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Dynamic Load Coefficient of Tyre Forces from Truck Axles

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Abstract: This study aims to predict the Dynamic Load Coefficient (DLC) of tyre forces from truck axles. Dynamic Load Coefficient is frequently used to characterise the dynamic loads generated by axles. It is a simple measurement of the dynamic variation magnitude of the axle load, for a specific combination of road roughness and speed. Under normal operating conditions, the DLC's value is typically in the range of 0.05-0.3, and close to zero when the truck's wheels are moving over a perfectly smooth road. To achieve the objectives of this study, which is to determine the DLC's value for seven different types of axles, a simple validated quarter-truck model was excited by a random road surface profile, in order to simulate a vehicle-road interaction. Points are equally spaced along the simulated road to generate dynamic loadings over a broad range of truck speeds. Multiple trucks gross-weight conditions were used to present realistic traffic behaviour. The results showed that irregular road profiles, exciting the vehicle as it travelled, caused continually changing tyre forces. Also, dynamic loading was seen to be fundamentally influenced by the type of suspension (i.e., air and steel), loading condition, and vehicle speed. For example, the DLC value of the tyre forces of the quarter-truck fitted with a steel suspension was found to be more than twice that of the truck fitted with an air suspension. Tyre forces of the one-third laden truck were more aggressive than any other loading condition, due to the uncertain body-bounce generated by the truck, which was strongly dependent on surface irregularities. At low speed, the DLC was greatly decreased if the load was increased. Furthermore, DLC value was always lower for trucks with air suspension over steel suspension, for the same load and vehicle speed. However, air suspension efficiency was clearly better for higher axle loads.

Introduction

Several factors influence the dynamic response of axle tyres, including vehicle speed, road irregularities, suspension type, loading conditions, etc. Vehicle speed has a very close relation with the roughness level of the road. The dynamic behaviour of a heavy vehicle axle fitted with wheel mounted instrumentation showed that dynamic loading patterns on road surfaces are very similar for the same vehicle at the same speed [1]. At low vehicle speeds, the vibration is slight, but at high vehicle speeds, the motion leads to significant vehicle vibrations, which not only reduce the ride quality of the occupants, but also generate additional inertia forces. The increase in dynamic load with speed is compensated for by the shorter duration of an applied axle load at an increased speed. Therefore, rutting may be diminished by the decreasing loading time at high speed. Also, as agreed by Cebon, road materials and structural responses are sensitive to vehicle speed and to the frequency content of the applied loads [2].

Axle loads applied at lower speeds can caused higher damage than loads applied at higher speeds; particularly on asphalt concrete pavement and at higher speeds, which produce higher dynamic loads. As speed increases, the peak strain (under a constant moving load) diminishes in amplitude, and occurs behind the point of load application. He also noted that the dynamic tyre forces applied to road surfaces by heavy vehicles caused a premature road failure. Moreover, Root Mean Square (*RMS*) dynamic wheel loads for various suspensions, tyres, and operating conditions, generally have broadly similar conclusions about the effect of suspension and tyre types on dynamic wheel loads

[3-7]. According to Sun and Kennedy, in their research investigating the quantitative influence of surface roughness, speed, and vehicle parameters, vehicle speed was shown to have a significant effect on the power spectral density of loads, which got worse for rougher surfaces [7]. Rough pavement, with a Pavement Serviceability Index (*PSI*) of 2.5 (note: smooth roads have a PSI above 4) experienced damage approximately 50% more than that of a smooth road, for most typical truck suspensions [9]. Over the typical range of International Roughness Index (*IRI*), of about 126 to 380 cm/km, the dynamic load coefficient will vary by a factor of 3, and relative damage will increase by 20% of Equivalent Single Axle Loads (*ESALs*). Soft suspension springs and tyres of a low vertical stiffness are desirable for minimising dynamic loads [3,10, and11]. For very low tyre stiffness's, low frequency force components are possible; and hence, the root mean square of force levels increase. The optimal level of viscous damping usually depends on suspension conditions and any dry friction in the suspension usually increases dynamic tyre forces. However, it is usually better to have too much suspension damping than too little [11].

According to Hahn, modern single leaf parabolic suspension, with good hydraulic damping, was reported to be 'not significantly worse' than stiff air suspensions. Meanwhile, triaxial suspensions were found to be better than tandem suspensions [12]. The theoretical increase in damage from the dynamic load of three tandem suspensions were compared with the damage caused by static wheel loads alone. The damage from torsion-bar, 4-spring, and walking beam, were 19, 22, and 37%, respectively [13]. Moreover, more dynamically active trucks, particularly those with walking beam tandem suspensions, will be more damaging on low strength pavements with a high roughness. Rubber walking beam suspension caused 17 to 22% additional theoretical damage, because dynamic and air suspensions contributed an additional 6 to 8% [14]. This is in agreement with other studies, that centrally-pivoted tandem drive axle suspension, as well as walking-beam and single point suspensions, were always found to generate the highest dynamic loads, because of their lightly damped pitching modes of around 8 to 10Hz. Furthermore, four-spring tandem suspensions were generally found to generate smaller dynamic loads than walking beams. Torsion bar and air suspensions generated the lowest loads [3, 4, 5, 6, and 15]. Additionally, more damage was reported when loads were applied to the road surface by a heavy vehicle fleet [9].

Literature Review

Dynamic Load Coefficient (DLC)

Dynamic Load Coefficient is frequently used to characterise dynamic loading. It is a simple measurement of the magnitude of the dynamic variation of the axle load for a specific combination ofroad roughness and speed. *DLC* is given as a ratio of the standard deviation of the dynamic load fluctuation over the static load, as follows:

$$DLC = \frac{\sigma}{F} \quad (2)$$

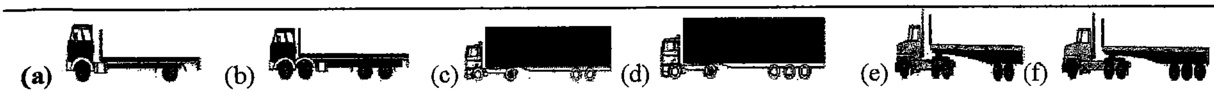
Where, σ is the standard deviation of the dynamic wheel load, F is the average or nominal static wheel load.The *DLC's* value is typically in the range of 0.05-0.3 under normal operating conditions and close to zero when the wheel axle of a truck is moving over a perfectly smooth road [9 and 16].

DLC values of less than 8% indicate moderately smooth pavements, greater than 10% is considered to indicate moderately rough pavement, and higher than 15% indicates very rough pavement surfaces [9]. A suspension with a natural frequency of less than 2Hz and a damping ratio of 0.2 or more is considered to be 'road friendly'. *DLC* values of different types of suspension at various vehicle speeds are shown in [4]. According to his study, *DLCs* for air, leaf spring, and walking beam suspensions, are from 0.03-0.1, 0.04-0.13, and 0.04-0.25, respectively; for smooth to rough road surfaces at vehicle speeds of 40 to 80km/h, with higher *DLCs* for higher vehicle speeds. *DLCs* are always lower for air suspension than steel suspension, for the same unevenness, loads, and vehicle speeds. However, the efficiency of air suspension is clearly better for high axle loads. At low speed, the *DLC* greatly decreases if the load increases. Meanwhile, *DLCs* are independent of loads at high speed, due to the adaptive suspension, which was designed to be efficient at high loads. A *DLC* value up to 0.4 is classified as a poor tandem suspension. At this level of dynamic loading, the axles spend a significant proportion of their time out of contact with the road's surface.

The dynamic tyre forces generated by vehicles are generally observed to be broad-band and close to Gaussian. The factors influencing the vertical dynamic include the mass and stiffness distribution of the vehicle's structure, payload mass distribution, suspension and tyres, road surface's longitudinal profile, and the speed of the vehicle [17]. The measured peak dynamic loads usually exceed the Root Mean Square (*RMS*) levels by a factor of about three. Dynamic tyre forces typically occur in the frequency range 1 to 15Hz, due to roughness features, with wavelengths of 0.5 to 15m, depending on speed [2]. The two dominant frequencies are body bounce and axle hops. The dynamic wheel forces generated by heavy vehicles fall into two distinct frequency ranges; (i) 1.5 to 4Hz for sprung mass bounce, pitch, and roll vibration mode, and; (ii) 8 to 15Hz for un-sprung mass bounce and roll, 'load sharing' suspension pitch modes [9]. These modes of vibration are excited by roughness irregularities with wavelengths of 6.9 to18.5m and 1.9 to 3.5m, respectively, at a vehicle velocity of 100km/h

Simulation Work

This section aims to predict the Dynamic Load Coefficient (*DLC*) of tyre forces from all types of truck axles, registered in Malaysia (as shown in Figure 1). All of the factors that are required to simulate tyre forces will be explained in the following sections.



(a) Rigid vehicle with 2 axles (1-1), (b)Rigid vehicle with 4 axles (1-1-2), (c) Articulated vehicle with 4 axles (1-1-2), (d) Articulated vehicle with 5 axles (1-1-3), (e) Articulated vehicle with 5 axles (1-2-2), (f) Articulated vehicle with 6 axles (1-2-3)

Figure 1. Types of HGV, according to their axle count

Quarter-truck model

Fluctuating loads are known as the dynamic tyre forces that are caused by the vehicle bouncing and pitching on the combined stiffness of the tyres and the suspension (i.e., body-bounce mode). In determination of the vertical forces, the wheel must be acting against a flat plane (i.e., the load cell must be recessed into the effective floor plane), or the suspension and tyre stiffness will combined with the wheel deflection to distort the results.

Therefore, the vehicle was a simplified model. Three kinds of vehicle models are commonly used for this purpose, namely quarter-truck, half-truck, or and full-truck. The quarter-truck model has been adopted by many previous researches [8, 20, 22, 23,and 24]. The simple quarter-truck model is generally used to evaluate vehicle loading. Indeed, it was identified with a personal computer to efficiently be used to predict pavement loading [20, 25, 26, and 27].It does not include detailed suspension nonlinearities and the complexities of body mass motions that are typical associated with heavy vehicles; however, the frequency content of the dynamic loads are sufficiently realistic for the purpose of studying pavement response. This point of view was verified in the work of Hardy and Cebon, who examined the importance of structural dynamics in the primary response of a flexible pavement to fluctuating, moving wheel loads, by means of a quarter vehicle model [20]. Besides, half-truck models are very complex, and thus require a detailed input. They also require longer execution times; even for simple problems. Hence, it is not popular amongst researchers.

For this purpose, a quarter-truck model with two Degrees of Freedom (*DOF*) to represent the bounce (vertical) motion of the vehicle was used (as shown in Figure 2). The figure shows a two degree of freedom truck model, with a simple suspension system, moving at a constant speed, *V* along a rough surface.

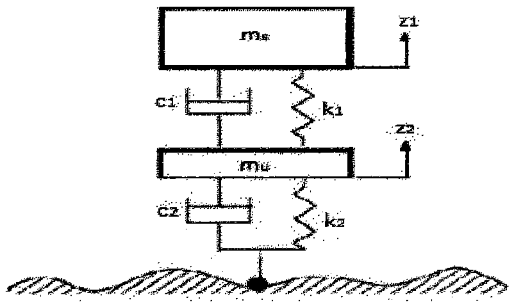


Figure 2: Quarter truck vehicle model

Based on Figure 2, k_1 is the suspension stiffness, $c_1$is the suspension damping, k_2 is tyre spring, c_2 is tyre damping. The vehicle body, supported by a suspension system, is approximately designed as the sprung mass (represented as m_1) and is considered rigid with mass properties that are concentrated at its centre of gravity, and a moment of inertia about the centre of gravity, in the pitch plane. The additional masses that are significant to the dynamic wheel load performance are proportions of the suspension linkage, denoted as un-sprung masses, represented as m_2. The un-sprung mass should include an appropriate contribution from all links, the moving parts of the damper, spring, and etc., and dominated by the wheel units. Sprung masses and un-sprung masses were both constrained to move vertically. A tyre spring and damper resulted from the rubber around the wheel. Hence, it can be seen that this model has two degrees of freedom i.e., the vertical displacements $z1$ and $z2$.

The input to the quarter truck model was a single sided spectrum of road displacement. The longitudinal profile of the pavement influenced the vertical excitation of the vehicle and the degree to which the sprung mass and axle motions causing dynamic loading was produced. The surface characteristics of the road must be represented in some way, i.e., considered deviation of the road from the simplest possible case (i.e., perfectly smooth, horizontal, and straight). The profile was described as the change in elevation of the pavement's surface with distance; or in terms of the dominant wavelengths in the profile and their amplitude of power spectral density of the profile.

As shown in the figure, the height of the surface above a fixed horizontal datum was plotted as a function of the distance along the road. The total number of intervals in the simulation was sufficient to ensure reasonable statistical accuracy in the results. A total of 1,002 point simulations were run, with an *IRI* of 2m/km. Long wavelength irregularities corresponded to the low frequency components in the time domain, whilst the short wavelength irregularities corresponded to the high frequency component.

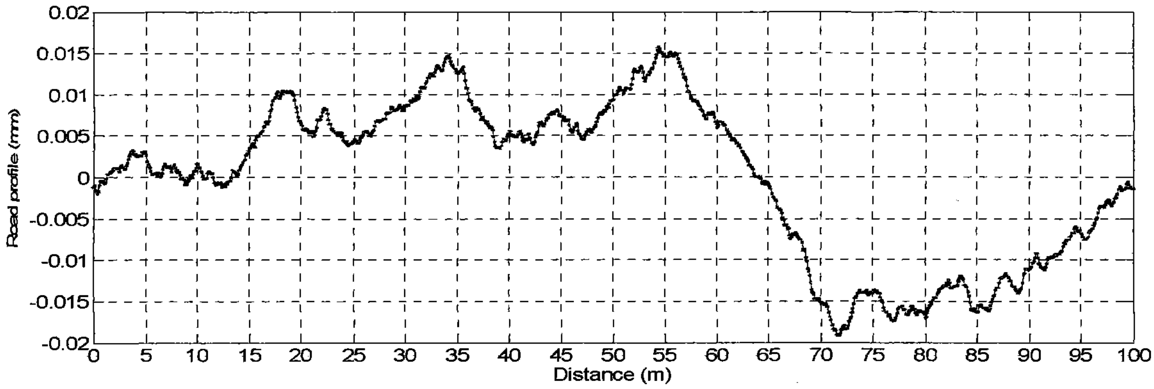


Figure 3: Road profile

Quarter-truck model vehicle parameters from the previous study are shown in Table 1. The steel suspension elements were based on a validated model that was developed by Francher et al. [28]. The parameters used in their models were largely based on the results obtained from validated articulated vehicle simulations by Collop [22]. The same parameters were used by Cole and Cebon [16]. It can be seen that, for a drive axle suspension, the suspension stiffness is lower for the air suspension, because the air suspension has a higher level of hydraulic suspension damping and there is no friction. The steer axle suspension has a lower suspension stiffness and a higher level of hydraulic suspension damping.

Table 1 Axle group model parameters [22]; M1=load for single tyre

Parameter	Steer axle	Single axle-steel suspension (SINS)	Single axle-air suspension (SINA)	Tandem axle- steel suspension (DUAS)	Tandem axle- air suspension (DUAA)	Triaxle - steel suspension (TRIS)	Triaxle axle-steel suspension (TRIA)
M_1 (kg) – full loads	X	X	X	X	X	X	X
M_2 (kg)	400	600	600	500	500	400	400
k_s (MN/m)	0.23	0.86	0.5	0.9	0.2	0.9	0.2
c_s (kNs/m)	1.5	6.5	13.0	1.0	8.0	1.0	8.0
k_t(MN/m)	1.0	2.0	2.0	2.0	2.0	1.3	1.3
c_t(kNs/m)	1.0	2.0	2.0	2.0	2.0	1.3	1.3

Quarter-truck sprung masses

In this study, un-sprung and sprung masses were calculated for the single wheel Quarter-truck model. The maximum load of each axle type (i.e., single, tandem, and triaxle) for each truck was based on the Weight Restriction Orders (as shown in Table 2) [29] for all trucks (as shown in Figure 1). The maximum sprung mass was calculated by dividing the maximum load for each axle by the total number of tyres.

Table 2 Maximum axle load [Weight restriction (Federal Road) [Amendment] Order 2003]

Type of Axle	Number of Wheels Per Axle	Maximum axle load $x10^3$ (kg)
Single	2	6.0
	4	12.0
Tandem	4	19.0
Triaxle	4	21.0
Tandem	2	9.6
	2 wheels for the 1st axle and 4 wheels for the 2nd axle	14.0
Triaxle	2	12.8
	2 wheels for the 1st axle, 4 wheels for the 2nd axle, and 3 wheels for the 3rd axle	17.0

As shown in the table, the sprung mass of each single tyre of each type of axle varies. Therefore, to simplify the computation, the maximum sprung mass for single, tandem, and tri-axle of 3000, 2400, and 2130kg, respectively, were determined for the next stage of the computation. In order to determine the total loading (sprung and un-sprung masses) for the quarter-truck model for 1/3 laden, 2/3 laden, fully laden, and a 1/3 overloaded truck; the proportion was multiplied by the maximum load of a single tyre (as shown in Table 3). The sprung mass was then calculated by substituting the total load with the un-sprung mass. The un-sprung mass was constant for all loading conditions (varies according to single, tandem, or triaxle).

The payload of both models was assumed to be uniformly distributed along the trailer. Figure 4 shows the tyre forces histories that were generated by both suspensions travelling over a 100m section at 20m/s. As expected, the dynamic tyre forces generated by the truck fitted with a steel suspension gave higher peaks than those generated by the truck fitted with air-sprung suspension. This shows that the amplitudes increased quickly at certain locations, due to the body bounce mode, through immediately going up or down on a level road surface. The dynamic load coefficient of the tyre forces of the quarter-truck fitted with a steel suspension was more than twice that of the truck fitted with an air suspension, which were 0.12 and 0.06, respectively.

Table 3 Loading Proportions of Single, Tandem and Triaxle

Loading proportion from fully laden	Single Axle (kg)	Tandem Axle (kg)	Triaxle (kg)
1/3	1000	800	710
2/3	2000	1600	1420
1	3000	2400	2130
4/3	4000	3200	2840

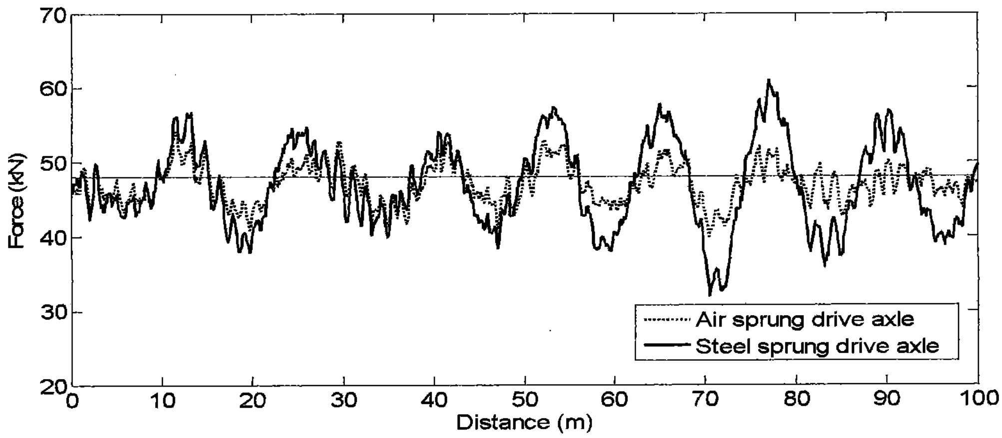


Figure 4 Tyre forces from the drive axle (single axle, single tyre) fitted with steel and air suspensions travelling at 20m/s. (Total sprung and un-sprung mass is 4800kg)

Result and Discussion

In this section, the quarter-truck model's parameters were replaced with drive air and steel suspension of the drive axle. The quarter-truck model shown in Figure 1 was used with the following parameter values (referred to as DUAA and DUAS in Table 4).

Table 4 Parameter Values of Air and Steel Suspensions

Quarter-truck components	Air suspension	Steel suspension
m_1 (kg)	4300	4300
m_2 (kg)	500	500
c_1 (Ns/m)	8×10^3	1×10^3
c_2 (Ns/m)	2×10^3	2×10^3
k_1 (N/m)	0.2×10^6	0.9×10^6
k_2 (N/m)	2×10^6	2×10^6

Study of the tyre force spectral density curves reveals that the normal effect of increasing suspension stiffness is to increase the response of the sprung mass model and decrease the response of the wheel hop mode. Figure 5 shows a comparison of the simulated tyre force spectral densities for trucks fitted with steel and air suspensions travelling at 20m/s. This indicates that, for these particular operating conditions, steel suspension fitted on a quarter-truck model increased the dynamic forces. As shown in the figure, the main spectral peak for both curves was at a frequency of about 1Hz; which means that the sprung mass was resonating on the suspension. However, the higher resonance that was experienced when steel suspension was fitted was represented by large discrepancies in the region of the sprung mass modes of 1 to 3Hz and 5 to 12Hz. The response accentuated at a frequency above the body resonance point was a result of wheel motion; with the greatest effect being seen at the resonance frequency of the wheel.

Figures 6(a-b) show a comparison of the tyre force spectral density of a truck travelling at different speeds i.e., 20m/s and 30m/s, for each type of suspension. The *DLCs* of the tyre forces were approximately 0.07 and 0.14, for trucks fitted with air and steel suspension, respectively, at 30m/s. As shown in Figure 4(a), there are big discrepancies in the region of the sprung mass modes of 1 to 3Hz and the un-sprung mass modes. At a higher speed (30m/s), the frequency of the main spectral peak increased from 1.1Hz to 2.5Hz. A similar effect is shown in Figure 6(b) for the truck with an air suspension drive axle. However, at 30m/s, apart from the main peak, a second peak appears at several points.

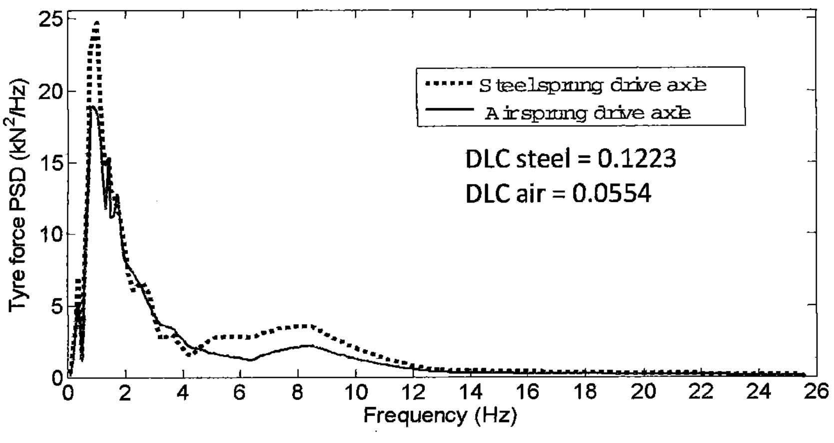
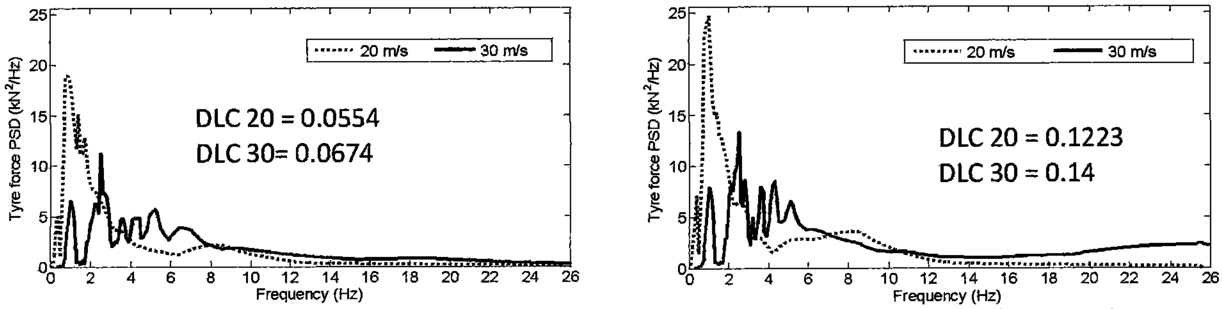


Figure 5 Tyre force spectral density

Effect of loading conditions on the dynamic wheel loads of trucks

It is a requirement to calculate the road damage for lower values of sprung mass, in order to predict the effects of vehicles operating partially laden. The dynamic force exerted on the road's surface by traffic, depends on the vehicle's characteristics and road roughness. The measured dynamic wheel loads, generated by particular fleet models, was assessed in terms of various loading conditions. Figure 7 shows the simulated dynamic tyre forces generated by a single axle with a steel suspension, under various loading conditions, in response to road displacement, at 20m/s. Ina fully loaded condition, the total sprung and un-sprung mass was 3000kg. As shown in the figure, tyre forces are smaller at lower loads, and vice versa. However, for certain conditions along the pavement, the tyre forces of a 1/3 fully laden truck are more aggressive; with high peaks, whose maximum is over 150% more than the mean average loads. This phenomenon is caused by the uncertainties of body bounce that are generated by the truck, which are strongly dependent on surface irregularities.

The percentage increase of forces for a truck that is 2/3 fully laden, above the mean average tyre forces, was lower at approximately 45%. This value decreased for higher truck loads and increased for lower truck loads. Figure 8 shows a comparison of tyre force spectral densities of a truck with different loading conditions, travelling at 20m/s. As shown in the figure, the main peak was similar for all loading conditions, at a frequency of about 1Hz. At this frequency, the sprung mass remained stationary, in a rough duplication of the road's input. However, there were big discrepancies in the regions of the sprung mass modes of 1 to 3Hz and the un-sprung mass modes. The wheel hop mode for a 1/3 laden truck is higher than any other loading condition. In this condition, the un-sprung mass of the tyre/wheel assembly goes into a vertical (hop) resonance mode; thus adding a small bump. This is because the suspension's stiffness and damping are insufficient to minimise the natural frequency that is required for a very lightly sprung mass (vehicles are manufactured primarily for fully laden use). As such, the suspension prevents establishing a natural frequency of the system in the bounce (vertical) mode. Therefore, due to impractical component assemblies (suspension stiffness, damping, etc.), the natural frequency of the wheel hop mode was evaluated at a frequency above the body's resonant point; 5 to 8Hz.



(a) Drive axle with air suspension (b) Drive axle with steel suspension

Figure 6. Comparison of tyre force spectral densities from a truck travelling at 20 and 30m/s. (Total sprung and un-sprung mass is 4800kg)

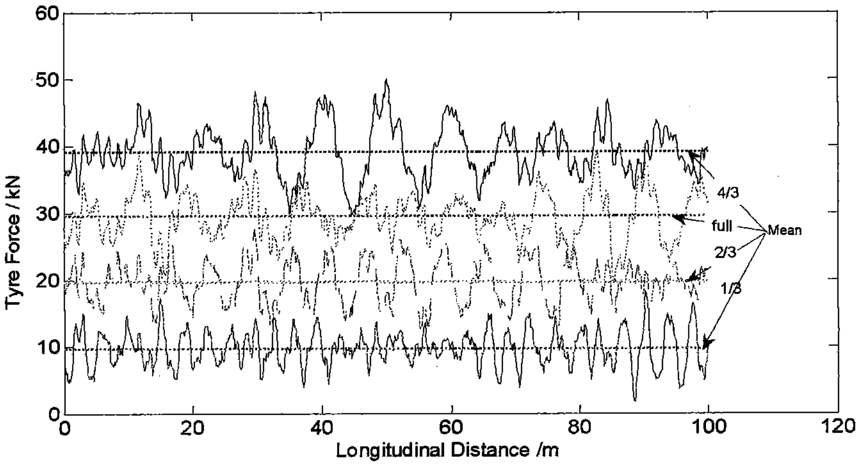


Figure 7 Tyre forces from a single axle truck fitted with steel suspension, with a 1/3 loading, 2/3loading, fully laden, and 1/3 overloading, at 20m/s. Dotted lines are the static forces, corresponding to each loading condition (total sprung and un-sprung mass was 3000kg).

Dynamic load coefficient of tyre forces

To ensure that the dynamic force level generated by the axle load group models is realistic, a level of dynamic variation of tyre force history and dynamic load coefficient was computed. This is the ratio of standard deviation of tyre forces to static force. The quarter-car model's parameters (as shown in Table 1) were used and loading conditions were varied, as shown in Table 3. As a result, the *DLCs* of the tyre forces for seven different axle types, over a broad range of truck speeds, are shown in Figure 9. Accordingly, each axle was fully loaded. The results were separated into two, namely high and low degrees of effect. The lower degree of *DLCs* were the tyre forces from trucks fitted with air suspension (except for the steering axle) and the higher degree was for tyre forces from trucks fitted with steel suspension. As expected, the DLCs for trucks fitted with steel suspension were greater than those fitted with air–sprung suspension. However, for both types of suspension, the *DLCs* were relatively insensitive to speed.

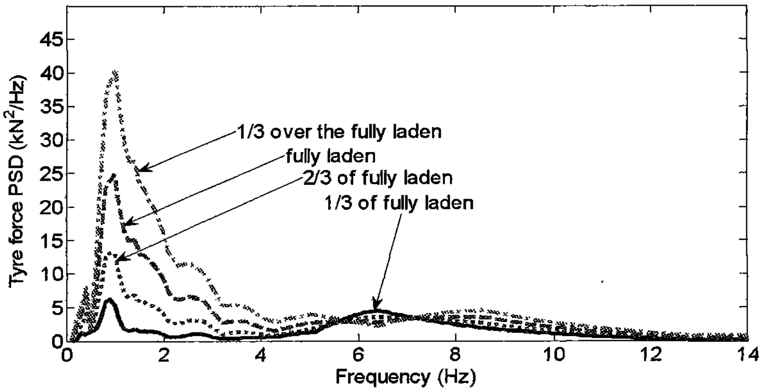


Figure 8.Tyre force spectral density plotted against time frequency, for trucks with various loading conditions, travelling at 20m/s. Quarter-truck parameters are shown in Table 4 (denoted as SINS with a total loading of 3000kg).

The *DLCs* of tyre forces from trucks with loadings that are lower and higher than fully laden are shown in Table 5.

Table 5. DLC for different axle suspension variation with speed

Axle & Suspension types	Loading conditions	DLCs				
		Speed (m/s)				
		10	15	20	25	30
STEER	FULL	0.047	0.065	0.080	0.088	0.092
(SINS)	1/3	0.157	0.212	0.271	0.316	0.366
	2/3	0.092	0.127	0.156	0.177	0.205
	FULL	0.074	0.103	0.115	0.142	0.164
	4/3	0.067	0.100	0.095	0.120	0.146
(SINA)	1/3	0.123	0.169	0.211	0.250	0.286
	2/3	0.056	0.078	0.097	0.115	0.132
	FULL	0.042	0.058	0.071	0.086	0.097
	4/3	0.034	0.048	0.059	0.070	0.081
(DUAS)	1/3	0.184	0.252	0.309	0.356	0.421
	2/3	0.100	0.142	0.168	0.208	0.226
	FULL	0.081	0.110	0.152	0.173	0.196
	4/3	0.074	0.096	0.120	0.149	0.174
(DUAA)	1/3	0.146	0.202	0.252	0.279	0.341
	2/3	0.067	0.094	0.117	0.139	0.158
	FULL	0.046	0.064	0.080	0.095	0.107
	4/3	0.035	0.050	0.062	0.072	0.082

(TRIS)	1/3	0.176	0.243	0.299	0.347	0.407
	2/3	0.103	0.139	0.180	0.207	0.233
	FULL	0.088	0.1215	0.144	0.173	0.195
	4/3	0.075	0.103	0.113	0.142	0.165
(TRIA)	1/3	0.109	0.150	0.187	0.221	0.254
	2/3	0.05	0.069	0.087	0.103	0.118
	FULL	0.035	0.048	0.061	0.072	0.081
	4/3	0.028	0.039	0.048	0.056	0.063

These results show that *DLCs* are sensitive to loading conditions. As the sprung mass of the truck was reduced, the *DLC* value increased and vice versa. For example, reduced truck loads (from the maximum permissible to 1/3 of maximum) increased *DLCs* from about 100% (for tri-axle fitted with a steel suspension) to about 218% (for tandem axle fitted with an air suspension). Overall, *DLCs* for the trucks fitted with steel sprung and air sprung suspensions, at 1/3 fully laden, were higher than the normal range of 0.1 to 0.3 (as presented by several researchers).The reason for this phenomenon is once again due to the body bounce mode, through immediately going up and down on the road's surface. In this condition, the reduction in road damage (caused by the lower static weight of the partially laden vehicle) is therefore offset by the increase in road damage caused by the increased dynamic forces. For other loading conditions, the effect of speed was less significant, due to less truck vibration than the lightest truck load's vibration. Consequently, the *DLC* of the tyre forces were always smaller for the truck fitted with air suspension than for the truck fitted with a steel suspension for the same unevenness, load, and vehicle speed. However, the efficiency of the air suspension is clearly better for high axle loads.

Summary and Conclusions

1. Dynamic loading was observed to be influenced by the type of suspensionand vehicle speed.
2. The tyre forces from of 1/3 laden were more aggressive than the other loading conditions. The maximum amplitude was greater than 150% mean average load. This was due to uncertainties of body bounce generated by the truck, which strongly depended on surface irregularities. For 2/3 laden, the maximum amplitude was lowered to about 45%, which decreased further for higher truck loads and vice versa.
3. At low speed, the *DLC* greatly decreased if the load increased. Meanwhile, the *DLC* was independent of the load at high speed, due to the adaptive suspension, which was designed to be efficient at high loads.
4. *DLCs* were always smaller with air suspension than steel suspension; for the same load and vehicle speed. However, the efficiency of the air suspension was clearly better for high axle loads. The worst was when the truck loaded was only 1/3 laden.
5. A higher resonance was experienced when a steel suspension was fitted, compared to the air suspension; with large discrepancies in the region of the sprung and un-sprung mass modes of 1 to 3Hz and 5 to 12Hz, respectively.
6. The main spectral peak was higher when the truck speed was higher; and at that higher speed, the second peak appeared at several points.

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