Modeling and Simulation to Study the Dynamic Response of a Railway Carriage Components Moving on Tangent Tracks at Critical Hunting Velocity

Karim H. Ali Abood*, R. A. Khan#

Department of Mechanical Eng., Jamia Millia Islamia, New Delhi-110025, India
*Corresponding author, Mobile No. 0091-9990367319; E-mail: karimhali@yahoo.com,
#Mobile No. 0091-1126943808; E-mail: rasheed_jmi@hotmail.com

ARTICLE INFORMATION

Article history
Received 07 January 2011
Revised 17 March 2011
Accepted 31 March 2011
Available online 30 April 2011

ABSTRACT

A mathematical model of railway carriage on tangent tracks with single-point and two-point wheel-rail contact is considered. The railway carriage is modeled by 31 degrees of freedom which govern vertical, lateral, pitch, roll and yaw dynamic responses of wheelset, carbody and bogies. Linear stiffness and damping parameters of longitudinal, lateral and vertical primary and secondary suspensions are provided. Linear Kalker's and nonlinear Heuristic model is adopted to calculate the creep forces introduced at wheel-rail contact patch area. Computer aided-simulation is constructed to solve the differential equations of the mathematical model using Runge-Kutta fourth order method. Principle of limit cycle and phase plane approach is applied to evaluate the critical hunting velocity. The numerical simulation model used to study dynamic responses of carbody and bogies subjected to specific parameters of wheel conicity and primary suspension characteristics at critical hunting velocity. A comparison to study the sensitivity of railway carriage components to dynamic responses is also presented.

Keywords
railway carriage
conventional bogies
tangent tracks
lateral response
yaw response
critical hunting velocity

1. Introduction

Dynamic response analysis is used to study the dynamic behavior of railway carriage due to some external inputs such as rail irregularities, sudden disturbances, rail maneuvers, breaking or accelerating and other imperfections. Problems arise due to these undesirable inputs when railway carriage begin to move in different directions as vertical, pitch, roll, lateral and yaw directions. These movements cause vibrations and damage in railway components with uncomfortable ride passenger. As railway carriage laterally moves, carbody has a tendency to move in vertical direction to introduce hunt or bounce motion seriously. Pitch motion or the rotation of railway carriage about the lateral direction causes damage to the components of railway carriage due to strong coupling impacts. Pitch together with bounce motion introduce uncomfortable ride passenger in railway carriage. Roll motions or rotation about the longitudinal direction arise as the wheelsets move in the lateral direction over the track due to any external disturbances. Roll motion in railway carriage is so complicated since the critical roll motion may cause railway carriage to overturn. Lateral displacements occur due to imperfections and irregularities in the track which cause different undesirable motions like roll, yaw, and pitch. Lateral forces arise in the wheel-rail contact patch plane due to interactions between...
the wheel and the rail which force wheelsets to move laterally and may climb the rail. These introduced forces called creep forces in which depend upon different creep coefficients. The magnitudes of creep coefficients depend upon the wheel-rail geometry, normal load, and material properties. Many investigations used different magnitudes of creep coefficients and a combination of linear Kalter’s theory (1979) and nonlinear Heuristic is used in the present study. The study of railway carriage dynamic behavior should take into account the responses of railway carriage to these displacements and movements and more degrees of freedom should be considered to verify the accuracy of the system model. Different railway carriage models with different degrees of freedom are investigated and presented by many papers concerning railway carriage dynamic response. Dynamic stability of railway vehicle wheelsets and bogies having profiled wheels was presented by Wickens (1969) in which two degrees of freedom model was suggested govern lateral and yaw angle of each wheelset. Nonlinear mathematical model of dynamic simulation has been established with 7 degrees of freedom by Jawahar et al. (1990) which govern lateral and yaw movements of wheelsets and lateral, yaw and roll movements for both conventional and unconventional bogies. Xu et al. (2002, 2006) studied the dynamic analysis of coupled train-bridge systems under fluctuating wind and stated a railway carriage model that each 4-axle vehicle in a train is modeled by 27 degrees of freedom dynamic system. A vehicle model by Wang (1992) was developed to represent a 23 degrees of freedom conventional freight car, consisting of a carbody, two bolsters and two truck assemblies where the carbody was assigned five degrees of freedom govern vertical lateral, yaw, pitch, and roll while each bolster was assigned three degrees of freedom vertical, lateral and roll motion. Nath et al. (2005) studied the influence of yaw stiffness on the nonlinear dynamics of railway wheelset used two degrees of freedom model which govern the lateral and yaw motions. Nonlinear differential equations modeled by 8 and 10 degrees of freedom of railway carriage moving on curved tracks are presented by Lee et al. (2005, 2006) rain vehicle model considered by Kumaran et al. (2003) conforming to Indian railways consists of a vehicle body, two bogies with four wheelsets in which the system is modeled by 17 degrees of freedom. Mohan (2003) suggested railway carriage model comprising carbody, two bogies with four wheelsets, in which the railway model assigned two degrees of freedom which govern lateral and yaw motions of each wheelset bogies whereas the full vehicle assigned three degrees of freedom which govern lateral, yaw and roll motions. Study the effects of railway track imperfections on track dynamic behavior, and the effect of unsupported sleepers on the normal load of wheel-rail were investigated by Zhang et al. (2007) in which the system is modeled by 35 degrees of freedom that consider the lateral and vertical displacement, roll, pitch and yaw angle for the carbody, front and rear bogie frames and the four wheelsets. An ideal truck model with full frame decoupling represented by Dukkipati et al. (2001, 2003) which is modeled by 8 degrees of freedom. An investigation of dynamic interaction of long suspension bridges with running trains is presented by Xia et al. (2000) in which a 27 degrees of freedom model is used. A new finite element model for three-dimensional analysis of high-speed train-bridge interactions is proposed by Song et al. (2003) in which the equations of motion of the vehicle-bridge were derived using Lagrange’s equation where the carbody is considered with four degrees of freedom which govern bouncing, swaying, pitching, and yawing whereas bouncing, sliding, swaying, pitching, rolling and yawing motions are considered for the bogie. Li et al. (2007) investigated the problem of railway vehicle suspension estimation in which lateral and yaw modes are important and wheelsets and bogie have two degrees of freedom which govern lateral and yaw motions. A nonlinear model of a single wheelset moving with constant speed on a purely straight track is presented by Pater (1980), and the equations of motion were written down either as six equations containing the normal forces, or as four equations which do not contain the normal forces. Yugat et al. (2009) presented an analytical model of wheel-rail contact force due to the passage of a railway vehicle on a curved track used equations of motion govern vertical and roll motion of right and left wheel while vertical motion of right and left rail. A nonlinear wagon-track model with 23 degrees of freedom is presented by Sun et al. (2008) used to study rail corrugation formation due to the wheel stick-slip process. Rajib et al. (2008) presented equations of motion govern vertical motion of front and rear wheelset, bounce and pitch motion of bogie and bounce motion of carbody to study the dynamic analysis of railway vehicle-track interactions.

Railway vehicle dynamics during motion along a curved track is examined by Zboinski (1998, 1999) in which the dynamic behavior of the system is studied using two different methods, the quasi-statical and dynamical approach. In additional the research concerned the influence of vehicle suspension parameters as well as conditions of motion (speed, super-elevation, curve radius, transition curve existence) on limit cycle occurrence. The present study considers a railway carriage consists of carbody, two bogies and four conventional wheelsets modeled by 31-degrees of freedom which govern bounce, pitch, roll, lateral, and yaw motions of the system. The procedure done in this study is to derive the second order governing differential equations of motion of the full railway
carriage and transformed these equations into a set of first order differential equations using especial technique to facilitate solving them with numerical methods. Computer-aided simulation is used to solve these equations with Runge Kutta fourth-order method and represent the dynamic behavior of the system by increasing the forward speeds of the system to reach the critical hunting velocity. Principle of limit cycle approach Mohan (2003) is used to specify the critical hunting velocity of the system in which subjected to different amounts of wheel conicity and primary suspension parameters. The dynamic responses of carbody and bogies components of the railway carriage subjected to specific parameters of wheel conicity and primary suspension characteristics at critical hunting velocity are investigated using the constructed numerical simulation model. A comparison to study the sensitivity of railway carriage components to dynamic responses is also presented at critical hunting velocity.

Figure 1 Front view of railway carriage components equipped with sets of primary and secondary suspension elements.

2. Mathematical Railway Carriage Model

The railway carriage is a combination of components and wheelsets joining together by a set of different primary and secondary suspension elements, in which the full railway carriage configuration model system consists of carbody, two conventional bogies, and four wheelsets as shown in Figure 1. A railway carriage model of 31 degrees of freedom is constructed in this research to study the dynamic responses at critical hunting velocity of railway carriage components moving on tangent tracks. The differential equations of motion govern lateral displacement \( Y_w, Y_b, Y_c \) vertical displacement \( Z_w, Z_b, Z_c \) roll angle \( \phi_w, \phi_b, \phi_c \) and yaw angle \( \psi_w, \psi_b, \psi_c \) of wheelset, bogie and carbody respectively while pitch angle \( \theta_b, \theta_c \) of bogie and carbody. Railway carriage model is equipped with eight longitudinal, lateral and vertical primary suspensions of spring stiffness \( K_{px}, K_{py}, K_{pz} \) respectively and viscous damping constant \( C_{px}, C_{py}, C_{pz} \) respectively. Also the system is provided with eight longitudinal, lateral and vertical secondary suspensions of spring stiffness \( K_{sx}, K_{sy}, K_{sz} \) respectively and viscous damping
constant $C_{sx}$, $C_{sy}$, $C_{sz}$ respectively. Symbols and notations are illustrated in the nomenclature in Table 1. Dynamic behavior of railway carriage is caused by wheel-rail interactions in which creep forces are introduced at wheel-rail contact patch area. Non-conservative forces and elastic deformations at the contact patch introduce a phenomenon of creep and combination of linear Kalker's theory (1979) and nonlinear Heuristic is considered to calculate the introduced creep forces. Vibrations are transmitted through connected suspensions to other railway carriage components and the dynamical behavior of the system is governed by the equations of motion of each component of railway carriage.

2.1 Wheelsets Differential Equations of Motion
The railway carriage model is equipped with four conventional wheelsets in which consists of two wheels attached together by a solid axle. Wheelsets are used to steer and support the carriage. Wheelsets equations of motion are derived using Newton's laws with suspension, creep and normal forces in which some of these forces are calculated by Sen et al. (2005). The vertical, roll, lateral and yaw equations of motion of single wheelset are:

$$m_w \ddot{Z}_{wi} + 2C_{pz} \dot{Z}_{wi} - 2C_{pz} \ddot{Z}_{bj} - 2C_{pz} I_{zbj} \theta_{bj} + \frac{2f_{11}}{V} \ddot{\phi}_{wi} + \frac{2f_{11} r_o}{V} \phi_{wi} + \frac{2f_{12}}{V} \phi_{wi} Y_{wi} + \frac{2f_{12}}{V} \psi_{wi} = 0$$

(1)

$$J_{wz} \ddot{\phi}_{wi} + \left( \frac{2f_{11} a r_o \ddot{\alpha}}{V} + \frac{2f_{11} r_o^2 \alpha}{V} \right) \phi_{wi} + \frac{2f_{11} (r_o + a \ddot{\lambda})}{V} \dot{Y}_{wi} + 2C_{pz} I_{z} Z_{wi} - 2C_{pz} I_{z} \ddot{Z}_{bj}$$

(2)

$$m_w \ddot{Y}_{wi} + \left( \frac{2f_{11} f_{12}}{V} - 2C_{py} \right) \dot{Y}_{wi} + 2C_{py} \phi_{bj} = 0$$

$$-2f_{11} \psi_{wi} + W \phi_{wj} = 0$$

(3)

$$m_w \ddot{Y}_{wi} + \left( \frac{2f_{11}}{V} - 2C_{py} \right) Y_{wi} + 2C_{py} \phi_{bj}$$

$$+ \frac{2f_{12}}{V} \dot{\psi}_{wi} + \frac{2r_0 f_{11}}{V} \phi_{wj} - 2K_{py} \psi_{wi} + 2K_{py} Y_{bj}$$

$$-2f_{12} \dot{Y}_{wi} = 0$$

(4)

$$J_{wz} \ddot{\psi}_{wi} + \left( \frac{2a^2 f_{13}}{V} + \frac{2f_{12}}{V} \right) + 2C_{ps} L_s^2 \psi_{wi}$$

$$-2f_{12} \dot{Y}_{wi} - \left( \frac{J_{wz}}{r_o} + \frac{2r_o f_{12}}{V} \right) \phi_{wi}$$

$$-2C_{ps} L_s^2 \psi_{bj} = 2a^2 f_{13} Y_{wi} - 2K_{ps} L_s^2 \psi_{bj}$$

$$+ [2K_{ps} L_s^2 (-2f_{12} + a \alpha W)] \psi_{wj} = 0$$

2.2 Bogies Differential Equations of Motion
Railway carriage model consists of two bogies in which each bogie has two conventional unconnected front and rear wheelsets and two vertical secondary suspension elements are used to connect bogies with carbody in additional to the set of primary suspension elements connected each bogie with the wheelsets. The bogies differential equations of motion govern vertical, pitch, roll, lateral, and yaw degrees of freedom are:

$$m_b \ddot{Z}_{bj} + \left( 2C_{sz} + 4C_{pz} \right) \ddot{Z}_{bj} - 2C_{sz} I_{zbj} \theta_{bj}$$

(5)

$$-2C_{sz} \ddot{Z}_{cj} - 2C_{pz} \ddot{Z}_{wi} - 2C_{pz} \ddot{Z}_{wi}$$

$$+ (2C_{sz} + 4C_{pz}) \ddot{Z}_{bj} - 2C_{sz} \ddot{Z}_{bj} - 2C_{sz} \theta_{cj} - 2C_{sz} \theta_{cj} = 0$$

$$J_{by} \ddot{\theta}_{bj} + 4C_{ps} \ddot{\phi}_{bj} - 2C_{ps} I_{zbj} \ddot{Z}_{bj} + 2C_{ps} I_{zbj} \ddot{Z}_{bj}$$

(6)

$$+ 4K_{ps} I_{zbj} \ddot{\theta}_{bj} - 2K_{ps} I_{zbj} \ddot{Z}_{bj} + 2K_{ps} I_{zbj} \ddot{Z}_{bj} = 0$$

$$J_{bx} \ddot{\phi}_{bj} + (4C_{ps} + 2C_{sz} \ddot{\phi}_{bj} - 2C_{sz} \ddot{\phi}_{bj}$$

$$- 2C_{ps} I_{zbj} \ddot{\phi}_{bj} - 2C_{ps} I_{zbj} \ddot{\phi}_{bj} - 2C_{ps} I_{zbj} \ddot{\phi}_{bj}$$

$$+ 4C_{py} L_{ca} \ddot{Y}_{bj} - 2C_{py} L_{ca} \ddot{Y}_{bj}$$

(7)

$$+ (2K_{ps} + 2K_{py}) \ddot{\phi}_{bj} - 2K_{ps} \ddot{\phi}_{bj} - 2K_{ps} \ddot{\phi}_{bj}$$

$$- 2K_{ps} \ddot{\phi}_{bj}$$

$$- 2K_{ps} \ddot{\phi}_{bj} - 2K_{ps} \ddot{\phi}_{bj} - 2K_{ps} \ddot{\phi}_{bj}$$

$$- 2K_{ps} \ddot{\phi}_{bj}$$

$$- 2K_{py} L_{ca} \ddot{Y}_{bj} - 2K_{py} L_{ca} \ddot{Y}_{bj}$$

$$- 2K_{py} L_{ca} \ddot{Y}_{bj}$$

$$- 2K_{py} L_{ca} \ddot{Y}_{bj} = 0$$
m_{w} \dot{Y}_{bj} + (2C_{cy} + 4C_{py}) \ddot{Y}_{bj} - 2C_{sy} \dot{Y}_{c} - 2C_{py} \dot{Y}_{wi} \tag{8}
- 2C_{py} \ddot{Y}_{wi} + (2K_{s} + 4K_{py}) \dddot{Y}_{bj} - 2K_{s} \dddot{Y}_{c} \tag{8}
- 2K_{s} \dddot{Y}_{wi} - 2K_{py} \dddot{Y}_{wi} = 0
J_{bc} \psi_{bj} - \left( 4L_{x}^{2}C_{px} - 4L_{y}^{2}C_{py} \right) \psi_{bj}
- 2L_{x}^{2}C_{px} \psi_{wi} - 2L_{x}^{2}C_{py} \psi_{wi} - 2L_{y}^{2}K_{py} \psi_{wi} \tag{9}
+ 2L_{y}K_{py} \psi_{wi} - (4L_{x}^{2}K_{px} - 4L_{y}^{2}K_{py}) \psi_{bj}
- 2L_{x}^{2}K_{px} \psi_{wi} - 2L_{x}^{2}K_{py} \psi_{wi} - 2L_{y}K_{py} \psi_{wi} \tag{9}
+ 2bK_{py} \psi_{wi} = 0

2.3 Carbody Differential Equations of Motion

Carbody is the heaviest component in railway carriage makes crush between wheel and track and elastic deformation is introduced at contact patch area to produce creep forces and moments. Carbody differential equations of motion govern bounce, pitch, roll, lateral, and yaw degrees of freedom are derived as:

\begin{align*}
m_{c} \dddot{Z}_{c} + 4C_{sc} \dot{Z}_{c} = 2C_{sx} \dddot{Z}_{bj} - 2C_{sc} \dddot{Z}_{bj} + 4K_{sx} \dddot{Z}_{c} - 2K_{sx} \dddot{Z}_{bj} - 2K_{sc} \dddot{Z}_{bj} = 0 \tag{10}
J_{cy} \dddot{\theta}_{c} + 4C_{lc} \dot{I}_{c} \dddot{\theta}_{c} - 2C_{lc} \dot{I}_{c} \dot{Z}_{bj} + 2C_{lc} \dot{I}_{c} \dddot{Z}_{bj}
+ 4K_{lc} \dot{I}_{c} \dddot{\theta}_{c} - 2K_{lc} \dot{I}_{c} \dot{Z}_{bj} + 2K_{sc} \dot{I}_{c} \dddot{Z}_{bj} = 0 \tag{11}
J_{cy} \dddot{\phi}_{c} + 4C_{lc} \dot{I}_{c} \dddot{\phi}_{c} - 2C_{lc} \dot{I}_{c} \dot{\phi}_{bj} - 2C_{sc} \dot{I}_{c} \dddot{\phi}_{bj}
+ 4C_{ly} \dddot{L}_{c} \dot{y}_{c} - 2C_{ly} \dddot{L}_{c} \dot{y}_{bj} - 2C_{ly} \dddot{L}_{c} \dot{y}_{bj} + 4K_{sc} \dot{I}_{c} \dddot{\phi}_{bj} - 2K_{sc} \dot{I}_{c} \dddot{\phi}_{bj} = 0 \tag{12}
\end{align*}

3. Numerical Simulation

Railway carriage runs on tangent tracks is modeled by the second order differential equations of motion (1-14). A simple and important technique used to transform the governing equations of motion into first order differential equations in suitable form known as state space equations. This technique is used to facilitate solving the equations with numerical integration methods. The transformed equations of motion are simulated with computer-aided simulation to be solved by fourth order Runge-Kutta numerical method. Table 2 represents the data used in numerical simulation from resources (Mohan A., 2003; Lee et al., 2005). Also initial conditions are assumed for the dynamic motions of the system. Simulation is executed to represent the dynamic responses of railway carriage components subjected to different parameters. Procedure is achieved by increasing the speeds to reach the critical velocity and principles of phase plane approach are utilized to represent the critical hunting velocity of the system.
### Table 2: The Numerical Simulation Data

<table>
<thead>
<tr>
<th>Notation</th>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
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<tbody>
<tr>
<td>( f_{11} )</td>
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<td>( N )</td>
<td>( 9.43 \times 10^6 )</td>
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<td>( f_{12} )</td>
<td>Lateral/spin creep coefficient</td>
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<td>( f_{22} )</td>
<td>Spin creep coefficient</td>
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<td>Forward creep coefficient</td>
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<td>Mass of wheelset</td>
<td>( kg )</td>
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</tr>
<tr>
<td>( m_b )</td>
<td>Mass of bogie</td>
<td>( kg )</td>
<td>( 3.086 \times 10^3 )</td>
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<tr>
<td>( m_c )</td>
<td>Mass of carbody</td>
<td>( kg )</td>
<td>( 4.820 \times 10^4 )</td>
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<td>( J_{wx} )</td>
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<td>( kg.m^2 )</td>
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<td>( kg.m^2 )</td>
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</tr>
<tr>
<td>( J_{wz} )</td>
<td>Yaw mass moment of inertia of Wheelset</td>
<td>( kg.m^2 )</td>
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<td>Centered rolling radius of wheel</td>
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<td>( a )</td>
<td>Half of track gage</td>
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<td>( d_p )</td>
<td>Half distances between primary longitudinal suspensions</td>
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<td>( \mu )</td>
<td>Coefficient of friction between wheel and rail</td>
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<td>Wheel conicity</td>
<td>( rad )</td>
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<td>( K_{py} )</td>
<td>Lateral primary suspension spring stiffness</td>
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<td>( 9048.2 )</td>
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<tr>
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<td>Vertical primary damping coefficient</td>
<td>( N.s/m )</td>
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<tr>
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<td>Lateral primary damping coefficient</td>
<td>( N.s/m )</td>
<td>( 1.75 \times 10^4 )</td>
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<tr>
<td>( C_{sz} )</td>
<td>Vertical primary damping coefficient</td>
<td>( N.s/m )</td>
<td>( 2.75 \times 10^4 )</td>
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<tr>
<td>( K_{rail} )</td>
<td>Lateral rail stiffness</td>
<td>( N/m )</td>
<td>( 14.6 \times 10^4 )</td>
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<tr>
<td>( C_{rail} )</td>
<td>Lateral rail damping coefficient</td>
<td>( N.s/m )</td>
<td>( 14.6 \times 10^4 )</td>
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</table>
A comparison is presented in this research to study the dynamic behavior of carbody and the two leading and trailing conventional bogies in which are components of single railway carriage at critical hunting velocity. The considered simulation railway carriage model is used to represent the dynamic behavior of carbody and leading and trailing bogies due to lateral, yaw, roll, vertical and pitch dynamic responses at critical hunting velocity. According to the principles of limit cycle approach which is handled to realize the critical hunting velocity of the system Figures 2-6 show that lateral and yaw dynamic behavior of carbody and the bogies are the most sensitive dynamic responses to critical hunting velocity with different magnitudes of lateral and yaw displacements. As shown in Figures 2-3 lateral and yaw displacements have less magnitudes in carbody than the two leading and trailing bogies in which means that accurate suspension and wheel parameters have been selected in this present simulation model to satisfy the ride passenger comfortable. It is also observed that lateral and yaw displacement of trailing bogie are less than the front bogie and that depends about the initial yaw moment of the carody. The roll dynamic response of carbody is also sensitive to the critical hunting velocity but with small magnitudes of roll displacement meanwhile roll displacements of leading and trailing bogies are not affected and have closer magnitudes as shown in Figure 4. Vertical or bounce dynamic behavior of railway carriage components is depicted in Figure 5 in which shows that the railway carriage simulation model presents acceptable magnitudes of vertical displacements for carbody and the leading and trailing bogies at critical hunting velocity. In addition vertical displacements of carbody has more magnitudes than the vertical displacements of the bogies in which can be interpreted that two sets of primary and secondary suspensions are joined with the bogies. Figure 6 presents the pitch dynamic behavior of carbody and the leading and trailing bogies with acceptable magnitudes of pitch displacement for the components of railway carriage. In the figure it can be observed less magnitudes of pitch displacement for carbody than the bogies and that satisfy the ride passenger comfortable. The constructed simulation model of railway carriage moving on tangent tracks used to analysis the dynamic behavior is able to represent the most dynamic responses and behaviors of carbody and bogies. It is concluded that lateral and yaw dynamic responses for carbody and bogies can be handled to improve the critical hunting velocity of railway carriage since these dynamic displacements are more sensitive at critical hunting velocity. Improvement of critical hunting velocity and eliminate the undesirable hunting phenomenon is achieved by increasing the critical hunting velocity of the railway carriage in which can be satisfied by using the appropriate values and magnitudes of suspension parameters to the components which are sensitive to the critical hunting velocity.

References


