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EVALUATING THE PERFORMANCE OF 3-PIECE BOGIES ON SHORT DEFECTS AND WITH UNEQUAL WHEEL DIAMETERS

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SUMMARY

Suspension tests on three piece bogies were conducted at the Centre for Railway Engineering Heavy Testing Laboratory. Two 3–piece bogies were tested; both had constant force type friction wedge dampers. One bogie had constant force wedges housed in the bolster. The other bogie had constant force wedges housed in the sideframes. The bogies tested were not reconditioned with wedges half worn or more.

Tests were completed using a coal wagon ballasted with scrap steel to simulate a loaded wagon. Track profile inputs were simulated using position controlled hydraulic cylinders. Both bogies were subjected to bounce tests at 3 and 7 Hz and sinusoidal rail profile inputs of wavelengths 3.0 m with simulated vehicle speeds of 40 and 80 kph. Further variation in suspension behaviour was investigated by simulating variations in wheelset wheel diameters. Wheelset diameter mismatches of 24mm were simulated for first and second axles on the bogie. Rail profile magnitudes tested were 24 mm peak to peak.

The bounce tests illustrated the inadequacy of the friction damping at resonance. The simulation of rail profile inputs, which resulted in pitch motion of the bogie sideframes resulted in unexpected hysteresis performance. The work demonstrates the limitations of the friction dampers commonly used on freight bogies and highlights the need for more detailed mathematical models in wagon simulation.

1 INTRODUCTION

The use of dry friction wedge dampers (snubbers) in three-piece bogies is extensive in freight bogies in North America and Australia [1],[2]. Designs differ but can be grouped into two groups defined by the method used to apply the force to the wedge. The methods are: Constant force (where a spring provides a constant force on the wedge, independent of the load in the wagon) and variable force (where the wedge springs support a proportion of the wagon weight and the force on the wedge varies with wagon load and dynamic movements of the bolster). The three common types of friction wedge arrangement are shown in Figure 1. Types (a) and (b) load the wedge with a constant preload. Type (a) is used on National 'C1' and 'C1 Wedgelock' bogies while type (b) is used on 'American Steel Foundries (ASF) Ride Control' and 'Super Service Ride Control' bogies. Type (a) with wedges located in the sideframe are less common than type (b). Type (c) is used on Standard Car Truck (Barber) bogies as well as ASF 'Ridemaster', GSI 'Aligned' and National 'Swing Motion' bogies.

2 EQUIPMENT AND METHOD

A 70 tonne capacity gondola wagon was arranged in the Heavy Testing Laboratory at the Centre for Railway Engineering as shown in Figure 2. The set up allows the suspension to be excited from displacement inputs applied to the axle box pedestals of the sideframes via shaker beams which take the place of the wheelsets. Input displacement to the wagon is provided by servo controlled cylinders. The displacement is measured by displacement transducers within the hydraulic cylinders. The force applied to the wagon system is measured by the difference in hydraulic pressures above and below the hydraulic cylinder piston. The movement of the bolster relative to the sideframe is measured at both the leading and trailing wedge interface. Data acquisition was completed using a sample rate of 1000S/s per channel. The wagon was ballasted to simulate loaded wagon behaviour. The bogies tested were not reconditioned and the snubber wedges could be described as half worn or more. The assessment of wear was based on wedge height measurements.

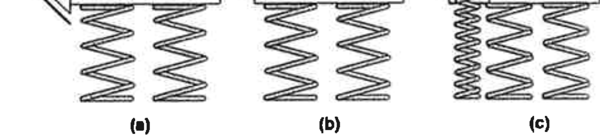

Figure 1 Common Wedge Damper Systems

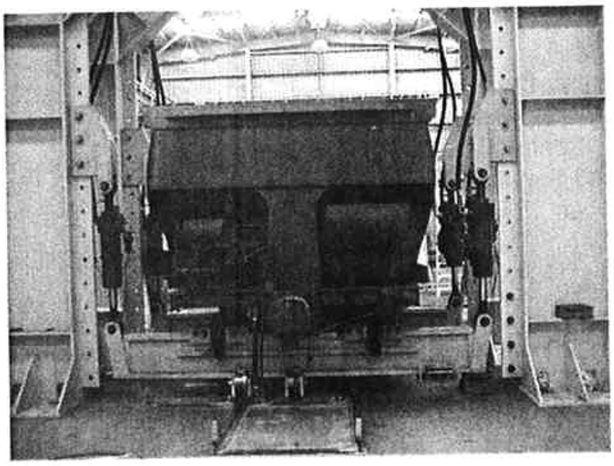

Figure 2 Wagon Suspension Test Rig

Control software was developed to allow any combination of track surface profiles to be applied by the hydraulic cylinders. Wagon speed is simulated by the displacement rate or frequency of oscillation of the cylinders. A large range of input magnitudes, frequencies and combinations were completed, however, due to the restrictions for this technical paper only the results from tests indicated in Tables 1 to 3 will be presented.

Table 1 Bogies Tested

Bogie Detail	Wedge Type
National C1 Friction Dampers	Figure 1,(a)
Super Service Ride Control	Figure 1,(b)

Table 2 Input Profiles

Mode	Z, mm (pk to pk)	Freq (Hz)	Defect Length (m) and Speed, (kph)
Bounce	24	3.0, 7.0	NA
Pitch	24	3.7; 7.4	3.0m, 40kph 3.0m, 80 kph

Table 3 Unequal Wheel Diameters

Test No.	Wheel Diameters
1	Equal
2	Front +24mm
3	Rear +24mm

Bounce profiles are achieved by synchronised inputs to all axle pedestals, Figure 3. Pitch mode involves the delay of the input to the second axle by the duration equivalent to the axle spacing divided by the simulated speed. The input for the pitch mode is a series of individual sine waves, Figure 4. The wheel diameter difference was simulated by adjusting the vertical position of the cylinders to simulate the mismatch in axle heights. The diameter difference of 24mm is simulated by a difference in the side frame pedestal positions of 12mm. The difference of 24 mm was selected to ensure that the effect could be identified. Rail operators usually allow a limit of about half this value.

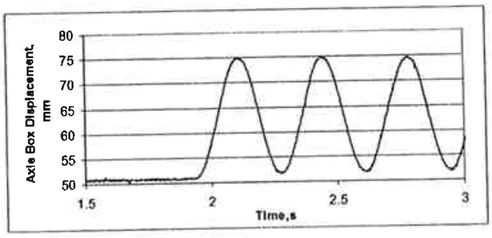

Figure 3 Bounce Mode Input

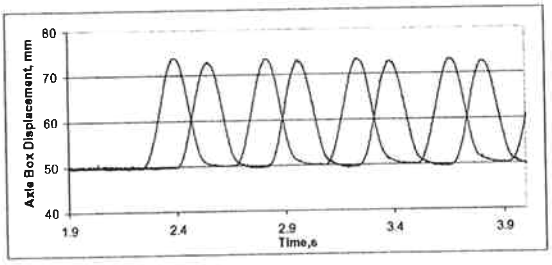

Figure 4 Pitch Mode Input

3 RESULTS

A selection of results plots are shown in Figures 5 to 15. A summary of these results are also given in Table 4. The forces plotted are the forces measured at the hydraulic cylinder. The hydraulic cylinder force includes the inertia due to the shaker beam mass. For a 7 Hz excitation at 24mm peak to peak magnitude, the inertia has a maximum value of 14 kN, or 20% of static load. The result is less significant for a 3 Hz excitation with a maximum of 2.5 kN or 3.5% of static load. The inertia term in a sinusoidal input is at a maximum at maximum displacements, so the effect of the inertia is seen in the rounding of force displacement hysteresis plots in the higher frequency plots. Due to the corrections needed to the force measurement the suspension movement is also plotted against the input displacement to indicate the damping behaviour using kinematic data only. In the force plots damping is indicated by the difference between the loading and unloading curves, the larger the difference the larger the damping force. Likewise in the kinematic data, larger damping levels are associated with larger difference between loading and unloading curves or larger 'phase lag'.

Figures 5 to 8 give a comparison of older design National 'C1' Type bogie damping system to the more recent design, the super service ride control snubber system. These tests are for bounce mode with equal wheel diameters. Figure 9 shows the same bounce mode for a super service bogie with the front wheelset with a 24 mm larger diameter.

Figures 10 to 15 show the responses of the bogie in pitch mode for 3m long track defects at 40 kph and 80kph and the three variations of wheel diameters.

Table 4 Results Summary

Bogie Detail	Test Details	1L Wedge Displ, mm	2L Wedge Displ, mm	1L Force Hysteresis, kN	2L Force Hysteresis, kN
National C1 Friction Dampers	Bounce, 3 hz, EWD, Figure 5	70	70	30	40
	Bounce, 7 hz, EWD, Figure 7	30 *	30 *	20 to 40 **	20 to 40 **
Super Service Ride Control	Bounce, 3 hz, EWD Figure 6	70	75	20	25
	Bounce, 7 hz , EWD, Figure 8	35 *	35 *	10 to 40 **	10 to 40 **
	Bounce, 3 hz, W#1 +24mm, Figure 9	75	75	15 to 20	15 to 20
	Pitch, 3.7 hz, EWD, Figure 10	30	25	25	8
	Pitch, 3.7 hz, W#1 +24mm, Figure 11	36	22	30	8
	Pitch, 3.7 hz W#2 +24mm, Figure 12	32	20	25	6
	Pitch, 7.4 hz, EWD, Figure 13	20 *	18 *	20 **	45 **
	Pitch, 7.4 hz, W#1 +24mm, Figure 14	20 *	20 *	20 **	40 **
	Pitch, 7.4 hz, W#2 +24mm, Figure 12	17 *	20 *	20 **	40 **

Notes: All inputs 24 mm peak to peak
EWD = Equal Wheelset Diameters
W#1 +24mm = 1st Wheelset diameter larger by 24mm
W#2 +24mm = 2nd Wheelset diameter larger by 24mm
* Total displacement during test, displacement per cycle ~ 10-15 mm.
** Stick slip behaviour

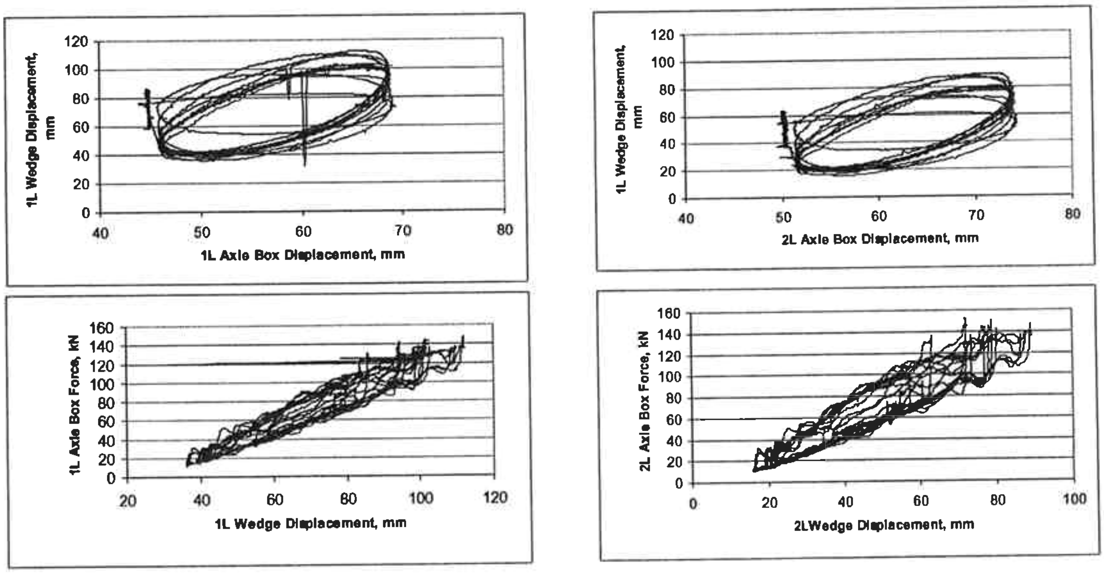

Figure 5 National 'C1' Type Bogie, 3 Hz Bounce Mode, Equal Wheelset Diameters

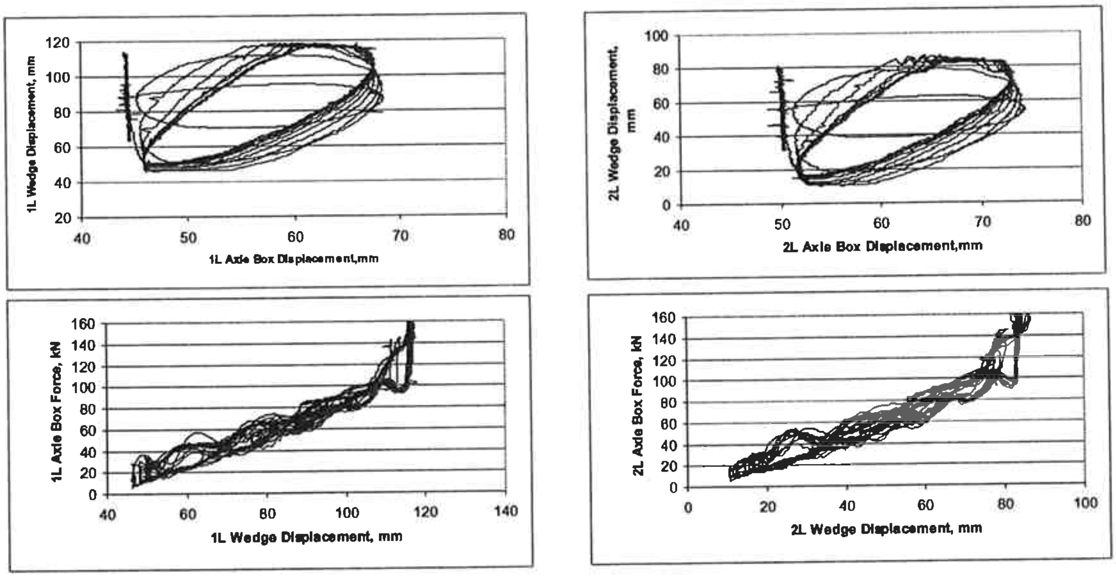

Figure 6 Super Service Ride Control Bogie, 3 Hz Bounce Mode, Equal Wheelset Diameter

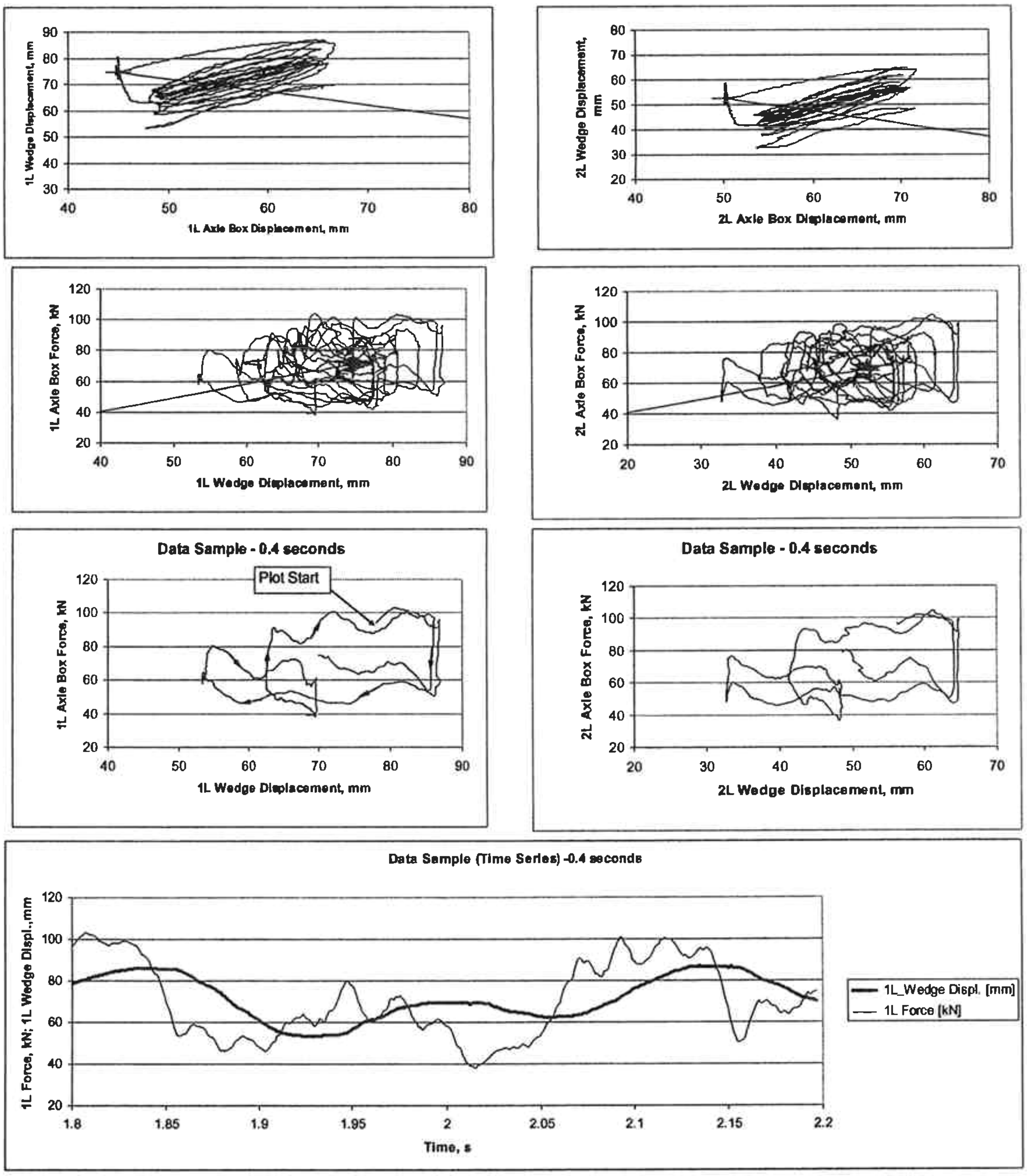

Figure 7 National 'C1' Type Bogie, 7 Hz Bounce Mode, Equal Wheelset Diameter

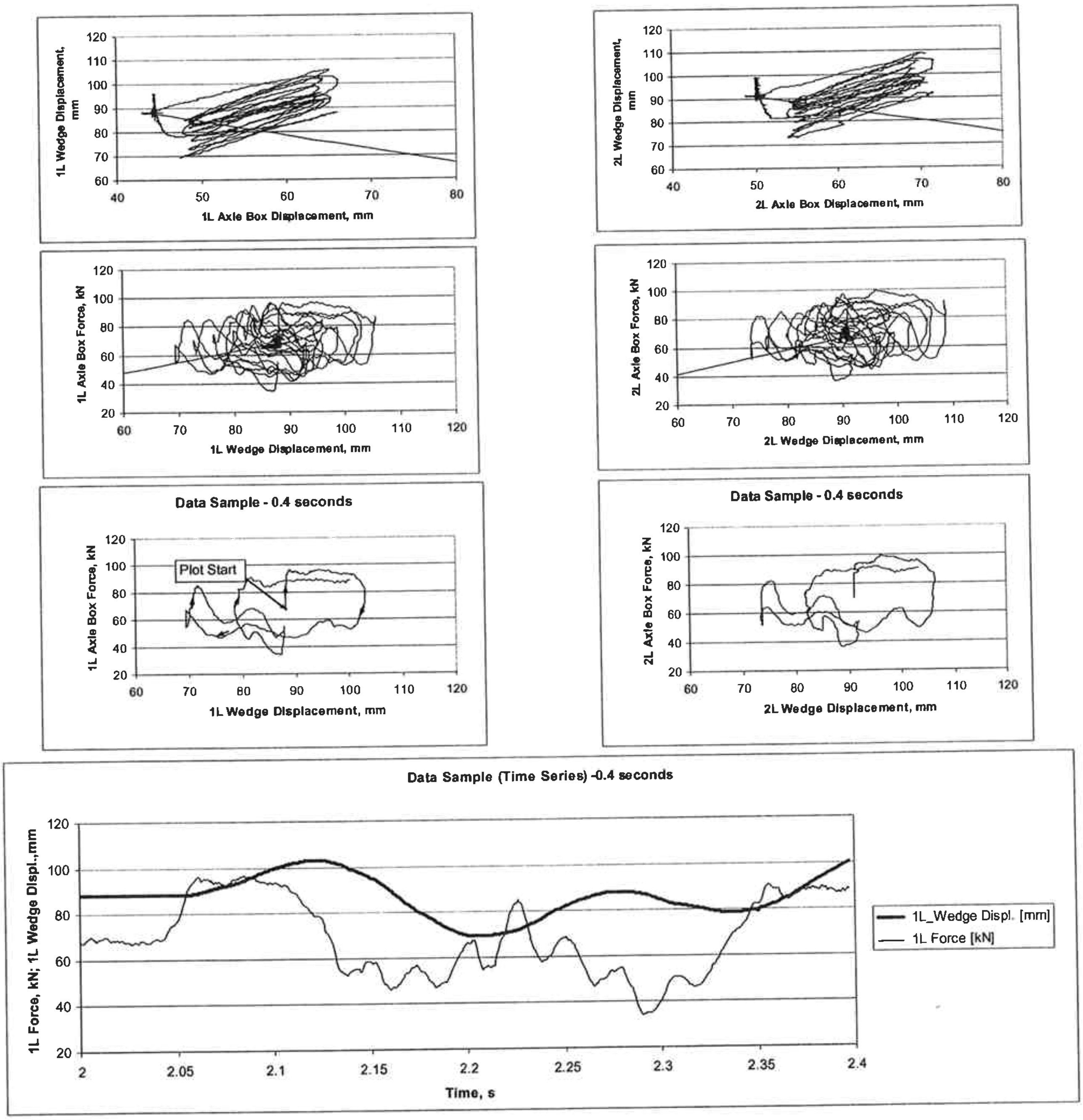

Figure 8 Super Service Ride Control Bogie, 7 Hz Bounce Mode, Equal Wheelset Diameters

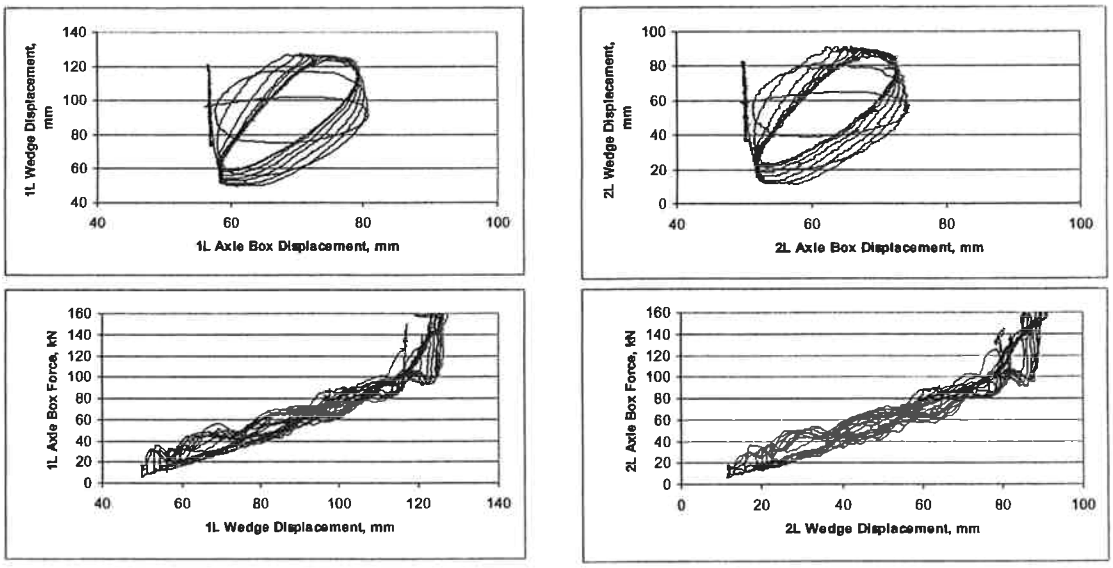

Figure 9 Super Service Ride Control Bogie, 3 Hz Bounce Mode, First Wheelset +24mm

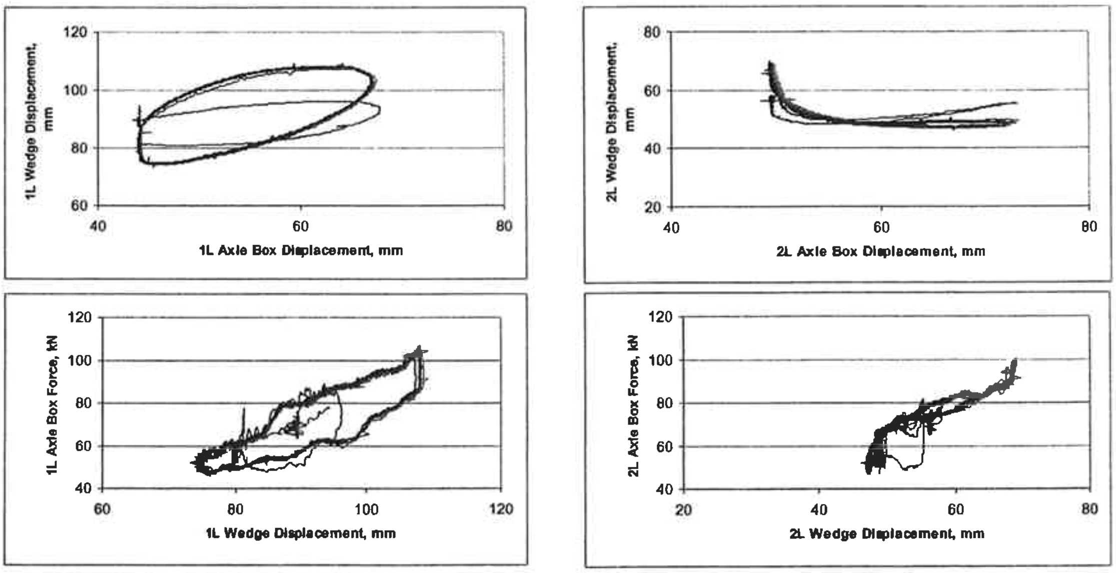

Figure 10 Super Service Ride Control Bogie, 40kph, 3.7 Hz Pitch Mode, Equal Wheelset Diameters

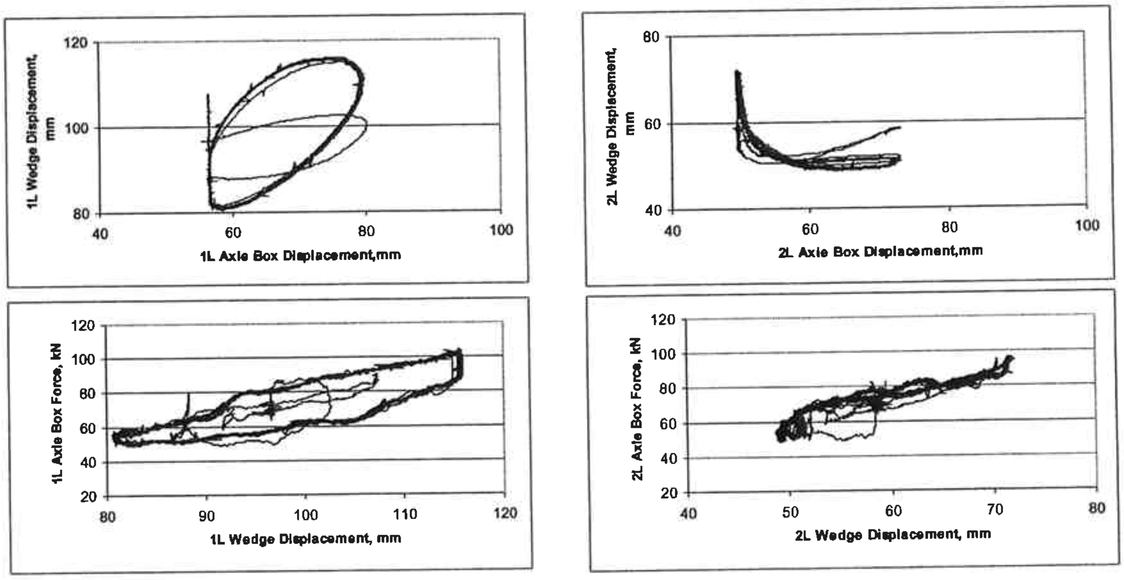

Figure 11 Super Service Ride Control Bogie, 40kph, 3.7 Hz Pitch Mode, First Wheelset +24mm

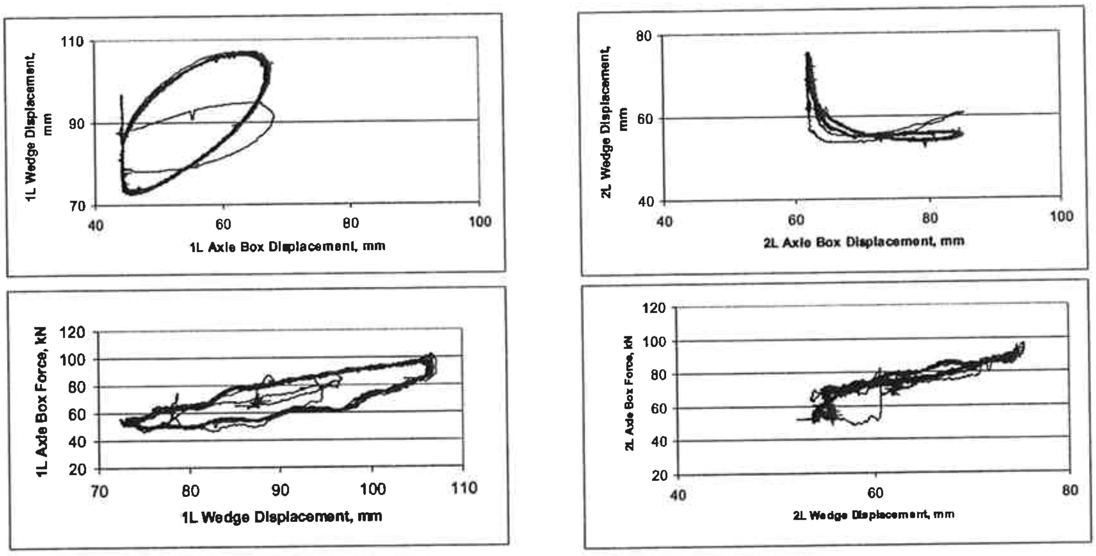

Figure 12 Super Service Ride Control Bogie, 40kph, 3.7 Hz Pitch Mode, Second Wheelset +24mm

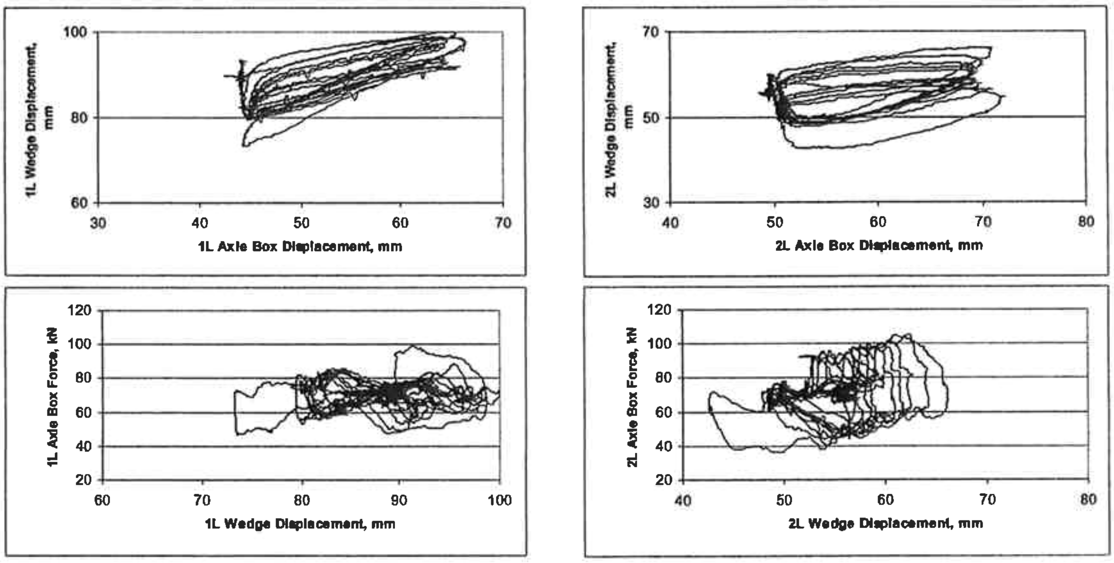

Figure 13 Super Service Ride Control Bogie, 80kph, 7.4 Hz Pitch Mode, Equal Wheelset Diameter

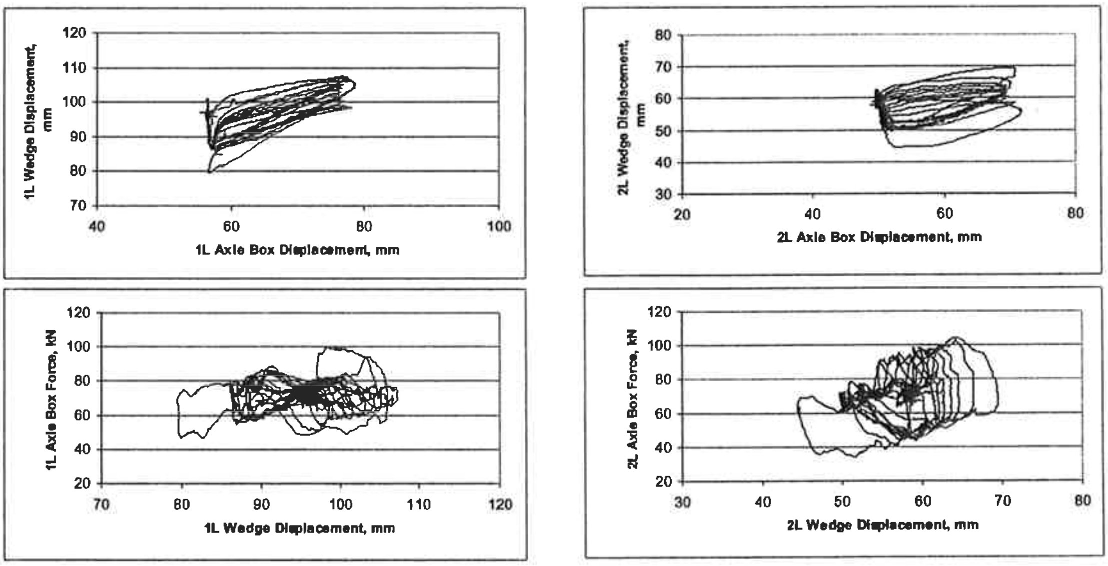

Figure 14 Super Service Ride Control Bogie, 80kph, 7.4 Hz Pitch Mode, First Wheelset +24mm

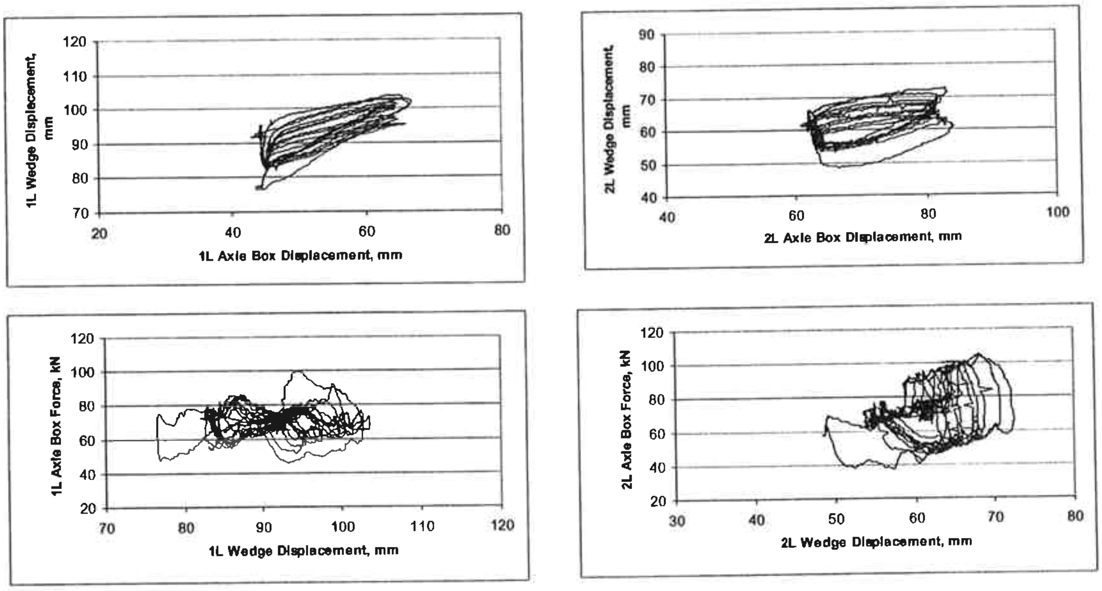

Figure 15 Super Service Ride Control Bogie, 80kph, 7.4 Hz Pitch Mode, Second Wheelset +24mm

4 DISCUSSION

Further to the notes regarding the measurement of force noted in the Results section, the force measurements actually give a close approximation of the vertical wheel-rail contact force. The reason for this is that the wagon shaker beam weighs approximately 1.3 tonnes, which is slightly heavier than a wheelset. As these tests are restricted to bounce and pitch modes, excluding roll modes, the roll inertia is not significant in the calculations. Also for bogie pitch modes, if it is desired to examine actual forces at the spring nest the following correction equation is required, (Figure 16).

Using the free body diagram in Figure 16 and applying moments the following corrections can be obtained,

$$W_1 = (1/d)[F_1(L+d)/2 - F_2(L-d)/2 + I\alpha - mad/2 - mgd/2]$$

$$W_2 = (1/d)[F_2(L+d)/2 - F_1(L-d)/2 - I\alpha - mad/2 - mgd/2]$$

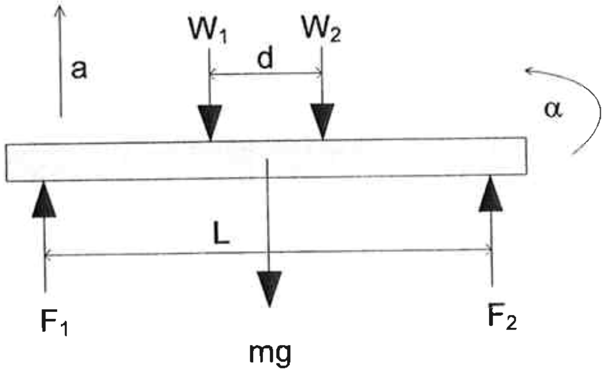

Figure 16 Free Body Diagram for Calculating Pedestal and Spring Nest Forces

Note that the equations can be used to correct forces for both the position of the cylinders on the shaker beam and the difference between forces at the axle box and at the spring nest. For this paper neither of these corrections were used due to the difficulty in obtaining accelerations for the inertia terms from digitised measured displacement data. . Rather than partially apply the corrections it was decided to just present the measured results. It is useful to see the effect of the suspension at the wheel-rail interface rather than at the spring nest.

The results in Figures 5 and 6 are as expected. Note that the suspensions give quite large amplitudes, ~70mm for the National 'C1' Type bogie and similarly 70mm for the Super Service Ride Control bogie, (see also Table 4). Note that there is evidence of the Ride control bogie reaching hard limit. These amplitudes represent a relative motion transmissibility factor of about 3.

The excitation of 3 Hz was chosen as it is close to resonance of the wagon with the ballast added. The resonant frequency is slightly higher than would be expected because the wagon is only half ballasted, (30 tonnes of steel scrap in one end and not placed in exactly the right position to simulate a fully loaded half wagon) and the motion is slightly rotational due to only one suspension being excited. The results highlight why cyclic top waves in track, that can induce bounce mode, present significant derailment risk. These results show that the damping system provided by both wedge designs do not adequately control resonance.

Higher frequency responses in Figure 7 and 8 show a loss of normal suspension behaviour. The average position of the bolster relative to the sideframes does not stay in the same place each cycle, i.e. the bolster appears to walk both up and down. While the net damping is reduced, as evidenced by the kinematic curves, the suspension still behaves well with amplitudes limited to ~30mm, and a relative motion transmissibility of ~1. As discussed earlier there would be reduction of the forces reported at the extremes of displacement due to inertia terms not accounted for in measurement. The stick slip behaviour is in fact so severe it is very difficult to see any effects of the inertia terms being neglected. Note that the suspension appears to have negative slope or "softening" response in some regions, see the additional short duration cross plots and time series plot. Following softening there is more normal response with positive stiffness and friction damping The behaviour can only be explained by stick behaviour followed by a sudden loosening of wedges to such an extent that the stiffness appears to be negative. Note also that the net hysteresis seems to reduce as the cycling continues, (see arrows in the 5th plot). The overall suspension response approximates to a softer suspension with damping retained.

A single example of bounce mode is included for different wheel diameters in Figure 9 for brevity. Only slight increases can be noted (+5 to +10mm) in suspension movement when compared with Figure 6. No change in performance due to wheel diameter mismatch is evident, although the maximum hysteresis is slightly reduced, Table 4.

The results in figures 10 to 15 all refer to pitch modes induced by a relatively short defect with a 3m wavelength simulated at 40 and 80 kph. Of interest is the response at 40 kph, 3.7 Hz which is close to resonance. The suspension provides a damped response to the leading axle and minimal damping to the trailing axle. It is not correct to infer from this that the trailing wedge provides no damping. Note the correction equations given earlier in the discussion. The actual forces measured at the suspension would show some hysteresis at both leading and trailing wedge positions. It must, however, be concluded that some of the damping force is lost due to the pitching motion of the sideframe. The pitch motion tests, Figures 10,11 and 12 show up to 30% reduction in the sum of the friction wedge forces as compared to the bounce mode, see Table 4. Comparing the cases of equal wheelset diameters with mismatched diameters, Figure 10, 11 and 12 it is noted that the sum of the friction wedge forces increase by 15% for the larger front wheelset and reduce by 6% for larger second wheelset. These changes are probably due to wedge bedding rather than design as it would be expected that damping would reduce for both wheelset cases. No significant change in behaviour due to wheelset size differences can therefore be noted.

The results in Figures 10, 11 and 12 present some interesting challenges for wagon modelling as it is not usual for the relative pitch of the bolster and side frame to be considered as a factor which could change the damping force provided by the wedges. Wedges are manufactured with a convexed surface to reduce the effects as noted. It was not possible to examine the wedges before testing so it could be that wear had reduced the capabilities of the suspension to maintain damping during sideframe pitch. This does not remove the need to understand these variations and develop mathematical models to correctly simulate the response. There are many cases where simulation is required of actual wagons rather than new wagons.

The results of pitch mode tests for higher speeds, Figures 13 to 15 does not give the same differentiation between lead and trailing wedge hysteresis. Wedge performance is better than for bounce cases at similar frequency with less evidence of "walking up/down", which was seen in Figure 8. Note that the hysteresis in the front wedges, Figures 13 to 15 , show similar loss of stiffness to examples in Figure 7 and 8. The stick slip behaviour is also evident on the rear wedge, but more overall suspension stiffness is retained.

5 CONCLUSIONS

Testing of two different three piece bogies with vertical excitations in bounce and pitch mode were completed with simulations for wheelsets with equal and mismatched diameters.

For the large track irregularity simulated, 24mm peak to peak, vibration close to resonant frequencies was not adequately damped by the snubber system with a relative motion transmissibility factor of approximately 3. This does not mean that the snubber system is inadequate for more normal track irregularities as in non-linear damped systems lower input amplitudes are controlled by suspension locking.

Higher frequency excitation resulted in stick-slip behaviour in bounce modes but response amplitudes were controlled with a relative motion transmissibility factor of approximately 1.4.

The negotiation of a short severe defect of 3m wavelength and 24 mm amplitude at a simulated speed of 40 kph demonstrated a loss of up to 30% of the friction damping force in the snubber system due to sideframe pitch.

There were no significant differences in suspension behaviour due to wheel diameter mismatch in the test data analysed. It should be noted that these tests did not investigate the effects on the damping of lozenging motions. It is therefore premature to conclude that greater levels of wheelset diameter mismatch can be tolerated.

Improved suspension modelling of the behaviour of wedge dampers during the pitch of the sideframe relative to the bolster is required to understand and simulate some responses due to short wavelength defect inputs.

6 ACKNOWLEDGEMENTS

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7 REFERENCES

[1] Hawthorne,V.T, *Evolution of the Modern Three Piece Bogie*, International Heavy Haul Conference, Montreal, Canada, 1996

[2] Guin,S.G, *Three Piece Truck, Yesterday, today and Tomorrow*, International Heavy Haul Conference, Montreal, Canada, 1996