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Paper title: **Analysis of heavy vehicle suspension dynamics using an on-board mass measurement system**

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Abstract (200 words):

Trucks with road friendly suspension (RFS) but with ineffective shock absorbers can damage the roads more than trucks with steel suspensions. Two measurements used to show that heavy vehicle suspensions are road friendly are the damping ratio and the natural frequency. Normally these are found by laboratory testing. A truck is mounted on a frame and subjected to calibrated jolts and vibrations.

This paper outlines low-cost test procedures and analysis methods for determining the road-friendliness of heavy vehicle suspensions. It is suggested that, by using the simple process of driving over a 50 mm pipe and analysing the data provided by the on-board mass measurement system, the damping ratio may be determined by simple calculation. Further, by driving an instrumented semi-trailer around the block, we can tell if its suspension has the correct natural frequency. The testing used a newly constructed, 34 t four-axle semi-trailer with on-board mass transducers.

Using on-board mass management systems provides the opportunity for determining the health of road-friendly suspensions without the expense and inconvenience of taking the truck off the road for testing. This will allow a 'win-win' circumstance for operators and road authorities regarding the health status of heavy vehicle suspensions.

Introduction

Trucks with road friendly suspension (RFS) but with ineffective shock absorbers can damage the roads more than trucks with steel suspensions (Transit NZ, 2001).

This paper outlines the groundwork for a low-cost testing method for finding out the natural frequency and damping ratio of heavy vehicle suspensions. Testing and analysis of the air suspension of a newly constructed, 34 tonne (t), four-axle semi-trailer was undertaken. The semi-trailer had on-board mass transducers. The tests occurred in February 2003.

The semi-trailer had known and certified suspension characteristics. When built, the suspension manufacturer of the trailer certified that it met the European Union (EU) requirements for road-friendly suspensions on heavy vehicles. The two measurements used to show that the suspension of a heavy vehicle is road friendly are whether the damping ratio is high enough and the natural frequency is low enough. The damping ratio is how much the shock absorbers reduce suspension bounce after the truck hits a bump. The damping ratio, zeta (ζ), is given as a value under 1 (eg. 0.3) or a percentage (eg. 30%). The natural frequency, (f_n) is how many times in a second the suspension wants to bounce after the truck hits a bump. Frequency is measured in Hertz (Hz) indicating cycles, or bounces, per second.

The European Union (EU) Standards require a minimum damping ratio of 0.2 (20%) and a maximum natural frequency of 2.0 Hz to be classified as “road friendly”. Australian road authorities require manufacturers to certify that their suspensions meet these standards if their products are to be regarded as road friendly.

Normally the damping ratio and the natural frequency figures are found by laboratory testing. A truck is mounted on a frame and subjected to calibrated jolts and vibrations. In this paper, the experimental results of the analysis are compared with the suspension characteristic values of natural frequency and damping ratio as certified by the manufacturer. Feasibility of determining road-friendliness of air suspensions for heavy vehicles without recourse to laboratory or workshop facilities is explored.

Test equipment

The truck used for the testing was a standard Kenworth prime mover, with air suspension on the drive axles, coupled to a quad axle semi-trailer with air suspension. The vehicle configuration had been deemed to fall under Queensland Transport’s “innovative vehicle” classification. This classification allowed operation of the combination under a permit on Queensland roads.

O'Phee Trailers built the trailer and the combination is owned and operated by Tronc's carrying service. The combination had been on the road since mid February 2003 operating under permit. One of the conditions of the permit was for the vehicle to be monitored for mass and location on the road network while operating. The gross combination mass (GCM) permitted depended on the combination's location.

Figure 1 The vehicle combination used for the testing. Note that the container plus contents weighed approx. 11t.

The freight task of the combination was general freight/general access when the GCM was 42.5t or less, whilst a GCM of 50t or less was permitted on a particular route between Acacia Ridge and Lytton in Brisbane.

The tracking was via GPS position fixes from a C-Track GPS reporting system relayed back to a base station at the premises of Digicore, a third party service provider. The C-Track system reported every hour via mobile phone link. The report contained the position of the vehicle in 6 minute intervals for the preceding hour. The prime mover was equipped, before this trial, with the C-Track GPS reporting system for fleet management purposes.

Axle load data from the drive axles of the prime mover and the semi-trailer axle group was measured indirectly from, and proportional to, the air pressure in the high-pressure air lines to the air suspension. Air pressure was converted to an electrical signal proportional to mass by a TRAMANCO on-board ChekWay® mass measurement system which sent the mass signals to the C-Track system as well as displaying the mass of the prime mover and the semi-trailer on a readout panel in the cabin. 40 kg increments were chosen for the digital mass measurement system.

An Australian Provisional Patent Application Number # 200 390 4423 has been allocated to the CHEK-WAY® Eliminator Ins-Com® System in the name of TRAMANCO Pty Ltd.

Air suspension for heavy vehicles uses air bags instead of conventional steel springs to provide a flexible interface between the axle assembly and the body of the truck or trailer. Installing instrumentation to measure mass on air suspension is cheaper than installing instruments to measure mass on steel spring suspensions. This is because the air bag pressure, which is relatively easy to measure with a pressure transducer, provides a direct and convenient output proportional to the load supported by the suspension. For the combination tested in this experiment, 3 air line transducers were used. Two were installed in the high-pressure air lines to the prime mover's drive (rear) axles and one on the high-pressure air line to the trailer's quad axle group. The air pressure in these air lines yielded a signal proportional to the mass being supported by the corresponding axle group. To perform the same measurements on steel suspensions, the installation of a large number of sensing elements such as strain gauges would be required. These would be on each axle or at each spring attachment point.

To contain Troncs' capital outlay, only the drive (rear) axle group on the prime mover and the semi-trailer quad axle group were instrumented. The steer (front) axle of the prime mover was not instrumented to measure mass. Since the steer axle did not have an air bag suspension, it would have been expensive to instrument this axle to determine the mass being supported by that axle. This was not of great concern as the geometry of the combination ensured a fairly constant mass on the steer axle, regardless of load on the trailer. This was because the position of the trailer coupling on the prime mover (the turntable location) ensured that any weight from the semi-trailer was supported evenly by the two drive (rear) axles without any weight being transferred forward to the steer axle. This is called zero coupling lead, that is, the turntable was located over the centre of the drive axle group.

Even without a direct mass transducer measurement from the front axle, it was fairly easy to determine the total mass on the prime-mover. This was done by allocating a constant 6t mass value and programming this to be and added to the drive axle group. Weighbridge testing and subsequent calibration of the on-board mass measurement system ensured that that this signal was proportional to the total mass on the prime mover. Similar testing and calibration was performed on the semi-trailer to ensure accurate mass measurement.

The semi-trailer had a York control system which raised the front axle when the trailer is empty and lowered it when a load is on board. The rear axle of the semi-trailer group was built so that it steered itself around corners at low speeds and was locked by a driver-actuated control at high speeds. This resulted in less scrubbing of the semi-trailer quad axle group tyres at low speeds and high-speed stability without the tendency to "shimmy" that self-steerable axles sometimes have.

Procedure

For the purpose of the testing, a container with freight weighing approximately 11t was loaded onto the trailer as shown in Figure 1.

3 tests were performed on the quad axle trailer suspension. The first test was a step test to attempt a replication of the EU step test as documented by Sweatman, McFarlane, Ackerman, George (1994) so that a damping ratio value could be derived. The second was an attempt to apply, by approximation, an impulse function to each of the quad axles so that a damping ratio value could be derived from examination of the reducing excursions of the mass

readings as shown by Milliken, De Pont, Meuller and Latto (2001). The third involved driving the combination on some normal, uneven suburban roads. We hoped that this would apply a random signal, in the form of normal road unevenness, to give data with which to determine the natural frequency (f_n) of the suspension.

The step test

Sweatman *et al* (1994) detail the EU step test as using an 80 mm step down to create a negative step input to the vehicle suspension for purposes of determining damping ratio and natural frequency of axle-to-body bounce.

The vehicle storage and marshalling yard of the freight operator presented an ideal opportunity to replicate the EU step test. A new warehouse was being built and the slab was finished, awaiting the superstructure, shown in Figures 2 and 3. This slab was 65 mm above the surrounding surface of the manoeuvring apron in the yard.

Figure 2 Detail of 65mm drop in level between warehouse slab and manoeuvring apron.

Figure 3 General view of the warehouse slab (middle of picture) and manoeuvring apron in foreground

The combination was driven off the warehouse slab onto the apron at approximately 5 km/h. Figure 4 shows the progression over the step between the slab and the apron. This test became known as "the step test".

The bump test

The combination was driven over a 50mm water pipe at approximately 5km/h to provide a positive impulse function as an input to the suspension of the quad axle group of the semi-trailer. Figure 5 shows the progression over the step between the slab and the apron. The pipe had a bar welded to either end to prevent rotation as the tyres moved over it. This test became known as "the bump test".

Figure 4 Showing progression of combination over the 65mm drop between slab and apron

Figure 5 Detail of progression of combination over the 50mm pipe for the bump test.

The on-road test

A final test was performed by driving the combination with the 11t container on the semi-trailer over some normal, uneven suburban roads. These roads were convenient to the Tronc's depot. The variation in the mass signal was recorded as the combination travelled along these roads at speeds up to 60 km/h.

Data from testing

Step test data

The step test results, plotted in Figure 6 show that the signal is varying slowly and gives a shape that could not be analysed to derive axle-to-body dynamic parameters. This result may not be entirely useless, however, as it may be useful for deriving the air suspension system time constant. This is explored later in the Findings section. Figure 6 shows the test signal as measured for the first 2 axles, the first axle signal on the left. The second axle provided the entirety of the negative-then-positive signal shape on the right from 2.25s to beyond 4.92s. The other 2 axles produced a similar shape to axle 1 and have not been included for brevity.

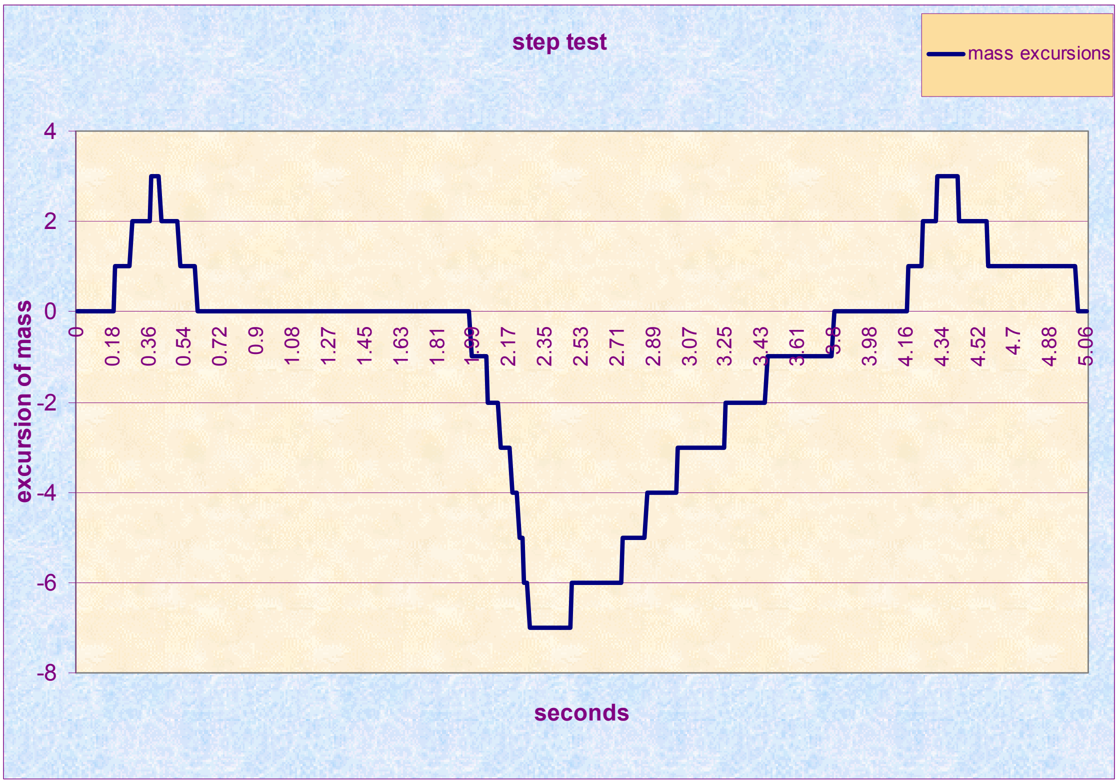

Figure 6 **Test signal from step test. Y-axis units are in 40kg increments**

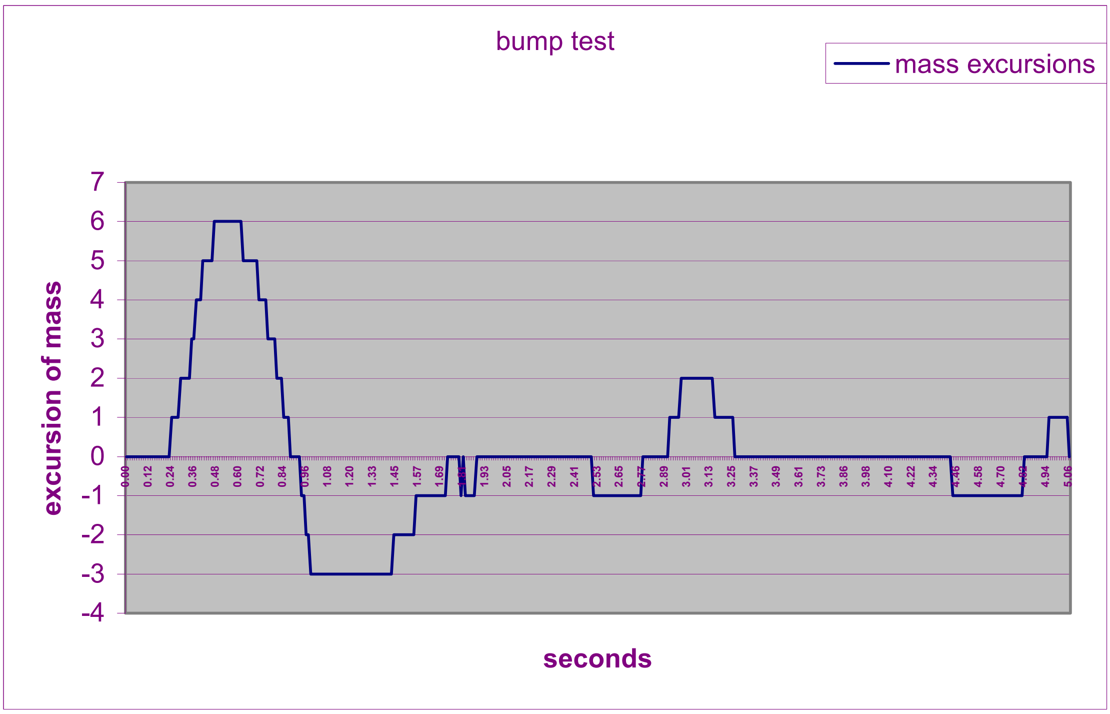

Figure 7 **The data from the bump test converted and plotted in 40kg increments. The 3 signals caused by the tyres from the 2nd, 3rd and 4th axles travelling over the pipe are shown here L-R respectively. Y-axis units are in 40kg increments**

Bump test data

The bump test yielded data that lent itself to meaningful analysis and these data are shown in Fig 7. The graph shows the signal excursions, left to right, produced by the second, third and fourth axles respectively. The signal generated by the first axle perturbation is not plotted here as it was similar to the two caused by axles 3 and 4, shown as the other two lumps in Fig 7.

On-road test data

To evaluate the validity of the proposal that a random signal applied to the semi-trailer axle group could be used to determine the natural frequency of the axle-to-body suspension, variation in the mass signal from the trailer axle group was of primary interest. For this evaluation the combination was driven around some normal, uneven suburban roads, Table 1 is a sample of how the resultant data from the on-board mass measurement system appeared in hexadecimal format.

By converting the data from the tests we were able to plot out the variation in mass induced by the dynamic forces on the combination, shown for information in Fig 8. We assumed that the signal derived from driving the vehicle on normal roads approximated to the response of the suspension system when subject to a random signal of multiple frequencies in the spectrum of frequencies of interest to us.

Table 1 **Sample of hexadecimal format test signal data from the on-board mass measurement system as the combination was driven around normal, uneven suburban roads.**

89 88 87 86 84 83 81 00 02 04 07 09 0C 0E 2A 02 05 07 09 0B 0C 0D 0D 0D 0D 0D
0D 0D 0D 0C 0B 09 07 05 03 01 00 82 84 86 89 8B 8D 8F 88 81 83 85 86 87 89 89 8A
8B 8B 8B 8B 8B 8B 8B 8B 8B 8B 8A 89 87 86 84 82 00 02 04 07 0A 0D 23 02 05 08
0A 0C 0D 0D 0E 0D 0D 0D 0D 0D 0D 0D 0C 0C 0A 08 06 04 02 00 82 85 87 8A 8C
8E 80 82 84 85 87 89 8A 8B 8B 8C 8C 8C 8C 8C 8C 8C 8B 8A 89 88 87 85 83 81 00
03 05 08 0B 0E 27 02 05 07 09 0A 0A 0A 0A 0A 0A 0A 09 09 09 07 06 04 02 00 81 83
86 88 8B 8D 8F F4 82 84 85 87 89 89 8A 8A 8B 8B 8B 8B 8B 8B 8B 8A 89 88 87 86
84 83 81 00 02 05 08 0A 0D 0F 26 02 03 05 06 07 07 07 07 07 07 07 07 06 05 04 02 00
81 84 86 88 8A 8D 8F 84 81 83 85 86 88 89 8B 8C 8C 8D 8D 8D 8D 8D 8D 8D 8D 8D
8C 8B 89 88 86 84 82 00 01 04 06 09 0B 0E 24 02 04 06 07 09 09 09 09 09 09 09 09 08
08 06 05 03 00 81 83 86 88 8B 8D 8F F6 82 83 85 86 88 89 8A 8B 8B 8B 8B 8B 8B 8B
8B 8B 8B 8A 88 87 85 84 82 00 01 04 06 09 0C 0E 1D 02 04 05 06 07 07 07 07 07 07
07 07 07 06 05 04 03 01 00 81 83 85 87 89 8B 8D 8F F3 81 83 85 86 87 89 89 8A 8A
8A 8A 8A 8A 8A 8A 8A 89 88 86 85 83 82 00 02 04 06 09 0B 0E 0E 02 04 05 07 08 08
09 09 08 08 08 08 07 06 04 02 00 81 84 86 89 8B 8D 8F E0 81 83 85 86 87

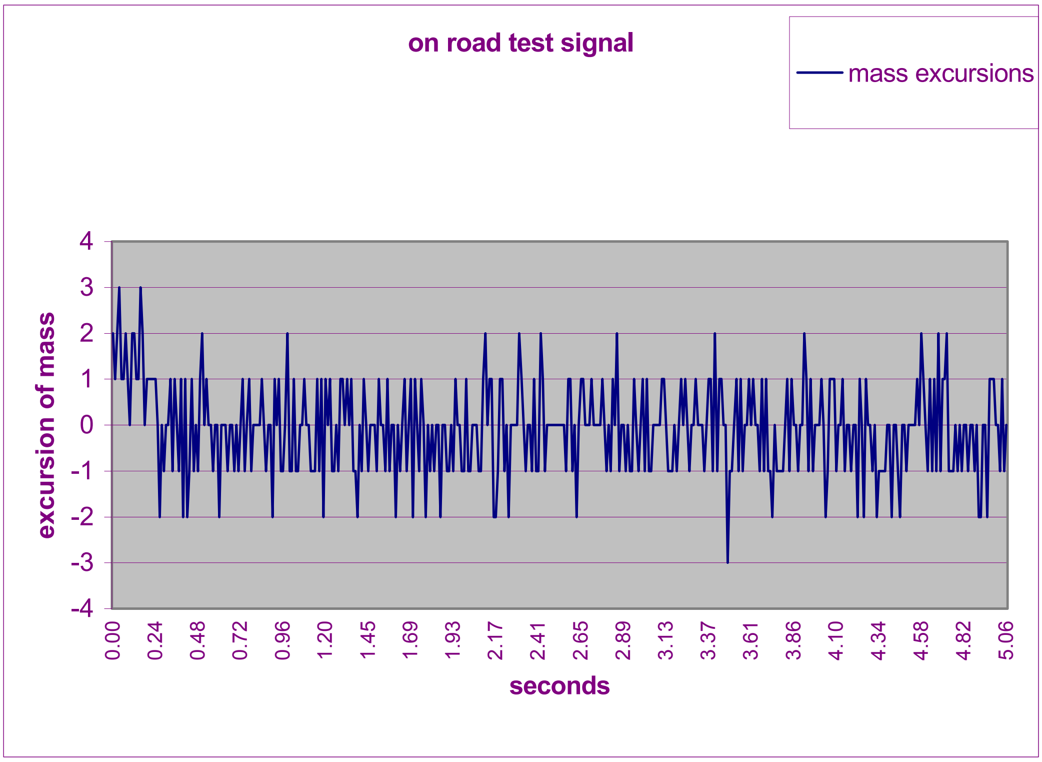

Figure 8 **Showing the variation of the air pressure (one increment is proportional to a mass value of 40kg) signal as the combination drove on some normal, uneven suburban roads. Y-axis units are in 40kg increments**

Findings

Analysis of the step test

The step test did not yield any data that could be analysed to provide meaningful results regarding axle or axle group dynamics with respect to the trailer body. It is noted that in Sweatman *et al* (1994), the EU test differs from our step test in that it is conducted at much higher speeds, it is done for one axle only and the drop is 80 mm, not 65 mm as used for our testing. We surmised that the effect of the air lines connecting the air bags were allowing any pressure differential between the air bags on differing axles to equalise. This would have caused the signal to behave in the manner shown in Figure 6, particularly for axle 2. It was thought that the 3 axles were restraining the first axle in the vertical plane as it went over the step but when the second axle encountered the step we thought that that was the point of equilibrium of the axle group and the group then teetered like a see-saw, giving the resultant signal. This signal was not easily analysed and we surmised that this was why the EU test was performed with only one axle. Further research and testing, as suggested by Prem H (2003), may yield results from this signal with respect to the dynamics, time constant and natural frequency of the load-sharing attributes of air bag suspensions.

Analysis of the bump test

The bump test yielded data that lent itself to meaningful analysis. 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. The pipe provided an impulse signal to the suspension of the combination. In this case the output signal of the system was the pressure in the air lines as the body/axle group interaction settled back to equilibrium. Milliken *et al* (2001) show that by determining the absolute values of the first excursion of the response data of an underdamped system to an impulse function, that is, the first and third excursions of the mass signal, the damping ratio (ζ) may be determined. Prem, Ramsay, McLean, Pearson, Woodrooffe and de Pont, (2001) use of the formula:

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

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

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

and

A_1 = amplitude of the first peak in the plot of the absolute value of the response and
A_2 = amplitude of the third peak, Milliken *et al* (2001), in the plot of the absolute value of the response.

The absolute values of the data created by the 2nd axle excursion shown in figure 6 have been plotted, expanded in time in Figure 9.

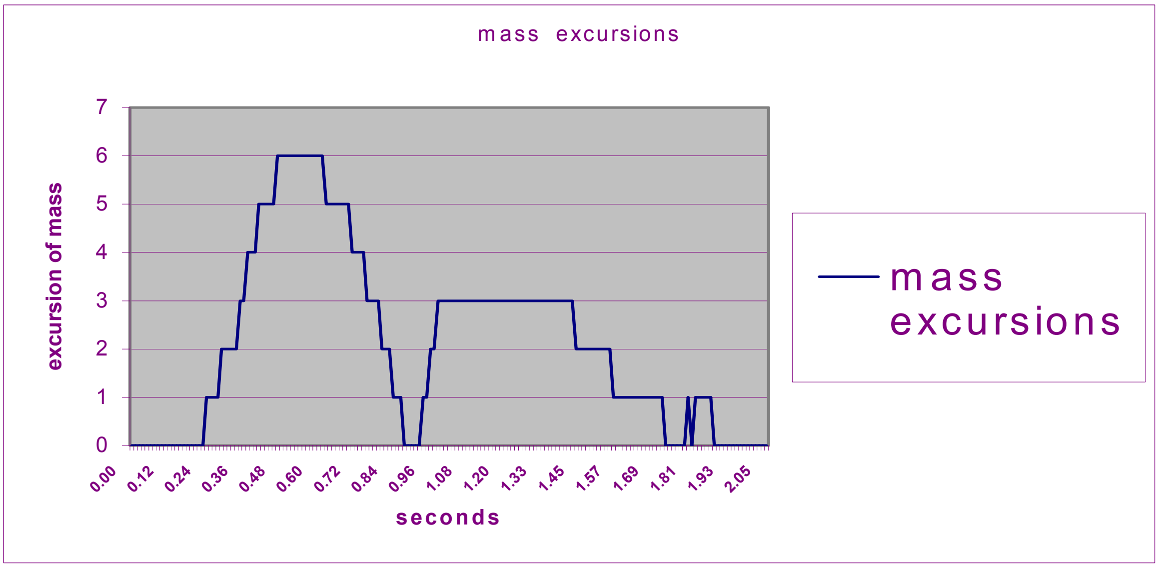

Figure 9 Absolute values of the useful data from the bump test plotted against time. Y-axis units are in 40kg increments

From Figure 9 we see that the value of A_1 = 6 x 40kg increments and A_2 = 1 x 40kg increment. It is noted that these values are the closest approximation to the actual values measured by the TRAMANCO mass measurement system given the approximately 1000 digitisation steps over the measurement range of 40 t, that is, 40,000kg $\div$ 1000steps = 40kg per increment.

Substituting the values of A_1 and A_2 as shown in Fig 9 into Equation 2 and deriving ζ from Equation 1 yields a damping ratio (ζ) of 0.27 $\pm\frac{0.11}{0.06}$ or 27% $\pm\frac{11}{6}$%. The inaccuracy is introduced from the discretisation of the mass values.

The EU standard (contained in ECC directive 92/7/ECC) for the damping ratio (ζ) for road-friendly heavy vehicle suspensions is 20% or greater. The manufacturer, Rizzo (2003), has confirmed that the suspension subject to these tests has a damping ratio (ζ) value of 0.263, which accords with the value derived from the analysis.

Fourier analysis of the on-road test

Chesmond (1982) showed that if a random input signal is applied to a system, characteristics of the system transfer function may be determined by Fourier analysis of the output signal resulting from that random input. The natural frequency (f_n) of the system is contained in that transfer function. Frequency is measured in Hertz (Hz). This is a metric unit indicating frequency in cycles, or bounces, per second.

The natural frequency is the frequency at which a system has a tendency to oscillate, or bounce, the most. Fourier transform analysis of a signal plotted against time, for example as shown in Figure 8, results in a plot which shows which frequencies are present and how large they are compared to other frequencies. The range of these frequencies is called the frequency spectrum. There is an approximate version of Fourier transform analysis called fast Fourier transform (FFT) analysis which is available in most personal computer spreadsheet programmes. Whilst widespread, FFT can produce noisy results.

By FFT analysis of the data from the on road test we were able to determine a value for the natural frequency of the semi-trailer's suspension. For this, a FFT was performed on the data from the semi-trailer's body-to-axle group frequencies induced by the forces on the combination as it drove around uneven roads.

For this analysis, we assumed that the signal derived from driving the vehicle on normal roads approximated to a random signal containing frequencies of interest to us. Road excitation covers a range of frequencies but is generally not uniform in magnitude. However, Prem (2003) advised we were only interested in a particular range of frequencies and their associated peaks in the frequency spectrum, and therefore the non-uniformity of the magnitude of the signal at any given frequency was not significant.

The plot in Fig 10 is a Fast Fourier Transform (FFT) of the on-road data. It is lumpy. The load data have been discretized at 40kg increments and therefore the positive and negative step increments have introduced sin (x)/x type noise and lumps and Prem (2003) has advised that this is because the signal is heavily digitized, not smoothed and not sampled particularly quickly.

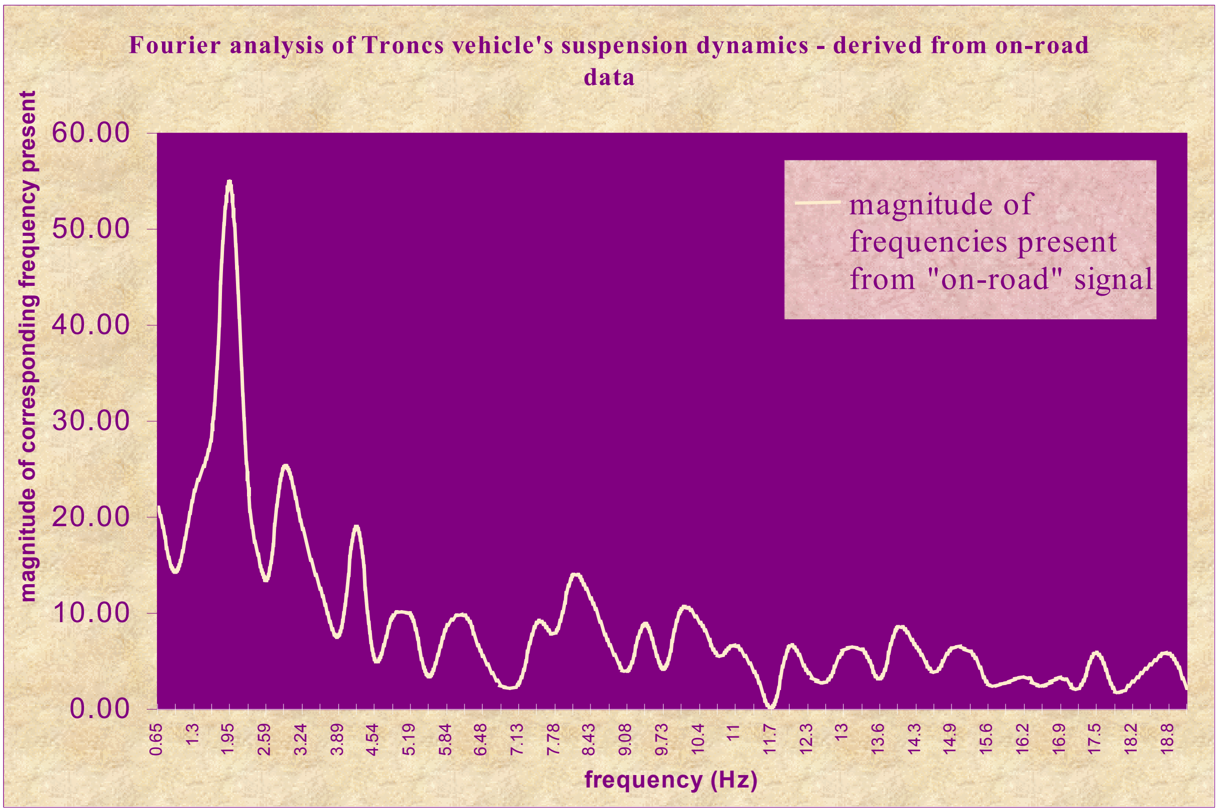

Figure 10 **Fourier plot of trailer axle group mass signal. Frequency (x-axis) is in Hertz (Hz).**

The FFT of the on-road signal data did not, therefore, yield pure clean plots of frequency magnitudes. Even allowing for the overall noise created by the FFT process, we can see that the greatest magnitude frequency present in the FFT of the on-road signal in Figure 10 is that of 1.9 Hz.

The EU Standard contained in ECC directive 92/7/ECC for the natural frequency, f_n of road friendly suspensions is that they must be 2.0 Hz (2 cycles per second) or less. This suspension has been certified as meeting the EU standard and the manufacturer via Rizzo (2003) has confirmed that the suspension subject to these tests has a natural frequency of 1.81 Hz.

It is therefore highly likely, given the suspensions' certification, that the plot in Figure 10 is indicating the natural frequency of the suspension.

It is to be noted that some lower frequencies appear to be present in the Fourier plot and that there are greater magnitude frequencies at approx. 3 Hz, 4 Hz, 8 Hz, 14 Hz and 17 Hz. Prem (2003) has advised that axle hop appears to be the explanation for the 14 & 17 Hz signals and the 3 & 4 Hz signals may be due to other modes of vibration such as prime mover pitching or "pig-rooting", vertical bending of the trailer frame and bending of the truck frame etc, but further research is needed on these signals. Cebon (1999) concluded that tyre elasticity and oscillation appears to be the explanation for the 8 Hz signal.

Conclusion

The test semi-trailer was certified that it met the EU Standard for damping ratio, ζ and natural frequency, f_n. The test results match the certified values of damping ratio and natural frequency. This provides a preliminary validation of the test methods used as the derived damping ratio and natural frequency match the certified values extremely well.

This exercise indicates that, by using the simple tests of driving over a 50 mm pipe and driving on normal, uneven roads, then analysing the data provided by the on-board mass measurement system, the characteristics of heavy vehicle suspensions may be determined by simple calculation and FFT plots.

More work needs to be done to analyse the data from these tests further and also to perform more tests, for example, analysis of a continuous signal from the pressure transducer instead of one that is digitised. However, what the foregoing shows is that the technology and analysis tools exist to indicate that accurate, on-board determination of suspension parameters using inexpensive methods are possible where instrumented vehicles are in the fleet. Should such testing be formalized by combining vehicle telematics and remote monitoring technologies, the possibility exists for heavy vehicle operators to have healthier suspensions on their fleet without recourse to expensive testing facilities. The labour costs of replacing any healthy suspension parts deemed to be beyond their expiry date or the damage to roads from suspensions outside their correct specification could be reduced by employing such low-cost testing. Road authorities would also have confidence in the road friendliness of any heavy vehicles thus analysed and reported, with mutual benefits to all concerned.

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