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Procedia Materials Science 6 (2014) 1891 – 1898
Procedia Materials Science
www.elsevier.com/locate/procedia

3rd International Conference on Materials Processing and Characterisation (ICMPC 2014)

Structural dynamic analysis of freight railway wagon using finite element method

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Abstract

This paper describes the development of a virtual freight wagon vehicle using virtual prototyping computer tools. The freight wagon considered as open type wagon *"BOXN25"* of the Indian Railways. The freight wagon vehicle comprises of car body structure and two bogies. *Solidworks* is used for modeling the freight wagon and the geometry is exported to finite element tool, ANSYS. A multi degree of freedom system has been reviewed and compared with system having infinite degree of freedom (continuous structure). The current problem falls in the category of large models (Block Lanczos Algorithm is used) and has high degrees of freedom (PCG Solver is used). The structural dynamic response for the virtual freight wagon is determined. It is seen that the car body deformation is influenced by its elastic underframe and sidewalls. The influence of vibration modes, which describe local deflection, on the comfort level and stability of laden goods is discussed. Mode shapes up to a frequency of 30 Hz are considered.

Selection and peer review under responsibility of the Gokaraju Rangaraju Institute of Engineering and Technology (GRIET)
Keywords: Structural dynamics; Ride comfort; Virtual prototyping

1. Introduction

Ever since the advent of railways, the rail tracks have been guiding the trains to operate in a safe manner. Further, development ensured that the tracks played an important role in economic aspects of railways. For the passage of the trains along the tracks to be smooth, the tracks need to be perfectly aligned as well as leveled; otherwise, the track irregularities may cause vibrations and oscillations. It has been seen all along that, the major cause of discomfort for passengers has been due to induced oscillation and vibrations. These vibrations and oscillations are also a major source of damage to the laden goods.

The traditional dynamic analysis approach for railway vehicles has been based on the assumption that the car body is modeled as a rigid one. It is quite evitable that for such rigid models, the critical positions in context to ride comfort or safety of laden goods are the vehicle extremities, as at these locations the vertical motion of the whole body tends to sum up with the pitching motion of the car body. The results based on above assumptions of rigid body often leads one to believe that vibration issues at the vehicle extremities are more significant and should be addressed with priority. But the possibility of the magnitudes of the acceleration levels at the center of vehicle floor being comparable with those at vehicle extremes cannot be neglected. With an ever increasing demand for logistic support provided by railways, efforts to improve speed in order to facilitate reduction in time for transport operation has received considerable attention. The past two centuries have seen increase in the travel speed from

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2211-8128
Selection and peer review under responsibility of the Gokaraju Rangaraju Institute of Engineering and Technology (GRIET)
doi:10.1016/j.mspro.2014.07.221

about 50 km per day by horseback to about 650 km per day by car or 200-650 km per hour by using high-speed rail (HSR), air passenger transport (APT) and also possibly the TRANSRAPID MAGLEV.

With reference to the performance of railways in comparison to any other transportation means, especially, air transport, railways have scored better with consideration to parameters like direct burdens/emissions, such as energy consumption, air pollution, noise, land take and land use, safety and congestion. Janic (2003) has in his work showed how high speed rail contributed towards mitigation of a cumulative environmental damage.

With the probability of higher speed of railways in context with increasing demand for logistics support, the comfort of passengers and the safety of laden goods, assumes great significance. A large number of excitation sources exist that may induce oscillations, vibrations, and noise in the train and in the track and its surroundings. But the effect of vibrations on the car body due to rail profile on account of increased speed has marked significance. Long wavelength geometric irregularities in the track alignment induce lateral displacements of the railway cars, and this will induce travelling discomfort for the passengers. Short wavelength irregularities will induce vibrations and noise. The same can be said for long and short wavelength irregularities of the track level/vertical profile (Dahlberg, 2006). Several studies have been performed in the past few years with regards to rail vehicle dynamics and rail vehicle system. These studies have helped to disintegrate the problem of vehicle dynamics for rail vehicle engineering into different categories which are dependent on the activities that these analyses support. The need to study Eigen behavior before the commencement of any dynamic calculations has been emphasized by Polach et al. (2006).

A study of the vehicle structural dynamics in connection with vehicle running dynamics is thus a necessity. With a consistent demand in the market, the vehicle development is driven by need for innovative designs and lower costs. Virtual prototyping has shown great potential for improving the product in the development process. Work using virtual prototyping computer tools has been done by Stribersky et al. (2002). A virtual vehicle was developed and used for prediction of the structural dynamics. The work was based on modular design concept and vehicle components for a metro train were modeled and stored as substructures in a specific vehicle component database. This database helped for assembling three-car train very quickly to perform structural dynamic analyses to predict the ride comfort.

Work on evaluation of vibration modes of passenger vehicle was presented by Diana et al. (2002). The author has stressed on the need of appropriate modeling of dynamic behavior of the vehicle components, particularly the flexibility of car body in the analysis of railway vehicle comfort performance. Popprath et al. (2006) investigated structural dynamic behavior of 1/10-scaled model of a metro vehicle car body using finite element method and experimental modal analysis.

In general, a computer model of a railway vehicle can be constructed and run in a virtual environment, and a range of possible designs or parameter changes can be investigated. Also the outputs from the model can be set up to provide accurate predictions of the dynamic behaviour of the vehicle and its interaction with the track. The inputs are normally provided at each wheelset of the model. The typical inputs are deviations in gauge and cross level and vertical and lateral track irregularities. A generalization of the simulation process was presented by Polach et al. (2006) (Fig. 1).

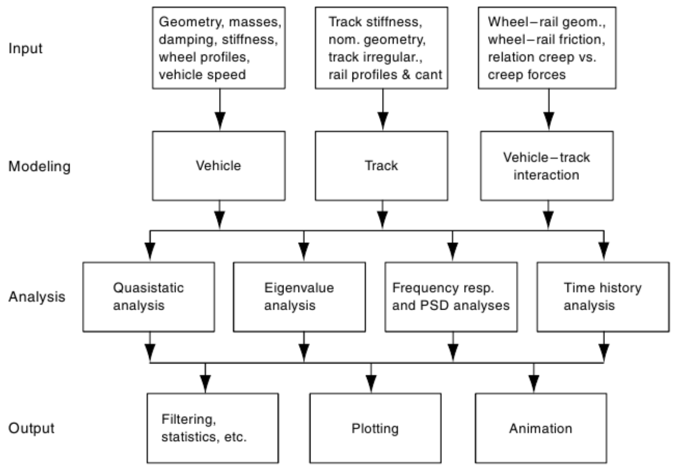

Fig. 1. Process of computer simulation of the railway system (Polach et al. 2006)

This paper describes the dynamic analysis of freight vehicle, which is modeled using *Solidworks* and its structural dynamic response determined using analysis tools (ANSYS). The dynamic effect of the response on the stability of car body and consequently the comfort of passenger and safety of laden goods is discussed.

2. Vehicle model description

The freight wagon vehicle comprises of car body structure and two bogies. The freight wagon considered in our case is open type wagon "BOXN25" of the Indian Railways. The bogie for this wagon is of conventional type and consists mainly of two wheelset, two side frames, and a bolster. No steering connection exist between the front and rear wheelset of each bogie frame. Each of the four wheelsets is identical and free to move independently.

The coupling between the first two wheelset is only through the front bogie frame, and the coupling between the two rear wheelsets is only through the rear bogie frame. The front and the rear bogie frame are coupled through primary suspension elements to the front and rear bolster respectively and the latter through the secondary suspension elements to the car body. The car body rests on the bolster centreplate and is supported at the centreplate and side bearings.

The freight wagon vehicle has been modeled virtually using *Solidworks* (Fig. 2). This virtual model consists of substructures which are interfaced with each other to form a complete vehicle. The various substructures include the carbody, the front bogie and the rear bogie. All these substructures have been modeled as elastic bodies keeping in view the structural dynamics. All the interfaces between various substructures are defined separately. Since the wheelset are always in contact with the rail they are not included in the structural dynamics.

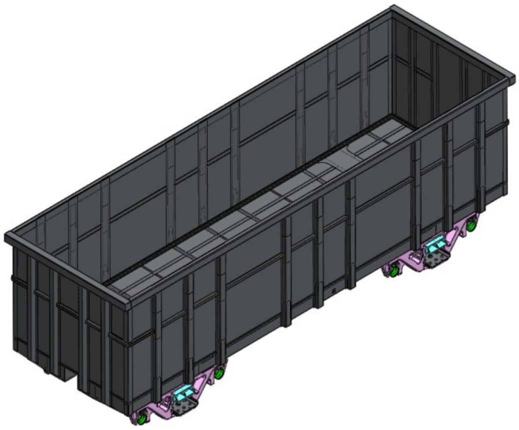

Fig. 2. Freight railway wagon of Indian Railways (Open wagon type - *BOXN25*)

The material defined for the model is IRS M-44 (Indian rail steel), having carbon content 0.03%. The mechanical properties like yield strength is in the range of 350-450 MPa, whereas the tensile strength is around 500 MPa. The percentage elongation is 25.

3. Vehicle vibration modes

3.1. Need for modal analysis

Frequency analysis comprises of two major analyses in frequency domain; they are eigenvalue and spectral density analysis. Hence, to determine the characteristics of all the bodies and elements that are used in track and vehicle, viz, mass, stiffness and damping coefficient, eigenvalue analysis is more appropriate. Thus, natural frequencies can be determined from this type of analysis and one can investigate the stability of the vehicle by studying the natural frequencies obtained. Also, eigenvalue analysis helps in detecting the eigenmodes. Therefore, the user can find the sensitive resonance frequencies of the vehicle elements and investigate, for example, the effect of track irregularities (reference) on the carbody vibration (response). Since eigenvalue analysis can be done on a linearized system all the non-linear relations in the dynamic behavior of the system have to be simplified by linearization.

3.2. Modal analysis for a continuous structure

Using D'Alemberts principle and discretization of the process of a continuous system with finite elements the following equation of motion can be derived:

$$m\ddot{} \qquad \dot{} \qquad f(t) \tag{1}$$

In the above equation *m*, *c* and *k* denote the structural mass, damping and stiffness matrices. The vectors of nodal accelerations, velocities and displacements are $\ddot{x}$, $\dot{x}$ and *x* respectively. $f(t)$ is the vector of applied forces. Dynamical equilibrium is obtained if Eq. (1) holds good for all times '*t*'. In fact, all problems in dynamics can be formulated on the basis of above equation of motion, Eq. (1).

Vibration characteristics of any system can be determined with modal analysis; especially for structure, it can be determined during the design process. Thus, modal analysis determines natural frequencies and mode shapes of a continuous structure. Considering Eq. (1), the right hand side is assumed to be zero, i.e. $f(t)=0$, and thus modal analysis forms a basis for other more detailed dynamic analyses such as transient or harmonic or even spectrum analysis based on the modal superposition technique. The modal analysis is always a linear analysis. The user has thus to ignore nonlinearity or the software itself will ignore any user specified non-linearity. However, effects of pre-stress can be considered for any given system.

A review of multi degree of freedom system (Fig. 3) is significant in the sense that it will help us to compare a limited degree of freedom system with that having infinite degree of freedom (continuous structure).

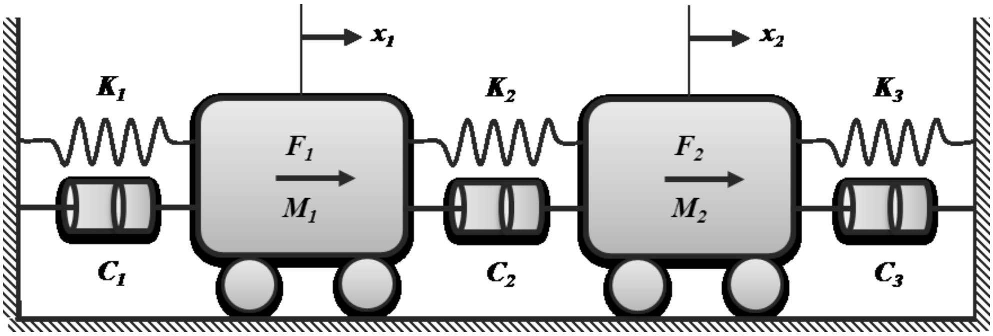

Fig. 3. System with two degree of freedom

The finite element method approximates the real structure with a finite number of degrees of freedom '*N*' and mode shapes can be found for a finite element method having '*N*' degrees of freedom. Modal Analysis is thus a process for determining the '*N*' natural frequencies and mode shapes. Given "suitable" initial conditions, the structure will vibrate at one of its natural frequencies and the shape of the vibration will be a scalar multiple of a mode shape. Given "arbitrary" initial conditions, the resulting vibration will be a superposition of mode shapes. The equation of motion for the system (Fig. 2) is same as that obtained in Eq. (1).

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=\{F\} \tag{2}$$

In the above equation, [*K*], [*C*] and [*M*] are constant. Thus, for a free vibration analysis, the natural frequencies (ω_i) and the mode shapes (ϕ_i) will be calculated by following relation:

$$\left([K]-\omega_i^2[M]\right)\{\phi_i\}=0 \tag{3}$$

Following assumptions have been made while evaluating mode shapes for freight wagon:

- The material is assumed to behave linearly elastic,
- No nonlinearities are included,
- Small deflection theory is used,
- [*C*] is not present, so damping is not included,
- {*F*} is not present, so no excitation of the structure is assumed,
- The structure can be constrained or unconstrained,
- The mode shapes {ϕ} are relative values, and not absolute values, and
- The unknown nodal displacements vary with time.

3.3. Solution algorithms for a modal analysis

By default the Subspace Method uses the Frontal Solver to obtain the first natural frequencies of a structure. This solver works efficiently for small models of up to 50,000 active degrees of freedom. However, if models consist mainly of solid elements with more than 50,000 active degrees of freedom, the Subspace Method combined with the PCG-Solver should be the preferred solution method. In ANSYS, the combination of the Subspace Method together with the PCG-Solver is called Power-dynamics Method. For large models of up to 10,000,000 degrees of freedom, this method significantly reduces solution time.

Following is the syntax extract from the solution information for our problem:

```
***** ANSYS SOLUTION ROUTINE *****
PERFORM A MODAL ANALYSIS
THIS WILL BE A NEW ANALYSIS
USE PCG LANCZOS MODE EXTRACTION METHOD
EXTRACT 50 MODES
NORMALIZE THE MODE SHAPES TO THE MASS MATRIX
```

From the syntax it is evident that the problem falls in the category of large models (as Lanczos method is used) with high degrees of freedom (as PCG Solver is used).

However, to be more precise about the model we have the following data:

- Total Mass of a single wagon assembly : 16080.98 kilograms;
- Total Volume : 2.51 cubic meters; and
- Surface area : 511.16 square meters
- Total number of elements : 296779
- Total number of nodes : 569003

3.4. Finite element model details

The model which was prepared external to ANSYS was imported. Engineering Data Manager was used for defining, storing, and organizing material properties. The contact regions were identified at: interface between carbody and centre pivot top, interface between centre pivot top and bottom, interface between centre pivot bottom and bolster, bolster and sideframe. A mesh with quality metrics was generated. Of the many metrics orthogonal quality, aspect ratio, and skewness were given special attention. Statistics of the orthogonal quality for the freight wagon is shown in Fig. 4. A major (approximately 70%) of the total elements lie in the 'very good' zone followed by remaining (approximately 27 %) in the 'good' zone and a meager amount (approximately 3%) in the acceptable zone. A very negligible amount of elements fall in the 'bad zone', i.e. from 0.01 – 0.001. Since, there are no elements in the 'unacceptable zone' of 0.001 – 0.00 as the minimum value obtained is 0.0117, thus the mesh quality has passed the orthogonality test.

Fig. 4. Orthogonal quality metrics on percentage of volume / area basis

The Skewness quality of the discretized elements is expressed in terms of the percentage of volume /area of the total elements (Fig. 5). The Skewness quality of nearly 50% of the elements is 'excellent' whereas nearly 28% of the elements have 'very good' skewness quality, followed by nearly 12% with a 'good' skewness quality tag. Nearly 9% are tagged as ones with 'acceptable' quality level. A mere 1% fall in the category of 'bad', there being extremely negligible elements with 'unacceptable' level '1'. Thus the Skewness quality of the elements is to be hailed.

Fig. 5. Skewness quality metrics on percentage of volume / area basis

The 'Aspect Ratio' which measure how much stretched the elements are, is also quite significant. The aspect ratio tells us about how much elongated or skinny an element is. For the three dimensional elements we have used, aspect ratio is the ratio of the lengths of the largest and the smallest edge. The minimum value achieved is 1.0454 while the maximum being 745.77. The average value is 2.5439. The acceptance criteria call for aspect ratio < 100. Since, the average value of aspect ratio is quite below the threshold, it indicate that nearly all the element have a good geometry.

4. Simulation results and discussions

The finite element model and the obtained eigenmodes represent the freight wagon which is not coupled to its actual environment. The car body deformation is influenced by its elastic underframe and sidewalls. The modes which describe local deflections have been considered, as they greatly influence the comfort level and stability of laden goods. Mode shapes were considered up to a frequency of 30 Hz. Thus with an upper frequency limit of 30 Hz, 8 different modes of the structure are identified and described in Table 1.

Up to a frequency of 20 Hz the model indicates five different mode shapes with dominant elastic deformations of the car body structure. The lowest frequency is 6.94 Hz which represents the first mode shape. The motion can be described as lateral swaying of side walls and this lateral swaying is also sometimes referred to as diagonal distortion since it causes change in the diagonal length of the carbody cross-section. Under excitation frequencies equivalent to the first modal frequency the objects located in the maximum displacement zone of side walls will sway and may be for continuous excitation this swaying motion can get converted in rolling action, thereby affecting stability in the lateral direction.

Table 1. Mode shapes and frequencies for freight wagon vehicle

Mode Number	Frequency (Hz)	Description of mode shapes
1	06.94	Lateral swaying of side walls. Fig. 6.(a)
2	07.03	Shell breathing in lateral direction. Fig. 6.(b)
3	12.07	Torsion and lateral swaying of side walls. Fig. 6.(c)
4	16.01	Lateral shell breathing with front & rear walls swaying. Fig. 6.(d)
5	17.64	Rolling and lateral swaying at central length. Fig. 6.(e)
6	23.21	Torsion and longitudinal diagonal distortion. Fig. 6.(f)
7	25.79	Shell breathing in lateral & longitudinal direction. Fig. 6.(g)
8	29.73	Rolling and lateral swaying at central and longitudinal ends. Fig. 6.(h)

The next modal frequency is 7.03 Hz which is fractionally greater than the previous one, but the behaviour is changed drastically from lateral swaying of side walls to shell breathing in the lateral direction. For this mode shape we can say that only the side walls will vibrate out of phase and therefore those objects which are in the region of maximum displacement at each side wall will sway under the influence of the excitation frequencies which are equivalent to value of second mode shape.

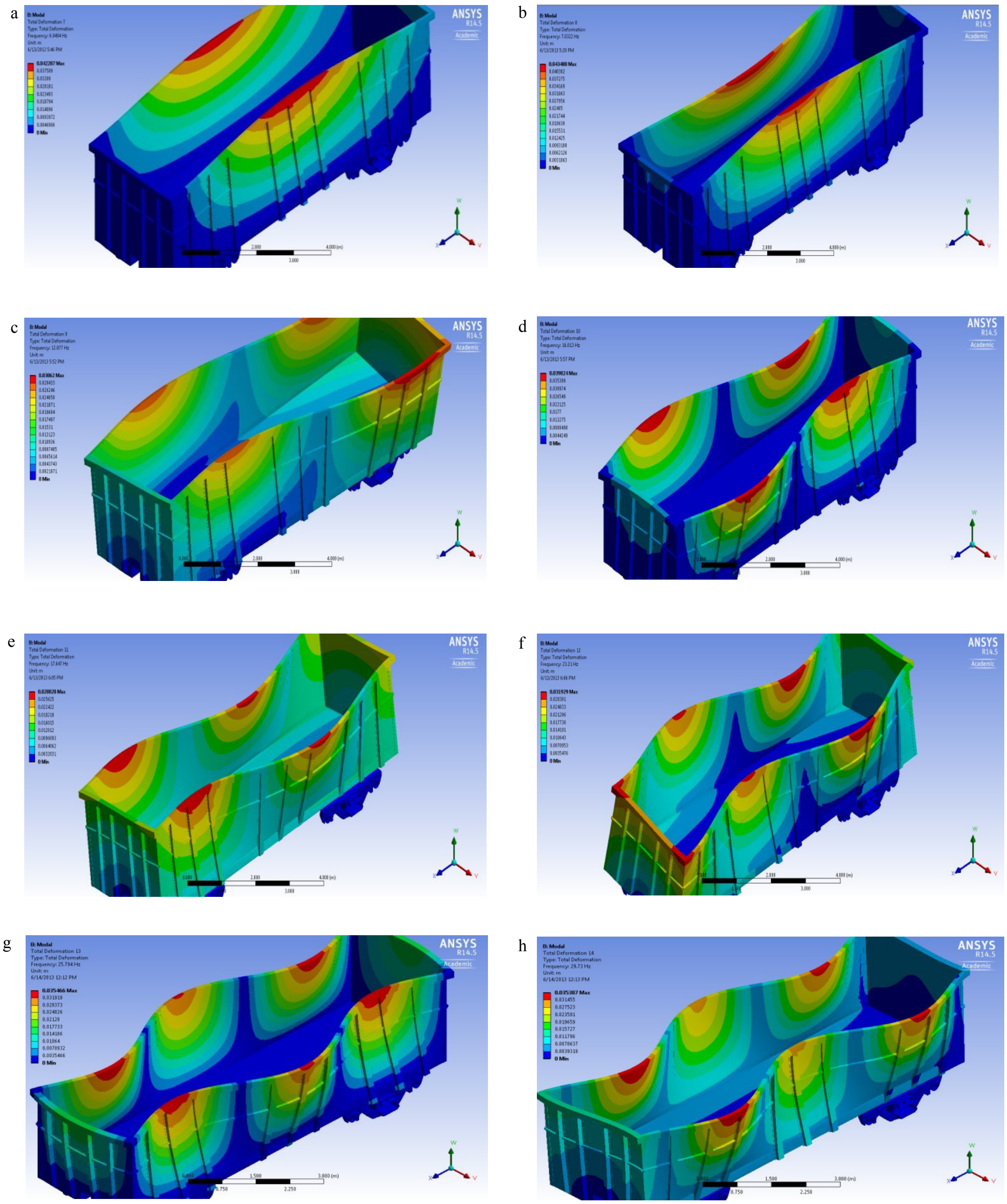

Fig. 6. Finite element analysis. (a) first flexural vibration mode, 6.94 Hz, (b) second flexural vibration modes, 7.03 Hz, (c) third flexural vibration modes, 12.07 Hz, (d) fourth flexural vibration modes, 16.01 Hz, (e) fifth flexural vibration mode, 17.64 Hz, (b) sixth flexural vibration modes, 23.21 Hz, (c) seventh flexural vibration modes, 25.79 Hz, (d) eight flexural vibration modes, 29.73 Hz.

The third mode shape occurs at frequency of 12.07 Hz and the carbody is subjected to torsion in longitudinal direction coupled with lateral swaying of side walls. In addition to swaying motion, the torsional behavior indicates that the entire length of the car body is influenced at this frequency, either by swaying or twisting motion in lateral direction. Thus those objects located at either ends of the carbody will be influenced with torsion / twisting motion in addition to the swaying motion because third modal frequency. Hence it should be noted that the objects within the vehicle at all locations are sensitive to lateral motion at this modal frequency. Therefore any excitation frequency in this range needs to be taken seriously.

The fourth mode shape occurs at a frequency of 16.01 Hz. The motion of the carbody is a kind of shell breathing but with side walls at longitudinal ends having opposed motion. Thus the objects at either ends are sensitive to lateral as well as longitudinal swaying motion and continuous excitation at this frequency can cause individual objects to roll in either direction. The fifth modal frequency is 17.64 Hz and has significant rolling motion with lateral swaying at the central longitudinal length. Therefore continuous exciting frequencies at this value can cause significant rolling motion of the entire carbody.

The sixth modal frequency occurs at 23.21 Hz and the carbody is twisted along with the longitudinal diagonal getting distorted. This indicates that the end walls of the car body are being influenced. This mode thus indicates the onset of vibrational motion in the longitudinal direction along with the lateral direction. The seventh modal frequency is 25.79 Hz and the mode shape can be described as shell breathing in lateral as well as longitudinal direction. This mode indicates the increase in the longitudinal motion along with the lateral motion. The final mode shape being considered is having frequency of 29.73 Hz. Here we can experience rolling of the car body with swaying at center and longitudinal ends opposed to rolling.

These results are further compared with the simulation results of Stribersky et al. (2002) and S. Popprath et al. (2006) in Table 2.

Table 2. Comparison of modal frequencies in (Hz) with earlier simulation results

Mode Number	Current Work	Stribersky et al. (2006)	Popprath et al. (2006)
1	06.94	10.30	15.20
2	07.03	10.90	17.70
3	12.07	12.40	19.50
4	16.01	14.50	20.30
5	17.64	15.30	--
6	23.21	16.60	--
7	25.79	18.20	--
8	29.73	--	--

5. Conclusions

The structural dynamic behavior of the freight railway vehicle has been carried out using finite element tool. The structural behavior of the freight wagon indicates that if there are external excitation frequencies because of irregularities in the vertical profile of the track and if those frequencies coincide with the modal frequencies of the system, then the instability of the carbody will increase. All the modal frequencies upto the upper limit of 30 Hz are indicating displacement in the lateral direction, thereby signifying the importance of rolling motion for the freight wagon. Comparison of the values of the obtained modal frequencies with the earlier published simulation results for metro (passenger) trains indicate that laden goods can be subjected to low frequency oscillations as compared to passenger trains. Hence, laden goods in freight trains can be sensitive to even low excitation frequencies due to track irregularities. The track irregularities on the vertical profile for a single track only are most crucial and can cause overturning of laden goods in the freight vehicles. Thus further work regarding numerical modeling of the dynamic behavior of the freight wagon should be concentrated around irregularities on the vertical profile of track.

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