Introduction References Biography Content

Pages 252-278

METHODS FOR EVALUATING THE DYNAMIC-WHEEL-LOAD PERFORMANCE OF HEAVY COMMERCIAL VEHICLE SUSPENSIONS

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ABSTRACT

The study of the dynamic interactions between heavy commercial vehicles and road pavements, and research into the pavement damaging effects of dynamic loading by heavy vehicles, has progressively increased since about 1980. Recent efforts have attempted to develop fundamental knowledge of these interactions and pavement wear and damage mechanisms to identify possible improvements to both vehicles and pavements, with an overall aim of achieving a nett gain in road transport system productivity, efficiency and safety.

Pavement wear is largely brought about by the interactive influence of pavement surface unevenness, structural variability and the dynamic wheel loads imposed by heavy vehicles. The latter is of considerable importance and has been a subject of ongoing attention, leading to various test methods for evaluating the dynamic wheel load performance of heavy vehicle suspensions. The Council of European Communities, for example, produced a directive in 1992 which contains one definition of a suspension considered to have desirable characteristics and a test protocol to check for acceptable performance. While a number of evaluation procedures set out in the EC directive can be used to prove compliance, these procedures are not entirely equivalent and it is not difficult to show that the “pass/fail” outcome can be procedure specific. The evaluation method has other deficiencies outlined in the paper.

To preserve our road network asset will require that appropriate methods be in place for evaluating the dynamic wheel load performance of existing and new heavy vehicle suspensions. These methods should: a) enable heavy vehicle and suspension manufacturers to evaluate suspension performance at the design and manufacturing stages of equipment development; b) allow operators to maintain equipment so that it functions at the required performance level over the entire service life; and c) provide an effective and easy to implement and administer system of compliance auditing for the regulatory jurisdictions. The paper critically reviews methods for evaluating the dynamic wheel load performance of heavy vehicle suspensions, and includes results of detailed investigations of steel- and air spring suspension types. A practical suspension evaluation method is proposed for widespread use in regulatory test stations and in vehicle maintenance facilities.

INTRODUCTION

The study of the dynamic interactions between heavy commercial vehicles and road pavements, and research into the pavement damaging effects of dynamic loading by heavy vehicles, has progressively increased since about 1980 (Sweatman, 1983; Cebon, 1985; Heath, 1988; Gillespie et. al., 1993; Cebon, 1993). Recent efforts have attempted to develop fundamental knowledge of these interactions and pavement wear and damage mechanisms to identify possible improvements to both vehicles and pavements, with an overall aim of achieving a nett gain in road transport system productivity, efficiency and safety (Gillespie et. al., 1993; OECD, 1997).

It has been found that pavement wear is largely brought about by the interactive influence of pavement surface unevenness, structural variability and the dynamic wheel loads imposed by heavy vehicles (Gillespie et. al., 1993; OECD, 1997). The latter is of considerable importance and has been a subject of ongoing investigation, leading to various test methods for evaluating the dynamic wheel load performance of heavy vehicle suspensions. The application of these methods to new vehicles to determine initial type certification, and to vehicles already in-service is becoming increasingly important, and methods that are both effective and practical are understandably being sought.

This paper examines methods of evaluating the dynamic wheel-load performance of heavy vehicles. To assist with the investigations, computer models were developed that simulate the dynamic response of heavy vehicles to a wide range of excitations, including measured road profile. The models feature complex non-linear behaviour of multi-leaf steel springs, air spring, and hydraulic dampers, and they have been used to study each of the methods presented. Finally, evaluation methods have been reviewed and broadly ranked on the basis of their suitability for suspension type approval and compliance testing.

FACTORS THAT CONTRIBUTE TO PAVEMENT DAMAGE

A wide range of both vehicle and pavement factors contribute to pavement damage, and suspension evaluation methods should consider these where possible. The following list is by no means exhaustive, but is intended to group the significant factors identified by various researchers in recent years.

1) Suspension type (multi-leaf steel, rubber, air, walking beam, etc.);
2) Low and high low frequency vibration characteristics (body bounce and axle hop);
3) Tyre and axle arrangements;
4) Tyre loads and tyre pressures;
5) Travel speed;
6) Wheel-base filtering;
7) Tyre-force time histories;
8) Dynamic interaction between the suspensions of tractors and trailers;
9) Pavement structure and structural variability;
10) Road surface unevenness;
11) Spatial repeatability.

Where possible, methods which are designed to evaluate suspension performance, particularly with respect to pavement damage potential, should address most of the items listed above.

HEAVY VEHICLE DYNAMICS MODELLING

Relatively simple mathematical models of heavy vehicle dynamics were developed and computer simulations were performed to determine vertical dynamic response to a broad range of the possible excitations specified in each of the suspension evaluation methods studied. These included response to road profile (measured and artificial), response to initial conditions (displacements and velocities), and response to sinusoidal sweep-type displacement excitations as well as time-varying force excitations imposed on specific components. Accurate prediction of the dynamic response was considered essential, and simple "quarter-truck" models were used that feature complex non-linear multi-leaf steel springs, air springs, and hydraulic dampers. The use of a quarter-truck model precludes study of prime-mover/trailer dynamic interaction, wheelbase filtering effects, and load sharing within an axle-group. However, for the purposes of this investigation that is concerned primarily with first order suspension effects, a quarter-truck model was considered adequate.

Quarter-Truck Dynamic Model

A description of the quarter-truck models is given in this section together with the parameter data sets defining the various components and the suspension element characteristics. Parameter values for each model were sourced from the literature and in most cases values reported are “average” and “typical” of the suspension type modelled. Overall, system natural frequencies, damping values and general responses to specific excitations were found to be consistent with those reported in the literature.

Mass properties

The parameter set listed in Table 1 is typical for a linear quarter-truck model (Cebon, 1993; Gillespie et. al., 1993; de Pont, 1994; Karagania, 1997). The unsprung mass, m_u, represents one-half of one-axle and is made-up of the combined masses of the axle, hub and wheel rim, brake parts and associated hardware. Tyre stiffness and damping, k_t and c_s, shown, are typical for a dual tyre set. The sprung mass, M_s, represents the portion of the total sprung mass supported by one-half of one-axle. The total axle load of an equivalent vehicle is therefore equal to twice the sum of the quarter-truck sprung and unsprung mass, $2(M_s+m_u) = 8.8$t. The suspension stiffness and damping values, K_s and C_s, reported in Table 1 are not used in this paper, but have been included for the interested reader.

Table 1. Quarter-truck model parameter set

Parameter	Value	Units
Sprung Mass, M_s	3900	kg
Suspension Stiffness, K_s	8.78E+05	N/m
Suspension Damping, C_s	1.75E+04	Ns/m
Unsprung Mass, m_u	484	kg
Tyre Stiffness, k_t	1.49E+06	N/m
Tyre Damping, c_t	1755	Ns/m

Note: K_s and C_s are not used in this study

Suspensions

In order to deal properly with the peculiar non-linear characteristics of heavy vehicle suspensions and accurately predict dynamic response, two suspension models were developed using two separate non-linear spring models, multi-leaf steel and air, and one non-linear hydraulic damper model. Tyres have been treated as linear elements.

Multi-leaf steel spring

The most common type of heavy vehicle suspension is the multi-leaf steel spring, which is available with either flat or tapered leaves. Leaf springs exhibit a high level of friction in their operation that produces very complex force-deflection characteristics. These have been studied in detail by Fancher et. al. (1980), who found the behaviour depends on the nominal stiffness of the spring, and coulomb friction force that is dependent on previous motions and the direction of displacement. **Fig. 1(a)** shows typical force-deflection characteristics of a leaf spring and the key parameters used in the model developed by Fancher et. al. (1980), which has been used in countless simulations and is well documented. Typical force-deflection characteristics from suspension motions of the leaf spring model used in this paper is shown in **Fig. 1(b)** which has the general characteristics and form of the force-deflection characteristics of Fig. 1(a). The leaf spring parameters listed in Table 2 have been sourced from Karamihas and Gillespie (1993).

Table 2. Multi-leaf spring parameters (flat leaf)

Parameter	Value	Units
Upper envelope stiffness	578.0	kN/m
Lower envelope stiffness	473.0	kN/m
Beta parameter	2.03E-3	m

Air spring

The load-deflection characteristics of an air spring are non-linear and the response properties shown in **Fig. 2** are typical. Air spring characteristics depend on the properties of the contained gas, initial pressure and volume, rate at which loads are applied and heat transfer between the contained gas and its surroundings. In general, the load-deflection characteristics will follow the ideal gas law, viz.:

$$PV^{\gamma} = const \qquad (1)$$

where:

P = absolute pressure (Pa)
V = volume (m^3)
γ = polytropic index (-)

When the applied loads change very slowly and there is time for heat transfer to take place between the contained gas and the surrounding surface such that the gas temperature remains essentially constant, the thermodynamic process is isothermal and the polytropic index, γ, is equal to unity. This occurs when a vehicle's load condition changes from empty to laden, for example. On the other hand, when a vehicle is travelling over an uneven surface and the suspension loads are changing very rapidly, very little heat will be exchanged between the contained gas and its surroundings. Under this condition the load-deflection characteristics of the air spring will also follow the ideal gas law, however, the process will be approximately reversible or adiabatic, and the polytropic index, γ, which is gas dependent, will be significantly greater than unity. The polytropic index for air undergoing adiabatic expansion or compression is 1.4. It is important to recognise this fundamental difference because the equivalent spring rate[1] of an adiabatic air spring is approximately 1.4 times that of an isothermal air spring. Rakheja and Woodroofe (1996), for example, have assumed an isothermal process in calculating air spring forces which would produce calculated body bounce natural frequencies about 20% lower than if an adiabatic process had been assumed.

The air spring parameters are listed in Table 3.

Table 3. Air spring parameters

Parameter	Value	Units
Design height	0.35	m
Design pressure	110.0	kPa
Air spring diameter	0.36	m
Polytropic index	1.4	-

The equivalent spring rate of an air spring will increase with applied load, which leads to an almost constant bounce natural frequency largely independent of the sprung mass load. The axle-hop frequency on the other hand will increase with load.

Air springs typically possess very little damping, relying on external means to dissipate energy, such as dampers or hydraulic shock absorbers.

Dampers

Dampers are one of the most complex components of the suspension system. They are non-linear devices that have velocity and excitation-amplitude dependent force generating characteristics. As such, dampers are normally characterised by force-velocity diagrams that show the response to a range of excitation amplitudes and excitation frequencies. One example showing the response of a shock absorber to a range of excitations is given in **Fig. 3**. (A detailed explanation of the workings of shock absorbers is beyond the scope of this paper, however, for the interested reader more information can be found in Segel and Lang (1981), Besinger et. al. (1995), Lang and Sonnenburg (1995), and Duym et. al. (1997)). In its simplest form the response of the shock absorber can be modelled using three damping rates, a low damping rate for bump, C_b, and a high rate for rebound[2], C_{r1}, reducing to a lower rate, C_{r2}, above the saturation velocity, v_{lim}. The three regions described and the break point in the bilinear rebound response at the saturation velocity is seen most clearly in Fig. 3(c).

Hydraulic shock absorbers used on air spring suspensions generally have higher damping levels than those found on multi-leaf steel spring suspensions because of the much lower inherent losses within the air spring and associated hardware. In practice, shock absorbers are not always fitted to multi-leaf steel spring suspensions because hysteretic losses are considered large enough to not warrant their use. For this reason two damper parameter sets were used in the quarter-truck models, one tuned for use on the air suspension, the other for the multi-leaf steel suspension. Hysteresis caused by compliance in rubber bushings contributing to additional losses at the higher frequencies has been ignored in this paper. The parameter values shown in Table 3 are based on information gathered from various sources, including published data reported in the literature (Uffelmann and Walter, 1994; Besinger et. al. 1995; Becher and Siebert, 1996). The force-velocity diagrams for the air suspension shock absorber is shown in **Fig. 4.**

Table 3. Hydraulic Dampers Parameters

Parameter	**Value**	**Units**
Saturation velocity, v_{lim}	0.150	m/s
Air Suspension		
Rebound rate below saturation velocity, C_{r1}	40000	Ns/m
Rebound rate above saturation velocity, C_{r2}	8000	Ns/m
Bump rate, C_b	4000	Ns/m
Multi-Leaf Steel Suspension		
Rebound rate below saturation velocity, C_{r1}	10000	Ns/m
Rebound rate above saturation velocity, C_{r2}	2000	Ns/m
Bump rate, C_b	1000	Ns/m

Tyres

The tyre model consists of a linear spring, k_t, and a linear viscous damping element, c_t. The parameter values listed in Table 1 are for a pair of tyres, or a dual set. Simulations involving travel over measured road profile or artificial bumps included a tyre-enveloping model to accurately reproduce the way the tyre envelops small bumps and the short, sharp unevenness features in the profile. This behaviour has been studied in detail by Gillespie et. al. (1981), and is also reported in recent heavy-vehicle studies (Gillespie et. al., 1993, for example). The tyre enveloping effect was included by applying a 300mm moving average to all road and artificial-bump profiles used as inputs to the quarter-truck models.

Computations

The equations of motion were derived from first principles, rearranged and put into a suitable form, and solved using a fourth-order Runge-Kutta numerical integration method. All calculations and computer simulations were performed using a standard spreadsheet package, giving a high degree of portability. An integration time-step size of between 5ms and 20ms was found to be adequate, and actual step-size depended in part on the type of excitation imposed. For example, where road profile was used as an input to the vehicle model the profile sampling interval and vehicle speed determined the integration step-size.

Response to Road Inputs and Validation

Measured road profiles taken with the ARRB TR Walking profiler (Auff, Tyson and Choummanivong, 1995) were used as inputs to the air- and steel-suspension quarter-truck models. Firstly to establish confidence in the models and perform a validation, and secondly to determine typical response magnitudes to two levels of road roughness. IRI values for the two measured profiles were 2.44 and 4.38 m/km, and these represent average and high unevenness levels.

A number of simulations were performed and the Dynamic Load Coefficient (DLC) was determined for both suspension types, including the air suspension without a damper. **Figs 5(a) and 5(b)** show that the DLC values, and general trends and form of the results, compare well with those reported in the literature (Sweatman, 1983; Gyenes, Mitchell and Phillips, 1992; Gillespie et. al., 1993; Cebon, 1993; Woodroofe, 1996).

The response to road inputs of the steel suspension springs and dampers is shown in **Figs 6(a) to 6(d)** for a typical simulation. Suspension travel for the air suspension spring is approximately 50mm on the high roughness profile and about half this value on the average profile. This is about twice the corresponding magnitude of suspension travel seen in the steel spring shown in the Figs. Damper peak velocities for the steel suspension are approximately 0.4m/s in bump and 0.5m/s in rebound, and marginally higher when compared to air on both roughness profiles. The suspension spring and damper non-linearity can be clearly seen in the response plots, which show, for example, that the range of motion in the damper under certain operating conditions does not cover the full extent of the non-linear characteristics.

REVIEW OF EVALUATION METHODS

A variety of suspension evaluation methods have been developed over the years aimed specifically at testing for heavy-vehicle/pavement interaction characteristics that are know to contribute to increased levels of infrastructure damage (Council of European Communities, 1992; Gyenes, Mitchell and Phillips, 1992; de Pont, 1996; Woodrooffe, 1996; OECD, 1997).

More recently the importance of heavy-vehicle bounce and axle-hop vibration modes in relation to dynamic loading of bridges and frequency matching has received wider attention (Heywood, 1996; Green, 1996; OECD, 1997). As a result of this new work, suspension evaluation methods must now also test for characteristics that can lead to unacceptable bridge loading. This will include evaluation of both the bounce and axle-hop frequencies, the former has importance in the performance of long span bridges, and the latter bridges with short spans.

One of the main requirements of a suspension is that it exhibits good performance across a wide range of operating conditions that are typical of the expected in-service levels. Equally, evaluation methods should be capable of identifying both desirable characteristics and deficiencies at an early stage before type approval has been granted and widespread use has taken place.

Methods

Evaluation methods can be classified under the following broad headings (Gyenes, Mitchell and Phillips, 1992):

a) Parametric – set acceptable limits for dynamic response parameters, eg body bounce and axle-hop natural frequencies and damping, hysteretic losses, etc.;

b) Relative – measure performance relative to known acceptable designs;

c) Simulated – simulate on-road conditions using servo-hydraulic actuators, for example, or a standardised artificial road profile; or use computer models to predict dynamic loads under a range of conditions;

d) Instrumentation – direct response measurements on real roads.

Parametric

Council Directive 92/7/EEC

The Council of European Communities has produced a directive outlining a procedure to test for equivalence between air and non-air suspension systems (Council of European Communities, 1992). The directive states that equivalence to air suspension is recognised when the mean damping ratio D is more than 20% of critical for the suspension in its normal condition with hydraulic dampers in place and operating. A further requirement is that the damping ratio of the suspension with all hydraulic dampers rendered ineffective is not more than 50% of D, and the frequency of oscillation of the sprung mass in free vibration must not be greater than 2 Hz. A recent major study has recommended the sprung mass bounce frequency be reduced to 1.5 Hz in order to help to reduce dynamic loading of pavements (OECD, 1997). Three specific test methods are described in the EC Directive to determine the two suspension parameters of prime interest.

- *pull-down/pull-up methods*

The damping ratio is established either by the "pull-down" method requiring the chassis to be pulled down until the axle load is 1.5 times its static value, or by the "pull-up" method, that requires the sprung mass to be lifted 80mm above the axle. The chassis is suddenly released from these positions and the ensuing oscillations analysed. The two methods are not equivalent, and the pull-down method will generally produce higher damping forces due to the non-linear characteristics of the hydraulic damper, which produce higher damping forces during the initial rebound phase. The work of Uffelmann and Walter (1994) show how the difference in damping estimates can be as large as 70%, and pass/failure is test method specific.

- *80mm step profile*

In the third method proposed, the vehicle is driven over the step profile shown in **Fig. 7** at approximately 5km/h and the transient oscillations once the wheels have left the step are analysed.

The air and steel suspension quarter-truck responses to the step profile are shown in **Figs 8(a) and (b).** The time period from about 0.5 to 2.5s corresponds to motion up and along the step profile. Once the "wheels" leave the ramp at about 2.5s, the initial suspension deflection responses for both air and steel are seen to be very similar in form and magnitude. Hysteresis in the steel is evident and reveals itself as the constant offset in the response, both initially and once the oscillations have decayed to small levels. This is due to migration from the upper to lower envelope stiffness suspension curves and "settling" of the suspension. The characteristic slight increase in oscillation frequency as the response amplitude decays to small levels is also apparent. Over small displacements the effective spring rate of multi-leaf springs may be as high as 3 to 10 times the nominal rate, which accounts for this behaviour, as illustrated in Fig. 1(a) and reported by Gillespie (1985). (Figure 1(b), given early in this paper, is the force-deflection response corresponding to the steel suspension step response shown in Fig. 8(a).)

Fig. 8(b) shows the air suspension oscillation is well damped at both large and small displacement amplitudes, and the frequency of the vibration is about 1.5 Hz.

Constant Displacement Sinusoidal Sweep

A constant amplitude sinusoidal sweep has recently been proposed as a means for assessing the dynamic wheel load performance of heavy vehicle suspensions, and as a practical means of assessing hydraulic damper condition (OECD, 1997). The method involves the application of a constant amplitude sinusoidal displacement input to the wheels using an excitation source, typically in the form of a servo-hydraulic actuator. The sinusoidal input frequency is designed to increase with time and generate excitation frequencies across the full range of interest, usually 0 to 20 Hz. Sweep rates and forms can be easily specified, and they can be linear or exponential. Typical examples are linear sweep rates of 0.5Hz per second (Karagania, 1997) and 5Hz per minute (Woodrooffe, 1996). The resulting vehicle response when plotted as a function of frequency will reveal resonances and damping levels (body-bounce and axle-hop). The following constant amplitude, sinusoidal sweep input was applied to the two quarter-truck models:

$$z(t) = A\sin\left[\pi t\left\{2f_1 + \frac{t}{T}\left(f_2 - f_1\right)\right\}\right] \tag{2}$$

where:

A = Amplitude (m)
f_1 = Start frequency (Hz)
f_2 = End frequency (Hz)
T = End time (s)

The resulting quarter-truck wheel force responses are shown in **Figs 9(a) and (b)**, which are largely consistent with the results reported in Woodrooffe (1996), Karagania (1997) and OECD (1997). Evident are the body-bounce and axle-hop frequencies that correspond to steel and air suspensions, respectively. The asymmetry in the response plot for air is due to the non-linearity in the damper and higher damping level in rebound. This asymmetry is not evident in the results reported in Woodrooffe (1996) or OECD (1997), that are based on actual tests performed on heavy vehicles. The difference is most likely due to the additional compliance and damping in the rubber bushes that provide some isolation between the shock absorber and the suspension system. At such small excitation amplitudes the contribution from the rubber bushes could be expected to be significant, and would need to be taken into account if the 1mm constant amplitude sinusoidal sweep test method were adopted.

Fig. 10 shows the steel suspension deflections for the constant amplitude sinusoidal sweep input, and the deflection magnitude is typical of the deflections for both air and steel suspensions to this type of input. When the result shown in Fig. 10 is compared with the responses to real road inputs, which are shown in Figs 6(a) and (b), it is clear the sweep excitation does not subject the suspension to motions, or forces, typically found on even the smoothest roads. As such, the conclusion by OECD (1997) that this method shows strong promise as a means for assessing the dynamic wheel load performance of heavy vehicle suspensions should be reconsidered.

Force Sinusoidal Sweep

A force frequency sweep applied to the unsprung mass is considered a viable alternative to the constant amplitude displacement sweep, as a practical and relatively inexpensive and simple method of assessing damper condition. Two alternative force sweep excitations were evaluated, a constant amplitude force sweep and a force sweep with the amplitude increasing with frequency.

- *Constant force amplitude*

A constant force sweep was applied to the axles of the quarter-truck models. The force magnitude was limited to values that could only be generated by portable, commercially available electrodynamic shakers. The force magnitudes (typically of the order of 450N) were found to be too small to be effective. Thus, the constant force frequency sweep was not pursued any further.

- *Increasing force amplitude*

Various devices are used in the vibration measurement industry for generating force excitations. One of the more common devices features two contra-rotating masses that have their mass centres offset from the axis of rotation to produce an “out-of-balance” force. Mass rotations are synchronised in such a way that the lateral force generated by one mass is exactly balanced by the other mass leaving only a sinusoidal vertical force. The magnitude of this force, *F*, can be calculated from the following expression:

$$F = m\omega^2 R \sin \omega t \tag{3}$$

where:

m = Total rotating mass (kg)
ω = Angular velocity (radians/s)
R = centre-of-mass offset (m)

Equation (3) shows that the force magnitude increases with the square of the rotational speed. A force frequency sweep was applied to the axle (unsprung mass) of the air suspended quarter-truck model with the rotational speed, ω, equal to the sweep frequency. Frequency was increased according to Equation (2). The amplitude *A* was set equal to $m\omega^2 R$, and so the force amplitude increased as the square of the rotational speed. **Figs 11(a) to (d)** show the resulting axle displacements and accelerations of the air suspended quarter-truck model both with and without dampers. Without dampers the axle displacement response of approximately ±10mm is considered large enough to be visible with the naked eye, and thus specialised equipment would not be necessary to identify air suspensions with totally inoperative dampers. Equally, the acceleration response at the axle-hop frequency is large enough that it would be possible to identify increasing levels of hydraulic damper deterioration with the use of accelerometers. The method is yet to be fully tested by application to heavy vehicles, but these initial results are very encouraging.

Simulated

Road Simulator

Applying standardised or measured road wheel inputs to suspensions with servo-hydraulic actuators is widely used by heavy vehicle manufacturers in product development to assess vehicle performance (Prem, 1987). In a recent major study it was found to be effective for evaluating suspension response to a wide range of conditions and replicating road inputs to the vehicle (OECD, 1997). However, in that study it was concluded that the high capital cost of establishing a full scale shaker system would be prohibitive for widespread use or in practise as a means of assessing suspension performance at set intervals during the life of the suspension. This method would be suitable for type certification, for example, and only one or a few such facilities would be required. Other methods presented in this paper, described earlier, could be used for in-service compliance testing. While there is evidence to suggest the method has not yet been perfected, see De Pont (1996) for example, the advantages offered would strongly support continuing research and development effort.

Computer Simulations

The development of multi-body computer codes over the last two decades has been significant (Kortum and Sharp, 1993), allowing large and complicated vehicle systems to modelled in great detail and their performance predicted with high accuracy (OECD, 1997; Elischer and Prem, 1997). In this approach computer models would be used to predict the dynamic response of the entire vehicle, or select sub-system, to standardised road inputs. The complexity of the computer models would depend to large degree on the specific problem being addressed. Some of the more advanced packages, for example, feature flexible elements, bushings, non-linear suspension elements, and comprehensive tyre models (MDI, 1998). However, the models when developed require validation before they can be regarded as reliable enough to be used for legislative purposes (Gyenes, Mitchell and Phillips, 1992). Modelling *per se* can assist with the evaluation of the suspension design but should not be the sole means of certification.

Instrumentation

Instrumentation for measuring dynamic wheel loads can be either vehicle based or pavement based. Developmental work and tests by Gyenes and Mitchell (1996) suggest vehicle based systems can measure dynamic wheel loads to an accuracy of 1-2%. Each vehicle tested would have to be fitted with instrumentation, limiting it to type certification.

An array of closely spaced load sensors installed along a section of pavement is another method of measuring dynamic wheel loads, and a semi-permanent installation on sections of pavement covering a range of road roughness levels could be used to test any number of suspensions and vehicles across a broad range of operating conditions. Work reported in Cole and Cebon (1993), for example, indicates an average sensor error for measured instantaneous loads of less than 4% RMS is possible. The method would test all axles and the interactions between tractor and trailer suspension groups, and whole-vehicle effects.

Studies of accuracy of WIM systems (Gillespie and Karamihas, 1996) clearly show that axle weight estimates on heavy vehicles fitted with suspensions known to produce large dynamic wheel loads, such as walking beam suspensions, or air suspensions with deteriorated shock absorbers, will exhibit higher error. There would therefore appear to be a direct relationship between dynamic wheel load performance of specific suspension types (and condition) and WIM accuracy. This suggests that WIMs could be used as an in-service screening device to identify trucks with poorly maintained suspensions. Once detected these trucks could be further tested using one of methods described in this paper.

SUMMARY AND CONCLUSIONS

Various methods of evaluating the dynamic wheel load performance of heavy vehicles have been presented and reviewed. Where possible they were studied in detail with “quarter-truck” models that feature complex non-linear multi-leaf steel springs, air springs, and hydraulic dampers. The models have helped demonstrate certain aspects of the methods. While the use of quarter-truck models precludes study of prime-mover/trailer dynamic interaction, wheelbase filtering effects, and load sharing within an axle-group, for the purposes of this investigation that is concerned primarily with first order suspension effects, the quarter-truck models proved to be very useful.

One of the main requirements of a suspension is that it exhibits good performance across a range of conditions that are typical of those expected in-service. Equally, evaluation methods should be capable of identifying both the desirable characteristics and the deficiencies at an early stage so they can be corrected before type approval has been granted and widespread use has taken place. Methods for in-service compliance checking should be simple and practical.

The methods have been put in one of two categories depending on whether they are considered suitable for type approval testing or in-service compliance checks. The following methods are considered suitable for type approval because they can replicate the full range of operating conditions, and they are sensitive to whole-vehicle and suspension effects:

- Road simulators (servo-hydraulic actuators, artificial surfaces);
- Vehicle or pavement based instrumentation (wheels or axles, multiple-sensor load sensors);
- Computer simulations supplemented by validation trials.

The following methods are considered suitable for in-service compliance checks, either in their present form or after further development, and would be used in conjunction with information obtained from type approval tests:

- Constant amplitude frequency sweep;
- Increasing force frequency sweep;
- EC bump test (including pull-up/pull down).

Introduction Content

Introduction Content

REFERENCES

Auff, A.A., Tyson, G. and Choummanivong, L. (1995). “Evaluation of the Road Roughness Measuring Capability of ARRB’s Prototype Walking Profilometer.” *ARRB Transport Research Ltd., Research Report No. 266. 11 pages including 9 figures and 6 tables.*

Council of European Communities (1992). Annex III of Council Directives 85/3/EEC on the weights, dimensions and certain technical characteristics of certain road vehicles, February 1992. *Council of European Communities, Brussels.*

Becher, H.O. and Siebert, A. (1996). “Electronic Shock Absorber Control as Part of an Integrated Suspension Management for Commercial Vehicles.” *Heavy Vehicle Systems, Int. J. of Vehicle Design, Vol. 3, Nos 1-4, pp. 36-54.*

Besinger, F.H.D., Cebon, D. and Cole, D.J. (1995). “Damper Models for Heavy Vehicle Ride Dynamics.” Vehicle Systems Dynamics, 24 pp. 35-64.

Cebon, D. (1985). “An Investigation of the Dynamic Interaction Between Wheeled Vehicles and Road Surfaces.” Ph.D. Thesis, University of Cambridge.

Cebon, D. (1993). “Interaction between Heavy Vehicles and Roads”. *Society of Automotive Engineers Inc., Special Publication SP-951, 81p.*

Cole, D.J. and Cebon, D. (1993). “Performance and Application of a Capacitative Strip Tyre Force Sensor.” *Proceedings of the 6th International Conference on Road Traffic Monitoring and Control, IEE, London, UK.*

de Pont, J. (1994). “Road Profile Characterisation.” *Transit New Zealand Research Report no. 29, Transit New Zealand*.

de Pont, J. (1996). “Experiences with Simulating On-Road Heavy Vehicle Suspension Behaviour using Servo-Hydraulics.” *Heavy Vehicle Systems, Int. J. of Vehicle Design, Vol. 3, Nos 1-4, pp. 127-139.*

Duym, S., Stiens, R. and Reybrouck, K. (1997). “Evaluation of Shock Absorber Models.” *Vehicle Systems Dynamics, 27, pp. 109-127.*

Elischer, M.P. and Prem, H. (1997). “Validation of Heavy Vehicle Dynamics Modelling using ADAMS - Lane Change Manoeuvre.” *ARRB Transport Research Ltd., Austroads Project NRUM 9501B, Contract Report OC6503V (Restricted Circulation)*.

Fancher, P.S., Ervin, R.D., MacAdam, C.C. and Winkler, C.B. (1980). “Measurement and Representation of the Mechanical Properties of Truck Leaf Springs”. *Society of Automotive Engineers Inc., Paper No. 800905, 16p*.

Gillespie, T.D., Sayers, M.W. and Segel, L. (1981). “Calibration of Response-Type Road Roughness Measuring Systems.” *National Cooperative Highway Research Program Report 228, Transportation Research Board, Washington D.C.*

Gillespie, T.D. (1985). "Heavy Truck Ride". *Society of Automotive Engineers Inc., Special Publication SP-607, 68p.*

Gillespie, T.D., Karamihas, S.M., Sayers, M.W., Nasim, M.A., Hansen, W. and Cebon, D. (1993). "Effects of Heavy-Vehicle Characteristics on Pavement Response Performance." *National Cooperative Highway Research Program Report 353, Transportation Research Board, Washington D.C.*

Gillespie, T.D. and Karamihas, S.M. (1996). " Feasibility of Multiple-Sensor Weighing for Increased Accuracy of WIM." *Heavy Vehicle Systems, Int. J. of Vehicle Design,* Vol. 3, Nos 1-4, pp. 149-164.

Green, M.F. (1996). "Assessment of Bridge-Friendliness of Heavy Vehicle Suspensions." *Heavy Vehicle Systems, Int. J. of Vehicle Design,* Vol. 3, Nos 1-4, pp. 165-179.

Gyenes, L., Mitchell, C.G.B. and Phillips, S.D. (1992). "Dynamic Pavement Loads and Tests of Road Friendliness for Heavy Vehicle Suspensions." *Proceedings of the 3rd International Symposium on Heavy Vehicle Weights and Dimensions, Cambridge, UK.*

Gyenes, L. and Mitchell, C.G.B. (1996). "Measuring Dynamic Loads for Heavy Vehicle Suspensions." *Heavy Vehicle Systems, Int. J. of Vehicle Design,* Vol. 3, Nos 1-4, pp. 191-221.

Heath, A.N. (1988). "The Mechanics of Dynamic Pavement Loading by Heavy Vehicles." *Ph.D. Thesis, University of Melbourne*.

Heywood, R.J. (1996). "Influence of Truck Suspensions on the Dynamic Response of a Short Span Bridge over Cameron's Creek." *Heavy Vehicle Systems, Int. J. of Vehicle Design,* Vol. 3, Nos 1-4, pp. 222-239.

Karagania, R.M. (1997). "Road Roughness and Infrastructure Damage." *Master of Engineering Thesis, Queensland University of Technology*.

Karamihas, S.M. and Gillespie, T.D. (1993). "Characterizing Trucks for Dynamic Load Prediction." *Heavy Vehicle Systems,* Special Series*, Int. J. of Vehicle Design, Vol. 1, No. 1, pp. 3-19.*

Kortum, W. and Sharp, R.S. (1993). "Multibody Computer Codes in Vehicle System Dynamics." *Supplement to Vehicle System Dynamics, Volume 22.*

Lang R. and Sonnenburg R. (1995). "A Detailed Shock Absorber Model for Full Vehicle Simulation." *Presented at the 10th European ADAMS Users' Conference, Frankfurt, 14-15 November.*

MDI (1998). "Internet website - http://www.adams.com/" Mechanical Dynamics, Inc., Ann Arbor, Michigan, USA.

OECD (1997). "Dynamic Interaction of Vehicle & Infrastructure Experiment – Asia-Pacific Concluding Conference." *OECD DIVINE Project Concluding Conference, Melbourne, November 5-7.*

Prem, H. (1987). "Overseas Trip Report September 1987: Profilometry, Road Unevenness, and Heavy Vehicle Research." *ARRB Internal Report AIR 435-1.*

Rakheja, S. and Woodrooffe, J. (1996). "Role of Suspension Damping in Enhancement of Road Friendliness of Heavy Vehicles." *Heavy Vehicle Systems, Int. J. of Vehicle Design, Vol. 3, Nos 1-4, pp. 363-381.*

Segel, L. and Lang. H.H. (1981). "The Mechanics of Automotive Hydraulic Dampers at High Stroking Frequencies." *Vehicle Systems Dynamics, 10, pp. 82-85.*

Sweatman, P.F. (1983). "A Study of Dynamic Wheel Forces in Group Suspensions of Heavy Vehicles." *Australian Road Research Board, special report SR27, 65p.*

Uffelmann, F. and Walter W.D. (1994). "Protecting Roads by Reducing the Dynamic Wheel Loads of Trucks." *ADAMS User Conference, Frankfurt.*

Woodrooffe, J. (1996). "Heavy Vehicle Suspensions: Methods for Evaluating Road-Friendliness." *National Research Council Canada, Centre for Surface Transportation Technology, Final Report.*

Introduction Content

Introduction Content

AUTHOR BIOGRAPHIES

Hans Prem has a Bachelor's degree and a Ph.D. in Mechanical Engineering from the University of Melbourne, which he received in 1979 and 1984, respectively. His interests are principally in vehicle dynamics and road roughness research. Hans first joined ARRB Transport Research in 1984, and was responsible for development of the prototype version of the ARRB laser profiler. In 1989 Hans left ARRB TR to take up a position in BHP Research, as a specialist in heavy haulage vehicles in mining equipment research, where he played a central role in the design and manufacture of a new and innovative off-highway haulage truck (220t payload capacity). He was also involved in blast-hole drill-rig automation and airborne reconnaissance technology. Hans returned to ARRB TR in 1997, where he is Research Coordinator of the Heavy Vehicles and Mining business area.

Rod George joined ARRB Transport Research in 1977 after serving a traineeship at the Aeronautical Research Laboratories in Melbourne. Rod is a member of the Heavy Vehicles and Mining research team at ARRB Transport Research and was appointed a Vehicle Design Associate of the International Journal of Vehicle Design. He has led many heavy vehicle research projects, and has a keen interest in heavy vehicle suspension performance, truck/trailer dynamics and performance-based standards for large combination vehicles. Rod is completing a Masters Degree in Engineering at Swinburne University of Technology.

John McLean was educated at Melbourne University where he received his Bachelor and Ph.D. degrees in Mechanical Engineering. He joined ARRB Transport Research in 1972, and led a number of traffic engineering research projects, mainly in the area of cost effective traffic design standards. From 1983 until 1989 he was Chief Scientist for ARRB Transport Research's Road Technology Group, with particular responsibility for setting up the Accelerated Loading Facility road pavement testing program, and was one of the authors of the NAASRA Strategy for Pavement Research and Development. In 1990, John became a Research Director. He is a member of the Institute of Engineers, Australia and the Institute of Transportation Engineers.

Introduction Content

FOOTNOTES

1 The equivalent spring rate is the slope of the tangent to the load-deflection curve at the static load point.

Return to paper

Content

2 Bump refers to motion causing a reduction in the distance between the ends of the shock absorber, during rebound this distance is increasing.

Return to paper

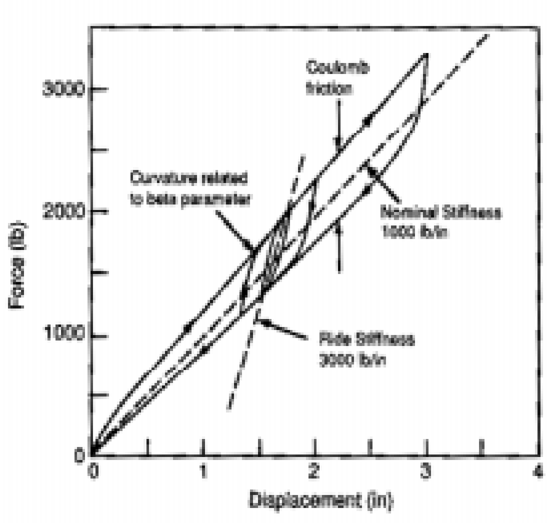

Figure 1(a) Leaf spring force-deflection characteristics (Gillespie et. al., 1993).

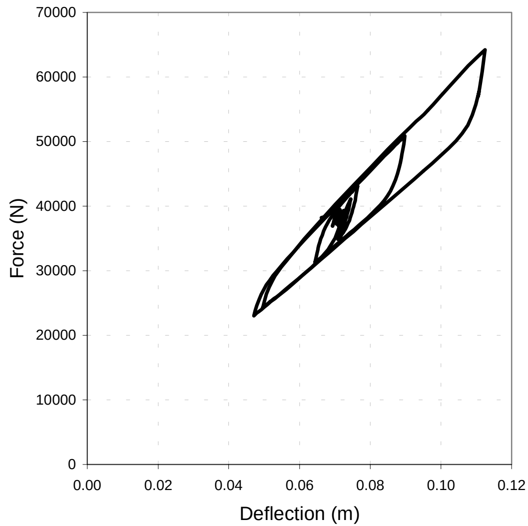

Figure 1(b) Characteristics of the quarter-truck multi-leaf steel spring used in this paper (typical).

Return to paper

Content

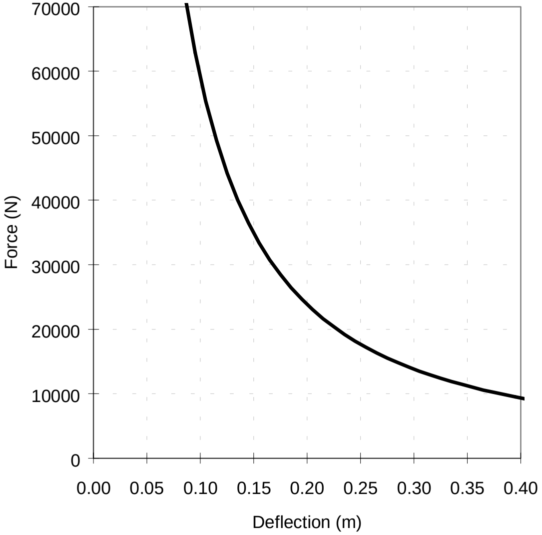

Figure 2 Force-deflection characteristics of the quarter-truck air spring (<u>note</u>: zero deflection corresponds to a zero volume "airbag")

Return to paper

Content

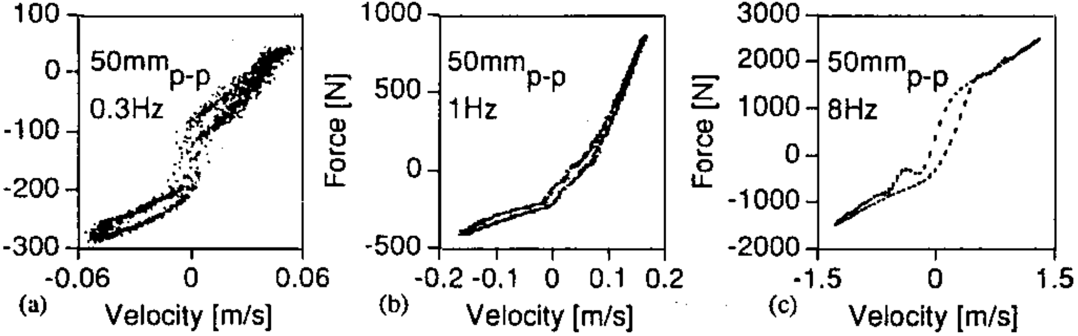

Figure 3 Typical damper force-velocity diagrams (reproduced from Duym et. al., 1997)

Return to paper

Content

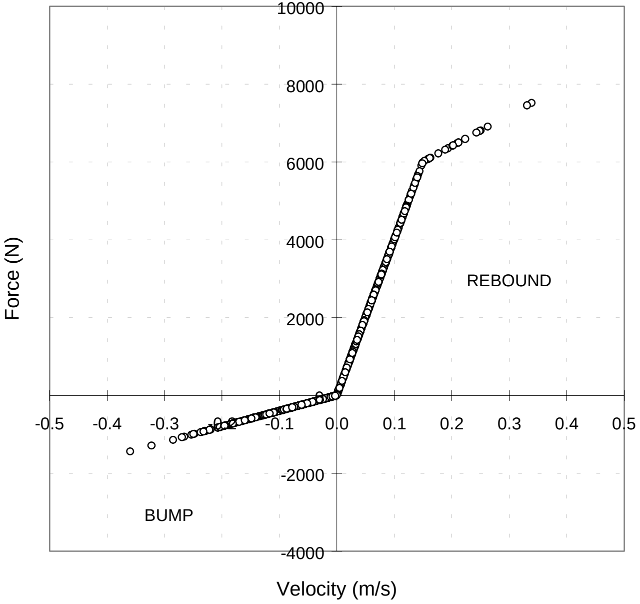

Figure 4 Hydraulic damper characteristics of the air suspension quarter-truck model used in this paper.

Return to paper

Content

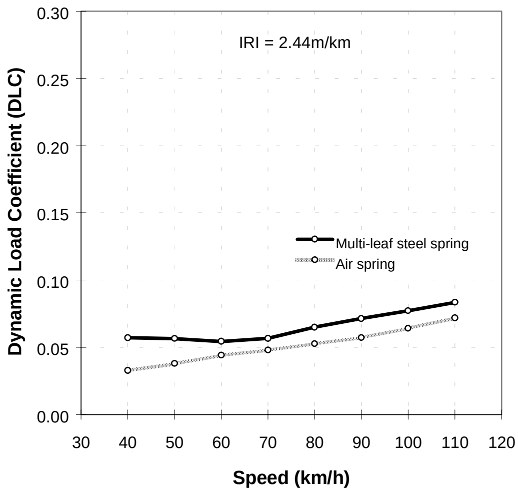

Figure 5(a) Air and steel spring quarter-truck DLCs for a medium roughness road profile.

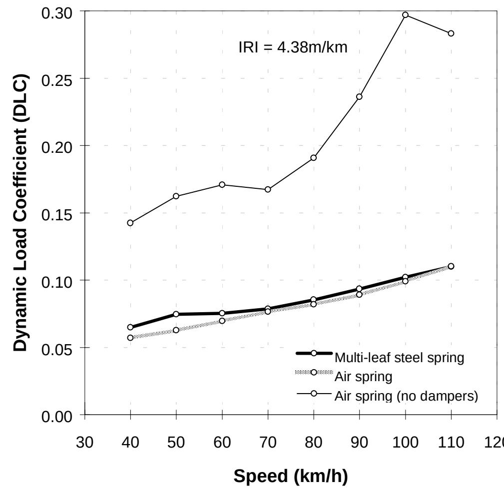

Figure 5(a) Air and steel spring quarter-truck DLCs on a high roughness road profile.

Return to paper

Content

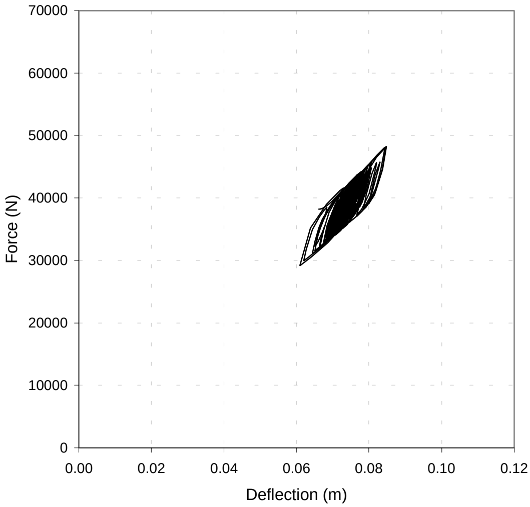

Figure 6(a) Quarter-truck steel spring response to high roughness profile.

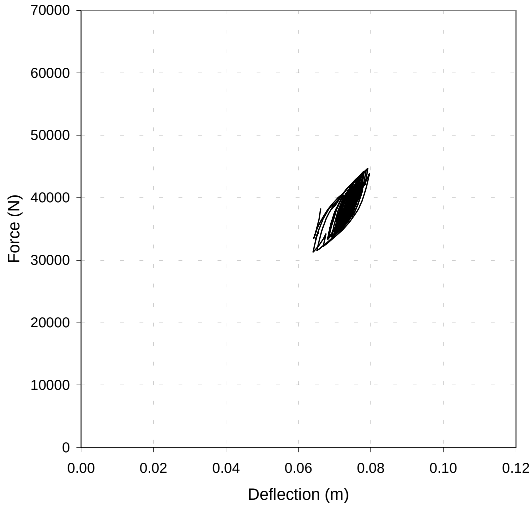

Figure 6(b) Quarter-truck steel spring response to average roughness profile.

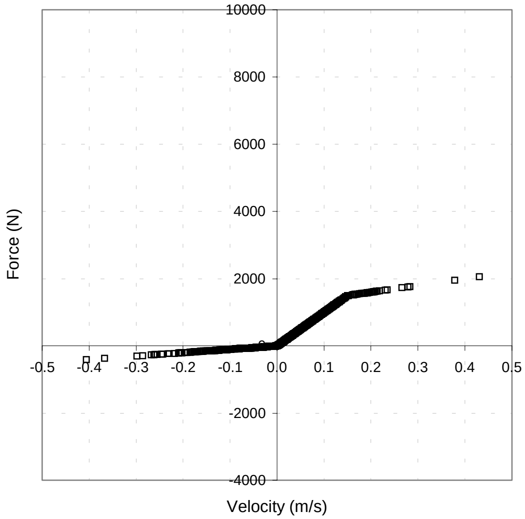

Figure 6(c) Quarter-truck damper response to high roughness profile (steel spring suspension).

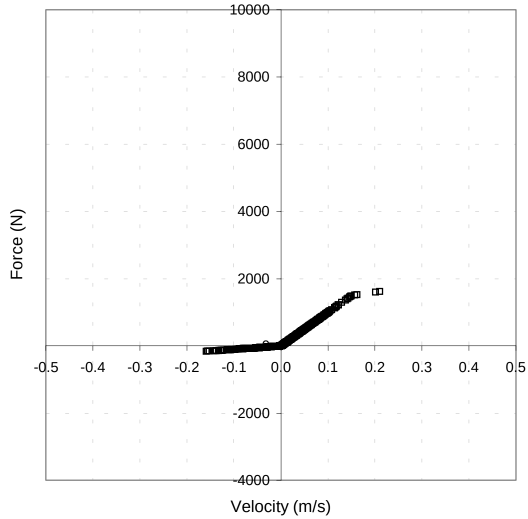

Figure 6(d) Quarter-truck damper response to average roughness profile (steel spring suspension).

Return to paper

Content

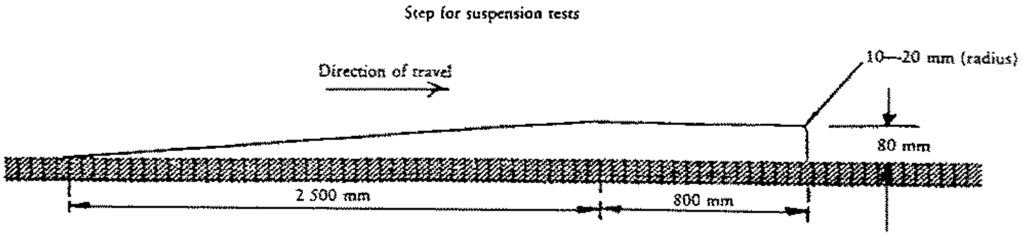

Figure 7 80mm step profile (reproduced from Council of European Communities, 1992).

Return to paper

Content

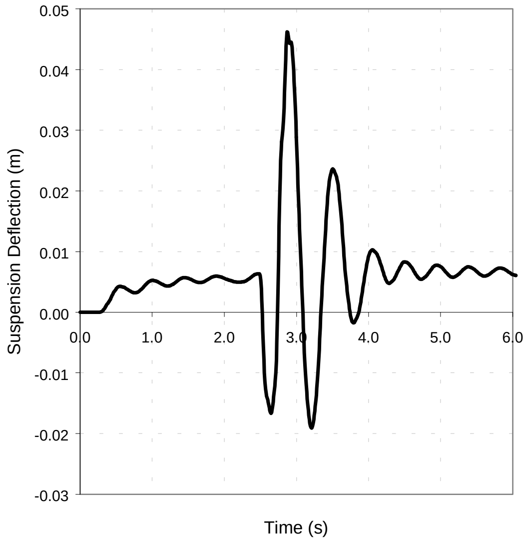

Figure 8(a) 80mm step response of quarter-truck multi-leaf steel spring.

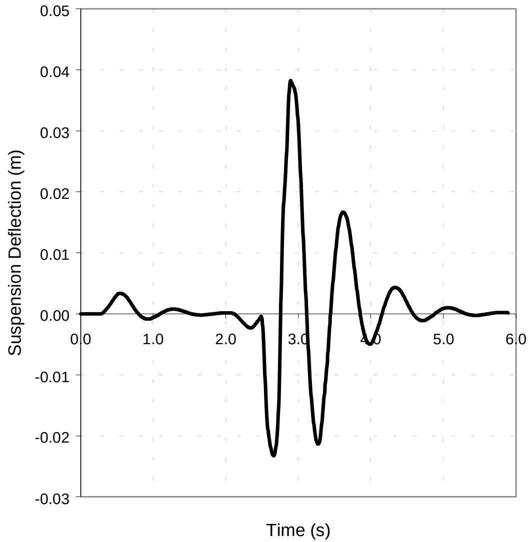

Figure 8(b) 80mm bump response of quarter-truck air spring.

Return to paper

Content

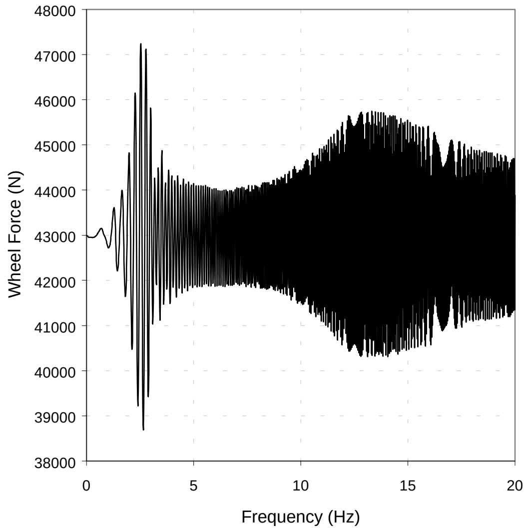

Figure 9(a) Constant 1mm displacement sinusoidal sweep input applied to the steel suspension quarter-truck model.

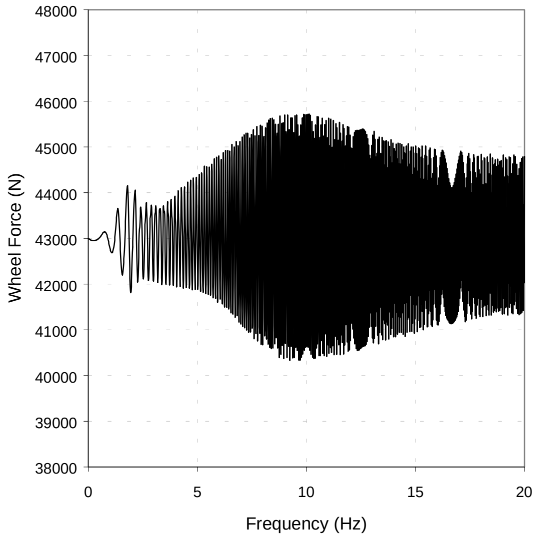

Figure 9(b) Constant 1mm displacement sinusoidal sweep input applied to the air suspension quarter-truck model.

Return to paper

Content

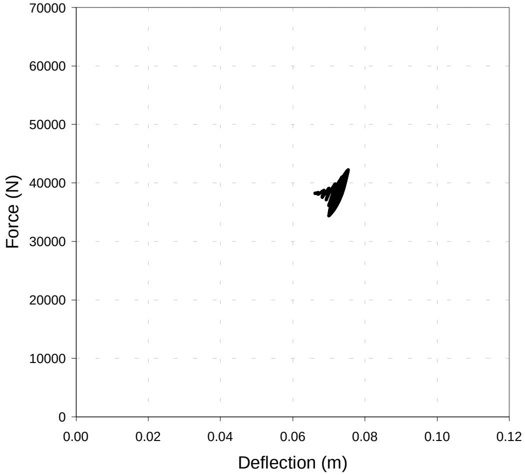

Figure 10 Steel suspension deflection for the 1mm constant amplitude sinusoidal sweep input.

Return to paper

Content

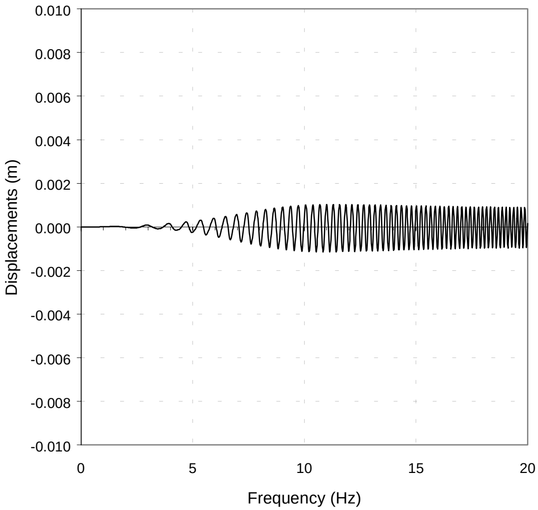

Figure 11(a) Axle displacement response of the air suspended quarter-truck model to an increasing-force frequency sweep (m=20kg, R=0.020m).

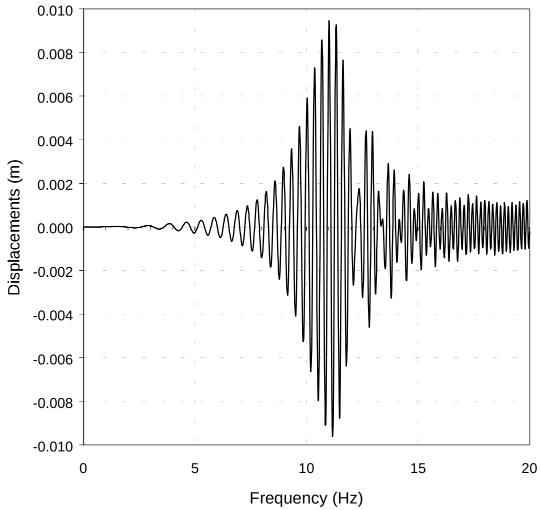

Figure 11(b) Axle displacement response of the air suspended quarter-truck model without dampers to an increasing-force frequency sweep (m=20kg, R=0.020m).

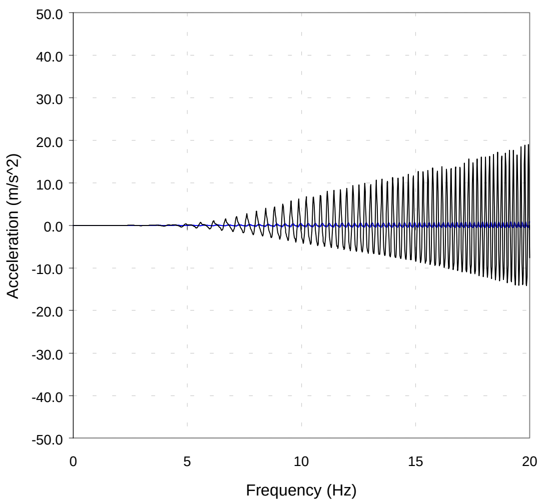

Figure 11(c) Axle acceleration response of the air suspended quarter-truck model to an increasing-force frequency sweep (m=20kg, R=0.020m).

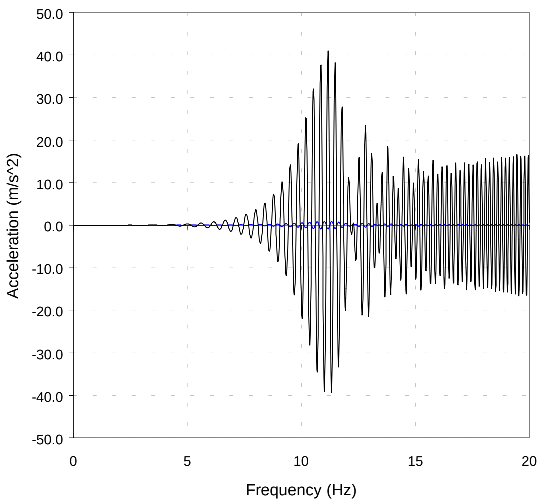

Figure 11(d) Axle acceleration response of the air suspended quarter-truck model without dampers to an increasing-force frequency sweep (m=20kg, R=0.020m).

Return to paper

Content