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Ride Comfort Analysis of Unsuspended Cabin Tractor Semi-Trailer

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ABSTRACT:

This research work uses parameter variation techniques for calculating the set of vehicle parameters that result in the best ride comfort for the driver. Tractor semi-trailer vertical dynamic ride model is developed using 9 degrees-of-freedom. The simulation of the model is conducted using MATLAB. The input to the system is a velocity amplitude function of the vertical road irregularities. Other inputs include the load conditions, vehicle speed and road conditions. The RMS vertical weighted acceleration at the driver's seat and at the centre of gravity of tractor and semi-trailer for different parameters of the vehicle are assessed for ride comfort analysis.

KEYWORDS:

Ride comfort; RMS vertical weighted acceleration; Unsuspended cabin; Tractor semi-trailer

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1. Introduction

Computer simulation of the tractor semi-trailer allows the parameters to be varied and the response to be studied with multiple types of loading conditions, road conditions, and speeds. Studies conducted by Foster [1] and Flower [2] concluded that Root Mean Square (RMS) accelerations were effective to analyze the effects of various cab suspensions and thereby improves the driver ride comfort. The greatest improvements in comfort were found to be in the frequency range 1 to 3 Hz in which the human body was found to be more sensitive. The linear equations of motion describing the dynamic behaviour of an articulated vehicle have been written for vehicle models ranging from a simple 3 degrees of freedom (DoF) model by Ellis [3] to a complex 38 DoF model by van Deusen [4]. ElMadany et al. [5] have utilized a frequency response analysis to obtain the optimum suspension of an articulated vehicle. By minimization of peak acceleration value of the cab, Walther et al. [6] compared theoretical frequency response analysis to an experimental investigation. The experimental results showed only marginal verification of the theoretical results.

Various tractor-towed models have been proposed in the literature for on-road [7-8] and off-road [9-10] operations. The models developed for off-road applications are of varying degrees of fidelity. Dynamic models for a tractor represent the system responses more closely than kinematic models [11] and contain tire lateral force models with several soil-tire interaction parameters. The inaccuracies in these parameters will affect the accuracy of the model-based responses as the responses are sensitive to these model parameters [12]. Frequency response methods allow the results to be easily compared to ride quality standards set forth by the International Standard Organization, ISO 2631-1 [13]. Different body types of drivers and seating positions make it nearly impossible to determine an exact comfort limit for every driver. The ride quality standards exist as upper boundaries of the RMS vertical accelerations measured at the driver's seat over the frequency range from 0.1 to 50 Hz. The boundaries represent the amount of time the driver can sustain that particular acceleration before the frequency level becoming uncomfortable. As one would expect, lower acceleration magnitudes can be tolerated as the driver operates the vehicle for longer periods of time.

This paper details the development of a mathematical model for ride comfort analysis of a tractor semi-trailer. The developed parametric model provides the user with the best set of values based on the vertical ISO weighted acceleration of the unsuspended cabin centre of gravity (CG). The model has 9 DoF and focuses on the vertical dynamic response. Amongst the outputs given by the parametric model, the RMS accelerations at the driver's seat are most relevant to the ride comfort analysis. In this paper, the time dependence of ride comfort is retained for health evaluation as it focuses on the vehicle with long periods of operation.

2. Vehicle dynamic model

The tractor semi-trailer was modelled with 5 DoF for the tractor and 4 DoF for the semi-trailer as shown in Fig. 1. The tractor DoF are the pitch and bounce (one steer axle and two drive axles). The semi-trailer DoF are the pitch and bounce of three drive axles. This model is two dimensional in the longitudinal and vertical plane. The model, based on six axle baseline vehicle configuration has one axle in the front, two in the tractor rear, and three semitrailer axles. Each of all three wheel and axle assemblies is represented by a lumped unsprung mass having vertical translational DoF only. The tires are modelled as systems having vertical spring and damping characteristics. The dynamics due to seat and cab suspensions is neglected and therefore seat, cab, engine and tractor chassis are considered as one rigid body. Semi-trailer with its chassis is also considered as one rigid body. Both units are allowed to translate in the longitudinal and vertical directions and to pitch except as constrained by the fifth wheel.

By considering the kinematic constraints imposed by the fifth wheel on the motion of the tractor and semitrailer, the model includes 9 DoF as given in Table 1. In addition, dynamic force balance in the longitudinal direction provides a relationship for the fore-aft motion of the tractor in terms of pitch coordinates. Therefore, the total DoF for the vehicle model are expressed by the remaining generalized equations. The governing equations were derived using the Lagrangian approach discussed by Meirovitch [14] which uses the kinetic and potential energies of the tractor semi-trailer elements. The equations of motion are arranged in matrix form,

$$[M]\{\ddot{X}\}+[C]\{\dot{X}\}+[K]\{X\}=\{F(t)\} \tag{1}$$

Where [M], [C] and [K] are the mass, damping and stiffness matrices respectively.

The matrix X is the vector of system unknowns as,

$$X^T=\left[y_{t,}\theta_{t,}\theta_{s,}y_{1,}y_{2,}y_{3,}y_{4,}y_{5,}y_6\right] \tag{2}$$

Where Y_t is tractor vertical displacement, θ_c and θ_s is pitch angle of tractor semi-trailer respectively. Y_1 is tractor front unsprung mass 1 vertical displacement, Y_2 and Y_3 is vertical displacement of tractor unsprung mass at 2 rear axles. Y_4, Y_5 and Y_6 are vertical displacement of semi-trailer unsprung mass at 3 rear axles. The matrix q^T is the vector of the road profile vertical displacement given by,

$$q^T=\left[q_{1,}q_{2,}q_{3,}q_{4,}q_{5,}q_6\right] \tag{3}$$

Where q is road inputs. It is convenient to use the time domain technique for ride quality investigation since most ride quality specifications consist of limits upon the accelerations of the vehicle as a function of frequency. The transfer function between x and q is defined by:

$$x(i\omega)=\left[-\omega^2M+i\omega C+K\right]^{-1}q(i\omega)_{6\times 9} \tag{4}$$

The articulation between the tractor and semi-trailer can be analysed using the formulated dynamic model. The equation of motion of the vehicle system together with the kinematic constraints is given below.

$$y_s=y_t+b_6\theta_t+b_5\theta_s \quad \text{and} \quad x_s=x_t-h_1\theta_t+h_2\theta_s \tag{5}$$

By integrating twice the equation of motion of the whole system in the longitudinal direction

$$\begin{aligned} &x_t=a(h_1\theta_t+h_2\theta_s), \quad a=m_8/(m_7+m_8)\\ &m_7=m_t+\sum_{i=1}^{3}m_i, \quad m_8=m_s+\sum_{i=4}^{6}m_i \end{aligned} \tag{6}$$

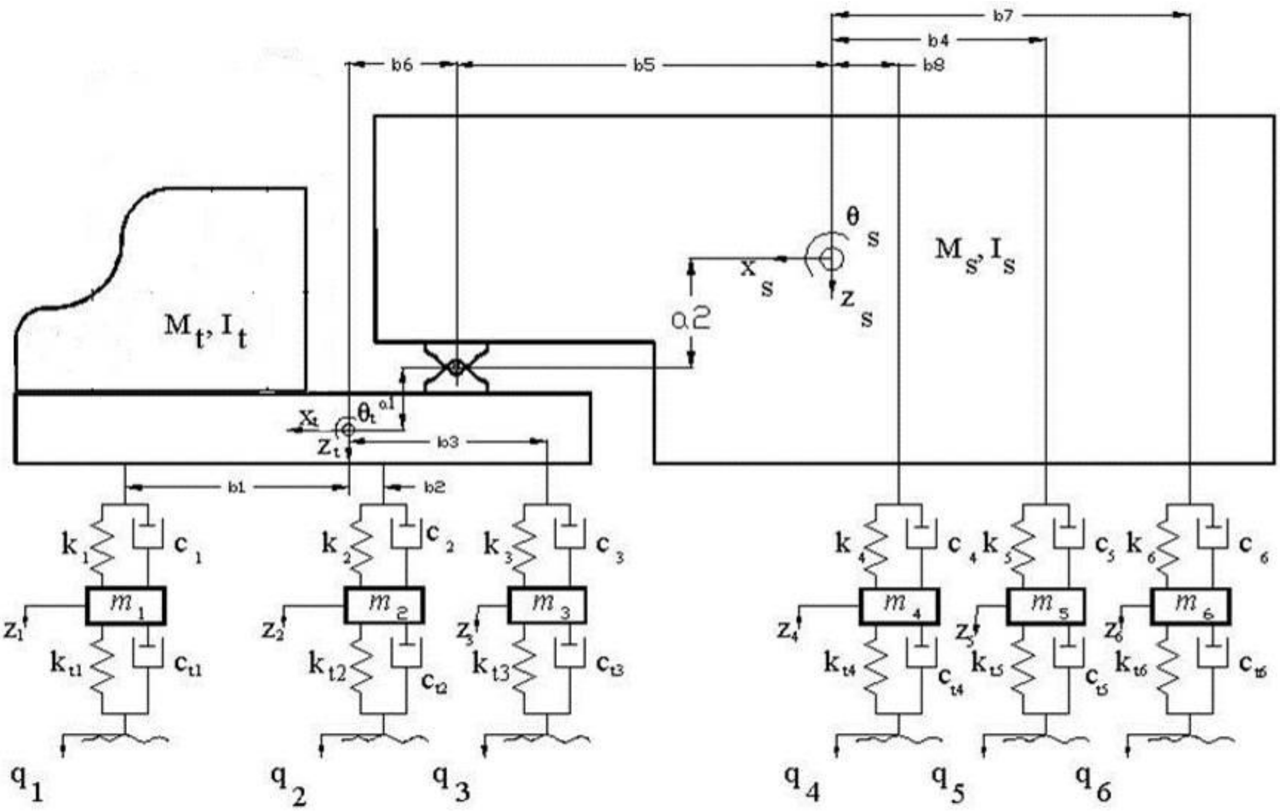

Fig. 1: Schematic representation of unsuspended cabin tractor semi-trailer vehicle model

The equation of motion of tractor body vertical and pitch motions are given by,

$$\begin{aligned}&(m_t + m_s)\ddot{y}_t + m_s b_6 \ddot{\theta}_t + m_s b_5 \ddot{\theta}_s + (c_1 + c_2 + c_3 + c_4 + c_5 + c_6)\dot{y}_t + ((-b_1 c_1) + b_2 c_2 + b_3 c_3 + b_6(c_4 + c_5 + c_6))\dot{\theta}_t \\ &+ ((b_5 + b_4)c_5 + (b_5 + b_7)c_6 + (b_5 + b_8)c_4)\dot{\theta}_s + (-c_1)\dot{y}_1 + (-c_2)\dot{y}_2 + (-c_3)\dot{y}_3 + (-c_4)\dot{y}_4 + (-c_5)\dot{y}_5 + (-c_6)\dot{y}_6 \\ &+ (k_1 + k_2 + k_3 + k_4 + k_5 + k_6)y_t + ((-b_1 k_1) + b_2 k_2 + b_3 k_3 + b_6(k_4 + k_5 + k_6))\theta_t + ((b_5 + b_4)k_5 + (b_5 + b_7)k_6 \\ &+ (b_5 + b_8)k_4)\theta_s + (-k_1)y_1 + (-k_2)y_2 + (-k_3)y_3 + (-k_4)y_4 + (-k_5)y_5 + (-k_6)y_6 = 0\end{aligned} \quad (7)$$

$$\begin{aligned}&(I_t + b_6^2 m_s h_1^2 a m_7)\ddot{\theta}_t + b_6 m_s \ddot{y}_t + (b_6 b_5 m_s + h_1 h_2 a m_7)\ddot{\theta}_s + (-b_1 c_1 + b_2 c_2 + b_3 c_3 + b_4 c_4 + b_5 c_5 + b_6 c_6)\dot{y}_t \\ &+ (-b_1^2 c_1 + b_2^2 c_2 + b_3^2 c_3 + b_4^2 c_4 + b_5^2 c_5 + b_6^2 c_6)\dot{y}_t + (b_6(b_5 + b_4)c_5 + b_6(b_5 + b_7)c_6 + b_6(b_5 + b_8)c_4)\dot{\theta}_s \\ &+ (b_1 c_1)\dot{y}_1 + (-b_2 c_2)\dot{y}_2 + (-b_3 c_3)\dot{y}_3 + (-b_4 c_4)\dot{y}_4 + (-b_5 c_5)\dot{y}_5 + (-b_6 c_6)\dot{y}_6 + (-b_1 k_1 + b_2 k_2 + b_3 k_3 - (b_9 + b_{11})k_7 \\ &- (b_9 + b_{10})k_8 + b_6 k_4 + b_6 k_5 + b_6 k_6)y_t + (b_1^2 k_1 + b_2^2 k_2 + b_3^2 k_3 + b_4^2 k_4 + b_5^2 k_5 + b_6^2 k_6)\theta_t + (b_6(b_5 + b_4)k_5 \\ &+ b_6(b_5 + b_7)k_6 + b_6(b_5 + b_8)k_4)\theta_s + (b_1 k_1)y_1 + (-b_2 k_2)y_2 + (-b_3 k_3)y_3 + (-b_4 k_4)y_4 + (-b_5 k_5)y_5 + (-b_6 k_6)y_6 = 0\end{aligned} \quad (8)$$

The equation of motion for semi trailer body pitch is given by,

$$\begin{aligned}&(I_S + b_5^2 m_s h_2^2 a m_7)\ddot{\theta}_S + b_5 m_s \ddot{y}_t + (b_6 b_5 m_s + h_1 h_2 a m_7)\ddot{\theta}_t + ((b_5 + b_8)c_4 + (b_5 + b_4)c_5 + b_6(b_5 + b_7)c_6)\dot{y}_t \\ &+ (b_6(b_5 + b_8)c_4 + b_6(b_5 + b_4)c_5 + (b_5 + b_7)c_6)\dot{\theta}_t + ((b_5 + b_8)^2 c_4 + (b_5 + b_4)^2 c_5 + (b_5 + b_7)^2 c_6)\dot{\theta}_s \\ &+ (-(b_5 + b_8)c_4)\dot{y}_4 + (-(b_5 + b_4)c_5)\dot{y}_5 + (-(b_5 + b_7)c_6)\dot{y}_6 + (b_5 + b_8)k_4 + (b_5 + b_4)k_5 + (b_5 + b_7)k_6)\dot{y}_t \\ &+ (b_6(b_5 + b_8)k_4 + b_6(b_5 + b_4)k_5 + b_6(b_5 + b_7)k_6)\theta_t + ((b_5 + b_8)^2 k_4 + (b_5 + b_4)^2 k_5 + (b_5 + b_7)^2 k_6)\dot{\theta}_s \\ &+ (-(b_5 + b_8)k_4)y_4 + (-(b_5 + b_4)k_5(y_5 + (-(b_5 + b_8)^2 k_6)y_6 = 0\end{aligned} \quad (9)$$

Tractor front axle unsprung mass vertical motion is given by,

$$m_1\ddot{y}_1 + (c_1 + c_{t1})\dot{y}_1 + (-c_1)\dot{y}_t + (b_1 c_1)\dot{\theta}_t + (k_1 + k_{t1})y_1 + (-k_1)y_t + (b_1 k_1)\dot{\theta}_t = c_{t1}\dot{q}_1 + k_{t1}q_1 \quad (10)$$

Tractor rear axles 1 and 2unsprung mass vertical motion are given by,

$$m_2\ddot{y}_2 + (c_2 + c_{t2})\dot{y}_2 + (-c_2)\dot{y}_t + (b_2 c_2)\dot{\theta}_t + (k_2 + k_{t2})y_2 + (-k_2)y_t + (b_2 k_2)\dot{\theta}_t = c_{t2}\dot{q}_2 + k_{t2}q_2 \quad (11)$$

$$m_3\ddot{y}_3 + (c_3 + c_{t3})\dot{y}_3 + (-c_3)\dot{y}_t + (b_3 c_3)\dot{\theta}_t + (k_3 + k_{t3})y_3 + (-k_3)y_t + (-k_3 b_3)\dot{\theta}_t = c_{t5}\dot{q}_3 + k_{t3}q_3 \quad (12)$$

Semi-trailer 1st, 2nd and 3rd axle unsprung mass vertical motion are given by,

$$\begin{aligned}&m_4\ddot{y}_4 + (c_4 + c_{t4})\dot{y}_4 + (-c_4)\dot{y}_t + (-b_4 c_4)\dot{\theta}_t + (-b_5)(-b_8)c_4\dot{\theta}_s + (k_4 + k_{t4})y_4 + (-k_4)y_t + (-k_4 b_6)\dot{\theta}_t \\ &+ (-b_5)(-b_8)k_4\theta_s = c_{t5}\dot{q}_4 + k_{t4}q_4\end{aligned} \quad (13)$$

$$\begin{aligned}&m_5\ddot{y}_5 + (c_5 + c_{t5})\dot{y}_5 + (-c_5)\dot{y}_t + (-b_5 c_5)\dot{\theta}_t + (-b_5)(-b_4)c_5\dot{\theta}_s + (k_5 + k_{t5})y_5 + (-k_5)y_t z + (-k_5 b_6)\dot{\theta}_t \\ &+ (-b_5)(-b_4)k_5\theta_s = c_{t5}\dot{q}_5 + k_{t5}q_5\end{aligned} \quad (14)$$

$$\begin{aligned}&m_6\ddot{y}_6 + (c_6 + c_{t6})\dot{y}_6 + (-c_6)\dot{y}_t + (-b_6 c_6)\dot{\theta}_t + (-b_5)(-b_7)c_6\dot{\theta}_s + (k_6 + k_{t6})y_6 + (-k_6)y_t + (-k_6 b_6)\dot{\theta}_t \\ &+ (-b_5)(-b_6)k_7\theta_s = c_{t6}\dot{q}_6 + k_{t6}q_6\end{aligned} \quad (15)$$

Table 1: Generalized coordinates of unsuspended cabin tractor semitrailer two dimensional vehicle model

Coordinates	Description of motion at	Symbol
bounce	Tractor vertical displacement	z_t
	Tractor front unsprung mass 1 vertical displacement	z_1
	Tractor rear unsprung mass 2 vertical displacement	z_2
	Tractor rear unsprung mass 3 vertical displacement	z_3
	Semi trailer unsprung mass 4 vertical displacement	z_4
	Semi trailer unsprung mass 5 vertical displacement	z_5
	Semi trailer unsprung mass 6 vertical displacement	z_6
pitch	Tractor pitch angle	θ_c
	Semi trailer pitch angle	θ_s

A description of the tractor semi-trailer model geometric parameters, inertial properties and suspension parameters can be found in Table 2, 3 and 4 respectively. The values have been collected from a number of different sources in an effort to create a model that accurately represents the intended test vehicle. It is assumed that the vehicle is symmetric about the longitudinal centreline of the tractor and trailer. Similarly, it is assumed that the left and right sides of the axles experience an identical road profile. These assumptions allow the left and right sides of the axles to be lumped into single masses and suspension elements.

Table 2: Geometric dimensions of the tractor semi-trailer model

Symbol	Description	Value
a_1	From the tractor CG to 5th wheel (vertical)	0.446 m
a_2	From the tractor CG to 5th wheel (vertical)	1.029 m
a_3	From the trailer CG to the cab CG (vertical)	0.538 m
b_1	From the tractor CG to axle #1	1.511 m
b_2	From the tractor CG to axle #2	1.642 m
b_3	From the tractor CG to axle #3	3.042 m
b_4	From the trailer CG to axle #5	3.91 m
b_5	From the trailer CG to 5th wheel (horizontal)	3.5 m
b_6	From the tractor CG to 5th wheel (horizontal)	1.991 m
b_7	From the trailer CG to axle #6	5.31 m
b_8	From the trailer CG to axle #4	2.51 m

Table 3: Inertial properties of the tractor semi-trailer model

Symbol	Description	Value
m_t	Mass of the tractor	3259 kg
I_t	Moment of inertia of the tractor	10113 kgm^2
m_s	Mass of the loaded semi-trailer	33400 kg
I_s	Moment of inertia of the semi-trailer	434000 kgm^2
m_1	Mass of the tractor front axle	426 kg
m_2	Mass of the tractor rear axle # 1	765 kg
m_3	Mass of the tractor front axle # 2	690 kg
m_4	Mass of the semi trailer rear axle # 1	690 kg
m_5	Mass of the semi trailer rear axle # 2	690 kg
m_6	Mass of the semi trailer rear axle # 3	690 kg

Table 4: Suspension parameters of the tractor semi-trailer model

Symbol	Description	Value
K_1	Steer axle spring coefficient	365150 N/m
K_2	#1 drive axle spring coefficient	1327350 N/m
K_3	#2 drive axle spring coefficient	1327350 N/m
K_4	#1 trailer axle spring coefficient	1327350 N/m
K_5	#2 trailer axle spring coefficient	1327350 N/m
K_6	#3 trailer axle spring coefficient	1327350 N/m
C_1	Steer axle damping coefficient	7168 Ns/m
K_{t1}	Steer axle tire stiffness	1660000 N/m
K_{t2}	#1 drive axle tire stiffness	4000000 N/m
K_{t3}	#2 drive axle tire stiffness	4000000 N/m
K_{t4}	#1 trailer axle tire stiffness	4000000 N/m
K_{t5}	#2 trailer axle tire stiffness	4000000 N/m
K_{t6}	#3 trailer axle tire stiffness	4000000 N/m
C_{t1}	#1 drive axle tire damping coefficient	700 Ns/m
C_{t2}	#2 drive axle tire damping coefficient	1200 Ns/m
C_{t3}	#1 trailer axle tire damping coefficient	1200 Ns/m
C_{t4}	#2 trailer axle tire damping coefficient	1200 Ns/m
C_{t5}	#3 trailer axle tire damping coefficient	1200 Ns/m
C_{t6}	#1 drive axle tire damping coefficient	1200 Ns/m

3. Results & discussions

A MATLAB simulation is created to investigate the effects of the various parameters of the model on driver ride comfort and pavement loading. For each element of the model, the vertical displacements have the positive direction defined as downward movement, and positive pitch rotations are defined as the front of the particular body moving up and the rear moving down. The road profile is an approximation to the vertical irregularities found on different types of roadways. The purpose of the ISO weighting factors is to assign greater importance to the frequencies which cause the driver to experience gross discomfort. These values in turn have a greater effect on the overall weighted RMS acceleration value (a_0). This value is calculated by ISO 2631-1 [13] as,

$$a_0 = \sqrt{(k_x\, a_{0\_L})^2 + (k_z\, a_{0\_v})^2} \tag{16}$$

Where k_x and k_z are the longitudinal and vertical acceleration frequency weighting respectively. a_{0_L} and a_{0_v} are the longitudinal and vertical weighted RMS acceleration respectively. When evaluating vehicle ride comfort, k_x =0 and k_z = 1. The overall weighted RMS acceleration value, a_0, can then be compared to the comfort ranges in ISO 2631-1.

The tractor semi-trailer simulations are allowed for any number of parameter configurations and model characteristics to be changed in whatever order desired. Based on time and frequency domain programs, properties can be altered and the effect of these properties has been closely studied on the system response. Table 5 presents the grand average z-axes cabin vertical vibration magnitudes measured on unsuspended cabin tractor semitrailers. This level of acceleration indicates an uncomfortable ride (1.185 m/s2) for unsuspended cabin tractor semitrailer.

Table 5: Cabin vertical acceleration values for unsuspended cabin tractor semi-trailer

Road condition	Load condition	Speed (km/h)	Simulated acceleration (m/s^2)
Paved road	Low load	40	1.269
		50	1.346
		60	1.518
	Medium load	40	1.260
		50	1.332
		60	1.421
	High load	40	1.250
		50	1.313
		60	1.407
Herringbone road	Low load	40	1.110
		50	1.259
		60	1.342
	Medium load	40	1.078
		50	1.179
		60	1.305
	High load	40	1.081
		50	1.184
		60	1.269
Smooth road	Low load	40	0.860
		50	1.040
		60	1.203
	Medium load	40	0.875
		50	0.964
		60	0.986
	High load	40	1.022
		50	1.058
		60	1.081
Grand average			1.185

The ride quality of a vehicle is strongly influenced by the type of road it traverses. Figs. 2 to 4 show the bounce acceleration spectra at driver-seat interface for paved road, herringbone road and smooth road and compared to ISO eight hours reduced comfort boundaries. The general shape of the response spectra for all types of roads is consistent. However, in the low frequency, the rate of increase of acceleration spectra for herringbone and smooth road is much higher than that for the paved road. There is an increase in the whole frequency range content of bounce vibration due to paved road profile. The bounce acceleration power spectral density for a paved road is about five times higher around the maximum peak value. The acceleration spectra for the smooth and paved road exceed the 8 hour ISO guide by a considerable margin. Thus, the ride vibration levels for the baseline vehicle are excessive in the bounce motions.

An increase in the vehicle speed from 40 km/hr to 60km/hr increases the bounce ride levels around the low frequency. Although the dominant peaks of bounce acceleration spectra are suppressed at lower vehicle speed of 40km/hr, there is a considerable increase in vibration levels in the frequency range, 3-5 Hz. For the same load condition, the variation in vehicle speed has little to no apparent effect on the cabin vertical vibration levels. A decrease in the load from 49 metric tonnes to 15 metric tonnes increases the bounce ride levels around the low frequencies. The fore-aft ride quality also improves by a considerable amount by increasing the load on the trailer. Although the dominant peaks of bounce as well as fore-aft acceleration spectra are suppressed to lower values by increasing the load from 15 to 40 metric tonnes, there is no apparent change in the frequency range of these dominant peaks.

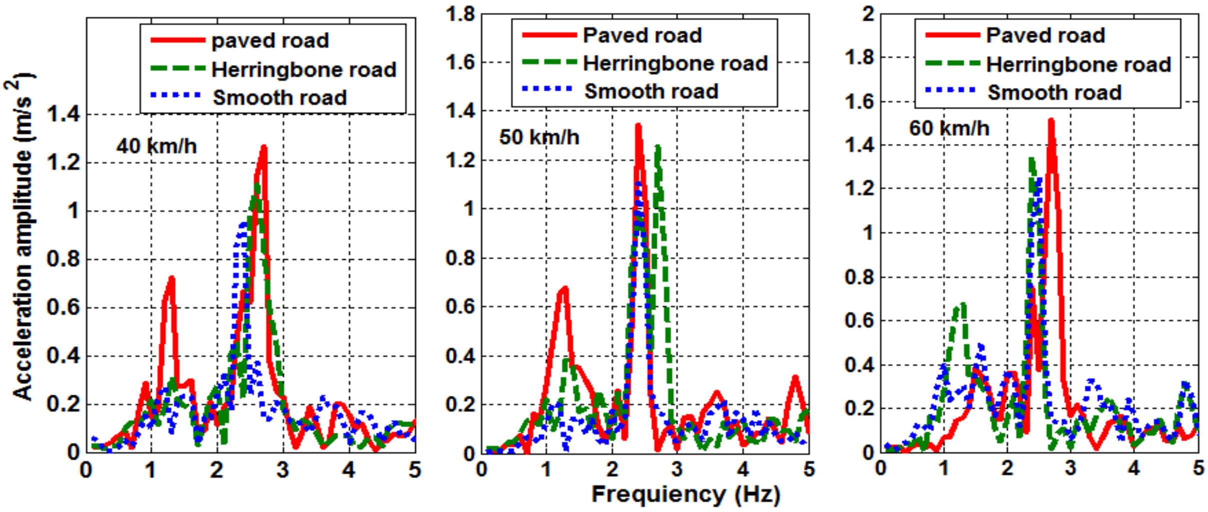

Fig. 2: Spectrum of the vertical acceleration measured on the cabin during low load (15 metric tonnes)

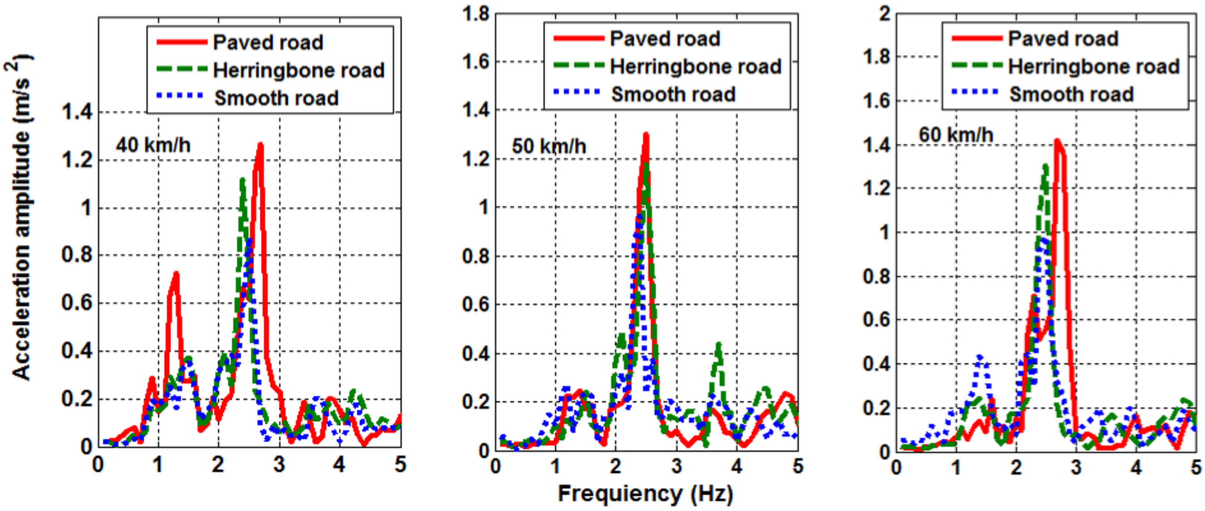

Fig. 3: Spectrum of the vertical acceleration measured on the cabin during medium load (25 metric tonnes)

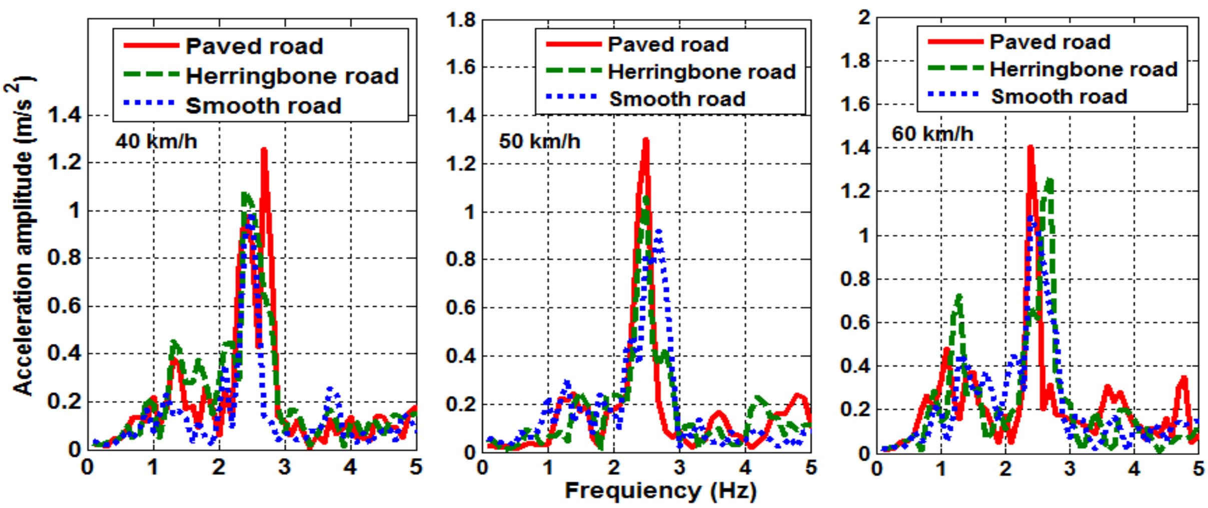

Fig. 4: Spectrum of the vertical acceleration measured on the cabin during high load (49 metric tonnes)

4. Conclusions

In this paper, the ride comfort analysis of a complex articulated vehicle has been studied using linear mathematical models subjected to real road input excitations. The analytical techniques employed to solve the mathematical models of baseline vehicle provide an attractive and convenient solution that yields a great deal of insight into the vehicle behaviour. The linear analysis of the articulated vehicle, in general is useful in studying the modal parameters of the vehicle such as system driver seat vibration and general effects on the ride quality due to generic changes in various vehicle parameters. A parametric study of linear vehicle model is carried out to establish the influence of various parameters on the articulated vehicle dynamic behaviour. The ride performance of the vehicle model is assessed with reference to ISO ride comfort criteria. Vehicle ride behaviour is considerably influenced by the road profile. Paved road surface deteriorates the vehicle ride quality in vertical direction. An increase in vehicle speed deteriorates the ride quality in vertical direction. A low loaded vehicle has a poor ride quality as compared to a high loaded vehicle. The average vertical acceleration level from simulation indicated a fairly uncomfortable ride for unsuspended cabin tractor semi-trailer.

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