THE PERFORMANCE BASED STANDARDS (PBS) REQUIREMENTS FOR HEAVY VEHICLES IN NEW ZEALAND (EFFECTIVE 22 MAY 2019)

INTRODUCTION

These new (2019) PBS requirements for New Zealand outline both the test procedures and the pass/fail criteria. These PBS requirements are used as a basis for the permitting of high productivity motor vehicles and other vehicles that fall outside of the specifications of the Land Transport Rule: Vehicle Dimensions and Mass 2016 (VDAM Rule).

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 and other Rules 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 has been shown to be the safe maximum for most routes, and 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 its 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)

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

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

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 fully 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.
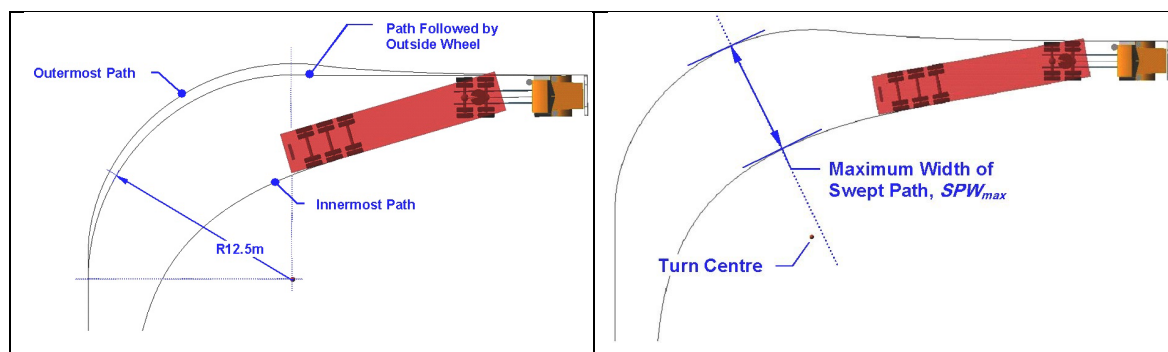

Figure A1. Illustration of the swept width performance measure reproduced from (National Transport Commission 2007)

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 vehicle 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.
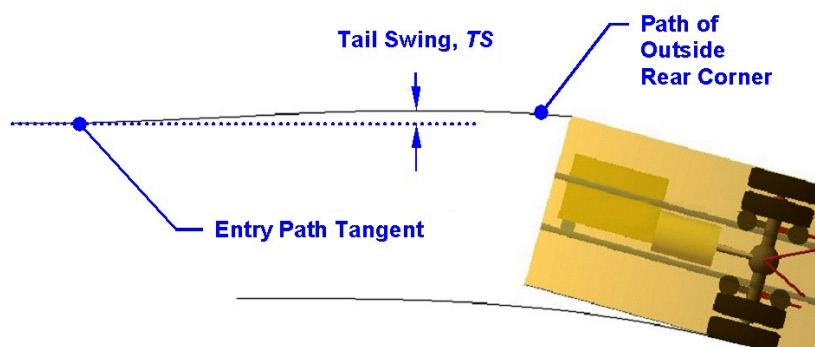

Figure A2. Tail swing as prescribed in the Australian PBS system.

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:

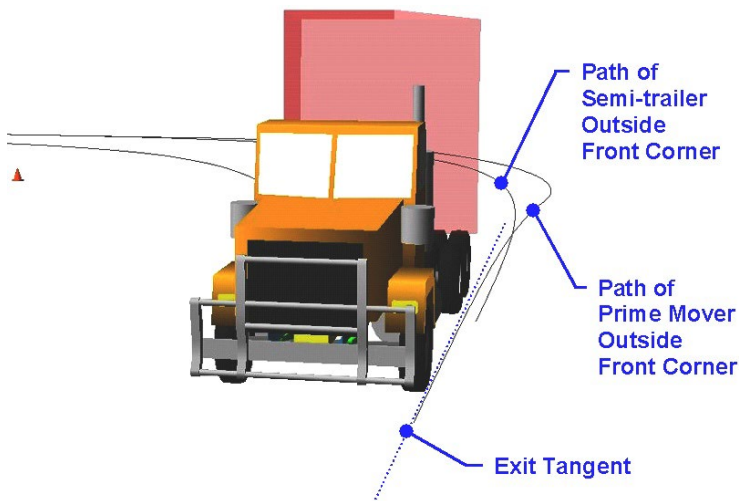| Maximum frontal swing ≤ 0.75m | for trucks and trailers |
| Maximum frontal swing ≤ 1.50m | for buses |

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 axles 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 is 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 width 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 online. 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:

$$\boxed{\text{Static Rollover Threshold} \geq 0.35g}$$

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 standards 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.
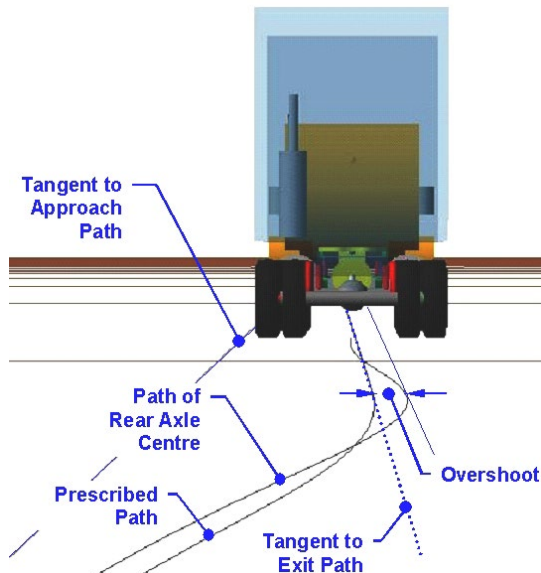

Figure A4. High speed transient off-tracking at the completion of the lane change (National Transport Commission 2007).

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.
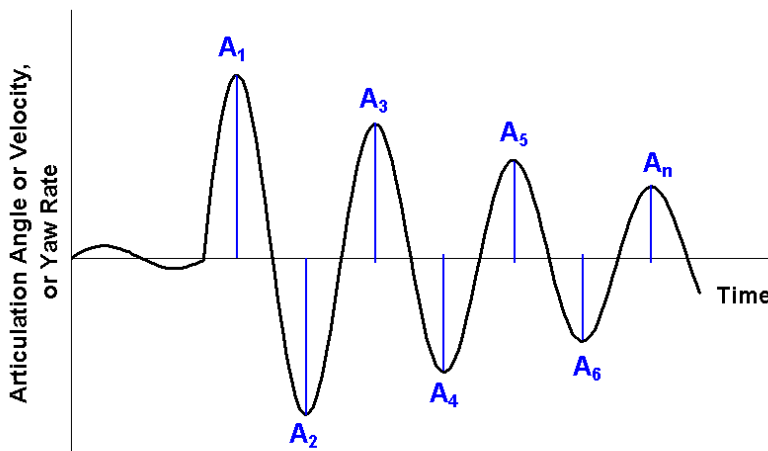

Figure A5. Determination of amplitudes for damping ratio calculation.

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

References

EC (1992). Directive 92/62/EEC adapting to technical progress Council Directive 70/311/EEC relating to steering equipment for motor vehicles and their trailers. 92/62/EEC. E. Parliament.

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