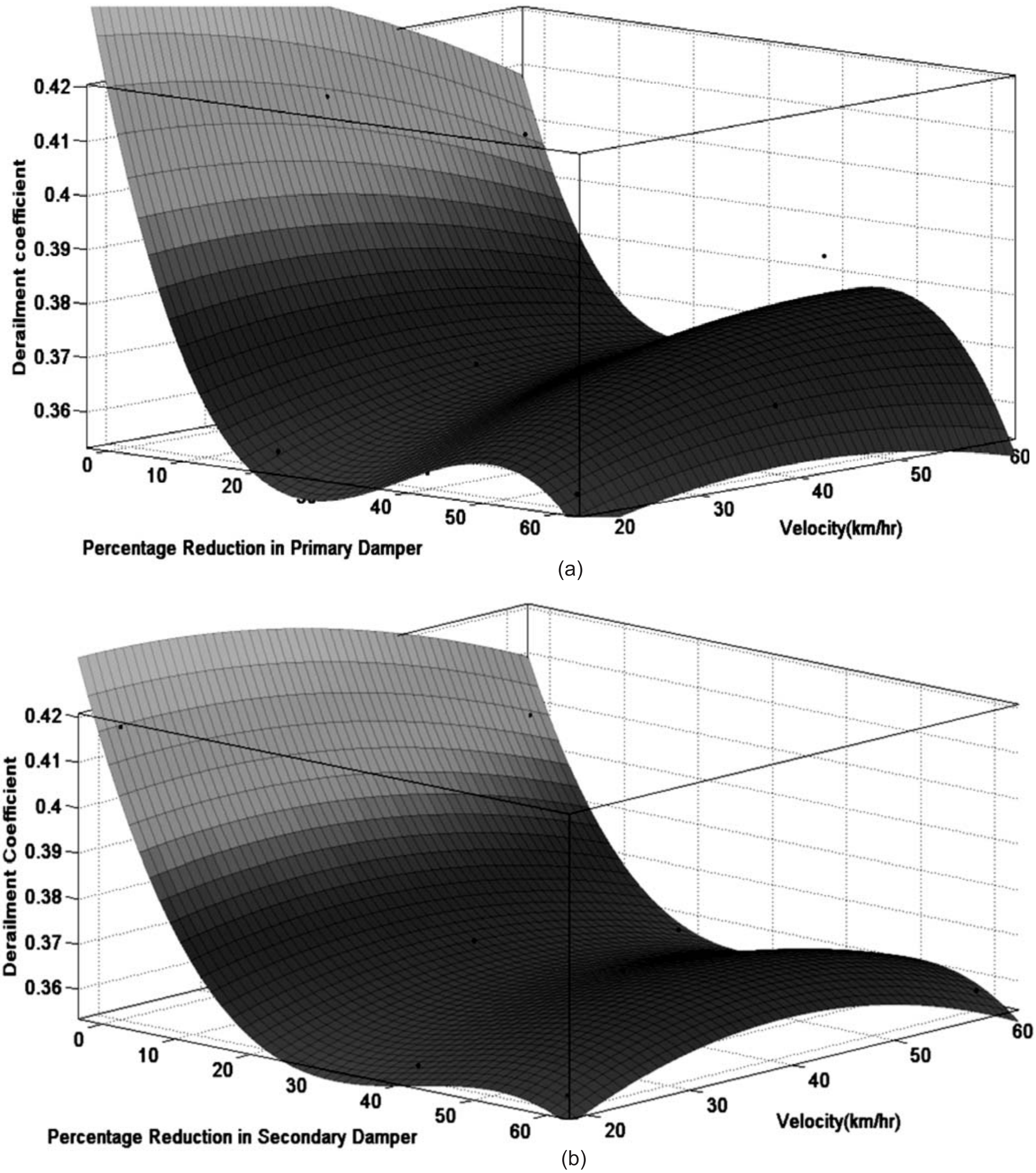


Figure 5: (*a*) Variation of derailment coefficient with variation in velocity and nonlinear stiffness of primary damper, (*b*) Variation of derailment coefficient with variation in velocity and nonlinear stiffness of secondary damper.

5.2. Variation of Ride Comfort with Changes in Parametric Combination of Undercarriage components

A similar process is undertaken as was done in the case of derailment coefficient. The variation of the ride comfort is measured as per Sparling ride index. The variation in ride comfort with the variation of velocity and percentage reduction in primary suspension, secondary suspension, primary damper and secondary damper are shown in Fig. 9 to Fig. 10, respectively. The impact of variation of both the primary and secondary suspension in ride comfort is shown in Fig. 11.

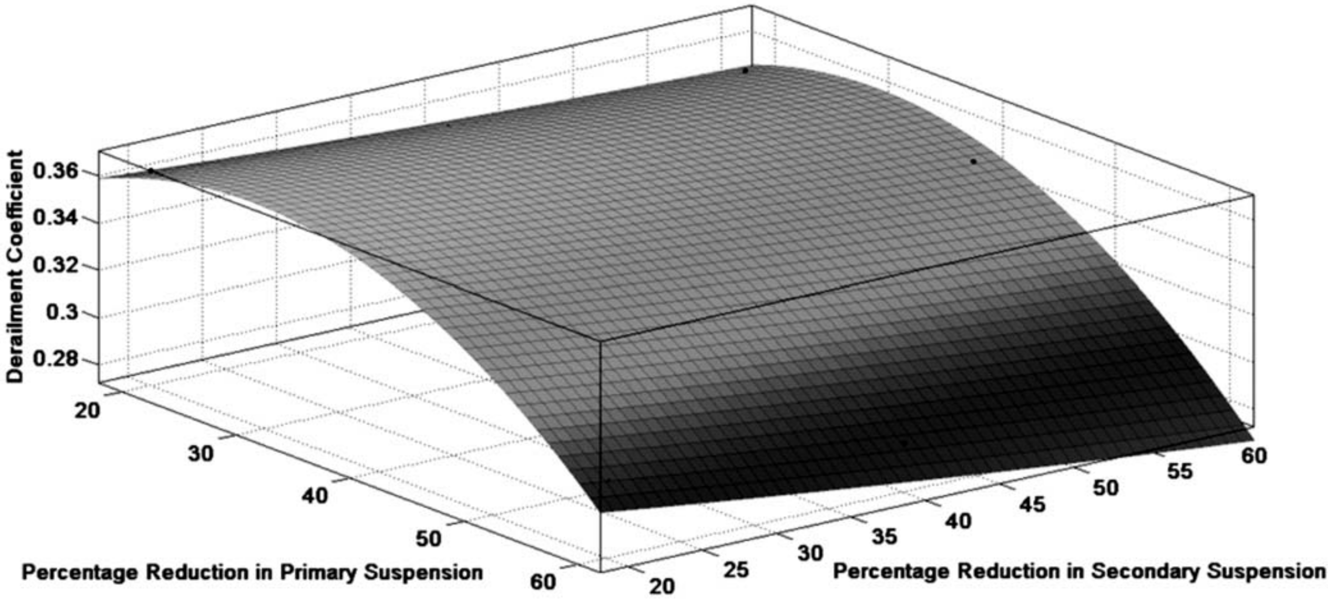
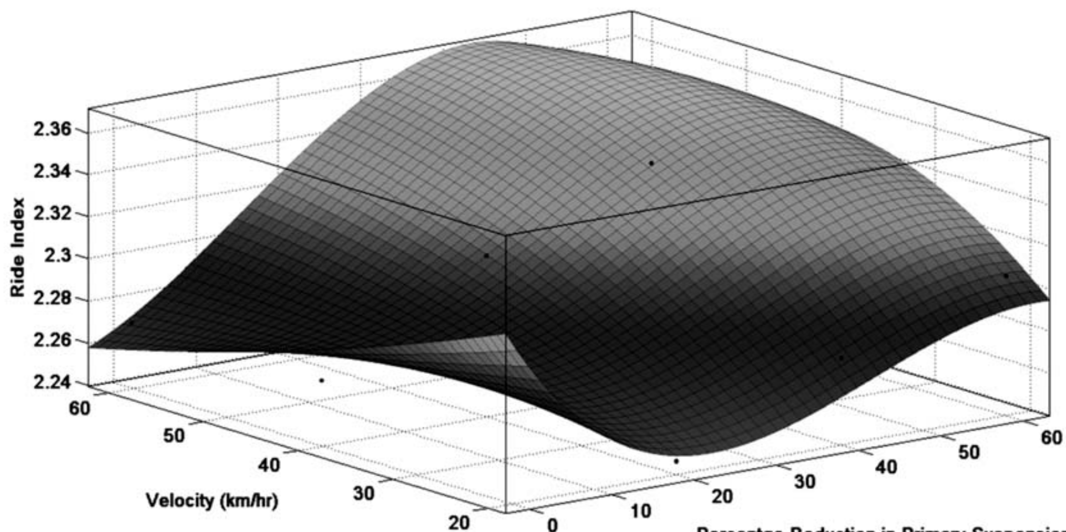
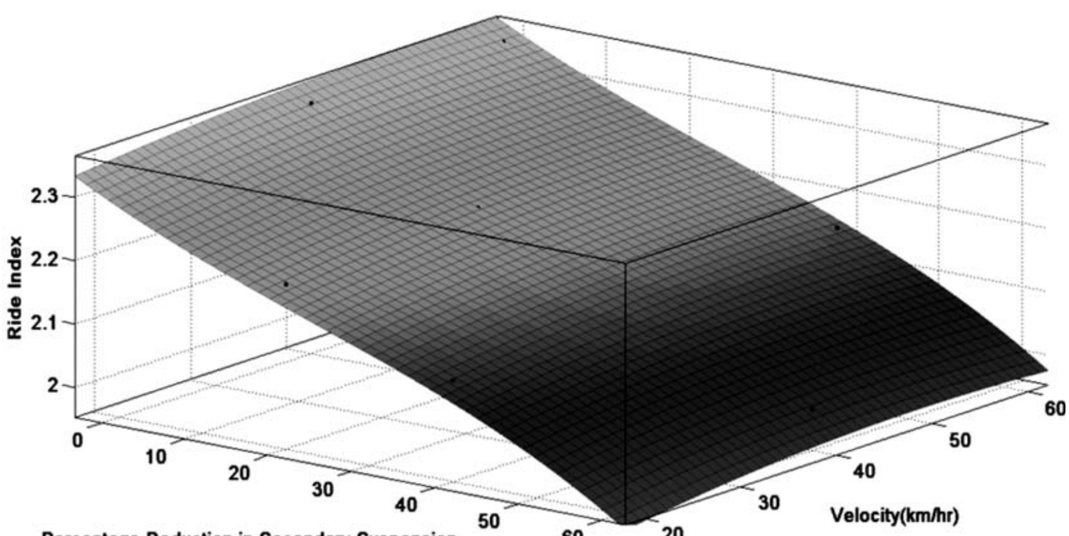


Figure 6: Variation of derailment coefficient with variation in both primary suspension and secondary suspension.



(a)



(b)

Figure 7: (*a*) Variation of ride comfort index with variation in velocity and stiffness of primary suspension, (*b*) Variation of ride comfort index with variation in velocity and stiffness of secondary suspension.

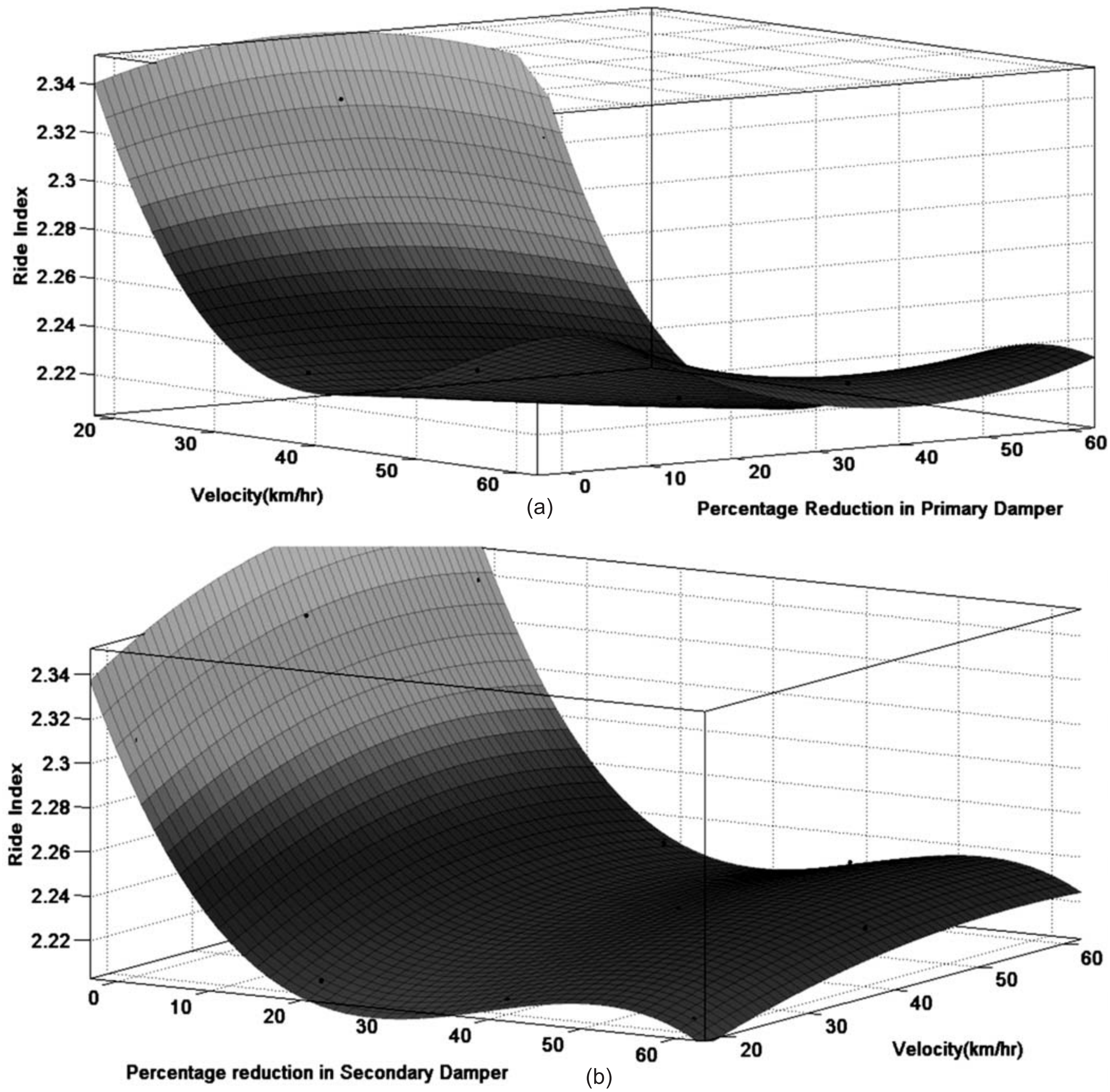


Figure 8: (*a*) Variation of ride comfort index with variation in velocity and nonlinear stiffness of primary damper, (*b*) Variation of ride comfort index with variation in velocity and nonlinear stiffness of secondary damper.

It is observed that a majority of the results obtained from the above analyses show coincidence with several reported literature. Fig. 6 reveals that the derailment coefficient has a small sensitivity to the changes in primary suspension [5][13]. It is found to be higher at low speeds and the reverse trend is observed at higher speeds [14]. The ride comfort, as expected, decays slowly with increasing velocity and loss of stiffness in primary suspension. For the secondary suspension, it is found that derailment coefficient is barely sensitive to changes in secondary suspension. The ride index, however, is very sensitive to changes in the secondary suspension. The ride comfort is found to be more for a softer spring [13]. This is because, a softer spring allows very less vibrations to pass from bogie and bolster to the car-body.

Primary and secondary dampers have similar behaviours. The derailment coefficient and the ride comfort have very little sensitivities to changes in either of the primary and secondary damping. In the case where the primary and secondary suspensions are varied simultaneously, it is found that, the derailment coefficient is more sensitive to the change in primary suspension while the ride comfort is more sensitive to the change in secondary suspension [15].

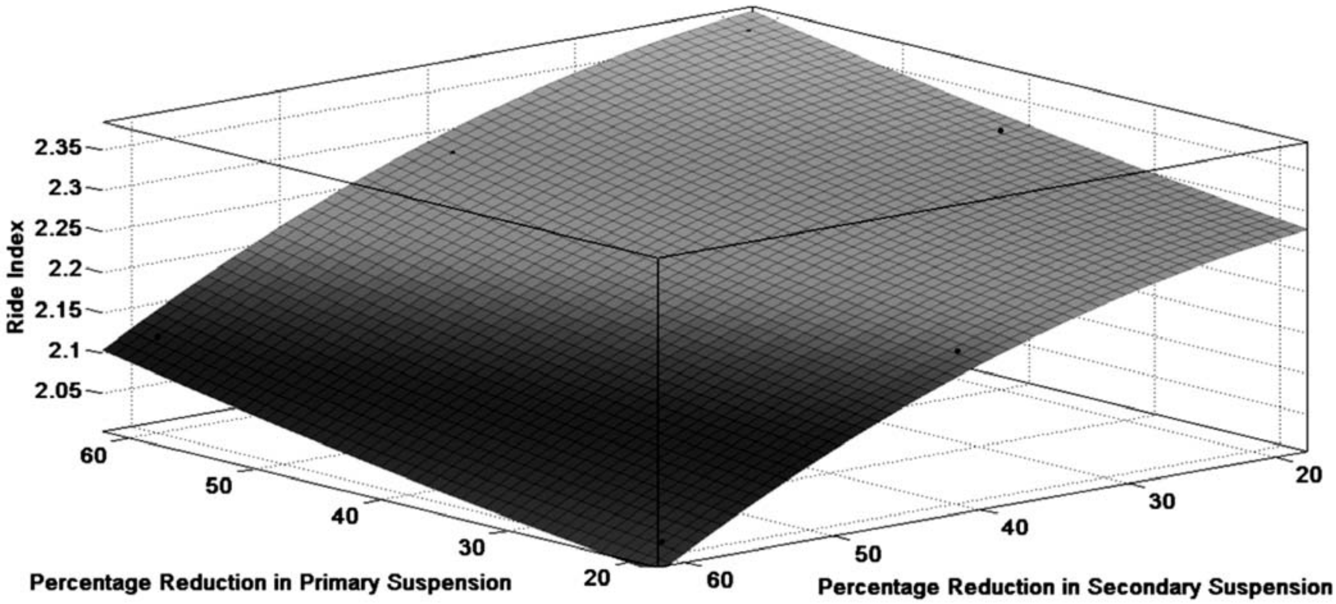


Figure 9: Variation of ride comfort index with variation in both primary suspension and secondary suspension

6. CONCLUSIONS

This paper finds the correlation between the derailment safety, ride comfort and undercarriage components of the Indian railway passenger vehicle. It is concluded that the ride comfort degrades slowly with the increase in velocity and magnitude of fault in the primary suspension. The ride comfort is very sensitive and degrades rapidly with the increase in the magnitude of faults in the secondary suspension. The status of primary and secondary dampers have very little influence on the derailment safety and ride comfort. When a fault appears in both the primary and secondary suspensions, it is concluded that the derailment coefficient is more sensitive to fault in primary suspension while the ride comfort is more sensitive to fault in secondary suspension. The relative sensitivities of other combinations of simultaneous faults will be taken up in the near future.

7. APPENDIX A

h_{bgsec}	Vertical distance from bogie frame centre of gravity to secondary suspension
h_{cbsec}	Vertical distance from car body centre of gravity to secondary suspension
h_{bgpri}	Vertical distance from bogie frame centre of gravity to primary suspension
l_b	Half of bogie centre pin spacing
w_b	Half of wheel-base
a	Half of wheelset contact distance
d_{pri}	Half of primary suspension spacing (lateral)
d_{sec}	Half of secondary suspension spacing (lateral)
m_{cb}, m_{bg}, m_{wst}	Mass of car-body, bogie, wheelset, respectively
I_{cb}, I_{bg}, I_{wst}	Moment of Inertia of car-body, bogie, wheelset, respectively
K_g	Creep moments induced by gravity forces
v	Velocity of vehicle
g	Acceleration due to gravity

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