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The Performance of Passive Cab Suspension Systems in Tractor-Semitrailer Vehicles

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Abstract. In this paper, the planar vibrational response of an articulated vehicle with a suspended cab is considered. The statistical performance of innovative linear passive suspensions in response to a randomly profiled road is evaluated. Design guidelines based upon the analytical model derived are provided. A procedure for optimizing driver comfort/cab deflection trade-off is presented. The trade-off is made explicit in the form of a performance index comprised of weighted mean squared driver acceleration and mean squared suspension dynamic deflection. In addition, the potential improvements offered by non-linear passive suspensions are investigated.

Nomenclature

A	system dynamics matrix
B	system input matrix
b_j	vehicle dimension
C	system input matrix
c_j	suspension damping rate
c_n	nonlinear damping coefficient
c_{tj}	tire damping rate
F_d, F_s	damping and spring forces, respectively.
h_j	vehicle dimension
I	unit matrix
I_c, I_s, I_t	cab, semitrailer and tractor pitch moments of inertia, respectively.
i	$\sqrt{-1}$
J	performance index
k_j	suspension spring rate
k_l	linear stiffness coefficient
K_n	nonlinear stiffness coefficient

K_{tj}	tire spring rate
km_j	mass of wheel-axle assembly
m_c, m_s, m_t	cab, semitrailer and tractor masses, respectively
n	spatial frequency of road roughness
n_0	reference spatial frequency ($n_0 = 1/2\pi$ c/m)
q	vector of correlated road inputs
q_j	road input at tire j
$S_x(\omega)$	state spectral density matrix
$s(n)$	spatial spectral density of road roughness
$s(n_0)$	mean square spatial spectral density of road roughness at the reference spatial frequency n_0
$s_q(\omega)$	temporal spectral density of road roughness
$s_q(\omega_o)$	mean square temporal spectral density of road roughness at the reference temporal frequency ω_0
t	time
τ_{jk}	time delay between inputs to the jth and kth tires
v	vehicle speed
x	state vector
$x(i\omega)$	system frequency response
x_c, x_s, x_t	longitudinal motion coordinates of the centers of gravity of the cab, semi-trailer and tractor, respectively.
y	cab deflection
y_j	vertical motion coordinates of vehicle wheel-axle assemblies
y_c, y_s, y_t	vertical motion coordinates of the centers of gravity of the cab, semi-trailer and tractor, respectively
$\theta_c, \theta_s, \theta_t$	rotational coordinates about the centers of gravity of the cab, semi-trailer and tractor, respectively.
∂	weighing factor
σ	expected average
ω	temporal frequency
ω_0	temporal reference frequency

1. Introduction

Reduction of cab vibration in on-highway tractors has been an important engineering objective. The vibrations induced in vehicle components as a result of riding over irregular road profiles are not only prejudicial to riding comfort but may also cause physiological damage and inefficient performance. This research describes work undertaken to alleviate, through the use of analytical methods and optimization techniques, some of the truck ride problems.

Analysis and ride comfort evaluation of truck response to road excitation have been the subject of considerable analytical work. The modelling techniques to truck ride problem have included lumped mass methods [1-5], finite element methods [6-8], and model analysis methods [9-11]. The solutions of the governing equations have included transfer function, spectral density approach [12-14], and covariance analysis for linear vehicle systems [15], whilst equivalent linearization approach [16] and simulation techniques [17, 18] for nonlinear vehicle systems. These analytical and simulation techniques have been very useful tools to evaluate the design alternatives at the design concept stage.

Several measures have been used to quantitatively describe vibration environment. Among these measures are: 1) acceleration power spectral densities compared with International Standards Organization (ISO) vibration tolerance criteria [19,20], 2) distribution of root mean square acceleration in one-third octave frequency bands weighted by ISO standards [20], 3) ISO "weighted sum" method (weighted root mean square acceleration) [21], 4) simple root mean square acceleration levels, and 5) absorbed power [22,23].

A number of studies have employed optimization techniques and various forms of optimization criteria in the design of vehicle suspension systems [11,17]. A review of various optimization criteria is included in reference [24]. These studies have recognized that good vibration isolation and small suspension deflection are contrary goals, and that the suspension design is essentially a trade-off between these competing goals. The trade-offs are made through the use of a performance index in formal optimization procedures or through the use of a trade-off diagrams in the analysis of suspension performance.

In this paper, the heave, fore-aft, and pitch motions of a tractor-semitrailer truck are considered, with primary emphasis on the evaluation of the statistical performance of the linear and nonlinear passive suspension systems in response to road irregularities. The main goal of this research is to develop an analytical model for cab suspension design, to evaluate suspension performance, and to provide design guidelines based upon the derived model.

A procedure for optimizing the comfort-deflection trade-off is presented herein. The trade-off is made explicit in the form of a performance index comprised of weighted mean squared acceleration and mean squared suspension dynamic deflection.

2. Heavy Trucks Dynamic Models

In order to predict the truck-road dynamic interaction and to describe its performance, several mathematical models varying in complexity have been developed. Because of the complexity of the problem, various assumptions regarding vehicle model and road representation have been made by the various researchers in order to model and simplify the system and to obtain a solution.

Ellis [25] used a three-degree-of-freedom vehicle model including tractor heave and pitch and semitrailer pitch. The model was excited by a harmonic function. Noon [26] considered the motion of the cab on the tractor chassis, and neglected the bounce and fore-aft motions of the tractor and trailer.

Walther *et al.* [27] used a model of seven-degree-of-freedom by including the bounce motions of the unsprung masses of the three wheel-axle assemblies. The fore-aft motions of the tractor and semitrailer were ignored and a stiff spring was substituted for the fifth wheel king pin. The road profile was assumed to consist of a flat surface with periodically spaced tar strips. Walker and Potts [18] employed a six degree-of-freedom model in which the fore-aft motions of the two units were considered, and the constraints caused by the fifth wheel in the vertical and fore-aft directions were made use of. The resulting degrees of freedom were tractor heave and pitch motions, semitrailer pitch motion, and the vertical motions of the three axles. The input to the vehicle system was sinusoidal excitation. Dokainish and ElMadany [3] applied the power spectral density approach to obtain the response of the six degree-of-freedom model to random road irregularities. The driver seat was modelled as a single degree of freedom system in reference [5].

Van Deusen [17] viewed the heavy truck-road system as a thirty degree-of-freedom system excited by single road profile spectral density. The model included the tractor frame flexibility which was modelled as twelve concentrated mass elements interconnected by beam equations, with each beam element being represented by two degrees of freedom. The two bogies of the tractor were modeled in three degrees of freedom each. The cab was represented by rigid body having two degrees of freedom on flexible cab mounts, while the driver and seat assembly was represented by a single degree of freedom. The semitrailer was modeled in five degrees of freedom, including vertical and pitching motions of the semitrailer body, the rotational motion of the walking beam assembly of its two front axles, and the vertical motions of the unsprung masses of its two rear axles.

Allen [2] investigated the limits of ride quality improvement with isolation systems only at the rear of the cab. A single degree of freedom representing cab pitching motion was used. The cab model accepted as inputs the frame vertical and longitudinal accelerations at the cab hinge and at the rear cab suspension pad. Recorded vibrations from a tractor frame were played into the simulated cab.

2.1. Nine-Degree-of-Freedom Model

The most desirable mathematical model of the vehicle is one which is not very complex, and which will be realistic enough to infer valid conclusions about the ride performance of the vehicle. With this objective the planar vehicle model shown in Fig. 1 is developed. The tractor chassis and semitrailer payload are represented by two planar rigid bodies interconnected at the fifth wheel. The cab is represented by a planar rigid body suspended on tractor frame. The masses of axles and wheels are represented by masses supported by the tires. The vehicle is modelled in fore aft, bounce and pitch for sprung masses and bounce only for the unsprung masses.

The vehicle motions are assumed to be small compared with its dimensions. This assumption indicates that all displacements and rotations are small and first order approximations are used. In addition, it is assumed that the constitutive relations for the vehicle suspensions are linear.

Each suspension for the tractor and semitrailer is composed of a spring and a damper. The tire is modelled as a spring in parallel with a viscous damper. The cab

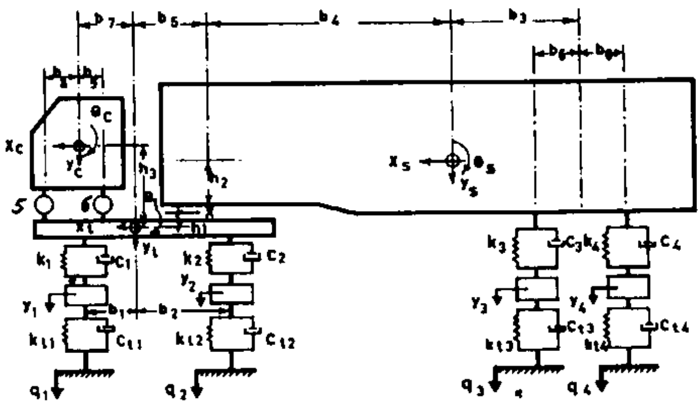

Fig. 1. Tractor-semitrailer truck model

is isolated from tractor chassis by different types of suspension systems comprised of springs and viscous dampers.

By considering the constraints imposed by the fifth wheel on the vertical and longitudinal motions of the tractor and semitrailer, the motions in the vertical plane give rise to nine degrees of freedom. The degrees of freedom of the vehicle model include cab and tractor vertical translations (y_c,y_t) cab, tractor and semitrailer pitch motions (θ_c, θ_t,θ_s), and vertical displacements of the four wheel-axle assemblies (y_1, y_2, y_3, y_4).

The main objective of the model developed herein is to study the effects of suspension system modifications on the vehicle ride quality. A frequency domain approach is implemented on digital computer to obtain vehicle accelerations. The frequency domain is preferred overtime domain since most ride quality specifications consist of limits upon the acceleration of the vehicle as a function of frequency.

3. Road Representation

The input excitation to the vehicle is assumed to be the apparent vertical roadway motion, caused by the vehicle forward speed along a road having an irregular profile. In order to apply spectral density approach to analyze and evaluate vehicle ride quality, a statistical representation of road irregularities is required. Measured data of road profiles indicate that the surface irregularities may be treated as a stationary Gaussian random process with zero mean and characterized by a single-sided power spectral density of the form [28].

$$s(n) = s(n_0)\,(n_0/n)^w, \tag{1}$$

where n is the spatial frequency, $s(n_0)$ is the vertical amplitude mean square spectral density at the reference spatial frequency $n_0 (n_0 = 1/2\pi$ c/m). The exponent w has two values, $w_1 = 2.0$ and $w_2 = 1.5$ for $n \leq n_0$ and $n > n_0$, respectively, $s(n_0)$ is taken to be 100×10^{-6} m^3/c.

A vehicle travelling at a constant forward speed v therefore sees an input of the form $s_q(\omega) = s_q(\omega_0)(\omega_0/\omega)^w$, where $\omega = 2\pi nv$ and $s_q(\omega_0) = s_q(n_0)/2\pi v$.

4. State Power Spectral Density Matrix

The system equations can be expressed in the form

$$\dot{x} = Ax + B\dot{q} + Cq \tag{2}$$

where x is the state vector, A is the vehicle system dynamics matrix, B and C are input matrices, q is the vector of correlated road irregularity inputs to the four tires, *i.e.*,

$$\begin{aligned} q^T(t) &= [q_1(t), q_2(t), q_3(t), q_4(t)] \\ &= [q_1(t), q_1(t\text{-}\tau_{12}), q(t\text{-}\tau_{13}), q_1(t-\tau_{14})] \end{aligned} \tag{3}$$

where τ_{jk} is the time delay between inputs to the j*th* and k*th* tires, and $q_1(t)$ is the input to the front tire.

Taking Fourier transform of equation (2) and using equation (3), the system response can be calculated from

$$X(i\omega) = [i\omega I - A]^{-1}[i\omega B + C] \begin{bmatrix} 1 \\ e^{-i\omega\tau_{12}} \\ e^{-i\omega\tau_{13}} \\ e^{-i\omega\tau_{14}} \end{bmatrix} q(i\omega) \tag{4}$$

The state spectral density matrix is given by

$$S_x(\omega) = [i\omega I - A]^{-1}[i\omega B + C] \begin{bmatrix} 1 \\ e^{-i\omega\tau_{12}} \\ e^{-i\omega\tau_{13}} \\ e^{-i\omega\tau_{14}} \end{bmatrix} S_q(\omega)\,[1\; e^{i\omega\tau_{12}}\, e^{i\omega\tau_{13}}\, e^{i\omega\tau_{14}}]\,[i\omega B + C]^T . \{[i\omega I - A]^{-1}\}^* \tag{5}$$

where * denotes complex conjugate transpose.

The diagonal elements of $S_x(\omega)$ representing the spectral densities of the states are real. The state cross-spectral densities in general are complex.

5. Statistical Performance Measures

In order to evaluate the performance of the different suspensions and to design an effective cab suspension system, some specific measures of that performance are needed. For the suspension types under consideration, the indicators of suspension performance are chosen to be cab acceleration levels (acceleration levels at the attachment point of the seat to the cab) as a measure of ride quality or ability of the suspension to isolate the cab from road disturbances, and the suspension excursion (suspension relative dynamic deflection) as a measure of stroke length needed.

The quality of ride offered by a vehicle is rather subjective. However, several attempts have been made to specify a vehicle ride quality criterion. Much work has been done to relate the various sensations experienced by a person undergoing wholebody vibration, to the frequency and level of that vibration. In this study, the ISO recommended frequencies weighting technique is used to produce a single number representation of ride. The acceleration spectrum to be assessed is modified throughout the frequency range using a weighting function. By integrating the resulting spectrum the weighted mean squared acceleration can be obtained. The freqeuncy range over which the integration is performed in this study is from 0.1 to 30 Hz.

To estimate the random vertical displacement of the suspension due to road irregularities, the root mean squared displacement is used.

6. Evaluation of the Suspension Alternatives

In this section an analysis is sought to improve the ride quality at the cab by redesigning the cab suspension systems. The statistical performance of the suspension is evaluated for a vehicle operated along a road whose irregularities are represented by the power spectral density function given by equation (1). For ride quality assessment of the vehicle, the acceleration spectra and weighted rms accelerations are used. The study is for a vehicle operating at 80 km/h, and the baseline values of the vehicle are listed in Table 1.

Table 1. Values of baseline vehicle parameters

Masses	Dimensions
m_c = 950 kg	b_1 = 0.65 m
m_s = 24800 kg	b_2 = 2.8 m
m_t = 3450 kg	b_3 = 3.68 m
m_1 = 803 kg	b_4 = 5.62
m_2 = 1503 kg	b_5 = 2.2 m
m_3 = 1503 kg	b_6 = 0.7 m
m_4 = 1503 kg	b_7 = 0.9 m
I_c = 800 kg.m^2	b_8 = 0.85 m
I_s = 200000 kg.m^2	b_9 = 1.15 m
I_t = 9500 kg.m^2	h_1 = 0.3 m
	h_2 = 1.0 m
	h_3 = 1.0 m

Suspension characteristics

k_1 = 400 kN/m	c_1 = 33 kNs/m
k_2 = 700 kN/m	c_2 = 58.3 kNs/m
k_3 = 1000 kN/m	c_3 83.3 kNs/m
k_4 = 1000 kN/m	c_4 = 83.3 kNs/m

Table 1 (continued)

Masses	Dimensions
Tire characteristics	
k_{t1} = 2000 kN/m	c_{t1} = 0.7 kNs/m
k_{t2} = 4000 kN/m	c_{t2} = 1.2 kNs/m
k_{t3} = 4000 kN/m	c_{t3} = 1.2 kNs/m
k_{t4} = 4000 kN/m	c_{t4} = 1.2 kNs/m

Three suspension systems are evaluated: the conventional, the elastically-isolated conventional (EIC), and the conventional with elastically connected damper (ECD). Figure 2 illustrates the different suspension types. Table 2 gives description of the different suspension types, and the corresponding definition of natural frequency and damping ratio.

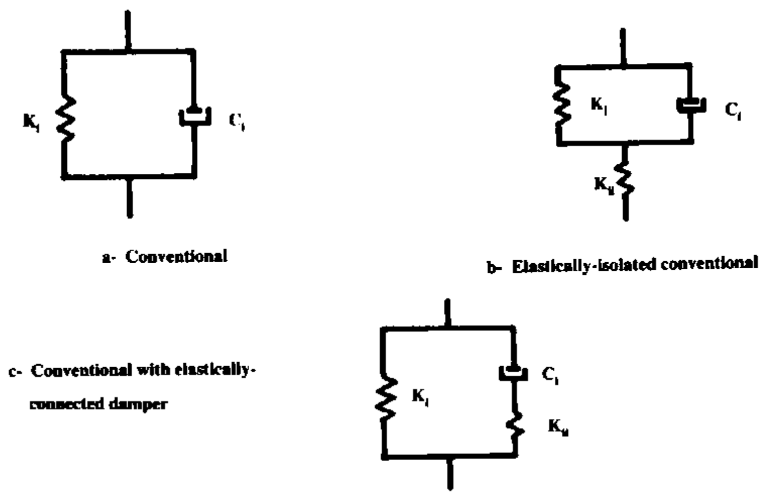

Fig. 2. Suspension types

Table 2. Description of the different suspension types

Suspension type	Description	Natural frequency, ω_n	Damping rate, ξ
1- Conventional	Second-order system. A spring and viscous damper connected in parallel.	$\sqrt{\dfrac{\Sigma k_j}{m_c}}$	$\dfrac{\Sigma c_j}{2\sqrt{m_c \Sigma k_j}}$
2-Elastically isolated conventional (EIC)	Third-order system. A conventional suspension system in series with a spring.	$\sqrt{\dfrac{k_e}{m_c}}$ $\left(k_e = \Sigma \dfrac{K_j K_{jj}}{(K_j + K_{jj})}\right)$	$\dfrac{\Sigma c_j}{2\sqrt{m_c k_e}}$
3- Conventional with elastically connected damper (ECD)	Third-order system. A spring connected in parallel with elastically-connected damper.	$\sqrt{\dfrac{\Sigma k_j}{m_c}}$	$\dfrac{\Sigma c_j}{2\sqrt{m_c \Sigma k_j}}$

Figures 3 and 4 show the ride quality/dynamic deflection trade-offs for the different suspension types. The weighted rms vertical and longitudinal accelerations versus average cab suspension dynamic deflection are given in Figures 3 and 4, respectively. The damping ratio is varied between 0.1 and 0.8. The data presented in the figures permit an evaluation of the different suspension types and the effectiveness of damping in reducing vehicle accelerations.

Several conclusions can be drawn from the results presented in Figures 3 and 4. The conventional suspension system gives better results than both the elastically-isolated conventional and the conventional with elastically connected damper. This indicates that most of the rms acceleration contributions are low frequency. Increasing the stiffness ratio in the elastically isolated conventional system will result in reduction in the acceleration levels, and at the end will approach the performance of the conventional system. A reduction in weighted rms vertical accelerations is achieved by increasing suspension damping until a minimum is reached, after which a further increase in damping impairs vertical ride quality. However, further decrease in the acceleration may be obtained only by decreasing the cab suspension spring rates. The dynamic deflection can be significantly improved by increasing the damping while leaving the stiffness at its baseline level. A trade-off exists between the level of rms cab vertical accelerations and suspension dynamic deflections. Thus as the vertical acceleration levels are reduced, larger dynamic deflections occur. Suspension design, therefore, represents a trade-off between achieving low dynamic deflections and good ride quality represented by low acceleration levels.

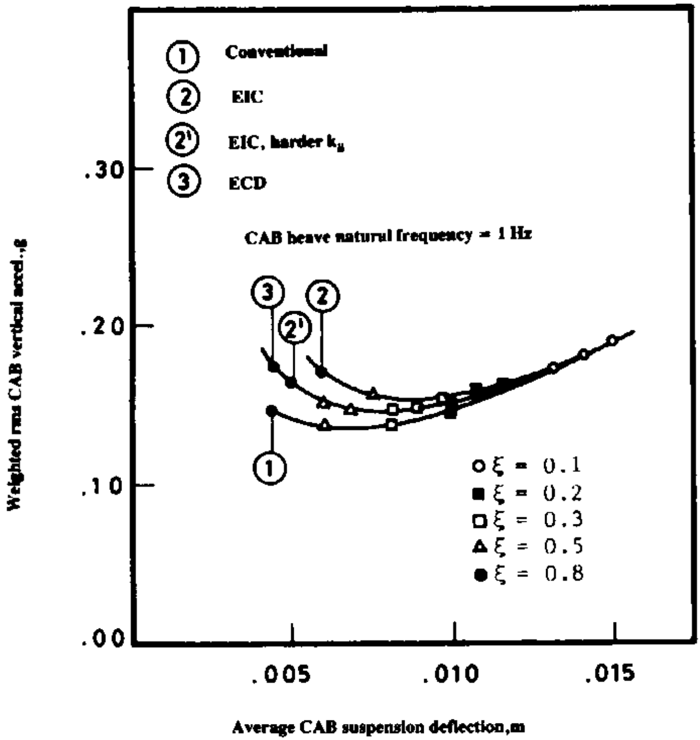

Fig. 3. Weighted rms CAB vertical acceleration versus average cab suspension dynamic deflection

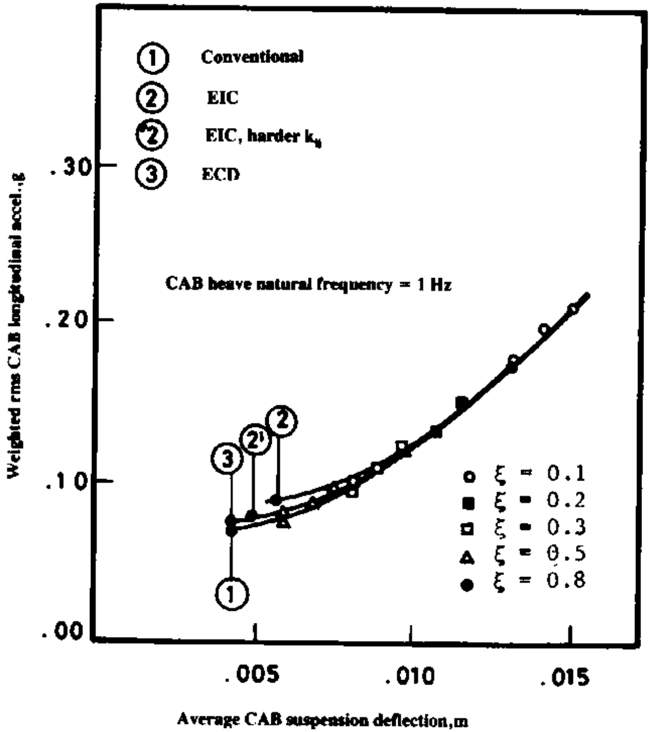

Fig. 4. Weighted rms cab longitudinal acceleration versus average cab suspension dynamic deflection

7. Vehicle Suspension Dynamic Optimization

The cab suspension dynamic response optimization technique is presented in this section. The dynamic performance measures include weighted mean squared acceleration in the vertical direction and mean squared suspension deflection. The design parameters are the cab suspension springing and damping characteristics.

To measure the overall suspension performance for suspension design purposes, the following performance index is used.

$$J = \sigma^2_{\ddot{y}_d} + \varrho\sigma^2_y \tag{6}$$

Where
σ denotes the expected average. This performance index is a quadratic functional.

$\sigma^2_{\ddot{y}_d}$ is the weighted mean squared vertical acceleration at the driver position; a measure of vehicle riding comfort.

σ^2_y is the average mean squared values of the cab front and rear suspension dynamic deflections.

The value of the weighting parameter ϱ may be taken arbitrarily. The choice for the value of ∂ sets the relative importance placed upon the two functions in equation (6). A trade-off curve is obtained through the determination of the value of ϱ.

Many mathematical programming techniques, for solving optimization problems, have been developed and evaluated for engineering optimization [29]. Optimization of the vehicle suspension considered herein is carried out using Hooke and Jeeves pattern search algorithm [30] which required function evaluation but not gradient information. It is an iterative algorithm which varies the system parameters in a search to minimize the performance index.

The computer optimization technique is applied on the three types of cab isolation systems of conventional, elastically isolated conventional and conventional with elastically connected damper. Two values of the weighting parameters ϱ (10^4 and 10^6) are used for each type. The performance index is minimized in each case and the optimum values of the cab springing and damping parameters are calculated. Table 3 gives the weighted rms values of the cab vertical and longitudinal accelerations, the cab front and rear deflections, and the optimized suspension parameters.

From examination of the data in Table 3, it can be stated that with large penalty on cab dynamic deflection, the deflection is reduced but at the expense of increasing cab acceleration in both the vertical and longitudinal directions. The conventional suspension offers the best results as regard to the riding comfort based on using the same weighting factor. The optimum values of the springing parameters, k_{55} and k_{66}, for the elastically isolated conventional and conventional with elastically connected damper are very large and in effect the two types of suspension approach a conventional one with a spring and damper in parallel.

The acceleration power spectra at driver's location in the vertical and longitudinal directions for $\varrho = 10^6$ are shown in Figures 5 and 6, respectively. The results are given for the three types of the linear passive suspensions. ISO for 1 and 8 hours fatigue decreased proficiency are shown in the figures for comparison. It can be seen that the vehicle system dynamics are characterized by several rigid body modes: bounce, pitch and wheelhop. The response spectra peak at the corresponding

Table 3. Optimum results

Suspension type	Weighting factor, ϱ	Weighted cab acceleration, g		Cab Deflection, m ($\times 10^{2}$)		Optimum spring rates, kN/m				Optimum damping rates, kNs/m	
		vertical	longitudinal	front	rear	k_5	k_6	k_{55}	k_{66}	c_5	c_6
Conventional	10^4	0.122	0.061	0.714	0.714	60	57.0			7.72	4.75
	10^6	0.146	0.078	0.537	0.462	116	69.6			9.63	5.8
Elastically isolated conventional	10^4	0.153	0.073	0.621	0.319	100	91.2	1720	1032	8.3	8.6
	10^6	0.188	0.080	0.374	0.307	172	103.2	1720	780	14.3	8.6
Conventional with elastically connected damper	10^4	0.126	0.061	0.810	0.430	58	80.4	1420	852	7.14	7.1
	10^6	0.180	0.082	0.436	0.356	172	103.0	1720	1032	14.3	8.6

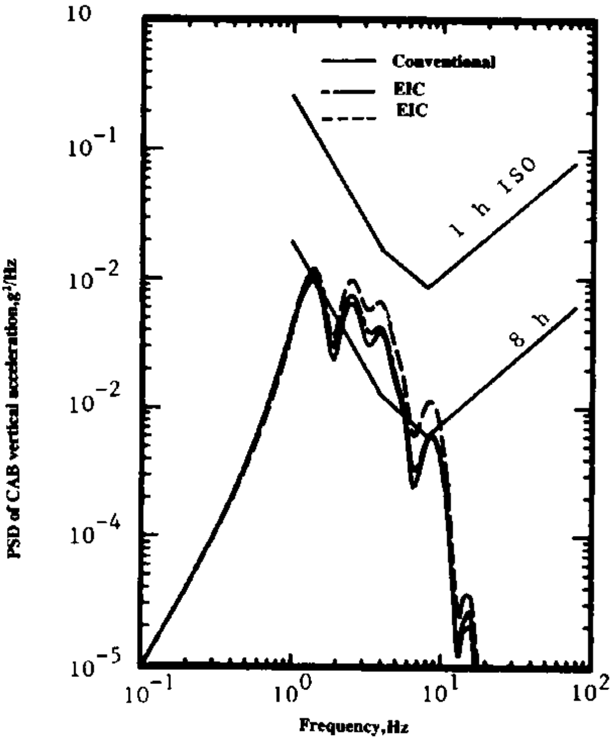

Fig. 5. Cab vertical acceleration powr spectra for the optimized suspensions with $\rho = 10^4$

frequencies. The pitching modes of the three units (cab, tractor and semitrailer) form a wide peak in the 1.3 to 4 Hz frequency band and contribute significantly to the vertical and longitudinal accelerations. The response spectra exceed the 8 hour ISO specification. Among the three passive suspension systems considered, the conventional one provides the best response and proves to be less fatigue causing particularly for the frequency range above 2 Hz. It can be concluded that the ride quality even for the optimized passive linear suspension is unacceptable. This shows clearly the limitation of the passive suspension systems in providing a good ride quality for such a particular combination of vehicle configurations and road characteristics.

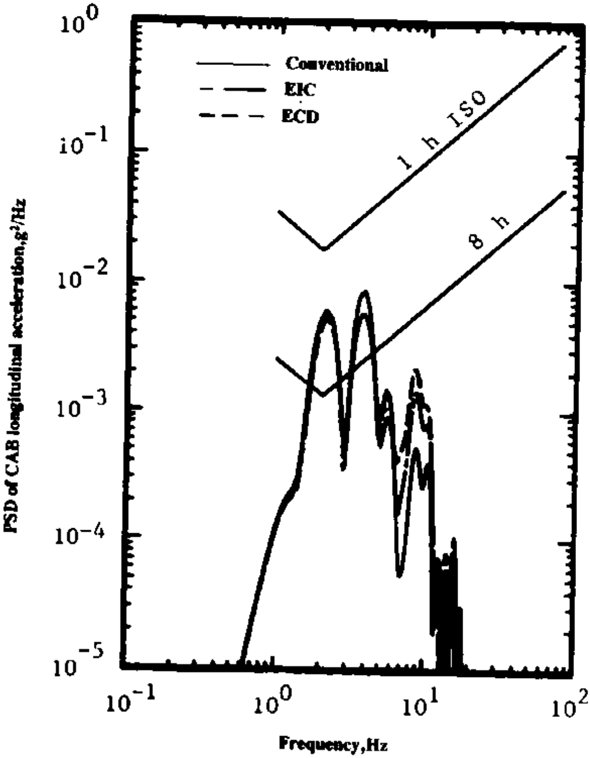

Fig. 6. Cab longitudinal acceleration power spectra for the optimized suspensions with $\rho = 10^4$

8. Performance of Cab Ride with Nonlinear Suspensions

In this section the improvement in cab ride quality through the use of nonlinear springing and damping characteristics is examined. The cab suspension system consists of springs in parallel with dampers. The modification to the suspension includes the use of linear-square and linear-cube type for the nonlinear springing characteristics, and square law type for the dampers. The stiffness and damping coefficients for the nonlinear suspension are determined by setting the force balance for the nonlinear suspension equal to the peak force obtained from the use of the optimal conventional suspension with $\varrho = 10^6$.

The linear-square and linear-cube system models for stiffness are given by the following force-displacement relationships:

$$F_s = k_\ell\, y + k_n\, y|y| \tag{7}$$

$$F_s = k_\ell\, y + k_n y^3 \tag{8}$$

where y is the relative motion between the cab and tractor frame at the suspension connecting point.

The force-velocity relationship for the square law damper is given by

$$F_d = c_n\, \dot{y}|\,\dot{y}| \tag{9}$$

The solution of the nonlinear stochastic differential equations describing the motion of the vehicle system is obtained through the use of the equivalent linearization technique [16] in which a nonlinear characteristic is replaced with an equivalent linear model.

The effects on cab performance of the variations in the stiffness coefficients of the nonlinear suspension systems are studied. Typical resutls are shown in Table 4 which are obtained using the optimum damping parameters for the conventional suspensions. It may be inferred from Table 4 that by using higher values for the linear part of the stiffness, the equivalent stiffness increases. This leads to an increase in the vertical and longitudinal accelerations, accompanied by a decrease in the suspension dynamic deflections. The rms acceleration levels are lower for the nonlinear suspensions (linear-square and linear-cube) than the corresponding linear ones, leading to improvement in ride quality. Comparing the results of the linear-cube with that of the linear-square type, the linear-cube system shows lower acceleration levels but higher suspension deflections. This results from the fact that the equivalent stiffness for the linear-cube system is lower than that of the comparable linear-square system. It can be said that improvement in the cab ride quality can be achieved by using nonlinear springs for cab suspensions.

Table 4. Results using nonlinear springing characteristics

Suspension type	Linear stiffness coefficient	Weighted cab acceleration, g		Cab deflection, m (x 10^2)		Equivalent stiffness, kN/m	
		Vertical	Longitudinal	Front	Rear	Front	Rear
Linear-square	0	0.130	0.069	0.596	0.536	68.5	42.7
	1/4 k^*_{op}	0.134	0.071	0.572	0.511	79.2	48.5
	1/2 k_{op}	0.138	0.073	0.568	0.504	90.8	54.8
	3/4 k_{op}	0.141	0.075	0.552	0.486	108.0	62.2
	1 k_{op}	0.146	0.078	0.537	0.462	116.0	69.6
Linear-cube	0	0.125	0.065	0.615	0.556	50.6	33.1
	1/4 k_{op}	0.129	0.068	0.599	0.534	65.2	41.2
	1/2 k_{op}	0.134	0.071	0.581	0.519	80.8	49.4
	3/4 k_{op}	0.140	0.074	0.560	0.493	97.5	58.7

*Optimum values of spring rates calculated for $\varrho = 10^6$

The results of using square law damper in cab suspension system together with linear springing characteristics are presented in Table 5. It can be seen that the rms acceleration levels and the relative dynamic deflections are larger than those obtained with the linear viscous dampers. In other words, the nonlinear square law damper provides less effective cab vibration control than the linear viscous damper for such vehicle system. This behaviour may be justified since the equivalent damping is lower than that of the linear viscous damper.

Table 5. Results using square law dampers

Weighted cab acceleration, g		Cab deflection, m (x 10^2)		Equivalent damping, kNs/m	
Vertical	Longitudinal	Front	Rear	Front	Rear
0.148	0.092	0.713	0.621	6.1	3.3

9. Conclusions

The dynamic performance capabilities of passive suspension systems in isolating the cab from the random road irregularities are evaluated. A tractor-semitrailer model incorporating different suspension systems in search for better riding quality at the cab is presented. Among the linear suspension alternatives, the conventional system provides the best ride quality/deflection trade-off. For the combination of vehicle system and road characteristics studied here, the optimized acceleration power spectra exceed the ISO fatigue decreased proficiency standards due to the imposed limitation on the cab dynamic deflection (stroke). Modest improvement in performance over the linear system can be obtained by the use of nonlinear springs in the cab isolation system. The square law damper is inferior to the ride quality compared with the linear viscous damper.

Recommended areas for future investigations include the incorporation of active suspension system for improving cab performance, and development of more complex models (*e.g.* frame flexibility).

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أداء أنظمة معلقات سلبية لكابينة القيادة الخاصة بعربات الطرق ذات الجرار والمقطورة

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ملخص البحث. في هذا البحث تمت دراسة استجابة الاهتزازات لعربة مفصلية لها كابينة قيادة معلقة، كما تم تقييم الأداء الإحصائي لمعلقات خطية سلبية مستحدثة وذلك نتيجة سيرها على طريق ذي سطح تعرجي عشوائي ويسرد هذا البحث الخطوط العريضة لتصميم المعلقات. وكذلك يشرح الأسلوب الأمثل لتصميم المعلقات على أساس دليل أداء مكون من متوسط مربع عجلة السائق ومتوسط مربع الإزاحة الديناميكية للمعلقات. وبالإضافة إلى ذلك فقد تم بحث التحسينات التي يمكن تحقيقها عن طريق استخدام معلقات سلبية غير خطية.