Assessing The Relative Road Damaging Potential Of HGVs

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ABSTRACT

Assessing the road damaging potential of heavy vehicles is becoming an increasingly important issue. In this paper, current vehicle regulations and possible future alternatives are reviewed, and are categorized as tests on individual axles and whole vehicles, and 'direct' and 'indirect' tests. Whole vehicle methods of assessing road damaging potential accurately are then discussed. Direct methods are investigated (focussing on using a force measuring mat), and drawbacks are highlighted. Indirect methods using a transient input applied to individual axles are then examined. Results indicate that if non-linearities are accounted for properly, indirect methods of assessing whole vehicle road damaging potential could offer the required accuracy for a possible future test procedure.

1. INTRODUCTION

Assessing the road damaging potential of heavy vehicles is becoming an increasingly important issue. The cost of repairing roads runs into billions of pounds per year in the U.K. alone [1]. Recent research has highlighted the dynamic component of heavy vehicle tyre forces as a significant contributing factor to road damage. Not only can this component be a sizeable proportion of the mean tyre force [2, 3, 4], but high 'spatial repeatability' of tyre forces from different vehicles [5] means that certain points along the road will continually be subjected to peak dynamic tyre forces, thus increasing the damage incurred at those points. Legislation has been in place for many years limiting the static weights of whole vehicles and individual axles [6], but only in the last few years have regulations controlling dynamic tyre forces started to be enacted [7].

Section 2 reviews current and possible future procedures for assessing dynamic tyre forces, and categorizes these as 'direct' and 'indirect' methods for whole vehicles and individual axles. It is argued that the current parametric test used in the EC is unrealistic, and design restrictive [8]. The two alternative methods which could be used to test whole vehicles are then examined. Direct test methods are discussed in section 3. One method uses a load measuring mat containing capacitive strip sensors. Tyre force histories for each axle of a vehicle can be measured directly on a typical road profile [9, 10]. Section 4 describes two indirect test methods. These involve exciting each axle of the vehicle in turn with a transient input, and measuring the responses. The measured transient responses can be used in conjunction with a computational scheme to predict the dynamic behaviour of the whole vehicle to a typical road profile. Finally in section 5, conclusions are drawn regarding the practicality and the accuracy of different methods,.

2. CURRENT AND PROPOSED VEHICLE ASSESSMENT METHODS

Figure 1 gives an overview of different methods available for assessing the relative road damaging potential of heavy vehicles. The horizontal axis is divided into tests of individual axles (such as the drive axle), and tests of whole vehicles. The most realistic test will include all axles of a vehicle, because all axles interact, and contribute to road damage to some extent [11]. Some researchers believe that the axles are independent and can be tested seperately, thereby considerably simplifying the testing procedures.

	Individual axles	Whole vehicle
Direct methods	Single axle shaker rig [15, 29] Instrumented axle [4, 19, 22, 23, 30]	**Load measuring mat** [9, 10, 31] Road simulator [32, 33] Instrumented vehicle [2, 17] Primary road response [34, 35, 36]
Indirect methods	Parametric test [7] Design based criterion	**Linear method** [18] **Non-linear method**

Figure 1. Methods of assessing 'road-friendliness'.

The vertical axis of figure 1 is divided into 'direct' and 'indirect' test methods. Direct test methods are those where a tyre force or road response history is recorded directly for the vehicle subjected to a realistic road roughness. Indirect test methods require a simple test to characterise the vehicle/axle dynamics in some way, and from this test, road damaging ability is estimated. The figure shows currently used and possible future test methods, and previous research into each method. Cost, complexity and accuracy of testing procedures generally increase from bottom left to top right of the figure.

Current legislation falls into the category of indirect tests on individual axles (bottom left). Two test methods are noted: 'parametric' tests and 'design-based' criteria (or 'type approval' tests). Design-based criteria have been in existence for a number of years, and are the simplest form of testing. Certain types of suspension group are either banned (as in Australia), or allowed a reduced weight limit (as in Germany and the U.K. [12, 13]) . Parametric tests measure vehicle parameters such as *natural frequency* and *damping ratio*. This type of test has been recently introduced in the European Community [7]. The procedure can provide some information about suspension performance, but there is no clear relationship between the measured parameters and road damage. For some suspensions however, the body bounce frequency can be correlated with the Dynamic Load Coefficient (*DLC*) [14] which is a measure of RMS dynamic tyre force.

Indirect tests of individual axles represent the least realistic form of testing; for example the EC step test has been shown under certain circumstances to 'pass' vehicles that cause a higher level of road damage than vehicles that 'fail' [11]. An assessment procedure should be objective, and not design restrictive. For this to be achieved, Cebon [8] provided a list of ten criteria that 'road-friendliness' assessment tests should ideally meet. Direct tests on whole vehicles can most easily meet these criteria as conditions most nearly represent those on the highway, but conversely are the most expensive methods. Direct tests on individual axles, and indirect tests on whole vehicles provide compromise solutions.

One method of carrying out direct tests on individual axles is to use instrumented axles, using strain gauged axles or instrumented wheel hubs. An alternative test method investigated by de Pont [15] is to use a hydraulic actuator to excite one axle of a heavy vehicle in an attempt to recreate the dynamic tyre forces measured for that axle on a typical road.

Measuring primary road response (stress, strain etc) has the advantage that the response can be directly related to theoretical road damage, but significant disadvantages due to (1) the large number of sensors needed, (2) the sensitivity of road response to environmental conditions (especially temperature), (3) the sensitivity to vehicle tracking, and (4) the non-uniformity of road response [8]. The other three test methods within this category all measure dynamic tyre forces. These can be converted to theoretical road damage using road response models [16]. All axles of a vehicle can be instrumented, and this method has been used by a number of researchers [2, 17] to investigate tyre forces. A road simulator (hydraulic vehicle shaker) offers a number of advantages including a laboratory environment and flexibility of input conditions. The main drawback is the capital and running costs of the equipment required. The other option for this class of test is to use a load measuring mat. This is discussed in more detail in section 3.

The final category of test in figure 1 is indirect testing of whole vehicles. Best [18] investigated a linear technique for estimating dynamic tyre forces in the frequency domain from tests performed on a single axle, and work is currently underway at Cambridge investigating two alternative methods using time domain calculations. These two methods are discussed further in section 4.

3. WHOLE VEHICLE DIRECT TESTS WITH A FORCE MEASURING MAT

As shown in figure 1, using a force measuring mat is one way of directly measuring the tyre force histories for all the axles of a heavy vehicle traversing a road profile. The main advantages of the load measuring mat are that the system is relatively inexpensive (compared with a road simulator), and once the mat has been installed, many vehicles can be tested quickly (unlike the instrumented axle technique which requires strain gauges to be fitted to all axles of a vehicle before it is tested). Figure 2 compares loads measured by the mat and an instrumented axle, and indicates that the mat is accurate for measuring dynamic loads (from [9]). The main disadvantages of the mat over the other two methods, are that only one road profile can be used in the tests, and the length of the test is governed by the number of sensors in the mat.

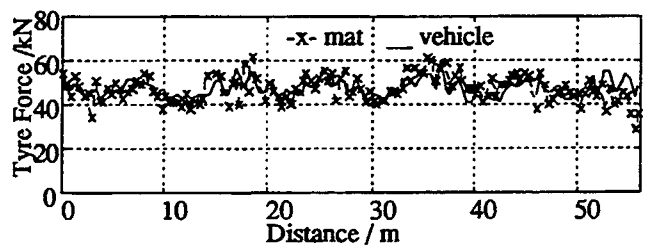

Figure 2. Tyre force history for axle traversing the force measuring mat (measured by mat and instrumented axle).

For a mat test (and indeed any other direct test) to be implemented successfully, there are a number of issues that need to be resolved. The most important of these are:

(i) the spacing between sensors (or sample rate),
(ii) the speed at which vehicles traverse the sensor array,
(iii) the length of the sensor array (sample length), and
(iv) the level of road roughness at the test site.

Sensor spacing (or sampling rate) is limited by the high frequency components of the tyre forces. The spacing between sensors must be small enough to record this high frequency component. For heavy vehicles, the wheel-hop vibration mode has the highest natural frequency f_{high}, and can be up to 15 Hz [8]. If the vehicle speed over the array is v, for the high frequency to be measured, the sensor spacing must be less than $v/2f_{high}$. This spacing corresponds to the Nyquist frequency of the wheel-hop vehicle vibration mode. The *maximum* spacing is therefore limited by the *minimum* speed at which vehicles are to be tested. As sampling rates with road simulator and on-vehicle instrumentation are likely to be much higher than the minimum required to measure the wheel-hop vibration mode, this is only likely to be an issue for mat tests. The mats used in tests performed by the authors had a sensor spacing of 0.4 m which was suitable for heavy vehicles travelling at highway speeds.

The speed of the vehicles during the test also affects the response of the vehicle. The vibration amplitude of a particular mode depends on road roughness and vehicle speed, thus affecting the ranking of different vehicles. The effect can be illustrated using a simple two degree of freedom model such as that shown in figure 3. Figure 4 shows the front tyre force PSDs for this model travelling at a range of typical highway speeds (40-60 mph). The different responses are caused by 'wheel-base filtering' ([19] describes this phenomenon). This figure shows that very different responses occur even with the typical range of highway speeds. Cole [20] examined the effect of wheel-base filtering using a four degree of freedom model with different suspension stiffnesses. He found that the ranking of different suspensions in terms of road damage done by the whole vehicle varied with vehicle speed due to wheel-base filtering. In order to take this into account, vehicles must be tested over a range of typical highway speeds, and ranked in terms of road damage. This means that individual vehicles will not be discriminated against because a particular test speed was used. This will be the case whatever test method is chosen, and means that vehicle tests at artificially low speeds cannot be used.

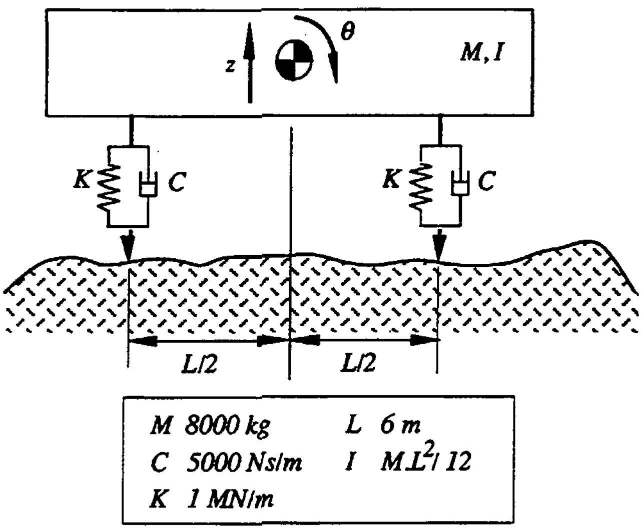

Figure 3. Two degree of freedom model used to investigate wheel-base filtering.

The length of the sensor array (the length of the test) has two effects. Most importantly, the array length must be sufficient to record a representative sample of the dynamic tyre forces generated by the low frequency vibration modes of the vehicle. The lowest natural frequency for heavy vehicles f_{low} is the body bounce frequency, which can be as low as 1.5 Hz [8]. For a required level of accuracy (n low frequency cycles), the minimum sensor array length for a vehicle speed v is given by nv/f_{low}. Consequently, the *minimum* array length is governed by the *maximum* vehicle test speed. Sampling times using a road simulator or on-vehicle instrumentation are relatively easily extended, so this is only likely to be an issue for mat tests.

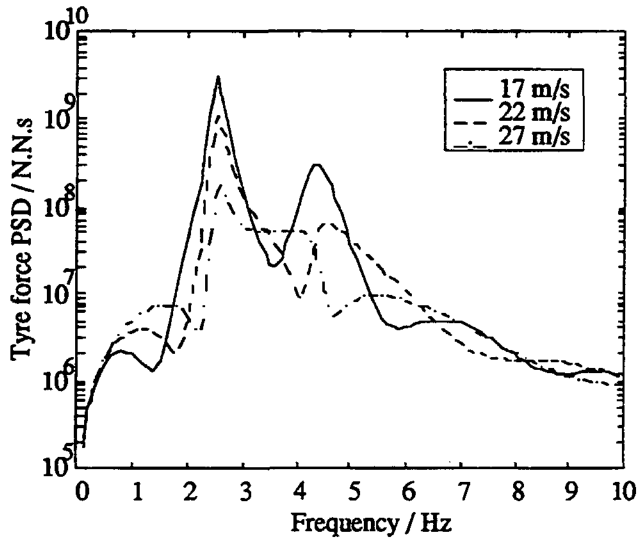

Figure 4. Comparison of front tyre force PSDs for model travelling at three typical highway speeds (40, 50 and 60 mph). Note: data is scaled by ratio of velocities to give magnitude of response at 22 m/s.

The second effect of mat length is important for the repeatability of tests between different sites, and for accurately relating the results of a test to highway conditions. If the roughness of a highway is considered to be a stationary random process of infinite length, finite length test sites with the same nominal roughness will give a range of results, due to the statistics of a finite length process [21]. The variability of results between test sites will increase for shorter array lengths. The quarter-car model shown in figure 5 was used to investigate this effect.

Figure 6 shows how the variation in simulated 95th percentile tyre force varies as a function of mat length. The dotted lines give theoretical 95th percentile limits. The roads used in this simulation all had the same nominal roughness, modelling a principal road. It is clear from figure 6 that, the statistical variation decreases with mat length (distance). For a mat length of 200 m, the variation (width of band) normalised by mean 95th percentile tyre force level is approximately 7%. If multiple test sites with different roughnesses are used to test vehicles, a 200 m length of mat would be required for suitable accuracy.

The level of road roughness at the test site is also an important variable. Should vehicles be tested over a single roughness of road, and if so what level of roughness should be used? Alternatively, should a range of road profiles be used, and vehicles tested over each roughness in turn? Researchers have drawn two general conclusions about the effects of road roughness on the magnitude of dynamic wheel loads:

(i) Dynamic wheel loads increase with increasing road roughness, although not necessarily monotonically [2, 4].

(ii) Although generic suspension types rank in approximately the same order, some particular suspensions have been identified where ranking changes at different speeds and roughness levels [2, 4, 22, 23].

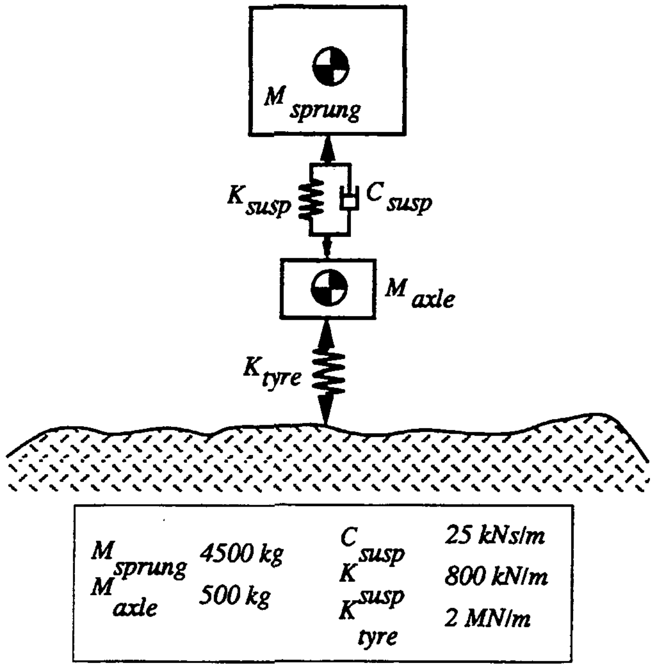

Figure 5. Two degree of freedom quarter car model.

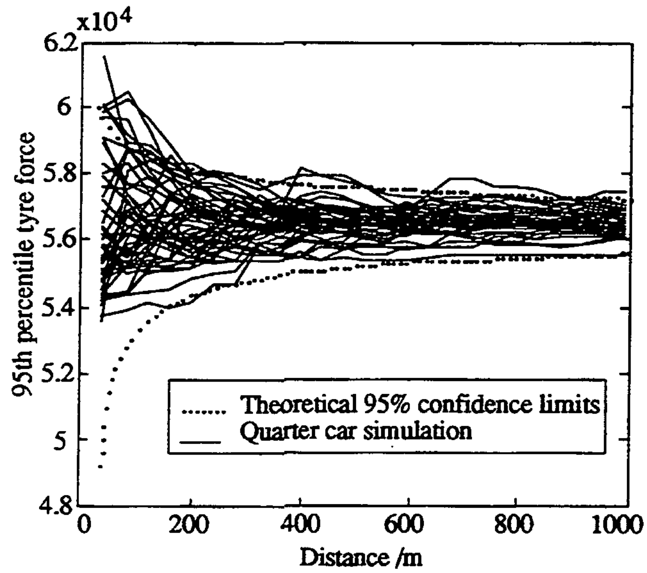

Figure 6. 95th percentile tyre force vs. track length for the vehicle model in figure 5. (Each line corresponds to a different sample of random road input.)

Wheel-base filtering effects require tests to be carried out at highway speeds. However, it is at highway speeds that the variation in dynamic tyre forces is largest, and the largest changes in ranking occur with road roughness particularly due to suspension non-linearities. If an assessment test is designed to rank suspensions over all conditions of roads in common usage by heavy vehicles, a series of mat tests at different roughnesses will be required. For road simulator and instrumented vehicle tests, this does not present much of a problem, because different road profiles can be input or traversed. However, the cost of a mat based assessment test would increase significantly if more than one roughness level was required.

The main conclusions that can be drawn about direct testing methods for whole vehicles, is that each method has particular drawbacks associated with it. For road simulator tests, the main drawback is cost, for instrumented vehicle tests, it is having to instrument each vehicle. For mat tests, it is the inflexibility of the testing procedure.

The following section discusses indirect tests which can be used to 'simulate' direct tests on whole vehicles.

4. WHOLE VEHICLE INDIRECT TEST METHODS

In this section, two indirect methods of assessing whole vehicle road damaging potential are discussed: the linear *convolution method*, and the non-linear *parameter estimation method*. Both methods use a step input test to characterise the dynamics of the vehicle. The measured characteristics are converted into a numerical model of the vehicle, and combined with a measured (or artificially generated) road profile to calculate dynamic tyre force histories for each axle, under representative conditions.

4.1. THE CONVOLUTION METHOD

The convolution integral is applicable to any linear system [24]. Consider a *linear* quarter-car vehicle model which has as its input a road profile, and as its output a dynamic tyre force history. The dynamic tyre force can be found for any road profile using the convolution integral, by measuring the impulse response function of the vehicle system $h(t)$. In this case $h(t)$ will be the tyre force measured due to an impulsive displacement of the road. The impulse response can be convolved with the road profile $u(t)$, to obtain the tyre force history $f(t)$ for the vehicle passing over this road profile according to:

$$f(t) = \int_{-\infty}^{t} u(\tau)h(t-\tau)d\tau \qquad (1)$$

In practice, it is easier to measure the step response function, and this can be convolved with the derivative of the road profile in a similar manner, to obtain the tyre force history. The convolution integral is easily evaluated numerically, and the method can be extended to vehicles with any number of axles. The main problem with this method however, is that the suspension elements of heavy vehicles are *non-linear* to varying degrees, and this makes the convolution method inaccurate.

A computer simulation was used to investigate the effect of typical vehicle suspension non-linearities on the accuracy of the convolution procedure. Two quarter-car models (based on the model in figure 5) were used, one with a leaf spring suspension, and the other with an air suspension. The leaf spring represents the most non-linear type of vehicle suspension in common use and was modelled using a simplified version of the UMTRI leaf spring equation of Fancher et al. [25], which has three independent parameters: friction, stiffness, and an exponential term describing the transition between sliding and sticking of the leaves. The air suspension was modelled as a linear spring in parallel with a validated non-linear hydraulic damper model containing a bi-linear force-velocity characteristic with 'saturation' at high force levels [26].

Figure 7 shows simulated step responses (scaled to a unit step) for step heights of 10 mm, 30 mm and 60 mm for the leaf spring model. (The 10mm and 60mm step responses have been shifted by ±1MN for clarity.) The non-linearity of response is clear, with the relative amplitude increasing and the dominant frequencies decreasing with step height.

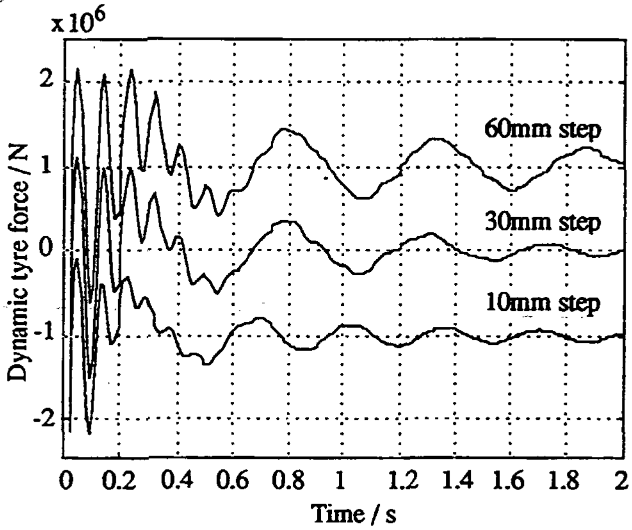

Figure 7. Unit step responses from three step profiles (leaf spring model).

For the simulations used to investigate the convolution method, the road had a roughness given by the spectral density $S(n) = c.n^{-2.5}$, where n is the wavenumber, and c is a constant [27], and the vehicle model traversed the profile at 20m/s. Figure 8 shows tyre force histories for the leaf spring model traversing a road with a roughness of $c = 50\times10^{-8}\, m^{1/2} cycles^{3/2}$. The simulation result was obtained using a non-linear vehicle simulation [28]. The convolved response was obtained by generating the step response for a 30 mm step, using the non-linear vehicle simulation, and convolving this step response with the differentiated road profile. The error between the two tyre force histories can be expressed as an Error Coefficient of Variation (*ECOV*), defined as:

$$ECOV = \frac{\sqrt{\overline{(g(t)-f(t))^2}}}{\sqrt{\overline{(g(t))^2}}}\times100\% \qquad (2)$$

where $g(t)$ is the simulation result,
$f(t)$ is the convolved response, and
$\overline{g(t)}$ is the mean value of $g(t)$.

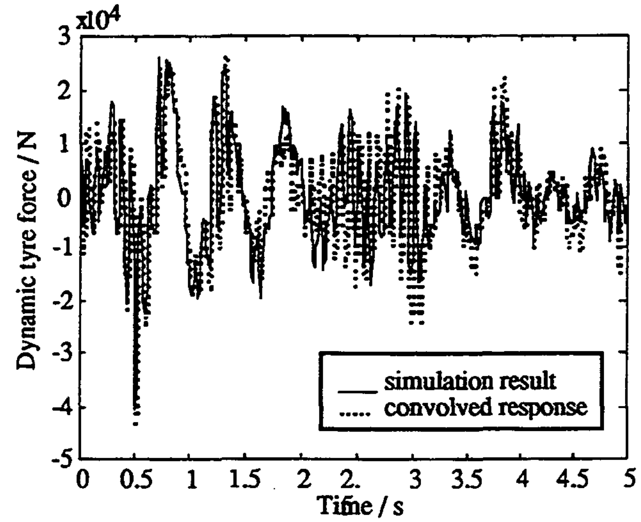

Figure 8. Tyre force histories for leaf spring model traversing medium roughness road.

For the data in figure 8, the *ECOV* is 47%. Figure 9 shows the *ECOV* between simulated and convolved responses for the leaf spring model, with various road roughnesses and step heights. For all the step responses, the tyre of the vehicle model was forced to remain in contact with the road, even if the tyre force became negative. For the simulations on the random road profiles, the tyre was allowed to lose contact with the road surface, although this only occurred for road profiles with a roughness of $c = 250\times10^{-8}\, m^{1/2} cycles^{3/2}$. The effect of this was investigated by including the 'no loss of contact' case for random profiles with this roughness, where the tyre was forced to remain in contact with the road. Loss of contact increases the *ECOV* by over 50% for this vehicle model.

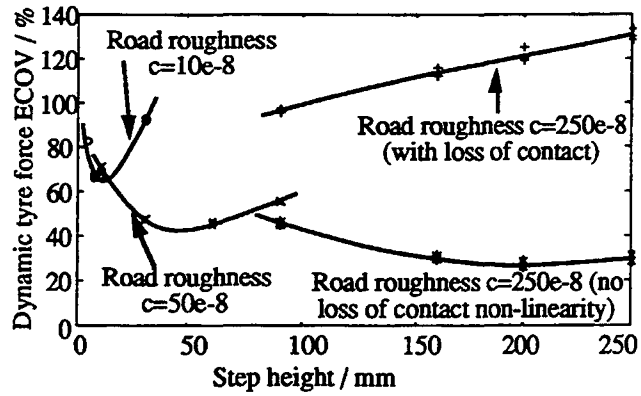

Figure 9. *ECOV* for convolution with different step heights (leaf spring model).

For each road roughness level, there is a step height at which the *ECOV* is minimum. The minimum level occurs when the leaf spring is subjected to similar displacements from the road and the step inputs. As the road roughness increases, the minimum *ECOV* decreases. This is because the force-displacement characteristic of the leaf spring becomes more linear at large amplitudes. The minimum level of *ECOV* (40%) is generated by a vehicle travelling at the maximum speed possible without significant loss of contact between the tyre and road.

Figure 10 shows a similar result for the air suspension model. Similar conclusions can be drawn, but in this case, the minimum *ECOV* is 25%. This is lower than the leaf spring model because the suspension non-linearities are smaller. The *ECOV*s for the two lower roughness roads are a minimum at the same step height due to the different type of non-linearity in the air suspension.

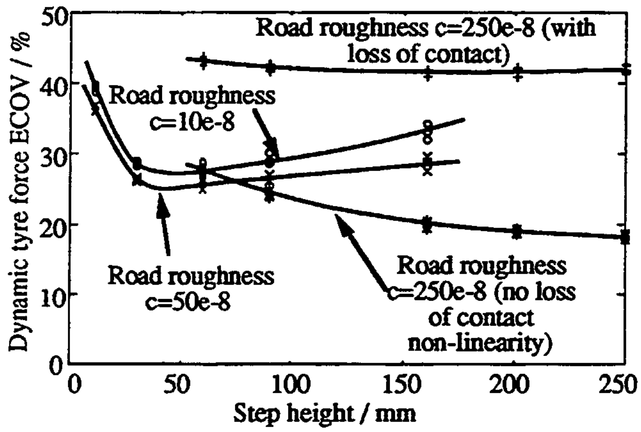

Figure 10. *ECOV* for convolution with different step heights (air suspension model).

In order to consider the effect of *ECOV* on errors between measured and estimated maximum tyre forces, the RMS dynamic tyre force error ($e=\sqrt{(g(t)-f(t))^2}$) must be normalized by the static tyre force F_{stat}. The normalised error with respect to the static tyre force (the Error / Static Load or *ESL*) will therefore be e/F_{stat}. The *DLC* is the ratio between the RMS dynamic tyre force and the static tyre force, so

$$ESL = DLC \times ECOV \quad (3)$$

A typical value of *DLC* for heavy vehicles travelling at highway speeds over a poor quality road is 0.25 [2, 3, 4]. Using this value, an *ECOV* of 40% will give an *ESL* of 10%. It can be shown that for typical heavy vehicles, this leads to a 10% error between the estimated maximum tyre force and the measured maximum tyre force. An *ECOV* of 40% is therefore considered to be a threshold value for an acceptable level of error.

This threshold level indicates that the convolution method can satisfactorily be used for air suspensions, but the errors caused by the non-linearities in leaf spring suspensions are too large for the convolution method to be used.

4.2. THE PARAMETER ESTIMATION METHOD

The parameter estimation method uses a non-linear mathematical model of the test vehicle to generate tyre force histories for each axle, using a non-linear vehicle simulation. The parameters for the model are identified from a simple test on the vehicle. A step input can be used to excite the vehicle, and its response can be measured with accelerometers, displacement transducers and tyre force transducers. These signals are then used as inputs to an algorithm that minimises the error between the model and experimental results, to give a set of estimated vehicle parameters.

Non-linearities can be fully accounted for using this method, thus removing the main source of error in the convolution method. The main error will be due to discrepancies between the vehicle and the model used to simulate the vehicle. It is therefore essential to use a model that has been validated thoroughly. The method has been shown to work exactly for a simulation model, with the correct parameter values being found. Further experimental work is currently being undertaken to assess the accuracy of the method for typical suspensions.

5. CONCLUSIONS

(1) Current regulations for assessing 'road-friendliness' of heavy vehicles is inadequate, because they do not reflect road-damaging potential.
(2) A variety of possible direct, whole vehicle testing methods have been investigated.
(3) Drawbacks are found with the load measuring mat method: different road profiles are required for a test, and for repeatability a mat length of at least 200 m is required. These factors mean that a large number of sensors would be required for this test.
(4) Assessment tests with instrumented vehicles would be impractical because of the large effort required to instrument each vehicle.
(5) Tests with a hydraulic shaker (road simulator) would be too expensive to perform on a regular basis for many vehicles.
(6) Indirect transient tests using the convolution method were found to be sufficiently accurate to assess air suspensions, but not accurate enough to assess leaf spring suspensions, due to non-linearities inherent with this type of suspension.
(7) The parameter estimation method takes into account non-linearities, and thus removes the main source of error with the convolution method. Initial tests indicate the method works satisfactorily.

6. REFERENCES

1. D. E. Newland and D. Cebon, "Roads to Ruin." New Scientist. Dec. 15: 1990.
2. C. G. B. Mitchell, "An experimental assessment of steel, rubber and air suspensions for heavy goods vehicles." *IMechE seminar*, London, 1991.
3. J. H. F. Woodrooffe and P. A. Le Blanc, "Heavy vehicle suspension variations affecting road life." *Proc. ARRB/FORS Symposium on Heavy Vehicle Suspension Characteristics*, Canberra, Australia, 1987.
4. P. F. Sweatman, "A study of dynamic wheel forces in axle group suspensions of heavy vehicles." ARRB Report SR 27, 1983.

5. D. J. Cole et al., "Spatial repeatability of measured dynamic tyre forces." *Submitted to J. Mech Eng Sci (IMechE)*, 1994.
6. J. M. Berry, "Vehicle engineering legislation, including the implications of 1992." *Transnet*, London, 1989.
7. Anon, "Council Directive 92/7/EEC." 1992.
8. D. Cebon, "Interaction between heavy vehicles and roads." *SAE SP–951*, 1993.
9. T. E. C. Potter et al., "An investigation of road damage due to measured dynamic tyre forces." *Proc. I. Mech. E. (Part D)*, 209 pp 9-23 1995.
10. T. E. C. Potter et al., "Road damaging potential of measured dynamic tyre forces in mixed traffic." *To be published in Proc. I. Mech. E. (Part D)*, 1995.
11. D. J. Cole and D. Cebon, "Assessing the road–damaging potential of heavy vehicles." *Proc. I.MechE. (Part D)*, 205 pp. 223–232. 1991.
12. T. Wilding, "The log and short of our weighty laws." *Roadway*, (March): pp. 25-29, 1990.
13. P. von Becker, "Commercial vehicle design - road stress: effect on transport policy decisions." *Strasse und Autobahn (Translated by TRRL as WP/V&ED/87/28)*, 12 pp. 493-498. 1985.
14. L. Gyenes, C. G. B. Mitchell and S. D. Phillips, "Dynamic pavement loads and tests of road–friendliness for heavy vehicle suspensions." *Heavy Vehicle Systems, Int. J. Vehicle Design*, 1(4): pp. 381-395, 1992.
15. J. de Pont, "Experiences with simulating on-road heavy vehicle suspension behaviour using servo-hydraulics." *Engineering Foundation Conference on Vehicle-Road and Vehicle-Bridge Interaction*, Noordwijkerhout, 1994.
16. W. J. Kenis, J. A. Sherwood and T. F. McMahon, "Verification and application of the VESYS structural subsystem." *Proc. 5th Int. Conf. on the Structural Design of Asphalt Pavements*, 1 pp. 333-345. 1982.
17. D. J. Cole and D. Cebon, "Simulation and measurement of dynamic tyre forces." *Proc. 2nd Int. Symp. on Heavy Vehicle Weights and Dimensions*, Kelowna, Canada, 1989.
18. A. Best and A. Schlesinger, "Predicting HGV road loads from rig tests with inputs on one axle at a time." *Engineering Foundation Conference on Vehicle-Road and Vehicle-Bridge Interaction*, Noordwijkerhout, Netherlands, 1994.
19. M. Sayers and T. D. Gillespie, "Dynamic pavement/wheel loading for trucks with tandem suspensions." *Proc. 8th IAVSD Symposium on the Dynamics of Vehicles on Roads and on Railway Tracks*, Cambridge, MA, Swets and Zeitlinger, 1983.
20. D. J. Cole and D. Cebon, "Truck suspension design to minimise road damage." Report CUED/C-MECH/TR64, Cambridge University Engineering Dept. 1994.
21. J. S. Bendat and A. G. Piersol, "*Random Data: Analysis and Measurement Procedures.*" John Wiley & Sons, Inc, 1971.
22. W. D. Hahn, "Effects of commercial vehicle design on road stress - quantifying the dynamic wheel loads for stage 3: single axles, stage 4: twin axles, stage 5: triple axles, as a function of the springing and shock absorption system of the vehicle." 1987.
23. J. H. F. Woodrooffe, P. A. LeBlanc and K. R. LePiane, "Effects of suspension variations on the dynamic wheel loads of a heavy articulated highway vehicle." Canroad Vehicle Weights and Dimensions Study Vol 11, 1986.
24. D. Newland, "*Mechanical vibration analysis and computation.*" Longman, Singapore, 1989.
25. P. S. Fancher et al., "Measurement and representation of the mechanical properties of truck leaf springs." *SAE Trans.*, Report. 800905 1980.
26. F. H. Besinger, D. Cebon and D. J. Cole, "Damper models for heavy vehicle ride dynamics." *Vehicle System Dynamics* 24 pp35-64, 1995.
27. C. J. Dodds and J. D. Robson, "The description of road surface roughness." J. Sound and Vibration 31(2):, pp. 175-183. 1973.
28. D. J. Cole and D. Cebon, "Validation of an articulated vehicle simulation." *Vehicle System Dynamics*, 21 pp. 197–223. 1992.
29. J. de Pont, "Using servo-hydraulics to assess heavy vehicle suspensions for road wear." *3rd Int. Symposium on Heavy Vehicle Weights and Dimensions*, Cambridge, 1992.
30. A. P. Whittemore et al., "Dynamic pavement loads of heavy highway vehicles." NCHRP Report 105, 1970.
31. D. Cebon and C. B. Winkler, "A study of road damage due to dynamic wheel loads using a load measuring mat." SHRP Report SHRP–ID/UFR–91–518, 1991.
32. L. Gyenes and C. G. B. Mitchell, "Measuring dynamic loads for heavy vehicle suspensions using a road simulator." *Engineering Foundation Conference on Vehicle-Road and Vehicle-Bridge Interaction*, Noordwijkerhout, Netherlands, 1994.
33. G. Hu, "Use of a road simulator for measuring dynamic wheel loads." *SAE Conference on Vehicle/Pavement Interaction, SAE 881194, SP-765*, Indianapolis, SAE, 1988.
34. W. J. Kenis, B. T. Kulakowski and D. A. Streit, "Heavy vehicle pavement loading - a comprehensive testing program." *3rd Int. Symp. on Heavy Vehicle Weights and Dimensions*, Cambridge, Swets & Zeitlinger, 1992.
35. J. T. Christison, "Pavements response to heavy vehicle test program: Part 2 - Load equivalency factors." Canroad Vehicle Weights and Dimensions Study Vol 9,1986.
36. W. D. Hahn, "Effects of commercial vehicle design on road stress - vehicle research results." (Translated by TRRL as WP/V&ED/87/38) 1985.