Further Development of In-Service Suspension Testing for Heavy Vehicles

Lloyd. Davis[1], Stephen. Kel[1], Roger. Sack[2]

[1] Queensland Main Roads, Brisbane, Qld, Australia
[2] Tramanco P/L, Rocklea, Qld, Australia

1 Introduction

Road authorities regard heavy vehicles (HVs) with “road friendly” suspensions (RFS) favourably. Accordingly, in Australia, RFS-equipped heavy vehicles (HVs) are allowed to carry more mass under certain conditions, such as the higher mass limits (HML) schemes, in each State. RFS in Australia generally incorporates air springs, although there are some steel-sprung RFS emerging onto the market.

“Road friendly” suspension (RFS) certification is provided under testing defined by the Department of Transport and Regional Services (DoTaRS 2004) specification VSB11 when the heavy vehicle is first registered for HML or other duties requiring RFS. The challenge for Australian road authorities is to ensure that those air-sprung heavy vehicles (HVs) with RFS are maintaining their “road friendliness” since trucks with air suspensions but with ineffective shock absorbers damage roads more than trucks with non-“road friendly” suspensions (OECD 1998, Cebon & Costanzi 2005). This is because dynamic forces generated by typical travel over uneven roads produce very high dynamic loads when RFS are lightly damped (OECD 1998, Cebon & Costanzi 2005). Air suspensions on HVs rely on suspension dampers (shock absorbers) almost entirely for damping. In contrast, steel springs on HVs provide residual damping via some inter-leaf Coulomb friction (that is, between the leaves of the springs) in the event that shock absorbers wear (Prem et. al 1998). An in-service test for road friendliness would be an advantage to both the transport industry and road asset owners. The former because worn shock absorbers could be replaced before vehicle and payload damage occurs; high-mileage but still serviceable shock absorbers need not be replaced (saving labour and equipment costs) and the latter because road and bridge asset rehabilitation costs would be reduced through less wear-and tear from RFS HVs with worn shock absorbers. Such a test has been mandated between two Australian States and the Commonwealth (DoTaRS 2005 a & b).

Australian road authorities in their agreements with the Commonwealth Government are progressing requirements for an in-service suspension test for RFS vehicles in-service HV suspension test (DoTaRS 2005 a & b). This should provide better assurance that the ‘road-friendliness’ portion of RFS is retained throughout the working life of the vehicle than under current arrangements. The official HV transport industry view of worn shock absorbers is that the tyre wear resulting from such a state of disrepair is detected quickly and rectified to prevent more tyre wear and increased cost of premature replacement of tyres. Despite this, the Marulan survey by Blanksby et. al (2006) showed that more than half the HVs sampled on the Hume highway did not meet at least one VSB11 suspension parameter. Hence, the industry indicator of using tyre wear to detect worn shock absorbers is far too late in the maintenance cycle to be effective at meeting the requirements of VSB11. Compounding this issue is the fact that there are no recognised low-cost in-service HV suspension tests in Australia although this has been discussed previously without a definitive way forward (Starrs et. al 2000, Sweatman et. al 2000).

Two measurements used to show that heavy vehicle suspensions are road friendly are the damping ratio and the damped free vibration (natural) frequency. Normally these are found by expensive laboratory testing where a truck is mounted on a frame and subjected to calibrated jolts and vibrations. VSB11 (DoTaRS 2004) specifies conformance to an upper limit for the body-bounce damped natural frequency and a lower limit for damping ratio. Both of these parameters are influenced by the health of the vehicle's suspension dampers. Davis & Sack (2004, 2006) and Davis (2005) documented two low-cost processes for testing heavy vehicle suspensions for in-service testing of RFS. They comprised methods where a truck was driven on typical roads and a method involving driving a heavy vehicle over a pipe. The signals induced by the on-road test were then analysed to derive the damped natural frequency of the body bounce. The pipe test signal, approximating an impulse input to the suspension via the wheels, was used to determine the damped natural frequency and the damping ratio of the test vehicle' body bounce. Derived values were in good agreement with the manufacturer's certified values.

This paper describes the testing and results of an expanded testing programme. It expands greatly on the results of previous test programmes as presented at ATRF 04 and ARRB 06 by Davis & Sack (2004, 2006). The expanded test programme comprised two measurement regimes made up of a combination of semi-permanent on-board systems and temporarily-attached transducers. These systems recorded data during pipe tests, a VSB11-style step-down test and on-road excitations.

On-road data were recorded for speeds from 40km/h to 90km/h over a variety of road surfaces ranging from smooth with long undulations to rough with short undulations. Vehicles for this series of tests were a 3-axle interstate coach, a 7-axle semi-trailer combination and a 2-axle school bus. Derived suspension parameters are compared (where available) with the manufacturer's corresponding certified values of those suspension parameters. Where not available, the values derived from our own VSB11-style step tests (Peters 2003) were used.

With some road authorities required, by agreement with the Australian Government, to develop an in-service HV suspension test (DoTaRS 2005 a & b) whilst others consider the effectiveness of methods for in-service HV suspension testing, this paper outlines a "proof-of-concept" for simple, low-cost testing methods. The next steps should be the implementation of in-service suspension testing for HVs. Further developments in implementation in-service procedures, who would perform them and field test equipment will be left, properly, to the National Transport Commission project (DoTaRS 2005 a & b).

This paper and any developments for an in-service test for RFS HVs do not address load-sharing of air-suspended vehicles, a topic which will be dealt with in other papers.

2 Background theory

2.1 Damped free vibration frequency and damping ratio

Characterising heavy vehicle (HV) suspensions is central to the EU (and Australian) test for "road friendliness" (European Council 1996, DoTaRS 2004). Two measurements used to show that heavy vehicle suspensions are road friendly are the damping ratio, zeta (ζ) and the damped free vibration frequency (f). The damped free vibration frequency is the frequency at which a truck's suspension has a tendency to bounce with the largest excursions whilst being damped by the suspension dampers (shock absorbers).

The damping ratio is a measure of how fast a system reduces its oscillations (and returns to quiescent or steady state) after a disturbance. It is a measure of the reduction in excursions of subsequent amplitudes of the output signal from a system. In HV suspensions, it is related to a measure of how quickly the shock absorbers and other components reduce body bounce and wheel hop after the truck hits a bump. The damping ratio, zeta (ζ), is a dimensionless number and is usually presented as a value under 1 (e.g. 0.3) or a percentage (e.g. 30%) denoting the damping present in the system as a fraction of the critical damping value (Doebelin 1980).

Chesmond (1982) showed that system parameters may be characterised in a number of ways. Amongst these are:

- application of a random input signal to a system: after which Fourier (or other frequency domain) analysis of the output signal resulting from that random input is used to determine the characteristics of the system transfer function. The damped free vibration frequency (f) of the system characterises that transfer function and will show up as the largest magnitude peak in the frequency spectrum of the output signal; or

- application of an impulse input signal to a system: impulse signals contain all frequencies in equal proportion. Again, Fourier analysis of the output signal is used to determine the characteristics of the system transfer function. Similar to random input signal excitation, the dominant (in this case, the damped free vibration) frequency will manifest as the largest peak in the frequency spectrum of the output signal for a given impulse input; and

- subjecting any system to an impulse signal and measuring the reducing excursions of the output signal enables the damping ratio of a system to be determined.

Milliken et. al (2001) showed that the damping ratio (ζ) may be determined by comparing the values of any two consecutive response peaks in the same phase (i.e. comparing the magnitudes of the 1st and 3rd excursions or the 2nd and 4th excursions) of the output signal of an underdamped system after an impulse function input has been applied. Prem, et. al (2001) used the formula:

$$\zeta = \frac{\delta}{\sqrt{\delta^2 + (2\pi)^2}} \qquad \text{(Equation 1)}$$

where δ is the standard logarithmic decrement (Meriam & Kraige 1993) given by the following formula:

$$\delta = \ln\left(\frac{A_1}{A_2}\right) \qquad \text{(Equation 2)}$$

where: A_1 is the amplitude of the **first** peak of the response and A_2 is the amplitude of the **third** peak (Milliken et. al 2001) of the response; to determine damping ratio.

Where it is desired to determine damping ratio from a signal with more than 2 peak values in the signal on the same side of the x-axis in a time series, Thomson & Dahleh (1998) provide:

$$\delta = \frac{1}{n}\ln\left(\frac{x_0}{x_n}\right) \qquad \text{(Equation 3)}$$

to substitute into Equation 1, where: x_n is the amplitude after n successive cycles have elapsed or, for the case where continuous successive peaks are present, Eykhoff (1979) provides:

$$\delta = \ln\left[\frac{1}{n-1}\left(\frac{x_1}{x_2} + \frac{x_2}{x_3} + ... + \frac{x_n}{x_{n+1}}\right)\right] \qquad \text{(Equation 4)}$$

Note: Equation 1 may be derived by solving for ζ in the following equation (Thomson & Dahleh (1998): $\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}$

2.2 Road noise excitation as a white noise input signal – input frequency spectrum

Road excitation covers a range of frequencies but is generally not uniform in magnitude across that range of frequencies. Prem (2003) and Sweatman (1983) have, however, proceeded on the basis that, whilst acknowledging that road noise excitation is not uniform in magnitude across the frequency spectrum of interest, there is enough stability in the range of frequencies of interest for it to be a useful input function in the characterisation of suspension parameters.

The series of tests performed by Davis & Sack (2004, 2006) using excitation from road travel was designed to test the theory that suspension characteristics might be derived from analysing the output of the suspension (body bounce) after the road noise into the suspension was used as a surrogate for a white noise input. These tests showed that the damped (dominant) frequency value may be derived from frequency-domain analysis of the output signal from the truck's suspension after such an excitation.

2.3 A bump test using a pipe – input frequency spectrum

The testing described later and used by Davis & Sack (2004, 2006) employed a nominal 50mm diameter pipe ('bump test') to simulate an impulse function to the suspension of a heavy vehicle when the vehicle was driven over it. This test was an attempt to apply a surrogate impulse function to the axle/s of interest on a test vehicle so that the suspension's damped free vibration frequency might be determined from frequency-domain analysis of the output signal measured off the truck's chassis and the damping ratio derived from the exponential decay after such excitation, using Equation 1.

3 Testing equipment and procedures

A 3-axle interstate coach, a 7-axle semi-trailer combination and a 2-axle school bus were instrumented as detailed below. The axle groups of interest were the tri-axle group of the semi-trailer, the drive/tag axle group of the 3-axle coach and the drive axle of the school bus. The vehicles were loaded to as close as was practical to the maximum legal load for the axle/axle group in question. This allowed comparison of our results with manufacturer's certified data or, in the absence of these values, the results of our own VSB11-style test. The coach rear axle group consisted of a drive axle and a tag axle. The drive axle on the coach had a longitudinal beam attached on either side with an air spring on each end of the beam. This arrangement supported the chassis with 4 air springs in total for that axle. The tag axle had an air spring mounted above it on either side.

3.1 Data measurement and recording

Air pressure transducers were connected to each air line at their points of entry to each air bag for the axle group of interest. This allowed measurement of body-to-chassis signals at each air spring. An accelerometer was fixed to each test axle end as close to the hub as possible. An advanced version of the TRAMANCO on-board CHEK-WAY® telemetry system was used to measure and record the dynamic signals from the pressure transducer outputs on each air bag and hub accelerometers. Figure 1 shows the air pressure transducers and the CHEK-WAY® recording system installed on the coach. The CHEK-WAY® system is subject to Australian Patent number 200426997 and numerous international application numbers and patents which vary by country.

Figure 2 shows an example of the two Lansmont SAVER™ 3X90 accelerometer/data recorders used to record dynamic signals at the chassis of the test vehicles during testing. These units had magnets to attach them to the chassis of the test vehicles above the axle group/s of interest.

3.2 Telemetry system and sampling frequency

The CHEK-WAY® system sampling rate was 1 kHz giving a sample interval of 1.0mS. Cebon (1999) documented that the natural frequency of a typical heavy vehicle axle as 10 - 15Hz compared with a relatively low 2 - 3Hz for sprung mass frequency (de Pont, 1999). Any attempt to measure relatively higher frequencies (such as axle-hop) using time-based recording will necessarily involve a greater sampling rate than when relatively lower frequencies (such as the body-bounce frequency) are to be determined (Chesmond 1982). Since axle-hop was the highest frequency of interest for the analysis undertaken, the sampling frequency used by the CHEK-WAY® system was more than adequate to capture the test signal data, since its signal sample rate was much higher than twice any axle-hop frequency.

Figure 1 LHS panel: Air pressure transducers (green sticker) connected to air springs. RHS panel: CHEK-WAY® data recorders (yellow) used during the testing.

Figure 2 Lansmont SAVER™ 3X90 accelerometer/data recorders (blue and white) used during the testing.

The sampling frequency of the Lansmont accelerometer/data recorders was set at 100Hz for the pipe and step tests and 50Hz for the on-road testing. Due to finite memory capacity, the reduction in sampling frequency resulted in the ability of the Lansmont units to record for longer during our road tests. Accordingly, the Nyquist sampling criterion (Shannon's theorem) was met during the testing.

3.3 Procedural detail

All vehicles were loaded to as close as practical to maximum legal mass on the axle groups of interest and a series of on-road, pipe and step tests were performed. The dynamic signals from the accelerometers and the air pressure transducers on each axle-end of the axle group of interest were recorded using the on-board telemetry. The Lansmont accelerometer/data recorder was attached to the chassis of the test vehicles above the axle group/s of interest to record signals during the same tests (Figure 2). This resulted in test data from the Lansmont unit in the form of a time-series signal of chassis acceleration as well as time-series signals recorded by the on-board telemetry system from each transducer from each axle-end of each test HV for the axle/s or axle groups of interest.

3.4 On-road (white-noise) testing

The tests comprised driving the HVs over a series of typical, uneven road sections and recording the data generated from each transducer. The sections of road varied in roughness from smooth with long undulations to rough with short undulations. The vehicles were driven over the test road sections at a variety of speeds from 40km/h to 90km/h.

3.5 Step (VSB11-style) testing

The combination was driven off an 80mm step to replicate the VSB11 step test. For our VSB11-style step tests all wheels were rolled off a set of blocks simultaneously (Peters 2003). The signals from the air pressure transducers and the Lansmont unit were analysed to provide a set of reference data for damped natural frequency and the damping ratio of the body bounce with which to compare the results of the on-road and pipe tests. Figures 3 to 5 show the detail of these tests for the coach, for example. An auxiliary series of step tests was performed on the trailer with only the temporary accelerometer attached to the chassis above the tri-axle group. For these tests, the step test was applied to the rear axle only as shown in Figures 6 & 8. Chains (top left, Figure 3) attached to the chassis were used to drag the blocks once the wheels had moved off them so that the wheels were not fouled as they rolled subsequent to the step-down action.

Figure 3 Before: showing preparation for the step test on the coach.

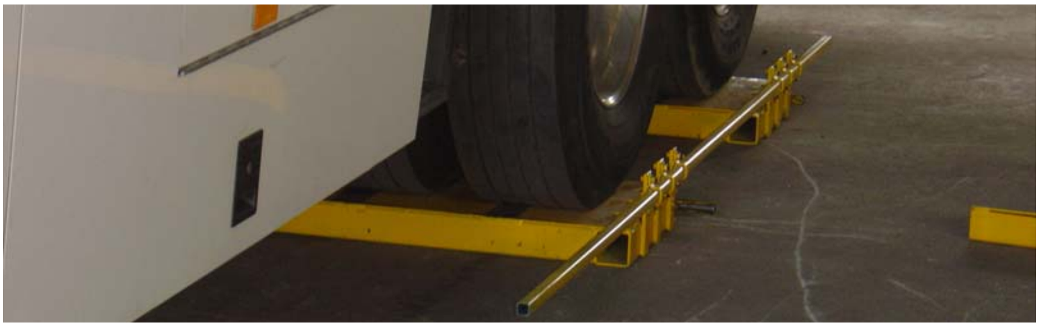

Figure 4 During: the rear axle group of the coach ready for the step test.

Figure 5 After: the step test which was set up in Fig 4.

3.6 Pipe testing

The combination was driven over a 50mm nominal diameter heavy wall pipe at a speed (as well as the driver could manage) of just above 5km/h. This was in order to provide a surrogate for an impulse input to the suspension of the test vehicle. The actual speeds of testing for the vehicles varied between 4.9km/h and 9.6km/h. The pipe had a bar welded to either end to prevent rotation as the tyres moved over it (Figures 6 and 7). Because the speedometers of the test vehicles did not register at speeds below 5km/h, the actual speeds for the pipe tests were measured by using a stopwatch elapsed time between two marks 10m apart on the test vehicles' paths. An auxiliary series of pipe tests was performed on the trailer with only the temporary accelerometer on the chassis above the tri-axle group.

Figure 6 Showing the test masses on the semi-trailer with the pipe for the "pipe test" in the foreground, bottom left.

Figure 7 Detail of a typical tyre rolling over the pipe during the pipe test.

Figure 8 Showing the preparation for step-testing the semi-trailer rear axle only.

4 Analysis

The signals from the air pressure transducers and on-axle accelerometers recorded during the road tests were analysed using a FFT (Davis 2006). The resultant FFT spectrum from the air pressure transducers showed the damped natural frequency of the body bounce of the test vehicles as the greatest peak in the spectrum. The signals from the Lansmont unit were analysed using a trial version of DADSIP software. Pitching, body roll, axle hop and wheel imbalance were also present on the derived spectra from on-road excitation. These were able to be distinguished easily as they had different frequencies from the body bounce frequency and were thereby able to be dismissed from the data (OECD 1998 pp 74-76).

For the pipe and the step tests, the Lansmont accelerometer data and the data from a representative sample of the air pressure transducers' signals were analysed by FFT to derive the damped natural frequency of the body bounce of the test vehicles. Analysis to derive the damping ratios was performed on the time-series test data using a logarithmic decrement method (Thomson & Dahleh 1998) where more than two successive peaks were available for analysis (Equations 1 and 3 or 4, above). Sometimes only two successive peaks were available in which case we used the formula shown in Equations 1 and 2 above. This yielded a value for the damping ratio for each axle tested. A double-check against the inverse of the time between successive peaks of the body bounce signal confirmed that the damped free vibration frequency was being shown as a peak on the FFT graph.

Sometimes the signals from the Lansmont unit on the chassis of the test vehicles during the pipe tests were a composite of the impulse from the first axle and subsequent axles, particularly if the test speed was verging toward 10km/h. In these cases, the damping ratio was not able to be determined since exponential decay was not apparent before the next impulse occurred. These data were not able to be analysed for damping ratio.

Body roll and axle roll were not present in the step test or the pipe test signals because the test disturbances were parallel to the axles.

5 Results

5.1 Input function characterisation

In determining the parameters that are important to an in-service test for HV suspensions, it is important to characterise the input function. Enough energy needs to be present in the input signal and at the frequencies of interest for the suspension system to be excited so that it will resonate at the damped natural frequency of the body or chassis. Accordingly, the signals from the accelerometers mounted on the test vehicle's axles were analysed to derive the input frequency spectrum for the pipe tests, our VSB11-style step tests and the on-road tests as described below.

5.2 Frequency spectra of input functions - white noise "on-road" test

We recorded the signals from the accelerometers mounted on the axles of the test vehicles. When subjected to our FFT, typical frequency spectra for the input signals from the "on-road" tests were derived and are shown in Figure 9.

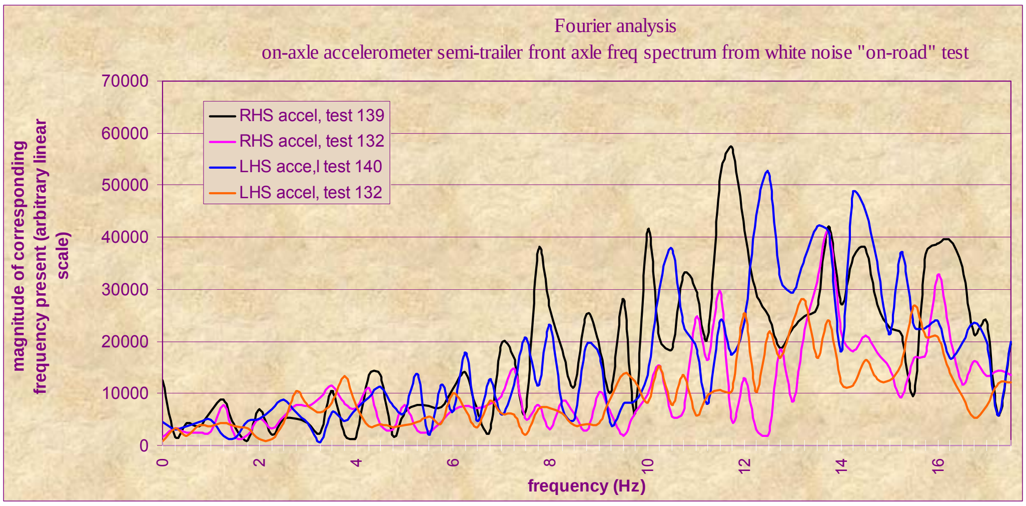

Figure 9 FFT of signals from typical road tests. These were from the accelerometers on the front axle of the semi-trailer but typical for vehicles tested.

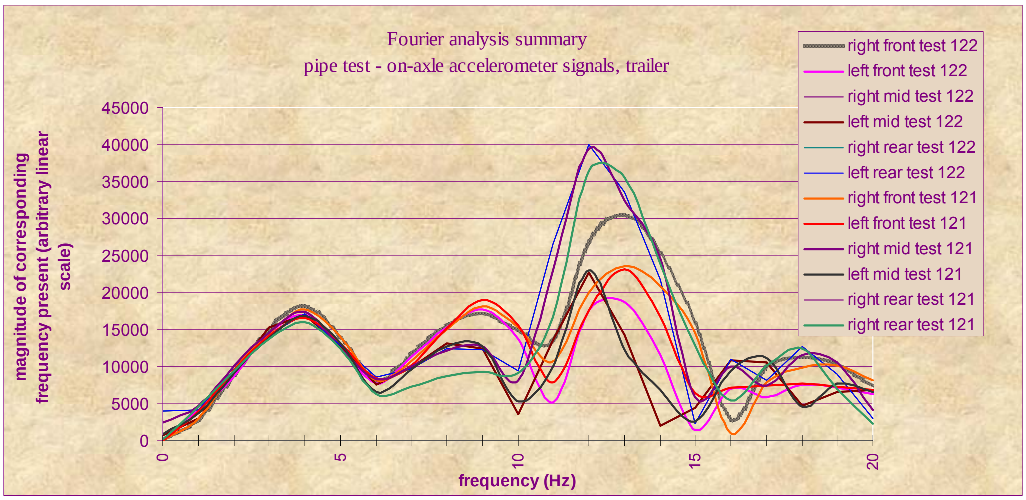

Figure 10 FFT of signals from pipe tests. These signals derived from the accelerometers on the axles of the semi-trailer but typical for vehicles tested.

5.3 Frequency spectra of input functions – "pipe test"

The frequency spectra of the signals from the accelerometers mounted on the axles of the semi-trailer are shown in Fig 10. These may be regarded as the input signal frequency spectra or the relative strength of the signal present from the pipe simulating an impulse signal into the HV suspension under test. More of these signals have been shown here than for the other tests since the novelty of the pipe test warrants more detail in the explanation. The speed of the test did not affect the input spectrum nor did the derived suspension parameters vary over the range of speeds used during testing for those tests which were closer to 5km/h than to 10km/h.

5.4 Step test

From the accelerometers mounted on the axles of the test vehicles, typical frequency spectra of the input signals from the step tests are shown in Figure 11. This test is defined in VSB11 so Figure 11 provides a useful comparison with the other spectra shown in Figures 9 and 10.

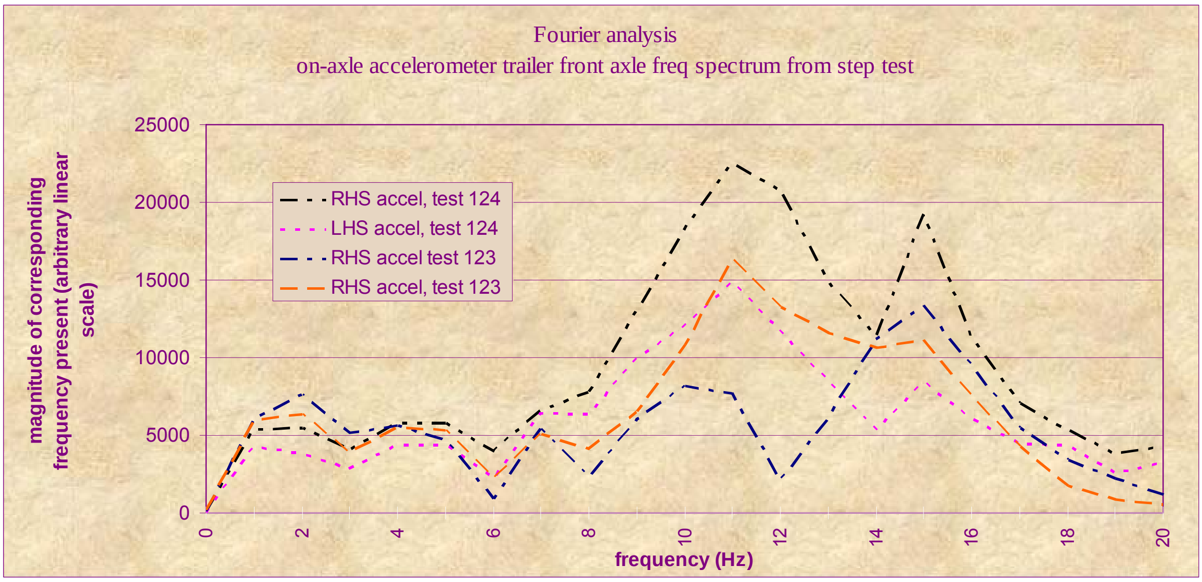

Figure 11 FFT of signals from step tests. These signals derived from the accelerometers on the front axle of the semi-trailer but typical for vehicles tested.

5.5 Derived parameters

Tables 1 to 5 show the results for damping ratio (ζ) and the damped free vibration frequency (f) for the various tests. It was apparent during testing, from preliminary results, that there were differing suspension parameters on the two rear axles on the coach. Detailed post-experimental analysis showed that the coach chassis signal being recorded by the Lansmont accelerometer/data unit comprised a combination of differing axle signals with different frequencies with different frequencies, damping, phases and time lags. Accordingly, the Lansmont data has been presented in Table 5 separately from the on-board data for the coach.

The entries in the tables are for averaged values where more than one value for any parameter was derived (including manufacturer's data, Colrain 2007). Coach and bus manufacturer's data was unavailable at the time of writing.

Table 1 School bus – drive axle, parameters from various test methods

Result	Test						
	Manufacturer pull up and drop	**Step (VSB11-style)**		**Pipe**		**Road**	
		Lansmont	on-board	Lansmont	on-board	Lansmont	on-board
f (Hz)	n/a	1.06	1.19	1.14	1.20	1.17	1.10
ζ (%)	n/a	23.31	18.5	25.84	17.5	n/a	n/a

Table 2 Trailer – all axles, parameters from various test methods

Result	Test						
	Manufacturer pull up and drop	**Step (VSB11-style)**		**Pipe**		**Road**	
		Lansmont	on-board	Lansmont	on-board	Lansmont	on-board
f (Hz)	1.89	1.61	1.67	1.53	1.64	1.56	1.73
ζ (%)	25.01	28.12	23.57	23.47	27.70	n/a	n/a

Table 3 Trailer – Lansmont data from rear axle excitation only

Result	Test		
	Manufacturer pull up and drop	**Step (VSB11-style)**	**Pipe**
f (Hz)	1.89	1.70	1.78
ζ (%)	25.01	18.58	19.33

Table 4a Coach – drive axle results from on-board system

Result	Test			
	Manufacturer pull up and drop	**Step (VSB11-style)**	**Pipe**	**Road**
f (Hz)	n/a	1.17	1.31	1.11
ζ (%)	n/a	32.00	17.49	n/a

Table 4b Coach – tag axle results from on-board system

Result	Test			
	Manufacturer pull up and drop	**Step (VSB11-style)**	**Pipe**	**Road**
f (Hz)	n/a	1.10	1.10	0.97
ζ (%)	n/a	29.45	17.90	n/a

Table 5 Coach – Lansmont results from chassis measurement

	Test		
Result	**Step (VSB11-style)**	**Pipe**	**Road**
f (Hz)	1.35	1.24	1.12
ζ (%)	29.72	8.81	n/a

6 Discussion

Davis & Sack (2006) noted that the tyre contact patch length may influence the input signal spectrum of the pipe test and that the resulting input signal may not be of sufficient magnitude at the frequencies of interest (0 – 4 Hz). We see from Figure 10 that these concerns may be dismissed since the pipe test frequency spectrum is strong at the body bounce frequencies with respect to the "gold standard" VSB11 step test. Doebelin (1980) is apt: 'We see that if [the input function's] duration is "short enough", the system responds in essentially the same way as it would to a perfect impulse of like area and that *the shape of* [the input function] *makes no difference whatsoever*.' Figure 10 shows that the pipe test's input function provides a valid input signal for determining body bounce response at speeds of 5km/h and above. Figure 11 shows that the strength of the input signal available using a VSB11-style test is actually lower than the strength of the signal available during the pipe and road tests, validating the view that both the road test and the pipe test provide useful excitation for HV suspension analysis, compared with the VSB11 step test.

Uffelmann & Walter (1994) and Prem et. al (1998) documented variations in the derived values for damping ratio, dependent on the excitation method due to, in part, the non-linearity of suspension dampers and their action non-symmetry, depending on direction of travel. A 60% variation was noted in derived damping ratio when comparing a "lift and drop" test with what they described as a "bump test" which was our step test.

The derived values for damped free vibration frequency and damping ratio from the pipe and on-road methods of testing were compared with the manufacturer's certified values, where available. Manufacturer's values are generally derived from a type-test by using a "lift and drop" technique on a single axle. Where manufacturer's values were unavailable at the time of writing, our step test results have been taken as the reference values. The results for the derived values for the damped natural frequency of the axles on the semi-trailer from the road tests, pipe tests and our step tests were within a 5% range of each other for any given recording system. The on-board system's results for f were also within 13% of the manufacturer's values and the agreement with the Lansmont was within 19% for the same criterion. Similarly, damping ratios derived from our tests using the on-board system on the trailer were within 10% of the manufacturer's certified value and varied over a 10% range. The Lansmont data for the damping ratio of the trailer axles ranged over 16% and were within 11% of the manufacturer's value. The bus f derived from the on-board system for pipe and road tests came within 10% of our step test result; the Lansmont data for the bus from pipe and road tests yielded a value of f which was within 17% of the reference step test value. Damping ratios from the bus on-board system were within 6% of the step test value; the Lansmont yielded a damping ratio for the bus which was within 10% of its step test result. The correlation found in these tests is good-to-excellent; especially given the potential variation as noted by Uffelmann & Walter (1994).

Our single-axle tests provided good correlation with the certified values of the trailer f but the damping ratio results had a difference of 34% when compared with the manufacturer's value. This was outside the reasonable scope of experimental error. VSB11 (DoTaRS 2004) allows axle groups or single axles to be tested for their suspension characteristics. It is noted that de Pont (1999) addressed testing per axle with the axle/s on the vehicle and showed that this characterisation for body bounce mode was valid. Our tests were performed both on individual axles whilst in a group on a vehicle and on entire groups. The results from the use of both methods show that either is valid for deriving f but we experienced difficulty in agreement when comparing derived damping ratio values for the coach and for the single-axle tests on the semi-trailer. The potential for variation (Uffelmann and Walter 1994) may be the explanation for the difference between the single-axle test results and the manufacturer's values but this consideration should have mitigated against the good results for testing axles as a group as described above.

When we tested multi-axle groups with the pipe test with speeds closer to 5km/h than 10km/h the suspension had time to settle before the excitation from the next axle. This was important for the on-chassis measurement of damping ratio from the Lansmont unit.

The signals from the coach were difficult to analyse due to the drive axle having a different suspension arrangement (and therefore different parameters) from the tag axle. As noted above, the Lansmont data from the coach chassis was especially difficult to analyse, being a composite of the signals from the two different axles. It is proposed that the disparity of axle types is the reason for the large differences in the derived parameters for the coach.

Blanksby et. al (2006) found that 54% of air suspended HVs did not meet VSB11 for damping ratio. Cebon & Costanzi (2005) modelled a fleet of HVs with 50% ineffective dampers which led to them concluding that, at Higher Mass Limits loadings, road damage to pavements and surfacings would be 20 - 30 % greater than a comparable freight task with a fleet equipped with dampers in good condition. That study was on the Newell highway which has considerably thicker pavements than those found in Queensland. Even if this quantum of costs was translated, without considering the comparative fragility of the Queensland network to higher loads (with respect to the Newell's >200mm thick pavements), a HV fleet with 100% functional shock absorbers would save Main Roads $45M/annum in 2006 dollars (Main Roads 2006) every year. This is essentially "free money" since HV suspensions should be maintained as a matter of course. Further, this consideration does not include the road safety aspects of a HV fleet with sub-optimal shock absorbers.

The analysis carried out for this paper was performed using the proprietary software available with the Lansmont accelerometer/data logger, a trial version of DADSIP software and an Excel-based FFT (Davis 2006) and by the use of a pocket calculator to find the inverse of a value read off a time-series graph. These tools are neither complex, scarce nor expensive. On-board instrumentation for HVs is becoming more common. Signals from such telemetry, where installed, have been documented here as able to be analysed to provide suspension parameters during defined perturbations such as the "pipe test". Where such systems are not installed, a simple perturbation in the transport operator's workshop or marshalling area would allow damped natural frequency and damping ratio to be determined from temporary attachment of an accelerometer/data logger during road &/or bump testing. Untrained personnel could attach a device such as the accelerometer/recorder used here and record HV chassis signal during a traverse over a "pipe test". Analysis of such signals took us minutes per signal. Hence a "suspension damper health check" would not be expensive and should be considered as one of the next steps in the national in-service HV suspension test (DoTaRS 2005 a & b) project.

7 Conclusion

The results of the more extensive testing described in this paper have indicated that the methodologies of excitation provided by the pipe impulse and the on-road white noise are sound for similar axles in a group. With careful choice of test speed bounds, the accuracies in derived parameters from both the Lansmont data and those derived from the on-board system data should be sufficient for 'pass/marginal/fail' testing. With more experience gained by undertaking continuous in-service monitoring it may be possible to obtain more accurate, more reliable and cheaper results than those from single pass laboratory testing such as VSB11.

Difficulties in determining suspension parameters in vehicles with differing axle types within a group may be experienced when designing in-service testing methods. More work needs to be done on single-axle testing within a group if this method is considered for in-service testing in future.

In-service testing of HV suspensions should be feasible using the tools, techniques and equipment which provided good results from this series of tests. With the increasing prevalence of on-board instrumentation for HVs (such as the CHEK-WAY® on-board telemetry system), the signals present from the suspension during typical travel tasks should be able to be analysed for damped natural frequency. Similarly, a simple perturbation in the transport operator's workshop or marshalling area would allow damped natural frequency and damping ratio to be determined with suitable analysis. For HVs not so equipped, the temporary attachment of an accelerometer/data logger during road &/or bump testing should yield the same parameters. There are now a number of choices for implementing low-cost in-service HV suspension testing (DoTaRS 2005 a & b) in order to save transport operators the cost of damage to vehicles and freight; and road authorities the cost of asset damage from HVs operating with out-of-specification or deficient shock absorbers.

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