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Investigation of vertical dynamic behaviour and modelling of a typical indian rail road vehicle through bond graph

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Abstract. Indian Rail transport is one of the major mode of transportation, so it must offer high comfort level for the passengers and the staff. However, the comfort that passengers experience is a highly complex and individual phenomenon. The improvement of passenger comfort while travelling has been the subject of intense interest for many train manufacturers, researchers and companies around the world. Although new techniques in manufacturing and design ensure better ride quality in railway carriages, still it is impossible to completely eliminate track defects or various ground irregularities. The dynamic behaviour of a train also depends on the load and the mechanical systems, such as springs, dampers, etc., which interact with the wheels, the train body and bogies.

The present study deals with the effects of railway track imperfections on dynamic behaviour, and investigates the effect of vehicle speed and the rail irregularity on ride comfort through numerical simulation. The numerical simulation of the vertical dynamic behaviour of a typical railroad vehicle will be done by modelling through bond graph technique. The model consists of 17 degree of freedom with wheel set, bogie and car. For assessment of ride comfort Sperling ride index used by Indian Railway is calculated using filtered RMS accelerations. The ride characteristics of the vehicle provide assessment of the dynamic behaviour of the vehicle through analysis of accelerations at the vehicle body, whereas ride comfort assesses the influence of vehicle dynamic behaviour on the human body. The bond graph modelling and its simulation, is performed using SYMBOLS SHAKTI software[1].

Keywords: sperling index, ride quality, bond graph modelling, dynamic behaviour, ride comfort

1 Introduction

Indian Rail transport is one of the major mode of transportation, so it must offer high comfort level for the passengers and the staff. However, the comfort that passengers experience is a highly complex and individual phenomenon. In several researches, noise and vibration have been identified as the most important factors for high comfort. The main sources of vibration in a train are track defects in welding or rolling defects, rail joints, etc. The nature of vibration itself is random and covers a wide frequency range[10]. The vibration accelerations consist of six components: three translational components along the $x-$, y and $z-$axes and three rotational accelerations round the axes x, y and z, respectively.

The improvement of passenger comfort while travelling has been the subject of intense interest for many train manufacturers, researchers and companies all over the world. Although new techniques in manufacturing and design ensure better ride quality in railway carriages, it is sometimes impossible to completely eliminate track defects or various ground irregularities. The dynamic behavior of a train also depends on the load and the mechanical systems, such as springs, dampers, etc., which interact with the wheels, the train body and bogies.

The dynamic performance of a rail road vehicle as related to safety and comfort is evaluated in terms of specific performance indices. The quantitative measurement of the ride quality is one of such performance

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indices. Ride quality is interpreted as the capability of the rail road suspension to maintain the motion within the range of human comfort and Sperling's ride index which is a measure of the ride quality and ride comfort used by Indian Railways[7].

Due to the unique dynamics that exist between the rail and wheel, rail vehicle dynamics are often difficult to model accurately. This velocity-dependent dynamics justify the importance of the track input to railcar modeling. In the physical system, the input comes from the actual track. In a model, a user-defined input is used to predict the actual track characteristics. The user-defined input can be created analytically or can be based on actual measurements. Measured track data are obtained by running a specialized railcar down the track. Analytic track data are created using mathematical shapes, such as cusps, bends, sines, to represent the track geometry[3].

There have been several studies, which are contributed by many researchers regarding the dynamic analysis and to enhance the ride comfort while travelling. Kumar and Sujata[7] presented the numerical simulation of the vertical dynamic behavior of a railway vehicle and calculated Sperling ride index for comfort evaluation. They modeled a typical Indian rail road vehicle running on broad gauge for the analysis. Kumaran et al.[8] discusses the dynamic response of a typical pre-stressed concrete rail track-sleeper due to wheel-track interaction dynamics. Nielsen and Igeland[9] investigated the vertical dynamic behaviour for a railway bogie moving on a rail which is discretely supported by sleepers resting on an elastic foundation. Effects of imperfections on the running surfaces of wheel and rail were studied by assigning irregularity functions to these surfaces. Bureika and Subacius[2] investigated the dynamics of vertical interaction between a moving rigid wheel and a flexible railway track. Gangadharan et al.[5] studied the influence of different track irregularities on dynamic response and coupling between vertical and lateral dynamics.

2 Model for ride comfort

Track irregularities, vehicle characteristics and vehicle speed generate motion quantities that are perceived by passengers. The combinations of these quantities affect the passengers' perception of ride comfort as shown in Fig. 1.

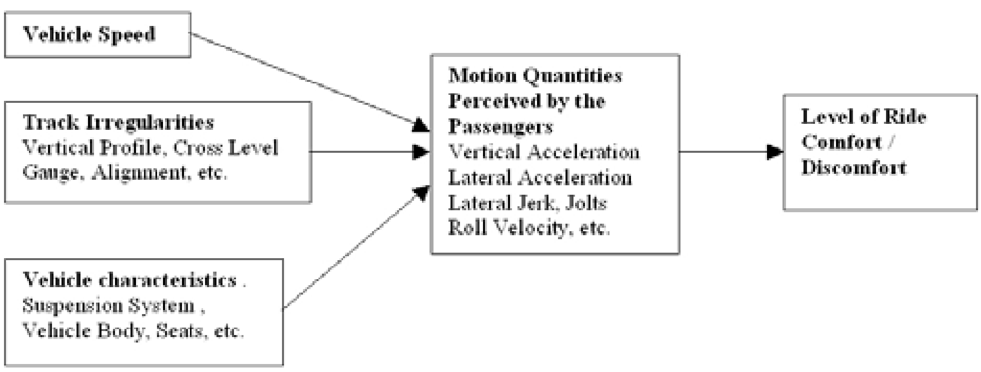

Fig. 1. Significant track and vehicle parameters for ride comfort

3 Modeling of rail road vehicle

To analyze the dynamic behavior of railway vehicles, usually the vehicle (and if necessary the environment) is represented as a multibody system. A multibody system consists of rigid bodies, interconnected via mass less force elements and joints. Due to the relative motion of the system's bodies, the force elements generate applied forces and torques. Typical examples of such force elements are springs, dampers, and actuators combined in primary and secondary suspensions of railway vehicles.

3.1 Assumptions

The assumptions made in formulating the model are as follows:

- Bogie and car body component masses are rigid.
- The springs and dampers of the suspension system elements have linear characteristics.
- Friction does not exist between axle and bearing.
- The vehicle is moving with constant velocity on rigid and constant gauge.
- All wheel profiles are identical from left to right on a given axle and from axle to axle and all wheel remains in contact with the rails.
- Straight track was assumed.
- An irregularity in vertical direction with same shape for left and right rail is considered in this model.

3.2 Rail road vehicle model

Fig. 2 illustrates the train vehicle model adopted in this study conforming to Indian railways. It consists of a vehicle body, two bogie frames and four wheel sets. Each bogie consists of the bogie frame, the bolster, and two wheel sets. The car body is modeled as a rigid body having a mass M_c; and having moment of inertia J_{bx} and J_{cy} about the transverse and longitudinal centroidal horizontal axes, respectively. Similarly, each bogie frame is considered as a rigid body with a mass $M_b(M_{b1}\text{and}M_{b2})$ with moment of inertia J_{bx} and J_{by} about the transverse and longitudinal centroidal horizontal axes, respectively. Each axle along with the wheel set has a mass M_w (for four axles $M_{w1}; M_{w2}; M_{w3}$ amd M_{w4}). The spring and the shock absorber in the primary suspension for each axle are characterized by spring stiffness K_p and damping coefficient C_p respectively. Likewise, the secondary suspension is characterized by spring stiffness K_s and damping coefficient C_s respectively. As the vehicle car body is assumed to be rigid, its motion may be described by the vertical displacement (bounce or Z_c) and rotations about the transverse horizontal axis (pitch or θ_c) and about the longitudinal horizontal axis (roll or Φ_c). Similarly, the movements of the two bogie units are described by three degrees of freedom Z_b; θ_b and Φ_b, each about their centriods. Each axle set is described by two degrees of freedom Z_w; and Φ_w. about their centriods. Totally, 17 degrees of freedom have been considered in this study for the vehicle model shown in Fig. 2. The detailed parameter regarding the moment of inertia and mass of different component is given in Tab. 1.

Table 1. Detailed parameter of rigid bodies

S.No	Name of Rigid Bodies	Mass (Kg)	Moment of Inertia(Kg-m^2)	
			I_{XX}	I_{ZZ}
1	Car Body	33700	5.24×10^4	7.36×10^5
2	Bogie-I	3150	2.02×10^3	3.56×10^3
3	Bogie-II	3150	2.02×10^3	3.56×10^3
4	Wheelset-I	1500	7.13×10^2	-
5	Wheelset-II	1500	7.13×10^2	-
6	Wheelset-III	1500	7.13×10^2	-
7	Wheelset-IV	1500	7.13×10^2	-

3.3 Vehicle parameters

Some parameters regarding the rigid bodies are already given in Tab. 1; however, the other parameters, which are essential for the simulation of the vehicle are presented in Tab. 2.

3.4 Bond graph model of a rail road vehicle

A typical rail road vehicle system is composed of various components such as car body, springs, dampers, Bogies, Wheel-set, and so forth. When such dynamic systems are put together from these components, one

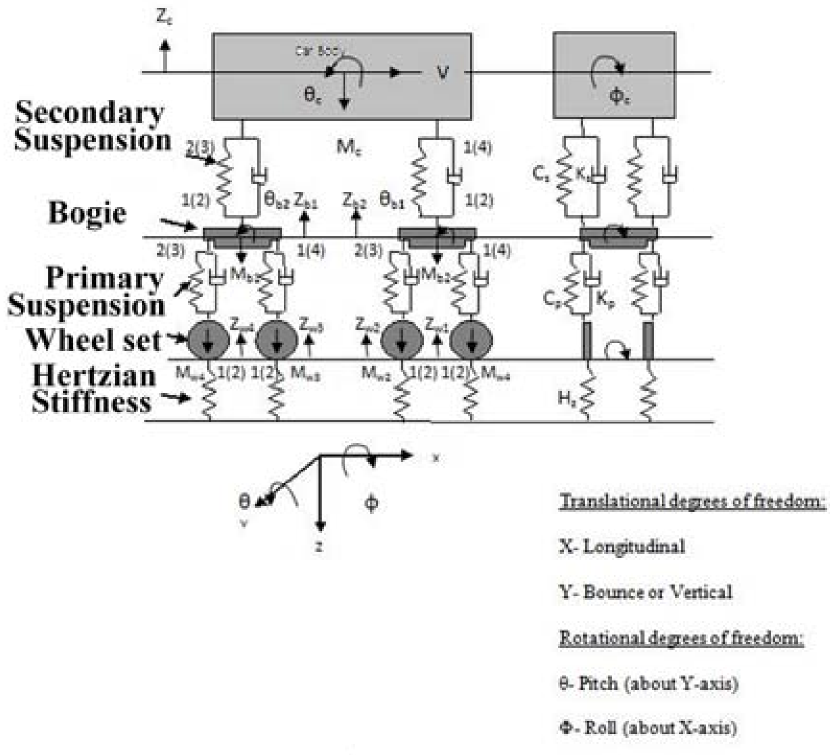

Fig. 2. Physical model of railway vehicle

Table 2. Vehicle parameters

S.No.	Parameter	Nomenclature	Values
1	Primary spring stiffness	K_p	$0.7 \times 106 N/m$
2	Secondary spring stiffness	K_s	$0.41 \times 106 N/m$
3	Primary damping coefficient	C_p	$5.88 \times 103 Ns/m$
4	Secondary damping coefficient	C_s	$22000 Ns/m$
5	Half of primary suspension spacing	b	$1.448m$
6	Half of distance between bogie centers	c	$7.315m$
7	Half of secondary suspension lateral spacing	d	$0.794m$
8	Semi gauge length	e	$0.864m$
9	Half of primary suspension lateral spacing	f	$1.127m$

must interconnect rotating and translating inertial elements with axial and rotational springs and dampers, and also appropriately account for the kinematics of the system structure. Bond graphs are well suited for this task. In deriving the bond graph models, the velocities in upward direction are assumed to be positive and springs and dampers are assumed positive in compression.

The bond graph model for the rail vehicle is made by enforcing the kinematic constraints correctly. Kinematic constraints are always related with the flow variables[6]. To add the flow according to the constraints, 0-junctions are used. The transformers are used to convert the angular velocity into linear velocity components, which is ωc and ωd at c and d points respectively. The complete bond graph model for rail vehicle system is shown in Fig. 3.

4 Track inputs to rail road vehicle

The dynamic wheel loads generated by a moving train are mainly due to various wheel/track imperfections. These imperfections are considered as the primary source of dynamic track input to the railroad vehicles. Normally, the imperfections that exist in the rail-track structure are associated with the vertical track profile,

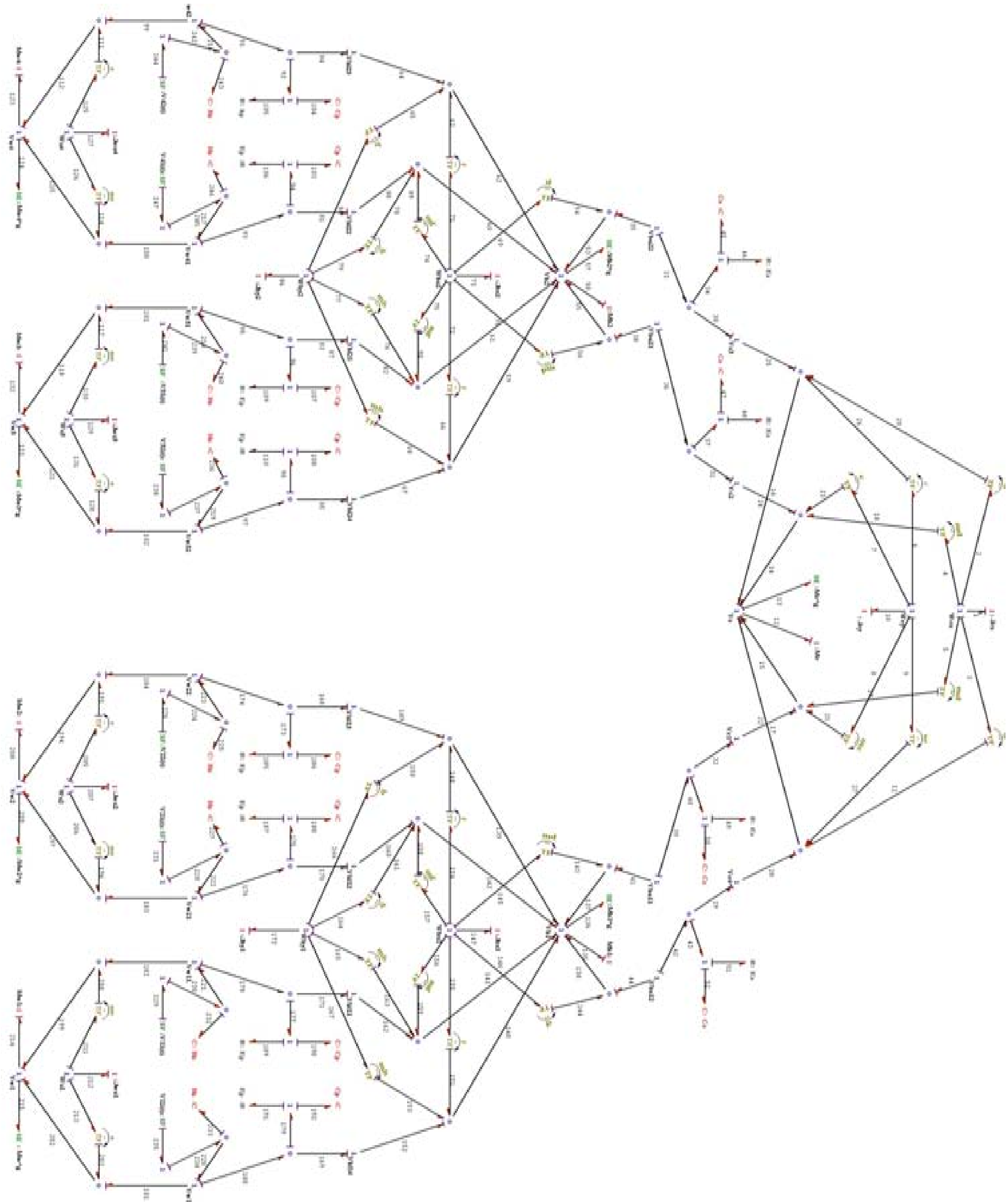

Fig. 3. Bond graph model

cross level, rail joint, wheel flatness, wheel/rail surface corrugations and sometimes uneven support of the sleepers.

In actual practice different type of periodic, aperiodic or random track irregularities may exist on the track, but in the present study bump type of irregularity is considered as shown in Fig. 4[3]. The shape of irregularity is assumed to be similar on left and right rail.

The bump excitation of the front wheel of leading bogie is

$$y = H \times \sin\left(\pi \times \frac{V}{L} \times t\right) \quad \text{for} \quad 0 \leq t \leq \frac{L}{V} \tag{1}$$

$$= 0 \quad \text{for} \quad t > \frac{L}{V} \tag{2}$$

The bump excitation of the rear wheel of leading bogie is

$$y = H \times \sin\left(\pi \times \frac{V}{L} \times \left(t - \frac{A_2}{V}\right)\right) \quad \text{for} \quad \frac{A_2}{V} \leq t \leq \frac{A_2 + L}{V} \tag{3}$$

$$= 0 \quad \text{for} \quad t > \frac{A_2 + L}{V} \tag{4}$$

where, A_1 is the distance between the front and the rear axle.

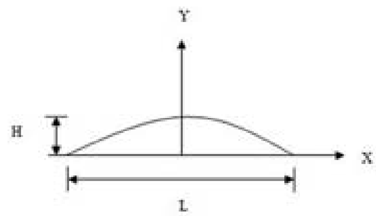

Fig. 4. Model of track irregularity

Thus the expression for flow variables are obtained through differentiating Eqs. (1) $\sim$ (4) w.r.t. time. In the present study, H is taken as $0.03m$ and L is taken as $1m$.

5 Simulation study

The Simulator of Symbols Shakti, which is the base post-processing module of SYMBOLS Shakti, is used for the simulation of the rail vehicle. Simulations were done under two conditions.

5.1 Preload analysis

Preliminary simulations were performed to find the static equilibrium of the vehicle and for checking the bond graph model of the vehicle. This was usually done to confirm since the vehicle is a symmetric vehicle, the various loads and deflections observed were equal. It was also essential to check whether the vehicle is correctly balanced or not.

5.2 Dynamic analysis

Dynamic analysis was carried out for vehicle at different speed of $15m/s$, $30m/s$, $45m/s$ and $60m/s$. The following output parameters are evaluated:

- Vertical acceleration at the floor of the car-body centre of mass.
- Vertical acceleration at the front and the rear bogie centre pivot.
- Vertical forces for all the wheel-sets.

The acceleration response of the car-body at speed of $15m/s$, $30m/s$, $45m/s$, $60m/s$ are shown in Fig. 5. Plots show that initially, the value of acceleration is nearly equal to $-9.8m/s^2$, which is mainly the acceleration due to gravity. Finally it goes to zero, when the vibration of the car body ceases and it stabilizes. The acceleration is generally within acceptable range and does not show any instability.

The acceleration response of front and the rear bogie with time is presented at different velocities of vehicle in Fig. 6. It is evident from the plots, which initially the wheels of the front bogies comes in contact with the track irregularity and the vibration starts in the front bogie and latter these vibrations are shifted to the rear bogie. The amplitude of the vehicle vibration also increased with vehicle speed.

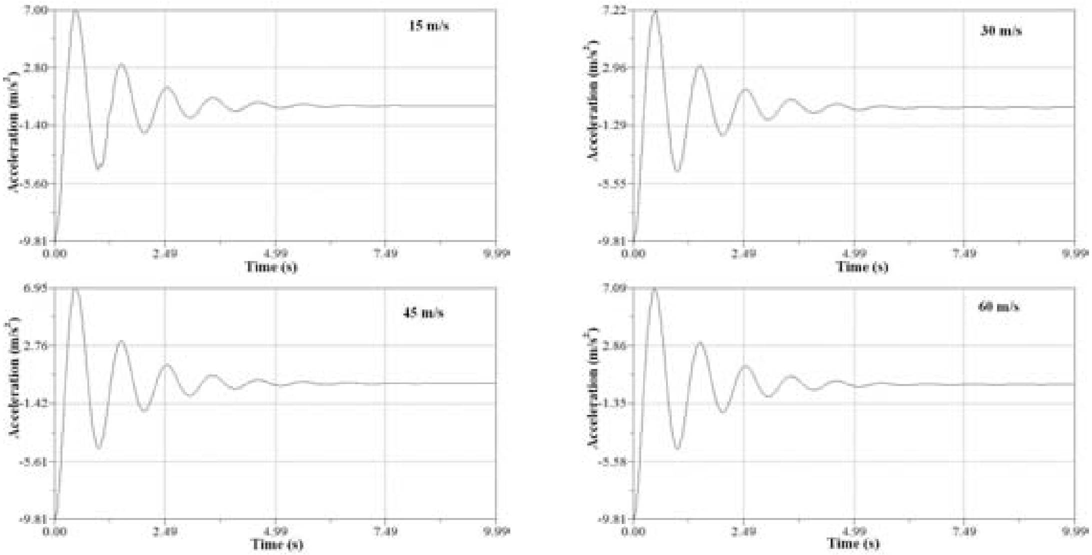

Fig. 5. Vertical acceleration at the floor of the car-body C. G for vehicle velocity of $15m/s$, $30m/s$, $45m/s$ and $60m/s$

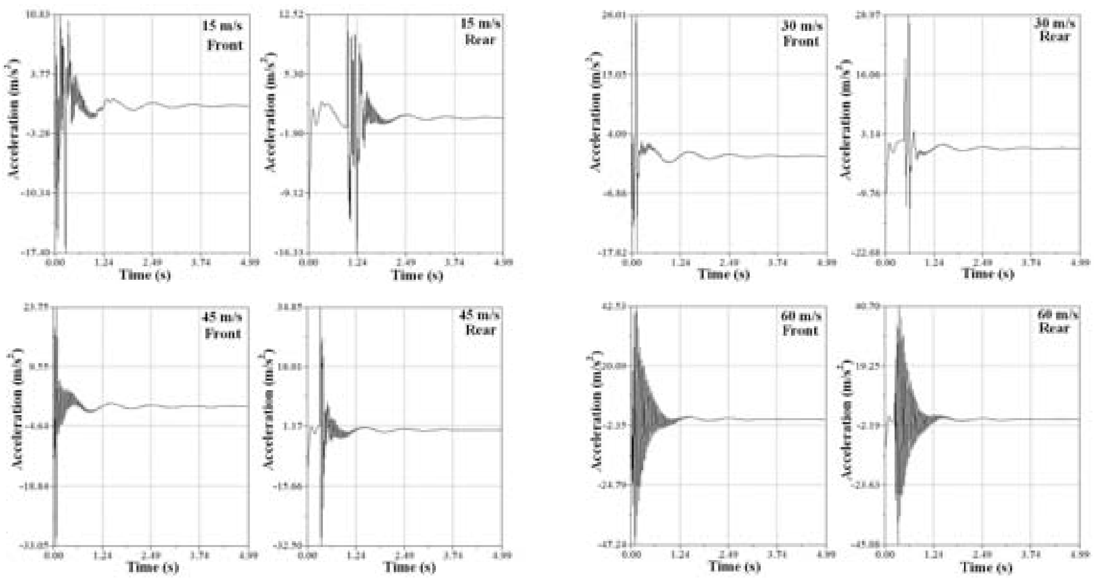

Fig. 6. Vertical acceleration at front & rear bogie centre pivot respectively for vehicle speed of $15m/s$, $30m/s$, $45m/s$ and $60m/s$

5.3 Sperling ride index

Sperling's ride index is defined as[4]:

$$W_Z = \sqrt[10]{\sum_{i=1}^{n_f} W_{Z_i}^{10}} \tag{5}$$

where n_f is the total number of discrete frequencies of the acceleration response of the railway vehicle identified by the FFT and W_{Z_i} is the comfort index corresponding to the ith discrete frequency, given by:

$$W_Z = \left[a_i^2 B(f_i)^2\right]^{\frac{1}{6.67}} \tag{6}$$

where a_i denotes the amplitude of the peak acceleration response (in cm/s^2) measured on the floor of the ith frequency identified by the FFT and $B(f_i)$ a weighting factor, given by:

$$B(f) = k\left[\frac{1.911f^2 + (0.25f^2)^2}{(1-0.277f^2)^2 + (1.563f - 0.0368f^3)^2}\right]^{\frac{1}{2}} \tag{7}$$

where $k = 0.737$ for horizontal vibration and 0.588 for vertical vibration.

5.4 Comfort evaluation

Acceleration v/s frequency plot was generated for car body at vehicle speed of $15m/s$ and is shown in Fig. 7 to calculate the Sperling ride comfort index. The FFT plot is generated for frequency range between 0 to 25 Hz, as the human beings are most sensitive in the frequency range 4 to 12.5 Hz.

Ride comfort analysis has been performed for speeds ranging from $15m/s$ to $60m/s$. The analysis has been performed on the system model to calculate the vertical acceleration of system. FFT output is taken to get peak acceleration frequency component. Comfort index has been calculated through Eqs. (5) $\sim$ (7), which are present in Tab. 3.

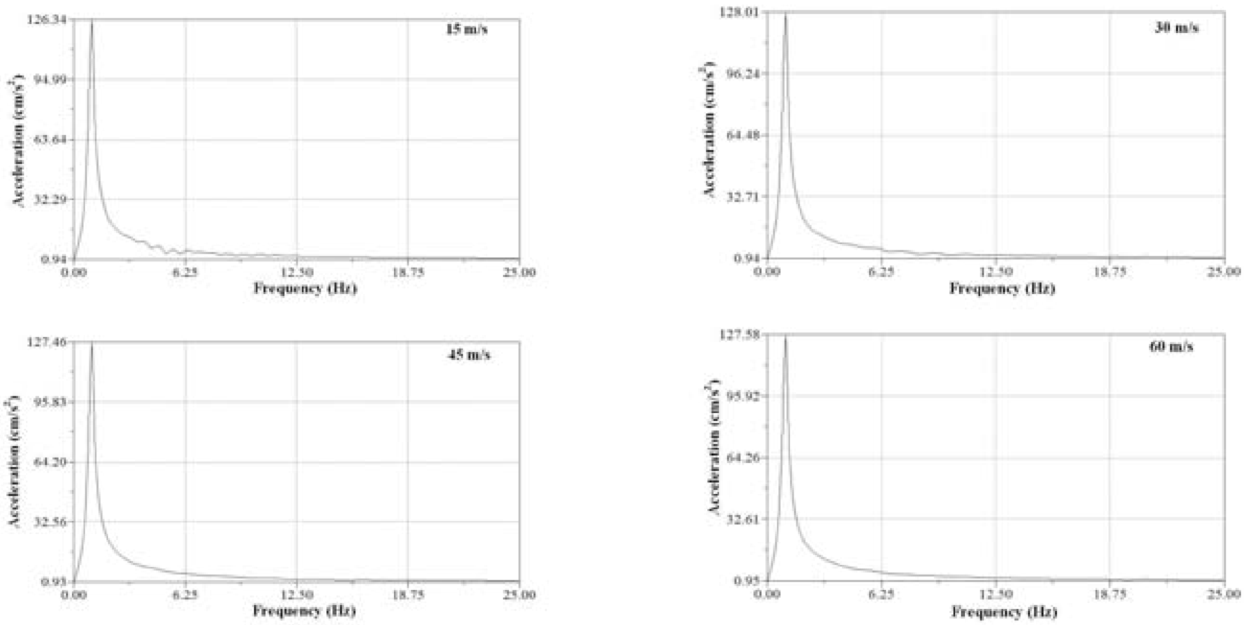

Fig. 7. Vertical acceleration at car body C.G. for vehicle speed of $15m/s$, $30m/s$, $45m/s$ and $60m/s^2$

Table 3. Sperling's ride comfort index for different vehicle velocities

S. No.	Vehicle Speed (m/s)	Sperling Index (Wz)
1	15	2.01
2	30	1.83
3	45	1.72
4	60	1.65

6 Conclusion

Vertical dynamic analysis has been carried out for a typical Indian Railway Vehicle of AC/EMU/T (Alternating Current/ Electrical Multiple Unit/ Trailer) type running on broad gauge. A 17 degree of freedom model is used for analysis. Velocity input at all the wheel set is given by considering similar bump irregularity at both right and the left rail. Vertical acceleration response at the car body C.G has been calculated in the frequency domain. Sperling Ride indices have been found out for the above vehicle. Comparing the result obtained with the standards set by the Indian Railways, the values are found well in the satisfactory limit. The Sperling Ride Index values at different speeds are presented in Tab. 3. The results showed a good agreement with the Indian Standard and validate the correctness of the model.

References

[1] Users manual of symbols shakti. Http://www.htcinfo.com/.

[2] G. Bureika, R. Subacius. Mathematical modeling of dynamic interaction between wheel-set and rail track. *Journal of Transport*, 2002, **XVII**(2): 46–51.

[3] V. Dukkipati, J. Amyot. *Computer-Aided Simulation in Railway Dynamics*. Marcel Dekker, New York, 1988.

[4] J. Forstberg. *Ride comfort and motion sickness in tilting trains - Human responses to motion environments in train and simulator experiments*. Master's Thesis, KTH/ FKT/D-00/28-SE, Division of Railway Technology, Department of Vehicle Engineering, KTH, 2000.

[5] K. Gangadharan, C. Sujatha, V. Ramamurti. Dynamic response of railroad vehicles: a frequency domain approach. *International Journal of Heavy Vehicle Systems*, 2008, **5**(1): 65 – 81.

[6] D. Karnopp, D. Margolis, R. Rosenberg. System dynamics, modelling and simulation of mechatronic systems. *John Wiley & Sons*, 2005.

[7] H. Kumar, Sujata. Vertical dynamic analysis of a typical indian rail road vehicle. **in:** *Proceedings of second international conference on computational mechanics and simulation, IIT*, India, 2006, 8–10.

[8] G. Kumaran, D. Menon, K. Nair. Dynamic studies of rail track sleepers in a track structure system. *Journal of Sound and Vibration*, 2003, **268**: 485–501.

[9] J. Nielsen, A. Igeland. Vertical dynamic interaction between train and track-influence of wheel and track imperfections. *Journal of Sound and Vibration*, 1995, **187**: 825–839.

[10] D. Skarlatos, K. Karakasis, A. Trochidis. Railway fault diagnosis using a fuzzy logic metho. *Applied Acoustic*, 2004, **65**(10): 951–966.