Sharma. 2016. *Int. J. Vehicle Structures & Systems*, 8(1), 1-6
ISSN: 0975-3060 (Print), 0975-3540 (Online)
http://dx.doi:10.4273/ijvss.8.1.01

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

Evaluation of Passenger Ride Comfort of Indian Rail and Road Vehicles with ISO 2631-1 Standards: Part 1 - Mathematical Modeling

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ABSTRACT:

Ride quality and ride comfort are the most important performance indices of road or rail vehicles and is affected by various factors, such as vibrations, acoustics, smell, temperature, visual stimuli, humidity and seat design. Among these vibration is a dominant factor that influences the performance indices the most. In this work the coupled vertical-lateral mathematical models of Indian rail and road vehicles have been formulated using Lagrangian. The roadway vehicles considered for this analysis are three wheel and light four wheel Indian passenger vehicle. The rail vehicles considered for this analysis are General sleeper ICF coach of Indian railway.

KEYWORDS:

Ride comfort; ISO 2631;Lagrangian method; Vehicle model; Frequency weightings

CITATION:

R.C. Sharma. 2016. Evaluation of Passenger Ride Comfort of Indian Rail and Road Vehicles with ISO 2631-1 Standards: Part 1 - Mathematical Modeling, *Int. J. Vehicle Structures & Systems*, 8(1), 1-6. doi:10.4273/ijvss.8.1.01.

NOMENCLATURE:

$m_{C,B}$	Mass of car body and bolster respectively
$m_{BF,W}$	Mass of bogie frame and wheel axle respectively
$I_C^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of car body respectively
$I_B^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of bolster respectively
$I_{BF}^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of bogie frame respectively
$I_W^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of wheel axle respectively
$k_{CB}^{z,y}$	Vertical (1/2 part) and lateral (1/2 part) stiffness between car body and bolster respectively
$c_{CB}^{z,y}$	Vertical (1/2 part) and lateral (1/2 part) damping coefficient between car body and bolster respectively
$k_{BBF}^{z,y}$	Vertical (1/4 part) and lateral (1/4 part) stiffness between bolster and bogie frame respectively
$c_{BBF}^{z,y}$	Vertical and lateral damping coefficient between bolster and bogie frame respectively (1/2 part)
$k_{BFWA}^{z,y}$	Vertical (1/2 part) and lateral (1/2 part) stiffness between bogie frame and corresponding wheel axle
$c_{BFWA}^{z,y}$	Vertical (1/4 part) and lateral (1/2 part) damping coefficient between bogie frame and corresponding wheel axle
$k_R^{z,y}$	Vertical and lateral track stiffness
$c_R^{z,y}$	Vertical and lateral track damping coefficient
t_B	Lateral distance from bolster c.g. to vertical suspension between bolster and bogie frame
t_C	Lateral distance from car body c.g. to side bearings
t_w	Lateral distance from bogie frame c.g. to corresponding vertical suspension between bogie frame and wheel axle
z_{12}	Vertical distance between the car body c.g. and the bolster c.g.
z_{24}	Vertical distance between the bolster c.g. and the bogie frame c.g.
z_{46}	Vertical distance between the bogie frame c.g. and the corresponding wheel axle c.g.
x_{12}	Horizontal distance between the car body c.g. and the bolster c.g
x_{46}	Horizontal distance between the bogie frame c.g. and the corresponding wheel axle c.g.
z	Lateral distance from bolster c.g. to vertical suspension between bolster and bogie frame
m_v	Total mass of three wheeler
m_s	Sprung mass of three wheeler
m_{fs}	Front steering mass of three wheel vehicle
m_{ft}	Mass of front wheel of three wheeler
m_{rt}	Mass of single rear wheel of three wheeler
$I_s^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of sprung mass about its c.g. of three wheeler
$I_{fs}^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of front steering mass about its c.g. of three wheeler
k_{fs}	Front suspension vertical stiffness of three wheeler
c_{fs}	Front suspension vertical damping coefficient of three wheeler
k_{ft}	Front tyre vertical stiffness of three wheeler
c_{ft}	Front tyre vertical damping coefficient of 3 wheeler
k_{ru}	Rear suspension vert. stiffness (1/2 part) of 3 wheeler

c_{ru}	Rear suspension vertical damping coefficient (1/2 part) of three wheeler
k_{rt}	Rear tyre vertical stiffness of three wheeler
c_{rt}	Rear tyre vertical damping coefficient of three wheeler
x	Wheel base of three wheeler
y	Wheel gauge of three wheeler
r_w	Radius of both front and rear wheel of three wheeler
α_s	Steering axis inclination angle of three wheeler
M	Sprung mass of four wheel vehicle
m_f	Front unsprung mass of four wheel vehicle
m_r	Rear unsprung mass of four wheel vehicle
$I_M^{x,y,z}$	Roll, pitch and yaw mass moment of inertia of sprung mass about its c.g. of four wheeler
k_{sf}	Front suspension vertical stiffness (1/2 part) of four wheeler
c_{sf}	Front suspension vertical damping coefficient (1/2 part) of four wheeler
k_{tf}	Front tyre vertical stiffness (1/2 part) of four wheeler
c_{tf}	Front tyre vertical damping coefficient (1/2 part) of four wheeler
k_{sr}	Rear suspension vertical stiffness (1/2 part) of four wheeler
c_{sr}	Rear suspension vertical damping coefficient (1/2 part) of four wheeler
k_{tr}	Rear tyre vertical stiffness of four wheeler
c_{tr}	Rear tyre vertical damping coefficient of 4 wheeler
x_w	Wheel base of four wheeler
y_w	Wheel gauge of four wheeler

1. Introduction

Ride comfort is one of the important characteristics in evaluating the performance of rail and road vehicles. Ride comfort is affected by many factors, such as vibration, acoustic noise, smell, temperature, humidity, visual stimuli and seat design. Another important aspect of comfort is structure borne and air borne noises. Structure borne noise has no sharp boundary to mechanical vibrations, particularly in the frequency range of 20-80 Hz, where vibrations are commonly both sensible and audible. Motions and vibrations of the train is also an important aspect in passenger comfort [1-2]. Motions and vibrations may be transient or stationary. In rail and road vehicles vibrations are very often more or less transient. This comfort aspect will, in this context, be designated as motion related comfort or more commonly as ride comfort. Traditionally, ride comfort is investigated in the frequency range of 0.5-20 Hz, i.e. in the non-audible range. However, the new proposed ISO and CEN standards recommend that vibrations up to 80 Hz should be considered [17]. The frequency range below 0.5 Hz should be considered as well, if the risk of motion sickness is to be evaluated [8]. Among the various factors of ride comfort, vibration, which originates from vehicle motion, is taken as the primary concern because its effect is relatively large in vehicles. Other non-motion factors also affect ride comfort. However, it is very difficult to consider overall factors simultaneously [6].

A study by Suzuki et al [16] attempted to consider non-motion factors such as acoustic noise, illumination and arrangement of seats, as well as vibration, in evaluating the ride comfort of railway vehicles. However, it is still a problem to establish a general ride comfort index including various non-motion factors because it needs a great amount of test data, which incurs a very high cost. For this reason, evaluating ride comfort in view of vibration is still an effective methodology and is generally accepted in vehicle engineering. There are various means by which the vibration can be expressed, such as displacement, velocity and acceleration. Of these physical quantities, acceleration is generally adopted as a preferred measure of quantifying the severity of human vibration exposure [6]. In this paper coupled vertical-lateral models of a 37 Degrees of Freedom (DoF) General Sleeper ICF coach, a 9 DoF three wheel and four wheel road vehicles have been formulated using Lagrangian dynamics. Rail and road vehicles have been formulated using coupled vertical-lateral models by different researchers [3, 7, 9, 12-15] and they have utilised those equations of motion in evaluating different aspects. Each and every country has its own criteria for ride comfort evaluation. The criteria adopted in India will be discussed in next section.

2. Ride comfort of rail and road vehicle according to Indian standards

The ride comfort of Indian roadway vehicle is evaluated using ISO 2631-1 1997 specifications. For Indian rail vehicle Sperling criteria is adopted. Human sensation of comfort is dependent on displacement, acceleration and the rate of change of acceleration. In other words, the product of displacement, acceleration and the rate of change of acceleration could be used as a measure of discomfort during travel. For sinusoidal vibration with β as the amplitude and ω as its periodicity, the Sperling's Ride Index (RI) can be derived as follows:

Displacement: $s = \beta \sin \omega t$

Velocity: $v = ds/dt = \beta\omega \cos \omega t$

Acceleration: $a = dv/dt = -\beta\omega^2 \sin \omega t$

Impulse: $I = da/dt = -\beta\omega^3 \cos \omega t$ (1)

Thus, level of discomfort: $\alpha = aIs$ (2)

Taking the maximum value of parameters over the half wave of displacement, the RI is given by,

$$RI = \alpha\left(-\beta\omega^2\right)\left(-\beta\omega^3\right)\beta = \alpha\left(\beta^3\omega^5\right) = k * \beta^3\omega^5 \quad (3)$$

For amplitude of acceleration $b = -\beta\omega^2$ and $\omega = 2\pi f$ with *f* being the vibration frequency, Eqn. (1) becomes,

$$RI = k\left(-b/\omega^2\right)^3 \omega^5 = -k\, b^3/\omega = K\, b^3/f \quad (4)$$

For an individual, the sensation of vibration varies according to an exponential law and thus RI for ride quality is given by,

$$RI = 0.896\left(b^3/f\right)^{0.1} \quad (5)$$

In order to take human reactions, Eqn. (5) is modified by taking into a correction factor and thus RI for ride comfort is given by,

$$RI = 0.896\left(b^3\phi(f)/f\right)^{0.1} \tag{6}$$

Where $\phi(f)$ is a frequency dependent factor expressing human vibration sensitivity and is different for vertical and lateral vibration components. The term ride comfort is equivalent to RI for locomotive and coaching stock and ride quality is equivalent to RI for freight stock. The classification of RI with reference to subjective appreciation is adopted on Sperling's scale are given in Table 1 and Table 2.

Table 1: Ride quality of rail vehicle

Ride Index	Appreciation
1	Very good
2	Good
3	Satisfactory
4	Accepted for running
4.5	Not accepted for running
5	Dangerous

Table 2: Ride comfort of rail vehicle

Ride Index	Appreciation
1	Just noticeable
2	Clearly noticeable
2.5	More pronounced but not unpleasant
3	Strong, irregular but still tolerable
3.25	Very irregular
3.5	Extremely irregular, unpleasant, annoying, prolonged exposures intolerable
4	Extremely unpleasant, prolonged exposure harmful

In the analogue or chart recorder method, the correction factors for frequencies below 0.5 Hz are not relevant since the average common frequency is always above 0.5 Hz. In DAS method, since the frequency of every half wave has to be considered individually, some of the half waves may have frequencies less than 0.5 Hz for which a correction factor of 0 has been assigned. This is because frequencies below 0.5 Hz have very low energy levels and do not affect the ride comfort. The accelerometers are placed on the floor level of the vehicle near the pivot centre for measuring the acceleration and calculating RI. The correction factors for various frequency values are given in Table 3. The traditional Indian method for calculation of RI by taking a common average frequency for all the half waves and average amplitude for each range of peaks (called Bin method) was essentially adopted from old SNCF (French Railway) practice as mentioned in ORE report No. 8 of C116 of 1977. This is an analogue method involving reading of acceleration oscillogram recorded on chart recorder and counting of peaks.

Table 3: Correction factors applied to different frequencies

Vertical mode	Lateral mode
0 for f < 0.5 Hz	0 for f < 0.5 Hz
0.325 f^2 for 0.5 < f< 5.4 Hz	0.8 f^2 for 0.5 < f< 5.4 Hz
400/f^2 for 5.4 < f < 20 Hz	650/f^2 for 5.4 < f < 20 Hz
1 for f > 20 Hz	1 for f > 20 Hz

3. Rail & road vehicle models

In this study, the following assumptions have been made in formulating the mathematical model of rail and road vehicles:

- The vehicle is moving with constant speed.
- All mass assemblies are considered to be rigid bodies and vehicle structure is not deformable.
- The rail/road vehicle is assumed to possess longitudinal plane of symmetry (c.g. of all masses lie in central plane).
- Wind drag forces do not affect the motion of the rail/road vehicle.
- All the springs and dampers are linear within the range of suspension travel [5].
- Track/tyre surface possess linear stiffness and damping properties.
- Wheel-track / tyre-road surfaces are in contact.
- Hertzian contact stiffness at the interface is accounted and contact patch is taken as ellipse.
- Creep forces and creep moments are linear functions of creepages for rail vehicle [10, 11].

A coupled vertical-lateral rail vehicle model (Fig. 1 and Fig. 2) has been formulated using Lagrangian dynamics with 37 DoF assigned to 9 rigid bodies i.e. car body (vertical, lateral, roll, pitch and yaw), front and rear bolsters (vertical, lateral and roll), front and rear bogie frames (vertical, lateral, roll, pitch and yaw) and four wheel axle sets (vertical, lateral, roll and yaw). The values of the different parameters of rail vehicle model are given in Table 4. The track may be divided into a super structure and a sub structure. The super structure includes rails, rail fastenings, pads, sleepers and ballast (i.e. soil). The sub grade or subsoil is the sub structure of a track. The track in the present analysis is assumed to be flexible in both vertical and lateral directions. Its flexibility is accounted by considering wheel to be in series with sleeper, soil and subsoil Fig. 3.

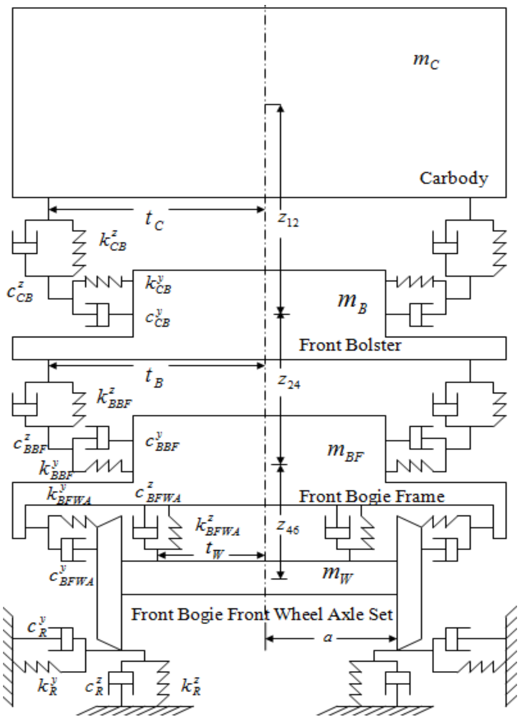

Fig. 1: Rail-vehicle model (side view)

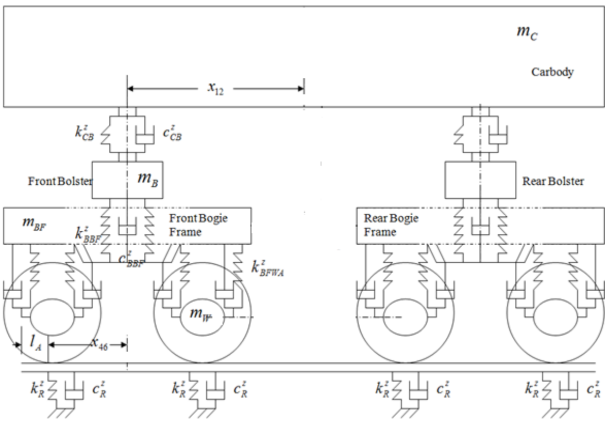

Fig. 2: Rail-vehicle model (front view)

Table 4: Parameters value for rail vehicle model

Parameter	Parameter value	Parameter	Parameter value
m_C	37960 kg	c_{CB}^z	0.035 MN-s/m
m_B	400 kg	c_{CB}^y	0.0175 MN-s/m
m_{BF}	2346 kg	k_{BBF}^z	0.42375 MN-s/m
m_W	1487 kg	k_{BBF}^y	0.2324 MN-s/m
I_C^x	63950 kg m^2	c_{BBF}^z	0.0589 MN-s/m
I_C^y	1470750 kg m^2	c_{BBF}^y	1 MN-s/m
I_C^z	1473430 kg m^2	k_{BFWA}^z	0.26935 MN/m
I_B^x	307 kg m^2	k_{BFWA}^y	11.5 MN/m
I_B^y	00	c_{BFWA}^z	0.0206 MN-s/m
I_B^z	336.5 kg m^2	c_{BFWA}^y	0.5 MN-s/m
I_{BF}^x	1546 kg m^2	t_C	0.8 m
I_{BF}^y	2893 kg m^2	t_B	1.127 m
I_{BF}^z	4298 kg m^2	t_W	1.079 m
I_W^x	1181 kg m^2	z_{12}	1.3275 m
I_W^y	108.5 kg m^2	z_{24}	0.1435 m
I_W^z	1181 kg m^2	z_{46}	0.194 m
k_{CB}^z	35 MN/m	x_{12}	7.3915 m
k_{CB}^y	17.5 MN/m	x_{46}	1.448 m

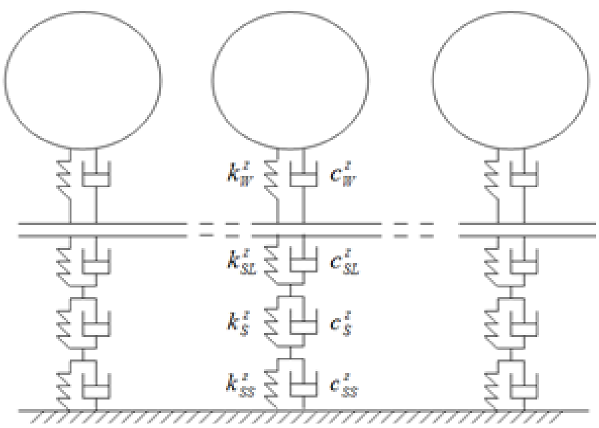

Fig. 3: Track model

A coupled vertical-lateral three wheel vehicle model (Fig. 4) has been formulated using Lagrangian dynamics with 9 DoF considered i.e. 5 DoF of sprung mass (vertical lateral, roll, pitch and yaw), 2 DoF for front wheel steering arm (vertical and angular motion of steering arm) and 1 DoF each for rear unsprung mass (vertical). The mass of front tyre is included in front steering mass and total rear tire mass is the same as rear unsprung mass. The values of the different parameters of three wheel vehicle model are given in Table 5. Fig. 5 and Fig. 6 show different views of a 9 DoF heavy duty roadway passenger vehicle considering 5 DoF of sprung mass (vertical lateral, roll, pitch and yaw) and 4 DoF each for unsprung mass (vertical) considering independent suspension. The values of the different parameters of four wheel vehicle are given in Table 6.

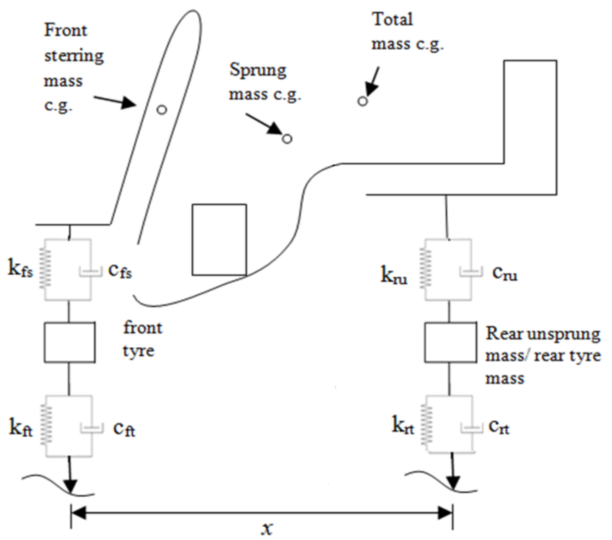

Fig. 4: Three wheel vehicle model

Table 5: Parameters value for three wheel vehicle model

Parameter	Parameter value	Parameter	Parameter value
m_v	532 kg	c_{fs}	3500 N-s/m
m_{fs}	12.5 kg	k_{ft}	238 kN/m
m_s	493 kg	c_{ft}	557 N-s/m
m_{ft}	8.5 kg	k_{ru}	49.8 kN/m
m_{rt}	9 kg	c_{ru}	2200 N-s/m
I_s^x	182 kg-m^2	k_{rt}	250 kN/m
I_s^y	170 kg-m^2	c_{rt}	436 N-s/m
I_s^z	163 kg-m^2	x	2 m
I_{fs}^x	1.9 kg-m^2	y	1.15 m
I_{fs}^y	1.07 kg-m^2	r_w	0.205 m
I_{fs}^z	1.3 kg-m^2	α_s	20°
k_{fs}	32.7 kN/m	-	-

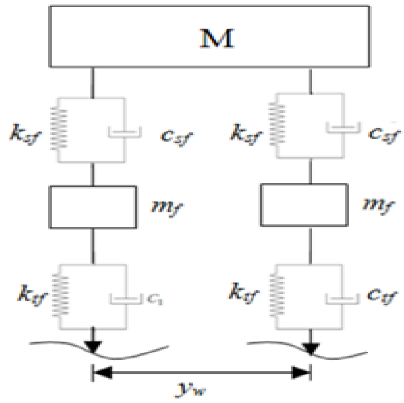

Fig. 5: Four wheel vehicle model (side view)

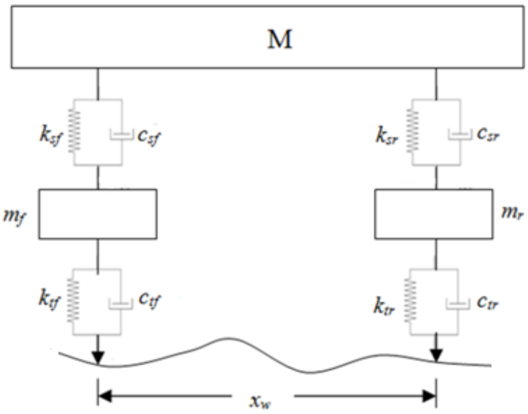

Fig. 6: Four wheel vehicle model (front view)

Table 6: Parameters value for four wheel vehicle model

Parameter	Parameter value	Parameter	Parameter value
M	581 kg	k_{tf}	400 kN/m
m_f	28.5 kg	c_{tf}	700 N-s/m
m_r	48.5 kg	k_{sr}	55 kN/m
I_M^x	725	c_{sr}	6300 N-s/m
I_M^y	1400	k_{tr}	400 kN/m
I_M^z	1200	c_{tr}	700 N-s/m
k_{sf}	50 kN/m	x_w	2.36 m
c_{sf}	6000 N-s/m	y_w	1.29 m

4. Railway track/road surface roughness

The vertical and lateral irregularities on rail track and road surface are considered as random and represented by power spectral density functions. For the railway track auto and cross-power spectral density functions of vertical and lateral irregularity for a straight track reported by Goel et al [4] are used and are also represented by following,

$$S(\Omega)=C_{sp}\Omega^{-N} \tag{7}$$

Where C_{sp} is an empirical constant and N characterizes the rate at which the amplitude decreases with frequency. For the road track auto and cross-power spectral density functions of vertical and lateral irregularity for a straight track reported by Ramji [14] are used and represented by same function as in Eqn. (7).

5. Ride analysis

The equations of motions for the rail and road vehicle systems are formulated using Lagrangian dynamics in the following form

$$[M]\{y_i\}+[C]\{y_i\}+[K]\{y_i\}=[F_r(\omega)] \tag{8}$$

Where [M], [K] and [C] are the mass, stiffness and damping matrices respectively for the vehicle. $[F_r(\omega)]$ is force matrix for displacement excitations at the wheel/rail or wheel/road contact points. Eqn. (8) may also be written as,

$$([M](-\omega^2)+[C](i\omega)+[K])y_i e^{i\omega t}=[F_r(\omega)]q_r e^{i\omega t} \tag{9}$$

$$[D_1]H_r(\omega)=F_r(\omega) \tag{10}$$

Where [D_1] is the dynamic stiffness matrix and $H_r(\omega)=(y_i/q_r)$ is the complex frequency response function for rth input. For a linear system subjected to random inputs, using input-output relationships for spectral densities, the auto and cross-spectral density matrix of the response may be written as

$$[S_{yy}(\omega)]=[H_r(\omega)][S_r(\omega)][H_r(\omega)]^T \tag{11}$$

The complex frequency response functions $[H_r(\omega)]$ can also be defined as the ratio of the response rate to unit harmonic input at a given point. The superscript T denotes transpose of matrix. It may be noted here that the above equation is used independently for vertical and lateral irregularities of the track. The mean square acceleration response (MSAR) at the car body mass center expressed as $(m/s^2)^2/Hz$, which is nothing but PSD of acceleration which may be written as,

$$MSAR=(2\pi f)^4[H_r(\omega)][S_r(\omega)][H_r(\omega)]^T \tag{12}$$

For root mean square acceleration response (RMSAR) at the center frequency, f_c, the power spectral density function is integrated over one third octave band using,

$$RMSAR=\sqrt{(2\pi)^4\int_{0.89\,fc}^{1.12\,fc}[S_{YY}(f)](f)^4\,df} \tag{13}$$

The RMSAR values of car body mass centre at a series of centre frequencies within the range of interest are obtained independently in vertical and lateral directions and then the ride comfort is evaluated and compared with the specified ISO 2631-1 standards [6]. In the present analysis principal frequency weightings with multiplication factors specified in ISO 2631-1 standards are applied to RMSAR values to obtain frequency-weighted acceleration for evaluation of passenger comfort in the sitting position.

6. Conclusions

The ‘rigid body’ assumption is valid in the frequency range of interest since the rail/road vehicle modes frequencies are in this range only and the vibration modes due to structural flexibility occur much above 30 Hz. For the rail vehicle track mass is not considered or it is not allotted to any DoF. This is because the random inputs are required to be considered from track itself. Considering DoF and inputs at the same time would be a very complex procedure. Although the models of rail/road vehicles are based on certain simplifying assumptions, the consideration of random inputs would be able to represent the practical aspects of the road/rail dynamics.

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