Developing a Set of Performance Based Standards for New Zealand

V2.5

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INTRODUCTION

The traditional method of regulating vehicle size and weight has been through the use of prescriptive limits. Thus, for example, we have limits for the maximum height, width and length of vehicles as well as limits on axle weight, axle group weight and gross vehicle weight. Originally the main purpose of size and weight regulations was to protect the infrastructure. In recent times, size and weight regulations have also been used to try to improve vehicle safety and transport efficiency.

The prescriptive limits approach is relatively straightforward to implement and enforce and equally relatively simple for operators and vehicle manufacturer's to comply with. However, the connection between the prescription limit values and the infrastructure protection or safety outcome is sometimes not very direct. Generally prescriptive limits have evolved over time by a process of incremental change.

Although this process is relatively slow, significant changes have occurred within the design life of the infrastructure elements. For example, the New Zealand Bridge Manual (Transit New Zealand 2003) specifies that permanent bridges shall be designed with a design working life of 100 years while for pavements a design life of 25 years is typical but lightly trafficked roads can last much longer than this. If we look back 50 years to the mid-1960s, most truck transport operations were legally restricted to maximum travel distances of 67km (40 miles) (KiwiRail 2014) with some exemptions for perishable goods and for journeys between origins and destinations not served by rail.   The Transport Act (1962) specifies that the maximum allowable weight on a single axle is eight tons provided the axle is more than eight feet from the nearest other axle. An exception is made for tractor units in an articulated vehicle where the minimum axle spacing is six feet. For more closely spaced axles the axle weight limit is six tons. No mention is made of whether the axles need to be fitted with dual tyres or not and so the implication is that single-tyred axles weighing up to eight tons were permitted. There is also no separate bridge formula specified in the Act, so the bridge loadings can be deduced from the axle load limits and the axle spacing requirements. It should also be noted, that these regulations pre-date the publication of the results of the AASHO road test and so the pavement designers would not have been using the fourth power relationship between axle loads and pavement wear as part of the design process. Remarkably, with maintenance and upgrading, the network has coped with the changes in truck size and weight and traffic volumes that have occurred in the intervening period.

An alternative approach to prescriptive limits is the use of performance-based standards (PBS). A performance measure consists of a prescribed test procedure during which some defined quantity is measured. A performance-based standard additionally defines acceptability levels for the performance measure. The use of PBS for regulating motor vehicles is not new. For example, one of the braking requirements in New Zealand is defined as a maximum stopping distance from 30km/h. This is a performance-based standard and it dates back to the late 1920s. Similarly there is a rollover stability requirement for buses in the UK that dates back to the 1920s. However, these are isolated performance standards relating to a specific aspect of vehicle performance rather than a system of performance-based standards for characterising the whole vehicle.

In the 1970s and 1980s, researchers at the University of Michigan Transportation Research Institute (UMTRI) undertook a number of studies investigating the factors influencing heavy vehicle performance (Bernard, Fancher et al. 1973, Fancher, Bernard et al. 1973, Winkler, Bernard et al. 1976, Fancher, MacAdam et al. 1977, Ervin, Fancher et al. 1978, Winkler and Fancher 1981, Ervin and MacAdam 1982, Winkler, Fancher et al. 1983, Fancher, Ervin et al. 1984). To characterise this vehicle performance they developed a number of performance measures. From these individual performance measures, the concept of using a set of PBS for characterising vehicles was developed in a study undertaken for the Road Transport Association of Canada (RTAC 1986). Each of the Canadian provinces sets its own vehicle size and weight requirements. As a consequence, these regulations varied from province to province and it was difficult to operate inter-provincial truck transport efficiently. The purpose of the RTAC study was to develop a set of standard vehicle configurations that could be accepted by all the provinces and would thus form the basis for inter-provincial road transport operations. The PBS were developed as a basis for quantifying the safety performance characteristics of the proposed vehicle configurations.

The RTAC study was published in 1986 (RTAC 1986) and the findings were disseminated at an international symposium held at Kelowna, British Columbia the same year. Following this symposium, New Zealand was one of the first jurisdictions in the world to start using PBS measures as a guide for assisting in the development of size and weight regulations. Subsequently some of the PBS were used as a basis for permitting vehicles outside of the standard size and weight limits. In 2002 the size and weight regulations were moved into the Vehicle Dimensions and Mass (VDAM) Rule (Land Transport Safety Authority 2001). Again PBS were used to develop some aspects of the Rule and most notably a rollover stability performance standard applying to nearly all large heavy vehicles was incorporated in the Rule. This was a world-first. Other jurisdictions specify rollover stability requirements for special vehicles where the potential outcome of a rollover is particularly serious (dangerous good tankers and buses) but nobody else has made it a requirement for the general fleet. The 2002 VDAM Rule also included some PBS for determining whether over-dimension vehicles can achieve the same turning performance as standard vehicles and hence whether they can be exempted from the travel time restrictions.

In 2010, there was an amendment to the VDAM Rule which provided for the introduction of High Productivity Motor Vehicles (HPMVs) (New Zealand Transport Agency 2010). This amendment provided for vehicles to be permitted to exceed the standard size and weight restrictions on the sections of the network that could accommodate them. To facilitate the uptake of HPMVs, the NZTA in conjunction with the transport industry promoted the concept of pro-forma designs for larger and heavier vehicles that could fit on the existing network. The pro-forma designs were developed using PBS and essentially are larger vehicles which have key dimensions constrained within a range which ensures that satisfactory performance is achieved. The critical aspect of performance for these longer vehicles was expected to be low speed turning and the NZTA specified a new performance standard based on a 120° turn to control this. In addition to the low speed turning requirement, the rollover stability requirements were made more stringent and a full set of performance measures based on the Australian PBS system was evaluated to ensure that there were no unexpected adverse performance impacts. HPMVs have also been permitted on a one-off basis where they have been individually assessed to ensure that they have comparable or better performance than the pro-forma designs.

PBS INTERNATIONALLY

Overview

The first stage of the project is to review the performance measures that are used in other jurisdictions and have been used in New Zealand. For each of these measures we will identify the threshold levels that have been used and how they have been derived.

As noted in the previous section the main purpose of performance measures in the RTAC study was to inform the development of a set of harmonised vehicle size and weight limits that would be acceptable to all the Canadian provinces. An initial memorandum of understanding (MOU) between the provinces and the federal government was signed in 1988 and this has been extended and amended a number of times since then. The vehicles included in the MOU are defined by a range of prescriptive limits forming vehicle envelopes where there is some flexibility in axle spacing, weight and hitch position. This provides industry with some design flexibility that keeps vehicle performance within acceptable limits and reduces administrative and compliance burden. In addition to this, some of the Canadian provinces have used the performance measures developed in the RTAC study as a basis for evaluating high productivity vehicles which they have then allowed to operate under permit within the province.

Internationally, Australia has championed the use of PBS as an optional alternative to prescriptive size and weight limits. This is a much more formalised PBS structure than we have in New Zealand. A comprehensive set of studies was undertaken between 1999 and 2007 (Prem, Ramsay et al. 1999, Sweatman, Prem et al. 1999, Prem, Ramsay et al. 2001, Prem, Ramsay et al. 2001, Prem, Ramsay et al. 2001, Prem, de Pont et al. 2002, Pearson, Prem et al. 2006) to develop and gain acceptance for the PBS framework. The system was finally implemented in 2008. Because of the rigor of the system, the process of undertaking an assessment and gaining a PBS approval is quite costly and cumbersome. Compared to HPMVs in New Zealand the uptake of PBS in Australia was initially very slow which caused concern for the regulators. One of the ways that this is being addressed is through the development of "blueprint" designs which are comparable to the pro-forma designs in New Zealand. It is worth noting that, although the Australian PBS system was developed with a view to being a complete alternative to prescriptive size and weight limits, a number of key dimensions such as height, width and length are effectively prescribed as are axle group weights and a bridge formula.

South Africa has followed Australia's lead and is trialling a PBS system with a number of vehicles (Nordengen, Prem et al. 2008, Thorogood, Bright et al. 2009, Nordengen 2010, Nordengen 2012). Rather than repeat all of the work that went into developing the Australian PBS system, South Africa has largely adopted it as is with some minor adjustments to local conditions.

Several European countries are running long combination high productivity vehicles using the European Modular System (EMS). These combinations are not based on any performance assessment and, in fact, some of the configurations that are permitted by the EMS have potentially relatively poor performance characteristics. There is some resistance to these vehicles in many European countries and at this stage they are only operating in Scandinavia, the Netherlands and Belgium.

Sweden was a leader in the introduction of the EMS-based long combination vehicles and just over a year ago commenced a major research programme (Kharrazi, Karlsson et al. 2015) to investigate the use of PBS for developing high productivity vehicles in Sweden. This research programme involves a consortium of researchers from the truck and trailer manufacturers, several universities and government agencies including the transport research institute. The programme includes computer modelling and vehicle testing and is scheduled for completion in 2016.

Reviewing the way that PBS has been used in these various jurisdictions, we can identify a number of key issues that need to be addressed.

The first is that although it might appear that PBS is a unique well-defined set of standards this is not the case. Each standard is designed to characterise a particular aspect of vehicle performance that is

considered important but different jurisdictions use different performance measures to characterise the same aspect of vehicle behaviour. While the performance measures appear to be the same, there can still be differences in the detail of both the test conditions and the performance measures that are assessed. Even when the performance measures are the same, there can still be differences in the acceptability thresholds and so the performance standard is different.

A second issue is the completeness of the PBS system. The Australian PBS system was intended to be complete in covering all critical aspects of safety and infrastructure performance. In practice there are still some limitations because it was found to be impossible to define acceptable performance standards for some factors, for example, ride quality and handling. Furthermore, although the existing set of standards appears to work reasonably satisfactorily for typical current vehicle configurations, there is the possibility that someone will invent an innovative vehicle that achieves all the existing performance standards but has some undesirable performance characteristics that are not covered by the current standards.

A third issue is how the pass/fail thresholds are set. In the current Australian system, each standard has a pass/fail threshold and it is necessary to achieve satisfactory performance for all of the standards. Thus each standard is equally important in determining whether or not a vehicle is acceptable. Furthermore, it implies that a vehicle that achieves a pass for a safety-related standard is safe for that aspect of vehicle performance and conversely, a vehicle that fails that standard is unsafe in that regard. In reality the criteria are not that black and white. For each of the safety-related performance measures, a worse score corresponds to an increased safety risk but there is no step change in risk to provide us with a natural pass/fail threshold. The difference in safety risk between a vehicle that narrowly passes and a vehicle that narrowly fails is usually very small. Note that this is also true for prescriptive limits. Furthermore, for many of the performance measures, the precise relationship between performance and safety is not known with any degree of certainty. As a consequence, the pass/fail thresholds for the different performance measures do not necessarily represent the same level of safety risk. This is important because for any vehicle configuration there are trade-offs between different aspects of performance, e.g. low speed manoeuvrability and high speed dynamics. The pass/fail criteria that are set for the relevant performance standards will be a significant factor in determining where the balance point between these aspects of performance lies. In the past we have had examples where operators have successfully argued a case for being given a concession on overall length in order to achieve improved rollover stability. This raises the question about whether it is possible to configure a PBS system so that these trade-offs can be taken into account.

Most of the current PBS thresholds have been set on the basis of the performance levels that are achievable by existing vehicle configurations that are considered to have acceptable safety performance. This is not unreasonable as the current vehicle configurations have evolved over time based on the experience of operators, regulators and policymakers. By a process of natural selection, these vehicles will tend towards the configurations that are best suited to the transport task being undertaken within the constraints of the operating and regulatory environment. Note, however, that "best suited" does not necessarily mean safest. Operators will be trying to maximise profitability. Safety is important to them because crashes are costly and because they do not want to put their staff at risk but there can be a trade-off between safety and productivity.

As noted above, performance analysis has been used to guide the development of the current VDAM Rule and has underpinned the development of the pro-forma designs for HPMVs and the permitting of other HPMVs that do not fit the pro-forma designs. The VDAM Rule was reviewed in 2016 and the 2016 version of the Rule came into force on 1 February 2017.

Current HPMV designs including the pro-forma designs have been developed primarily on the basis of a low speed turning performance criterion. The performance standard that is used was developed by NZTA and based on the performance characteristics of the worst-case standard legal vehicle which is the 19m quad-axle semitrailer. Designs that achieved this performance standard were then tested against a set of PBS standards based on the level 1 requirements of the Australian PBS system. Generally these designs have no difficulty at all achieving these Australian PBS requirements. Video trials on some of the longer vehicles that have been permitted have shown that at some "pinch points" on the network these vehicles occupy the full lane width. Clearly if two of these vehicles were to meet at one of these corners this would

cause difficulties. These "pinch points" are generally low speed curves with advisory speeds of 25 or 35 km/h in difficult terrain where it would be difficult to widen the road. The fact that these vehicles occupy the full lane width does not necessarily indicate the PBS approach used for designing them was flawed. It is quite likely that 19m semitrailer combinations would similarly occupy the full lane width on these corners. Nevertheless it does represent a safety risk. With some exceptions quad-axle semi-trailers are not extensively used for linehaul operations and so the number of these passing through the "pinch points" is limited. Truck-trailer and B-train HPMVs, however, are very popular as linehaul vehicles and so are more likely to meet each other at a "pinch point".

What this highlights is the need to adapt the PBS regime to New Zealand conditions. The performance measures and thresholds used in Australia are not necessarily appropriate for New Zealand conditions. The purpose of this project is to review the available set of performance standards and to determine a set of standards that are appropriate for New Zealand. These standards would include those used in the Rule to determine whether an over-dimension vehicle has adequate turning performance for exemption from the travel time restrictions. It is conceivable that the existing pro-forma designs will not meet the proposed set of performance standards. If this is the case it will be necessary to revise these designs.

Canadian Performance Standards

Introduction

The RTAC study (RTAC 1986) specified nine performance measures to be evaluated for characterising heavy vehicle performance. These were:

- Payload Volume
- Payload Weight
- Static Rollover Threshold
- High-Speed Offtracking in a Steady State Turn
- Amplification-Induced Rollover[1]
- Transient High-Speed Offtracking
- Low-Speed Offtracking
- Friction Demand in a Tight Turn
- Braking Efficiency

They also considered including a handling performance measure, which was "Tractor Understeer Coefficient at 0.25g Lateral Acceleration", but decided that the link between this measure and safety performance had not been demonstrated and therefore its inclusion would be premature.

For each of the nine performance measures they defined a reference value. The performance of each of the vehicle configurations that they analysed was then presented on a normalised scale as a percentage better or worse than the reference value. Although the reference values were not strictly a pass/fail criteria, they were used as a benchmark for acceptable performance. We will now review each of these performance measures and describe how they are evaluated and how the reference value was determined.

Payload Volume

This is a measure of the productivity of the vehicle for volume-constrained freight operations. It was determined by simple geometric calculations and the reference value was determined from the "standard"

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[1] This measure evolved into two separate measures both evaluated in the same manoeuvre. These two performance measures are rearward amplification and load transfer ratio and this is the standard terminology that has been used since.

Canadian 14.6m semi-trailer. This freight "box" is 14.6m long by 2.59m wide by 2.74m high. The height dimension assumes that the floor height is 1.37m and the overall height is 4.11m. Clearly, for volume-constrained loads, vehicle configurations with a greater payload volume will be able to undertake the same freight task with proportionately fewer trips. This implies a safety benefit from reduced exposure.

Payload Weight

This measure is the equivalent of the previous one for weight-constrained freight operations. Again the reference value was based on the baseline 5-axle semitrailer combination and was set at 25 tonnes. This was based on a gross combination weight limit of 39.5 tonnes and an assumed tare weight of 14.5 tonnes. The purpose of this measure was to quantify the safety benefit from reduced exposure for weight-constrained loads.

Static Rollover Threshold

The RTAC study states that the preferred test method for Static Rollover Threshold (SRT) is a tilt-table test and it sponsored the construction of a 25-metre tilt table for research and compliance purposes. However, the study then highlighted a number of factors that can potentially influence the test results and may affect the fidelity of the tilt-table results as a predictor of the in-service performance.

The RTAC study also determined SRT by computer simulation and this was the main method used to investigate the influence of different vehicle parameters on performance. The computer models of the vehicles were "driven" through a quasi-steady turn manoeuvre at a constant speed of 100km/h. During this manoeuvre the front wheel steer is slowly increased at a steady rate of 0.04 degrees/second. Thus the vehicle drives through a slow spiral and the lateral acceleration gradually increases until wheel lift-off occurs.

The reference level for Static Rollover Threshold was set somewhat arbitrarily at 0.4g. The baseline vehicle in this study was a standard Canadian 5-axle semi-trailer combination which was loaded with homogeneous freight with a density of 545kg/m^3, which results in a centre-of-gravity height for the payload of 2.03m. Note also that the width of this "standard" vehicle is 2.59m. In this configuration the vehicle has a Static Rollover Threshold of 0.437g. The baseline vehicle was also modelled with a high load where the centre-of-gravity height was 2.67m. In this configuration the Static Rollover Threshold was 0.315g.

High-Speed Offtracking in a Steady State Turn

This measure was defined as the amount of outboard offtracking of the last axle in the vehicle combination during a high speed turn where the lateral acceleration is 0.2g. The test manoeuvre was a 393m radius turn which is traversed at 100km/h. The offtracking is measured as the offset between the path of the tractor (or truck) steer axle and the path of the last axle in the combination.

The 0.2g lateral acceleration value was chosen as a reasonably high practical level which can reliably be achieved and steadily maintained without the possibility of oscillatory behaviour or actual rollover if the vehicle has poor rollover stability. The reference value used was 0.46m. This value was arbitrarily selected on the basis of a 2.44m wide tractor following a path down the centreline of a 3.66m lane and still having 0.15m of clearance to the outside of the lane. It is interesting that justification for the reference value is based on a 2.44m wide vehicle when the baseline vehicle used in the study was 2.59m wide. On the other hand, the justification is also based on the tractor following a path down the centreline of the lane, which effectively assumes that the driver does not take the offtracking into account when steering through a curve. We would expect that an experienced driver would be aware of the offtracking behaviour of the vehicle and would position the tractor to the inboard side of the lane when going through a curve so that the vehicle is centred in the lane as much as possible throughout the turn.

Amplification-Induced Rollover

For this aspect of performance, the RTAC study considers two measures; Rearward Amplification (RA) and Load Transfer Ratio (LTR). The manoeuvre used for determining this measure is a high speed lane change.

The report (RTAC 1986) notes that RA is the classical measure used for this performance characteristic and gives the definition of RA as the ratio of the peak value of lateral acceleration of the mass centre of the rearmost trailer to the peak value of lateral acceleration of the mass centre of the tractor. They state that this definition of RA has some shortcomings when the lateral acceleration response of the tractor is rather asymmetric. The standard lane change produces at symmetric lateral acceleration response at the steer axle which follows the prescribed path. If the centre of mass of the tractor is a significant distance back from the steer axle, the lateral acceleration response of the tractor can be somewhat different to that of the steer axle. Thus this issue of an asymmetric lateral acceleration response by the tractor tends to be more significant for long wheelbase tractors and for rigid truck and full trailer combinations. A suggested solution to this issue is to use a modified lane change path so that the resulting tractor lateral acceleration is a symmetric sine wave.

The discussion in the RTAC report also notes that RA as defined above fails to account for the benefits of roll-coupled hitching arrangements as in B-trains and C-trains. This is the reason that they developed LTR measure. LTR is the absolute value of the difference in the total load on the right side wheels and total load on the left hand side wheels divided by the total load on all wheels. The calculation is applied to a "rolling" unit and thus roll-coupled units are considered as a single entity. Where the "rolling" unit includes the tractor, the steer axle loads are not included in the calculation. Clearly when the LTR is unity, all of the load has transferred from one side of the "rolling" unit to the other and the unit is on the point of rollover. The reference value for LTR used by RTAC is 0.6. No reference value is provided for RA in this report.

The RTAC study investigated various options for the period and speed of the lane change manoeuvre and settled on a test speed of 100km/h with a period of 3.0 sec and a lateral acceleration amplitude of 0.15g. They suggested that if the lateral acceleration response of the tractor was excessively asymmetric (which they suggest is a difference of more than 10-15% between the positive and negative peaks), additional steps would be needed to obtain a more symmetric response.

Transient High-Speed Offtracking

This performance measure was developed by the RTAC study. It was defined as the peak value of the offset, normal to the path, between the path of the outside front tyre on the tractor and the path of the most outboard trailing axle.

For the RTAC study, the high speed lane change manoeuvre specified for the RA and LTR was used for this measure as well. The report does discuss whether some other manoeuvre might be more appropriate for evaluating this aspect of performance. For this performance measure based on the high-speed lane change manoeuvre a reference value of 0.8m was used. This was selected arbitrarily as the nominal mid-range value for the vehicles examined in the study.

Low-Speed Offtracking

This measure quantifies the amount of road space the vehicle requires when executing a small radius low speed turn. There are some minor inconsistencies in how the manoeuvre and the performance measure are described in the RTAC report. The turning manoeuvre used is a 90 degree turn executed at 8.25km/h. The definition of the manoeuvre specified an 11m radius at the outside front steer tyre but the computer simulations used a 9.8m radius turn at the centre of the steer axle. With a 2.44m wide tractor these two manoeuvres are approximately, but not exactly, the same.

Low-Speed Offtracking (LSO) was defined as "the maximum extent of lateral excursion of the last trailer axle relative to a circular arc subtended by the tractor front axle". However, elsewhere in the report it is defined as "the peak offset in wheel paths measured from the outside of the outer front tyre to the inside

of the innermost-trailing trailer tyre." The first definition implies that the measure is the offset between the axle paths measured at the axle centres while the second definition is a swept width which includes the width of the vehicle. Thus the first measure provides an indication of the additional road width required while the second measure indicates the total road width required. The reference value given is 6m and the baseline tractor-semitrailer combination is shown as having an LSO of 5.906m. We have built and run a model of the RTAC baseline tractor semitrailer and found that the LSO measured as the offset between the paths of the centre of the front and rear axles was 5.894m. Thus the 6m reference value corresponds to this offset and not the swept width.

In the discussion on LSO, the RTAC report also considered the swing out of the rear of the trailer at the start of the turn. This is the measure we call tail swing. This was not investigated further but they suggested that a value of 0.3m might be appropriate as a maximum acceptable level.

Friction Demand in a Tight Turn

This measure evaluates the net friction demand on the tractor drive axles during the same 90 degree turn that was used for LSO. The friction demand is a measure of the force that the tractor has to apply to the kingpin to pull the trailer around the turn. The magnitude of this force increases when the axle spread on the trailer is large but it usually only becomes an issue when the tyre/road friction is low due to ice or snow. In these circumstances excessive friction demand can lead to jack-knifing during a low speed turn. The reference value used for this performance measure was 0.1. This is rather low but recognises that the measure does not take into account all the sources of friction demand on the tyres (such as the tractive forces) and thus it allows some leeway for these additional forces.

The RTAC report notes that in snow conditions the friction level for car tyres is in the range of 0.2 to 0.4 but that there is no data for truck tyres. This performance measure would appear to have limited applicability to New Zealand because generally we don't operate vehicles in snow and we don't use widely-spaced trailer axles.

The reference value of 0.1 for this measure is remarkably low. There is a margin of safety included but it appears to imply tyre/road friction values of around 0.2. In this situation, a vehicle would not be able to achieve more than 0.2g deceleration when braking and would slide out in any corner when its lateral acceleration exceeds 0.2g. Driving in these conditions is inherently quite dangerous and should be avoided.

Braking Efficiency

Braking Efficiency was defined and the percentage of the available tyre/road friction limit that can be utilised in achieving an emergency stop without incurring wheel lockup. In the RTAC study this measure was determined by dividing the vehicle deceleration in g's by the highest friction demand at any axle. This measure was evaluated at two deceleration levels, 0.1g and 0.4g. 0.1g represents a typical in-service braking level while 0.4g represents the hard braking associated with an emergency stop.

A reference level of 70% was used for comparing vehicles. Quite low levels of brake efficiency were reported for some of the partially loaded vehicle configurations. It appears that the brake systems that were modelled did not have load proportioning or ABS or EBS. Consequently, under-loaded axles were over-braked. Vehicles complying with the requirements of the New Zealand Brake Rule should automatically have satisfactory performance in this regard.

Validation

In the RTAC study the performance measures were evaluated by calculation either directly or by computer simulation. For most of the measures, the Yaw-Roll multi-body simulation software developed by the University of Michigan Transportation Research Institute (UMTRI) was used. In addition to the simulations a series of experimental tests were conducted and a comparative analysis using different software packages was undertaken.

The experimental programme was quite extensive. Nine different vehicle configurations were tested. Each vehicle was subjected to ten tests but only three of these were simulated. These were:

- sinusoidal steer,
- lane change,
- steady circular turn.

Thus the purpose of the tests were not simply to verify the simulation but also to provide additional data on the vehicle performance characteristics

The authors state that this work was not a validation of the computer simulation method because they did not measure the properties of the actual vehicles and they did not try to adjust the component data to achieve a better match to individual runs. Rather the objective was to achieve reasonable agreement between test and simulation results using generic component data for individual runs and for the trend over a number of runs. They further state that this objective was achieved.

All the vehicle configurations that were tested used the same tractor which was extensively instrumented. The trailers were fitted with outrigger wheels to prevent rollover and instrumented to measure articulation angle, lateral acceleration, roll angle, outrigger touchdown and brake chamber pressures. Although the instrumentation was quite extensive, it was not comprehensive enough to be able to evaluate all of the performance measures. Thus most of the comparisons between the actual vehicles and the simulated vehicles were based on the vehicle responses measured by the instrumentation rather than the derived performance measures.

Australian Performance Standards

Introduction

The Australian PBS system was developed with the intention of being an alternative compliance regime for size and weight. Because of this there was a recognition from the outset that the system had to be complete and cover all of the critical aspects of vehicle performance. Like Canada, Australia had issues arising from the jurisdictional boundaries between the states and the federation. The states own and operate the roading network within the state and consequently reserved the right to regulate size and weight within state boundaries.

The National Road Transport Commission (NRTC) was established in 1991 as an inter-state body to develop regulatory and operational reform to the road transport system for improved productivity and efficiency. In 2003, its role was expanded to also cover rail transport and intermodal operations and it was renamed the National Transport Commission (NTC). One of its more significant initiatives was the development and implementation of a PBS system for regulating vehicle size and weight.

The process of developing the PBS system commenced with a project identifying a set of as many performance measures as possible (Sweatman, Ramsay et al. 1999) for potential inclusion in the system. This resulted in over 100 possible measures. A process of filtering was applied based on seven regulatory principles (Prem, Ramsay et al. 2001) and the number of performance measures was reduced to 26 which are listed in Table 1. In addition, this initial study identified 17 performance standards that were already required by existing standards and regulations. These are shown in Table 2. These additional 17 performance standards were not repeated in the initial PBS set.

These proposed performance measures were presented to stakeholders in a series of workshops. They were then further refined as the research project proceeded. Measures that were found to be strongly correlated to other measures were eliminated as being redundant. For some measures it was difficult to find a consensus approach to evaluation and to pass/fail thresholds. Some of these were eliminated and some were retained but implementation has been deferred. The final set of performance measures that make up the Australian PBS system is shown in Table 3. It is notable that load transfer ratio and high-speed

steady-state offtracking are not included in the final list. These were both significant measures in the Canadian study.

Twenty performance measures are listed but only 17 of these are currently active and, of those, compliance with the four infrastructure standards is determined through prescriptive limits and a vehicle can be deemed to comply with the directional stability under braking requirement if its braking system meets the requirement of Australian Design Rules (ADR) 35 and 38. Thus an Australian PBS assessment typically involves evaluating twelve performance measures to check that they comply with the standards.

Table 1. Initial reduced set of performance measures for Australia.

SAFETY RELATED	INFRASTRUCTURE RELATED
Static roll stability	Payload mass per ESA
Rearward amplification	Horizontal tyre forces
Load transfer ratio	Tyre contact pressure distribution
High-speed transient offtracking	Upper bound on axle/axle-group load
High-speed steady-state offtracking	Upper bound on GVM/GCM
Yaw damping	Axle spacing mass schedule
Tracking ability on a straight path	Critical design vehicle (bridges)
Braking stability (in a straight line)	
Braking stability (in a turn)	
Handling quality (understeer/oversteer)	
Low-speed offtracking	
Frontal swing	
Tail swing	
Friction demand (steer tyres in corner)	
Ride quality	
Startability	
Gradeability	
Intersection clearance time	
Overtaking time	

Prior to the introduction of PBS, the Australian road network had four levels of access with certain vehicle configurations being restricted to limited parts of the network. These four levels are:

- general access – virtually the entire network, vehicles up to 19m long.
- B-double routes – major intercity arterials and feeder links, 26m B-doubles[2]
- Type 1 road train routes – remote areas but up to cities in some states, 36.5m A- doubles[1]
- Type 2 road train routes – remote areas, 53.5m A-triples.

The PBS system was aligned to this concept of differential access and thus there were four levels of PBS vehicle assessment. The same performance measures are used for all levels but the pass/fail criteria differ

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[2] In New Zealand we call these vehicles A-trains and B-trains and they have no more than two trailers. In Australia, they can have more than two trailers and so they distinguish the two trailer combination from the three trailer combination by calling them A or B doubles and triples.

for different levels. Table 3 shows the pass/fail criteria for level 1 which is general access. These are the most relevant to New Zealand as we do not have an established hierarchy of roads in our network. With the introduction of HPMVs we are identifying routes that are capable of carrying higher weights but, to date, we have not considered having allowing different levels of safety performance on different categories of route.

Table 2. Performance standards included in existing regulations and standards.

#	POTENTIAL PERFORMANCE MEASURE	DESCRIPTION	EXISTING STANDARD OR REGULATION
1	Coupling strength	Fatigue strength of coupling under multi axis dynamic loading	ADR62
2	Braking - thermal capacity	Tendency of braking systems to lose effectiveness (fade) under prolonged use	ADR35/38
3	Operating Speed	The maximum speed that the vehicle is capable of operating at, through gear or road speed limiting	ADR/AVSR
4	Turning circle (wall-to-wall)	Ability to turn in a circle of a defined diameter measured from the edge of the tyre track at ground level	ADR/AVSR
5	Conspicuity	Degree to which the vehicle and its extremities are perceived visually by other road users	ADR/AVSR
6	Noise (external)		ADR28
7	Emissions	Gaseous by-products of the vehicle engine's combustion process	ADR70
8	Braking efficiency	Tendency of particular axle(s) to lock up prematurely under severe braking	Air Brake Code of Practice
9	Interchangeability of trailers	Degree to which trailer and coupling designs allow these vehicles to be interchanged	AS1771
10	Load restraint	Ability of the vehicle and its restraint systems to keep the load on the vehicle under severe manoeuvres and disturbances	M&L/LRG
11	Load shift - (compliance)	Ability of the vehicle and its restraint systems to prevent the load shifting on the vehicle under severe manoeuvres and disturbances	M&L/LRG
12	Noise (internal)	Sound Pressure Level (SPL) generated by a vehicle; peak levels outside the vehicle and levels that driver is exposed to need to be considered	OH&S (?)
13	Gradeability	Sustainable speed travelling up a defined grade	Only in some States for some vehicles
14	Dynamic wheel loads	Degree to which vehicle suspension is road-friendly in terms of dynamic loading	VSB11
15	Load sharing coefficient	Degree to which vehicle suspension is road-friendly in terms of load-sharing between axles	VSB11
16	Braking - stopping distance (entire vehicle)	Distance required to stop from a given speed	AVSR
17	Tyre inflation pressure	Inflation pressure required for the tyre size and load	AVSR

Table 3. Final set of performance standards in Australian PBS system.

PERFORMANCE STANDARD	Level 1 Requirements
Safety Standards	
1. Startability	≥ 15%
2. Gradeability:	
a. Maximum grade	≥ 20%
b. Speed on a 1% grade	≥ 80km/h
3. Acceleration capability	Time to travel 100m from rest ≤ 20s
4. Overtaking Provision	[reserved] Vehicle length ≤ 20m
5. Tracking Ability on a Straight Path	≤ 2.9m
6. Ride Quality (Driver Comfort)	yet to be defined
7. Low-Speed Swept Path	≤ 7.4m
8. Frontal Swing:	
a. Maximum Frontal Swing	≤ 0.7m for trucks, 1.5m for buses
b. Maximum of Difference	≤ 0.40m
c. Difference of Maxima	≤ 0.20m
9. Tail Swing	≤ 0.30m
10. Steer-Tyre Friction Demand	≤ 80% of maximum available
11. Static Rollover Threshold (Worst)	≥ 0.35g (0.40g for buses and dangerous goods)
Static Rollover Threshold of last unit	≥ 0.35g (0.40g for buses and dangerous goods)
12. Rearward Amplification	≤ 5.7 times static rollover threshold
13. High-Speed Transient Offtracking	≤ 0.6m
14. Yaw Damping Coefficient	≥ 0.15
15. Handling Quality (Understeer/Oversteer)	yet to be defined
16. Directional stability under braking	Deemed to comply if brakes meet ADR35 and 38
Infrastructure Standards	
17. Pavement Vertical Loading	Axle group weights meet current limits
18. Pavement Horizontal Loading	Max spread for non-steerable axles in groups
19. Tyre Contact Pressure Distribution	Current limits for min. width and max. pressure
20. Bridge Loading	Mass (M) and Length (L) formula; M = 3L + 12.5 for M <= 42.5 t; M = L + 32.5 for M >= 42.5 t

Aligning the PBS levels with the existing road classes has not been totally successful in Australia and one of the main barriers to a greater uptake of PBS has been the difficulties in getting access to the network (Arredondo 2012). This is currently being addressed.

Swedish PBS

In 2013 Sweden initiated a major study to investigate the applicability of PBS in Sweden. The aim of the project is to propose a PBS scheme for regulating "High Capacity Transports (HCT)" and their access to the road network. This a very large multi-agency study which will take several years to complete. The first significant report on this work was published recently (Kharrazi, Karlsson et al. 2015). This presents a review of current vehicle regulations in Sweden and the PBS approaches used in other countries. This study is very useful because in a number of key areas it addresses the same issues as this current work and thus it provides a valuable cross-check on our findings.

In addition to the review of the safety and infrastructure related performance measures used around the world, the Swedish study also looks at performance measures related to environmental impact. The three key components considered were exhaust emissions, fuel consumption and noise. Exhaust emissions and noise of heavy vehicles are currently regulated in Europe and elsewhere and fuel consumption regulations are being developed as a mechanism to reduce greenhouse gas emissions. The question raised in the report was whether these regulations are adequate for HCTs. This question was not answered although it was noted that noise levels may be different for HCTs because of differences in the engine load, and the numbers of trailers, axles and tyres.

This study also reviews the safety performance of HCTs. The only research relating performance measures to crash rates that they identify is the work that TERNZ undertook in New Zealand (Mueller, de Pont et al. 1999) and in Tasmania (de Pont 2005). They also analysed at the in-service safety performance of HCTs in different countries where they have been operating, namely various European countries, Canada, USA, Mexico, Australia and South Africa. For most of the European countries the data was based on trials involving relatively small numbers of vehicles which in turn had relatively few crashes. Although this data was generally favourable, it was not statistically significant. The exception to this was Sweden where 10 years of data for long combination vehicle operations existed. The Swedish data showed that the rate of fatal or serious crashes reduced as vehicle length increased. However, there was no data on the roads used by the longer vehicles and thus there is always the possibility that the longer vehicles were typically operating on safer roads. The Canadian data was based on HCT vehicle operations in the province of Alberta. These were found to have the best safety performance of all vehicle types. These vehicles are subject to quite strict permit conditions which limit the routes they can use, restrict the time of day for their operations, restrict their speed, require enhanced driver qualifications and limit operations when adverse road or weather conditions exist. Thus there are many factors that could contribute to improved safety performance. The US data was found to be problematic in that there was no reliable data of vehicle configurations involved in crashes. One study (Craft 2000) is quoted as finding that HCT vehicles were not significantly more or less safe than other heavy vehicle combinations. In Mexico, large A-trains are permitted to operate. Apart from a warning sign on the rear, the vehicles are no more restricted than conventional tractor-semitrailer combinations. A recent study (ANTP 2014) was reported showing that the two-trailer combinations had lower crash rates than single trailer combinations and rigid trucks. In Australia a comprehensive study was undertaken using claims data from the largest truck insurer (Hassall 2014). This found that high productivity vehicles had a crash rate for serious and major accidents that was 76% lower than that of conventional vehicles. This was based on the rate per 100M veh-kms and so these safety benefits are substantially greater when the productivity gains are taken into account. It is not possible to link these safety benefits directly to the superior performance characteristics of the high productivity vehicles.

PBS in New Zealand

As noted in the introduction, New Zealand has been an early adopter of PBS principles for regulating size and weight. The first uses were to inform the development of prescriptive regulations for size and weight (White 1989). For these applications, the Canadian RTAC performance measures were used but some of

the reference values were modified. For example, the reference static rollover threshold value used in the RTAC study was 0.4g while the New Zealand studies used 0.35g. It is noteworthy that Canada does not enforce the 0.4g target because initially it was recognised that it would be difficult to do.

Subsequently performance standards were used as a basis for permitting some vehicle configurations to exceed their standard weight limits, most notably 44-tonne A-train milk tankers (White 1990). For these vehicles, the critical aspects of performance were identified and performance standards were specified for these critical measures. In the case of the A-trains, the RTAC study showed that the main performance problems with this configuration were load transfer ratio high speed transient off-tracking. Both these measures use the high speed lane change manoeuvre. Locally there was some concern about rollover stability. Thus the three standards specified were that the static rollover threshold (SRT) should be greater than 0.45g, the load transfer ratio (LTR) should be less than 0.6 and the high speed transient off-tracking (HSTO) should be less than 0.5m. The SRT and HSTO limit values represent better performance than the RTAC reference values so these requirements promoted good performance rather than just adequate performance.

The UMTRI and RTAC performance measures were also used as a basis for permitting some alternative vehicle configurations, for example, the use of tridem drive trucks for log transport (White 1994). This analysis used the following performance measures with the target values shown in brackets:

- Static Roll Threshold (≥ 0.35g)
- High-Speed Steady-State Offtracking (≤ 0.6m)
- Dynamic Load Transfer Ratio (≤ 0.6)
- High-Speed Transient Offtracking (≤ 0.6m)
- High-Speed Friction Demand (≤ 0.1)
- High-Speed Lateral Friction Utilisation (≤ 0.8)
- Low-Speed Offtracking (≤ 4.2m)
- Low-Speed Friction Demand (≤ 0.1)
- Low-Speed Lateral Friction Utilisation (≤ 0.8)
- Yaw Damping Ratio (≥ 0.05)
- High-Speed Handling Stability (3-point method developed by (El-Gindy, Woodrooffe et al. 1991)

The last two of these performance measures were not used in the RTAC study. Of the others, four of them have target values that are different from the reference values used in RTAC. Two of them (static rollover threshold and high speed steady-state offtracking) have less demanding target values than RTAC while the other two (high-speed transient offtracking and low-speed offtracking) have more demanding target values. We have not been able to find any documentation describing how these target values were set. From comments in the reports it appears that they are loosely based on the performance characteristics of existing vehicle configurations that are considered acceptable.

In 2002, the size and weight regulations were extensively reviewed and reformulated as the Vehicle Dimensions and Mass Rule (VDAM) 41001. A number of performance analyses using the UMTRI/RTAC performance measures were undertaken to inform the development of the rule (Mueller and Baas 2001). Several of the Rule's requirements (e.g. the truck-trailer mass ratio, and the maximum hitch offset) were a direct result of these analyses. In addition the Rule incorporated some PBS requirements. The most notable of these was the minimum static rollover threshold requirement of 0.35g applying to all large trucks and trailers. In 2016 the VDAM Rule was reviewed and the new version of Rule came into effect on 1 February 2017. The PBS aspects of the previous version of the Rule were retained but with some modifications. The section numbers in the following discussion refers to those in the current 2016 version of the Rule.

There are a number of low speed turning performance standards specified in sections 6.28, 6.33 and 6.34 of the Rule. Section 6.28 and 6.33 specify the same set of low speed turning performance requirements which are based on the performance characteristics of a maximum-sized standard vehicle. Although the requirements are described as "low speed" turning performance neither the turn radius nor the maximum speed at which the assessment should be done is specified explicitly. Both will have a significant effect on

the vehicle's performance. However, the Rule does say that the assessment must be done using a method that is approved by the Agency and published on the Agency's website and so the test parameters can be specified here and can be modified if necessary without amending the Rule. Section 6.34 specifies a 50m radius wall-to-wall turn of 90° and specifies that the swept width of the vehicle should not exceed 4.7m. This requirement does specify a maximum test speed of 5km/h

In 2010, the VDAM Rule was amended to allow for the introduction of high productivity motor vehicles (HPMVs). HPMVs operate under permit and are allowed to exceed the maximum size and weight requirements for standard vehicles on approved routes that are capable of handling them. To facilitate the uptake of HPMVs the NZTA promoted the development of pro-forma designs (de Pont 2010). These designs were based on a template with dimensional envelopes that were constrained to ensure adequate performance. As these vehicles were longer than the standard vehicles, the main issue was expected to be low speed turning performance. The worst case standard vehicle for low speed turning performance is the 19m quad-axle semi-trailer combination and thus its performance was used as the basis for acceptability. The standard low speed turning manoeuvre used in various PBS analyses is the 90° turn. However, the NZTA were concerned about the case of a vehicle making a right turn at roundabout where the turn angle is greater than 90° and thus they proposed a performance measure based on a 120° degree wall-to-wall turn with 12.5m outside radius. From the performance of a reference 19m quad-axle semi-trailer they determined that, for acceptable performance, no part of the vehicle should encroach on a concentric inner circle of 4.9m radius. Although this was the primary performance standard, a full suite of performance measures based largely on the Australian PBS system was evaluated to ensure that the vehicle's overall performance was satisfactory. Because these vehicles were generic designs they were modelled with standardised suspensions with relatively poor performance. The truck and tractor suspensions were based on Kenworth walking beam suspensions while the trailers were based on Reyco leaf spring suspensions. Most new vehicles use air suspensions and thus it was expected that, in practice, vehicles built to these designs would usually have superior performance to that predicted by the PBS assessment.

The standardised suspension data is based on rather dated suspensions. It has been suggested that, going forward, data from more current suspensions should be used for the generic designs. These reference suspensions should still be at the lower end of the performance scale compared to what is currently available to be conservative but should represent current practice.

At standard weights, vehicles permitted under this pro-forma design approach were given general access. They could operate at higher weights on specified routes as provided for in their permit. The routes available for higher weight operations were somewhat fragmented as local road controlling authorities had concerns about the impact on their infrastructure. Thus for many operators there were difficulties in utilising the provisions for higher weights cost-effectively. To overcome this the NZTA developed the concept of the 50Max designs. By requiring the vehicles to have at least nine axles and a first-to-last axle spacing of 20m or more, they could achieve gross combination weights of 50 tonnes without increasing pavement wear and also being able to use nearly all bridges without significant strengthening work. This approach has been accepted by most of the local road controlling authorities and so 50Max vehicles have access to most of the network.

A PBS SET FOR NEW ZEALAND

Introduction

Australia has led the way in developing a compliance regime based on PBS and thus it is logical for a New Zealand PBS system to be as compatible as possible with Australia. However, the transport environment in New Zealand is different to that in most parts of Australia and thus it is quite likely that the pass/fail criteria in New Zealand will not necessarily be identical to those used in Australia. Furthermore, in pioneering their PBS regime, Australia has made decisions which, in hindsight, may be less than optimal. We have the opportunity to learn from their experience.

One of the main regulatory principles in the Australian PBS system was that all performance measures should be able to be evaluated by physical testing as well as by computer simulation. This principle is carried through into the Rules for PBS assessment and for each performance measure the requirements are specified for both physical testing and computer simulation. In practice, the physical testing is rarely undertaken. I am not aware of any full PBS assessment in Australia that has been done by physical testing. Occasionally physical testing has been done to validate one or two performance measures or more simply to validate the computer simulation responses. There are two main difficulties with physical testing. Firstly, for some performance measures, the instrumentation required is extensive and would be costly to implement. Secondly, finding suitable test site venues for some of the manoeuvres could also be difficult. Some problematic performance measures such as load transfer ratio and high-speed steady-state offtracking were eliminated from the Australian PBS set partly because of these difficulties (Prem, de Pont et al. 2002). However, some of the performance measures that have been retained such as steer friction utilization and tracking ability in a straight path have similar issues. While it is desirable that the performance measures should be able to be evaluated by physical testing as well as by computer simulation, it is not essential. It is possible to validate the computer simulation process without being able to easily measure the specific performance characteristic.

There was an underlying philosophy in the development of the Australian PBS system that there should be no prescriptive limits at all and that all vehicle size and weight limits should be derived from performance standards. In practice this has not been possible. A number of size and weight limits are imposed by the limitations of the infrastructure and by traffic engineering considerations rather than vehicle performance. An obvious example is vehicle height, but vehicle width and length also fall into this category. In my view, it is better to specify these constraints prescriptively rather than to try to manage them artificially through performance requirements.

The Australian PBS system has four infrastructure standards. These are pavement vertical loading, pavement horizontal loading, tyre contact pressure distribution and bridge loading. In practice, these have been largely implemented as prescriptive limits based on the limits applying to standard legal vehicles. In the development of the PBS system, productivity-based performance measures were considered for characterising pavement wear but these were not implemented because of difficulties establishing a consensus view of their validity.

Some performance characteristics could also be converted to prescriptive limits. For example, the Australian PBS system includes requirements for startability, gradeability and acceleration capability. These drivetrain performance requirements can be largely met by specifying a minimum power-to-weight and torque-to-weight ratios for the vehicle and requiring the drive axles to carry a minimum proportion of the total weight. In theory, there is an additional requirement that the vehicle is appropriately geared for both low and high speed operations. This requirement can be specified explicitly but it is unlikely that a truck with suitable engine power would be supplied with inadequate gear ratios.

By using prescriptive requirements where appropriate we can reduce the number of performance standards that need to be assessed. This reduces the cost and increases the reliability and robustness of the PBS system. The more standards that need to be assessed, the greater the chance that different assessors will get different results.

Having identified the aspects of performance that can best be covered by prescriptive requirements the next task is to identify performance measures that address the outstanding aspects of performance. Where different performance measures have been used internationally to characterise the same aspect of performance, the process of identifying the measures determines which of these measures is best suited to New Zealand. The process then needs to determine suitable pass/fail criteria to convert the performance measures to performance standards.

Prescriptive Dimensions

The 2010 amendment to the VDAM Rule allows for larger and heavier vehicles to operate on roads that can cope with them. Potentially this could lead to hierarchical road classes similar to those in Australia albeit on a lesser scale. However, even in Australia, where there is a large difference in the maximum size of vehicles allowed on different road classes, the prescriptive maximum height and width limits are the same across all road classes.

As noted previously, the Australian PBS system was developed with four levels of vehicle performance. Three of these (levels 2, 3 and 4) have been subdivided into two access classes (National Transport Commission 2007) which are based on the network classification guidelines. For each of these levels and road access classes, a maximum vehicle length is specified. General access vehicles are limited to 20m overall length while longer vehicles are allowed on restricted parts of the network.

These prescriptive height, width and length limits are based on the capacity of the infrastructure and traffic engineering considerations and not on the performance of the vehicle. Vehicle performance requirements may impose additional restrictions on height and/or length, but this not necessarily so. Conversely, satisfactory vehicle performance may be achievable at greater height, length or width.

A recent Austroads study (Elischer, Eady et al. 2012) undertook a review of the Australian PBS level 1 (general access) length limit. This study considered the safety and productivity implications of increasing the allowable length and found that a maximum length of up to 23m could be considered. When HPMVs were introduced in New Zealand in 2010, the initial pro-forma designs were based on 22m overall length limit which was seen as an acceptable increase over the standard vehicle length limit of 20m. This 22m limit was selected simply on the basis of being a reasonable increase with minimal safety risk and not on the basis of any vehicle performance constraints. Log trucks had already been operating at 22m under permit for a number of years with positive safety gains through improved rollover stability so there was evidence that this length did not pose undue safety risks. Subsequently pro-forma designs at 22.3m and then 23m maximum overall length were developed. These length increases came about to achieve productivity gains where, for example, a small increase in deck length enabled an additional row of pallets to be accommodated. Although the low speed turning requirements do ultimately constrain the overall length of the vehicle, it is possible to meet the current low speed turning requirements with vehicle lengths greater than 23m. Several trial vehicles with overall lengths between 23m and 25m have been permitted. Following the publication of the Austroads report, the NZTA has applied a maximum overall length limit of 23m and no new permits have been issued for vehicles over 23m.

The Austroads study was based on a risk assessment of different vehicle lengths at 55 sites in Australia. These sites were selected as being representative but there were no sites from New South Wales (the most populous state), Tasmania and Northern Territory, while there were 40 sites from South Australia, six from Queensland, five from Australian Capital Territory, three from Victoria and one from Western Australia. Thus the distribution of sites does not reflect either population or network length. The risk scores were banded into one of three categories low, medium or high. This banding process resulted in the appearance of sudden increase in risk with increasing vehicle length above 23m. However, the underlying points score indicates a more gradual increase in risk. Nevertheless, it is logical that if the infrastructure has been designed on the basis of a particular maximum vehicle length, then as the vehicle length increases beyond this maximum, an increasing number of infrastructure elements will be found to be inadequately designed and so the risk profile will increase at an accelerating rate. At some vehicle length value, the risk level will become unacceptable. The difficulty is determining the maximum acceptable length value for the New Zealand network. The Austroads study is based on Australian road network which does not necessarily have the same characteristics as the New Zealand network but we do not have a comparable analysis for

sites on the New Zealand network. Prior to the introduction of the PBS, the maximum length general access vehicle in Australia was 19m compared to 20m in New Zealand. Thus the Australian infrastructure design was based on a similar but slightly shorter maximum vehicle length. The maximum length pro-forma vehicles in New Zealand are 23m long and many of these have been operating for several years now without any evidence that this length is problematic (Stimpson & Co 2014). It therefore seems appropriate to suggest that this maximum length should remain for PBS vehicles with general or widespread access. There is the potential to allow longer vehicles on specific routes which can be risk-assessed in detail.

In summary, I believe that, for general or widespread access PBS vehicles, the overall dimensional envelope should be prescribed. The maximum height and maximum width limits should be the same as those for standard vehicles. Under the new 2016 VDAM Rule these limits are 4.3m and 2.55m respectively. The overall length limit should also be prescribed and in the absence of better information, the currently used value of 23m is appropriate. Note that all three of these limits are based on the perceived capacity of the existing infrastructure. They are not based on vehicle performance and this is the reason why they should be prescribed rather than the subject of performance assessment.

Infrastructure Standards and Productivity Measures

Pavement wear is generated by repeated loading applied by the axles and tyres of the vehicles passing over it. The relationship between the magnitude of the applied axle load and the amount of pavement wear that results is not linear. As well as being non-linear it also varies with pavement type and with the pavement wear mechanism being considered. The road user charges (RUC) regime in New Zealand assumes that the average relationship is a fourth power one. RUCs are based on road users paying for the full cost of the pavement wear that they generate and thus, in theory, there should be no need to limit axle weights explicitly because the RUCs associated with higher weights should be so high that it is uneconomic to operate at these weights.

In practice, the situation is more complicated than this. The fourth power model of pavement wear is analogous to Miner's Law for metal fatigue and describes the gradual accumulation of wear over a large number of loading cycles. Very high loads can cause damage to the pavement in a relatively small number of load applications and so maximum axle load limits are set to prevent this overloading effect.

In both Australian and Europe higher weight limits are allowed for axles fitted with "road-friendly" suspensions. As a vehicle moves along the road the axles and body move up and down in response to the unevenness of the road surface. The loads applied to the road by the axles are not constant but vary up and down about an average weight. If we assume that there is a fourth power relationship between axle load and pavement wear, then the additional pavement wear caused when the axle load is higher than average is greater than the reduced pavement wear when the axle load is lower than average. Thus dynamic wheel forces generate more pavement wear than constant wheel forces. Using a "road-friendly" suspension reduces the magnitude of the dynamic wheel forces and thus causes less pavement wear than a suspension that is not "road-friendly". Calculating the magnitude of this effect is very difficult. Dynamic wheel forces vary with vehicle speed, road roughness and vehicle load as well as suspension type. The pavement wear effect also depends on the spatial distribution of these dynamic loads and whether we consider average wear or localised peak wear.

Generally modern air suspensions are "road-friendly" and older mechanical suspensions are not, but air suspensions can cease to be "road-friendly" if their dampers (shock absorbers) are worn and not functioning properly while some modern mechanical suspensions are "road-friendly". A number of studies in the 1980s and 90s investigated the dynamic wheel forces generated by different suspensions (Sweatman 1983, Magnusson, Carlson et al. 1984, Woodrooffe, LeBlanc et al. 1986, Hahn 1987, Mitchell and Gyenes 1989, de Pont 1997). As a typical example, Mitchell and Gyenes estimated pavement wear reductions for using air suspension instead of steel suspension of between 8% and 24% at 90km/h on a road of medium roughness. Although these figures sound substantial, using the fourth power model that they are based on, this is equivalent in effect to an increase in static axle load of 2% to 5.5%. In other words, a tandem group fitted with steel suspension at 15 tonnes is equivalent to a tandem axle group with air suspension (in good condition) loaded to somewhere between 15.29 tonnes and 15.83 tonnes. At lower speed or on smoother road the effect is even less. Given that a regime that provides concessions for "road-friendly"

suspensions would require a significant administrative overhead and cost to approve suspensions and enforce in-service compliance and also that most new vehicles are being built with "road-friendly" suspensions anyway there would seem to be little net benefit in giving a weight concession for these suspensions.

The axle load limits embedded in the VDAM Rule provide the basis for pavement design practice in New Zealand. These same limits should apply to PBS vehicles. The 2010 Amendment to the VDAM Rule included some increases to these axle limits for approved routes. Presumably these higher limits are based on the pavements of the routes being designed for greater weight capacity and so these higher limits should also apply to PBS vehicles when operating on these same routes. For some axle group configurations such as the tridem and the quad, the axle group weight limits appear to be based on bridge formula considerations rather than pavement strength because the same weight limits apply to twin-tyres and wide single tyres. For all other axle configurations, the wide single tyres have a lower maximum weight than twin tyres.

The previous discussion concerned primarily vertical axle loads although this was not explicit. In the limit, the maximum horizontal tyre loads are determined by the vertical load and the friction coefficient between the tyres and the road. These maximum loads can occur during emergency braking and on tyres in multi-axle groups during extended tight turns but these situations occur relatively infrequently. In normal operations the horizontal forces are usually well below the friction limit.

The main vehicle factors affecting the magnitude of these horizontal forces are the vertical load and the number of axles in the axle groups, the axle spacing and whether or not any of the axles are steerable. These factors are all controlled in the current VDAM Rule. This allows for single, tandem, tridem and quad axle groups and specifies maximum weights for each based on spacing. It also specifies a maximum spread for each axle group configuration and a requirement that one axle in a quad set must be steering and that half or fewer of the axles in any axle group may steer.

The tyre scuffing forces generated by multi-axle groups in turns has been investigated in depth (Taramoeroa and de Pont 2008) but there have been some changes in the allowable configurations since then. When an axle group travels around a curve, the tyres cannot all achieve a zero slip angle. At each wheel position the magnitude of the slip angle is approximately proportional to the distance of the wheels from the axis of the axle group. Thus a tridem axle set, where the axle spread can be 3m, will have greater slip angles at the first and last axles than a tandem axle set, where the maximum spread is 2m. For the same turn radius, the slip angles for the outside axles of a maximum spread tridem group will be approximately 50% greater than those of a maximum spread tandem group. The magnitude of the cornering forces generated by the tyres are related to the size of the slip angles and the vertical load but this relationship is not linear. Figure 1 shows the relationship between cornering force and slip angle at different vertical loads for a common truck tyre. It is notable that as the slip angles get larger the rate of increase in cornering force reduces. Although not quite as easy to see from the figure the same effect is true for increasing vertical load particularly for small slip angles. For a tandem axle set with dual tyres at 15 tonnes loads, the vertical load on each tyre is 1875kg. For a tridem axle group with dual tyres at 18 tonnes the vertical load on each tyre is 1500kg. Thus the tridem axle requires a larger slip angle to generate the same level of horizontal tyre force as the tandem axle and can tolerate a larger spacing as is permitted in the VDAM Rule. The quad axle group is required to have a caster steering axle and it is implicitly assumed that this has a zero slip angle. In terms of horizontal tyre forces this results in the quad axle set behaving like a tridem with a maximum axle load of five tonnes and a spread of 2.67m. The resulting horizontal tyre forces are less than those of the tridem axle group which has an axle load of six tonnes and a spread of 3m. In practice, the caster steer axle typically has some centring force which results a slip angle being developed. This effectively increases the distance from the axis of the front axle of the quad set and hence increases the horizontal tyre forces. However, because of the lower axle weights these forces are still lower than those of a worst case tridem. If the turn radius is small enough and the turn angle is large enough the caster steer angle will eventually reach its maximum turn angle. From then on it does generate horizontal tyre forces and it effectively increases the axle group spread. In the 2008 study, the caster steer axles were modelled with a maximum steer angle of 15 degrees and, on a 13.75m radius turn, they typically hit the stops when the turn angle was greater than 110 degrees.

Furthermore, even then, the peak scuffing forces for the quad axle semi-trailer vehicle were significantly lower than those of the tridem semi-trailer.
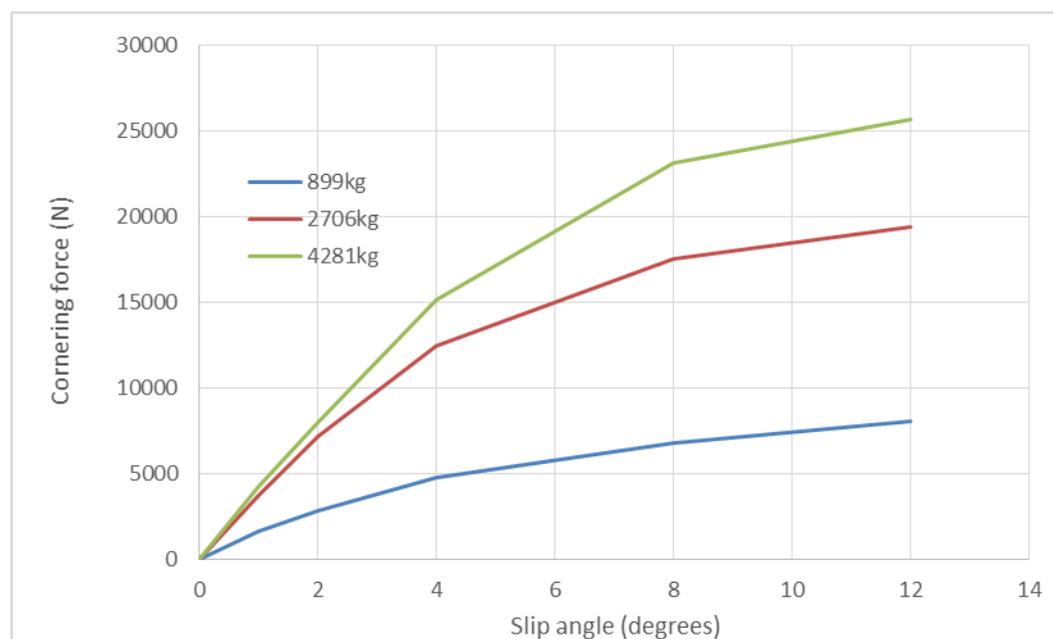

Figure 1. Cornering force vs slip angle at different vertical loads for 11R22.5 tyres.

We could specify a performance-based standard which would consist of a maximum allowable horizontal tyre force during a standard low speed turning manoeuvre. Potentially this would make it possible to have a greater axle spread for axle groups containing steering axles. However, some road controlling authorities have on various occasions indicated their concern about the damaging effects of scuffing forces from multi-axle groups at intersections and roundabouts where tight turns are undertaken. The simplest option therefore is to manage horizontal tyres forces by requiring PBS vehicles to comply with the standard axle group configuration requirements for numbers of axles, maximum load, maximum spread and steerable axles.

The VDAM Rule does not currently specify minimum centring force requirements for steerable axles. There is a generic requirement in the Rule for vehicle manufacturers and equipment suppliers to ensure that vehicle systems are safe but there are no explicit centring force requirements. Prior to the introduction of the VDAM Rule in 2002, there was a Land Transport Safety Authority LTSA) policy document (LTSA, 1996) which did specify minimum centring force requirements for these axles. The Australian requirements for quad axle sets (NHVR, 2018) specify that the axle group must have a rear steerable axle which has an effective centring mechanism. The VDAM Rule requirements for steerable axles on standard legal vehicles should be sufficient for PBS vehicles also. It may be desirable for the VDAM Rule to be amended so that the centring force requirements in the Rule are more explicit.

The third infrastructure standard in Australia is tyre contact pressure distribution and the Australian requirements are that PBS vehicles must comply with the rules that apply to standard vehicles. Tyre contact pressure distribution affects the forces that are applied to the road at and just below the surface. Clearly excessively high contact pressures can cause localised damage to the road surface without damaging the underlying structure. However, damage to the surface layer will enable water to penetrate and ultimately will undermine the structure. Land Transport Rule, Tyres and Wheels 2001, Rule 32013 details the requirements for wheels and tyres including specifying a maximum inflation pressure. There does not appear to be any basis for introducing separate requirements for PBS vehicles and thus we propose meeting the requirements of Rule 32103 is sufficient.

The final infrastructure standard in the Australian PBS system is bridge loading. There were originally proposals to base this on the bending moments and shear forces generated by the PBS vehicle not exceeding those generated by reference vehicles but ultimately the requirements have been simplified to a

series of formulae relating mass and length. In New Zealand, equivalent formulae exist but they are presented in tabular form rather than as equations. The New Zealand tabular form has been configured so that the weights increase in 1000kg intervals and thus the formulae are essentially a series of steps rather than a smooth line. In most cases this has no significant impact. However, some of the axle weight limits are not specified in whole tonnes. For example, the single axle with wide single tyres is 7200kg, the twin steer axle group is 11,000kg, and the tandem axle with dual tyres on one axle and wide singles on the other is 14,500kg. Combinations involving these axle groups will often not have whole number weight limits. Using a mathematical formula rather than a tabular approach would allow these vehicles to have slightly shorter overall axle spacings while still achieving their maximum load capacity. Although the differences in spacing limits are relatively small, some PBS vehicles will be approaching the performance limits in other aspects such as low speed turning performance and a small reduction in axle spacing may be useful.

Because of rounding the tabulated values shown in the VDAM Rule do not exactly correspond to piecewise linear formulae but they are quite close to it. The relevant equations are shown in Table 4. While it may be undesirable to allow very small weight increments because of difficulties with compliance and enforcement, it would be relatively straightforward to allow 0.5 tonne or even 0.25 tonne weight increments using the formulae. With this approach there would be less incentive for vehicle designers to artificially distort their vehicle geometry in order to maximise weight capacity.

Table 4. Proposed Bridge Formulae

Access Category	Bridge Formula
General access and 50MAX	$1.8m \leq L < 5.8m \quad W \leq 10 + 3L$ $5.8m \leq L \quad W \leq Min (18 + 1.6L, 50t)$
HPMV	$1.8m \leq L < 8m \quad W \leq 10 + 3L$ $8m \leq L \quad W \leq Min (18 + 2L, 62t)$

The development of the Australian PBS system also considered productivity measures based on pavement wear per tonne of payload or pavement wear per tonne of gross weight. The pavement wear contribution was evaluated on the basis of the number of equivalent standard axles (ESA) applied by the laden vehicle. Conventionally ESA are calculated using a fourth power relationship and the reference standard axle is an 8.2 tonne single axle with dual tyres. Options to use a higher power (up to twelve) for calculating ESA were also considered. Ultimately none of these measures were included on the PBS system. In New Zealand, the RUC system is based on the principle of the full cost of pavement wear being recovered from the users. Thus, in principle, any PBS vehicle that generated high levels of pavement wear would incur high levels of RUCs. Operators would only choose to use such a vehicle if it were economically efficient to do so. Historically, New Zealand vehicles have been configured with more axles than the minimum number required to carry the load because these incurred lower RUCs and were more economically efficient. That is, the savings in RUCs were greater than the loss of revenue from reduced payload capacity. Provided the RUCs accurately reflect the transport system costs, this is the optimum solution for the system. For PBS vehicles, we can assume that the RUCs will reflect the transport system costs imposed by the vehicles and thus operators will choose whether or not to use a particular configuration on the basis of its economic efficiency. There is no need to impose any further productivity performance requirement.

Drivetrain Performance

In the Australian PBS system there are three performance measures relating to the vehicle's drivetrain performance. These are startability, gradeability and acceleration capability. Gradeability is further subdivided in two components, low speed and high speed. In the Australian system these measures are usually evaluated by simulation but, in my view, this is an unnecessary level of complication. Furthermore, the results often depend on some driver performance parameters, such as gear change time, and some vehicle parameters such as aerodynamic drag coefficient, which are largely speculative. The key issues are:

- Can the engine and drivetrain generate enough power and torque to achieve satisfactory performance?
- Can the drive wheels transmit this power and torque to the road?

The startability requirement is the maximum grade on which the vehicle must be able to start from rest while the low speed gradeability is the maximum grade on which the vehicle can maintain forward motion. The startability requirement will always be less than the low speed gradeability requirement for two reasons. The first is that for gradeability the vehicle merely has to maintain speed while for startability it has to accelerate as well albeit at a minimal rate. Secondly, gradeability can be achieved with the engine operating at its peak torque while startability is based on the clutch engagement torque which is always less.

The Australian PBS requirement for low speed gradeability on general access routes is 20%. This means that the vehicle has to be able to maintain forward motion on a 20% grade. If we assume that the tyre-road friction coefficient is 0.8, this requires that 25% or more of the gross combination weight is on the drive axles. The current VDAM Rule does not specify a minimum weight for the drive axles. It does require vehicles over 39 tonnes to have at least two driven axles. However, this does mean that vehicles of 39 tonnes may have only one driven axle. If this is a single axle, its maximum weight is 8.2 tonnes (21% of 39 tonnes) while if it is part of a tandem axle set its maximum weight is 7.5 tonnes (19% of 39 tonnes). Thus some existing standard legal vehicles would not be able to achieve 20% gradeability. The more common combinations are 44 tonnes vehicles with two driven axles. At full load, the drive axles would only need to be loaded to more than 11 tonnes and this would almost always be true. The startability requirements for general access in Australia is 15%. Thus if the weight on the drive axles weight is adequate for low speed gradeability it is also adequate for startability.

In winter conditions, the presence of snow and/or ice can reduce the road friction coefficient which would negatively impact on startability and gradeability performance. However, because these are relatively rare events and New Zealand vehicles are not set up to operate winter conditions the NZTA takes a conservative approach to managing the network in these circumstances. Treatments are applied to mitigate the effects of ice and to remove snow and where this has not yet been done roads are closed either to towing vehicles, vehicles without chains or all vehicles. If the friction levels are deemed to be too low to be safe, the roads should be closed.

This raises the question as to what are the appropriate levels for startability and low speed gradeability in New Zealand. The State Highway Geometric Design Manual (Transit New Zealand 2002) specifies that a maximum grade of 10% applies to State Highways but notes that there may be circumstances where steeper grades are unavoidable e.g. in mountainous terrain. Baldwin St in Dunedin has a 35% grade but this is exceptional and is not a truck route. Nevertheless, there are roads on the network which have grades in excess of 10%. Generally, New Zealand's terrain is more mountainous than Australia's and thus it is unlikely that a lower standard for low speed gradeability and startability is appropriate.

We can assume that, for low speed gradeabilty and startability, aerodynamic drag forces are negligible. If we assume that the rolling resistance coefficient for heavy trucks is 0.01 (AUSTROADS 2002) then for low speed gradeability the tractive force required is

$$Force_{tractive} = Mg.(0.01 + \theta)$$

where   M = mass of the vehicle in kg

Θ = gradient of the road in m/m = 0.20 minimum

From this we can calculate the minimum engine peak torque required for a 20% grade as

$$Torque_{peak} \geq \frac{0.21Mg.r}{final\ drive\ ratio.first\ gear\ ratio}$$

where   r = radius of the drive wheels in m.

The same approach applies to startability except that we are calculating the minimum clutch engagement torque rather than the peak torque and we have to allow for a minimal level of acceleration. A calculation of startability by Volvo Truck Corporation for a South African PBS assessment (de Saxe 2012) uses an acceleration value of 0.4m/s^2. Using this we get the following equation for minimum clutch engagement torque when the startability limit is 15%.

$$Torque_{clutch\ engagement} \geq \frac{0.20 Mg.r}{final\ drive\ ratio.first\ gear\ ratio}$$

Note that with this 0.4m/s^2 acceleration rate, the minimum clutch engagement torque is only 5% less than the minimum peak torque. In practice, this will mean that the clutch engagement torque is always the critical value and if the vehicle has sufficient clutch engagement torque, it will be guaranteed to have sufficient peak torque. If we specify a lower minimum acceleration rate for the startability criterion, then both torque criteria may be relevant. With a minimum acceleration rate of 0.2m/s^2 the minimum clutch engagement torque is 86% of the minimum peak torque.

For high speed gradeability, we could use a similar approach but we would have to determine which gear the vehicle would be in at the higher speeds. It is simpler to consider the power requirements. At higher speeds aerodynamic drag forces need to be included. These depend on the aerodynamic drag coefficient, C_D, and the frontal area, A. For articulated trucks (AUSTROADS 2002) lists typical values of 0.65 for C_D and 8.5m^2 for frontal area. The Volvo Truck Corporation calculation (de Saxe 2012) uses 1.0 for C_D and 11.5m^2 for frontal area. The variation in these numbers illustrate the uncertainty associated with the simulation modelling approach. In New Zealand, the maximum height is 4.3m and the maximum width is 2.55m so the maximum frontal area is 10.965m^2. If we use this area and the Volvo drag coefficient, which are both more conservative that the Austroads figures, we find that the minimum power in watts required to achieve at least 80km/h on a 1% grade is given by

$$Power \geq 4.36M + 73702$$

Associated with this is a requirement that the vehicle gearing is high enough for it to travel at 80km/h or more when the engine speed is at the level required for peak power.

The third performance requirement in the Australian PBS system is acceleration capability. The other requirements already defined cover low speed and high speed performance. If the vehicle meets both of these it should have satisfactory acceleration capability. It is theoretically possible to for an engine's torque characteristics and its gearbox to be so poorly matched that it is not possible to achieve satisfactory performance but no operator would ever buy such a vehicle.

The torque and power requirements shown above did not include any efficiency losses for the drivetrain. Generally these are quite small but the engine power and torque requirements could be boosted by 5% or so to allow for losses.

Applying the above requirements to a 50MAX vehicle with a tyre rolling radius of 0.52m, a first gear ratio of 12.5 and a final drive ratio of 3.46 results in a minimum peak torque requirement of 1238Nm, a minimum clutch engagement torque of 1179Nm (with the 0.2m/s^2 acceleration requirement this value would be 1062Nm), a minimum power requirement of 295kW and with the drive axles loaded to at least 12.5 tonnes.

Braking Performance

In Australia it was initially proposed that there should be a requirement for the vehicles to comply with the American FMVSS 121 requirements for braking in a turn. This test measures the lane-keeping of a vehicle during full brake application while travelling at speed through a turn on a low friction surface. Essentially this is a test to measure the effectiveness of the vehicle's ABS braking system in preventing wheel lock-up. Without ABS or EBS, full brake application on a low friction surface should cause wheel lock-up and, because of the lateral acceleration generated by travelling through a curve at speed, the vehicle will slide

sideways and will not stay in its lane. At the time, ABS was not mandatory in Australia and so clearly many vehicles would not be able to achieve this performance.

Subsequently the PBS requirements were altered so that the performance test is undertaken in a straight line on a high friction surface. General access vehicles are required to achieve average deceleration rates of 0.4g for rigid vehicles and 0.35g for semi-trailers without significant wheel lock-up and while staying within their lane. The test is done with the vehicle in an unladen condition. Essentially this performance standard requires the vehicle's brake system to apply balanced braking across all axles.

Although the performance requirements are detailed in the PBS rules, vehicles are deemed to comply if they have ABS brakes or they have a load proportioning brake system that complies with Australian Design Rules 35 and 38.

The New Zealand Heavy Vehicle Brake Rule (2006) specifies requirements for heavy vehicles to have balanced brake performance in all road-legal load conditions. These requirements are more rigorous than the Australian Design Rules. Thus, in my view, compliance with the requirements of the New Zealand Heavy Vehicle Brake Rule is sufficient to ensure satisfactory braking performance for PBS vehicles and no additional performance standards are required.

Steady Turn Directional Performance

There are two key aspects to steady speed turning performance of vehicles. At very low speeds, there is no lateral acceleration (sideways force) generated and thus the tyres are not required to generate any sideways force to counter the lateral acceleration. In principle all the tyres can roll in the direction of travel with zero slip angle. In practice, this is not always possible. If the centre of a steer axle is following a circular path, the required turn angles for the left and right steer tyres to have both zero slip are different. The steering linkages are usually configured to approximately achieve the correct turn angles but a perfect match is not achievable. When a tandem axle group follows a curved path, all of the wheels must necessarily operate at a slip angle and thus all of them generate some sideways forces. Across the whole vehicle these sideways forces cancel out and there is no net sideways force. The effect of this "zero" slip requirement is that the paths of the rear axle groups track inside the paths of the axle groups ahead of them and so the vehicle track inboard of the turn and the road width required is greater than the width of the vehicle. The axis of an axle group is the location where the group could be replaced by a single axle which would have zero slip. For the purposes of prescriptive regulations the axis of an axle group is usually defined geometrically but this is an approximation and is not exactly correct. The magnitude of the low speed offtracking is primarily influenced by the vehicle geometry, the radius of the turn and the turn angle. Vehicle load and tyre properties also have a small effect. If a vehicle is held in a constant radius low speed turn, it will reach a steady state situation where all the axles are following concentric circular paths with a different radius for each axle. The turn angle required to achieve steady state varies with vehicle configuration and turn radius. Shorter rigid vehicles will typically achieve steady state offtracking within 180° of turn while longer combination vehicles can take 270° or more (Taramoeroa and de Pont 2008).

On high speed turns, centrifugal forces are generated. If the vehicle is not to slide out of the curve, these forces must be balanced by equal and opposite forces generated by the tyres at the road interface. The tyres generate these forces by operating with a slip angle. That is, there is an angle between the direction that the tyres are pointed and the direction that the axle is moving. To achieve the appropriate forces (in both magnitude and direction) the rear of the vehicle must offtrack outboard of the path of the zero slip condition.   On large radius turns where the vehicle speed is high the rear of the vehicle tracks outboard of the front of the vehicle and the road width required is greater than the width of the vehicle.

For most real world turns, the vehicle's offtracking behaviour is a combination of these two effects. On low speed small radius turns, the low speed offtracking effect dominates and the vehicle offtracks on the inboard side while on high speed large radius turns the high speed offtracking effect dominates and the vehicle offtracks on the outboard side. At some intermediate speed and radius combinations, the two effects will cancel each other and there is virtually no offtracking. Note that these zero offtracking conditions may be slightly different for each axle group and so, when the vehicle as a whole has zero offtracking there may still be some offtracking at some of the axle groups.

The approach used in most performance assessments is to consider these two components of offtracking separately and to specify performance standards for each. For low speed offtracking the most widely used test manoeuvre is a 90° circular turn with a tangential approach and exit. Various approaches have been used for specifying the turn radius. The computer simulations in the RTAC study were based an 11m radius for the path of the outside of the steer tyre, but they were undertaken using a radius of 9.8m for the path of the centre of the steer axle which is approximately, but not exactly, the same. The reason for this is that the driver model in their computer simulation software controls the path of the centre of the steer axle. The experimental testing in the RTAC study used an inner turn radius of 15m for the 90° turn. Thus they constrained the path of the worst case inboard tyres to be a circular arc and they allowed the driver to drive the tractor through whatever path was necessary to achieve this. In the Australian PBS system, a 12.5m radius path is used for the edge of the outside steer tyre.

With the introduction of HPMVs the NZTA introduced a 120° turn manoeuvre. This has an outside radius of 12.5m but is a wall-to-wall turn and the outside radius determines the path of the front outside corner of the vehicle.

The RTAC study experimental programme also included a 30m radius 360° turn to determine the steady state offtracking of the vehicle. This steady state offtracking can also be determined theoretically from the "equivalent wheelbase" (de Pont 2010) of the vehicle. This offtracking calculation is described in (Heald 1986). The RTAC report states that the difference between the measured and the computed offtracking values was less than 0.5% for all axles and vehicles evaluated.

For any of these test manoeuvres a test speed must be prescribed. In the RTAC simulation study, a speed of 8.25km/h was used. The experimental tests were conducted at 5km/h or less ("creep speed"). The Australian PBS system specifies a maximum speed of 5km/h. The RTAC simulations were done with the vehicles laden as was the 360° turn experimental test. The 90° turn experimental test were done with the vehicle unladen. The Australian PBS system specifies that the tests should be done both laden and unladen.

The primary low speed turning performance characteristic measured also varies. The RTAC study uses offtracking, which is the maximum offset between the path of the steer axle and the path of the worst case rear axle. This is also the measure that we have used in New Zealand for the 90° turn. It quantifies the additional road width required on a kerb-to-kerb basis, i.e. on the road surface. The Australian PBS system uses swept width, which is also, effectively, the measure used in the NZTA 120° turn manoeuvre. This quantifies the total road width required on a wall-to-wall basis, i.e. including the space above the road surface. The difference between the two approaches is primarily in how the front overhang of the truck/tractor is treated. With the offtracking approach it has no effect on the road width requirements while with the swept width approach it does increase the road space requirements. Arguments can be made for either case. In my view, the space above the road surface is important to other road users such as oncoming vehicles on an undivided highway and pedestrians on footpaths adjacent to corners. Thus I propose using swept width as the primary performance measure.

For consistency, I propose using the 12.5m radius 90° kerb-to-kerb turn at 5km/h as the basic manoeuvre for evaluating low speed turning performance. This is the Australian PBS manoeuvre. Using a kerb-to-kerb turn rather than a wall-to-wall turn simplifies the computer simulations because the driver models are usually based on controlling the position of the steer axle. It has no significant effect on the outcome because the pass/fail criteria are based on performance with respect to the specific manoeuvre. The basic performance measure will be swept width. The Australian PBS system specifies a maximum value for swept width of 7.4m for level 1 or general access. The 19m quad-axle semi-trailer with a single steer axle, which was used as the reference vehicle for the development of the pro-forma HPMV designs has a swept width of 6.87m. The 18m tridem semi-trailer which was the reference vehicle prior to the introduction of the quads has a swept width of 6.60m. The limit case 23m pro-forma B-train designs have swept widths of 6.99m-7.00m, while the limit case 23m truck-trailer designs have swept widths of 6.93m-6.94m. There is no absolute "right" answer for what the pass/fail threshold should be. The NZTA publishes tracking curves for various design vehicles at various turn radii (NZTA 2007). These tracking curves are in graphical form with scale drawings and are available on-line. The most recent version is dated 2007 and the design combination vehicle in these drawing is the RTS18 which is a 17.9m long quad axle semi-trailer with two

caster steer axles in the quad axle group.   The swept width of this vehicle is 6.68m using the Australian PBS manouevre. If we were to set the maximum swept width to 7.0m, all of the existing pro-forma designs should be compliant. For any that are not compliant the modifications required will be minor. The worst performing standard vehicles will also comply. However, the NZTA has some concerns about the road width used by the poorer performing combinations on low speed highway curves (those with advisory speeds of 15 and 25 km/h). At some locations these vehicles require more than the lane width available.

The preceding discussion on swept width limit values is all based on 2.5m wide vehicles. The 2016 VDAM Rule now allows 2.55m wide vehicles and thus the swept width of these limit vehicles would increase. Because the vehicles are at an angle to the swept width measurement direction the potential increase is greater than 0.05m but generally, the front of the truck/tractor is not the full width of the vehicle and thus a 0.05m additional width allowance should be sufficient.

If we set the maximum swept width limit to a lower value, say 6.75m which reflects the performance of the reference vehicle in the current tracking curves with an additional 0.05m, then most if not all of the current pro-forma designs would not comply and the limit case 19m quad semi-trailer would also not comply. The pro-forma design envelopes would all have to be reworked to achieve compliance and generally this would mean reductions in the maximum wheelbases and combined wheelbases. It is likely that the 23m overall length limit could still be achieved but it is doubtful whether the 20m first-to-last axle spacing which is required for 50MAX vehicles could still be attained. A study undertaken by TERNZ for NZTA showed that reducing the first-to-last axle spacing to 19m would enable this level of low speed turning performance to be achieved. For HPMVs operating under the higher weight limits, there is also likely to be reduction in the axle spacings attainable and hence in their gross weight capacity.   Although the NZTA wishes to achieve better low speed turning performance in order to reduce the road width demand it does not want to compromise the productivity gains that have been achieved with HPMVs. Thus we propose to set the low speed width limit at a maximum of 6.95m (including the additional 0.05m for the change in legal maximum width). This represents better low speed turning performance than is achieved by the current HPMVs but not quite as good as that achieved by the RTS18 vehicle. This particular performance measure relates primarily to the vehicles' performance at intersections and roundabouts. Their performance on low speed highway curves is better captured by the low speed steady state performance standard discussed below.

Several other performance measures are typically extracted from the low speed turning manoeuvre. The most common are tail swing, frontal swing, steer tyre friction demand. In the RTAC study, they also evaluated a tight-turn jack-knife measure. This measure is the friction demand on the drive axles during a tight turn and quantifies the propensity of the vehicle to jack-knife in low friction conditions (ice and snow). These conditions do not normally occur in New Zealand and, in my view, there is no need to include this measure in a New Zealand PBS system. It was not included in the Australian PBS system for the same reason.

The tail swing controls the extent to which the rear body work of the vehicle can cross over a kerb or a lane while the vehicle's wheels do not cross. In both the RTAC study and the Australian PBS system, tail swing is defined as the maximum excursion of outside rear corner of the vehicle beyond the plane of the outside edge of the vehicle at the start and finish of the low speed turning manoeuvre. The RTAC study uses a reference value of 0.3m for this measure and the Australian PBS system similarly uses a maximum value of 0.3m for general access. The same level has traditionally been used in New Zealand based on the Canadian and Australian practice. The European requirements (EC 1997) provide for a maximum tail swing of 0.8m for rigid vehicles and 1.2m for articulated vehicles but the test manoeuvre is different and will generate a larger value for tail swing. There does not appear to be an evidential basis for determining the 0.3m limit value but it has been used for some time without any indication of problems associated with it. Thus there appears to be no reason to set a more restrictive limit. It is less clear whether there are any reasons to set a higher limit. Longer buses with larger rear overhangs will generate more tail swing but these are also the vehicles where this poses a safety risk. In the absence of evidence we recommend a tail swing limit of 0.3m. Note that for a conventional vehicle without steerable axles on the trailer, tail swing occurs at the start of the turn. With steerable axles it is possible for tail swing to also occur at the end of the turn.

Frontal swing is analogous to tail swing but in relation to the front body work. In the Australian PBS system frontal swing is quite a complex performance measure. The simplest component is the frontal swing of the

truck or tractor. This is the extent to which the front outside corner of the vehicle travels outside the path of the outside edge of the outer front steer tyre. The maximum value occurs as the vehicle straightens out at the end of the low speed turn. The Australian PBS system specifies a maximum value of 0.7m for trucks and tractors and 1.5m for buses and coaches. This frontal swing value depends primarily on the front overhang of the vehicle and its wheelbase. Larger values for either of these parameters results in greater frontal swing. Frontal swing of the truck/tractor poses less of a safety risk than tail swing because the driver is in a better position to see what's happening. In general, there is little incentive for operators to select trucks or tractors with large front overhangs because this reduces the axle spacings and hence the load capacity under the bridge formulae. However, many European trucks have a frontal underrun protection which requires a larger front overhang, typically between 1300mm and 1500mm. This is a significant safety feature which should not be discouraged. The provision of a larger limit for buses and coaches is because these vehicles tend to have a relatively long wheelbase and a large front overhang is necessary to accommodate the passenger door forward of the front axle. The provision of access capability for wheelchairs requires a wide door which leads to a larger front overhang. Note that in this discussion, front overhang refers to the distance from the first steer axle to the front of the vehicle. In the VDAM Rule, front overhang is measured from the front edge of the driver's seat to the front of the vehicle and this measure is limited to a maximum of 3m (except for agricultural vehicles where 4m is allowed).

The more complex aspect of the frontal swing requirements in the Australian PBS systems are the semi-trailer requirements. The front corners of a semi-trailer also exhibit frontal swing at the exit of the turn. This is controlled through two measures called "difference of maxima" (DoM) and "maximum of differences" (MoD). These quantities are illustrated in Figure 2 below. The pass/fail levels are no greater than 0.40m for DoM and no greater than 0.20m for MoD. Thus the frontal swing requirements for semi-trailers are linked to the frontal swing performance of the tractor towing it. Although we can see the rationale for this approach, it can lead to undesirable design practices. We are aware of a case overseas where a bull-bar was fitted to a tractor unit to worsen its frontal swing so that the vehicle as a whole would meet the maximum DoM requirements. The complexity of these frontal swing requirements makes them quite difficult for the non-specialist to understand. In my view the complexity is not warranted by the safety risk posed and it is better to specify a simple maximum value for both the truck/tractor and the trailers. The VDAM specifies a maximum front overhang for semi-trailers using a 2.04m arc centred on the kingpin. If we assume that the vehicle is 2.55m wide, then the maximum possible frontal swing of the trailer is 0.765m (2.04 – 2.55/2). In practice it will be less than this. This would indicate that the maximum frontal swing should be set at about 0.7 or 0.75m for tractors, trucks and trailers. For buses and coaches there is a need to allow more frontal swing because of the large front overhang needed to accommodate the passenger door. The Australian PBS limit of 1.5m is based on the requirements of a 14.5m bus. Buses this length are permitted in New Zealand under permit with a route assessment. European buses are often shorter than this but with quite large front overhangs. From various manufacturer web-sites, 2.5m-2.7m is not unusual. The European regulations specify a wall-to-wall low speed turning requirement but not a separate frontal swing requirement. It seems reasonable to allow the 1.5m frontal swing limit for buses and coaches in New Zealand based on compatibility with the Australian PBS requirements, although existing standard legal length buses based on European designs may well not be able to meet this standard. We have not found any basis for quantifying a relationship between frontal swing and safety

risk.
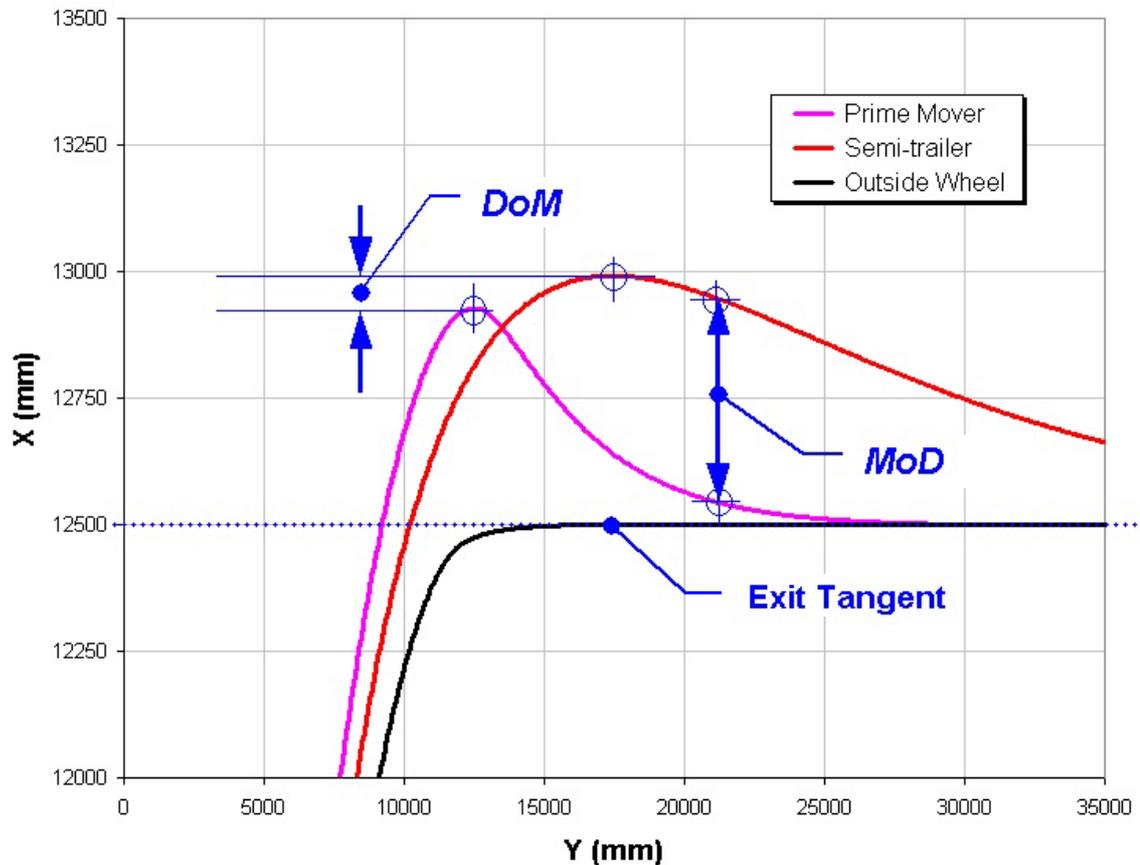
Figure 2. Illustration of frontal swing performance measures.

The third low speed turning performance measure that is often derived from the same manoeuvre is steer tyre friction demand. This is the sum of the absolute values of the horizontal tyre forces divided by the sum of the vertical tyre forces of the front steer axles. This needs to be less than the available friction or the steer tyres will not be able to generate enough cornering force to turn the vehicle. In the Australian PBS system the friction demand value is divided by the peak available tyre/road friction so effectively the measure becomes friction utilisation. However, the available road friction is assumed to be 0.8 and so this is simply a scaling factor. This measure can become critical for short wheelbase trucks/tractors with a wide spread and relatively high weights on the rear axles. This is particularly an issue with tridem drive axle groups. The limit value in the Australian PBS system is 80% of the available friction which, based on a peak friction value of 0.8, gives an absolute limit of 0.64. In my view, it is clearer if the peak tyre/road friction value is not embedded in the measure. The 0.8 peak friction value is typical for truck tyres on a dry road surface. For gravel roads, friction values of 0.6 are quoted while for wet roads values of 0.50 to 0.70 are given (Wong 1993). For wet roads, the friction is higher at lower speeds and this test is a low speed turn. With some margin of safety this would suggest that an upper limit of about 0.50 is appropriate for steer tyre friction demand. Note that this is more restrictive than the Australian PBS limit. Currently the 23m 5680 pro-forma B-train has the shortest tractor. This was modelled with a tractor wheelbase of 4220mm and a front overhang of 1460mm. The limit case of this vehicle has a steer tyre friction demand of 0.32 and this vehicle is right on the limit of acceptable low-speed turning performance. Potentially we can increase the payload volume of this vehicle design by reducing the tractor wheelbase and increasing the trailer wheelbases. Shortening the tractor wheelbase will increase the steer tyre friction demand and thus the performance measure is likely to be the one that limits the extent to which the tractor can be shortened. If we reduce the tractor wheelbase to 2900mm we can increase the combined trailer wheelbase by 800mm. In this case the steer tyre friction utilisation increases to 0.47. European vehicle manufacturers offer minimum tractor wheelbase options ranging from 2900mm to 3200mm. North American tractors tend to

have longer wheelbases. Again this suggests that a steer tyre friction demand upper limit of somewhere between 0.45 and 0.50 is appropriate.

With the introduction of HPMVs, the NZTA introduced a new low speed turning performance measure. This was a 120°, 12.5m radius wall-to-wall turn with a requirement that the vehicle does not cross a concentric inner circle of 4.9m radius. Effectively this is the same as specifying a 7.6m maximum swept width. The philosophy behind the 120° turn was that the worst case low speed turning situation is at small radius roundabouts where the vehicle is make a right turn. In this situation the turn angle is greater than 90° and so a larger turn angle was specified. Although this approach makes sense, there is very little difference in outcome between using the 90° turn and the 120° turn. For the same vehicle, the pass/fail threshold should change with turn angle but the outcome for the two turns will be similar. Interestingly, the pro-forma HPMV designs were based on achieved the same performance as a reference 19m quad-axle semi-trailer on the 120° turn. If we look at their performance on the 90° turn manoeuvre we see that the quad semi-trailer has a swept width of 6.87m where the pro-forma 23m B-trains had a swept width of about 7.0m and the 23m truck-trailers had a swept width of about 6.94m. Thus, if the pro-forma designs had been based on the performance of the quad semi-trailer using the 90° turn, their dimensional envelopes would have been slightly more restrictive than they are currently.

As noted above, the off-tracking behaviour of the vehicle during intermediate speed turns is a combination of the low speed (geometric) and high speed (kinetic) effects. The low speed turning performance characterised by the 90° 12.5m radius manoeuvre is not the steady state low speed off-tracking. For long combination vehicles on relatively small radius turns a large turn angle is required to achieve steady state off-tracking. However, for larger turn radii, steady state off-tracking is achieved more rapidly. Thus for the intermediate speed turns which involve a larger turn radius, we need to consider the steady state low speed off-tracking. There are two obvious approaches that can be used for characterising the low speed steady state off-tracking. One is to use a turning manoeuvre with a much larger turn angle, say 360°. The second is to simply consider the equivalent wheelbase of the combination vehicle. The experimental testing component of the RTAC study found that the measured steady state low speed off-tracking was within 0.5% of the value calculated using the equivalent wheelbase for all the vehicles that they tested. The equation for equivalent wheelbase is shown below.

$$WB_{equivalent} = \sqrt{\sum_i (WB_i)^2 - \sum_i (Hitch\_Offset_i)^2}$$

where WB_i is the wheelbase of vehicle unit I,
               $Hitch\_Offset_i$ is the distance between the rear axis and the coupling on vehicle unit i.

If we are going to base the off-tracking limit on a wall-to-wall turn, then the wheelbase of the truck or tractor should be its forward length i.e. the distance from the rear axis to the front of the vehicle.

Although the equivalent wheelbase calculation approach is quite attractive because it has been shown to be reliable and it is relatively unambiguous and robust and it can also be easily applied by vehicle designers as an initial check before any PBS assessment is undertaken, it runs into difficulties when trailers are fitted with steerable axles or when the axles in a group are unequally spaced. For a vehicle with axle groups, the wheelbase is measured from the axis of that group. For a standard axle group, where the axles are equally spaced and do not steer, the axis is assumed to be at the geometric centre of the group. This is an approximation but it is sufficiently accurate for these calculations. If the axle group has a self-steer axle(s), the VDAM Rule specifies that the axis should be determined by ignoring this axle(s). Effectively this assumes that the self-steer axle generates no cornering forces. However, in practice, a self-steer axle will have a mechanism that provides some centring force and this, in turn, generates cornering forces. Furthermore, the self-steer axle will have a maximum turn angle which it does not exceed. On an extended small radius turn, the self-steer axle will achieve this maximum turn angle and then stop turning. At that point it effectively becomes a fixed axle and can generate quite high cornering forces. With unequally spaced axles in a group, the problem is simpler but again the axis does not lie at the geometric centre and calculations are needed to determine the location of the axis.

The alternative approach is to undertake a low speed turn of sufficient turn angle so that steady state off-tracking is achieved. Either the off-tracking or the swept width can then be used as the performance measure. With a small turn radius and a long vehicle, it takes quite a large turn angle to achieve steady state and thus there is an advantage to using a larger radius for this manoeuvre. For convenience we have chosen the 25m radius wall-to-wall turn specified in the EC requirements for steerable axles (EC 1992) undertaken at 5km/h and used swept width as the performance measure.

Applying the equivalent wheelbase equation and this steady state swept width measure to the limit cases of the HPMV pro-forma designs gives the results shown in Table 5. The first point to note is that the 19.45m quad semi is essentially the same as a standard legal 19m quad semi and its wheelbase calculation ignores the self-steer axle. From this we can see that the B-trains are all a little "shorter" than the reference 19m quad semi-trailer while the truck and trailer combinations are all a little "longer". The reason for this appears to be that the B-trains develop their low speed off-tracking more quickly than the truck and full trailers. In the 90° turn, the B-trains have slightly more low speed off-tracking than the truck and trailer combinations, in the 120° turn they have the same off-tracking, because this is how they were designed, and in the steady state case with a large turn angle, the B-trains have slightly less low speed off-tracking than the truck and trailer combinations. The steady state swept width results follow approximately the same trend as the equivalent wheelbase but the correlation is not exact.

Table 5. Equivalent wheelbase and steady state swept width values for the limit case pro-forma HPMV designs.

Pro-forma HPMV Configuration	Equivalent wheelbase (m)	Steady state swept width (m)
23-metre truck and full trailer	11.37	5.20
23-metre truck and simple trailer (eg car transporter)	11.09	5.14
23-metre B-train 5680	10.97	5.04
23-metre B-train 5700	10.97	5.09
23-metre B-train long tractor	11.05	5.07
23-metre B-train long trailer	11.03	5.08
23-metre log truck and full trailer	11.37	5.02
19.45-metre quad semi	11.15	5.15
23-metre truck and long trailer	11.39	5.25
23-metre truck and trailer long drawbar	11.23	5.21

Both the 90° turn off-tracking performance and the steady state off-tracking performance are important to the overall fit of the vehicle on the network. Setting the maximum allowable equivalent wheelbase at 11.4m or the steady state swept with at 5.25m will allow all the existing pro-forma designs to continue to be used. This is comparable to setting the maximum swept width limit for the 90° turn at 7.0m. However, the NZTA is concerned about the low speed off-tracking performance of the larger vehicles in the fleet on lower speed highway curves and thus both these limits should be reduced. As noted above we propose to set the maximum swept width for the 90° turn to 6.95m. The RTS18 vehicle which we have used as a reference has a steady state swept width of 5.15m. Because of the two rear self-steer axles on this vehicle it is not possible to calculate a reliable equivalent wheelbase for it. Because of the difficulties with the equivalent wheelbase calculation when the vehicle has a castor steer axle or unequal spacing in an axle group, we recommend that steady state low speed turning performance should be characterised by measuring the swept width when the vehicle undertakes a 25m radius wall-to-wall turn at 5km/h and has achieved steady state off-tracking. Determining the appropriate pass/fail threshold is more complicated. At higher speeds some high speed (outboard) off-tracking will also occur which reduces the low speed off-tracking. The Austroads design guide (AUSTROADS 1999) provides recommended values for road widening on curves based on single unit trucks. The smallest curve radius considered is 30m and the guide states that for curves with a radius less than 30m, the actual paths of articulated vehicles should be found from

templates or computer simulation. It is not clear if this recommended methodology has been applied in practice in New Zealand. In principle, the steady speed off-tracking of the RTS18 quad-axle semi-trailer should be accommodated because it is the reference vehicle for tracking curves and thus, a threshold value of 5.20m should be acceptable (including an additional 0.05m for the new maximum width limit). This would include the pro-forma B-train designs but would require the pro-forma truck and trailer designs to be modified slightly. It should be noted that anecdotal evidence based on the video trials undertaken by NZTA suggests that there are some "pinch" points on the network where the level of curve widening is not adequate for a quad-axle semi-trailer. Generally drivers are aware of these locations and adjust their driving behaviour to mitigate any safety risk.

In the RTAC study high speed steady off-tracking (HSO) was measured on a 393m outside radius turn at 100km/h. This corresponds to a lateral acceleration of 0.2g. An earlier study in the USA (Fancher and Mathew 1987) used a 366m (1200ft) radius turn undertaken at 88.5 km/h (55mph). This corresponds to a lateral acceleration of 0.17g. Off-tracking is defined as the lateral offset between the path of the first steer axle and the path of the last axle on the rear trailer. The reference value used in the RTAC study was 0.46m while the earlier Fancher study proposed a target maximum level of 0.305m (1ft).   A New Zealand study (Latto and Currie 2012) into the off-tracking performance of quad-axle semi-trailers proposed some HSO target values for New Zealand. They used the RTAC turning manoeuvre of 393m radius and proposed using three levels of lateral acceleration, 0.2g, 0.25g and 0.3g. Their recommended pass/fail thresholds for these three lateral acceleration levels were 0.35m-0.42m, 0.6m and 0.8m respectively. It should be noted that these three lateral acceleration levels correspond to speeds of 100km/h, 112km/h and 122km/h respectively. Thus this performance standard can really only be undertaken by simulation for most vehicles.

The Australian PBS system initially included high speed steady off-tracking in it set of performance measures but then eliminated it in favour of tracking ability in a straight path (TASP) which is a related measure. TASP measures the swept width occupied by the vehicle when travelling at 90km/h on a straight road with a cross-slope of not less than 3% and a roughness of not less than 3.8 m/km IRI. The cross-slope requirement means that the vehicle is subject to a lateral acceleration of 0.03g while at the same time being subjected the vertical dynamics associated with a medium roughness road. The vertical dynamics will cause some roll motions of the vehicle which induces some roll steer at the axles and will cause some lateral wandering of the vehicle. Thus TASP measures the combined effect of two different largely unrelated factors. Furthermore, the level of lateral acceleration is quite low and the tyre slip angles required will be relatively small. In my view, it is better to measure high speed off-tracking performance separately from roll-steer effect and at lateral acceleration levels that are more representative of in-service cornering. The tyre response is non-linear and so the vehicle response at, say, 0.2g is not simply double the response at 0.1g.

The HSO manoeuvre with a 393m radius turn is intended to characterise the high speed off-tracking component independently of the low speed geometric off-tracking component. However it is impossible to separate the two completely. Using an equivalent wheelbase value of 11m (see Table 5) we can calculate the steady state low speed off-tracking. For a 393m radius turn it is 0.154m on the inboard side. Thus, if the total off-tracking is 0.42m on the outboard side, the actual high speed off-tracking component is 0.574m. Given that the legal speed limit for truck in New Zealand is 90km/h, it is appropriate that the HSO test manoeuvre should be based on this speed. For a lateral acceleration of 0.2g this implies a turn radius of 319m. Based on Latto and Currie's study the threshold for HSO should be 0.42m. The additional tests proposed by Latto and Currie are essentially designed to identify the problems associated with castor steer axles not generating enough lateral force when the demand is high and the tyres on the non-steering axles are reaching their saturation limits. In my view only one of these additional tests is necessary and this should be the 0.25g lateral acceleration. With a 319m radius turn this requires a speed of 100.7 km/h which, although illegal, is within observed speed behaviour.

The pass/fail criteria proposed by Latto and Currie (0.42m for the 0.2g turn on 319m radius and 0.6m for the 0.25g turn) is not achieved by a number of the current pro-forma HPMV designs. The 0.42m requirement using 319m radius turn is very similar to the RTAC reference value of 0.46m using a 393m turn. In New Zealand we have historically mostly used a reference value of 0.6m for the RTAC HSO manoeuvre. In practice there is a trade-off between low speed and high speed turning performance

(Fancher and Winkler 2007). Thus if we impose more restrictive HSO requirement as proposed by Latto and Currie, it is likely that we will not be able to achieve the required level of low speed turning performance at the current vehicle lengths and axle spreads. If we use the 319m radius turn for the manoeuvre, as proposed, at two speeds (90km/h and 100km/h) then offtracking upper limits of 0.46m and 0.68m respectively would enable most, if not all, of the current pro-forma designs to meet these standards. While accommodating existing vehicles is not of itself a basis for setting the performance standard pass/fail criteria, it does raise the question as to whether these existing vehicles have caused any safety concerns in relation to this aspect of performance. We are not aware of any evidence that this is the case.

An alternative performance test is specified in EC regulation 92/62/EEC (EC 1992) which specifies requirements for steerable axles on vehicles and trailers. The test in this EC directive is a 25m radius wall-to-wall turn which is negotiated twice; once at 5km/h and once at 25km/h. The vehicle continues turning until steady state is achieved. The 5km/h turn produces negligible lateral acceleration (less than 0.01g) while the 25km/h turn produces a lateral acceleration of 0.21g. The test standard is that the difference in offtracking between the low speed (5km/h) and the high speed (25km/h) turns should be less than 0.7m. This test does have the advantage that it can be done physically as well as by simulation. It would require a test site that has a paved area that is at least 50m x 50m but this is likely to be easier to find than a long sweeping curve of a specific constant radius, such as 319m, with minimal cross-slope. By measuring both the low speed and high speed offtracking on the same turn radius we can separate the two effects. However, the high speed offtracking component is measured while the vehicle is simultaneously experiencing substantial low speed offtracking and thus the tyre slip angles are quite different to what they are on a large radius curve at high speed when the vehicle is offtracking on the outboard side of a curve. Because the tyre cornering stiffness properties are non-linear, the high-speed offtracking value measured in this test is not the same as that measured for a 393m radius turn at 100km/h even though the lateral acceleration is roughly the same.

Relating this performance measure to real-life road space requirements is complicated. Thus our recommendation is to use the 319m radius turn as the HSO manoeuvre with speeds of 90km/h and 100km/h. The recommended pass/fail criteria are 0.46m and 0.68m respectively although these could be reviewed.

Stability and Dynamic Performance

The rollover stability of a vehicle during steady state cornering is characterised with a performance measure called static rollover threshold (SRT). Since 2002, there has been a requirement for most large heavy vehicles in New Zealand to have an SRT of not less than 0.35g. Thus SRT is well-established with the heavy transport industry in New Zealand and reasonably well understood. Several vehicle types where the consequences of a rollover event are likely to be more serious are subject to more demanding SRT requirements. Specifically, dangerous goods tank vehicles are required to have a minimum SRT of 0.45g, double decker buses and coaches are required to have a minimum SRT of 0.53g and single decker buses and coaches are required to have an SRT of 0.70g. The bus and coach requirements are defined as tilt angles based on a tilt table test. For HPMVs, the NZTA has required an SRT of not less than 0.35g and that the vehicle is fitted with roll stability control. Older trailers without roll stability control can be approved but a minimum SRT of 0.40g applies.

The Australian PBS system specifies a minimum SRT of 0.35g with 0.40g required for bulk dangerous goods vehicles and buses and coaches. There have been suggestions in Australia that the SRT requirements could be reduced for vehicles fitted with rollover stability control but this has not been implemented. The AUTOFORE project (CITA 2007) found that electronic vehicle systems were no more reliable than mechanical systems. This suggests that relaxing the SRT requirements on the basis of the electronic systems preventing the rollover situation is a risky approach. In my view, the current approach to SRT used in New Zealand is soundly based and has been effective. The New Zealand road environment is demanding with a substantial proportion of the network rated as hilly or mountainous. Consequently there are a lot of sections of winding road with relatively tight curves leading to a relatively high rollover risk.

Experimentally SRT has typically been determined using a tilt table test. There is an SAE recommended practice for undertaking these tests (SAE 1998). This is an option but, as far as we are aware, there are no tilt table facilities in New Zealand capable of undertaking this test.

For standard vehicles, compliance with the SRT requirements is usually undertaken using the SRT Calculator. This is an easy-to-use tool which is designed to provide a conservative estimate of the vehicle's SRT. For combination vehicles it can only estimate the SRT of the component vehicle units and not of the combination as a whole but again this is conservative because the SRT of the combination should not be worse than the SRT of the poorest performing vehicle unit in the combination. Thus the SRT can be determined using the SRT Calculator. However, if a computer simulation model of the vehicle has been built in order to evaluate the other performance measures, it is relatively straightforward to use this model to determine SRT and this should be more precise than the SRT Calculator.

The RTAC approach to determining SRT is to drive the model at a constant speed through a turn where the driver applies an initial small and slowly increasing steer angle so that the radius of the turn gradually decreases and the lateral acceleration steadily increases. Eventually the vehicle will get the point where the wheels start lifting off the ground. SRT is defined as the lateral acceleration when all the vehicle wheels on one side, except the steer tyres, have lifted off the ground. In the case of vehicle units that are not roll coupled (such as a truck and trailer) the SRT is achieved when the first of the vehicle units achieves wheel lift-off. The turning manoeuvre can be done in conjunction with the HSO manoeuvre. That is, the vehicle start with a constant radius turn to establish steady state off-tracking and then progresses to a gradually reducing turn radius. Thus it is logical to specify the test speed as 90 km/h. To avoid dynamic effects it is important that the turn radius does not change too quickly. The RTAC study uses a rate of change of steer tyre angle of 0.04 degrees per second.

The Australian PBS system uses an alternative approach where the turn radius is held constant and the vehicle speed gradually increases. The main reason for using this approach appears to be the response time of their very long combination vehicles. That is, the change in turn radius of the trailers lags behind that of the tractor (or truck) by some time. PBS assessment in New Zealand has mostly been done using the Yaw-Roll multi-body simulation software developed by the University of Michigan Transportation Research Institute (UMTRI). This is the same software that was used in the RTAC although it has been refined somewhat. Yaw-Roll uses a constant velocity model of vehicle and so implementing the Australian PBS approach to SRT in Yaw-Roll is rather clumsy because it is necessary to undertake repeat runs of the turning manoeuvre at incremental speeds. There does not appear to be any advantage in doing this for general access and 50Max vehicles and therefore we recommend using the RTAC method.

In the RTAC study there are three performance measures associated with the high-speed lane change manoeuvre. These are rearward amplification (RA), load transfer ratio (LTR) and high speed transient off-tracking (HSTO). The path of the high speed lane change manoeuvre is designed to produce a single sine wave lateral acceleration response at the tractor centre of gravity with an amplitude of 0.15g. In the RTAC study the test speed was 100km/h and three different paths were used to give periods for the sine wave of 2.0, 2.5 and 3.0 seconds. In practice it was found that the 3.0 second period path was the worst case (RTAC 1986) and the analyses were undertaken with this path. Note also that the driver model in the computer simulation controls the path of the centre of the steer axle rather than the centre of gravity of the tractor. For a typical tractor, these two points are quite close together and so this doesn't make much difference but for a rigid truck in, say, a truck and trailer combination, the centre of gravity of the truck could be quite some distance behind the centre of the steer axle and if the target path results in a 0.15g amplitude sine wave response at the steer axle, the response at the centre of gravity can be quite different.

Subsequent to the RTAC study, an ISO standard for stability testing has been developed (ISO 2000). The lane change manoeuvre in this standard is broadly the same as that used in the RTAC but with the details more rigidly specified. The Australian PBS measures use the ISO standard to define their lane manoeuvre but they specify a test speed of 88km/h and a period of 2.5 seconds so their manoeuvre is different from the RTAC manoeuvre.

There are also some differences in how the performance measures are defined and the pass/fail criteria. Rearward amplification is the ratio of the peak lateral acceleration of the rear trailer to the peak lateral acceleration of the tractor or truck. LTR is the peak of the proportion of the total weight that is transferred from one side of the vehicle to the other during the lane change manoeuvre. LTR is evaluated for each roll unit in the combination. Thus for a B-train, where all the vehicle units are roll-coupled, it is evaluated across the whole (excluding the steer axle) while for a truck and trailer combination it is evaluated for the truck and for the trailer separately.

The RTAC study introduced the LTR measure as a way of distinguishing between roll-coupled vehicles and non-roll-coupled vehicles. These two vehicle types could have a similar RA but the roll-coupled vehicle would have a lower LTR which reflects the fact that it is less likely to roll the rear trailer in an evasive manoeuvre. The reference values used were a maximum of 2.0 for RA and a maximum of 0.6 for LTR.

In developing the Australian PBS system it was found that RA, LTR and SRT are correlated. This is quite logical. The lane change manoeuvre applies a lateral acceleration to the truck or tractor. The RA describes how much this lateral acceleration is amplified at the rear trailer and the SRT of the rear trailer determine how close it is to rollover which, in turn is what LTR also describes. When comparing the results from different simulation packages it was found that the LTR results were the least consistent of the performance measures evaluated (Prem, Ramsay et al. 2001). These differences were attributed to differences in the paths taken by the driver models in the three simulation programmes used even though all three followed the prescribed path within the allowable tolerance (±0.15m). Because of the correlation between the three measures (SRT, RA and LTR) it was decided that only two were needed and thus LTR was eliminated from the Australian PBS system. To take into account, the roll-coupling effect mentioned in the previous paragraph, the RA measure was modified so that the peak lateral acceleration of the rear trailer became the peak acceleration of the rearmost roll-coupled unit (not including the tractor or truck). Thus for a B-train, the RA is the ratio of the peak lateral acceleration of both trailers treated as a single vehicle to the peak lateral acceleration of the tractor. Consequently with this approach the RAs of B-trains are much lower than those of truck and trailer combinations. Furthermore they specified the pass/fail criterion for RA so that it is 5.7 x SRT. This means that if the vehicle's SRT is 0.35g, the RA limit is 2.0 which is the same as the RTAC reference values. However, if the SRT is higher, then a higher RA value is acceptable. In an attempt to improve the consistency of the results they also reduced the allowable tolerance on the lane change path to ±0.03m. This reduced path tolerance effectively makes it impossible to undertake the test experimentally because it is beyond the capability of a human driver.

Thus for the New Zealand PBS system we have to decide whether to include both RA and LTR or just RA. We also have to choose the test manoeuvre, which definition of RA to use and the appropriate pass/fail criteria. In my view, the appropriate test speed should reflect New Zealand's operational speed limits. Thus the Australian 88km/h manoeuvre is more appropriate than the Canadian 100km/h manoeuvre. The 88km/h speed value is the result of the manoeuvre specification being based on a United States test with a 55mph speed. The most appropriate speed is 90km/h and it is a relatively simple matter to determine the path required to achieve the 0.15g lateral accelerations at this speed. For some consistency with Australia, I would recommend using the 2.5 second period for the manoeuvre. We could similarly apply the ±0.03m path tolerance requirement to reduce the variability between assessments.

Both RA and LTR have their place in characterising the dynamic performance of the vehicle. The RA definition should be the RTAC one which does not take into account roll coupling and a pass/fail criterion of a maximum value of 2.0 should be used. I don't think that there is a good justification for allowing vehicles with an SRT that is better than the minimum required to have a worse RA. It is likely that this condition was introduced in Australia to enable some of their road train configurations to pass.

The pass/fail threshold for LTR will be a sensitive issue in New Zealand. The reference value in the RTAC study was 0.6 but a number of truck and trailer combinations in New Zealand will not achieve this level particularly when modelled with generic suspensions. A study undertaken in New Zealand relating performance measures to crash rates (Mueller, de Pont et al. 1999) shows that, in terms of crash risk, an LTR of 0.74 has approximately the same relative crash risk as an SRT of 0.35g. Note, however, that these risk values are based on the curve fits shown in the report and there is some uncertainty associated with these. (Sweatman, Woodrooffe et al. 1998) also considered what levels of performance measure were

appropriate based on safety performance. They suggested that the LTR limit should be 0.8 for vehicles up 36.3 tonnes (80kips) and reducing in steps to 0.6 for vehicles over 54.4 tonnes (120kips).

Based on the crash rate analysis and the Sweatman et al results we could set the LTR pass/fail criterion for all vehicles up to 50 tonnes gross combination weight at no more than 0.7. For higher weight vehicles we can set the limit at 0.6. The alternative is to set the limit at 0.6 for all vehicles. Truck and trailer combinations can achieve this if they are suitably configured and are fitted with high roll stiffness suspensions. The difficulty with this approach will be in developing suitable pro-forma designs. Currently the pro-forma designs are based on using generic suspensions with relatively poor performance. If we use data from more modern suspensions for the generic suspensions we may well be able to achieve the 0.6 LTR value for the pro-forma truck and trailer designs. Alternatively it could become necessary to specify minimum suspension performance characteristics for some of the pro-forma truck and trailer designs.

The third performance measure that is determined from this lane change manoeuvre is HSTO. This is the maximum lateral excursion of the path of the rearmost axle from the path of the steer axle. The reference value in the RTAC study is 0.8m. In the Australian PBS system the pass/fail threshold is 0.6m for level 1 which is general access and 0.8m for level 2. There does not appear to be any strong evidence to support either of the two options from a safety point of view. From the results of the performance assessments of the pro-forma designs, it appears that a pass/fail level of no more than 0.6m is attainable for most vehicles currently operating in New Zealand and thus we recommend this as the threshold level.

The ISO standard 14791:2000 (ISO 2000) specifies an alternative test for determining RA. This involves applying pseudo-random steer inputs over a range of frequencies and amplitudes for a test run of at least 12 minutes although this can be made up of a series of run at least 30 seconds each. The standard specifies that the test frequencies should span a range from 0.1Hz to, at least, 1Hz and that the amplitude of the steer inputs should vary with the ratio of the greatest to the least being not more than 4:1. There is a requirement that the lateral acceleration levels are kept with the range where the vehicle exhibits linear behaviour. The data processing for this test requires spectral analysis which is relatively complex and requires specialist software. This testing approach has an advantage that it identifies the frequency at which the worst case RA occurs and determines the RA at that frequency. On the other hand, it is limited to amplitudes where the vehicle response is linear. With some more active vehicle configurations, the lane change manoeuvre generates relatively large slip angles on the rear tyres and thus the vehicle response will be non-linear. The random steer input method will not capture this effect. Thus in my view it is better to use the lane change manoeuvre for determining RA.

The final dynamic performance measure that is used in both the RTAC study and the Australian PBS system is yaw damping ratio (YDR). The manoeuvre used for this measure is a pulse steer input undertaken at high speed. This initiates lateral sway of the trailer(s) which then decays over a number of oscillations. YDR measures the rate as which this decay occurs. The test manoeuvre used in the Australian PBS system is that defined in the ISO standard (ISO 2000). This standard also details the method for calculation the YDR from the amplitude of the successive peaks in either the yaw rate, articulation angle, or articulation angular velocity. Both the RTAC study and the Australian PBS system have a minimum acceptable level for YDR of 0.15 or 15%.

For New Zealand we could apply a test speed of 90 km/h in line with the speed limits for heavy vehicles. However, YDR is highly speed dependent and thus we propose specifying a test speed to 100km/h as it is conservative and it accounts for non-compliant speed behaviour and the possibility of speed limits being increased on some roads at some time in the future. The pass/fail criterion should be that the yaw damping ratio is no less than 0.15.

DEFAULT SETTINGS AND DATA REQUIREMENTS

Loading

The Australian PBS system states that the assessor should consider the worst case loading situation but gives no indication as to what this is. Thus this requirement is rather open-ended. It is possible to load a vehicle poorly and have a substantial negative impact on its safety performance but it is not reasonable to expect a PBS assessor to envisage all the possible ways in which the vehicle could be poorly loaded. We have to assume that the operator will follow sound loading practices so that as far as practicable:

- the longitudinal weight distribution results in the weight being distributed between the axle groups in proportion to the axle group's load capacity and,
- the lateral weight distribution is such that the centre of gravity on the load lies on the longitudinal axis of the vehicle and,
- the vertical load distribution minimises the height of the centre of gravity of the load.

Crashes have occurred because vehicles have been badly loaded and, as a consequence, their performance characteristics have been compromised. However, it is not practical to require a vehicle to have good safety performance regardless of how badly it has been loaded.

Low speed turning performance is usually a little worse for empty vehicles than for laden vehicles. Thus the low speed turning performance standards should be evaluated for both the laden vehicle and the unladen vehicle.

High speed performance, both dynamic and steady state, is influenced by weight and is generally worse for higher weights. Thus all of these measures should be evaluated for the laden vehicle.

There are some instances where partial loads can have adverse effects on performance, particularly for dynamic manoeuvres. This will typically occur when the partial load has quite a different weight distribution from the full load. Where this adverse partial loading occurs as part of the normal operating conditions of the vehicle, the high speed dynamic performance standards should be assessed for this load case. If the adverse partial loading is simply the result of poor loading practice and is avoidable through good loading practice no additional performance assessment should be required.

It is easy to find examples to illustrate these two cases. First let us consider a curtain-sider full trailer with a mezzanine floor such that two decks of pallets can be loaded into it. If the vehicle is used to deliver goods to a number of locations and the loading is not carefully planned, the situation can arise where the lower deck is unloaded first and the vehicle is travelling with a 50% load, all on the upper deck. This will adversely affect the rollover stability of the vehicle. However, this is the result of poor loading practice and is easily avoided by better planning. Even with poor load planning it would be possible to rearrange the load en route to avoid this unsafe situation. The PBS requirements should not be required to have to cope with such poor operating practice. Now let us consider a truck and trailer combination which has a 3000kg forklift that is transported by being mounted on a special deck at the rear of the trailer. When the combination is fully loaded the vehicle weight is appropriately distributed across all the axle groups. However, when the vehicle is unloaded, the effect of a 3000kg load applied behind the rear axle group of the trailer results in an unbalanced weight distribution which could cause yaw instability at high speed. This is a problem that is inherent in the vehicle's design and thus it is appropriate that the PBS assessment should evaluate the performance of the vehicle in this load state. Note that these two examples are based on actual vehicles that the author has encountered in the past.

Although these examples highlight the principles that should apply to determining whether a partial load or empty vehicle also needs to be assessed or not, they do not provide us with any hard and fast criteria for determining whether an additional assessment at partial or no load is necessary. Performance issues will occur when the axle load distribution differs significantly from that of the loaded vehicle, particularly if it is more heavily biased to the rear of the vehicle. Therefore we propose the following procedure for determining whether or not a partial load case should be performance-assessed:

- Determine the axle group weights and sprung mass centre of gravity heights of each vehicle unit for the fully laden case. For each axle group calculate the weight as a percentage of the total weight of the whole vehicle.
- Consider whether there are any partially laden situations (including empty) in normal operations where the sprung mass centre of gravity height of any vehicle unit is higher than it was for the laden case. If any such load cases exist they need to have a performance assessment done.
- Consider whether there are any partially laden situations (including empty) in normal operations where the proportion of the total vehicle weight carried by any of the trailer axle groups is higher than it is when the vehicle is fully laden. If any such load cases exist they need to have a performance assessment done.

Tyres

The tyres are the conduit for transmitting forces between the vehicle and the road and the characteristics of the tyres do have a significant effect on some aspects of vehicle performance. Of the performance measures outlined above, HSO, HSTO, LTR, RA and YDR are all significantly influenced by the cornering stiffness of the tyres (Fancher, Ervin et al. 1986).

The amount of lateral force generated by a tyre is related to both the slip angle and the amount of vertical load on the tyre. Both of these relationships are non-linear, as illustrated in Figure 1, and thus the tyre properties need to be measured for a range of vertical loads and a range of slip angles. These measurements are most usually done on a test rig and there are two basic types. One uses a roller drum and the other uses a flatbed, which is like an exercise treadmill. There are differences in the detail of the tyre-road surface contact zone between these two types of test machine and thus they do not produce identical results.

This problem of differences between the results from different testing machines came to the fore in New Zealand in the early 1990s with the 44-tonne A-train permits. A number of A-train milk tankers were granted permits to operate at 44-tonnes (instead of 39-tonnes) on the basis of a PBS assessment of their SRT and DLTR. These PBS assessments were based on a particular make and model of tyre for which data from a flatbed tester was available. Subsequently, the operators of these vehicles requested a variation to their permits to be able to use an alternative make and model of tyre. Roller drum test data for these tyres was provided by the tyre manufacturer. The PBS assessment was re-run with this new tyre data and the vehicles failed to meet the required standards. This was a surprise to the tyre manufacturer concerned as they believed that their tyre had similar performance characteristics to the one it was replacing. Following these results, the replacement tyres were sent to UMTRI, where the original tyres had been tested, for testing on their flatbed tester. These tests showed that the two brands of tyre were very similar in performance and thus that the differences in the measured data could be attributed to the differences in the two types of test machine.

However, there are also real differences in performance between different makes and models of tyres of the same size resulting from differences in design and materials used. Moreover, tyre properties change as the tyre wears and as it ages. The question of differences in tyre properties has been an issue in the Australian PBS system. PBS assessments are generally undertaken using the data for a particular make and model of tyre and the PBS permit then specifies that the approval is based on using this tyre. However, heavy vehicle tyres are a consumable item and are replaced at reasonably frequent intervals either because they are worn out or because they fail. Having to replace the tyres with identical tyres (same make and model) can be difficult particularly if the tyre has failed unexpectedly in a remote location. As well as this, competing tyre suppliers will be offering operators alternative tyres, which their PBS permit conditions prevent them from accepting. To overcome this, the Australian Road Transport Suppliers Association (ARTSA) have proposed the use of generic tyre data in two categories; standard tyres and superior tyres. The PBS assessment would be undertaken using one of these two categories of tyres and the PBS approval would be based on the tyre category used. All tyre suppliers would have any tyres that they wished to supply to the PBS vehicle market tested and rated as being suitable for either "standard" or "superior" tyre applications. The operator would then be able to use any make and model of tyre with a suitable rating on their vehicle.

This approach of using generic tyres has considerable merit operationally. It gives all tyre suppliers equal access to the market and gives operators the flexibility to choose any suitable tyres. If this is adopted in Australia, it makes sense for the New Zealand PBS system to align itself closely to the Australian categories. Tyre suppliers will get their tyres rated for the Australian market and these ratings would then be directly applicable to New Zealand.

As far as we are aware, the generic tyre approach has not yet formally been adopted as part of the PBS system in Australia. Many of the tyre suppliers in Australia do have manufacturer-supplied tyre test data available for their products to facilitate their acceptance for PBS vehicles. Generally this data is for 11R22.5 and 295/80R22.5 tyres which are the most widely used in Australia. In New Zealand we do also see the use, on trailers particularly, of wide-based single R22.5 tyres and of smaller R19.5 tyres. We currently have only a limited amount of data for these types of tyres. Comparing this tyre data with the data for an 11R22.5 tyre from the same manufacturer and the same model series is not as easy as might be expected. Tyre data typically consists of measurements at a range of slip angles with several vertical load options. However the vertical loads used are not necessarily the same for each tyre tested and so the data is not directly comparable. This same issue also makes it difficult to compare tyre data from different manufacturers even when the tyre size is identical.

The simplest solution to this is normalise the data by the vertical load so that instead of comparing the cornering force, F_y, we compare the cornering coefficient, F_y/F_z. These normalised data are illustrated in Figure 3 and Figure 4 which show the cornering coefficient data for an 11R22.5 and a 385/65R22.5 tyre respectively. The three graphs in each figure represent different levels of vertical loading. Both tyres are made by the same manufacturer and have the same model code but the measurements were taken at different times. Clearly the vertical loads used for the two tyres are different and the range of slip angles tested are also different. The obvious question is: how do we compare these two sets of data?
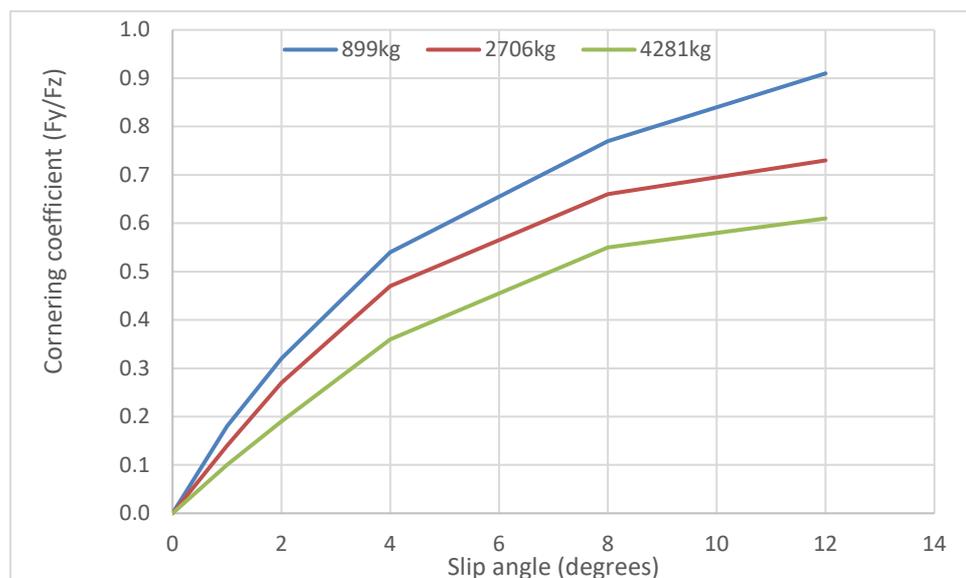

Figure 3. Cornering coefficient for 11R22.5 tyres.

The first thing that is clear is that the cornering coefficients reduce with increasing vertical load. Thus the worst case performance occurs when the tyres are more heavily loaded. Secondly the slope of the curves reduces as the slip angle increases. Thus for comparison purposes we need to consider the cornering coefficient at higher vertical loads and with larger slip angles. However, the vertical load at which the tyres are compared should not be the same for each tyre size and type. A wide single tyre will carry more load than a standard tyre in a dual tyre configuration and so it will need to generate more cornering force. We therefore propose that the cornering coefficient shown be evaluated at a vertical load that represents the maximum that the tyre can be loaded to legally at a slip angle of 6 degrees. The test data will not necessarily include this data point exactly but it can be evaluated by linear interpolation.

Single standard tyres can be loaded to 3000kg when in a single tyre axle configuration. Wide single tyres can be loaded to 3600kg in the same configuration. At 6 degrees of slip and 3000kg vertical load, the 11R22.5 tyres shown have a cornering coefficient of 0.54 while the wide single tyres (385/65R22.5) at 3600kg and 6 degrees of slip have a cornering coefficient of 0.68. The lower profile version of the standard single tyre (295/80R22.5) has a cornering coefficient of 0.65. All these tyres are from the same well-known manufacturer. If we are going to specify two levels of tyre, it is still not clear where the threshold between "standard" and "superior" should lie. From a practical point-of-view it is highly desirable that we should use the same threshold as Australia.
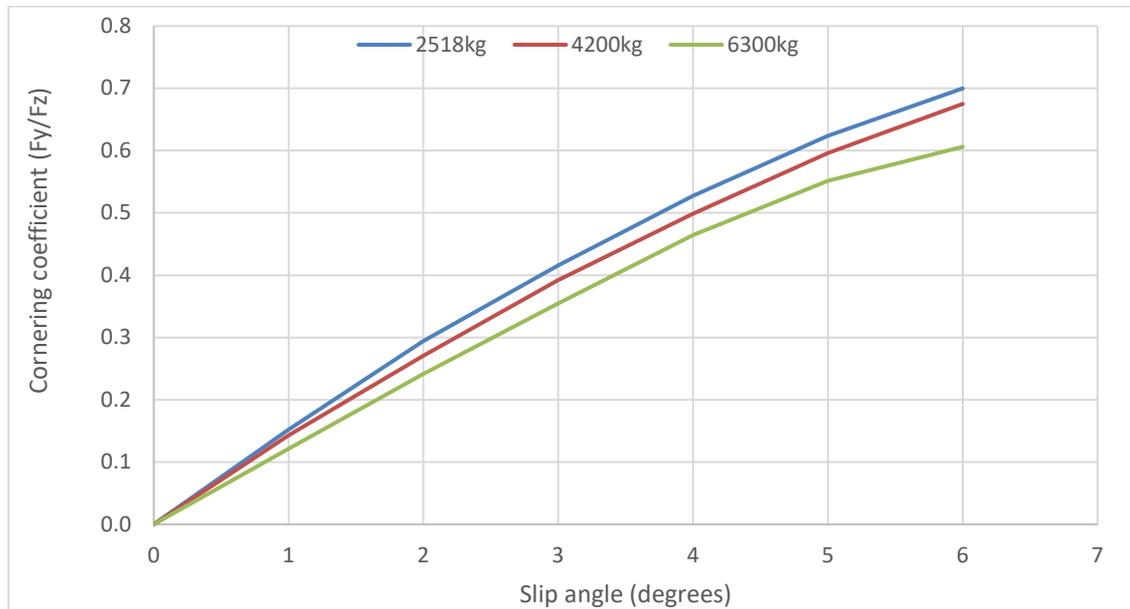
Figure 4. Cornering coefficient for 385/65R22.5 wide single tyres.

Suspensions and steerable axles

There are a number of suspension properties required by multi-body simulation software packages in order to model the vehicle's responses during the various PBS manoeuvres. Some of them are much more important than others in terms of their impact on the PBS measures. Generally the suspensions have relatively little effect on the low speed performance measures but have a significant effect on the high speed dynamic performance. The two most important mechanical properties of the suspensions in this regard are roll stiffness and roll centre height (Fancher, Ervin et al. 1986). An additional parameter, damping, has a medium effect on rearward amplification.

Ideally suspension data should be provided by the suspension supplier but this is not always forthcoming. New Zealand has had an SRT requirement for most large heavy vehicles since 2002. In order to calculate the SRT precisely it is necessary to know the suspension roll stiffness and thus for many suspensions sold in New Zealand roll stiffness data is available. Roll centre heights are less well known but there are methods for determining these from the suspension geometry (UMTRI 1988).

Damping data is often not easy to obtain. The Australian road-friendly suspension provisions require qualifying suspensions to have at least 20% damping with at least half of this as viscous damping. Thus for suspensions that are approved as "road-friendly" in Australia it is possible to determine the minimum level of damping that they have.

In general, there is a trade-off between low speed manoeuvrability and high speed directional stability. For long combination vehicles the way to improve low speed turning performance is to add more articulation points, use shorter wheelbases and increase the hitch offsets. However, this makes the vehicle more active dynamically and worsens its high speed directional stability. One of the ways that vehicle designers have tried to address this issue is to use steerable axles and, for the best results, steerable axles that lock

up when not at low speed. A variation on this is the Wabco Opti-turn system which reduces the vertical load on the rear axle of a tridem group during low speed turns. With reduced vertical load the axle generates less cornering force and the axis of the axle group moves forward and shortens the effective wheelbase of the vehicle. Thus it has a similar effect to a castor steer axle but it does not actually steer.

Steerable axles can be actively or passively steered. Active steering based on mechanical linkages is often relatively straightforward to model. In some cases, such as logging jinkers, the steering action changes the geometry of the vehicle and this may not be easy to represent accurately with some modelling systems but workarounds are possible. There are now also active steering systems which are computer-controlled and use algorithms that may or may not be disclosed. Modelling vehicles fitted with these systems is potentially very difficult and may be impossible. This leaves us with the option of physical testing but, as we have seen, some of the proposed performance measures are very difficult to evaluate experimentally. The solution may be to use a hybrid approach where a computer simulation model that approximates the vehicle is built and physical testing is used to validate the reliability of the model. The model can then be used for the performance measures that are difficult to determine experimentally.

Castor steer axles in New Zealand are usually used at the rear of axle groups to improve the low speed turning performance of the vehicle and reduce the tyre scuffing forces while maintaining the load carrying capacity of the axle group. The prescriptive dimensional requirements in the VDAM Rule ignore castor steering axles when determining the position of the axis of an axle group. Effectively this is assuming that the castor steer axle has negligible centring force and thus generates no significant cornering forces. For low speed turning performance this is a desirable characteristic but for high speed dynamic performance it is not. When the quad axle group for semi-trailers was introduced into the VDAM Rule in 2002 there was a requirement that two of its axles should be castor-steering. Without significant centring force on these steering axles, all of the lateral force for this axle group had to be provided by the two remaining non-steering axles which were only carrying half of the axle group load. As we have seen from the discussion on tyres, the amount of cornering force that a tyre can generate depends on the vertical load and the slip angle. Thus with reduced vertical load, greater slip angles are required resulting in more off-tracking. With more centring force on the steering axles, these can provide some cornering force and reduce the amount of force required from the non-steering axles. To overcome the difficulties experienced with some quad-axle trailers, the VDAM Rule was amended; initially to allow one of the castor steer axles to be locked at higher speeds (above 30km/h) and subsequently to allow quad axle groups with only one castor steering axle.

The centring force characteristics of the castor steer axle impacts on both low speed turning and high speed dynamic performance of the vehicle. If the vehicle is close to the acceptability threshold for either of these aspects of performance, the magnitude of the centring force may be critical. Data of the centring force characteristics is not always readily available from the suspension suppliers and some effort may be needed to obtain this data.

SUMMARY

The purpose of this study is to formalise a performance-based standards (PBS) framework for New Zealand that reflects local operating conditions and requirements.

We began by reviewing the history and use of PBS for managing vehicle size and weight around the world. Although New Zealand was an early adopter of this technology and has used it extensively for informing regulation and policy as well as for assessing permit vehicles, the approach has mostly been rather ad hoc using a selection of the performance standards developed in Canada and more recently Australia with some local adaptations. Some formal PBS requirements have been introduced into regulation; most notably the minimum Static Rollover Threshold (SRT) requirement for most large heavy vehicles, which was introduced as part of the 2002 VDAM Rule. There are also specific performance requirements for dangerous goods tank wagons. However, there is no formally recognised comprehensive set of PBS that is specific to New Zealand conditions.

Australia led the world in developing an alternative compliance regime based on PBS. The process of developing this system commenced in around 2000 and the system was finally implemented in 2008. Thus it is underpinned by a substantial body of research work. The Australian PBS is rigorous and comprehensive but it has had some difficulties as is to be expected for a world-first implementation.

Given our proximity to Australia and our close economic ties it makes sense for the New Zealand PBS framework to be aligned with the Australian system as much as is practicable. However, we should also learn from the Australian experience and eliminate or modify aspects of their system that do not work well. Furthermore, the New Zealand road transport environment is quite different from that in most of Australia and thus the pass/fail criteria set for the Australian PBS system are not necessarily appropriate for New Zealand.

Underpinning the development of the Australian PBS system was a principle that all performance measures should be able to be evaluated by physical testing as well as by computer simulation. This influenced the selection of the performance standards that were included in the system. In addition there was an underlying philosophy that all vehicle requirements should be specified in terms of performance standards with no prescriptive limits at all.

In practice neither of these aims was completely achievable. A number of the performance measures are very difficult to evaluate by physical measurement because of the expense of the required instrumentation and difficulties in finding suitable test venues. Very few if any of the Australian PBS assessments that have been undertaken have used the physical testing option for any of the performance standards. Furthermore some prescriptive limits have continued to be applied to PBS vehicles, for example, height, width and length. The main reason for this is that these limits are imposed by the capacity of the infrastructure rather than by the vehicle's performance characteristics.

For the New Zealand PBS framework we took the view that some prescriptive limits are unavoidable and furthermore that some aspects of performance can be adequately and more reliably characterised by prescriptive limits. Because prescriptive limits are much simpler to assess than performance standards, using prescriptive limits where appropriate simplifies the PBS framework and reduces assessment costs.

Secondly we recognised the fact that most PBS assessments will be undertaken by computer simulation and thus, although it necessary to be able to validate the computer simulation software by physical measurements, it is not necessary to be able to evaluate every performance measure by physical measurement. Hence we did not eliminate performance standards from consideration simply because they would be difficult to assess by physical measurement.

Finally there are two vehicle performance issues that have caused the NZTA concern in recent years that are not adequately reflected in the Australian PBS standards and thus we have introduced two additional performance standards to address these. These issues both relate to off-tracking on highway curves. The first relates to lower speed highway curves where the vehicle off-tracks on the inboard side of the curve

and additional lane width is required to maintain clearances between vehicles travelling in opposite directions. The second relates to high speed highway curves where the vehicle off-tracks on the outboard side of the curve and again additional lane width is required. This issue is exacerbated when the vehicle is travelling at higher than the recommended speed for the curve.

The result of this a set of twelve prescriptive requirements and twelve performance-based standards. Three of the test manoeuvres are identical to those in the Australian system and these cover eight of the performance standards. Five of these eight performance standards use the same performance measures as the Australian system but two of them have different pass/fail criteria. The other three performance measures are more closely aligned to the original Canadian RTAC performance measures. Similarly the method for determining SRT is the Canadian RTAC method rather than the Australian PBS method. These two methods give similar results but the Canadian method is easier to simulate. The final three standards do not have a match in the Australian PBS system but two of them are similar to one of the Canadian RTAC measures and one is comparable to a European measure.

Overall the proposed New Zealand PBS framework is substantially based on the Australian system with some modifications that are designed to make it simpler and easier to use. It also incorporates some inputs from Canadian and European practice to address some specific issues that have been identified in New Zealand. The pass/fail criteria have also been adjusted to reflect New Zealand operating conditions which are generally more demanding than Australia's.

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APPENDIX

PBS Requirements for New Zealand

Introduction

This appendix presents a summary of the proposed PBS requirements for New Zealand outlining both the test procedures and the proposed pass/fail criteria. These PBS requirements will be used as a basis for the permitting of high productivity motor vehicles and other vehicles that fall outside of the specifications of the VDAM Rule. The mechanism by which this PBS framework will be applied is yet to be determined by the NZTA and is not within the scope of this document.

Prescriptive Requirements

The limitations of the infrastructure impose certain size and weight constraints on all vehicles. Over time these limits may be modified as the infrastructure is upgraded. However, because these constraints are imposed by the infrastructure rather than by vehicle performance, it is expected that these requirements would apply to PBS vehicles as well. These limits are:

- Vehicle width
- Vehicle height
- Vehicle overall length
- Axle weights and axle group weights
- Axle spacing
- Combined axle set weight and spacing limits
- Tyre size and pressure

All of these requirements are specified in the VDAM Rule and cannot be violated based on a PBS assessment. A partial exception applies to vehicle overall length where a higher limit (23m) has been used for high productivity motor vehicles compared to the standard legal vehicle limit. However, this higher limit is a prescriptive limit which cannot be overruled on the basis of a performance assessment. It could potentially be exceeded on a route approval where the route has been assessed for it capacity to cope with longer vehicles.

Drivetrain Requirements

The drivetrain performance requirements are specified in terms of prescriptive limits. These should deliver the desired level of performance.

The following parameters need to be known:

M = gross combination weight at which the vehicle will be operating (tonnes)

W_{drive} = total weight on the drive axle(s) (tonnes)

T_{clutch} = clutch engagement torque (Nm)

T_{peak} = peak engine output torque (Nm)

G_{low} = lowest gear ratio

G_{high} = highest gear ratio

D_{final} = final drive ratio or differential ratio

P_{peak} = peak engine output power (kW)

Ω_{peak} = engine speed at peak power (rpm)

R   = rolling radius of drive axle tyres (m)

Assuming that the transmission system has 95% efficiency, the requirements are:

$$W_{drive} \geq 0.25.M$$

This ensures that the drive axles can transmit the required level of tractive forces onto the road. It needs to be evaluated with the vehicle full loaded and empty. The assessor also needs to consider whether there are any critical partial load situations that can occur in normal operations.

$$T_{clutch} \geq \frac{1859.M.R}{G_{low}.D_{final}}$$

This is the startability criterion. It needs to be evaluated with the vehicle fully loaded.

$$T_{peak} \geq \frac{2169.M.R}{G_{low}.D_{final}}$$

This is a low speed gradeability criterion. It needs to be evaluated with the vehicle fully loaded.

$$P_{peak} \geq 75.2 + 4.59M$$

This is a high speed gradeability criterion. It needs to be evaluated with the vehicle fully loaded.

$$\frac{0.12.\Omega_{peak}.\pi R}{G_{high}.D_{final}} \geq 90$$

This criterion ensures that the gearing is high enough for the vehicle to exceed 90km/h. It is unlikely that this condition will ever be critical. It is independent of load.

These drivetrain requirements do not include an acceleration requirement as is the case in Australia. If the above power and torque requirements are met, the acceleration capability will be satisfactory unless the gear ratios are very poorly matched to the engine's torque characteristics.

Low Speed Turning Performance

The primary low speed turning test manoeuvre is the same one that is used in the Australian PBS system (National Transport Commission 2007). The specified path consists of straight tangent approaches to a 90° circular arc of 12.5m radius. The approaches to the turn must be of sufficient length to ensure that the vehicle is straight at the point when the turn commences and, at the conclusion of the turn, the vehicle

must travel far enough so that the maximum swept width has been achieved. The vehicle must traverse the path at a speed no greater than 5km/h with the outermost point of the forward most outside steered wheel following the specified path to within ±50mm. This test should be undertaken with the vehicle laden and unladen. It may be done by either computer simulation or by field-testing.

Swept Width

During this manoeuvre the paths of innermost and outermost points of the vehicle are traced. The distance between these two paths is the swept width and the maximum value of the swept width is the performance measure. This is illustrated in Figure A1 below.
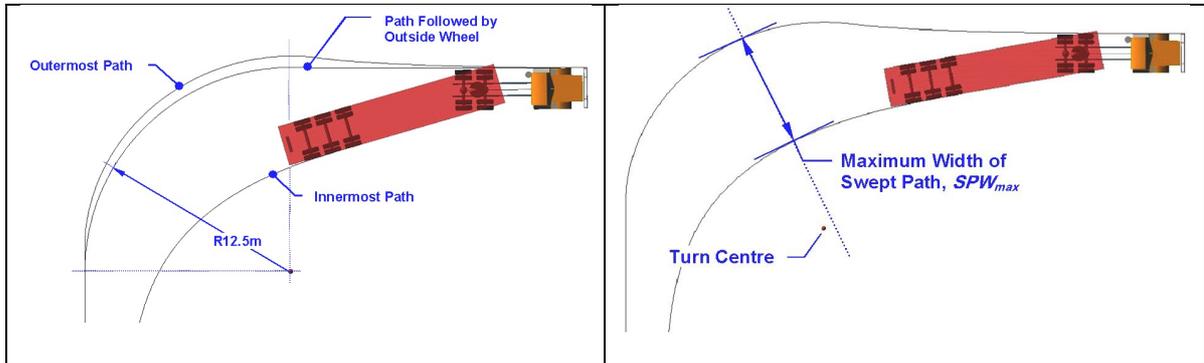
Figure A1. Illustration of the swept width performance measure reproduced from (National Transport Commission 2007)

The low speed swept width standard is:

Maximum width of swept path ≤ 6.95m

Tail Swing

If the vehicle has significant rear overhang, then at the start of the turn the outer rearmost corner can swing outboard of the original vehicle path. With a conventional vehicles this outswing will only occur at the start of the turn but when trailers have steerable axles it can also occur at the end of the turn. Both need to be checked. This measure is identical to that used in the Australian PBS system as illustrated in Figure A2.
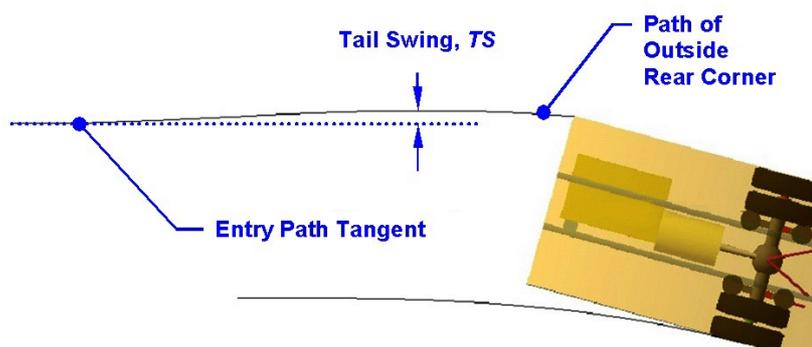

Figure A2. Tail swing as prescribed in the Australian PBS system.

The tail swing is defined as the maximum excursion of the rear outermost corner of the vehicle outside the entry and exit path tangent. The performance standard is:

Maximum tail swing ≤ 0.30m

Frontal Swing

The frontal swing behaviour is analogous to tail swing and is the result of front overhangs. It occurs at the end of the turn. Both the towing vehicle and the trailer(s) can exhibit frontal swing. This is illustrated in Figure A3. The frontal swing is the maximum excursion of the front outermost corners of the vehicle outside the exit path tangent. The performance standard for frontal swing is:

|	
Maximum frontal swing ≤ 0.75m	for trucks and trailers
Maximum frontal swing ≤ 1.50m	for buses
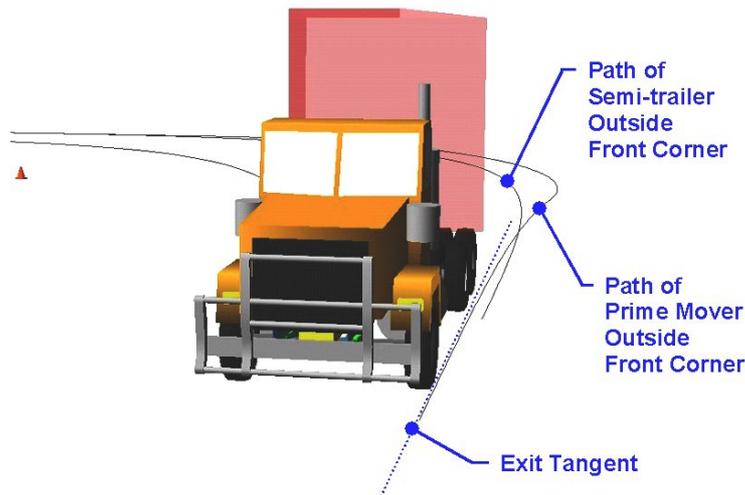
Figure A3. Frontal swing behaviour of a tractor semitrailer combination.

Steer-Tyre Friction Demand

The low speed turning behaviour is based on keeping the vehicle speed low enough that there is no significant lateral acceleration and thus vehicle tyres do not need to produce any net sideways force to counter this lateral acceleration. However, if the vehicle is fitted with axle groups, these axle will have a tendency to go straight ahead and thus the steer tyres have to generate the cornering forces needed to make these axle groups turn. If they cannot generate enough cornering force the vehicle will tend to "sledge" straight ahead particularly in situations where the road friction is reduced such as rain, snow or loose gravel. The steer-tyre friction demand measure quantifies the lateral force demand on the steer tyres as a proportion of the weight on the steer tyres. Effectively this the minimum level of tyre-road friction required for the vehicle to be able to steer. This measure uses the same 12.5m outside radius 90° turn as used for the previous three measures.

$$Steer\ tyre\ friction\ demand = \frac{\sum_{n=1}^{N} \sqrt{F_{xn}^2 + F_{yn}^2}}{\sum_{n=1}^{N} F_{zn}^2}$$

where: F_{xn} = longitudinal tyre force at nth tyre (N)

F_{yn} = lateral tyre force at nth tyre (N)

F_{zn} = vertical tyre force at nth tyre (N)

N = number of tyres on the steer axle or axle group

The performance standard for steer tyre friction demand is:

> Steer tyre friction demand ≤ 0.50

Steady State Low Speed Swept Width

The low speed swept width measure evaluated on the 90° turn does not represent the maximum low speed swept with for long combination vehicles because, with a 12.5m radius, these vehicles require more than 90° of turn angle to achieve maximum off-tracking. For normal intersection turns, the 90° turn is sufficiently representative to control this aspect of performance but for longer lower speed highway curves such as those marked with 25km/h or 35km/h advisory speed signs, the vehicles can achieve steady state off-tracking and thus it is important to also control this aspect of performance.

The test manoeuvre for this performance standard is a 25m radius wall-to-wall turn as described in (EC 1992). The turn should be sufficiently long that steady state off-tracking is achieved. The performance measure is swept width which is measured in exactly the same way as the swept width for the 90° turn above.

The steady state low speed swept width performance standard is:

> Maximum width of swept path ≤ 5.20m

Steady State High Speed Directional Performance

High Speed Steady State Off-tracking

During high steady speed turns vehicles off-track on the outboard side. That is, the path of the rear axles is outboard of the path of the front axles. This increases the road width required by the vehicle. Excessive high speed off-tracking can result in the rear outside tyres going off the edge of the road during the turn. The magnitude of this off-tracking depends on the vehicle configuration and loading, the curve radius and the vehicle speed. Because the tyre characteristics are not linear, we cannot simply scale the speed effect. Thus this performance standard assess the high speed off-tracking at two speeds on the same radius curve.

The test manoeuvre is a 319m radius turn of sufficient length to achieve steady state behaviour. The vehicle undertakes the manoeuvre at 90km/h and at 100km/h. These speeds generate lateral accelerations of 0.2g and 0.25g respectively. These lateral accelerations are below the level at which rollover will occur (see static rollover threshold requirements) but are both faster than recommended driving practice. For an 85km/h advisory speed curve undertaken at 85km/h, the lateral acceleration is 0.18g. The NZTA recommended practice for heavy vehicles is to go at 10km/h less than the advisory speed which would result in a lateral acceleration of 0.14g.

The high speed off-tracking is defined as the maximum distance between the path of the centre of the front steer axle and the path of the worst case rear axle. This may be the last axle but is sometimes the second to last axle.

The steady state high speed off-tracking standard is:

> High speed off-tracking at 90km/h ≤ 0.46m
>
> High speed off-tracking at 100km/h ≤ 0.68m

Static Rollover Threshold

The static rollover threshold is the maximum lateral acceleration that the vehicle can withstand before the onset of rollover. The onset of rollover is defined as the point where all of the axles except the steering

axles on one side of the vehicle have lifted off the ground. Currently most large heavy vehicles are required to achieve a static rollover threshold of 0.35g. In New Zealand this requirement is usually assessed using the SRT Calculator which is a web-based tool that can be accessed on-line. The SRT Calculator is applied to individual vehicle units rather than combination vehicles. This reflects the fact that the vehicle units are registered separately. For vehicles that are not roll-coupled, such as trucks and trailers, this method is appropriate because the vehicles' rollover initiation behaviour is largely independent of each other. Rollover will occur when the least stable unit in the combination rolls over. For roll-coupled vehicles, such as tractor-semitrailers and B-trains, this method is approximate as the more stable vehicle units will contribute to the stability of the less stable units and rollover does not occur until the whole combination is unstable. PBS assessments will apply to combination vehicles as a single entity and thus it is appropriate to assess the static rollover threshold for the whole vehicle combination rather than for its component vehicle units.

The manoeuvre used for this performance measure is the ramp steer manoeuvre used in the RTAC study (RTAC 1986). The vehicle speed is set to 100km/h with zero steer angle. A steering input is then applied which increases at a rate of 0.04 deg/s. This causes the vehicle to follow a spiralling path inwards. As the turn radius gradually decreases, the lateral acceleration steadily increases and eventually the point of wheel lift-off is reached. The lateral acceleration of the vehicle at that instant determines the static rollover threshold.

Combinations of units within a vehicle that are roll-coupled (i.e. connected by a turntable or a standard fifth wheel) will roll together while vehicle units that are not roll-coupled (i.e. connected by a pin coupling) will roll independently. Rollover is deemed to have occurred when any vehicle unit or group of units that can roll independently has achieved wheel lift-off. Note that for trucks and tractors, the steer axles do not have to lift-off in order for the onset of rollover to be deemed to have occurred.

When the rollover unit is a group of vehicle units, such as a B-train, each of the vehicle units will have a slightly different lateral acceleration at the point of wheel lift-off. To determine the static rollover threshold we use a weighted average of these lateral accelerations calculated as follows:

$$AY_{rcu} = \frac{\sum m_i h_i AY_i}{\sum m_i h_i}$$

where: AY_{rcu} = resultant lateral acceleration of the roll-coupled units (m/s^2)

m_i = sprung mass of vehicle unit i (kg)

h_i = height of sprung mass centre of gravity of vehicle unit i (m)

AY_i = lateral acceleration of sprung mass centre of gravity of vehicle unit i (m/s^2)

Static rollover threshold can also be determined by physical testing using a tilt table test. The requirements for undertaking a tilt table test are laid out in SAE Recommended Practice Guide J2180 (SAE 1998). A tilt table test must follow this procedure or some alternative international guidelines acceptable to the NZTA.

The static rollover threshold standard for acceptable performance is:

> Static Rollover Threshold ≥ 0.35g

High Speed Dynamic   Performance

Undertaking a high speed evasive manoeuvre with a combination vehicle generally results in an amplified response at the rear of the vehicle. This causes a load transfer from one side of the vehicle to the other which, if it is severe enough can cause rollover. Typically the path of the rear vehicle unit off-tracks outside the path of the towing vehicle.   The following three performance standards are designed to control this behaviour.

The manoeuvre used for these three performance standard is the high speed lane change manoeuvre defined in (ISO 2000). This is a 1.46m sinusoidal lane change executed in 2.5 seconds. The path is the path of the steer axle and it generates a peak lateral acceleration at the steer axle of 0.15g. The ISO standard specifies that the path should be followed to within 150mm which reflects the capability of a human driver when a physical test is undertaken. However, this level of path variation can produce significant variations in the results particularly for load transfer ratio. The Australian PBS system specifies a maximum path deviation of 30mm which is achievable for a computer simulation. This 30mm limit applies here.

Load Transfer Ratio

The load transfer ratio calculation is undertaken for each set of roll-coupled units within the vehicle. The load transfer ratio is defined as:

$$\text{Load Transfer Ratio} = \frac{\left|\sum\left(F_L - F_R\right)\right|}{\sum\left(F_L + F_R\right)}$$

where: F_L = vertical load on tyres on left side of vehicle (N)

F_R = vertical load on tyres on right side of vehicle (N)

The steer axle(s) is omitted from the computation due to its low roll stiffness and negligible influence on load transfer. Note that this performance measure is zero when the vehicle is stationary on level ground and rises to one when the vehicle is at the point of rollover.

The requirement for the load transfer ratio performance standard is:

Load transfer ratio ≤ 0.7,   if GCW ≤ 50 tonnes

Load transfer ratio ≤ 0.6,   if GCW > 50 tonnes

Rearward Amplification

During the high speed lane change manoeuvre, the maximum lateral acceleration of the steer axle is 0.15g. This effect is amplified for the trailing vehicles and thus the lateral acceleration of the centre of mass of the trailing units is usually greater than 0.15g. The rearward amplification is defined as:

$$Rearward\ amplification = \frac{max(|AY_{last\ vehicle\ unit}|)}{max(|AY_{steer\ axle}|)}$$

The performance standard for rearward amplification is:

Rearward amplification ≤ 2.0

High Speed Transient Off-tracking

At the completion of the lane change manoeuvre, the path of the rear trailer will tend to "overshoot" the path of the towing vehicle as illustrated in Figure A4. Clearly excessive "overshoot" is undesirable.
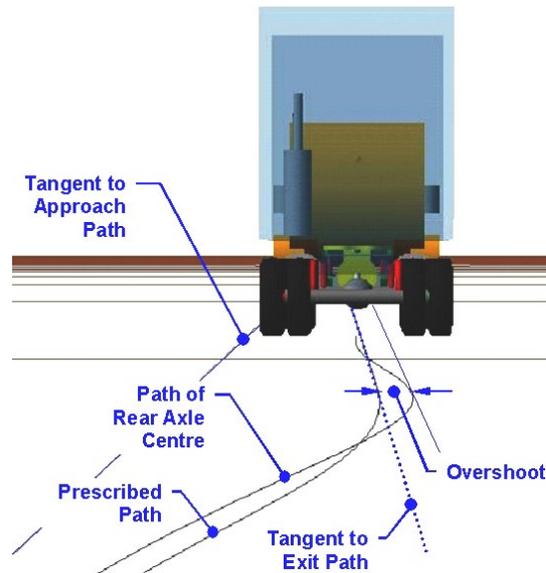
Figure A4. High speed transient off-tracking at the completion of the lane change (National Transport Commission 2007).

High speed transient off-tracking is defined as the maximum magnitude of the overshoot beyond the tangent to the exit path of the worst case rear axle. Usually this is the last axle of the vehicle.

The performance standard for high speed transient off-tracking is:

High speed transient off-tracking ≤ 0.6m

Yaw Damping Ratio

If a sudden sharp steering input is applied in a combination vehicle it will cause the trailers to oscillate from side to side. For safe operations these oscillations should decay rapidly. The yaw damping ratio is a measure of how quickly these oscillations decay.

A pulse steer test manoeuvre is defined in ISO standard 14791 (ISO 2000) but this is quite broad and allows considerable flexibility. (El-Gindy 1995) and Austroads (Prem, Ramsay et al. 2000) propose using a ±3.2° pulse steer input at the wheels with a period of 0.1s while travelling at 100km/h. These parameters are compatible with the requirements of ISO 14791 and thus we will specify this as the manoeuvre.

The procedure for calculating the yaw damping ratio is as follows:

The motion variable can be the articulation angle, or articulation angular velocity, between adjacent units, or the yaw rate of a unit, which gives the lowest damping of the vehicle combination. From the time history of the motion variable, all amplitudes starting with the first largest amplitude, A_1, after application of the specified steer input must be determined, as illustrated in Figure A5.
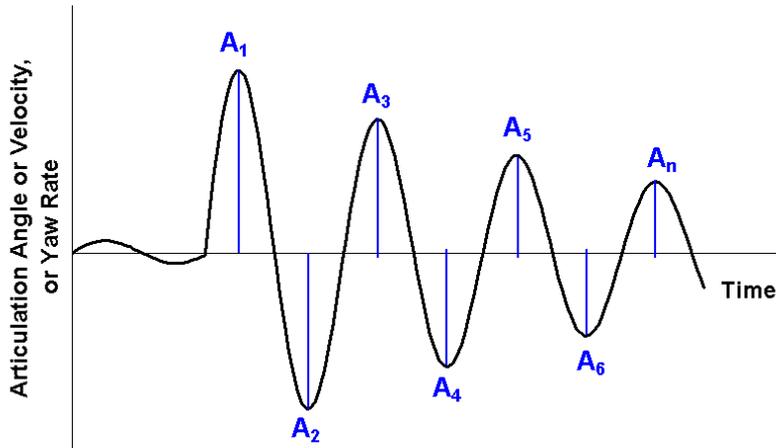
Figure A5. Determination of amplitudes for damping ratio calculation.

The mean value of the amplitude ratios, Ā, must be calculated separately for each articulation joint, or unit, using the following equation:

$$\bar{A} = \frac{1}{n-2}\left(\frac{A_1}{A_3} + \frac{A_2}{A_4} + \frac{A_3}{A_5} + \cdots. + \frac{A_{n-2}}{A_n}\right)$$

Amplitude A_n must be at least 5% of A_1 and the calculation of Ā must be based upon at least 6 amplitudes. The damping ratio, D, is calculated according to the following formula:

$$Yaw\ damping\ ratio = \frac{\ln(\bar{A})}{\sqrt{(2\pi)^2 + [\ln(\bar{A})]^2}}$$

If the 5% limit referred to above is reached before the 6th amplitude, then the following formulae may be used in place of the previous two equations:

$$\bar{A} = \frac{1}{n-1}\left(\frac{A_1}{A_2} + \frac{A_2}{A_3} + \frac{A_3}{A_4} + \cdots. + \frac{A_{n-1}}{A_n}\right)$$

$$Yaw\ damping\ ratio = \frac{\ln(\bar{A})}{\sqrt{(\pi)^2 + [\ln(\bar{A})]^2}}$$

The performance standard for yaw damping ratio is:

> Yaw damping ratio ≥ 0.15