A BRAKING FIGURE OF MERIT

PhD (1990) from Monash University, Victoria Australia. Peter has worked as a consulting vehicle-engineer for the past 17 years. He is the author of the Australian Heavy Vehicle Braking Strategy and has been the Chairman of the Australian Road Transport Suppliers Association (ARTSA) since 2007. Specialist domains are combination vehicle braking, forensic engineering and electrical and electronic designs.

P. M. HART
Chairman, ARTSA
Hartwood Consulting Pty Ltd
Melbourne, Australia.
pmhart@hartwood.com.au

Abstract:

This paper proposes a braking-performance *figure of merit* for heavy combination vehicles. A *braking figure of merit* could be a useful indicator of likely braking capability of combination vehicles before coupling. It's utility might be compared to that of the *Static Roll-Over Threshold (SRT),* which is a figure of merit for roll-over performance without the need to test.

A useful *braking figure of merit* should reliably predict 'acceptable' combinations of vehicles according to some desirable performance standard. The *braking figure of merit* that is proposed here is the calculated straight-line stopping distance with the brake control set to achieve threshold wheel lock-up. The calculated stopping distance is then normalized by the theoretical stopping distance for a vehicle that has perfect brake balance on the same surface. The result is called the:

Stable Stopping Distance (SSD %) =

$$\frac{100 \text{ x Minimum straight-line stopping distance without wheel lock-up occurring}}{\text{Theoretical minimum stopping distance } (V^2/2g\mu)}$$

The closer the *SSD* is to 100%, the shorter the stopping distance whilst maintaining directional control.

This paper builds upon braking stability tests that were conducted by ARRB Group on a semi-trailer as part of the ARTSA Brake Code of Practice project. A prime-mover and trailer were instrumented to allow the brake level of each vehicle to be varied independently. The semi-trailer was braked to a stop on a wet, sealed and curved roadway from the same initial speed, and the maximum deceleration for which the vehicle stayed in a 3.7m lane width, was determined. Four different load conditions were used. These tests provided a ranking of stable braking performance for different vehicle brake set-ups. The ranking was compared with that predicted by the *SSD* calculation. The tests also illustrate the relative braking performance that can be achieved by a European and Australian (similar to a North American) vehicles.

Keywords: Truck brakes, air brakes, dynamic stability during braking, brake balance.

1. Introduction

Stable braking is very important for the road safety of combination vehicles. Stable braking exists when the driver can confidently stop the vehicle at a high rate without losing directional control. Adequate braking performance of heavy (combination-) vehicles depends upon both the brake capability and brake balance. Even though each brake on a combination vehicle might be capable of generating say, 6 kN / tonne at full brake control, the ability of the combination vehicle to safely achieve even 50% of its theoretical minimum stopping distance whilst remaining directionally stable, depends upon the brake balance of the vehicle.

Foundation brake capacity can usually be assessed using manufacturer's data and the vehicle brake capacity can be estimated once the air-system characteristics are accounted for. Brake balance is more conceptual. It assesses the extent to which the brake effort is shared between the brakes in proportion to the load carried. A vehicle with poor brake balance wastes the available road friction because tyres lock-up at low brake levels and directional control becomes the limiting factor. The driver will avoid making harsh brake applications if the brake balance is poor.

Brake balance is affected not only by brake capability but also by vehicle dimensions, load distribution, load height and suspension reactions. Further, it changes slightly with the severity of the brake application and with brake temperatures. Achieving reasonable brake balance irrespective of load is the fundamental challenge of heavy-vehicle braking.

The Australian market has European, North American, Japanese, Chinese and Australian vehicle makes and there are different brake-system designs reflecting the design philosophies of the region of origin. There are also different brake technologies encompassing disc, drum, load-sensing, antilock and electronic distribution within each pedigree.

Australia never mandated compliance with unladen brake compatibility limits or antilock brakes, but some vehicles have technologies that arise from these requirements. European trucks are likely to have Electronic Brake Control Systems (EBS) that manage brake performance of the truck (although they are typically set to expect a European trailer is being pulled). Consequently, there is a wide range of possible brake technologies used on Australian combinations. Poor brake balance is likely to exist on many vehicles because some technologies that should not be mixed, are mixed.

Therefore, the minimum stopping distance that a (combination) vehicle can achieve depends upon both the brake capacity, the brake balance, brake technologies, vehicle dimensions and the load level and distribution. Whilst it is possible to brake test a combination vehicle, this is problematical because of the propensity of combinations to become directionally unstable. Therefore the question then is: *Can the braking performance of a combination vehicle be reliably predicted using relatively simple computations?* This paper seeks to answer to this question.

2. Compatibility Testing

Stopping tests were conducted on a semi-trailer that had been modified so that the brake capacity at each axle could be varied. Effectively, the brake balance could be altered between each of the three axle groups. The maximum deceleration that could be achieved without the combination vehicle leaving the test lane was tested. These results allowed a performance ranking to be made that could be compared (in Section 6) with a calculated *braking figure of merit* (‘*Stable Stopping Distance’*).

The tests were conducted on a curved concrete test track that was continually wetted (see Figures 1 & 2). The tests were modeled on the ‘braking in a curve test’ that is in the US braking rule FMVSS 121. This rule requires demonstration of directional control on a 500 ft (152.4 m) radius path of 9 ft (3.7 m) width, by a semi-trailer vehicle that has a prime-mover with antilock brakes. The test entry speed was nominally 30 mph (48 km/h).

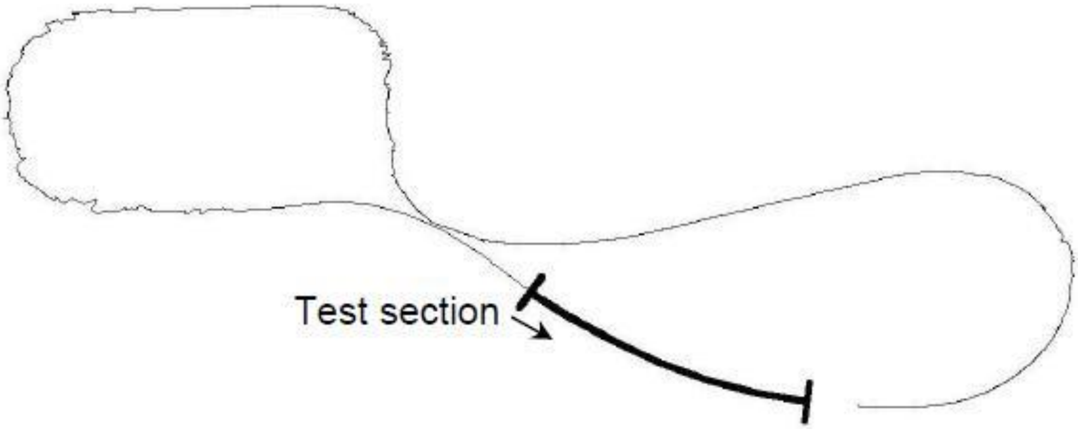

Figure 1 A GPS recorder display of the test track at DECA Shepparton, Victoria.

By varying the relative levels of the regulated air supplies, the test vehicles could be set-up to simulate common Australian and European arrangements. For some tests the lightly-laden and half-laden trailer group was set-up to simulate load-sensing brakes (with a 65% valve setting). The brake set-ups are described in Section 3.

Whilst about 10% of Australian heavy vehicles now have advanced electronically-controlled brake controls, it was not intended to explore this performance. The test vehicles had any antilock or EBS controllers disabled.

The test prime-mover and its semi-trailer were modified so that the foundation brakes on each of the three-axle groups were supplied from an independently-controllable and regulated air pressure tanks. Braking was triggered by depressing the clutch pedal, which initiated sudden electrical operation of three solenoid valves and thereby applied the preset air pressure to the brake actuators of each axle group, with the engine disconnected.

The brake control level was increased progressively until the vehicle could not be stopped within the 3.7 m wide lane. The penultimate test was a pass and its deceleration and retardation force levels are reported here. The penultimate test was repeated twice to ensure consistent performance.

A data logging system was used to record time, speed (GPS based) and (lat-long) position. Six air pressures were also recorded, which were the air tanks supplying the three axle groups and one actuator in each axle group. The electrical signal that initiated brake action was also recorded. Stopping distance and deceleration were calculated from the speed v time record.

The semi-trailer had a load frame fitted so that the added load was lifted by about 1.5 m (see Figure 3). This was done to produce a relatively high center of mass so that weight transfers during braking were exaggerated.

Tests were conducted with four different load levels, which were:

- Laden: General access load limits on each axle group.
- Half Laden Even: The added load on the trailer needed to achieve the Laden condition was halved. The added load was distributed evenly on the trailer.
- Half Laden Uneven: The added load was half that for the Laden case. This load was placed at the front of the trailer deck.
- Unladen: No added trailer load. The loading frame was left on the trailer.

The test loads are shown in Tables 1 & 2. Note that the position and height of the Centres of Mass in Table 1 are for the load component and not the (laden) vehicle. The heights of the Centers of Mass (C of M) are estimated. The longitudinal positions of the Centers of Mass have been calculated, based upon the known static laden axle weights.

Figure 2 The test vehicle on the test track.

Figure 3 The load at the front of the trailer was at floor level. A 1.5 m high load frame was used to increase the height of the load towards the rear.

Table 1 shows the measured (and estimated) load centers of the vehicle parts and Table 2 the values for the three axle-groups.

Table 1 Locations and heights of the load components of the vehicle parts.

Load component	**Weight**	**Position of C of M**	**Height of C of M above axle height**
Prime mover without trailer	10.71 t	2.03 m from steer axle	0.5 m
Trailer without load	8.6 t	5.85 m from kingpin	0.7 m
½ load (added) – drive heavy	12.2 t	1.14 m from kingpin	1.3 m
½ load (added) – even distribution	11.9 t	4.66 m from kingpin	1.7 m
Full (added) load	24.2 t	4.55 m from kingpin	1.75 m

Table 2 Calculated axle-group and kingpin weights for each test condition

Axle Group Loads	Unladen (t)	Half Load, Even (t)	Half Load, Drive Heavy (t)	Laden (t)
Prime mover – steer axle	4.90	5.19	5.47 / 5.8	5.51 / 5.9
Prime mover – drive group	5.81	10.92	15.81 / 15.7	16.50
King Pin imposed load	2.70	8.10	13.27	13.99
Trailer – tri-axle group	5.90	12.40	7.53	18.81
Load weight and height	0	11.90 at 4.66 m back from kingpin. C of M height of load is 2.2m	12.20 at 1.14m back from kingpin. C of M height of load is 1.8 m	24.22 at 4.55 m back from kingpin. C of M height of load is 2.25m
Total vehicle weight	16.61	28.51	28.81	40.82

3. Vehicle Features

The test prime-mover was an Isuzu Giga 6 x 4 truck with disc brakes. The trailer was manufactured by Maxitrans and has a tri-axle rear group with disc brakes. The trailer has an air-bag suspension whilst the truck has a spring suspension on both axle groups. The steer tyres were 295/80R whilst all other tyres were 11R22.5.

Initial testing determined the contact pressure and the fully-established average brake force for each axle. The results of contact pressure tests for wheels on each axle group are the x-axis intercepts in Figure 4. The fully developed average brake force values were calculated from the vehicle deceleration tests with brakes active on one axle group.

The deceleration tests were conducted in a straight line, from a speed of 60 km/h on a wet road, with control pressures measured at the trailer control coupling up to 270 kpa (40 psi). At this control level, wheel lock-up did not occur. Figure 4 shows the total vehicle-retardation average forces ascribed to the braking axle group.

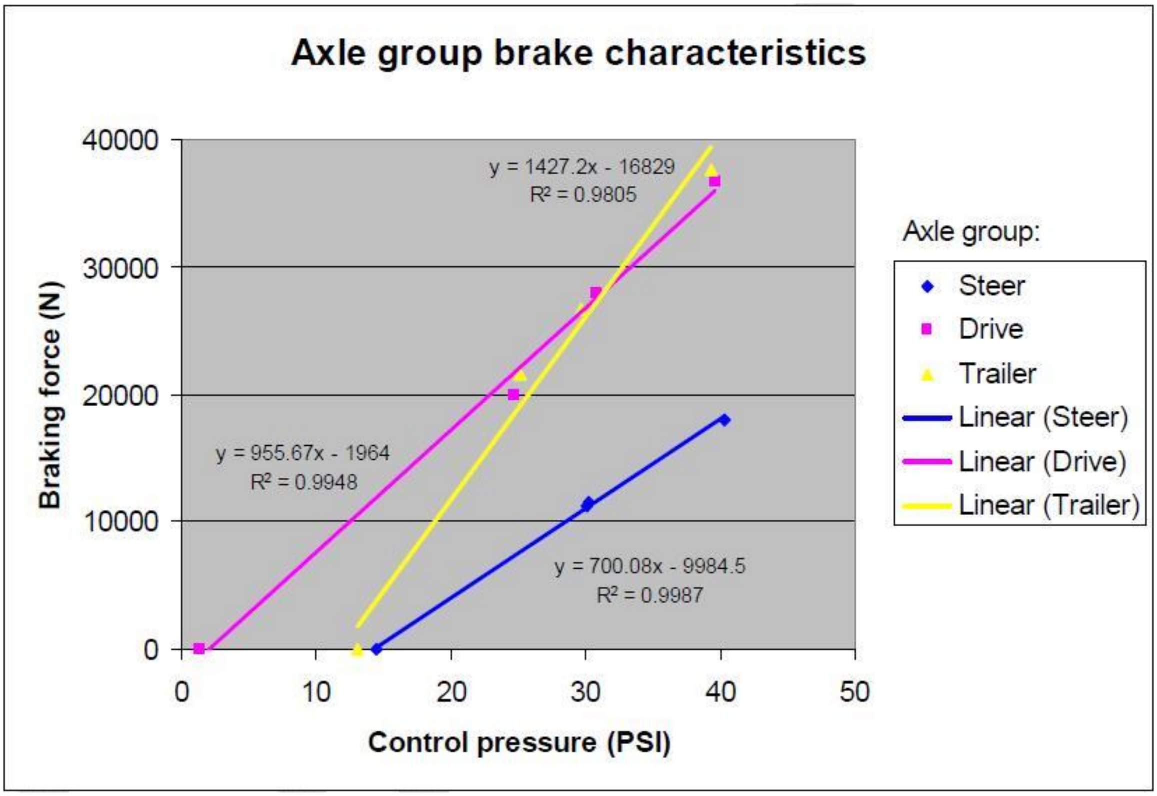

Figure 4 Axle-group force characteristics.

The characteristics in Figure 4 show:

- The contact pressures of the steer and trailer (tri-axle) groups are about the same. The contact pressure of the drive-axle group is significantly lower (by ~ 77 kPa or 11psi). This combination vehicle has poor contact pressure balance. Because all testing was done with control levels in the range 0.2 – 0.6E (160 – 350 kPa), the threshold pressure balance is unimportant.
- The retardation force curves are approximately proportional to the control pressure above the applicable contact pressure. The proportionality suggests that other non-braking retardation forces were negligible.
- The relative axle group torques are:

Steer	18,500 / 27 psi = 99.3 N/kPa
Drive axles	36,500 / 38 psi = 139.2 N/kPa
Trailer axles	37,000/ 27 psi = 198.6 N/kPa

4. Test Vehicle Set-Ups

Six different vehicle set-ups were tested. The combinations were:

- 'ADR' truck with 'ADR' trailer.
- 'ADR' truck with 'ECE' trailer.
- 'ADR' truck with 'ADR' trailer with simulated load-sensing brakes (LSB).
- 'ECE' truck with 'ADR' trailer.
- 'ECE' truck with 'ECE' trailer.
- 'ECE' truck with 'ADR' trailer with simulated load-sensing brakes (LSB).

Note that throughout this paper the term “truck” is used to mean the prime-mover and “trailer” means the semi-trailer. “Vehicle” means the combination of the truck and trailer.

The ADR and ECE terms refer to the assumed typical settings of vehicles that comply respectively to the **A**ustralian **D**esign **R**ules 35 (trucks) & 38 (trailers) and with the **E**conomic **C**ommission for **E**urope Regulation 13. Figure 5 shows a comparison of the compatibility limit lines from the applicable ADR rules with those in ECE Regulation 13, together with the test truck and trailer settings. The differences between the ADR and ECE limits for a prime-movers are relatively minor. For a semi-trailer the differences are substantial. The ECE trailer is set to comply with the appropriate ECE R13 limits for each loading condition.

The ECE Regulation 13 limit lines for a semi-trailer are calculated using a formula that accounts for the centre of mass and the kingpin – center axle dimension (‘S-dimension’). The ECE R13 limits shown in Figure 5 have been calculated using the assumed centre of mass of the laden trailer (2.25m above the ground). This is relatively high because a load frame was used to achieve a high center of mass; which promotes load transfer forward during braking and hence a worst-case braking condition.

Whilst these rules allow a range of brake settings, typical settings identified by the author were used. These nominal settings are illustrated in Figure 5, which show the three-truck and three-trailer theoretical vehicle characteristics on the ADR compatibility limit plane. The ADR truck was set-up so that its compatibility characteristic when laden, was half way between the limit lines at 450 kPa control pressure; passing through the point (450, 0.43).

The ADR trailer was set-up so that its compatibility characteristic when laden was half way between the upper limit and the center line between the limits. That is, passed through the point (450, 0.48). For some tests a Load Proportioning Valve (LPV) was simulated. The trailer set-up for the unladen tests and the drive-heavy tests was set at 65% of the laden trailer set-up. The trailer set-up with half-load evenly distributed was set at 82.5% of the laden set-up, which was the expected mid-load automatic performance.

The ECE truck and trailer set-ups were calculated using an ECE calculation program provided by Knorr Bremse. The prime-mover and trailer characteristics were set to be half way between the applicable upper and lower limit lines in both laden and unladen conditions.

The curves in Figure 5 do not adequately illustrate the differences in brake distribution between the ADR truck and the ECE truck. The ECE truck was set with significantly greater steer axle brake power than the ADR truck. Compared with the ADR truck, the ECE truck brake power is:

Steer:	130 %
Drive:	97 %
Total truck:	105 %

Hence it is important to recognize that there are differences in the brake distribution between the ECE and ADR prime-movers. The ADR prime-mover has brake distribution biased to the drive-group axles whereas the European prime-mover has less bias between the front- and rear-groups.

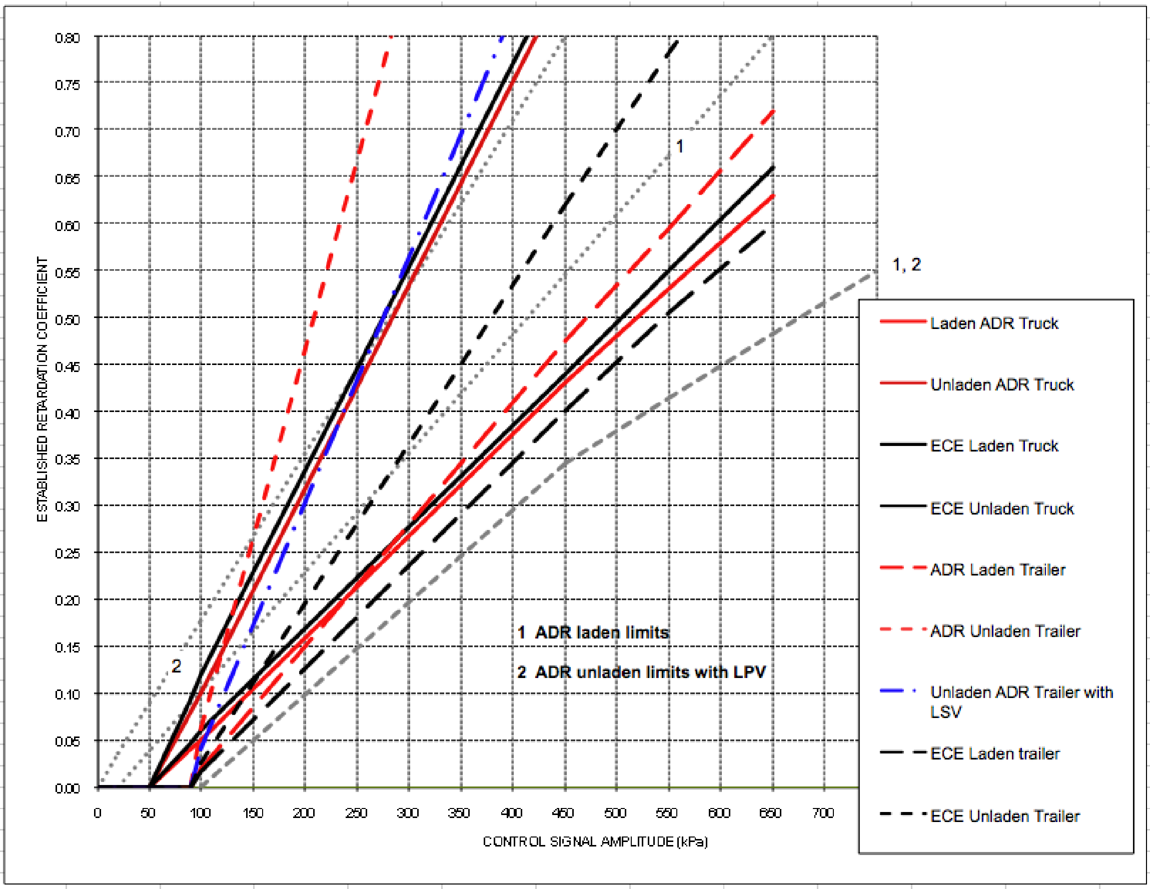

Figure 5 The nominal vehicle set-ups. The tests were conducted with control pressures in the range 160 – 390 kPa.

5. Deceleration Results

The deceleration results for the penultimate-tests (for which the test vehicle stopping within the lane width) for all test configurations are shown in Table 3. Note that "INT" denotes the load-sensing valve set-up on the trailer that was set to 100% (full load) and 65% (light load).

The graphs shown in Figures 6 – 13 show the established deceleration performance without the vehicle crossing the course boundaries. That is, the penultimate test performance. The number of locked-axles that were observed during the failed (ultimate) test is also reported in the Figures. An axle that is reported to have locked-up may have one or two locked wheels. The dynamic mode (under-steer, jack-knife, trailer swing) is not shown on the graphs because firstly; several lock-up modes often existed and secondly; the two observers (who were on different sides of the vehicle) sometimes reported different lock-up modes.

The friction utilization (retardation force / weight) values shown in Figures 6 - 13 were calculated using dynamic axle-group weights that were estimated using the known deceleration and the static axle weights. The axle retardation forces are assumed to be those predicted from the preliminary characterizing tests (Figure 4).

Table 3 Maximum stable average deceleration results, given in normalized (g) units. E is the applied control level. E =1 corresponds to 625 kPa.

Truck	**Trailer**	**Laden**	**Half Laden (Even)**	**Half Laden Drive Heavy**	**Unladen**
ECE	ECE	0.273g E=0.6	0.246g E=0.55	0.260g E=0.5	0.217g E=0.4
ECE	ADR	0.214g E=0.4	0.259g E=0.55	0.162g E=0.25	0.156g E=0.2
ECE	INT	* (same as ECE + ADR)	0.242g E=0.5	0.202g E=0.35	0.161g E=0.25
ADR	ECE	0.223g E=0.45	0.246g E=0.5	0.254g E=0.48	0.221g E=0.3
ADR	INT	* (same as ADR+ADR)	0.256g E=0.5	0.248g E=0.45	0.245g E=0.3
ADR	ADR	0.260g E=0.5	0.246g E=0.4	0.185g E=0.3	0.244g E=0.3

The applied control level values E in Table 3 indicate indicates how hard a driver might push on the brake pedal to achieve the stop.

Figures 6 – 9 pertain to the ADR-truck tests. The ADR truck and ADR trailer combination (Figure 6) exhibits gross wheel lock-up for most tests. The dominant failure mode is trailer swing although jack-knife might also occur. This combination is the dominant semi-trailer set-up used in Australia.

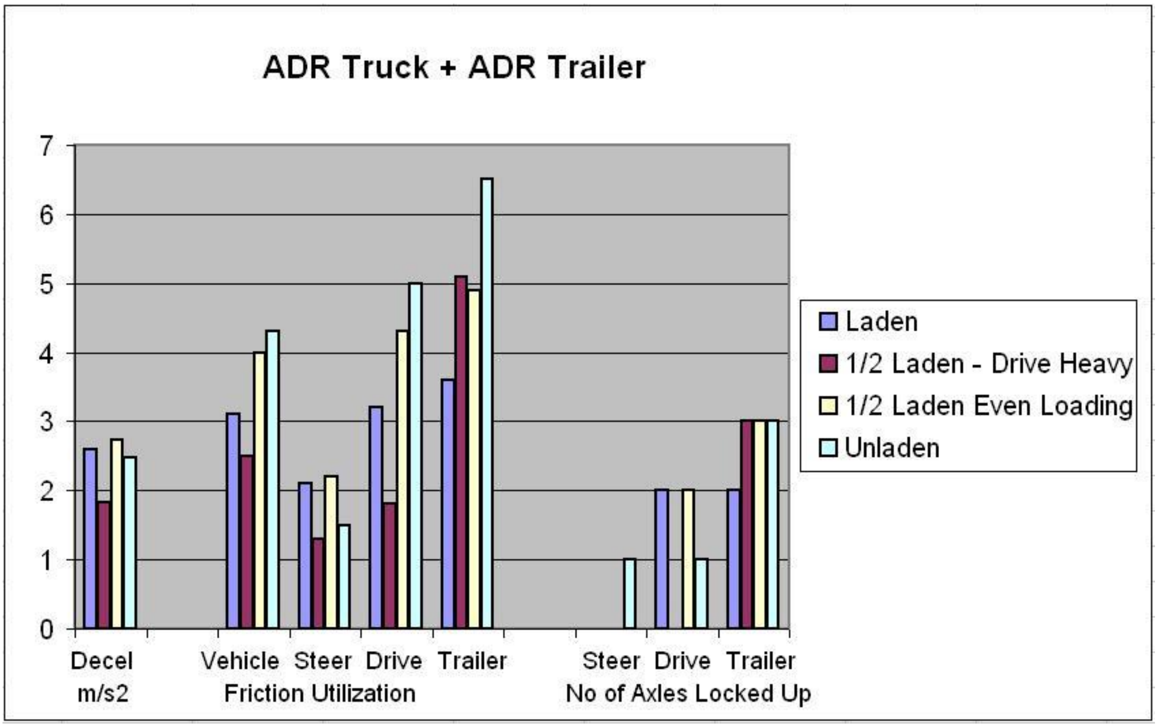

Figure 6 ADR truck & ADR trailer.

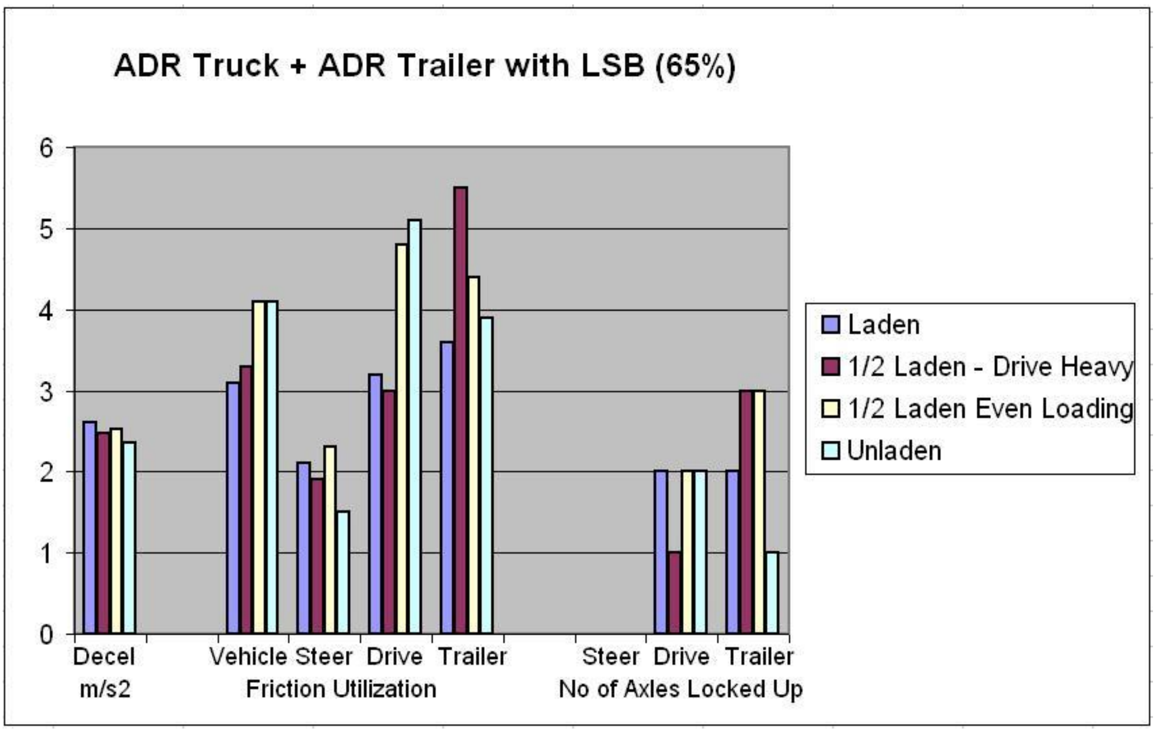

Figure 7 ADR truck & ADR trailer with an LPV set to 65% (unladen).

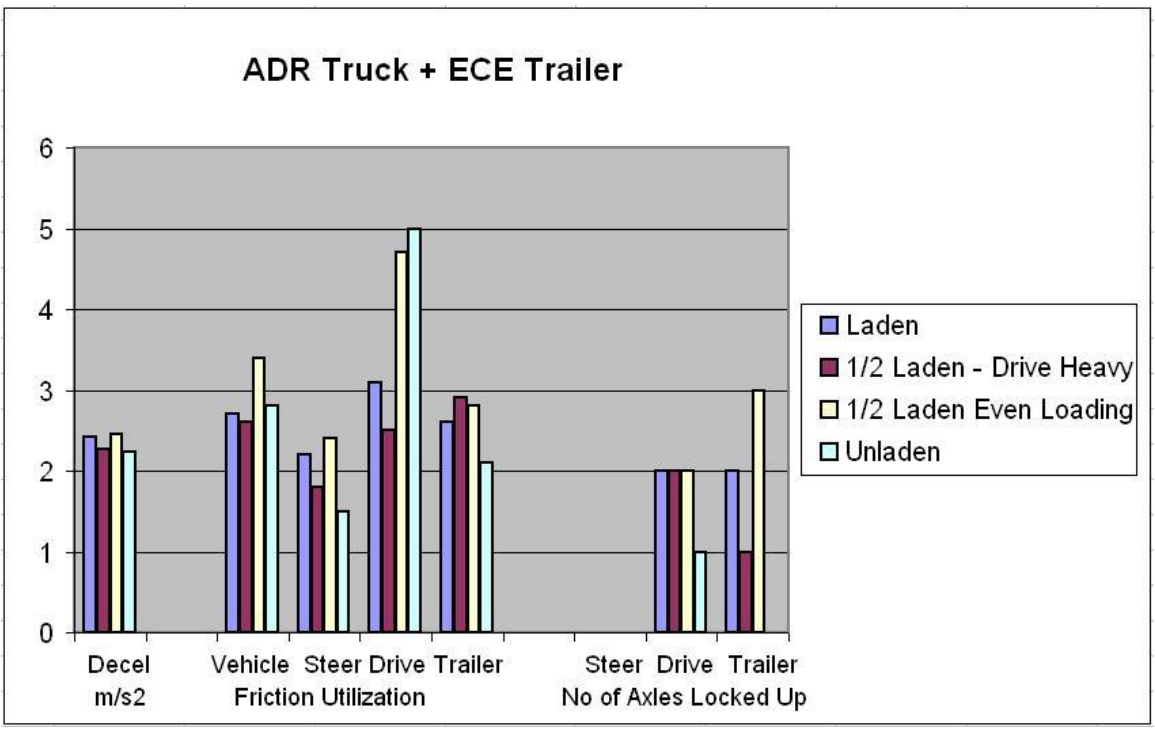

Figure 8 ADR truck & ECE trailer.

The application of a load-sensing valve to the ADR trailer reduces the extreme trailer utilization levels (compare Figures 6 and 7). The performance for the drive-heavy and unladen load conditions are improved, as might be expected. The failure mode might be jack-knife or trailer swing. Lowering the LSB setting is unlikely to improve the performance because this will increase the propensity for jack-knife.

When the trailer brake power is reduced further (ECE trailer, Figure 8) the performance is about the same as that of the ADR trailer with the 65% LSV setting. The dominant failure mode is now jack-knife. No steer axle lock-up was observed.

Figures 9 – 11 pertain to the ECE truck tests. Steer axle wheel lock-up is more prevalent for the ECE truck than the ADR truck because of the greater brake capacity on the steer axle of the ECE truck. The unladen performance of the ECE truck with the ADR trailer is about the same as the ADR truck with the ADR trailer.

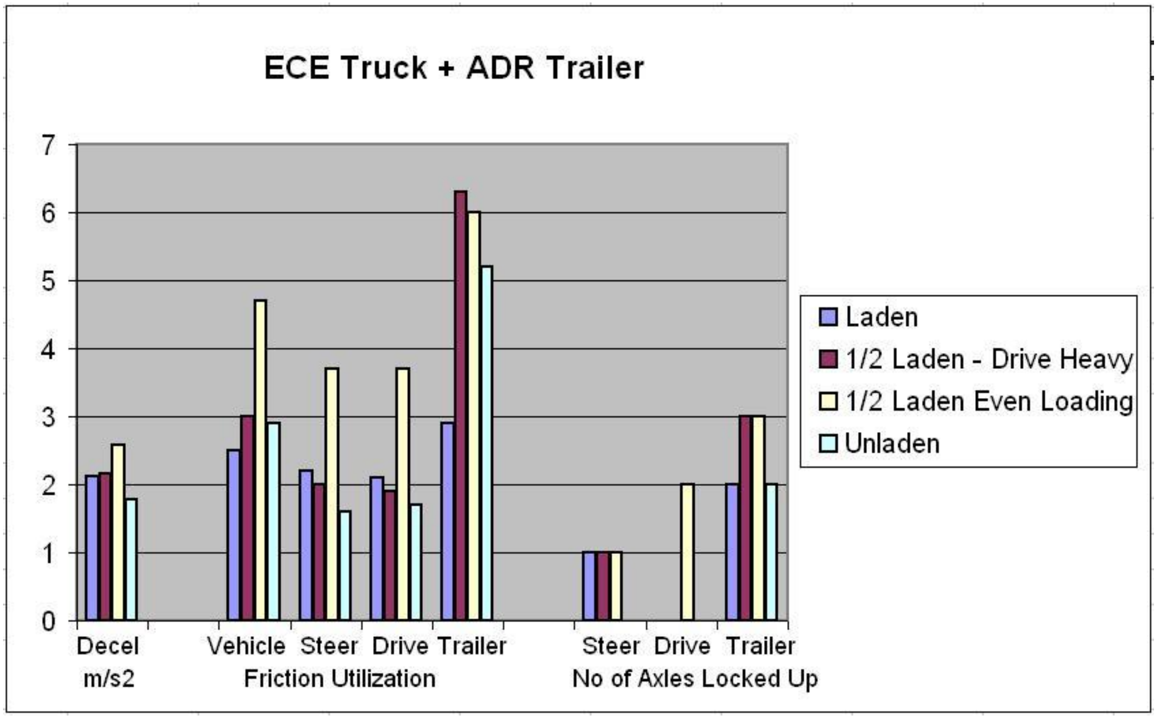

Figure 9 ECE truck & ADR trailer.

The ECE truck attached to the ADR trailer (Figure 9) is marginally better than the ADR truck with the ADR trailer (Figure 6), because the steer axle is doing more work. Consequently there is a lesser tendency for drive-group lock-up to occur. There is no significant change in the deceleration levels that are achieved.

The deceleration performance of the ECE truck with the ECE trailer (Figure 10) is about equal best (along with the ADR truck with the LSB trailer – Figure 7). The significant difference is that the steer axle on the ECE truck is doing more work and under-steer (with jack-knife) occurred routinely. The ECE truck + ECE trailer has the best brake balance of all combinations. Utilization numbers up to 4.5 occur.

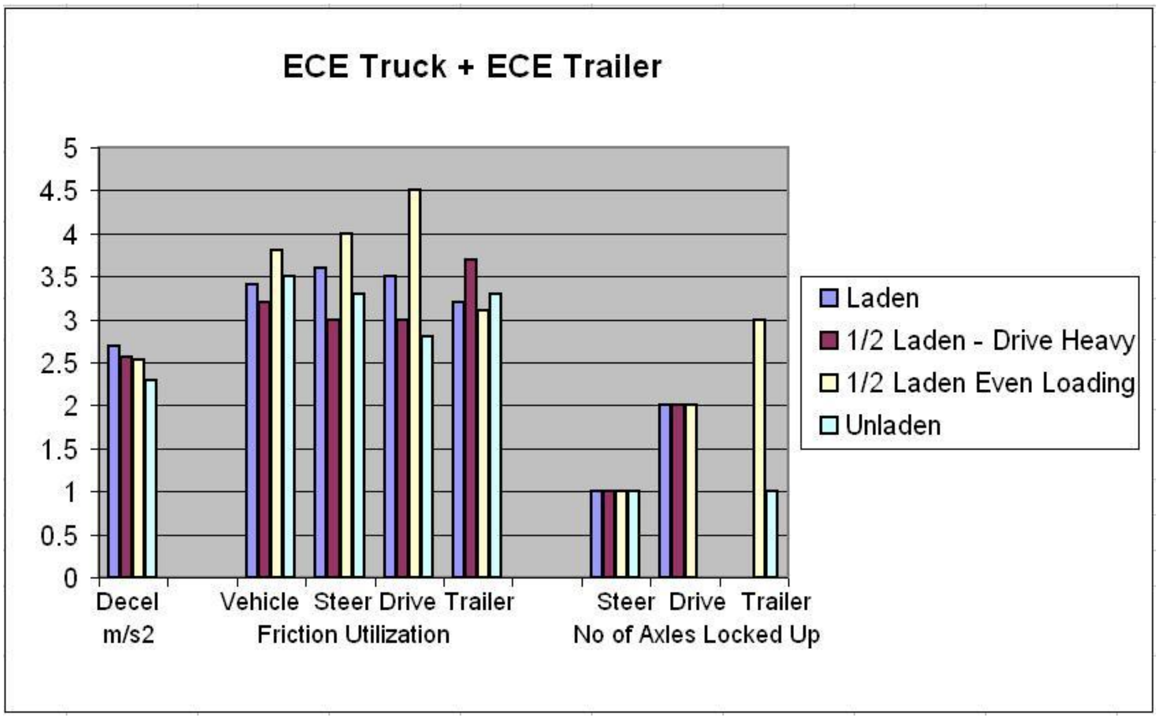

Figure 10 ECE truck & ECE trailer.

The ECE truck pulling the ADR trailer with a 65% LSV (Figure 11) has good overall performance except that the unladen performance was low due to steer-axle lock-up. The driver reported lack of steering.

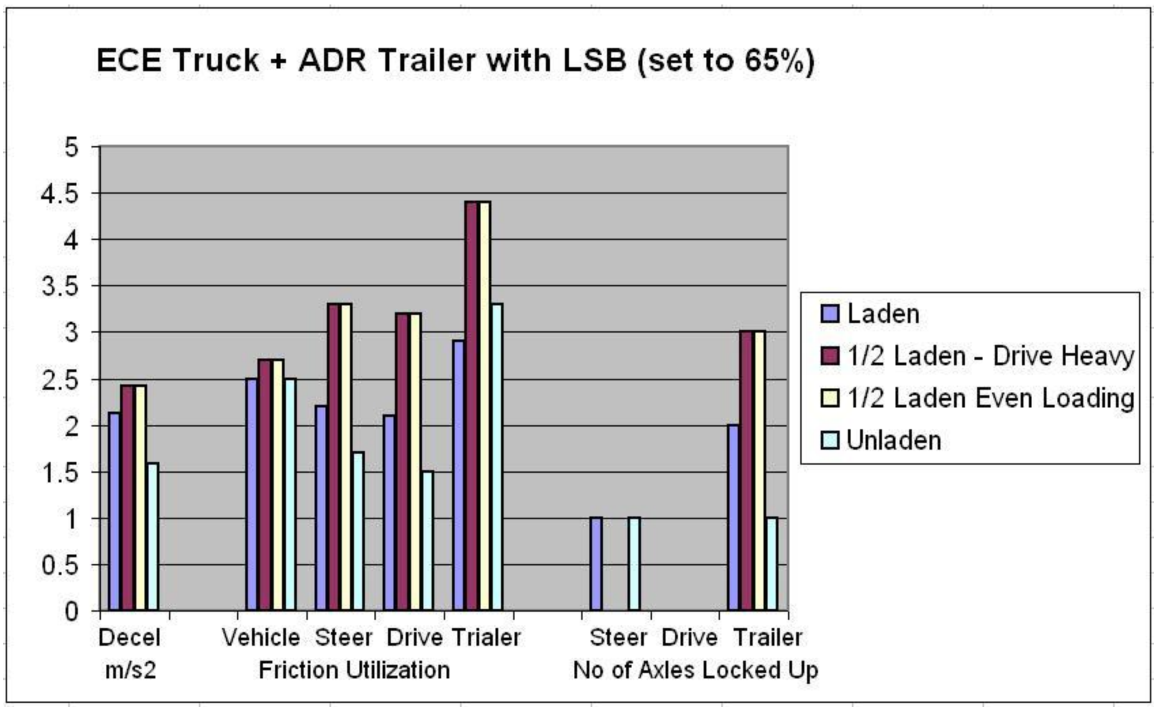

Figure 11 ECE truck & ADR trailer with LSV set to 65%.

6. A Braking Figure of Merit

Table 4 shows the implied straight line stopping distances for each of the maximum stable average decelerations results given in Table 3. These distances have been calculated using the straight-line stopping distance formula $s = V^2/2a$. Whilst there is a systematic error involved because the vehicle was not braked in a straight line, the distances do facilitate a comparison of the various configurations. Thus the ECE truck with the ADR trailer would take about 1.6 times longer to stop safely for the half-laden drive-heavy condition than the half-laden even condition.

Table 4 also gives (in brackets) the indicative stopping distances normalized by the theoretical stopping distance with perfect brake balance, which is $V^2/2g\mu$. The purpose of doing this is to provide an index of performance that is independent of the actual tyre-friction level μ. The problem however, is the μ was not accurately known for the tests. The nominal friction level of the dry test track for truck tyres is 0.9 and it is estimated that the friction of the liberally-wetted track was 0.5. This was however, not measured. The point of estimating these normalized stopping distances is that they can be compared with calculated values.

The calculated stopping distances shown in Table 4 assume the test vehicle was stopped in a straight line, when it was not. The limiting performance in the tests was loss of directional control rather than maximum longitudinal friction utilization. This raises the issue of what criteria should be applied for loss of directional control.

Table 4 Implied stopping distances based upon the average decelerations in Table 3; and the *Normalized Stable Stopping Distance* (expressed as a % in brackets) for an assumed tyre-road friction level of $\mu = 0.5$.

Truck	**Trailer**	**Laden**	**Half Laden (Even)**	**Half laden Drive Heavy**	**Unladen**
ECE	ECE	33.2 m (183%)	36.8 m (203%)	34.8 m (192%)	41.8m (230%)
ECE	ADR	42.4 m (233%)	35.0 m (193%)	56.0 m (308 %)	58.1 m (321 %)
ECE	INT	* (same as ECE + ADR)	37.5 m (207%)	44.9m (248%)	56.3 m (311%)
ADR	ECE	40.6 m (224%)	36.8 m (203%)	35.7m (197%)	41.0 m (226%)
ADR	INT	* (same as ADR+ADR)	35.4 m (195%)	34.6 m (202%)	37.0 m (204 %)
ADR	ADR	34.8 m (196%)	36.8 m (203%)	49.0m (270 %)	37.4m (205%)

The straight-line stopping distances with one axle group on the point of wheel lock up were predicted for all using a relatively simple simulation that was implemented on a spreadsheet. The steps involved are:

1.	The brake force capacity of each axle group is calculated by applying the brake control level (E) that was used for each reported test (see table 3) to the axle-force characteristics (Figure 4).

2.	The simulated brake control level E was increased in small steps until the friction utilization (F/W_s) of one axle group equals or exceeds the assumed road friction level μ. W_s is static weight.

3.	The vehicle deceleration is calculated and then the dynamic weights on each axle group are calculated.

4.	The friction utilization on each axle group is recalculated using the dynamic weights (F/W_d). W_d is dynamic weight.

5.	The steps 2 - 4 are repeated using the dynamic weights W_s as stating weight. A second estimate of the static weights and hence of the friction utilization of all axles when lock-up first occurs, is obtained.

6.	A better estimate of the vehicle deceleration is obtained at which wheel lock- up just occurs (i.e. $F/Ws = \mu$ first occurs on one axle group) is re-estimated. The stopping distance is calculated using $s = V^2/2a$.

7.	Load shifts within an axle group are ignored as are timing delays.

8.	The stopping distance results are normalized by the theoretical stopping distance that can be achieved with perfect brake balance; which is $V^2/2\mu g$.

So the simulation estimates the straight-line deceleration for which one axle group has wheels at the point of lock-up ($F/Ws = \mu$). This is taken to be the limiting stable braking deceleration because above this braking level, wheel lock-up will occur and the vehicle might slide sideways. Once the limiting declaration is known, the stopping distance can be calculated and then the *Stable Stopping Distance SSD* is calculated:

Stable Stopping Distance (SSD %) =

$$\frac{100 \text{ x Minimum straight-line stopping distance without wheel lock-up occurring}}{\text{Theoretical minimum stopping distance (} V^2/2g\mu)}$$

The assumed road-friction level μ is irrelevant in the calculations because it affects both the top and bottom lines in the *SSD*. Table 5 gives the calculated *Stable Stopping Distances (SSD %),* which can be compared with estimated test stable stopping distances shown in Table 4.

The definition of stable braking that has been used is different for the test measurement compared to the simulation. Both assessments are concerned with potential loss of directional control during braking. Inability to stop within a 3.7 m lane was an obvious loss of control during testing whereas lock-up of one wheel was not necessarily evident. In

contrast, it is easy to determine by simple simulation that wheel lock-up has occurred, whereas the simulation cannot predict lateral loss of directional control, because only straight-line braking is simulated. Therefore, the criteria for loss of directional control, are different.
The simulation can be applied to vehicles that have an antilock braking system because the control level that is used is below the wheel-lock-up threshold. If the truck has an advanced brake control system such as EBS, the simulation would not be applicable without further development. This is an aspect that is under further consideration.

Table 5 Calculated *Stable Stopping Distance %* for each test configuration and loading level. The control level that was used in the simulation is also given.

Truck	**Trailer**	**Laden**	**Half Laden (Even)**	**Half laden Drive Heavy**	**Unladen**
ECE	ECE	28.5m (E = 0.65)	26.9m (E=0.6)	28.7m (E=0.55)	34.1m (E=0.45)
ECE	ADR	35.7m (E=0.5)	26.9m (E=0.65)	52.7 (E=0.25)	36.7m (E=0.35)
ECE	INT	(same as ECE + ADR)	28.4m (E=0.5)	42.3m (E=0.35)	40.4m (E=0.3)
ADR	ECE	37.5m (E=0.5)	27.2m (E=0.55)	33.3m (E=0.475)	35.2m (E=0.30)
ADR	INT	(same as ADR+ADR)	23.6m (E=0.55)	38.6m (E=0.50)	32.9m (E=0.35)
ADR	ADR	30.0m (E=0.55)	26.8m (E=0.45)	39.1m (0.50)	34.5m (E=0.35)

Finally, Table 6 gives a comparison of the *SSD%* values (for the assumed $\mu = 0.5$) for both the test decelerations (curved track) and the simulated decelerations (straight line). As expected, the SSD% values are have a different level for each of the two cases however, the ranking of the vehicles is in good agreement. This indicates that the likely stable braking performance of a semi-trailer vehicle under variable load-conditions, can be indicated by a simple simulation.

7. Conclusions

A simple *figure of merit* for heavy vehicle braking has been proposed for heavy combination-vehicles. The figure of merit is the *Stable Stopping Distance (SSD)*. *SSD* is relevant to road safety because it gives an estimate of the distance required to stop a combination vehicle without losing directional control. Further work is progressing to propose suitable maximum values for the *SSD*. The Australian PBS braking standard is based on achieving an *SSD* < 200%.

The *Stable Stopping Distance (SSD)* can be determined by road tests if wheel lock-up can be sensed, however, it is more conveniently determined by simple simulation. Indeed, this is the benefit of the *SSD*. It can be calculated before a vehicle is configured and it will provide a guide to the likely braking stability of the vehicle. The simulation determines the performance at threshold wheel lock-up.

Braking in a curve stopping tests were conducted that provide a basis for the considerations in this paper. The tests provide an interesting comparison between a European and an Australian (or North American) semi-trailer. No one combination is consistently the best for each loading condition. The calculated *SSD* gives an accurate ranking of the measured performances of the various test vehicle configurations.

Table 6 Comparison of the SSD% values for the stopping in a curve tests and the simulated straight-line stopping tests. The ranking of the trucks for each load condition, obtained by both test and simulation are also given. (μ = 0.5)

Laden Condition

Configuration	Test Ranking	Test *SSD%*	Simulation Ranking	Calculated *SSD%*
ECE + ECE	1	183%	1	157
ECE + ADR	4	233%	3	197
ADR + ECE	3	224 %	4	207
ADR + ADR	2	196 %	2	166

Half Laden (Even) Condition

Configuration	Test Ranking	Test *SSD%*	Simulation Ranking	Calculated *SSD%*
ECE + ECE	3	203 %	3	149%
ECE + ADR	2	196 %	2	148 %
ECE + INT	6	207 %	5	157 %
ADR + ECE	3	203 %	4	150 %
ADR + INT	1	195 %	1	130 %
ADR + ADR	3	203 %	2	151 %

Half Laden (Drive Heavy) Condition

Configuration	Test Ranking	Test *SSD%*	Simulation Ranking	Calculated *SSD%*
ECE + ECE	1	192 %	1	158 %
ECE + ADR	6	308 %	6	291 %
ECE + INT	3	197 %	4	213 %
ADR + ECE	4	202 %	3	183 %
ADR + INT	2	195 %	2	156 %
ADR + ADR	5	270 %	3	215 %

Unladen Condition

Configuration	Test Ranking	Test *SSD%*	Simulation Ranking	Calculated *SSD%*
ECE + ECE	4	230 %	3	190 %
ECE + ADR	6	321 %	5	203 %
ECE + INT	5	311 %	6	223 %
ADR + ECE	3	226 %	4	194 %
ADR + INT	1	204 %	1	181 %
ADR + ADR	2	205 %	2	189 %

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