VALIDATION OF AN ARTICULATED VEHICLE SIMULATION

by D J Cole and D Cebon
(Department of Engineering, University of Cambridge)

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VALIDATION OF AN ARTICULATED VEHICLE SIMULATION

ABSTRACT

Tests were performed on a typical UK articulated vehicle to measure dynamic tyre forces and sprung mass accelerations. The measured road profile data and vehicle response data are used to determine some of the important characteristics of articulated vehicle vibration behaviour. In particular, roll motions and their effect on dynamic tyre forces are examined. The measured data are used to validate two and three-dimensional computer models of the vehicle. Attention is given to modelling the tandem leaf-spring trailer suspension. The conditions under which a two-dimensional model can accurately simulate vehicle behaviour are examined.

1 INTRODUCTION

Lack of understanding about the relationship between vehicle design and road damage has limited the effectiveness of vehicle legislation in minimising the road-damaging ability of vehicles. The simulation of vehicle vibration is valuable for understanding the ways in which dynamic tyre forces of heavy lorries are generated, and for predicting the effect of vehicle design changes on road damage.

Previous work on heavy vehicle vibration simulation was extensively reviewed by ElMadany et al [1] in 1979 and by Cebon [2] in 1985, and therefore will not be repeated in detail here. Cebon noted that very few heavy vehicle simulations had included experimental validation. These included a two-dimensional non-linear simulation by Van Deusen [3]. Agreement with measurement was reasonable, the main sources of error being attributed to vehicle test speed variation, differences between test track and simulated road profile spectra, and frame torsional modes not included in the simulation. Sayers and Gillespie [4] simulated a tandem axle 'walking-beam' suspension using a vehicle model with three degrees of freedom. Simulated acceleration and tyre force spectral densities agreed closely with measured values at frequencies above 5Hz. Tyre forces at lower frequencies were over predicted because
the sprung mass of the vehicle was lumped into one degree of freedom. Heath and Good [5] created linear models of five different three-axle tractors, but lack of information about the test vehicles and the road profile limited the extent of the validation.

Cebon developed a vibration simulation program suitable for modelling lumped parameter non-linear vehicle models of up to three dimensions [2, 6]. Two and three-dimensional non-linear models of a three-axle rigid fuel tanker were validated with data from experimental tests. The vehicle and test conditions were not representative of typical freight transport operating conditions. The test lane was a rough paved track and the maximum test speed was 4.2m/s. Agreement of the simulation with the test results was satisfactory considering the harsh operating conditions. The main errors were attributed to inadequate modelling of the bogie suspension spring. A frame bending mode at 6Hz did not significantly affect the tyre forces. Cebon concluded that two-dimensional models were satisfactory for predicting tyre forces at frequencies above the major sprung mass resonances. The two-dimensional models were less accurate at lower frequencies than three-dimensional models because of sprung mass roll motions, excited by roll roughness of the paved test track. Cebon also performed simulations of three different articulated vehicles in a parametric study, but these models were not validated.

Several heavy vehicle simulation studies have been reported since the review by Cebon [7-13], but only two of these have included comparison with measured responses. Hu [11] described a simulation model used for the study of articulated vehicle ride dynamics. The model was two-dimensional and incorporated non-linear spring characteristics and a first order model of frame bending. A limited comparison of simulated responses with measurements indicated good agreement. Sakuma et al [13] created two-dimensional non-linear models of rigid vehicles using a finite element program, and flexible frame modes were included. Agreement between measured and simulated vertical acceleration of the cab was good.

Most theoretical work on dynamic tyre forces has considered the pitch and bounce motions of rigid framed vehicles only. Consequently, there is no definite
opinion on whether roll motions and frame flexibility are significant in contributing to dynamic tyre forces.

In summary, many heavy vehicle simulations have been developed, but only a small number have been compared with experimental measurements, and no articulated vehicle simulations have been convincingly validated. Further work is needed to determine the accuracy that can be expected from realistic simulations of typical heavy vehicles operating under typical conditions of speed and road roughness.

2 EXPERIMENTS

2.1 Vehicle Tests

The test vehicle was a typical UK articulated lorry, with four axles and a maximum allowable gross mass of 32.5 tonnes (figure 1). It was provided and operated by the Transport Research Laboratory (TRL). The tractor had two axles, each fitted with multi-leaf springs, hydraulic dampers, and anti-roll bars. The trailer was fitted with a wide-spread tandem axle group having mono-leaf springs and a load equalising mechanism (figure 2). The steering axle was fitted with single radial-ply tyres and all other axles were fitted with dual radial-ply tyres.

The vehicle was instrumented by TRL to measure dynamic tyre forces, by means of strain gauges bonded to each axle between the spring mounting and brake back plate. Accelerometers were fitted near to the ends of each axle so that a correction could be made for the inertia of the mass outboard of each strain gauge [2, 14]. Static force components were measured with the vehicle stationary using portable weighpads. The static forces were added to the dynamic components to give the total tyre forces. This method for measuring dynamic tyre force is thought by Sweatman to be the best compromise between accuracy and complexity [15], but it assumes that the distance between the strain gauge and the line of action of tyre force remains constant. Tractor and trailer frame vertical accelerations were measured by eight piezo-electric accelerometers, three on the tractor and five on the trailer.
The outputs of the twenty-four transducers were recorded onto magnetic tape. The recorded signals were subsequently digitised (12-bit resolution) and transferred to a mainframe computer for further analysis.

The vehicle was tested at three speeds (13, 18, 22 m/s or 30, 40, 50 mph) and three payloads (unladen, half laden, fully laden). The payload consisted of concrete blocks (each 500 kg) placed at the front and rear of the trailer load bed.

### 2.2 Road Profile

The vehicle was tested on a 530 m long straight section of the TRL test track which had a jointed concrete surface with joints at 30 m intervals. The profiles of left and right hand wheel tracks were measured by TRL using their high-speed profilometer [16], which provides measurements at 0.107 m intervals along the road. The measured height is the average height over the preceding 0.107 m distance along the road. This averaging process and the effects of aliasing result in a lower wavelength limit of about 0.5 m [16]. The upper limit is several hundred metres.

For the slowest of the three testing speeds (13 m/s), the minimum wavelength (0.5 m) corresponds to a frequency of 26 Hz. For the highest speed (22 m/s), the maximum wavelength (200 m) corresponds to about 0.1 Hz. These frequencies are outside the main frequencies of interest in this study.

The profilometer cannot measure a pair of wheel tracks simultaneously; each track must be measured separately. For the profile measurements of the test lane, synchronisation between nearside and offside wheel tracks to within the 0.107 m measuring interval was achieved by using reflective strips on the track to put event markers on the profile records.

The displacement spectral densities of the measured vertical profiles of the nearside and offside wheel tracks are given in figure 3, which shows that the two wheel tracks have similar roughness.
Dodds and Robson [17] presented a method for classifying road profiles, which was the British proposal for an ISO standard [18]. Road profile spectra (single-sided) were defined in the following way:

\[
S(n) = S(n_0) \left( \frac{n}{n_0} \right)^{-w_1} \quad n \leq n_0
\]

\[
S(n) = S(n_0) \left( \frac{n}{n_0} \right)^{-w_2} \quad n \geq n_0
\]

\(n = \text{wavenumber, cycles/m}\)

\(n_0 = \text{reference wavenumber, cycles/m}\)

\(S(n) = \text{displacement spectral density, m}^3/\text{cycle}\)

\(S(n_0) = \text{spectral density at reference wavenumber, m}^3/\text{cycle}.\)

Values in [17, 18] for the parameters \(w_1, w_2, n_0, \) and \(S(n_0)\) were determined by analysing a wide range of European road profile measurements [19]:

\(w_1 = 2.0\)

\(w_2 = 1.5\)

\(n_0 = 1/(2\pi) \text{ cycles/m}\)

\(S(n_0)\) as follows:

<table>
<thead>
<tr>
<th>Road Class</th>
<th>(S(n_0) / 10^{-6} \text{m}^3/\text{cycle})</th>
</tr>
</thead>
<tbody>
<tr>
<td>A very good</td>
<td>2–8</td>
</tr>
<tr>
<td>B good</td>
<td>8–32</td>
</tr>
<tr>
<td>C average</td>
<td>32–128</td>
</tr>
<tr>
<td>D poor</td>
<td>128–512</td>
</tr>
<tr>
<td>E very poor</td>
<td>512–2048</td>
</tr>
</tbody>
</table>

Motorways are generally type A or B, principal roads type A to D, and minor roads type C to E.
Robson [20] subsequently reported that these values of \(w_1\) and \(w_2\) were incorrect and should be larger by a factor of 1.5, to give \(w_1=3.0\) and \(w_2=2.25\). These corrected values were used for plotting the classifications shown in figure 3. This figure shows that the test lane does not fit the classifications exactly, but corresponds approximately to the 'good' classification (poor motorway or average principal road).

The nearside and offside profiles \((u_1\) and \(u_2\)) were transformed into a vertical 'bounce' displacement \((u_1 + u_2)/2\) and a 'roll' displacement \((u_1 - u_2)/2\). The ratio of roll spectral density to bounce spectral density for the test lane is shown in figure 4a. The ratio is close to zero at small wavenumbers, and increases to about one at higher wavenumbers. This is typical of many road surfaces [21, 22]. The ratio is generally less than 0.5 at wavenumbers below 0.2cycles/m.

Figure 4b contains the same data as figure 4a, but the wavenumber axis is now plotted on a circular arc. At the centre of the arc is the origin of a set of rectangular axes, with speed on the vertical axis and frequency along the horizontal axis. A straight line drawn between the origin of the rectangular axes and a point on the roll/bounce ratio curve defines the combinations of speed and frequency that correspond to this point.

The line labelled A on figure 4b corresponds to a typical sprung mass mode frequency of 3Hz and a typical vehicle speed of 24m/s. The corresponding value of roll/bounce ratio is below 0.5, and so there is comparatively little roll excitation of

\[ S(n) = c|n|^{-w} \]

where \(w = 2.5\), and \(c\) is as follows:

<table>
<thead>
<tr>
<th>Road Class</th>
<th>(c \times 10^{-8}) m (^{0.5}) cycle (^{1.5})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motorway</td>
<td>3–50</td>
</tr>
<tr>
<td>Principal road</td>
<td>3–800</td>
</tr>
<tr>
<td>Minor road</td>
<td>50–3000</td>
</tr>
</tbody>
</table>
vehicles at (or below) this frequency. Therefore, the sprung mass roll modes, which
generally have frequencies below 3Hz, are unlikely to contribute significantly to the
dynamic tyre forces [21]. (However, this may not be the case for poorer roads where
the nearside wheel track is sometimes rougher than the offside wheel track [23]). The
line labelled B corresponds to a typical unsprung mass mode frequency of 15Hz and
a speed of 24m/s. The corresponding roll/bounce ratio is above 0.5, and therefore
unsprung mass roll modes will have significant excitation at this speed.

If the vehicle speed is decreased from 24m/s, the frequency corresponding to
a given point on the roll/bounce ratio graph will decrease, and the excitation
of the sprung mass roll modes will become comparable to the excitation of the
bounce modes. The unsprung mass modes (above 10Hz) will normally be excited
at wavenumbers where the ratio of roll to bounce excitation is close to one. These
results were derived for one particular road surface but it is likely that the general
trends will hold for most highway surfaces [21, 22].

The measured vehicle responses are used in the next section to examine the
relationship between road profile and vehicle roll motions.

2.3 Vehicle Response

Roll motion. The nearside and offside tyre forces \( (F_1(t) \text{ and } F_2(t)) \) measured on
the tractor drive axle and the leading trailer axle were transformed into ‘bounce’
\( (F_1(t) + F_2(t))/2 \) and ‘roll’ \( (F_1(t) - F_2(t))/2 \) components. Figure 5 shows the spectral
densities of these components for the fully laden vehicle travelling at 22m/s.

In the frequency range of the sprung mass modes (1–4Hz), the roll components
of the drive axle force (figure 5a) and trailer axle force (figure 5b) are negligible
compared to the bounce components. In the frequency range of the unsprung mass
modes (10–15Hz) the bounce and roll components have similar magnitudes. These
results are generally consistent with the measured roll/bounce characteristics of the
road profile described above.
Figure 6 shows spectral densities of measured bounce and roll accelerations (units \((\text{m/s}^2)^2/\text{Hz}\) and \((\text{rad/s}^2)^2/\text{Hz}\) respectively) for the fully laden vehicle at 22m/s. Both spectral densities are plotted on the same axes. This is equivalent to comparing the average \((\ddot{z}_1 + \ddot{z}_2)/2\) and difference \((\ddot{z}_1 - \ddot{z}_2)/2\) of vertical accelerations \(\ddot{z}_1\) and \(\ddot{z}_2\) at points 1m each side of the longitudinal vehicle centreline.

Figures 6a,b show that in the range 1–4Hz the sprung mass roll accelerations measured at the rear of the tractor and the rear of the trailer are small compared to the bounce accelerations. (The peak at 7Hz in these graphs is due to trailer bending, described later). This confirms that low frequency sprung mass roll motions are small. However, figure 6a shows that in the range 12–15Hz the roll acceleration at the rear of the tractor frame is very large, particularly when compared to the roll acceleration of the trailer, figure 6b. This is likely to be because the rear portion of the tractor frame has low mass and is torsionally flexible. The rear of the tractor frame appears to follow the roll motion of the drive axle closely, figure 6c.

The bounce and roll acceleration spectral densities of the leading trailer axle are shown in figure 6d. The peak at 14Hz corresponds to the tandem axles rolling in antiphase with respect to each other, and the peak at 15Hz corresponds to the axles bouncing in antiphase with respect to each other. In these two modes there is significant motion of the levelling beams. The peak at 22Hz corresponds to the axles rolling in phase with respect to each other, and the peak at 24Hz corresponds to the axles bouncing in phase with respect to each other, but in each of these modes there is little motion of the levelling beams.

These results suggest that for simulating the sprung mass modes (1–4Hz) of heavy lorries travelling under typical operating conditions, it is satisfactory to model the vehicle in only two dimensions. The simulation of the unsprung mass modes (10–15Hz) in two dimensions may or may not be satisfactory, depending on the characteristics of the suspension and the roll behaviour of the sprung masses.

For this particular vehicle the contribution of the unsprung mass modes to the dynamic tyre forces is generally small compared to that of the sprung mass
modes. This is typical of many heavy vehicles [5, 24, 25, 26]. For these vehicles, two-dimensional models are likely to be adequate for simulating dynamic tyre forces.

Some types of suspension do generate large tyre forces at high frequencies. The worst cases are tandem suspensions with lightly damped pitch modes (for example, walking beam suspensions [26]). Other suspensions, such as those with air springs, can generate low dynamic tyre forces with contributions of similar magnitudes from sprung and unsprung mass modes [25]. In these cases a three-dimensional model may be necessary for accurate simulation of dynamic tyre forces.

**Wheel non-uniformities.** Several experimental studies have noted a contribution to dynamic tyre forces from wheel out of balance, radial run-out, or circumferential stiffness variations [4, 5, 14, 21, 26]. The effects of wheel non-uniformities on dynamic tyre forces are usually only detected when the excitation from the road roughness is small [21]. The wheel rotation frequency for the test speed of 22m/s is 7.1Hz. The tyre force spectral densities shown in figures 5a,b do not show any significant peaks at this frequency.

**Trailer bending.** Figure 7a shows the measured 'bending' acceleration response of the half laden trailer at 13m/s and 22m/s. The graph indicates a lightly damped peak at 9 to 10Hz, with slightly greater response at the higher test speed. Figure 7b shows the corresponding responses for the fully laden trailer. The resonant frequency has decreased to 7Hz, as would be expected from the increase in trailer mass. The response at the higher test speed is significantly greater than at the lower speed. This is likely to be because wheel rotation frequency at a vehicle speed of 22m/s coincides with the 7Hz resonant frequency. In all four cases of speed and payload, examination of the corresponding tyre force spectral densities revealed very little response at the trailer bending resonant frequency. This is thought to be because the nodes of the first frame bending mode are close to the locations of the suspension systems [2, 25]. Cebon [2, 6] observed similar behaviour of a rigid three-axle vehicle.

† The 'bending' acceleration is the difference between the accelerations of the centre and the ends of the trailer.
The implication of these results is that it is satisfactory not to include frame bending modes when simulating dynamic tyre forces. This is in contrast to simulating ride accelerations, when frame bending can be very important [1, 7, 8, 9, 13, 21]. These results may depend on the distribution of payload on the vehicle. During the vehicle tests the payload was concentrated at the two ends of the trailer (figure 1). Greater influence of frame bending modes on tyre forces may have been observed if the payload was distributed nearer the middle of the load bed.

3 VEHICLE MODEL

3.1 Vehicle Models

The non-linear vehicle vibration simulation program developed by Cebon [2, 6] was used to create three models of the test vehicle:

i) A 6 degrees of freedom (DOF), two-dimensional trailer suspension model.
ii) An 11 DOF, two-dimensional tractor and trailer model.
iii) A 21 DOF, three-dimensional whole vehicle model, shown in figure 8.

Features of these models are:

i) Non-linear leaf spring elements, the hysteresis characteristic being modelled according to the equation devised by Fancher et al [27, 28].
ii) Anti-roll bars on the tractor axles (21 DOF model).
iii) Viscous damping on the tractor axles, with different rates in bump and rebound.
iv) Tyres modelled as linear springs in parallel with light viscous dampers. The tyres can lose contact with the road [6, 29].
v) Simple tyre contact patch averaging for the envelopment of short wavelength irregularities [6, 29].
vi) Rigid tractor and trailer sprung masses.

Inertia properties of the sprung and unsprung masses were estimated using the measured static tyre forces and dimensions of the test vehicle. Tyre stiffness values were derived from the manufacturer’s force-deflection data. Parameter values for the three-dimensional model are given in [30].
3.2 Tractor Suspension Model

Tractor suspension properties were provided by the manufacturer. The leaf spring data consisted only of values for linear (large deflection) stiffness. The hysteresis characteristics were estimated from results of tests on springs of similar type [27, 28, 31, 32].

3.3 Four-Spring Trailer Suspension Model

The four-spring trailer suspension consisted of two mono-leaf springs attached to each axle. The springs had slipper mounts at each end, and longitudinal location of each axle was provided by two radius arms (figure 2).

The force–deflection characteristic of a single trailer leaf spring tested on a laboratory rig (with slipper end mounts, but without radius arms), was provided by the suspension manufacturer and used to determine suitable values for the parameters of the leaf spring model. The manufacturer’s data and the fitted characteristic are shown in figure 9.

The effects of the forces in the radius arms on the behaviour of the suspension were modelled using the following assumptions:

i) Small displacements about the equilibrium position.

ii) Friction forces at the slipper ends are horizontal and normal forces are vertical.

iii) Pitch moments applied by the axle tubes and by rotational inertia of the axle assemblies are zero.

iv) Longitudinal acceleration of the unsprung masses is zero.

v) The leaf springs are treated kinematically as a rigid beams.

The idealised suspension geometry used for this analysis is shown in figure 10a. Each leaf spring is considered to consist of a non-linear vertical stiffness and a linear pitch (‘wind-up’) stiffness $\tau_y$ (located at the point RQ in figure 10a). The vertical displacement of the axle is $z$ and the pitch angle is $\theta_y$. The pitch angle across the two
ends of the leaf spring is \( \theta'_y \), and is directly related to the pitch angle of the levelling beam \( \theta_l \) by

\[
\theta_l b = -2a \theta'_y. \tag{3}
\]

The angular deflection across the pitch stiffness \( \tau_y \) is therefore

\[
\theta_w = \theta_y - \theta'_y, \tag{4}
\]

where \( \theta_w \) is the 'wind-up' angle. When there is no force in the radius arm \( \theta_w \) is zero. The radius arm provides a kinematic constraint such that

\[
\theta_y = -\frac{z \tan \beta}{c}. \tag{5}
\]

Thus from (3) to (5)

\[
\theta_w = -\frac{z \tan \beta}{c} + \frac{\theta_l b}{2a}. \tag{6}
\]

This equation gives the angular deflection \( \theta_w \) across the pitch stiffness of the spring in terms of the displacement of the axle \( z \) and the pitch angle of the levelling beam \( \theta_l \), assuming that there is no horizontal movement of the spring due to sliding at the slipper ends of the spring.

Now consider the forces acting on the spring and axle assembly (figure 10b). The torque \( T_y \) necessary to cause the wind-up \( \theta_w \), and the corresponding horizontal force \( F_h \) applied by the radius arm are given by

\[
T_y = \theta_w \tau_y, \tag{7}
\]

\[
F_h = \frac{T_y}{c}. \tag{8}
\]

The horizontal component of force in the radius arm \( F_h \) is reacted by the horizontal friction forces \( F_{f1} \) and \( F_{f2} \) at the spring ends. There is therefore a limit to the force \( F_h \) that can be generated, determined by the coefficient of friction \( \mu \) and normal forces \( F_{r1} \) and \( F_{r2} \) at the spring ends,

\[
|F_h|_{\text{max}} = (F_{r1} + F_{r2}) \mu. \tag{9}
\]

If the horizontal force \( F_h \) calculated from (8) is greater in magnitude than \( |F_h|_{\text{max}} \) then sliding at the slipper ends is assumed to occur, and \( F_h \) is limited to \( |F_h|_{\text{max}} \).
When sliding occurs, the vertical component of the arm force $F_v$ and the torque $T_y$ acting on the spring are given by

$$F_v = \frac{F_h}{\tan \beta}$$  \hspace{1cm} (10)

$$T_y = cF_h,$$  \hspace{1cm} (11)

where $F_h$ is limited according to (9).

The tandem suspension is modelled by treating each spring as a rigid massless beam with a non-linear leaf spring element at each end. The vertical component of the arm force $F_v$ (from eq. 10) is applied between the axle and chassis, along the vertical centreline of the axle. Horizontal suspension forces are not calculated in the vehicle simulations, therefore the two slipper friction forces and the equal and opposite horizontal component of radius arm force are applied as the pitch moment $T_y$ (from eq. 11), between the axle and chassis. The levelling beam pivots on a bush with linear pitch stiffness $\tau_l$.

Published measurements from tests performed by TRL on the test trailer [33] were used to determine values for the parameters of the tandem suspension model. In the TRL tests, the laden trailer was supported at the fifth wheel by a crane, and weighpads were placed under each wheel to measure the tyre forces. The fifth wheel was raised and lowered through several cycles to give a trailer pitch angle amplitude of $0.8^\circ$. The force amplitude at the tyres was measured to be 10.3kN for the leading axle and 7.6kN for the trailing axle. The maximum tyre force occurred on each axle when the corresponding axle to trailer frame distance was a minimum. The measurements were repeated with PTFE inserts in the slipper end mountings, and the measured force amplitudes were 3.5kN and 2.8kN. The static inclination of the radius arms $\beta$ was measured to be $12^\circ$.

The 6 DOF trailer suspension model was used to simulate the TRL tests, by running the model slowly over a sinusoidal road profile with displacement amplitude $\pm 14$mm and wavelength equal to twice the axle spacing. The values of the tandem suspension model parameters were chosen to give best agreement with the TRL
measurements [30]. Figure 11 compares the TRL measurements with the results of the simulation.

The difference in force amplitudes for the leading and trailing axles measured in the tests could not be simulated. Use of unequal friction coefficients $\mu$, unequal wind-up stiffnesses $\tau_y$, and unequal radius arm inclinations $\beta$ between the two axles in the simulation all gave equal force amplitudes.

The main features of the simulated tyre force response shown in figure 11b will now be described. Starting at the point of minimum tyre force on the leading axle (point A), the tyre force increases at a rate controlled mainly by the wind-up stiffnesses of both leading axle and trailing axle springs. At point B, sliding begins at the slipper ends of the now lesser-laden trailing axle spring. The leading axle tyre force then increases at a lower rate, because the trailing axle radius arm cannot exert any further force. On reversing the tyre displacements at point C, sliding at the ends of the trailing axle spring stops, and the tyre forces again change at a rate controlled by the wind-up stiffnesses of both springs. At point D, sliding begins at the ends of the leading axle spring.

The corresponding measured characteristic in figure 11b does not show the distinct changes in stiffness predicted by the simulation; this is likely to be because in practice the transition from sticking to sliding occurs gradually. Non-linearity in the wind-up stiffness $\tau_y$ may also cause discrepancy between measurement and simulation.

Displacement of both axles in phase with each other was simulated, and the force–displacement behaviour was very similar to that of the single spring without radius arm (figure 9). This is because the high vertical stiffness allows only a relatively small vertical displacement to occur, which is insufficient to generate large forces in the radius arms. However, it is clear that the radius arms strongly affect the pitch behaviour of the suspension.
4 VEHICLE SIMULATION RESULTS

4.1 Three-Dimensional Model

Response time histories for the 21 DOF three-dimensional model were calculated by numerical integration, using the measured test lane profile as the input.

Figure 12 shows measured and simulated tyre force histories for the nearside wheels of the fully laden vehicle travelling at 22m/s. In this figure, the horizontal axes show the position of each axle along the test section. Slight variations in test vehicle speed mean that the measured and simulated force histories are not always exactly synchronised: the maximum discrepancy in longitudinal position apparent from these graphs is about ±0.5m.

Figures 13 compares measured and simulated response spectral densities for the fully laden vehicle at 22m/s. Figures 13a-d correspond to the tyre forces in figure 12. Figure 13e is the vertical acceleration of the leading trailer axle, and figure 13f is the pitch acceleration of the tractor sprung mass. Overall the agreement between measurement and simulation is very good.

The tyre force spectral densities show some small discrepancies in the region of the sprung mass modes (1–4Hz), although the frequencies agree quite closely (figures 13a-d). In the region of the unsprung mass modes (10–15Hz) the steer and drive axle tyre forces and the leading trailer axle bounce acceleration agree quite closely (figures 13a,b,e), but the agreement is not so good for the trailer axle tyre forces (figures 13c,d). These results suggest that further refinement of the trailer suspension model is needed to simulate its complex behaviour accurately. Comparison of measured and simulated responses for other conditions of speed and payload showed similar agreement (see [30]).

Some of the low frequency discrepancies may be partly due to the assumption of rigid sprung masses. For example, in figure 13f the measured tractor pitch response shows a small contribution from trailer bending at 7Hz (see also figure 7). The
measured sprung mass 'whole body' motions (roll, bounce, pitch) may be inaccurate because the accelerometer data reduction cannot distinguish exactly between 'whole body' motion and flexible frame motion, as described in [2, 6]. The tractor sprung mass was treated as one rigid mass. In practice the cab and engine are flexibly mounted to the chassis, and have modes of vibration in the frequency range of interest.

An additional source of error may be the tractor suspension parameters, which were derived entirely from the manufacturer's data and published measurements on similar components. Ideally the suspension properties would be determined from 'in-situ' tests [31, 32]. The simple bilinear characteristic used for the damper simulation may also be inaccurate. Detailed studies of automotive damper behaviour [38, 39] have revealed highly non-linear and hysteretic behaviour.

Experimental error and data processing may contribute to some of the discrepancy between measurement and simulation. It is estimated that the vehicle speed varied by up to ±5% during each test run. The error in the tyre forces measured by the lorry instrumentation was estimated to be 1.5%rms [30].

It is difficult to determine how each of the possible sources of error contributes to the overall discrepancy. However, none of the experimental or numerical errors are thought to be excessive, and therefore the results of this study indicate the likely agreement that can be obtained when using a non-linear rigid body model with accurate road profile input and good estimates of the vehicle properties. Overall, the agreement is considerably better than that achieved in previous heavy vehicle simulation validation studies [2-6, 11, 13, 14, 29].

4.2 Comparison of Two and Three-Dimensional Models

Response time histories for the 11 DOF model were calculated using the nearside wheel track of the measured lane profile as input. Figure 14 compares nearside tyre force histories and spectral densities of the 21DOF and 11 DOF models (fully laden, 22m/s). Similar results were obtained for the offside wheel track.
Agreement between the two models is good in the region of the sprung mass modes. At this speed, sprung mass roll modes are not excited significantly (section 2.3), and therefore the two models are expected to give similar results. (The two models give identical results when the 21 DOF model has a symmetrical road profile input).

There is significant difference between the tyre force responses of the two models in the region of the unsprung mass modes (above 10Hz). This is because the unsprung mass bounce and roll modes are excited by the road profile and they do not have similar frequencies and damping.

In summary, a two-dimensional model may be satisfactory for predicting the tyre forces of a heavy vehicle if:

i) the vehicle speed is high enough to prevent excitation of sprung mass roll modes, and

ii) the unsprung mass modes have similar frequencies and damping in bounce and roll, or the contribution of unsprung mass modes to the tyre forces is small.

The benefits of using a two-dimensional model are significantly lower computation time and simpler model creation. Simulation of 20 seconds real time for the three-dimensional model required 500 seconds of CPU time, compared with 170 seconds for the two-dimensional model.
5 CONCLUSIONS

1. Tests were performed on a leaf-sprung articulated vehicle. Analysis of the measured tyre forces and road profile indicated that under normal operating conditions sprung mass roll motions do not contribute significantly to tyre forces. The contribution of unsprung mass modes to the tyre forces was also small. The effect of wheel non-uniformities on tyre forces was negligible.

2. A trailer frame bending mode at 7–10Hz did not significantly influence the dynamic tyre forces. Torsional frame resonance was not apparent, but torsional flexibility of the tractor frame and fifth wheel was thought to influence the roll response of the rear of the tractor frame and the drive axle.

3. The measured responses of the vehicle were compared in time and frequency domains with predictions from a 21 DOF three-dimensional non-linear model. Agreement was generally good, and better than that achieved in previous studies.

4. The main sources of error in the simulation were thought to be the trailer suspension model properties and the assumption of rigid sprung masses.

5. The tyre force responses of an 11 DOF two-dimensional model were compared with those of the three-dimensional model. The results suggest that the resonant frequencies and damping of the unsprung mass roll modes could be significantly different from those of the bounce modes. It may therefore be necessary to use a three-dimensional model when the unsprung mass roll modes contribute significantly to dynamic tyre forces.

6. A two-dimensional model should be satisfactory for predicting the tyre forces of typical leaf-sprung articulated vehicles with well damped suspension modes, operating under typical conditions of speed and road roughness.
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All the figures in this report are previously published in [36]. The authors are grateful to Swets and Zeitlinger BV for permission to reproduce them here.

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Figure 1 The instrumented test vehicle, shown in the half laden condition.
Figure 2 Schematic view of the trailer suspension.
Figure 3 Displacement spectral densities of the test lane, and the corrected ISO classifications (see section 2.2).

--- rearside wheel path
----- offside wheel path
(a) Roll/bounce ratio against wavenumber.

(b) Roll/bounce ratio on a radial plot. A radial line drawn through the desired speed and frequency combination determines the corresponding roll/bounce ratio.

**Figure 4** Ratio of roll displacement spectral density to bounce displacement spectral density of the test lane.
Figure 5  Bounce and roll components of tyre forces - fully laden, 22m/s.

(a) Drive axle.

(b) Leading trailer axle.
Figure 6 Bounce and roll acceleration spectral densities - fully laden, 22m/s.

--- bounce (units: (m/s^2)^2/Hz)  --- roll (units: (rad/s^2)^2/Hz)

(a) Rear of tractor frame.
(b) Rear of trailer frame.
(c) Drive axle.
(d) Leading trailer axle.
(a) Half laden.

(b) Fully laden.

Figure 7 Trailer bending acceleration.
Figure 8 21DOF three-dimensional whole vehicle model. For clarity, the force elements necessary for simulating the trailer suspension radius arms are not shown. A double-headed arrow indicates angular motion or torque.
Figure 9 Measured and simulated behaviour of a single trailer leaf spring.
(a) Trailer suspension geometry. (All displacements are relative to the trailer sprung mass).

(b) Forces acting on the idealised spring and axle.

Figure 10 Trailer suspension model.
measured leading axle tyre force
○○○

measured trailing axle tyre force
△△△

simulated tyre force
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(a) Test with low friction PTFE inserts in the slipper end mountings.

measured leading axle tyre force
○○○

measured trailing axle tyre force
△△△

simulated tyre force
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(b) Test with standard slipper ends (metal to metal sliding).

Figure 11 Measured and simulated trailer tilt tests. The measurements are from tests performed by TRRL. The horizontal axis is the road profile displacement seen by the leading axle in the simulation.
Figure 12 Measured and simulated nearside (ns) tyre force histories - fully laden, 22m/s.

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(a) Steer axle.

(b) Drive axle.

(c) Leading trailer axle.

(d) Trailing trailer axle.
Figure 13 Measured and simulated spectral densities - fully laden, 22m/s.

--- measured

--- --- simulated
Figure 14 Tyre force responses from 21 DOF and 11 DOF models - fully laden, 22m/s.

--- 21 DOF three-dimensional model    --- 11 DOF two-dimensional model