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Stress and Vibrational Analysis of an Indian Railway RCF Bogie

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ABSTRACT:

In the present work, static and dynamic finite element analyses are carried out on an Indian railway 6 ton RCF sleeper bogie. The geometrical CAD model of railway vehicle has been developed in UG-NX7.5 and has been exported to ANSYS12.1 package where finite element modelling and the required static and dynamic analyses have been performed. For dynamic response, modal, harmonic and transient dynamic analyses are carried out. First few natural modes of vibration of the bogie are extracted in Eigen frequency analysis and it is observed that the roll mode attained at a frequency which is well matched with the fundamental natural frequency calculated analytically. The harmonic peaks obtained are matching well with the natural frequencies obtained in modal analysis. Response to the ground excitation when the bogie passes over a bump is simulated in transient analysis.

KEYWORDS:

Railway bogie; Static analysis; Transient analysis; Finite element modelling; Laden sleeper; Unladen sleeper

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1. Introduction

Railway engineers and researchers strive for better ride comfort of coaches with robust designs. Static and dynamic behaviour of railway bogie plays a crucial role in the development and evolution of railway coaches and its components over the past few decades. Static analysis is done to determine the stresses and deformations in the bogie under different load conditions, whereas the dynamic analysis is carried out in terms of free and forced vibration analysis to extract the natural frequencies and understand the influence of these natural frequencies under operating conditions of the railway coach. Finite element (FE) stress analysis of three wheeler chassis using SDRC IDEAS software have been obtained by researchers [1-3] under critical loads, simulating Indian railroad conditions by considering dead weight of vehicle, passengers, driver and ground excitations. The dynamic response of a passenger vehicle in terms of acceleration and strain had been computed at all nodes by giving PSD of acceleration as input to the tires of a passenger vehicle using random response FE analysis [4-5].

Researchers [6-7] had studied dynamic performance of a chassis structure using FE techniques. They have predicted natural frequencies and mode shapes and these results are compared with experimental results. A computer-based FE model of a diesel engine chassis mounting bracket was developed and modal analysis was performed by for varying boundary conditions using ANSYS software. Static analysis was performed to evaluate the stress and displacements at various points under operating load conditions. Work was carried out on FE modelling and 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 using IDEAS software [8-9]. Simulations of railway vehicle through FEM have been performed by several authors in the past [10-14]. Harmonic and PSD response spectrum analyses of an Indian railway RCF bogie have been carried out using FE analysis [15].

Researchers [16] had worked on three types of practically important imperfections in the vehicle/track system were investigated. The rail corrugation and wheel flat were assumed as sinusoidal functions. Coupled vertical-lateral mathematical model of an Indian railway general sleeper coach using Lagrangian dynamics and its motion has been studied. It was concluded that in developing the mathematical model to study vertical response, it would not be adequate to include bounce, pitch and roll degree of freedom (DoF) of the components but yaw and lateral DoF also need to be considered [17-19]. The ride behaviour of the rail vehicle has been studied by varying its one parameter at a time in order to estimate its individual effect on vertical and lateral ride [20]. Wheel axle is usually modelled rigid however in order to analyse high-frequency effects at the wheel/rail contact its flexible modes have been considered [21-22]. The current work carried out is static and dynamic FE analyses conducted on an Indian railway 6 ton RCF sleeper bogie.

Four static analyses i.e. unladen sleeper, laden sleeper, unladen 3 tier AC and laden 3 tier AC bogie are performed each with their corresponding loads. The loads are half the dead weight of the entire coach for unladen case and half the dead weight plus weight of 36 passengers for the laden case. Static analysis results revealed that the maximum deformation of 3 tier AC bogie is on the higher side when compared to sleeper bogie for both laden and unladen conditions because of higher mass on 3 tier AC bogie. The location of maximum von-misses equivalent stress occurred at the connecting point of bolster spring-suspension hanger and the side frame of bogie for both laden and unladen conditions of sleeper and 3 tier AC bogies. The von-misses stresses obtained in all the cases lie well below the standard limits. In order to investigate the dynamic response, modal and transient dynamic analyses are carried out. Response to the ground excitation when the bogie passes over a bump is simulated in transient analysis. The response at key locations within the vicinity of the centre of gravity, front portion and rear portion of the bogie has been plotted against time.

2. Modelling of railway bogie

The bogie consists of a two stage suspension and two pairs of wheels and axles as shown in Fig. 1. 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. The wheel and axle sets are mounted in spherical roller bearings on either ends of the axle. The frame of the RCF (Railway Coach Factory) bogie is a fabricated structure made up of mild steel channels and angles welded to form the main frame of the bogie. There are two types of bolsters in a RCF coach i.e. body bolster and 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. Model is formulated with following assumptions:

- Half of the car body superstructure mass is lumped at the center of gravity of the bogie frame.
- 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.
- The suspensions are modelled as linear spring elements in the FE Model.

Bogie frame, primary and secondary suspension, wheel set, axle set, bogie bolster are modelled in the analysis and remaining components are neglected. Fig. 1 shows the general bogie arrangement with labelling of its major components. UNIGRAPHICS NX 7.5 is used for geometric modelling of the Indian railway 6 ton RCF bogie manufactured by ICF/RCF is shown in Fig. 2. This model is exported to ANSYS in par solid format. Fig. 3 represents the FE model generated after tetrahedral meshing using SOLID92 elements. Primary and secondary suspension modelled as spring elements using COMBIN14. FE mesh is made up of 380217 nodes and 214545 elements.

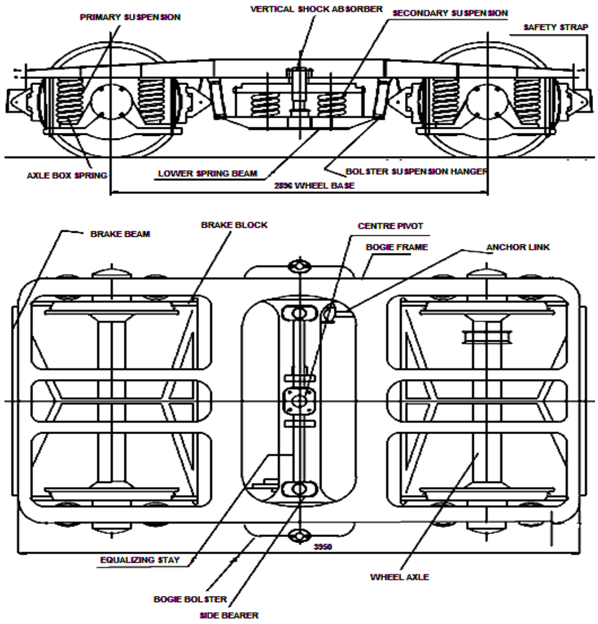

Fig. 1: Bogie general arrangement [21]

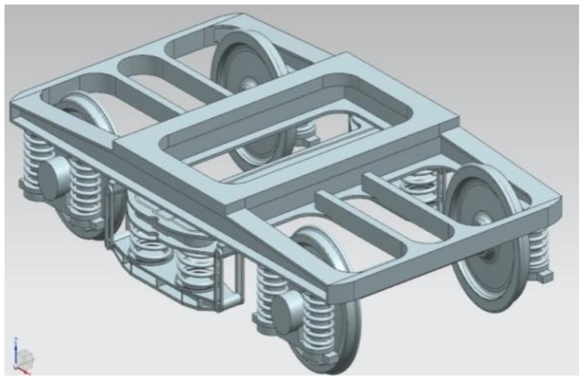

Fig. 2: Geometric model of RCF bogie (NX7.5)

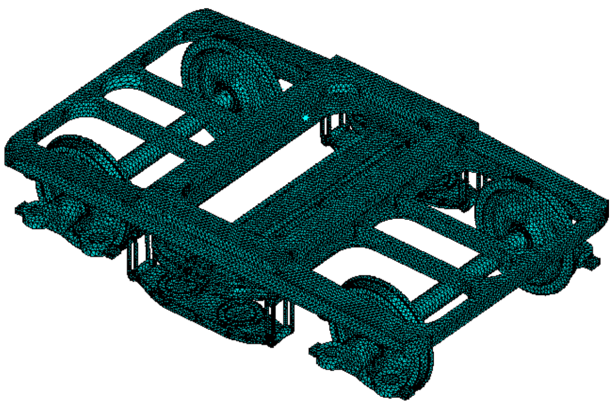

Fig. 3: FE model of bogie

Various constants considered for primary and secondary suspension stiffness and damping have been taken from the Indian railways maintenance manual. The dead and live load values, stiffness and damping values of suspensions used in the analysis are presented in Tables 1 to 7 respectively. The mass of the railway coach chassis is lumped at appropriate nodes in z-direction. 6 ton RCF bogies of sleeper and 3 tier AC coaches are selected for analysis purpose. The specifications of the bogies are shown in Table 1. All bogie components are assumed to be made of steel and its properties are given in Table 6.

Table 1: Unladen and laden loads

Weight	Unladen sleeper	Unladen 3 tier AC	Laden sleeper	Laden 3 tier AC
(N)	1.962×10^5	2.4525×10^5	2.19168×10^5	2.68218×10^5

Table 2: Primary stiffness values

Primary suspension spring (at wheel set)	(N/m)
Vertical stiffness between wheel and bogie frame (K_{pz})--(Z direction)	1.077×10^6
Lateral stiffness between wheel and bogie frame (K_{py})--(Y direction)	23×10^6

Table 3: Secondary stiffness values

Secondary suspension spring (at centre of bogie frame)	(N/m)
Vertical stiffness between bogie frame and bolster (K_{sz})--(Z direction)	1.695×10^6
Lateral stiffness between bogie frame and bolster (K_{sy})--(Y direction)	0.4648×10^6

Table 4: Material properties of steel

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

Table 5: Primary suspension damping coefficients

Primary suspension (at the wheel set)	(N-sec/m)
Vertical damping coefficient between wheel and bogie frame (C_{pz}) (Z direction)	0.082×10^6
Lateral damping coefficient between wheel and bogie frame (C_{py}) (Y direction)	1×10^6

Table 6: Secondary suspension damping coefficients

Secondary suspension (at the center of the bogie frame)	(N-sec/m)
Vertical damping coefficient between bogie frame and bolster (C_{sz}) (Z direction)	0.118×10^6
Lateral damping coefficient between bogie frame and bolster (C_{sy}) (Y direction)	2×10^6

3. Static analysis of sleeper bogie

For static analysis wheels are considered as constrained in all the directions for both laden and unladen conditions. Static analysis is carried out for laden and unladen conditions for both sleeper and 3 tier AC bogies. The load values of sleeper and 3 tier AC bogie used in the analysis are specified in Table 7. For unladen condition, half of the dead weight of the coach is lumped at the centre of gravity of the bogie. According to Indian railways Research Division Standards Organisation (RDSO) the maximum pay load or dead weight of a sleeper coach is 52 tons including the weight of the two bogies. As centre of gravity is in a hollow portion nodal coupling is given to the centre of gravity and to the components of the bogie. For laden condition, weight of 36 passengers along with the half the dead weight of coach is applied at centre of gravity of bogie frame. Average weight of each passenger is assumed as 65 kg i.e. 638 N.

Table 7: List of nodes at which dead and live loads are lumped & load values

Node	Unladen mass (N)	Laden mass (N)
C.G for sleeper bogie	1.962×10^5	2.19168×10^5
C.G for 3 tier AC bogie	2.4525×10^5	2.68218×10^5

Applied loads and boundary conditions are represented in Fig. 4. In unladen static analysis; half dead load of coach is applied at center of gravity with wheels constrained. In laden static analysis; along with half dead load of coach, live load of 36 passengers is also applied at center of gravity of the bogie frame with wheels constrained. For both unladen and laden conditions the deflections, principal, shear and von-misses stresses are extracted. Overall deformation of the sleeper and 3 tier AC bogies under both unladen and laden conditions is shown in Figs. 5 to 8. Figs. 9 to 12 represent maximum principal stress values of the sleeper and 3 tier AC bogies under both unladen and laden conditions. Of the three shear stresses (S_{xy}, S_{yz}, S_{zx}) the maximum occurs in S_{zx} and its variation for the sleeper and 3 tier AC bogies under both unladen and laden conditions is shown in Figs. 13 to 16. The maximum value of von-Misses stress of the sleeper and 3 tier AC bogies under both unladen and laden conditions is shown in Figs. 17 to 20. In each case, it is observed that the 3 tier AC bogie under laden condition is subjected to highest deformation and worst condition of stress components. Hence the bogie has to be designed considering these parameters. The maximum von-misses stress obtained for all cases is less than yield strength 250 MPa of the bogie material.

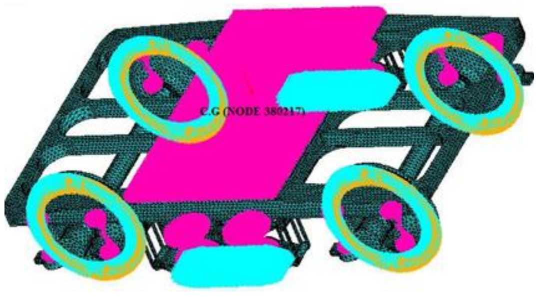

Fig. 4: Applied loads and boundary conditions

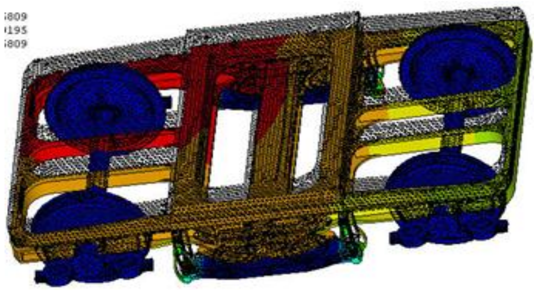

Fig. 5: Sleeper bogie overall deformation (Unladen)

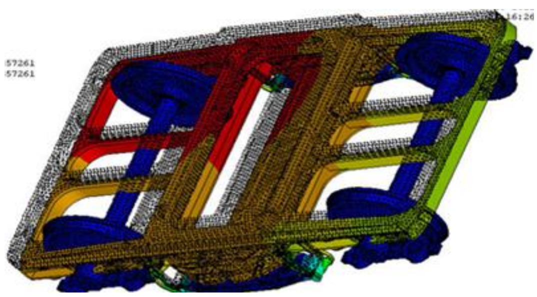

Fig. 6: 3 tier AC bogie overall deformation (Unladen)

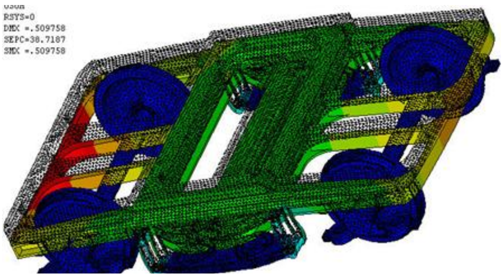

Fig. 7: Sleeper bogie overall deformation (laden)

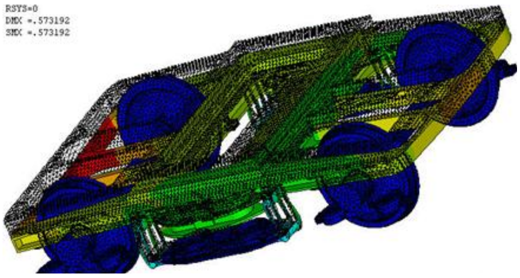

Fig. 8: 3 tier AC bogie overall deformation (laden)

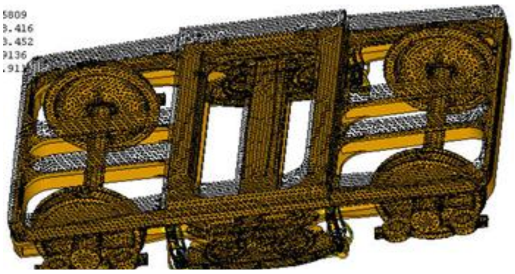

Fig. 9: Sleeper bogie principal stress (Unladen)

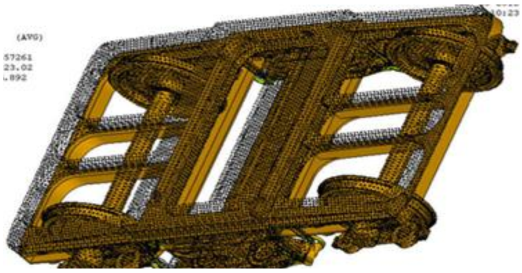

Fig. 10: 3 tier AC bogie principal stress (Unladen)

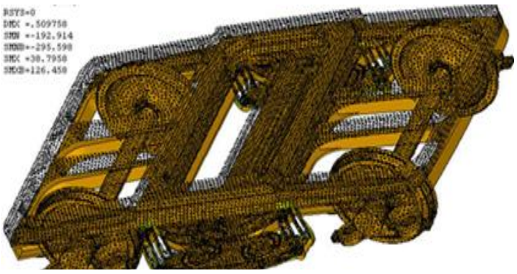

Fig. 11: Sleeper bogie principal stress (laden)

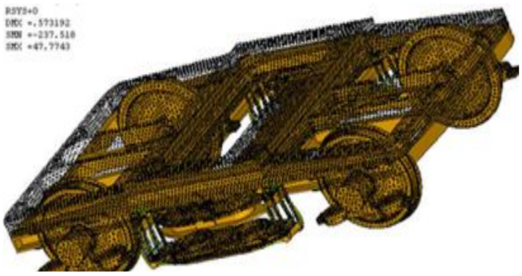

Fig. 12: 3 tier AC bogie principal stress (laden)

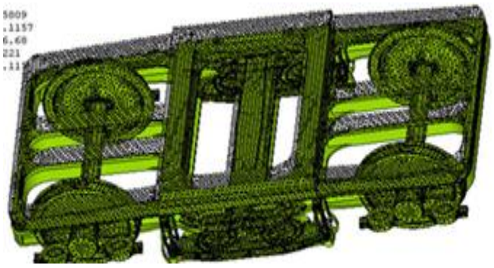

Fig. 13: Sleeper bogie shear stress distribution along XZ (Unladen)

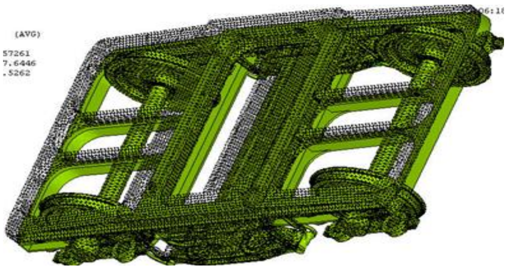

Fig. 14: 3 tier AC bogie shear stress distribution along XZ (Unladen)

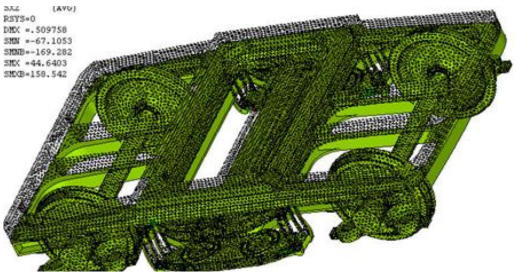

Fig. 15: Sleeper bogie shear stress distribution along XZ (laden)

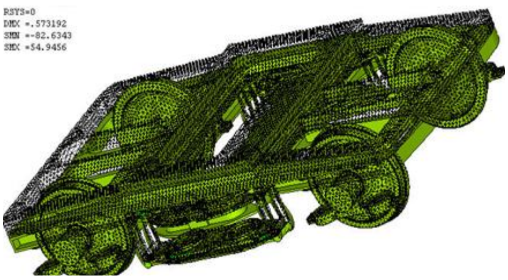

Fig. 16: 3 tier AC bogie shear stress distribution along XZ (laden)

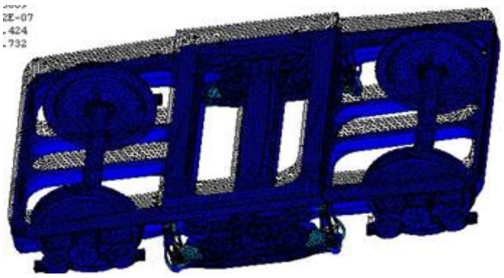

Fig. 17: Sleeper bogie von-misses stress distribution (Unladen)

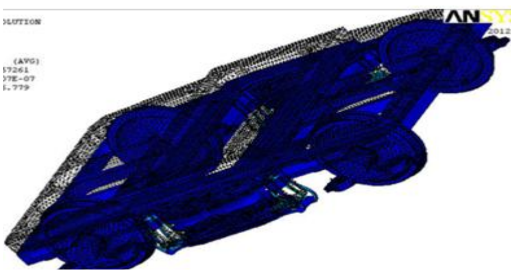

Fig. 18: 3 Tier AC Bogie von-misses stress distribution (Unladen)

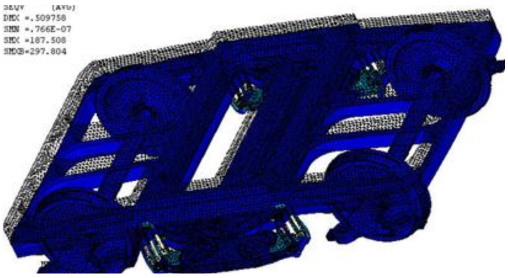

Fig. 19: Sleeper bogie von-misses stress distribution (laden)

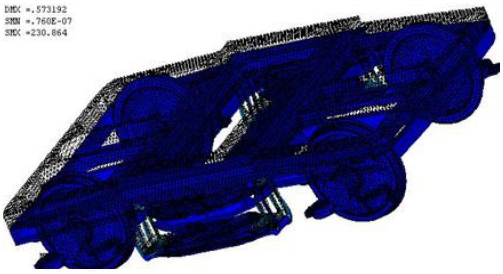

Fig. 20: 3 tier AC bogie von-misses stress distribution (laden)

Table 8: Static analysis results of key parameters for various load conditions of bogie

Parameter	Unladen sleeper	Unladen 3 tier AC	Laden sleeper	Laden 3 tier AC
Max. displacement (mm)	0.2858	0.3573	0.5097	0.5732
Principle stress (MPa)	178.42	223.02	199.30	243.91
Shear stress (MPa)	41.221	51.526	44.64	54.9456
Max. von-misses stress (MPa)	173.42	216.78	187.51	230.664

4. Modal analysis

For modal analysis contact points of the wheels are considered as constrained in all directions. 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. In Ansys, Block Lanczos method is used as this method computes natural frequencies and mode shapes for large and symmetric structures efficiently. Since the bogie design and mass is same for RCF sleeper and 3 tier AC coaches modal analysis results are same for both the sleeper and 3 tier AC bogies. The first few natural frequencies for unladen and laden condition are extracted and are tabulated as shown in Table 9.

Table 9: Natural frequencies of bogie

Mode no	Frequency for unladen condition (Hz)	Frequency for unladen condition (Hz)
1	1.0443	1.04137
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

5. Transient dynamic analysis of bogie

Assuming that the bogie passes over a bump, the variation in displacement at different key locations of the bogie i.e. at center of gravity (C.G) of bogie frame, center of gravity left corner (C.G.L) and center of gravity right corner (C.G.R) locations has been plotted against time as shown from Figs. 21 to 23. The total time taken by the bogie frame to cross over the bump is 0.144 sec. The variation in vertical displacement at C.G of bogie frame, C.G.L and C.G.R position with respective to the time as shown in Fig. 21. It shows that there is no variation in displacement for fraction of seconds i.e., up to 0.045 sec because the vehicle has not reached the obstacle which is at 100 mm away. When the vehicle starts to cross the bump the transformations of displacement from wheel bottom points to various locations of bogie frame can be observed in the time period 0.04 sec to 0.144 sec. The disturbances are shown high at C.G. portion than others while vehicle passing over the bump. The maximum displacement response attained at C.G. portion at time 0.136 sec is 0.82 mm when the front wheels reach the top of the bump and when the front wheel descending from top of the bump the response shows in decreasing order. And when rear wheels cross the bump the response is shown in downward direction with a maximum displacement of 0.8 mm at time 0.142 sec at C.G. portion.

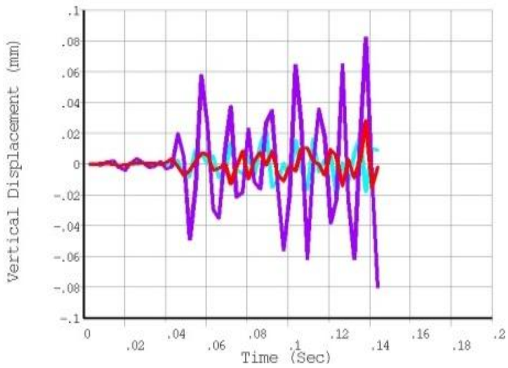

Fig. 21: Displacement vs. time for C.G, C.G.R and C.G.L locations of the bogie

Fig. 22 shows the displacement response at front left bogie bottom (F.L.B.B), front left bogie top (F.L.B.T), front right bogie top (F.R.B.T) and front right bogie bottom (F.R.B.B) positions. Minimum or negligible disturbances are observed up to time 0.045 sec as the vehicle has not reached the obstacle. It shows that increase in displacement response started at time 0.054 sec when front two wheels start to ascend the bump. It shows that disturbances are same for F.L.B.T and F.R.B.T over the entire time period and observed that maximum displacement obtained at time 0.082 sec is 0.18 mm. The response at rear left bogie top (R.L.B.T), rear left bogie bottom (R.L.B.B), rear right bogie top (R.R.B.T) and rear right bogie bottom (R.R.B.B) portions over the time period is shown in Fig. 23. It is observed that there is negligible displacement for fraction of seconds, since the vehicle has not reached the obstacle and is 100 mm away. It can be shown that the response pattern at any location of rear portion is same over the entire time period. The maximum displacement obtained at time 0.082 sec is 0.23 mm. While the rear wheels passing over the bump the response shows high in downward direction in time period of 0.088 sec to 0.144 sec. The maximum disturbances observed at these locations at time 0.142 sec is 0.19 mm.

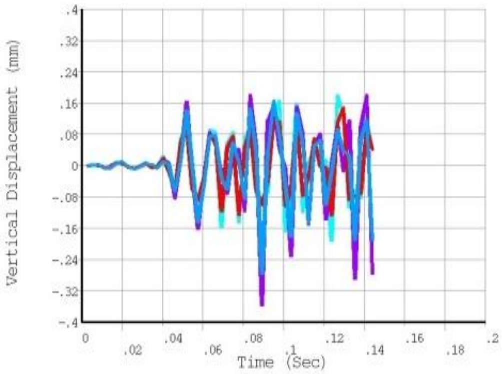

Fig. 22: Displacement vs. time for F.L.B.T, F.R.B.T, F.L.B.B and F.L.B.T locations of the bogie

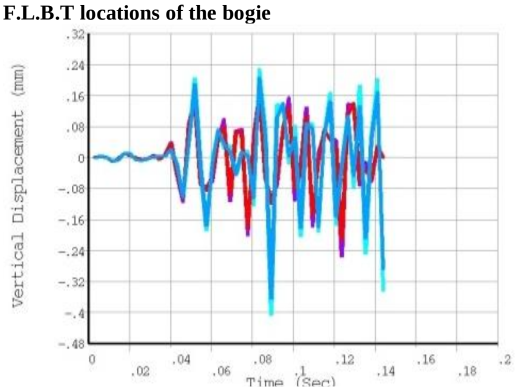

Fig. 23: Displacement vs. time for R.L.B.T, R.R.B.T, R.L.B.B and R.L.B.T locations of the bogie

6. Conclusion

Static stress analysis has been performed for laden and unladen conditions of sleeper and 3 tier AC bogies and the corresponding displacements, principal, shear and von-misses stresses have been plotted. Eigen value analysis has been performed and the first few mode shapes were extracted. Transient dynamic analysis has been performed and the displacement response of key locations of the bogie frame has been plotted with respect to time when the train is crossing a bump thereby understand the vibrational effect of rail surface irregularities. The following conclusions are drawn from the presented results:

- The location of maximum deformation for all the cases of static analysis occurred at the connecting point of bogie side frame and transom.
- Maximum deformation of 3 tier AC bogie is higher than sleeper bogie for both laden and unladen conditions because of higher mass on 3 tier AC bogie.
- Modal analysis results reveal that for both unladen and laden conditions the bogie attains similar natural frequencies with minor deviations. It means the effect of passenger load is minimal when compared to the overall pay load of the coach.
- Various mode shapes of the bogie illustrate that bounce, pitch and twist modes are predominant which influence the dynamic behaviour significantly.
- From the results it is concluded that the transient dynamic analysis is effectively used to analyse the system behaviour under most real time input excitations as the bogie passes over a semi-circular bump.
- From transient analysis it can be concluded that while train is moving at 100 kmph over the ellipsoidal bump the displacement response at bogie rear portion is higher than front portion.

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