Palli et al. 2015. *Int. J. Vehicle Structures & Systems*, 6(4), 16-20
ISSN: 0975-3060 (Print), 0975-3540 (Online)
doi: 10.4273/ijvss.7.1.03

International Journal of
Vehicle Structures & Systems
Available online at www.maftree.org/eja

Dynamic Analysis of Indian Railway Integral Coach Factory Bogie

Srihari Palli[a], Ramji Koona[b], Rakesh Chandmal Sharma[c] and Venkatesh Muddada[d]

[a,d]*Mech. Engg. Dept., Aditya Institute of Technology and Management, Tekkali, India*
[a]*Corresponding Author, Email: srihari.palli@gmail.com*
[d]*Email: venkateshmuddada328@gmail.com*

[b]*Mech. Engg. Dept., Andhra University College of Engineering, Visakhapatnam, India*
Email: ramjidme@gmail.com

[c]*Mech. Engg. Dept., Maharishi Markandeshwar University, Mullana, India*
Email: drrcsharma@mmumullana.org

ABSTRACT:

Dynamic response of railway coach is a key aspect in the design of coach. Indian railway sleeper and 3 tier AC coaches consist of two railway bogies, where the central distance of the center of gravity between the bogies is 14.9 m. Analysis of railway bogie forms a basis for investigating the behaviour of the coach as a whole. The current work carried out is, vehicle dynamic response in terms of Eigen frequency modal analysis and harmonic analysis of a Indian railway 6 Ton Integral Coach Factory (ICF) bogie using finite element (FE) method. The entire bogie model is discretized using solid92 tetrahedral elements. The primary and secondary suspension systems are modelled as COMBIN14 elements in the FE model of the bogie. Modal analysis of the bogie model using Block Lanczos method in ANSYS is carried out to extract first few natural modes of vibration of the bogie. The roll mode frequency attained in Modal analysis is in good agreement with the fundamental frequency calculated analytically. Sinusoidal excitation is fed as input to bottom wheel points to analyse the harmonic response of the bogie in terms of displacement at different salient locations. Harmonic response results reveal that the bogie left and right locations are more vulnerable than the locations near the centre of gravity of the bogie.

KEYWORDS:

Dynamic response; Modal analysis; Harmonic analysis; Finite element model; ANSYS

CITATION:

S. Palli, R. Koona, R.C. Sharma and V. Muddada. 2015. Dynamic Analysis of Indian Railway Integral Coach Factory Bogie, *Int. J. Vehicle Structures & Systems*, 7(1), 16-20. doi:10.4273/ijvss.7.1.03.

ACRONYMS AND NOMENCLATURE:

CG	*Center of Gravity of Bogie*
CGL	*Center of Gravity Left of Bogie*
CGR	*Center of Gravity Right of Bogie*
DoF	*Degrees of Freedom*
FE	*Finite Element*
FLBT	*Front Left Bogie Top*
FLBB	*Front Left Bogie Bottom*
FRBT	*Front Right Bogie Top*
FRBT	*Front Right Bogie Bottom*
ICF	*Integrated Coach Factory*
PSD	*Power Spectral Density*
RLBT	*Rear Left Bogie Top*
RLBB	*Rear Left Bogie Bottom*
RRBT	*Rear Right Bogie Top*
RRBT	*Rear Right Bogie Bottom*

1. Introduction

If a passenger riding in a rail vehicle does not think about vibration, which is because rail-vehicle dynamics engineers have succeeded in achieving good ride comfort. The current trend towards lighter vehicles and higher speeds makes the issue of car body's structural flexibility crucial in the design and development of competitive vehicles. Also, the demand for better comfort calls for a better understanding of the passenger car body interaction [12]. The vibrations are mainly caused by track irregularities, from which they are transmitted via the bogies and the car body to the passengers. The car body is not rigid, but bends and twists from the excitation coming from the bogies.

The railway coach motion on the rails is a combination of vertical and lateral motions. Dynamic performance of chassis structure was studied using FE techniques by finding the Natural frequencies and mode shapes [10]. Dynamic performance of chassis structure was studied by developing a FE model of a diesel engine chassis and performed modal analysis for varying boundary conditions using ANSYS software [9]. The dynamic interaction of the vehicle-track system was analyzed by assuming the vehicle model as a rigid body subjected to a concentrated force, and represented a bogie carrying half of the car body weight [8]. The rail was characterized as a finite Euler beam with discrete supports, and the sleeper was modelled as a rigid body. Corrugation of the rail surface is assumed as a sinusoidal wave. Random process theory was adopted to analyze inspected track data from two railway lines in India and concluded that irregularity of the longitudinal level may be modelled as a stationary Gaussian random process [7]. The primary and secondary hunting speeds of the railway vehicle were determined to investigate the dynamic stability. Critical parameters which influence the railway vehicle dynamic stability were analysed [1]. The dynamic response of a passenger vehicle in terms of acceleration and strain was computed at all nodes by giving PSD of acceleration as input to the tires of a passenger vehicle using random response [6, 13]. FE technique was used for analysis of bogie frame under load conditions such as vertical loads, transverse loads, self weight of bogie frame, torque arm reaction loads with the usage of spring, shell, rigid and gap elements [11]. Three types of practically important imperfections in the vehicle/track system were investigated [5]. The rail corrugation and wheel flat were assumed as sinusoidal functions. The ride behaviour of the rail vehicle was studied by varying its one parameter at a time in order to estimate its individual effect on vertical and lateral ride [3]. Coupled vertical-lateral mathematical model of an Indian Railway General Sleeper coach using Lagrangian dynamics and its motion has been studied [4]. It was concluded that in developing the mathematical model to study vertical response, it would not be adequate to include bounce, pitch and roll DoF of the components but yaw and lateral DoF also need to be considered [2].

Literature survey reveals that various methodologies have been adopted by researchers across globe in the study of dynamic behaviour of the railway coach and bogie/chassis. FE analysis is used in the study of railway bogie and analysing its dynamic response. FE software has been found vital in performing dynamic analysis and also finding the natural frequencies of the vehicle under operating conditions. The work carried out in this paper is Eigen frequency modal and harmonic analyses conducted on an Indian Railway 6Ton ICF bogie. The first few natural frequencies of the ICF bogie are obtained in modal analysis. Sinusoidal excitation is fed as input for the harmonic analysis and the harmonic peaks obtained are compared with natural frequencies obtained in modal analysis. The details of methodology for design and modelling of bogie, modal and harmonic analysis results are presented followed by conclusions.

2. Design and FE model of ICF bogie

The frame of the ICF bogie is a fabricated structure made up of mild steel channels and angles welded to form the main frame of the bogie. The ICF bogie consists of a body bolster and a bogie bolster. The body bolster is welded to the coach body whereas the bogie bolster is a free floating member which takes the entire load of the coach through the body bolster. The body bolster transfers the dead weight of the coach body to the bogie frame [14]. The bogie consists of a two stage suspension and two pairs of wheels and axles. The suspension between the axles and the bogie is called primary suspension and that between the bogie and the car body is called secondary suspension. UNIGRAPHICS NX 7.5 is used for geometric modeling of the Indian Railway 6 Ton ICF bogie as shown in Fig. 1. The assumptions in geometric modeling of bogie are:

- Geometric features which are insignificant from load bearing point of view are suppressed.
- The curvature of the bogie frame where cross-section changes takes place is neglected.
- Bogie frame, wheel set, axle set and bogie bolster are modelled and remaining components are neglected.

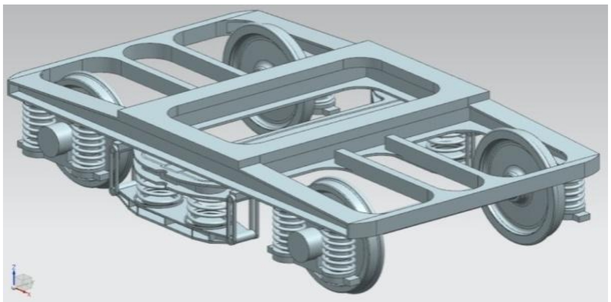

Fig. 1: Geometric model of ICF bogie

The geometric model of bogie is exported to ANSYS in parasolid format. Primary and secondary suspensions modelled in geometric model have been replaced with spring elements using COMBIN14. Fig. 2 represents the FE model generated after tetrahedral meshing using SOLID92 elements in ANSYS. The FE mesh is made up of 3,80,217 nodes and 2,14,545 elements. The assumptions in FE modeling of bogie are:

- The primary and secondary suspensions are modelled as linear spring elements in the FE model.
- As majority of the material in the bogie body is steel, material properties of steel are considered entirely for the element types used in the FE model of the bogie.

The material properties considered for different bogie components of steel are given in Table 1. Various constants considered for primary and secondary suspension stiffness and damping have been taken from the Indian Railways maintenance manual of BG coaches and given in Table 2 and Table 3 respectively.

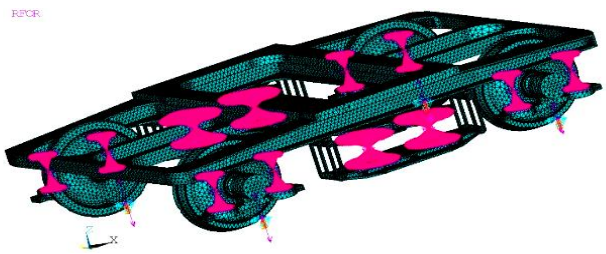

Fig. 2: FE model of ICF bogie

Table 1: Material properties of Steel

Property	Density ρ (kg/mm^3)	Young's modulus E (N/mm^2)	Poisson's Ratio (υ)
Value	7.85×10^{-9}	2.0×10^{5}	0.3

Table 2: Stiffness values

Parameter	Value (N/m)
Primary suspension - Vertical stiffness between wheel & bogie frame (Kpz)	1.077×10^{8}
Primary suspension - Lateral stiffness between wheel & bogie frame (Kpy)	2.3×10^{9}
Secondary suspension - Vertical stiffness between bogie frame & bolster (Ksz)	1.695×10^{6}
Secondary suspension - Lateral stiffness between bogie frame & bolster (Ksy)	4.648×10^{5}

Table 3: Damping coefficients

Parameter	Value (Ns/m)
Primary suspension - Vertical damping coefficient between wheel & bogie frame(Cpz)	8.2×10^{4}
Primary suspension - Lateral damping coefficient between wheel & bogie frame(CPy)	1×10^{6}
Secondary suspension -Vertical damping coefficient between bogie frame & bolster (Csz)	1.18×10^{5}
Secondary suspension - Lateral damping coefficient between bogie frame & bolster(Csy)	2×10^{6}

3. Modal analysis results and discussion

The first step in any dynamic analysis of a system is to understand how it behaves when it is just disturbed momentarily and then left to oscillate freely. The natural frequency and mode shapes of a given vehicle system are extracted by using free vibration analysis. The Block Lanczos Eigen value solver in ANSYS is used to predict the natural modes of vibration. Block Lanczos method is used as this method computes first few natural frequencies and mode shapes for large and symmetric structures efficiently. The first few natural frequencies for unladen (without passenger load) condition and laden (with passenger load lumped at center of gravity of the bogie) condition are extracted and are tabulated as given in Table 4. The bounce mode of the bogie is observed for both unladen and laden conditions in the second mode. The pitch mode is observed for both unladen and laden conditions in mode 5. Similarly, the twist mode can be observed in mode 6 for unladen and in mode 7 for laden conditions. The mode shapes of the bogie obtained are shown from Fig. 3 to Fig. 8.

Table 4: Natural frequencies of bogie

Mode No	Frequency for Unladen Condition(Hz)	Frequency for laden condition (Hz)
1	0.5463	0.5413
2	7.6805	7.6794
3	7.8939	7.8939
4	12.029	9.3747
5	13.012	13.145
6	23.344	23.959
7	24.652	24.623
8	28.651	32.947
9	32.692	33.316
10	33.316	33.335
11	33.334	34.746
12	35.301	38.590
13	39.812	39.921

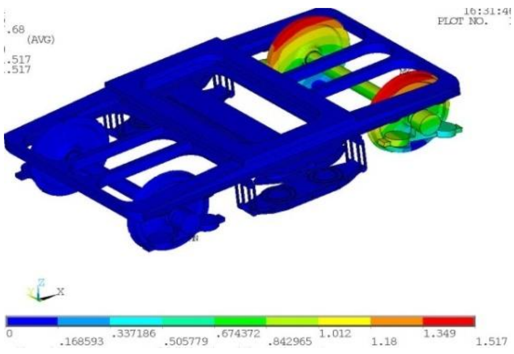

Fig. 3: Bounce mode of ICF bogie (unladen)

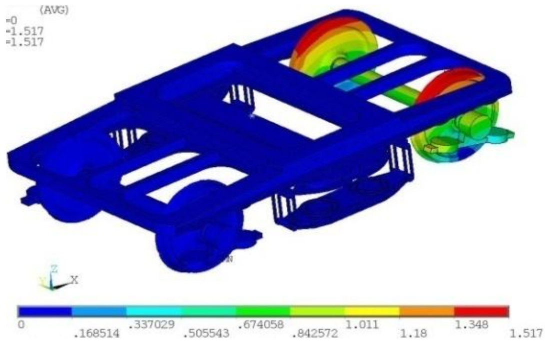

Fig. 4: Bounce mode of ICF bogie (laden)

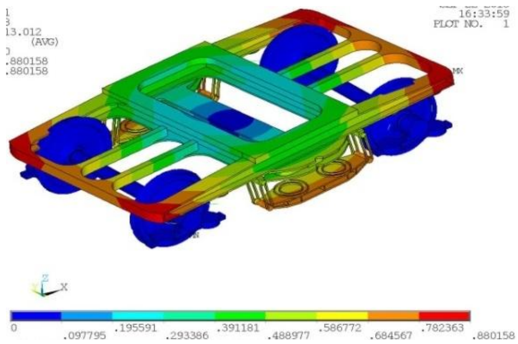

Fig. 5: Pitch mode of ICF bogie (unladen)

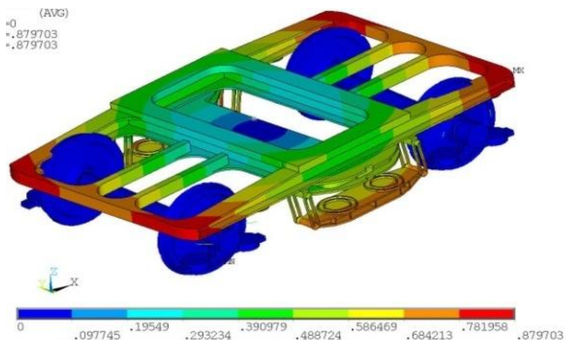

Fig. 6: Pitch mode of ICF bogie (laden)

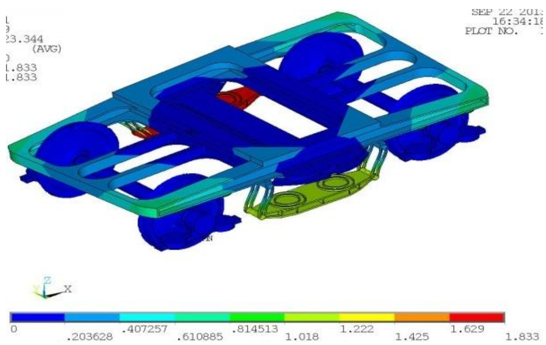

Fig. 7: Twist mode of ICF bogie (unladen)

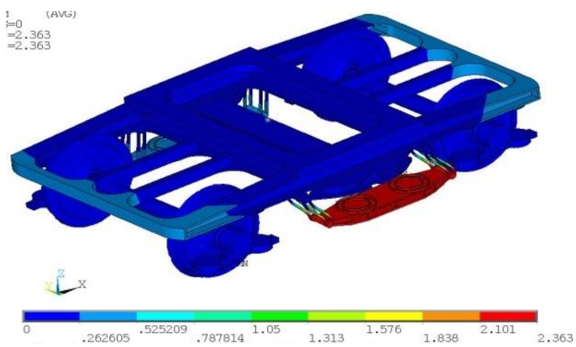

Fig. 8: Twist mode of ICF bogie (laden)

4. Harmonic analysis results and discussion

Harmonic response of the bogie is found by giving sinusoidal excitation to identified wheel rail contact nodes in the form of frequency and amplitude. Sinusoidal excitation is fed as input to the identified wheel bottom nodes in the vertical direction by considering maximum amplitude of 25 mm and frequency range of 0-50 Hz. Fig. 9 represents the various prominent locations of the bogie frame.

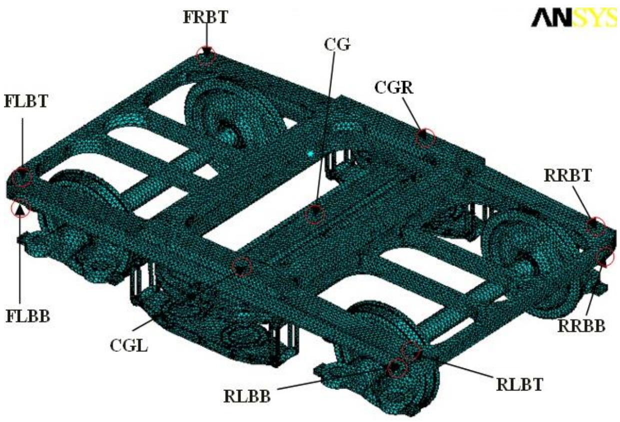

Fig. 9: Prominent locations of bogie

The variation in response due to this excitation at identified prominent locations of the bogie frame is plotted against frequency as shown in Figs. 10 to 12. The harmonic peaks attained are compared with natural frequencies obtained during Eigen frequency modal analysis of the bogie model. Variation of displacement with frequency at CG of bogie frame, CGL and CGR locations is shown in Fig. 10. The first harmonic peak obtained at a frequency of 7.7 Hz i.e., at bounce mode frequency. The domination of amplitude observed at first harmonic peak (7.7 Hz) i.e., at bounce mode frequency at CGL and CGR locations than CG location with an amplitude difference of 0.12 mm. The second harmonic peak observed at a frequency of 40 Hz i.e., at pitch 2 mode frequency. There is more displacement observed from graph at CGR location than other portions at pitch mode. The difference in amplitude at second peak in between CG and CGL is 0.26 mm. And another peak is observed at a frequency of 41.5 Hz at CGL location. The difference in amplitude at second peak between CG and CGR is higher than at first peak. It can be concluded from graph that the more disturbances are experienced at CGL and CGR locations than CG location in higher frequency range. In the frequency range from 8 Hz to 22 Hz the response comes down.

The variation in amplitude over the entire frequency range at FLBT, FLBB, FRBT and FRBB is shown in Fig. 11. Similar to CGL and CGR locations, for FLBT, FLBB, FRBT, and FRBB the first harmonic peak attained at a frequency of 7.7 Hz i.e., at bounce mode frequency. Few other peaks can be observed at frequencies of 24.7 Hz, 28.5 Hz, and 40 Hz matching well with the natural frequencies of modal analysis. It can be concluded from the graph that at any portion of bogie frame shows the same response over the entire frequency range and is subjected to higher displacement at third harmonic peak (28.5 Hz) at than other frequencies. The variation in displacement at RLBT, RLBB, RRBT and RRBB is shown in Fig. 12. It shows the same response that observed from front portion with small amplitude difference. The first peak obtained at a frequency of 7.7 Hz with amplitude of 0.067 mm. Subsequent peaks can be observed at 24.7 Hz, 28.5 Hz, 33.5 Hz and 40 Hz respectively matching with the natural frequencies. High displacement response observed at fifth harmonic peak for RLBT i.e., at 40 Hz with an amplitude of 0.29 mm. In the frequency range from 8 Hz to 17 Hz the disturbances are less at rear portion as compared with other frequencies.

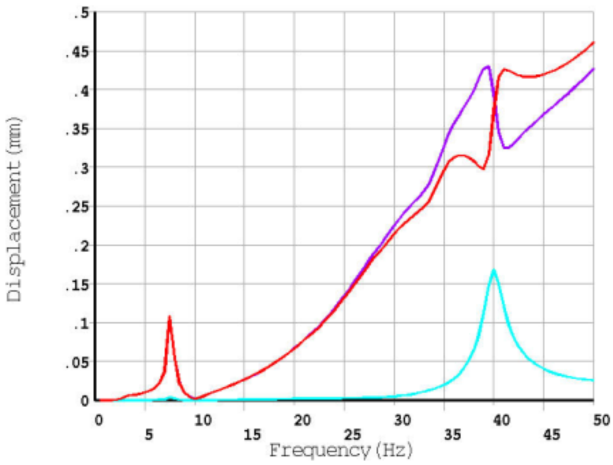

Fig. 10: Response of and around CG of bogie

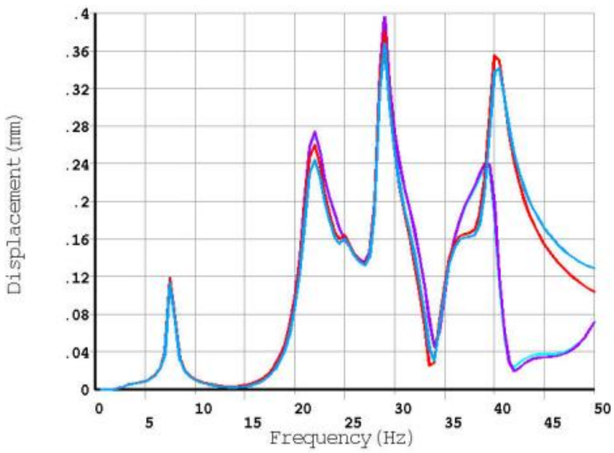

Fig. 11: Response of prominent locations at front portion of bogie

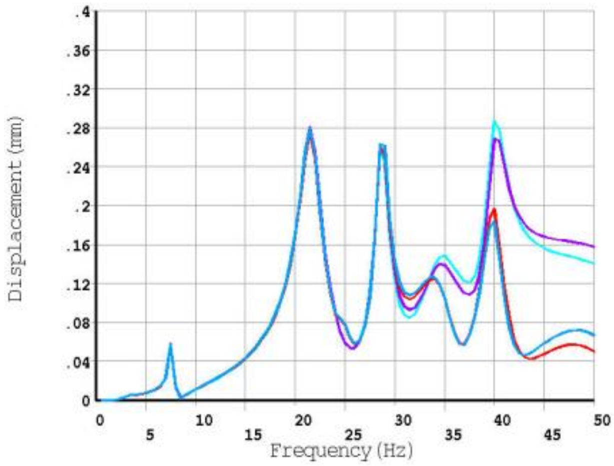

Fig. 12: Response of prominent locations at rear portion of bogie

5. Conclusions

Modal analysis results reveal that for both unladen and laden conditions the bogie attains similar natural frequencies with minor deviations. This means passenger load has less significance on the natural frequencies and modes of the ICF bogie. Further various mode shapes of the bogie illustrate that bounce, pitch and twist modes are predominant which influence the dynamic behaviour significantly. From harmonic response, it can be observed that first peak response is matching with first fundamental frequency of the bogie and further peaks are also attained at frequencies nearer to consequent fundamental frequencies of the bogie. The displacements at front and rear bogie CGL and CGR locations are much higher than that of CG at first resonance frequency. It should be avoided by controlling the operating speed of vehicle by running at above or below the first resonant frequency. The displacement response for bogie is lower than that of the first peak at higher frequencies i.e. between 8 Hz - 17 Hz than that of the first peak which 7.7 Hz. It seems that by increasing mass the displacement response is decreasing. It can be concluded that the displacement of the bogie left and right locations are more vulnerable than the locations near to the centre of gravity of the bogie.

REFERENCES:

[1] R.C. Sharma. 2013. Stability and eigenvalue analysis of an Indian railway general sleeper coach using Lagrangian dynamics, *Int. J. Vehicle Structures & Systems*, 5(1), 9-14. http://dx.doi.org/10.4273/ijvss.5.1.02.

[2] R.C. Sharma. 2012. Recent advances in railway vehicle dynamics, *Int. J. Vehicle Structures & Systems*, 4(2), 52-63. http://dx.doi.org/10.4273/ijvss.4.2.04.

[3] R.C. Sharma. 2011. Parametric analysis of rail vehicle parameters influencing ride behaviour, *Int. J. Engg. Sci. & Tech.*, 3(8), 54-65.

[4] R.C. Sharma. 2011. Ride analysis of an Indian railway coach using Lagrangian dynamics, *Int. J. Vehicle Structures & Systems*, 3(4), 219-224. http://dx.doi.org/10.4273/ijvss.3.4.02.

[5] C.Y. Li and T.H. Young. 2003. Vertical vibration analysis of vehicle/imperfect track systems, *Vehicle System Dynamics*, 40(5), 329-349. http://dx.doi.org/10.1076/vesd.40.5.329.17912.

[6] N. Karuppaiah, P.S. Deshpande, C. Sujatha and V. Ramamurthi. 2003. Vibration analysis in a light passenger vehicle by rigid body/finite element modeling, *Advances in vibration Engineering*, 2(2), 106-120.

[7] R.N. Iyengar and O.R. Jaiswal. 1995. Random field modelling of railway track irregularities, *J. Transportation Engineering*, 121(4), 304-308. http://dx.doi.org/10.1061/(ASCE)0733-947X(1995)121:4(303).

[8] T.J.S. Abrahamsson, and J.C.O. Nielson. 1992. Coupling of physical and modal components for analysis of moving non-linear dynamic systems on general beam structures, *Int. J. Numerical Methods in Engineering*, 33(9), 1843-1859. http://dx.doi.org/10.1002/nme.1620330906.

[9] J.K. Haghighi and G.W. Krutz. 1988. Dynamic analysis of an engine chassis mount bracket using the finite element method, *SAE Int. Technical Paper*, 881854, 6.1750-6.1758.

[10] R. Ali, J.L. Hedges, B. Mills, C.C. Norville and O. Gurdogan. 1970. The Application of finite element techniques to the analysis of automobile structure, *Proc. IMechE*, 185, 44-71.

[11] P.S. Paul, M.D.M. Gift, C.P.J. Sakthivel. 2002. Finite Element analysis of bogie frame, *Proc.18th National Convention of Mechanical Engineers*, 331-338.

[12] P. Carlbom. 2000. Carbody and Passengers in Rail Vehicle Dynamics, *PhD Thesis*, Dept. of Vehicle Engineering, Royal Institute of Technology, Sweden.

[13] N. Karuppaiah, C. Sujatha and V. Ramamurthi. 1999. Modal and vibration/stress analysis of a passenger vehicle by FEM, *SAE Int. Tech. Paper*, 990003, 129-133.

[14] Indian Railways Centre for Advanced Maintenance Technology, A Directorate of RDSO. *Maintenance Manual for BG coaches of ICF design*, 3, 1-36.

EDITORIAL NOTES:

Edited paper from International Conference on Newest Drift in Mechanical Engineering, 20-21 December 2014, Mullana, Ambala, India.

GUEST EDITOR: Dr. R.C. Sharma, Dept. of Mech. Engg., Maharishi Markandeshwar University, Mullana, India.