Modeling and Simulation for Vertical Rail Vehicle Dynamic Vibration with Comfort Evaluation

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Abstract. Investigation of vibration is an important topic for the purposes of ride comfort in railway engineering. The vibration of rail vehicles becomes very complex because it is affected by the condition of vehicles, including suspensions and wheel profile, condition of track sections, including rail profile, rail irregularities, cant and curvature. The present study deals with the effects of railway track imperfections on dynamic behavior, and investigates the effect of vehicle speed and the rail irregularity on ride comfort through numerical simulation. The numerical simulation of the vertical dynamic behavior of a typical railroad vehicle will be performed using Largrangian dynamics. The model consists of 17 degrees of freedom with 4 wheelsets, 2 bogies and a car body. For the assessment of the ride comfort, the Sperling ride index (ISO2631) is calculated using filtered RMS accelerations. The ride characteristics of the vehicle provide an assessment of the dynamic behavior of the vehicle through the analysis of the accelerations at the vehicle body, whereas the ride comfort assesses the influence of the vehicle dynamic behavior on the human body. A parametric study was carried out to suggest design modifications in order to improve the level Sperling index.

Keywords: Sperling index, ride quality, rail vehicle, dynamic behavior, ride comfort, vertical dynamic vibration.

1 Introduction

Rail transport is one of the major modes of transportation, so it must offer a high comfort level for passengers and crew. However, the comfort that passengers experience is usually perceived differently from one individual to another. In several research works, 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

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in welding or rolling defects, rail joints, etc. The nature of vibration itself is random and covers a wide frequency range [1]. The improvement of the 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 and its mechanical components, 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 indices. Ride quality is interpreted as the capability of the rail road suspension to maintain the motion within the range of human comfort. Sperling's ride index is a measure of the ride quality and ride comfort used by ISO 2631[2]. Due to the complex dynamics that exists between the rail and wheel, rail vehicle dynamics are often difficult to model accurately. This velocity-dependent dynamics justifies 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, and harmonic functions, to represent the track geometry [3].

There have been several studies, which dealt with the dynamic analysis of rail vehicles in order to enhance the ride comfort while travelling. Nejlaoui et al [4] optimized the structural design of passive suspensions in order to ensure simultaneously passenger safety and comfort. Abood et al [5] investigated the Railway carriage simulation model to study the influence of vertical secondary suspension stiffness on ride comfort of railway car body. Kumar and Sujata [2] presented the numerical simulation of the vertical dynamic behavior of a railway vehicle and calculated Sperling ride index for comfort evaluation. Nielsen and Igeland [6] investigated the vertical dynamic behavior 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.

2 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 multi body system. A multi body system consists of rigid bodies, interconnected via massless 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.

2.1 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 the axle and the bearing.
- The vehicle is moving with constant velocity on a 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 remain in contact with the rails.
- Straight track.
- An irregularity in the vertical direction with the same shape for left and right rails.

2.2 Rail Road Vehicle Model

Figure.1 illustrates the train vehicle model adopted in this study. It consists of a vehicle body, two bogies frames and four wheelsets. Each bogie consists of the bogie frame, and two wheel sets. The car body is modeled as a rigid body having a mass Mc; and having moment of inertia J_{bx} and J_{cy} about the transverse and longitudinal axes, respectively. Similarly, each bogie frame is considered as a rigid body with a mass M_b (M_{b1} and M_{b2}) with moment of inertia J_{bx} and J_{by} about the transverse and longitudinal axes, respectively. Each axle along with the wheel set has a mass M_w (for four axles M_{w1}; M_{w2}; M_{w3} and M_{w4}). The spring and the shock absorber in the primary suspension for each axle are characterized by a spring stiffness K_p and a damping coefficient C_p, respectively. Likewise, the secondary suspension is characterized by a spring stiffness K_s and a 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 Φ_c) and about the longitudinal horizontal axis (roll or θ_c). Similarly, the movements of the three bogie units are described by three degrees of freedom Z_b; θ_b and Φ_b, each about their centers. Each axle set is described by two degrees of freedom Z_w; and Φ_w. about their centers. Totally, 17 degrees of freedom have been considered in this study for the vehicle model shown in Figure.1. The detailed parameters regarding the moment of inertia and mass of different component are given in Table 1.

Table 1 Detailed parameter of rigid bodies

Name of Rigid Bodies	*Mass (Kg)*	*Moment of Inertia(Kg.m²)*	
		I_{XX}	I_{ZZ}
Car Body	*6,7 ×10⁵*	10^5	10^6
Bogie-I and II	10^5	10^5	10^5
Wheel-set-I,II,III and IV	*4000*	*4000*	*4000*

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

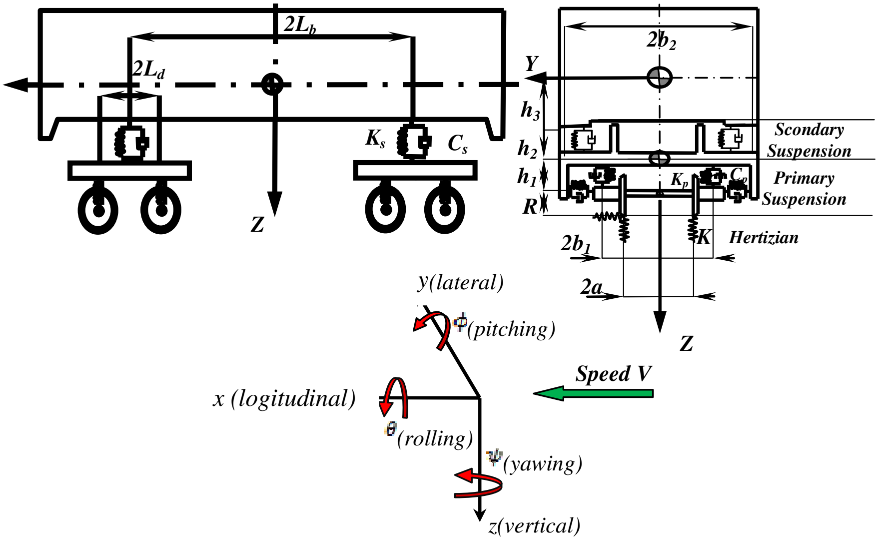


Fig. 1 physical model of railway vehicle

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 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.

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

In actual practice different types of periodic, a-periodic or random track irregularities may exist on the track, but in the present study bump type of irregularity is considered as shown in Figure.2 [3]. The shape of the irregularity is assumed to be similar on the left and the right rails.

Table 2 Vehicle parameters

Parameter	Nomenclature	Values
Primary spring stiffness	K_p	10^6 N/m
Secondary spring stiffness	K_s	$1,7\times10^6$ N/m
Primary damping coefficient	C_p	6×10^4 Ns/m
Secondary damping coefficient	C_s	10^5 Ns/m
Vertical hertz spring stiffness	K_{hz}	35×10^9 N/m
Longitudinal distance between bogies I and II and car body mass center	L_b	6 m
Longitudinal distance between wheel-set and corresponding bogie origin	L_d	1,4 m
Lateral distance between a longitudinal primary suspension and corresponding wheel-set	d_p	1 m
Lateral distance between longitudinal secondary suspension and corresponding bogie origin	d_s	1 m
Lateral distance between contact point of wheel–rail and corresponding wheel-set origin	a	0,7163 m
Lateral distance between vertical primary suspension and corresponding wheel-set origin	b_1	1 m
Lateral distance between vertical secondary suspension and car body mass center	b_2	1 m
Nominal wheel radius	R_1	0,61 m
Vertical distance between wheel-set and bogie mass centers	h_1	0,3 m
Vertical distance between bogie mass center and lateral secondary suspension	h_2	0,2 m
Vertical distance between lateral secondary suspension and car body mass center	h_3	1,3 m

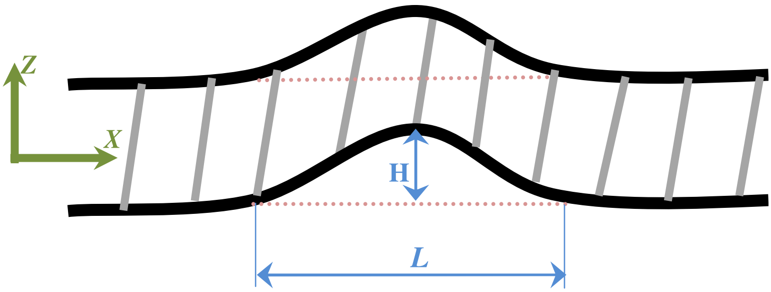


Fig. 2 Model of track irregularity

The bump excitations of the left wheels (Figure. 3) of leading bogies are as follows:

$$Z_{ri} = \begin{cases} \frac{H}{2}\left[1-\cos\left(2\pi\frac{V}{L}\left(t-t_{di}\right)\right)\right] & for \quad t_{di} \le t \le \frac{L}{V}+t_{di} \quad (i=1..4) \\ 0 & otherwise \end{cases} \tag{1}$$

$$\text{Where } \left[t_{d1}, t_{d2}, t_{d3}, t_{d4}\right] = \left[0, \frac{2L_d}{V}, \frac{2L_b}{V}, \frac{2L_b+2L_d}{V}\right] \tag{2}$$

In the present study, H is taken as $0.03m$ and L is taken as $1m$.

4 Simulation Study

The models were built in the MATLAB/ Simulink® environment. The fixed step solver ODE-45 (Dormand-Prince) was utilized, with the sampling time Ts=0.0001. Ts is smaller than the fastest half-car Active Vehicle Suspension Systems (AVSS) model dynamics, enabling observation of all model dynamics [**7, 8**].

Dynamic analysis was carried out for the vehicle at different speeds: 15m/s, 30m/s, 45m/s and 60m/s.

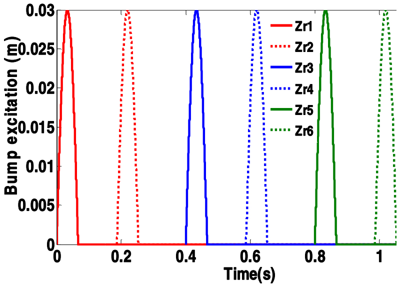


Fig. 3 The bump excitations of the left wheels vehicle speed of 15m/s

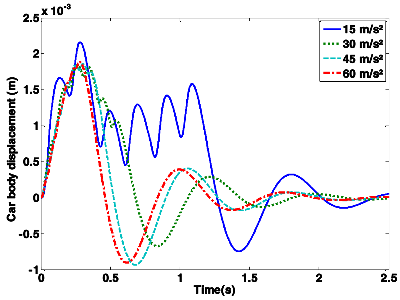


Fig. 4 The vertical Car body displacement for differents vehicle speeds

4.1 Dynamics Analysis

The following output parameters are evaluated:

- Vertical displacement and acceleration at the floor of the car-body center of mass.

Vertical acceleration at the front and the rear bogie center pivot.

The displacement and acceleration responses of the carbody at speeds of 15m/s, 30m/s, 45m/s, 60m/s are shown, respectively, in figure.4 and figure.5. Plots show that initially, the value of acceleration is nearly equal to 0 m/s^2, which is mainly the acceleration without gravity. Finally it goes to zero, when the vibration of the car body ceases and stabilizes. The acceleration is generally within an 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 figure.6.a) and figure.6.b), respectively. It is clear from the plots that initially the wheels of the front bogies come in contact with the track irregularity and the vibration starts in the front bogie and later these vibrations are shifted to the rear bogies. The amplitude of the vehicle vibration also increased with the vehicle speed.

4.2 Sperling Ride Index

Sperling's ride index is defined as [**9, 10, 11**]:

$$W_Z = \left(\sum_{i=1}^{n_f} W_{Z_i}^{10} \right)^{\frac{1}{10}} \tag{3}$$

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

$$W_Z = \left[a_i^2 B(f_i)^2 \right]^{\frac{1}{6.67}} \tag{4}$$

Where a_i denotes the amplitude of the peak acceleration response (in *cm/s²*) measured on the floor of the itch frequency identified by the FFT and $B(f_i)$ a weighting factor, given by:

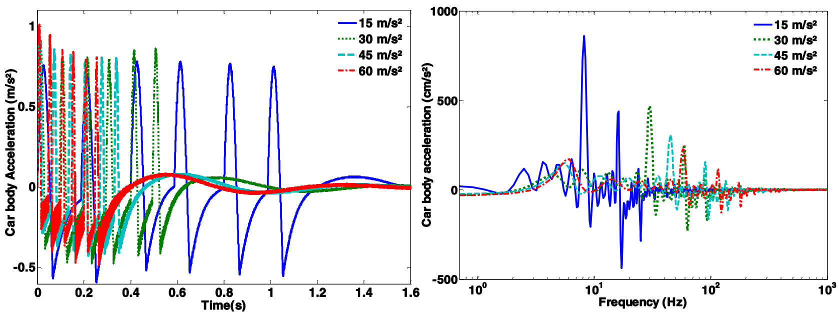
$$B(f) = k\sqrt{\frac{1.911f^2 + \left(0.25f^2\right)^2}{(1-0.277f^2)^2 + (1.563f - 0.0368f^3)^2}} \tag{5}$$

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

4.3 Comfort Evaluation

Acceleration frequency response plots were generated for car body at vehicle speeds of 15m/s to 60 m/s and are shown in figure.5.b) to calculate the Sperling ride comfort index. The FFT plot is generated for a frequency range between 0 to 25 Hz, as the human beings are most sensitive in the frequency range of 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 the system. FFT output is taken to get the peak acceleration frequency component. Comfort index has been calculated through Eqs[3-5], which are presented in Table.3.

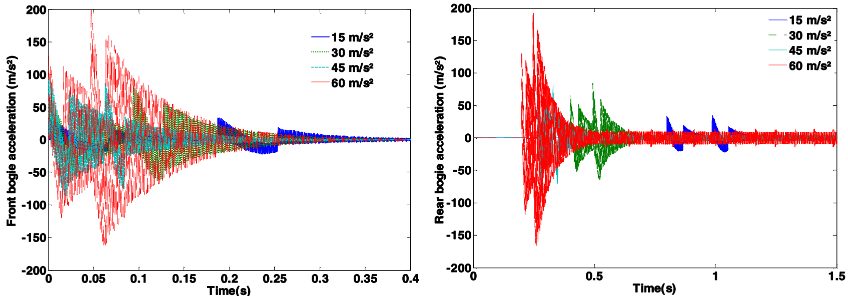


(a) (b)

Fig. 5 The vertical Car body acceleration in time domain (a) and frequency domain (b) for vehicle speed of 15m/s, 30m/s, 45m/s and 60m/s

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

Vehicle Speed (m/s)	Sperling Index (W_z)	Ride comfort evaluation
15	2.53	More pronounced but not unpleasant
30	2.03	Clearly noticeable
45	1.72	Just noticeable
60	1.62	Just noticeable



(a) (b)

Fig. 6 The vertical Front (a) and Rear (b) bogie acceleration for vehicle speed of 15m/s, 30m/s, 45m/s and 60m/s

The maximum and minimum ISO Sperling Index values are respectively 2.53 and 1.62 for the rail vehicle speed respectively 15 m/s and 60 m/s. These values respectively indicate "The more pronounced but not unpleasant" and "just noticeable" zones. Figure 7 show that the comfort index decreases as the vehicle speed increases while maintaining an acceptable level of comfort. This means that the passengers are not much affected by the vibration as they are exposed to low level of vibrations.

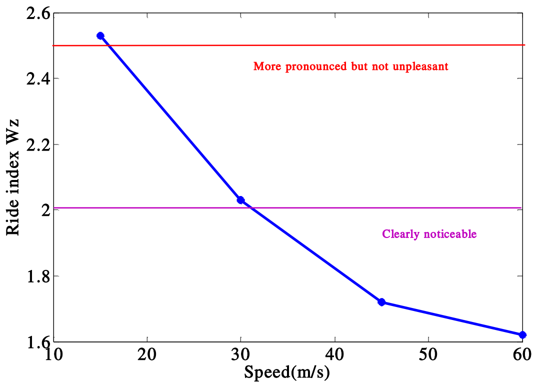
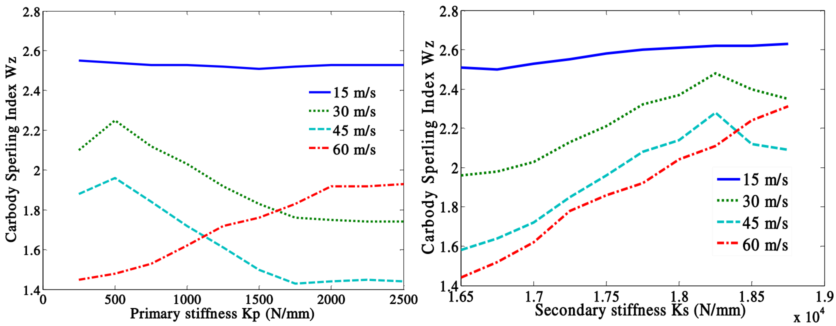


Fig. 7 The Sperling index comfort variation for different speeds

4.4 Parametric Study

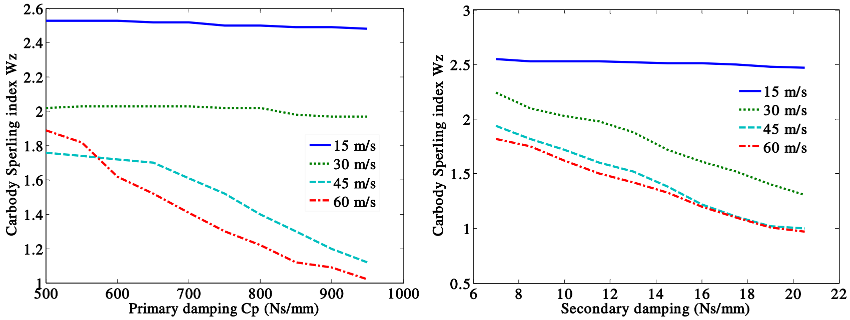
A parametric study was undertaken to find the influence of different suspension parameters on the Sperling index (Wz). The parameters considered for the analysis were primary stiffness, secondary stiffness, primary damping and secondary damping. The results of the parametric study have been plotted in terms of performance index such as the Wz index versus vehicle parameter. Figures 8 and 9 show the variation of Wz index as a function of stiffness and damping respectively (primary and secondary). The increase in the primary suspension stiffness reduces the Wz index at the carbody marginally up to a speed of 60 m/s. The influence of secondary stiffness has been found to be just the opposite of the primary stiffness. A reduction of the secondary stiffness value from the present value reduces the Wz index at the carbody at all the speeds. The variation of the primary damping is seen to have little influence on the Wz index at the carbody. At speeds above 45 m/s, increase in the primary damping is shown to produce marginal reduction in the Wz index at the carbody. The secondary damping has great influence at speeds higher than 30 m/s.



a) b)

Fig. 8 The influence of Primary a) and secondary b) stiffness

An increase in the secondary damping reduces the Wz index at speeds greater than 30 m/s, whereas up to 30 m/s speed, the secondary damping has little influence. It has also been observed that all the primary damping have very limited influence at low speeds.



a) b)

Fig. 9 The influence of Primary a) and secondary b) damping

5 Conclusion

Vertical dynamic analysis has been carried out for a Railway Vehicle. A 17 degree of freedom model is used for the analysis. Velocity input at all the wheelset is given by considering similar bump irregularities at both right and left rail. A vertical acceleration response at the car body has been calculated in the frequency domain. Sperling Ride index has been calculated for the above vehicle. The Sperling Ride Index values at different speeds are presented. The calculated values of the Sperling index are found well in the satisfactory limits defined by the ISO 2631 standard which means that the passengers are not much affected by the vibration as they are exposed to low level of vibrations. A parametric study has been carried out with emphasis on better ride index. The parametric study has brought out possible design changes required in different parameters to deliver better Sperling comfort index. It should be noticed that the parametric study was carried out to suggest design modifications to improve Wz index, but others dynamic and control behaviors like stability, control of secondary suspension were to be considered when implementing the design modifications.

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