# Performance Based Analysis of Current South African Semi Trailer and B-Double Trailer Designs

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# Preface

I, Rhys Lloyd Thorogood, hereby declare that the whole of this dissertation is my own work and has not been submitted in part or in whole to any other University. Where use has been made of the work of others, it has been duly acknowledged in the text.

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(Signature of Candidate) Date

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(Signature of Supervisor) Date

# Abstract

South African heavy vehicles are currently designed according to prescriptive standards designed and enforced by the National Department of Transport (DoT); these standards are regulated in terms of mass, dimensions and vehicle configuration. However, the current prescriptive standards leave little room for innovation in terms of heavy vehicle design.

Performance Based Standards, or PBS, is a new Australian based innovative alternative to the current heavy vehicle prescriptive standards, mass, dimensions and vehicle configuration. PBS seeks to align actual vehicle performance efficiencies, productivity and safety objectives as well as road and bridge infrastructure to the current road network. Vehicle performance measures are based on engineering and science, supporting superior safety and known road and bridge wear performance criteria. PBS produces “a result orientated approach” to improved heavy vehicle operations and safety rather than a ‘one size fits all approach’ utilised by the current prescriptive legislation.

Currently, dynamic vehicle simulations are not carried out on South African manufactured vehicle combinations. Evidence exists that this has, in some cases, resulted in safety compromises. The computer dynamic vehicle simulation technology developed and validated could be employed for the credible assessments of the vehicle design concepts/prototypes for compliance with PBS. This service, which includes vehicle performance simulation and testing, development of high productivity vehicle concepts, assessment and development of risk management strategies, advice on safety and productivity issues, would have a substantial commercialisation potential for the implementation in the larger transport industry in South Africa.

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# List of Symbols

| | |
|---|---|
| $A$ | Amplitude |
| | Area |
| $\bar{A}$ | Mean amplitude ratio |
| $AY$ | Lateral acceleration |
| $A_L$ | Area with respect to load |
| $\lvert AY \rvert_{max}$ | Maximum absolute value of the lateral acceleration |
| $AY_{rcu}$ | Resultant lateral acceleration of the roll-coupled units |
| $A_V$ | Area with respect to volume |
| $C_\alpha$ | Tyre cornering stiffness |
| $C_f$ | Coulomb friction |
| $C_s$ | Suspension damping co-efficient |
| | Longitudinal stiffness parameter |
| $d$ | Lateral offset from ground |
| D | Damping ratio |
| $E$ | Modulus of elasticity |
| $F_b$ | Vertical force of sprung mass |
| $F_{env}$ | Force relating to upper and lower boundaries with in the envleope |
| $F_w$ | Tyre/wheel non-uniformity force on the unsprung mass |
| $F_x$ | Longitudinal force (longitudinal force) |
| $F_y$ | Lateral force (lateral force) |
| $F_z$ | Vertical force (vertical force) |
| $g$ | Gravity |
| $h$ | Height |
| $K$ | Spring constant |
| $K_e$ | Effective spring rate |
| $K_L$ | Spring rate of leaf spring |
| $K_P$ | Constant pressure spring rate |

| | |
|---|---|
| $K_s$ | Vertical stiffness of suspension |
| $K_t$ | Vertical stiffness of tyre |
| $k_x$ | Stiffness parameter |
| $L$ | Length |
| | Load |
| | Wheelbase |
| $m$ | Mass |
| $M_L$ | Moment around steer axis due to tyre lateral forces |
| $M_T$ | Moment around steer axis due to tyre tractive force |
| $M_v$ | Moment around steer axis due to tyre vertical force |
| $M_x$ | Over turning moment |
| $M_y$ | Rolling resistance moment |
| $M_z$ | Aligning torque |
| $P$ | Pressure |
| $P_{at}$ | Atmospheric pressure |
| $r$ | Tyre radius |
| $R$ | Radius |
| RA | Rearward amplification |
| $s$ | Slip |
| $SF$ | Stiffness factor |
| $t$ | Tread |
| | Track width |
| $V$ | Forward Velocity |
| $V_o$ | Internal volume of air spring |
| $w$ | Width |
| $x$ | Forward direction with respect to vehicle axis system |
| | Distance |
| $X$ | Forward direction of travel with respect to earth-fixed axis sysem |

| | |
|---|---|
| $y$ | Lateral direction with respect to vehicle axis system |
| $Y$ | Lateral direction of travel with respect to earth-fixed axis sysem |
| $z$ | Vertical direction with respect to vehicle axis system |
| $Z$ | Vertical direction of travel with respect to earth-fixed axis sysem |
| $Z_r$ | Road profile elevation |
| $Z_u$ | Vertical displacement of unsprung mass |
| | |
| $\alpha$ | Slip angle |
| $\delta$ | Deflection |
| $\delta_i$ | Steering angle for inner wheel in a turn |
| $\delta_o$ | Steering angle for outer wheel in a turn |
| $\delta_x$ | Longitudinal deflection |
| $\delta_y$ | Lateral deflection |
| $\Theta$ | Pitch angle |
| $\lambda$ | Lateral inclination angle of steer axis |
| $\mu$ | Co-efficient of friction |
| $\mu_{peak}$ | Peak co-efficient of friction |
| $v$ | Caster angle of steer axis |
| $\Phi$ | Roll angle |
| $\psi$ | Yaw angle |
| $\omega$ | Angular velocity |

# Chapter 1 - Introduction

This chapter provides an introduction into the current methods of heavy vehicle operations in the South African transportation industry, the introduction of Performance Based Standards (PBS) as well as its aims and benefits, an overview of international PBS operational methods used in Canada, New Zealand and Australia, as well as the aim of this research.

## 1.1 South African Transportation Industry

South Africa’s social and economic development is directly influenced by its current transportation system, as a large percentage of manufactured goods overheads are derived from transportation and logistical costs. The transportation industry spends billions of Rands annually, and thus the need to incorporate modern technological advancements in order to improve productivity, and as such reduce these overhead costs, is substantial.

Road and rail are the two most utilised modes of transportation in South Africa, and thus are often considered to be the ‘arteries which keep the life blood of the economy flowing’. Approximately 70 percent of the freight moved in South Africa is done on road, with an expected increased growth of approximately four percent per annum [1].

A major increase in the cost of rail transportation, a move by the rail industry towards the transportation of fewer commodities, and an increased need by companies to move towards a more time efficient door-to-door logistical approach, has forced many organisations to transition towards road transportation.

The number of heavy goods vehicles on the South African road network has doubled over the last 30 years, (1970 - 2000), in comparison to the 100 million km of national and provincial road network available, Figure 1.1. This number is expected to increase at a more substantial rate over the next 10-20 years, thus giving rise to increased road infrastructure damage, congestion and the number of heavy vehicle accidents.

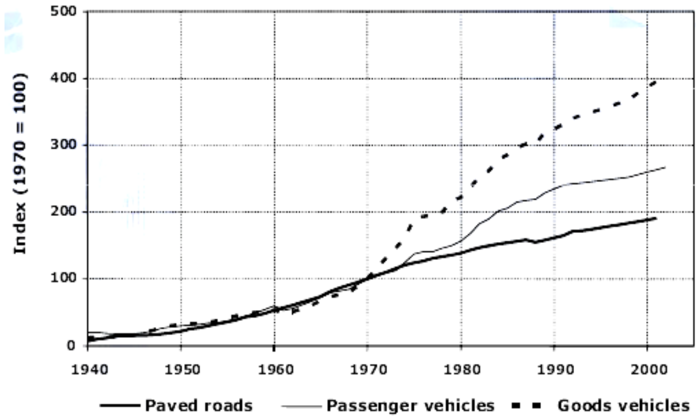


**Figure 1.1: The increase in number of Goods and Passenger vehicles in comparison to the increase of road network infrastructure [2]**

Figure 1.2 below illustrates the rapid increase in the number of heavy vehicle accidents on the South Africa road network over the last 70 years, with a dramatic increase since the mid 1970's, this is in direct relation to the increase in the number of heavy vehicles on the current South African road network.

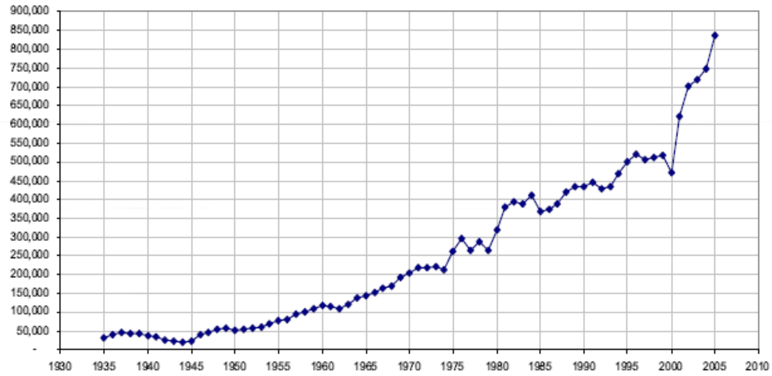


**Figure 1.2: History of road accident in South Africa [2]**

Figure 1.3 below illustrates the number of heavy vehicle fatalities, per 100 million kilometres travelled on South African roads. It is evident that South Africa has in the region of 300 – 500 percent greater number of heavy vehicle fatalities in comparison to that of other competitive countries. This is once again due to the increase in the number of vehicles, as well as the lack of safety performance of heavy vehicles on the South African road network.

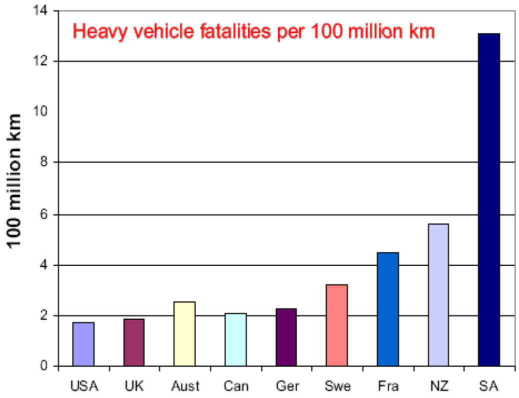


**Figure 1.3: Heavy vehicle fatalities per 100 million km [2]**

One of the major areas of concern for the South African transportation industry is heavy vehicle overloading and road safety. However, even with increased efforts by the Department of Transport and road traffic authorities; this is still an area of great concern. Overloading not only has a direct impact on the damage caused to the road network infrastructure, but also to the handling, stability and safety of heavy vehicles [3].

Thus the introduction of a new modernised regulatory system such as Performance Based Standards (PBS) in conjunction with higher management standards such as Road Traffic Management System (RTMS), aims not only reduce the damage to road network infrastructure, but also improve the safety and productivity of heavy vehicles on South African roads as well as promote long term sustainability.

## 1.2 Performance Based Standards

Traditionally heavy vehicles have been regulated by tightly defined prescriptive standards, which provides little scope for innovation and have no guarantee of good performance or dynamic stability. The introduction of Performance Based Standards provides an improved regulatory system that encourages innovation and provides a better match for heavy vehicles and the road network upon which they travel.

Performance Based Standards, or PBS, is a new innovative national alternative to the current heavy vehicle prescriptive standards which regulates the mass, dimensions and vehicle configuration of current heavy vehicles. PBS seeks to align actual vehicle performance efficiencies, productivity and safety objectives as well as road and bridge infrastructure to the current road network. Vehicle performance measures are based on engineering and science, supporting superior safety and known road and bridge wear performance criteria. PBS produces “a result orientated approach” to improve heavy vehicle operations and safety [4] rather than a ‘one size fits all approach’ [5] utilised by the current prescriptive legislation.

PBS is a voluntary initiative which operates in parallel with the current prescriptive legislation. A set of standards have been developed with which the vehicle must comply. An assessment of the heavy vehicles compliance towards these standards must be under-taken by certified vehicle assessors, either through numerical modelling or by means of physical testing.

PBS regulates the performance of a vehicle, how it is driven and operated, and the characteristics of the road network directly, rather than indirectly by limiting it with regard to dimensions mass, and vehicle configuration, thus creating more flexibility for innovative designs, increased productivity and improved safety, without compromising infrastructure impact.

PBS aims to improve productivity by reducing the number of vehicles on the road network and thus reducing the crash risk exposure rate, improve vehicle safety, reduce the wear and damage on the road network infrastructure and create an improved cohesion between the vehicles and the road network upon which they travel.

The benefits of which would result in the encouragement for innovative designs, an increase regulatory transparency, improved heavy vehicle safety, a reduction in vehicle down time and thus an increase in overall fleet productivity.

## 1.3 International Initiatives

The three major leaders in the Performance Based Standards scheme are Australia, New Zealand and Canada, this section provides a brief overview into the PBS operations utilised in each country.

### 1.3.1 Canada and United States of America

Performance Based Standards were developed in Canada in the late 1980's due to the fact that many of the heavy vehicle policies varied considerably with regard to weight and dimension between the various states. This therefore provided a great problem as vehicles would need to travel across various states in order to gain access to various coastal ports [6].

A comprehensive truck size and weight study was under taken and managed by the Roads and Transportation Association of Canada. This study was designed to determine what effects changes in size and weight limitations would have on the design, productivity, safety and vehicle performance characteristics of heavy vehicles on Canadian roads.

This study led to the development of a vehicle design envelope, within which various vehicle configurations and their relevant dimensions and weights were incorporated. This method of PBS gave the designers more room for innovation and flexibility. By projecting a number of size and weight scenarios, RTAC developed a basis from which the safety performance characteristics of vehicles can be easily assessed, when variations in size or mass transpire.

The average vehicle gross combination mass and vehicle length has increased from 40, 000 lbs (18,133kg) – 80,000 lbs (36,287 kg) and 30 ft (9m) – 48ft (15m) respectively, from 1950's to 1990's [7]. This posed significant impact on the road infrastructure, and therefore the need for road infrastructure impact analysis, bridge fatigue and construction costing was imperative.

Canada has also introduced the concept of a "special permit" where by a vehicle would be allowed to operate under PBS even though it did not fully comply with the PBS criteria. If the vehicle failed to comply with the requirement policies in place, this special permit could be revoked.

These requirement policies include:

- Routing specification
- Driver training and qualification
- Weather conditions and time of day restrictions
- Speed control
- Safety monitoring and evaluation
- Seasonal weight limitation

This created a feeling of trust instilled between the operator and the Department of Transportation; this therefore meant the operator would ensure that he/she closely followed the policies agreed upon with regard to maintenance, operation and safety measures.

The USA have not introduced the PBS approach as of yet, however, research and development into this new initiative are underway. University of Michigan Truck Research Institute (UMTRI) in conjunction with Mechanical Simulation Corporation (MSC) are currently looking at the development and simulation of various Australian PBS performance measures. The Federal Highway Administration is also looking into the practicality of implementing PBS standards in the near future.

### 1.3.2 New Zealand

The design and manufacture of heavy vehicles in New Zealand was largely regulated by prescriptive standards which regulated the dimensions, mass and configurations of heavy vehicles. Over the years the standards have been continuously altered and adjusted, these alterations in legislation lead to variations and inconsistencies between States and Territories [8], thus the need for a consistent national legislation was required.

In the mid 1990's various studies were under taken by the New Zealand government into the heavy vehicle accidents, these studies indicated an increase in the number of fatalities caused by heavy vehicle accidents, 18 - 21 percent from 1997 - 1998, [9].

Studies into the relationship between heavy vehicle stability performance and crash rates, (Mueller, du Pont and Baas, 1999), were conducted, the results of which clearly indicated that there is a direct relationship between Static Rollover Threshold (SRT), Dynamic Load Transfer Ratios (DLTR) and rollover crashes. Low performance results for SRT, and high DLTR results have a tendency to increase the likelihood of a rollover accident, Figures 1.4 and 1.5.

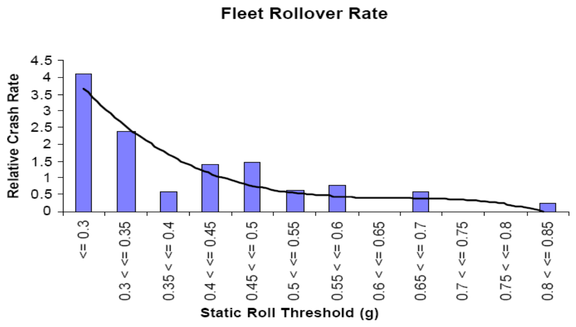


Figure 1.4: Relationship between SRT and relative crash rate [9]

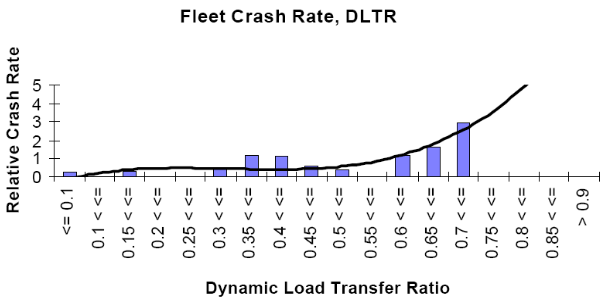


Figure 1.5: Relationship between DLTR and relative crash rate [9]

There has since been a growing interest in the development of an alternative legislative system of performance-based standards to regulate the design of heavy vehicles, with the aim of directly analysing the safety outcomes of vehicles.

Only permit vehicles with indivisible loads, which operate outside of the dimension and mass prescriptive limits, have been required to undertake stability performance assessments since the 1990's, as it was seen to be too costly to assess vehicles which conform to prescriptive standards [10].

In order to keep the cost of performance analysis conformity affordable the Land Transport Safety Authority (LTSA), who are a government agency responsible for maintaining the safety of road transport, in collaboration with Transport Engineering Research New Zealand Limited (TERNZ), have developed a simple algorithm to calculate the Static Rollover Threshold (SRT). This has led to the development of a free publically accessible internet analytical approach which allows users to calculate the SRT from easily accessible inputs, known as the SRT calculator.

In 2002 the LTSA introduced a new Vehicle Dimension and Mass Rule 41001 (LTSA, 2002), which is a world first, requiring most of the larger heavy vehicles operating on New Zealand roads to undergo a roll stability assessment, irrespective of special permit status. This rule requires vehicles, trucks of class NC (greater than 12 tons GVM) and trailers of class TD (greater than 10 tons GVM), to achieve a minimum Static Rollover Threshold (SRT) of 0.35g. Due to the development of the SRT calculator this new rule was seen as a feasible option [11].

New Zealand has, over time, also incorporated six other performance measures used to manage the stability and control of heavy vehicles, namely: Dynamic Load Transfer Ratio (DLTR), Rearward Amplification (RA), High Speed Transient Off-tracking (HSTO), High Speed Steady Off-tracking (HSO), Yaw Damping Ratio (YDR) and Low Speed Off-tracking (LSO). These performance measures may differ slightly, in comparison to those used by other countries, therefore care must be taken when developing computational models, as well as assessing vehicle performance results [10].

PBS has been introduced to a limited degree through codes of practice; these codes of practice are designed to manage the design, loading and performance of heavy vehicles in order to improve vehicle safety. Some of these codes of practice include: the design of drawbars, load securing, braking performance and the design of bolsters of timber vehicles. In conjunction with various initiatives namely: load height restrictions, driver education, speed control and safety monitoring.

The introduction of PBS, to a limited degree, in New Zealand has had a significant effect on the safety of heavy vehicles, introduction of safer heavy vehicle result in a reduction in heavy vehicle rollovers, as well as a positive effect on the country's economy, with a reduction in transportation costs, lower consumer costs, reduced fuel consumption, and safer road networks.

### 1.3.3 Australia

Prior to 1999 Australian heavy vehicles were regulated by stringent prescriptive standards, which regulated the mass and size limitation of various vehicle configurations. However, these prescriptive standards have continuously evolved over the years and as such seemed to differ between various States and Territories [12].

In 1999 the National Road Transport Commission (NRTC) and Austroads initiated a joint venture into the development of a new set of standards which would regulate the dynamic performance of vehicle capabilities, rather than regulate vehicle design according to mass, size and vehicle configuration. This new set of standards would provide a better cohesion between heavy vehicle and the road network upon which they operate.

NRTC has undertaken numerous studies into the rapid growth forecast of the road freight vehicles operating on the Australian road network, these studies have indicated that the number of heavy freight vehicles are to increase substantially over the next 5-10 years, thus indicating the drastic need for an improved regulatory system which would ensure improved vehicle safety, improved productivity and the development of a more sustainable transportation system.

An initial set of over 100 standards were proposed [12], these were then narrowed down to a set of 20 standards, 16 of which assess the dynamic performance and safety of vehicles, whilst the remaining 4 deal with the safety and preservation of the road infrastructure [13]. These 20 performance measure are listed in Table 1.1 below.

**Table 1.1: The 20 safety and infrastructure performance measures**

| | |
|---|---|
| **Safety performance measures** | |
| 1 | Startability |
| 2 | Gradeability |
| 3 | Acceleration Capability |
| 4 | Overtaking Provision |
| 5 | Tracking Ability on a Straight Path |
| 6 | Ride Quality |
| 7 | Low Speed Swept Path |
| 8 | Frontal Swing |
| 9 | Tail Swing |
| 10 | Steer Tyre Friction Demand |
| 11 | Static Rollover Threshold |
| 12 | Rearward Amplification |
| 13 | High Speed Transient Off-tracking |
| 14 | Yaw Damping |
| 15 | Handling Quality |
| 16 | Directional Stability Under Braking |
| **Infrastructure related performance measure** | |
| 17 | Pavement Vertical Loading |
| 18 | Pavement Horizontal Loading |
| 19 | Tyre Contact Pressure Distribution |
| 20 | Bridge Loading |

Under the prescriptive legislation general heavy vehicles had the ability to make use of the entire road network, irrespective of their mass, dimensions and vehicle configuration, whilst abnormal vehicles which operated under a special permit or exemptions, were required to travel on specified routes. This method has been incorporated for the purpose of the PBS initiative; a further study of the Australian road and highway network was conducted, classifying the routes into four major categories, known as Level 1 to Level 4, namely:

**Table 1.2: Four road classification levels and their respective access routes [14]**

| | |
|---|---|
| Level 1 | General Access |
| Level 2 | Restricted access – Major arterials and approved routes |
| Level 3 | Major freight routes and remote area combinations |
| Level 4 | Remote area designation for larger combinations |

From the 20 performance measures, in conjunction with the relevant four major road categories, a set of values, known as performance levels, were developed. In order for a vehicle to operate under the PBS scheme on a specific route, it must ensure that the vehicle fully complies with the performance levels laid down by the National Road Transport Commission.

Since the implementation of PBS, Australia has refined and developed each performance measure and performance level specific to the four road classification levels applicable, and as such are now considered the world leaders in the heavy vehicle performance-based approach.

## 1.4 South Africa

Currently, South African heavy vehicles are designed according to prescriptive standards designed and enforced by the National Department of Transport (DoT); these standards are regulated in terms of mass, dimensions and vehicle configuration. However, the current prescriptive standards leave little room for innovation in terms of heavy vehicle design.

With the rapid increase in modern technological advancements of vehicle safety and design, such as Electronic Braking Stability (EBS), Central Tyre Inflation (CTI), and active distance control, the transportation industry struggles to exploit the opportunities, as it is constantly hampered by the slow evolution of the current prescriptive standards, which leaves little room for innovative vehicle design, thus reducing the country's economic productivity and competitiveness. As such South Africa is looking to implement the PBS scheme in conjunction with the current prescriptive legislation.

The need for the introduction of a self-regulatory initiative, such as PBS, was first identified by the National Overload Control Strategy [3], which aims to limit the amount of overloading by heavy vehicle on South African roads. Thus another very important reason for the introduction of PBS in South Africa is due to the excessive amount of heavy vehicle overloading, and the resultant damage caused by these vehicles on the South African road infrastructure. All vehicles which aim to achieve PBS status must first be certified in accordance with the RTMS accreditation scheme.

The objective of the research is to develop a benchmark of current South African heavy vehicle configurations according to the Australian PBS initiative. This benchmark study would also form part of a legislative investigation into the introduction of Performance Based Standards (PBS) for heavy vehicles in South Africa.

Some of the main outcomes for the introduction of PBS in South Africa are to improve the current transportation productivity, improve the safety and stability of the vehicles on South African roads, and obtain a more sustainable transport system by limiting the number of heavy vehicles required to transport a specific amount of freight, thus reducing congestion, road damage and fewer vehicle on the road network resulting in the reduction of potential heavy vehicle accidents.

South Africa is currently running two PBS demonstration projects in the timber industry, both projects operating out of the KwaZulu-Natal region commenced operation in late 2007. The need for the introduction of a demonstration project came about from the need to gain practical experience in the design, manufacture and operation of various PBS projects as well as to determine the potential positive and negative productivity and safety outcomes of this initiative in the South African environment [15].

The demonstration projects are commissioned by Mondi Business Paper (Mondi) and Sappi Forests (Pty) Ltd (Sappi), each company having a single PBS vehicle in operation, a 24.0 m 64 100 kg GCM B-double vehicle, and a 27.0 m 67 500 kg GCM rigid draw-bar vehicle, respectively.

These demonstration projects have shown a dramatic positive improvement in comparison to the base line, 22.0 m and 56 000kg, vehicle. Both vehicles indicate an increase payload efficiency of approximately 18.5%, a fuel consumption saving of approximately 12.5 % and a fleet reduction size of 17 % [15]. These results demonstrate a very positive outlook to the introduction of a fully nationalised move towards the PBS scheme. However, the data collected from these vehicles was seen to be a small sample in order to accurately establish comparative performance results, therefore the introduction of a further 28 vehicles have been approved by the KwaZulu-Natal Department of Transport, in order to generate a larger PBS demonstration project in order to obtain a more substantial dataset.

The aim of my master's research project was to:

- Develop computer simulation models for each safety performance measure – Chapters 2,4 and 5
- Select vehicles which closely resemble the current SA fleet – Chapter 4
- Benchmark the current South African heavy vehicle fleet according to PBS standards – Chapter 5
- Validate these results through field testing or an analytical approach – Chapter 5
- Build up local expertise in South Africa

## 1.5 Road Transport Management System (RTMS)

RTMS is an industry-led, voluntary self-regulation scheme that encourages consignees, consignors and transport operators engaged in the road logistics value chain to implement a vehicle management system that preserves road infrastructure, improves road safety and increase the productivity of the logistics value chain [16].

RTMS, together with the Department of Transport's National Overload Control Strategy, was developed in order to ensure the integrity of the road transportation system, to ensure a fair competitive environment for all industry operators, and to encourage the responsible transportation of freight legally, through means of self-regulation.

This initiative is designed to regulate the standards on loading, driver wellness and training, vehicle operations and productivity [17], whilst also providing support to the various key operators and stakeholders. This initiative would lead to significant financial and time savings in the heavy vehicle industry, thus improving the logistical costs by optimising vehicle loading.

RTMS is a non-compulsory scheme and as such various incentives are emplace in order to encourage participation, namely: reducing in delays at weigh bridges and roadside checks thus reducing vehicle turnaround time and increasing productivity, receive support in order to obtain PBS accreditation, and the consideration of reduced toll fees and insurance costs is being undertaken

It is essential that all PBS members are RTMS accredited, thus ensuring that the specified vehicles are not overloaded, the drivers are well trained and the vehicles continuously obey road traffic legislation.

## 1.6 Conclusion

The increase in the number of heavy vehicle on the South African freight network, the dramatic increase in the number of heavy vehicle accidents in comparison to other competitive nations, and the increase in the number of heavy vehicle overloads, which has led to the increase in road damages, has forced the South African transportation industry to seek alternate methods for vehicle legislations.

PBS is an alternative legislative regulatory system, which allows vehicles to be designed according to their performance capabilities, rather than the prescriptive ‘one size fits all’ approach which regulates a vehicles design according to mass, dimension and vehicle configuration.

Canada were the first country to implement a performance based approach with regard to heavy vehicle design, and since then other countries such as New Zealand and Australia have also adopted the same approach. Australia are currently considered the world leaders in terms of heavy vehicle performance based standards, and have implemented a set of 20 performance measures, 16 of which regulate the safety aspects of the vehicle, whilst the remaining four standards are concerned with infrastructure damage. New Zealand has implemented PBS to a lesser extent.

South Africa are currently in the process of investigating the implementation PBS, two demonstration projects are under way in order to gain practical experience in the design, manufacture and operation of various PBS projects as well as to determine the potential positive and negative productivity and safety outcomes of this initiative in the South African environment.

Due to the fact that Australia is currently the world leader in terms of PBS, it was decided to utilise their current set of performance standards in order to analyse the current South African fleet, with the future intention of adjusting the performance measures and levels in order to accommodate for the South African conditions. The Chapter 2 provides an introduction in PBS as well as the relevant safety performance measures used to analyse the selected sample of South African heavy vehicle fleet.

The aim of this research is to develop computer simulation model for each of the concerned PBS manoeuvres, select vehicles which resemble the current South African heavy vehicle fleet, and assess them according to these standard in order to develop a benchmark for future PBS vehicles.

# Chapter 2 - Performance Based Standards

The Performance-Based Standards (PBS) scheme in Australia utilises 20 standards to assess a vehicles dynamic and safety performance, 16 of which are based on the safety and performance of the vehicles, while the remaining four standards deal with vehicle impact on the current road infrastructure. This chapter provides a brief overview of the Australian road classification guidelines, as well as an overview of 13 performance standards that were selected for analysis.

## 2.1 Road Classification

Australia's National Transport Commission underwent a large-scale operation in order to classify their current road network into four levels of network access, with the purpose of providing a match between the performance of a vehicle and the route on which it may operate. Level 1 allows for general access to the road network and thus requires more stringent performance standards, whilst levels 2, 3 and 4 are more lenient as they are intended for B-doubles (interlinks), double road train and triple road train configurations, respectively, as illustrated in Table 2.1 below. If a vehicle is deemed to comply with level 1 then is capable of travelling on roads of levels 2, 3 and 4 due to the hierarchy of the structure. Due to the fact that road trains are not permitted on South Africa roads, only levels 1 and 2 with their corresponding performance measures and required performance levels were tabulated.

**Table 2.1: Four road classification levels, their respective access routes, and general vehicle description**

| PBS Level | Route Access | Vehicle Description |
|---|---|---|
| Level 1 | General Access | General Access |
| Level 2 | Significant Freight Routes | B-doubles |
| Level 3 | Major Freight Routes | Double Road Trains |
| Level 4 | Remote Areas | Triple Road Trains |

## 2.2 Performance Based Standards

This section provides a brief description of 13 Australian performance standards developed by the National Transport Commission (NTC). The remaining three performance standards, overtaking provision, ride quality and handling quality, have not yet been fully developed, and as such have been excluded. This information was taken from NTC 'Performance Based Standards Scheme – The Standards and Vehicle Assessment Rules – July 2007' report [13]. Table 2.2 below provides a list of the 13 performance measures which were assessed, whilst a full description of the 16 performance measures can be found in Appendix A.

**Table 2.2: Summary of the 13 performance measures assessed**

| Safety Performance Measures | |
|---|---|
| **Low-speed Longitudinal Performance** | |
| 1 | Startability |
| 2 | Gradeability |
| 3 | Acceleration Capability |
| **High-speed Longitudinal Performance** | |
| 4 | Tracking Ability on a Straight Path |
| **Low-speed Direction Performance** | |
| 5 | Low Speed Swept Path |
| 6 | Frontal Swing |
| 7 | Tail Swing |
| 8 | Steer Tyre Friction Demand |
| **High-speed Directional Performance** | |
| 9 | Static Rollover Threshold |
| 10 | Rearward Amplification |
| 11 | High-Speed Transient Off-tracking |
| 12 | Yaw Damping Co-efficient |
| 13 | Direction Stability under Braking |

### 2.2.1 Low-speed Longitudinal Performance

This group of standards is used to determine the longitudinal performance of a vehicle at low speeds. This group of performance standards include: Startability, Gradeability and Acceleration Capability.

#### *2.2.1.1 Startability*

The startability performance measure is used to determine the maximum percentage grade[1] upon which a vehicle, operating at its maximum combination mass, has the ability to commence and

[1] Percentage grade is taken to be 100 times the change-in-height divided by the distance over which the height change occurs. For example a 5% grade would correspond to a grade line of 1:20.

maintain forward motion from a stand-still position. Steady forward motion is deemed to be achieved when a vehicle maintains a constant speed, or increases speed for a distance of five metres. This performance measure is designed to reduce the safety risk to other road users, ensuring that a vehicle has the ability to start on steep grades.

#### 2.2.1.2 Gradeability

Gradeability test is in place in order to ensure that vehicles have the ability to maintain a forward motion on specified grades, when loaded to maximum combination mass. There are two aspects to this performance measure, the first, is the ability for a vehicle to maintain a forward motion on a specified minimum grade, however, this differs from startability in that the vehicle has an initial forward speed.

The second aspect of gradeability is for a vehicle to maintain a specified minimum speed on grade of not less than 1%; an initial speed is once again acceptable.

Both startability and gradeability are capable of being tested by numerical modelling and field testing.

#### 2.2.1.3 Acceleration Capability

The Acceleration Capability performance measure is designed to determine the time taken for a vehicle, loaded to its maximum combination mass, to accelerate from rest, on a 0% grade, and travel a distance of 100m. The results of which would then be compared to the performance level specified in the PBS standards and vehicle assessment rules.

This performance measure posed various complications as numerous parameters pertaining to the engine performance characteristics were not available, due to company disclosure, and as such this performance measure was not computationally simulated. It is however recommended that this manoeuvre be assessed during field testing.

### 2.2.2 High-speed Longitudinal Performance

This group of standards is used to determine the longitudinal performance of a vehicle at high speed. This group of performance standards include Tracking Ability on a Straight Path (TASP).

#### *2.2.2.1 Tracking Ability on a Straight Path*

Tracking Ability on a Straight Path (TASP) is a performance measure designed to determine the total lateral swept width of a vehicle whilst travelling down a straight road. When a vehicle, loaded to least favourable load condition, travels down a straight road at a speed not less than 90 km/h with a specified road surface unevenness and cross fall, the rear of the vehicle experiences dynamic lateral movement, this movement is then recorded and compared to the performance level specified in the PBS requirements.

This ensures that the vehicle does not track outside its specified lane width, thus ensuring the safety of other vehicles on the road and reducing the damage of the road infrastructure.

The road profile used in the computational modelling and simulating process was developed from the work undertaken by Hans Prem for Austroads, and is supplied to the assessors such that all assessments are undertaken under the same external disturbances, thus ensuring uniform conditions.

The road pavement test section must be at least 1000 metres long and the surface must have an overall unevenness level in each wheel path of not less than 3.8m/km IRI (International Roughness Index). The unevenness level in each wheel path reported every 100 m must be not less than 3.0 m/km. The entire test section must have an average crossfall, falling to the left when viewed in the direction of travel, of not less than 3.0%. The average crossfall must have a crossfall deviation of not less than 1.0%.

The Figures 2.1 and 2.2 below illustrate the lateral sway of the rear of the vehicle, as well as the maximum swept width.

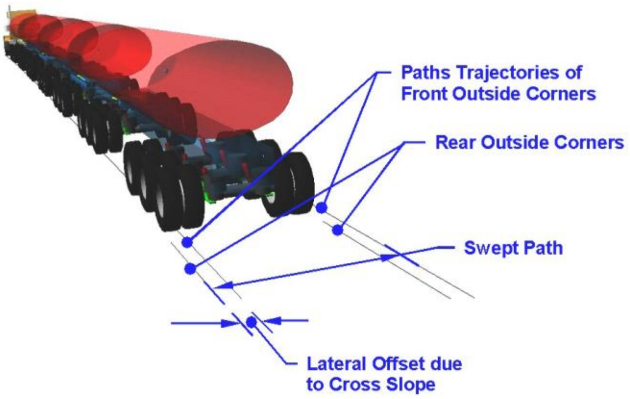


**Figure 2.1: Illustration of the path trajectories of the front and rear outside corners, and the swept path [13]**

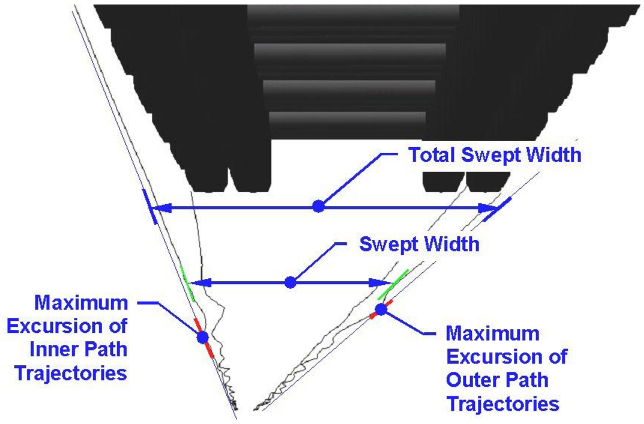


**Figure 2.2: Underside view of the vehicle illustrating the maximum excursions of the inner and out path trajectories, and the total swept width [13]**

### 2.2.3 Low-speed Directional Performance

This group of standards is used to determine the directional performance of a vehicle during cornering at low speeds. The standard is generic for measuring various low-speed directional outputs, such as Low-Speed Swept Path (LSSP), Frontal Swing (FS), Tail Swing (TS) and Steer Tyre Friction Demand (STFD).

The vehicle being tested has to follow a prescribed path of a 90 degree turn, of radius 12.5 m, at a speed no greater than 5 km/h as illustrated in Figure 2.3. The vehicle must be tested under both maximum laden and unladen mass conditions.

#### *2.2.3.1 Low-Speed Swept Path*

Low-Speed Swept Path (LSSP) is measured in order to limit the safety risk imposed by vehicles during cornering at low speeds. When a vehicle makes a low-speed turn the rear of the vehicle does not follow the path taken by the front of the vehicle but rather tracks inside this path.

A high value of LSSP is undesirable as the vehicle will require more road space to perform a low-speed turn, thus may result in collisions with oncoming traffic users, or damage to roadside objects. The maximum width of the swept path, $SPW_{max}$, is the maximum distance measured between the two path trajectories, perpendicular to their respective tangents (see Figure 2.4).

The two path trajectories of concern are, the outermost path scribed in the ground plane by the vertical projection of the furthest forward of outside point, or points, on the vehicle on the outside of the turn; and the innermost path scribed in the ground plane by the vertical projection of the points, or points, on the vehicle on the inside of the turn.

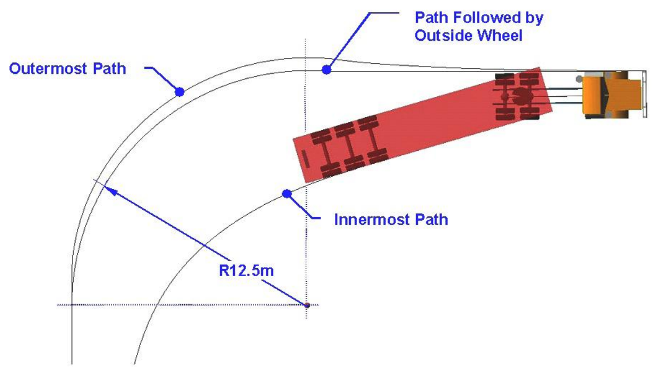


**Figure 2.3: Illustration of low speed swept path [13]**

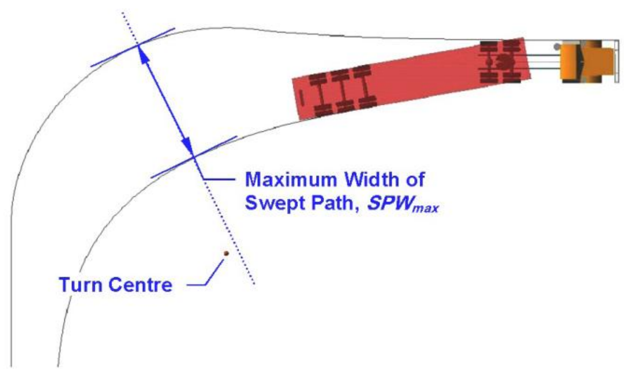


**Figure 2.4: Illustration of maximum width of swept path $SPW_{max}$ [13]**

#### *2.2.3.2 Frontal Swing*

Frontal swing is measured in order to minimise the safety risk of a vehicle when performing a tight turn at low-speed. During the low speed turn the front overhang of the hauling unit will cause the front outside corner to track outside the intended path to be followed by the front wheel. A large amount of frontal swing is undesirable as the vehicle will therefore require more road space to perform a low speed turn, thus encroaching into other lanes, endangering pedestrians or colliding with roadside objects (see Figure 2.5).

There are three parts to frontal swing, Part A deals with the prime mover, whilst Part B and Part C are concerned with the trailing units.

Part A determines the maximum width of the frontal swing that the prime mover requires when performing low-speed turn. The maximum distance, $FS_{max}$, is the straight line segment intersecting both trajectories perpendicularly to their respective tangents, at the intersecting points. The swept path must be defined by the path trajectories of:

(a) The outer most path scribed in the ground plane by the vertical projection of the furthest forward or outside point, or points, on the vehicle on the outside of the turn; and

(b) The path scribed in the ground plane of the outer most point on the outer tyre side wall nearest to the ground, on the forward most outside steered wheel, Figure 2.8.

Part B, Maximum of Difference (MoD), is the maximum difference between the swing-out of adjacent vehicle units when performing a low-speed turn. The difference between the frontal swing-out distances must be determined from the path trajectories of the outermost path scribed in the ground plane by the vertical projection of the furthest forward or outside point, or points, on each of the two adjacent vehicle units, one of which is a semi-trailer.

Frontal swing MoD is the maximum value of the straight-line segment intersecting both trajectories perpendicular to the low-speed turn exit tangent, Figure 2.6.

Part C, Difference of Maxima (DoM), is the difference between the maximum frontal swing-out distances between adjacent vehicle units when performing a low-speed turn. The difference between the maximum values of frontal swing-out distances must be determined from the path trajectories of the outermost paths scribed in the ground plane by the vertical projection of the furthest forward or outside point, or, points on each of the adjacent vehicle units, one of which is a semi-trailer.

Frontal Swing DoM is the maximum value of difference between the tangents, parallel to the exit tangent, of the two path trajectories, Figure 2.6.

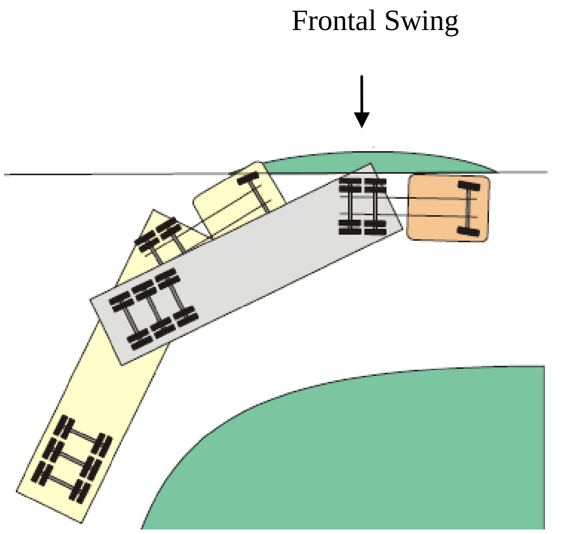


Figure 2.5: Illustration of frontal swing Part A [4]

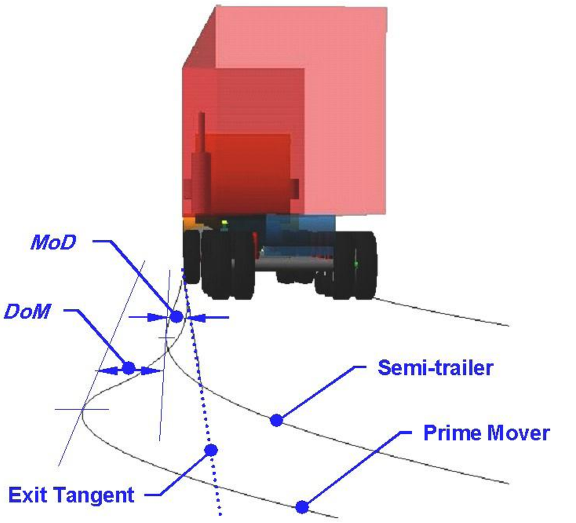


Figure 2.6: Illustration of Part B and C of frontal swing, indicating the Maximum of Difference (MoD) and the Difference of Maxima (DoM) [13]

#### *2.2.3.3 Tail Swing*

Tail swing is of great importance to vehicles with a large amount of rear overhang performing tight turns, at low-speed. A high value of tail swing is undesirable as the vehicle would therefore pose a severe safety risk to other road users by tracking into adjacent lanes, resulting in collisions. Two measurements are of interest during this procedure, $TS_{entry}$ and $TS_{exit}$ measured at the initial and final stages of the low-speed turn, respectively.

Tail swing must be determined from the path trajectory of the outermost path scribed in the ground plane by the vertical projection of the furthest rearward or outside point, or points, on the vehicle unit having the greatest tail swing.

On the entry side of the turn, tail swing is the length of the longest line segment perpendicular to the low-speed turn entry tangent intersecting it and the path trajectory, Figure 2.7.

On the exit side of the turn, tail swing is the length of the longest line segment perpendicular to the low-speed turn exit tangent intersecting it and the path trajectory.

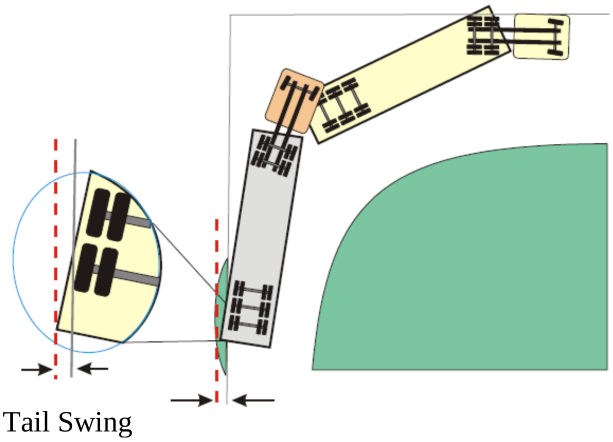


Figure 2.7: Illustration of Tail Swing [4]

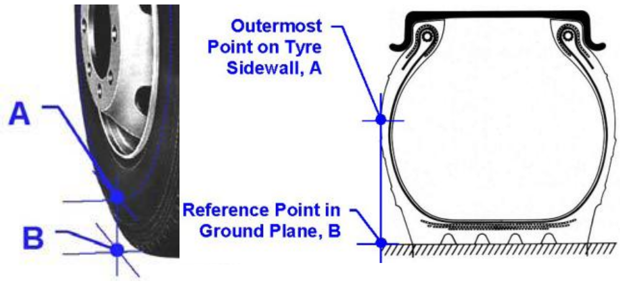


Figure 2.8: Illustration of outside wheel reference point [13]

During the simulation various points of interest on the vehicle were tracked and their trajectories were then plotted. These plots allow one to measure the required lateral displacements; the values are then compared to the corresponding PBS performance level to determine if they meet the required performance level.

#### *2.2.3.4 Steer Tyre Friction Demand*

The Steer Tyre Friction Demand (STFD) performance measure is used to limit the loss of steering control when a vehicle performs a tight turn at low-speeds. A loss of steering control will result in a vehicle exhibiting large amounts of understeer, resulting in the vehicle continuing straight ahead. This lack of steering would drastically increase the chances of a vehicle collision with other motorists as well as roadside objects.

The loss of steering occurs when a vehicle, operating at a maximum combination mass as well as unladen, performs a low speed turn and the available tyre/road friction limit at the steer-tyre is exceeded.

The friction demand required by the steer-tyre is calculated using the following formula:

$$Friction\ Demand(\%) = 100\left(\frac{Friction\ required}{Friction\ avaliable}\right)$$

$$= \frac{100 \left| \frac{\sum_{n=1}^{N} \sqrt{F_{xn}^2 + F_{yn}^2}}{\sum_{n=1}^{N} F_{zn}} \right|}{\mu_{peak}}$$

**Equation 1**

Where:

$F_{xn}$ = Longitudinal tyre force at *n*th tyre (N)

$F_{yn}$ = Lateral tyre force at *n*th tyre (N)

$F_{zn}$ = Vertical tyre force at *n*th tyre (N)

$N$ = Number of tyres on the steer axle or axle group

$\mu_{peak}$ = Peak value of prevailing tyre/ road friction

STFD is generally only of concern for multi-combination vehicles, and is not of great concern to vehicles with single or dual drive axles. However, it was decided to simulate this performance manoeuvre in order to gain experience for future analysis purposes.

### 2.2.4 High-speed Directional Performance

This group of standards is used to determine the directional performance of a vehicle at high-speeds. This group of performance measures include: Static Rollover Threshold (SRT), Rearward Amplification (RA), High Speed Transient Off-tracking (HSTO), and Yaw Damping.

#### *2.2.4.1 Static Rollover Threshold*

Static Rollover Threshold (SRT) is a measure of the lateral acceleration a vehicle can withstand without rolling over during a constant radius turn, or on a tilt table test. It can also be defined as a decrease in lateral acceleration with an increase in roll angle. The aim of this performance measure is to limit the likelihood of a rollover of a vehicle when it performs a steady state turn at high-speed.

When a vehicle travelling at high-speed enters a steady turn it is subjected to an outward lateral acceleration, which could result in the vehicle rolling over. High values of SRT are desirable as it is an indication of increased resistance to rollover. SRT is expressed as a fraction of acceleration due to gravity in units of ‘g’, where 1 g represents an acceleration of 9.807 $m/s^2$ corresponding to the force exerted by the earth’s gravitational field.

SRT is arguably the most important performance standard in terms of vehicle stability, as it has been strongly linked to crashes involving rollovers.

All vehicles being tested must be loaded to maximum combination mass and least favourable load condition, the maximum steady state lateral acceleration a vehicle can withstand without rolling over, must be recorded and compared to the required performance level.

There are two test procedures used to determine the SRT for vehicles, namely: a constant radius turn and a tilt table test (Figures 2.9 and 2.10, respectively). In order to get a good insight into the dynamic stability of the vehicle configurations, both procedures were simulated.

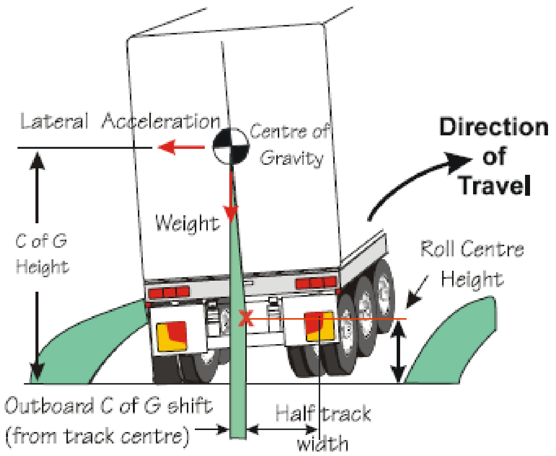


Figure 2.9: Illustration of SRT circular test [4]


Figure 2.10: Illustration of SRT tilt table test [18]

The rollover stability for multi-combination vehicles is much more complex than for single rigid units and depends on the type of coupling between trailers. Trailers that are connected through a turntable are said to be 'roll-coupled' and will rollover together as connected units.

For single unit vehicles, such as rigid trucks, the rollover threshold is the lateral acceleration of the sprung mass centre of gravity measured at the point of rollover instability. For multi-combination vehicles, the rollover threshold is the resultant lateral acceleration, $AY_{rcu}$, of any roll-coupled set of units.

For the purpose of the two trailer roll-coupled rear unit illustrated in Figure 2.11, the resultant lateral acceleration can be calculated by the following formula:

$$AY_{rcu} = \frac{\sum_{n=1}^{N} m_n h_n AY_n}{\sum_{n=1}^{N} m_n h_n}$$

Equation 2

Where:

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

$m_{1,2}$ = Semi-trailer sprung mass $(kg)$

$h_{1,2}$ = Height of sprung mass centre of gravity $(m)$

$AY_{1,2}$ = Lateral acceleration of sprung mass centre of gravity $(m/s^2)$

$N$ = Number of roll-coupled rear units

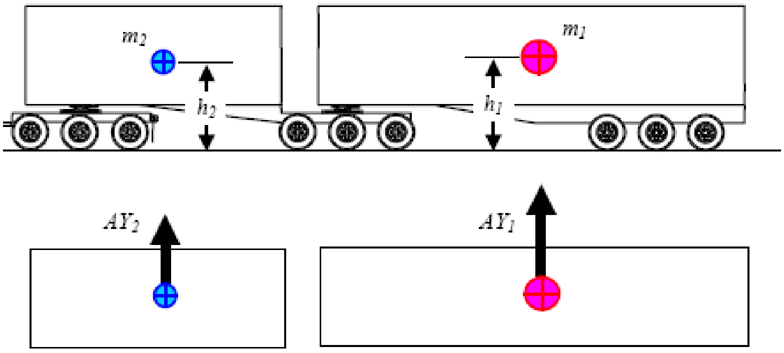


**Figure 2.11: Side and plan view illustration of 2 roll-coupled trailers showing sprung masses, sprung mass centre of gravity heights, and lateral accelerations [13]**

#### *2.2.4.2 Rearward Amplification*

Rearward Amplification (RA) is a performance measure that is designed to limit the lateral directional response of a vehicle performing an avoidance manoeuvre at high-speeds. This performance measure is more of a concern for vehicles with two or more articulation points.

As the name suggests, the lateral acceleration of each unit is an amplification of the unit directly ahead of it. Thus the rear unit in the vehicle combination will experience the highest level of lateral acceleration (Figure 2.12), which could result in rollover; the required performance level for this manoeuvre is therefore directly related to Static Rollover Threshold. This value must not exceed 5.7 times the SRT value for that particular vehicle.

The vehicle being assessed must be loaded to the permissible maximum combination mass and least favourable load conditions, and must perform a single lane change manoeuvre in accordance with ‘Single Sine-Wave Lateral Acceleration Input’ specified in International Standards Organisation (ISO) documentation [20].

RA is calculated by the ratio of the maximum lateral acceleration response of the rear most unit, measured at the centre of mass, to the lateral acceleration of the input, measured at the front steer axle.

$$RA = \frac{|AY|_{max} of\ the\ following\ vehicle\ unit}{|AY|_{max} of\ the\ first\ vehicle\ unit}$$

**Equation 3**

Where:

$|AY|_{max} of\ the\ following\ unit$ = maximum absolute value of the lateral acceleration of the centre of mass of the last vehicle unit $(m/s^2)$

$|AY|_{max} of\ the\ first\ vehicle\ unit$ = maximum absolute value of the lateral acceleration of the centre of the front axle $(m/s^2)$

As with SRT, the resultant lateral acceleration of roll coupled-units, $AY_{rcu}$, in multi combination vehicles is calculated from Equation 2.

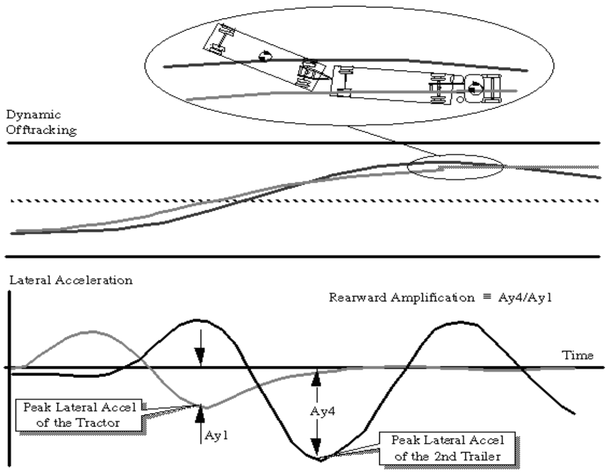


**Figure 2.12: Illustration of Rearward Amplification [22]**

### *2.2.4.3 High-Speed Transient Off-tracking*

High-Speed Transient Off-tracking (HSTO) is a performance measure used to limit the lateral displacement of the rearmost trailer of an articulated vehicle, whilst performing an avoidance manoeuvre at high-speeds.

When a vehicle, loaded to it maximum allowable mass and least favourable load condition, performs an avoidance manoeuvre the rear end of the rearmost trailer may overshoot the final path of the front steer axle; this measure of lateral overshoot is referred to as HSTO. The avoidance manoeuvre that is to be performed is based on the same ISO lane change manoeuvre described above in 'Rearward Amplification' above.

HSTO is determined by measuring the maximum lateral displacement of the centre rearmost axle of the rearmost vehicle unit from the exit tangent of the desired path, Figure 2.13. A HSTO overshoot is represented as a positive value, whilst an undershoot is represented as a negative value. Figure 2.14 below illustrates an overshoot as well as an undershoot scenario.

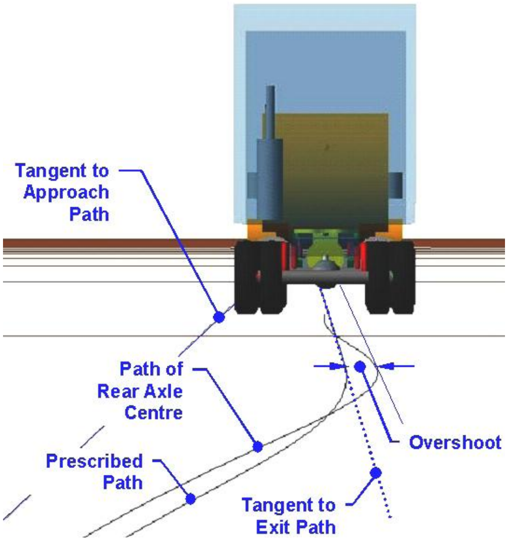


Figure 2.13: Illustration of a HSTO overshoot, indicating the desired path and the path of the centre rear axle [13]

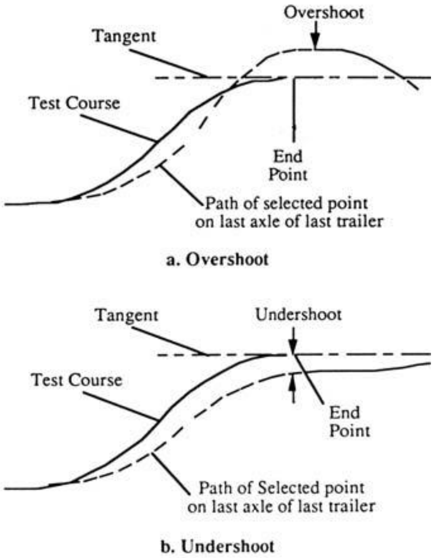


Figure 2.14: An illustration of the HSTO overshoot and undershoot scenario [13]

### 2.2.4.4 Yaw Damping Co-efficient

This performance measure is designed to reduce the yaw oscillations a vehicle experiences when performing a high-speed manoeuvre. When a vehicle, loaded to its maximum combination mass and least favourable load condition, performs a high speed manoeuvre the rear of the vehicle experiences sway or 'yaw' oscillations. The rate at which these oscillations settle down is known as Yaw Damping.

The performance manoeuvre that the vehicle is required to perform is a short duration steer input, which is in accordance with 'Pulse Input' specified in International Standards Organisation (ISO) documentation [21].

The damping ratio, D, must be calculated from the vehicles yaw rate and then compared to the required performance level specified in the PBS standards guidelines. The damping ratio is calculated using the following formula:

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

Equation 4

Where:

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

Equation 5

As can be seen from Figure 2.15 the amplitudes $A_{1 \to n}$ must be calculated in order to determine the mean value of the amplitude ratios, $\bar{A}$, and thus determine the damping ratio, D. The amplitude $A_n$ must be at least 5% of $A_1$, whilst $\bar{A}$ must be based upon at least 6 amplitudes.

However, if the 5% limit is reached before the 6$^{th}$ amplitude then the formula may be slightly modified, and then the new formula may be used to determine the damping ratio. This new formula can be found in 'Australian Road Transport Suppliers Association. PBS Explained – Performance Based Vehicles for Road Transport Vehicles (2003), page 65' [4].

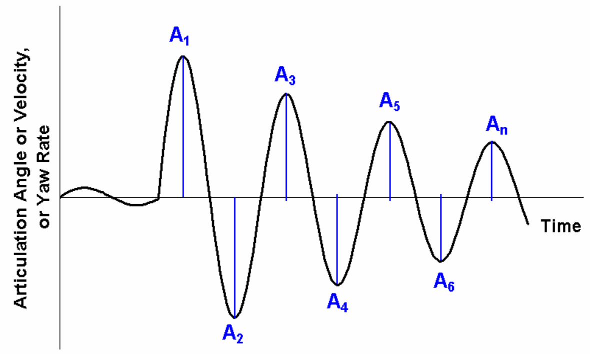


Figure 2.15: Time history graph of the yaw rate amplitudes [16]

### 2.2.4.5 *Directional Stability under Braking*

The direction stability under braking performance measure is designed to ensure the stability of a vehicle under heavy braking. When a unladen vehicle travelling at 60km/h applies the brakes it must certify that the vehicle does not experience gross wheel lock up and must ensure that the vehicle does not track outside the specified lane width.

Heavy vehicles experience complex combinations of forces during braking which place severe demand on both driver skill and vehicle performance, thus high levels of stability are desirable as it reducing the risk of rollover or loss of control.

The performance measure determines the average deceleration of a vehicle, from the vehicles initial speed and the stopping distance achieved. This average deceleration is determined by using the following formula:

$$Average\ deceleration = \frac{\frac{1}{2}(inital\ speed\ in\frac{m}{s})^2}{(g\ \times stopping\ distance\ in\ m)}$$

**Equation 6**

Where:

$$g = 9.81\ m/s^2$$

This average deceleration is then calculated and compared to the performance level, specified in the PBS guideline, according to the specific vehicle class to determine is the vehicle meets the required performance level. Unlike the previous performance measures, where the vehicle is measured according to the road classification guidelines, this performance measure is measured according to vehicle class. Low levels of average deceleration indicate poor performance.

If the vehicle passes the performance measure in its unladen condition, it is therefore considered to fulfil it in its laden condition.

**Table 2.3: List of the 13 PBS measures and their corresponding performance levels, according to road classification levels 1 and 2**

| **Safety Standard** | | **Performance Level** | |
|---|---|---|---|
| | | **Level 1** | **Level 2** |
| Startability | At least | 15% | 12% |
| Gradeability | | | |
| Part A - maintain forward motion | At least | 20% | 15% |
| Part B - maintain minimum speed | At least | 80 km/h | 70 km/h |
| Acceleration Capability | No greater than | 20 sec | 23 sec |
| Tracking Ability on a Straight Path | No greater than | 2.9 m | 3.0 m |
| Low-Speed Swept Path | No greater than | 7.4 m | 8.7 m |
| Frontal Swing | | | |
| Part A - Prime Mover | | No greater than 0.7 m | |
| Part B - Trailing unit (MoD) | | No greater than 0.4 m | |
| Part C - Trailing unit (DoM) | | No greater than 0.2 m | |
| Tail Swing | No greater than | 0.3 m | 0.35 m |
| Steer Tyre Friction Demand | | Not greater than 80% max. available tyre/road friction limit | |
| Static Rollover Threshold | | Not less than 0.35 g | |
| Rearward Amplification | | No greater than 5.7 X SRT | |
| High-Speed Transient off Tracking | No greater than | 0.6 m | 0.8 m |
| Yaw Damping Co-efficient | | Not less than 0.15 at the certified vehicle speed | |
| Safety Standard | | Performance Level | |
| | | Semi Trailers | B-double |
| Directional Stability Under Braking | Avg. Deceleration | 0.35 g | 0.3 g |

## 2.3 Conclusion

This section provided a short description of the four Australian road classification levels, their route access and a general description of the vehicles which used them.

This section also provided an overview of the 13 Australian safety performance measures, developed by the National Transport Commission (NTC), that were computationally analysed. The remaining three safety performance measures, Overtaking provision, Ride quality and Handling quality were left out as these have not yet been fully developed.

Table 2.3 above, provides a summary of the 13 performance manoeuvres and their relevant performances measures according to road classification Levels 1 and 2, which closely resemble heavy freight vehicles utilised on the South African road network.

A fully detailed description of the 16 performance measures can be found in Appendix A.

Chapter 3 below provides a short introduction into the various software packages that were utilised during the analysis, as well as a short introduction into the theory of dynamic vehicle mechanics, tyres, suspensions and steering systems, which were used in development of the vehicle dynamics simulation package, Trucksim.

# Chapter 3 - Theory

The following section provides an overview of the software packages that were used during the modelling and simulation sections of this research, as well as an introduction into the vehicle dynamics, the axis systems utilised during the modelling process, and a short theory section of various components and subsystems.

## 3.1 Software Analysis

### 3.1.1 Hellberg Transport Management – HTM

Hellberg Transport Management (HTM) is a company which has over the past 30 years dedicated itself to the development of a computer based software package which is designed to assist vehicle end-users in South Africa by providing a consultation service to the South African transportation industry. This software package, Transolve, aims to simplify the process of selecting the correct vehicle for a specific application, as well as calculating the costs incurred through the running of the vehicle.

Transolve consists of various software modules, namely:

- Loading
- Routing
- Performance
- Finance
- Maintenance
- Costing Specification
- Reference

This software allows one to optimise the vehicle configuration design, determine the maximum legal payloads for each vehicle unit, determine the operating costs of the vehicle, determine various finance options, manage vehicle performance, as well as generate a comparison between different vehicles in the same vehicle class.

HTM is currently utilised in excess of 90% of the South African commercial vehicle manufacturer’s market and have over 750 software installations country wide.

Transolve software was utilised in order to determine the maximum gross combination mass of each specific vehicle configuration, the tare mass of each vehicle unit, as well as the maximum legal payload of each vehicle unit. The software was also used to generate eight of the ten vehicle configurations that were utilised in the PBS computer simulation analysis.

### 3.1.2 MANEX and MANCAS

Manex and Mancas are in-house software packages developed for MAN Truck and Bus Company, these software packages were obtained from MAN South Africa, and were utilised in order to determine various mass and parametric data, as well as various drive-line performance characteristics of the prime movers.

### 3.1.3 Trucksim

Trucksim is a sophisticated vehicle dynamic simulation software package, which allows the user to model, simulate and analyse the dynamic behaviour of various truck-trailer configurations. It makes use of a primary Graphical User Interface (GUI), which allows the user to select various vehicle configurations, input controls and parameter settings and analyse the results through an Engineering Plotter as well as a post-processing animation feature.

Trucksim is a commercially available software package that is based on over 40 years of research and development through experimental testing and specialised laboratory analysis. The modelling assumptions used in the Trucksim math models were developed and validated at the University of Michigan Transportation Research Institute (UMTRI).

Autosim is a multibody computer simulation program that was developed in the 1980’s, this generic program was then validated and replaced many of the software simulation programs run at UMTRI, as well as being licensed to various other organisations and universities.

The development of Mechanical Simulation Corporation lead to the continuous development Autosim, and all of the mathematical models have since been utilised in the development of Carsim, Bikesim and Trucksim.

Trucksim was utilised throughout the PBS analysis to model each vehicle configuration, develop the required PBS performance manoeuvres, simulate and analyse the output data of each run.

### 3.1.4 Alternate Software Packages

Additional multi-bodied simulation software packages, such as MSC ADAMS/Car (Automated Dynamic Analysis of Mechanical Systems), SIMPACK and DADS (Dynamic Analysis and Design System) were investigated, in order to determine which software package was best suited for the needs of this research.

Numerous factors, such as financial limitations, software capability, training, assistance and data recourses etc. led to these alternate software packages not being incorporated during the modelling process of this research.

## 3.2 Theory

This section of the report was sourced from work presented at ‘Mechanics of Heavy Duty Truck Systems’ conference in Cape town April 2008, presented by Chris Winkler, Tom Gillespie and Richard Radlinski, as well as from additional work published by Tom Gillespie [22], [23], [24], [25] and [26].

### 3.2.1 Vehicle Dynamics

Vehicles dynamics is the study of the interaction between vehicle motion and the road surface. Each vehicle is made up of various components and subsystems, the break down and analysis of the subsystems allows one to determine the forces acting on the components and how these forces interact, in order to fully understand the dynamics of a vehicle.

Two methods are used in order to accomplish the understanding of vehicle dynamics modelling, namely: empirical and analytical approach [24]. Empirical relates to a trial and error approach [24], which has been learnt through past experience, whilst the analytical approach requires one to understand the mechanics of the interaction between subsystems through the use of basic laws of physics [24], in order to develop equations and analytical models.

However, neither method is fool-proof, due to incorrect assumptions made during the modelling process, as well as other analytical errors etc. Therefore many computational systems and engineers make use of a combination of both methods when assessing the dynamics of a vehicle.

### 3.2.2 Vehicle Axis System

Numerous multibody modelling systems make use of the Society of Automotive Engineers (SAE) conventions, earth-fixed axis system and the vehicle axis system, in order to accurately describe the position and orientation of vehicle units. The earth-fixed axis system (X, Y, Z) is represented by a right-hand orthogonal axis system which is fixed to the earth. The trajectory of the vehicle is described with respect to this earth-fixed axis system. The X- and Y- axes are in the horizontal plane and the Z-axis is directed downward.

The vehicle axis system (x, y, z) is represented by a right-hand orthogonal axis system which is fixed in the vehicle and located at the vehicle centre of gravity; this axis system is illustrated in Figure 3.1 below. A vehicle travelling steadily in a straight line on a level road, the x-axis is substantially horizontal, points forward, and is in the longitudinal plane of symmetry, the transverse y-axis points to the driver’s right-hand side and the vertical z-axis points downward, as illustrated in Figure 3.1 [22].

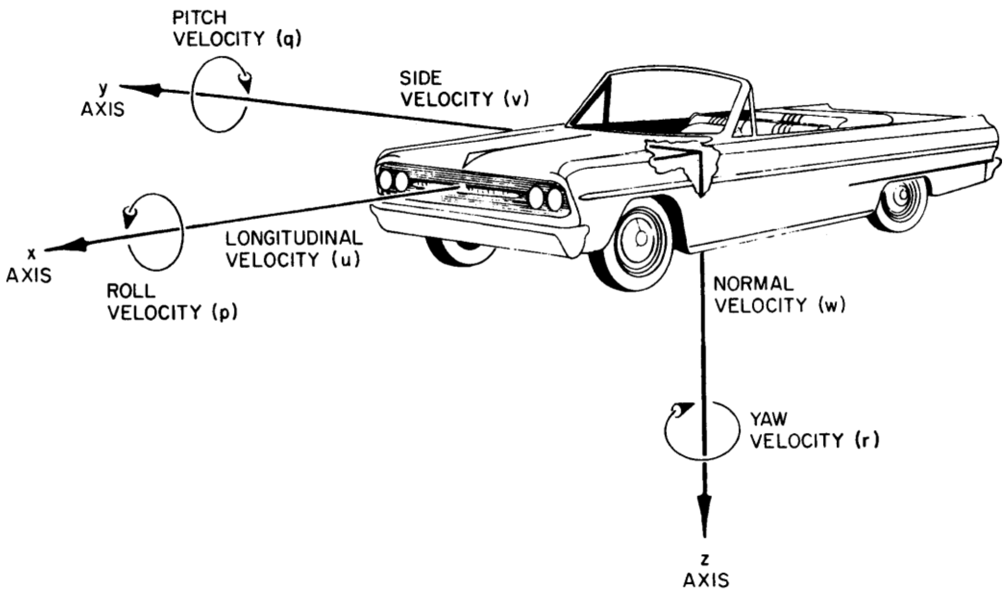


**Figure 3.1: SAE Vehicle Axis System [22]**

The orientation of the vehicle axis system (x, y, z) with respect to the earth-fixed axis system (X, Y, Z) is given by a sequence of three angular rotations, starting from a condition when the two sets of axes are initially aligned [22]:

a) A yaw rotation, ψ, about the aligned z- and Z-axis
b) A pitch rotation, Θ, about the vehicle y-axis
c) A roll rotation, Φ, about the vehicle x-axis.

### 3.2.3 Tyre Axis System

The tyre axis system utilised throughout the modelling process was sourced from the proposed SAE 'Vehicle Dynamics Terminology' [22] and is illustrated in Figure 3.2. The origin of the tyre axis system is located at the 'centre of the tyre contact' patch with the road surface, a point determined by the vertical projection of the spin axis of the wheel onto the road plane. The X'-axis is the intersection of the wheel plane and the road plane with the positive direction forward, the Z'-axis is perpendicular to the road plane with a positive direction downward, and the Y'-axis is in the road plane, its direction being chosen to make the axis system orthogonal and right-hand.

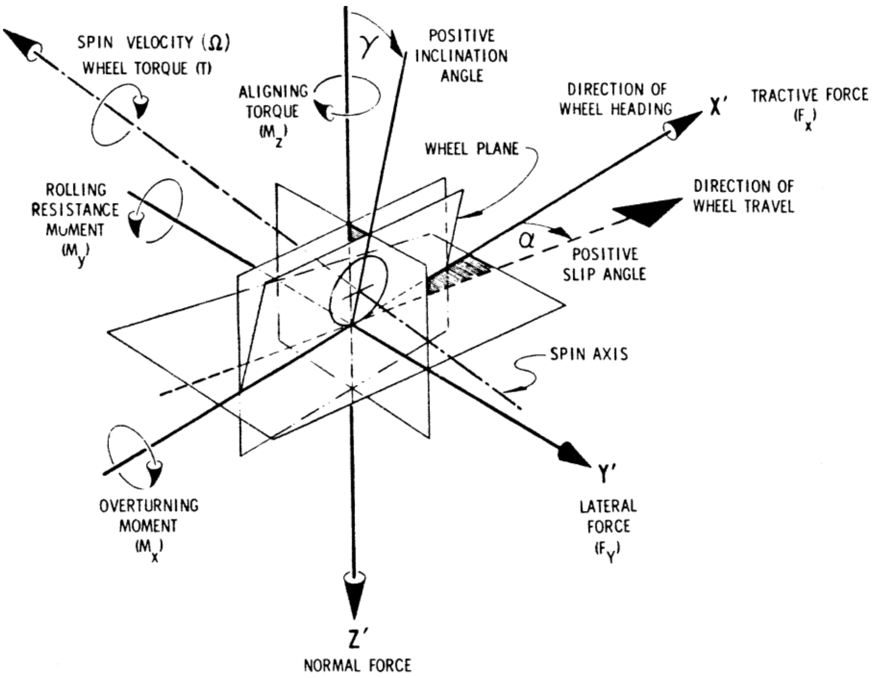


Figure 3.2: SAE Tyre Axis System [22]

### 3.2.4 Vehicle Systems

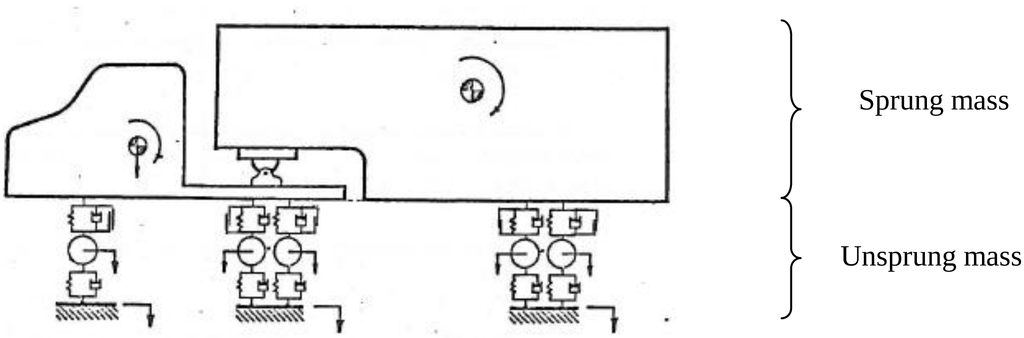


Figure 3.3: Vehicle system representation [23]

The heavy vehicle model is made up of various components and subsystems (suspension, tyres, axles, masses etc.), the interaction between these components and subsystems are developed and are resembled in the mathematical model. The vehicle body may be represented in two ways depending on the application, and simulation outcomes desired, namely: a single lumped mass located at its centre of gravity (CG), with mass and inertia properties [24], or as two separate systems, sprung mass (vehicle body), and unsprung mass, (isolating the wheels, axles and suspension), each of which having a mass and inertia properties. The latter of which is illustrated in Figure 3.3 above.

The unsprung mass can be represented as dynamic system, the most basic level is to assume it consists of a sprung mass, which supported by a suspension system, spring and damper. The wheel, tyre and axle are then further represented as the unsprung mass, which is in turn support by the tyre spring stiffness. This representation is illustrated in Figure 3.4 below.

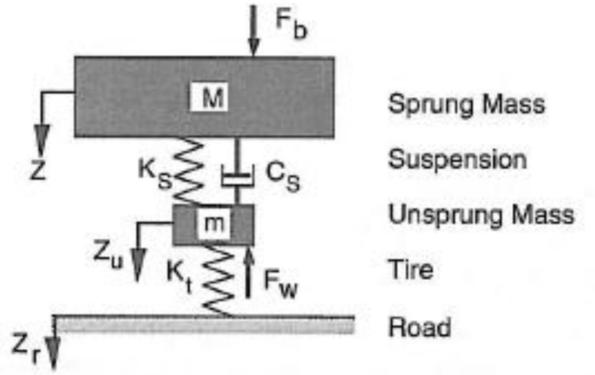


**Figure 3.4: Representation of the interaction between the sprung and unsprung mass models [24]**

The following sections discuss the tyre properties and modelling process, suspension systems as well as the steering systems.

### 3.2.5 Tyre modelling

It is often stated that the forces acting on a vehicle are developed through four, hand sized, patches where the tyre contacts the road surface, in order to understand the dynamics of a vehicle one must have knowledge of the forces and moments generated by the tyres at the road surface interface [24].

The handling characteristics and directional response of vehicles are strongly influenced by the forces and moments generated from the contact patch between tyre and road surface interface. These forces and moments generated strongly influence the acceleration, braking and handling capability of the vehicle.

In order to accurately generate a vehicle tyre model, the tyre force and moment characteristics must first be obtained, either through estimation or physical testing by means of experimental tests. Traditional means of obtaining tyre data was through the use of dynamometers and other tyre measuring systems in order to measure the forces and moments, through various camber, slip angles, and vertical load inputs.

Most recently mathematical functions have been developed in order to generate best fit equations which accurately model the tyre characteristics. These mathematical equations have then been used to develop mathematical tyre models.

Two factors that must just be mentioned prior to the discussion of the development of tyre forces and moments are slip and slip angle.

When a vehicle brakes or accelerates the longitudinal forces acting on the tyre either act to slow the down the wheels rotational speed, ω, or speed it up, respectively, with regard to the road surface. Therefore slip, s, is defined as the ratio of slip velocity at tyre contact patch (forward velocity, V, – tyre circumferential speed, $\omega r$) to forward velocity, V, as illustrated in Equation 7. These velocities are illustrated in Figure 3.5 below.

$$s = \frac{(V - \omega r)}{V}$$

**Equation 7**

When a vehicle is subjected to lateral tyre forces it is possible for the wheel to have a velocity vector which does not coincide with the direction of travel. This variation between direction of tyre heading and direction of travel is known as slip angle, $\alpha$. This is illustrated in Figures 3.2 and 3.9.

### *a) Longitudinal force properties*

The development of a method used to represent the longitudinal tyre force properties is essential in order to assess braking and acceleration studies. An analysis of a rotating tyre model was developed in order to model the deflection and shear force characteristics from which the tyre parameters could be established.

The tyre tread is imagined to consist of small elongated rubber segmented elements, which under baking and acceleration deform to develop longitudinal forces, these longitudinal forces vary with the slip. Each thread element is assumed to be deflected by a determinable amount at each point in the contact area.

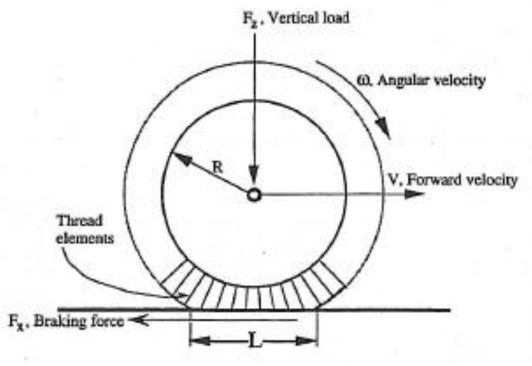


**Figure 3.5: Sketch of an idealised tyre [23]**

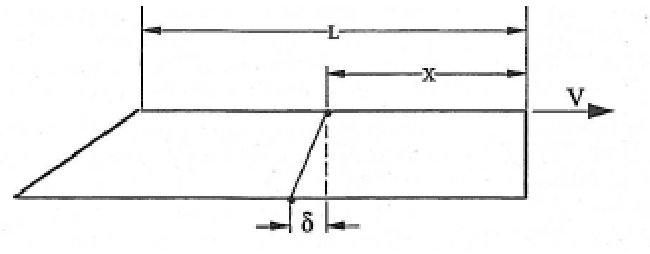


**Figure 3.6: Longitudinal deflection, $\delta$, of tread element at location x in the contact patch [23]**

From Figures 3.5 and 3.6, it can be seen that for a tread element at distance x, $\Delta t_x$, from the front of the contact patch, a deflection, $\delta$, of the element may be determined from the longitudinal slip, s. Deflection of the element at point x is given by:

$$\delta(x) = (V - R\omega)\Delta t_x$$

Where $x = R\omega\Delta t_x$

$$\frac{\delta(x)}{x} = \frac{(V - R\omega)\Delta t_x}{R\omega\Delta t_x} = \frac{V}{R\omega}\left(1 - \frac{R\omega}{V}\right)$$

Since $s = \left(1 - \frac{R\omega}{V}\right)$

$$\therefore \delta(x) = \frac{xs}{1 - s}$$

**Equation 8**

Figure 3.7 below illustrates a situation where the deflection pattern along the length of the contact patch (0 – L) for a situation in which no elements are sliding with respect to the road surface.

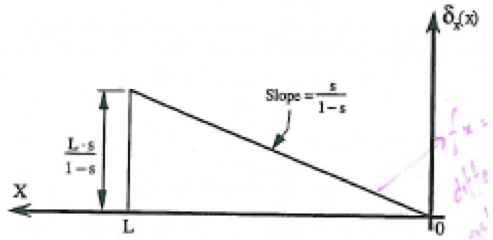


**Figure 3.7: Tyre deflection pattern, no sliding [26]**

In order to calculate the longitudinal force, the tyre is assumed to be characterised by a stiffness per unit area (A = Lw) of the contact patch. Where $k_x$ is the stiffness parameter, $F_x$ is the braking force when no sliding occurs:

$$F_x = \int_{x+0}^{L} \delta(x) k_x w dx$$

Substituting for $\delta(x)$ from Equation 8 and evaluating the integral across its length,

$$F_x = \left(\frac{k_x L^2 w}{2}\right)\left(\frac{s}{1-s}\right) = \frac{C_s s}{1-s}$$

Equation 9

Where $C_s = \frac{k_x L^2 w}{2}$ is the longitudinal stiffness parameter.

From this equation it can be seen that sliding starts to occur at the point when the friction potential per unit area cannot support any more deflection, this is further explained in Equation 10 below.

$$\frac{\mu F_Z}{A} = \frac{k_x x_s s}{1-s}$$

Equation 10

Where:

$\mu$ is the tyre/road friction co-efficient

$A$ is the area of the contact path (A=Lw)

$F_z$ is the vertical load

$x_s$ is the value of x at which sliding starts

Figure 3.8 illustrates a situation where the deflection pattern with sliding at the rear of the contact patch.

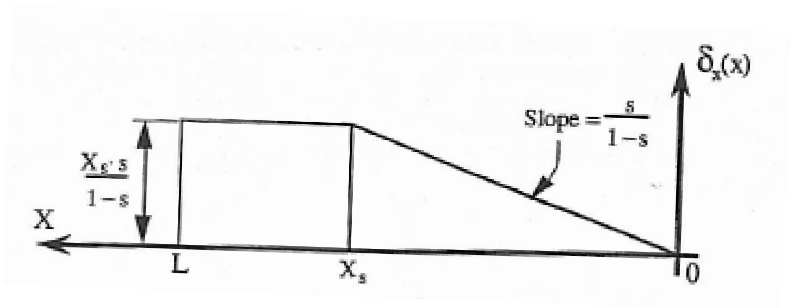


Figure 3.8: Tyre deflection pattern, with sliding [26]

Similarly,

$$F_x = \int_{x+0}^{L} \delta(x) k_x w dx + \frac{\mu F_z}{A} w(L - x_s)$$

Substituting for $\delta(x)$ and evaluating the integral across its length, the following formula solve to:

$$F_x = \frac{(\mu F_z)^2}{4C_s}\left(\frac{1-s}{s}\right) + \mu F_z(1 - \frac{x_s}{L})$$

Equation 11

Equations 9 and 11 provide a very simplified model of the longitudinal forces between the tyre and the road surface, during a situation with no slip and a situation with slip, respectively.

### *b) Lateral Force Properties*

The development of lateral force by a tyre is necessary in order to control the direction of the wheel, generate lateral acceleration in corners, and also to resist external forces acting on the tyre. When a tyre is subjected to a lateral force it deforms laterally under stress. This deformation creates an angle, slip angle ($\alpha$), between the direction the tyre is heading and the direction of travel. This deformation is illustrated in Figure 3.9 below.

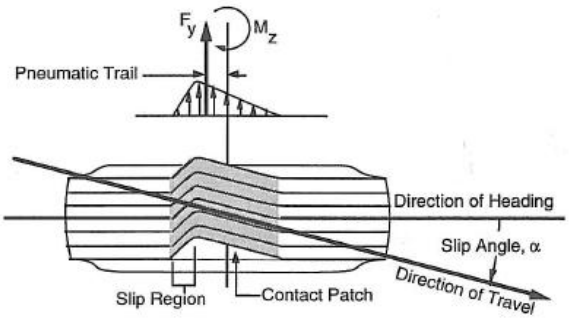


**Figure 3.9:Tyre deformation under lateral force [24]**

Figure 3.10 below is a representation of the lateral deformation of the tyre, operating at a small slip angle ($\alpha$), without the presence of longitudinal slip, $s$.

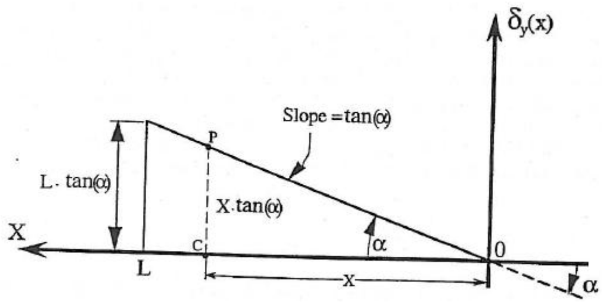


Figure 3.10: Lateral deformation of the tyre, with zero longitudinal slip [26]

The lateral deformation, $\delta_y$ of the tread element at a distance x from the front of the contact patch is:

$$\delta_y(x) = x\tan\alpha$$

Where $k_y$ is the lateral stiffness per unit area of the contact patch. An integration of the lateral force over the tyre contact area produces a lateral force, $F_y$.

$$F_{y(\alpha)} = -\int_0^L k_y(x\tan\alpha)Lwdw$$

$$F_{y(\alpha)} = C_\alpha \tan\alpha$$

Where, $C_\alpha = -\frac{k_y L^2 w}{2}$ is the tyre cornering stiffness. Note: that the negative sign has been chosen such that the lateral force is opposite to that of the slip angle.

### *c) Combined Longitudinal and Lateral Slip*

This following section provides a simple theoretical method for determining the longitudinal and lateral tyre forces under a combination of longitudinal and lateral slip conditions. An example of this situation would either be a vehicle accelerating or braking in a turn.

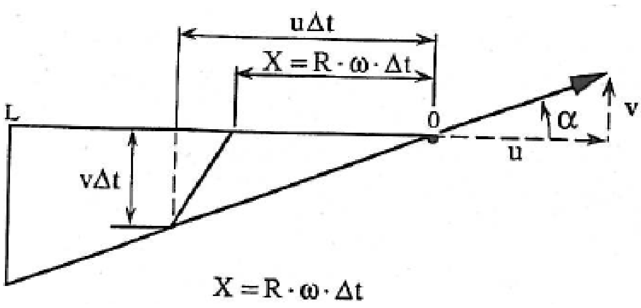


Figure 3.11: Combined slip model [26]

From Figure 3.11 above it can be seen that longitudinal slip increases the amount of lateral deflection. Using the prior equations developed in sections (a) and (b) above, the following equations express the lateral and longitudinal deflection in terms of slip.

$$\delta_y(x) = v\,\Delta t$$

$$\delta_x(x) = (u - R\,w)\Delta t$$

$$\delta_y(x) = \frac{x\tan(\alpha)}{1-s}$$

$$\delta_x(x) = \frac{s\,x}{1-s}$$

The total sliding velocity is given by the following equation:

$$V_s = \sqrt{[(u-R\omega)^2 + v^2]}$$

The angle of friction,$\theta$, is dependant of the direction of sliding therefore:

$$\sin\theta = {}^{v}/_{V_S}\ and \cos\theta = {}^{(u-R\omega)}/_{V_S}$$

In order to account for the directional influence of friction, the longitudinal and lateral friction components solve to:

$$\mu_x = \mu\cos\theta\ and\ \mu_y = \mu\sin\theta$$

From this the longitudinal and lateral force under combined slip is:

$$F_x = \left[C_s\left({}^{x_s}/_L\right)_x^2 s/(1-s)\right] + [(1-(x_s/L)_x)\mu_x F_z]$$

Equation 12

$$F_y = -\left[C_\alpha\left({}^{x_s}/_L\right)_y^2 \tan\alpha/(1-s)\right] - \left[(1-(x_s/L)_y)\mu_y F_z\right]\sin\alpha$$

Equation 13

### 3.2.6 Suspension

Heavy vehicle suspension systems are designed in order to satisfy various performance requirements, this includes, the ability to provide vertical compliance of the sprung and un-sprung masses [23] between the tyres and the road surface in order to isolate the vehicle from external disturbances, thus improving ride quality. The suspension also ensures correct orientation of the wheels with respect to the road surface and the vehicle through which the forces and moments generated can be transmitted to the vehicle body.

Mechanical leaf spring suspensions are utilised for the steer axle of heavy commercial vehicles, whilst the use of air suspensions are generally used for the drive axles. The type of suspension used for trailing axles is often determined by the vehicle application, the majority of heavy vehicles operating on South African roads make use of mechanical leaf spring suspension systems, due to the variation in load as well as the road conditions.

#### *a) Mechanical leaf spring suspension*

The properties of leaf spring suspensions can be approximated into an idealised diamond shape; this ensures that under load the lateral cross section is subjected to the same bending stress [23]. Figure 3.12 below is a representation of the semi-empirical idealised diamonds shape beam element.

One of the most important relationships which describes the fundamental properties of a leaf spring is that of the force-deflection curve. Mechanical leaf spring suspensions exhibit complex force-deflection properties, due to various factors, mainly coulomb friction; Figure 3.13 below illustrates a typical force-deflection behaviour of a leaf spring.

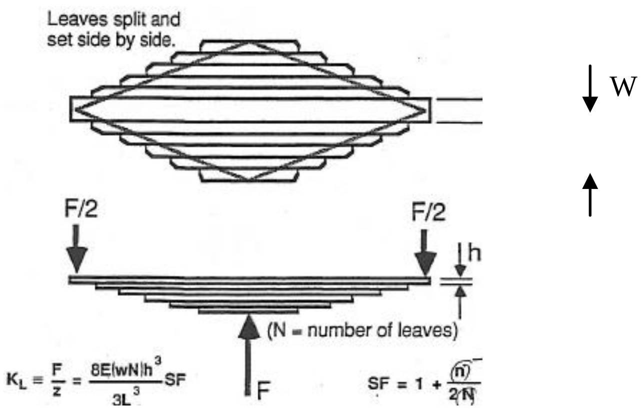


**Figure 3.12: Illustration of the idealised diamond shape simple beam leaf spring suspension [23]**

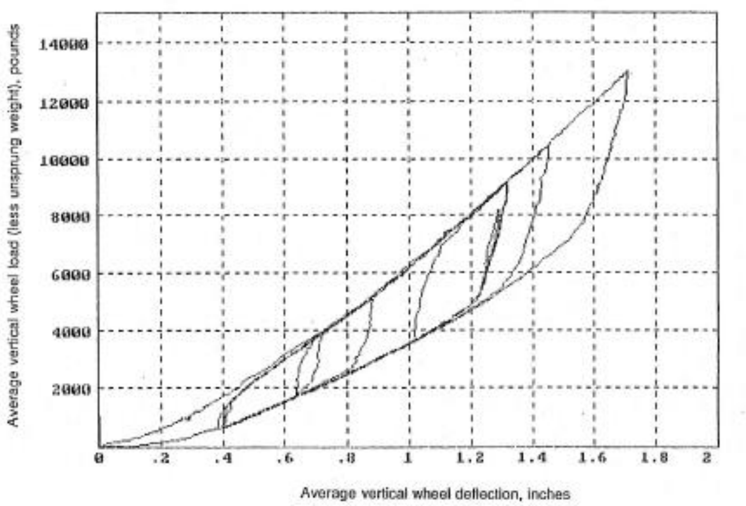


Figure 3.13: Typical leaf spring focre-delfection curve [23]

The spring rate, $K_L$, of this idealised leaf spring model is given as:

$$K_L = \frac{Load}{Deflection} = \frac{8E(wN)h^3}{L^3} SF$$

Equation 14

Where:

$E$ is the modulus of Elasticity of the material

$N$ is the number of leaves

$SF$ is the stiffening factor, and defined as:

$$SF = 1 + \frac{n}{2N}$$

Equation 15

Where:

$n$ is the number of full length springs

The most basics model of a leaf spring suspension system can be envisioned as a separate spring and damping functions, Figure 3.14 below illustrates the basic representation of the leaf spring model.

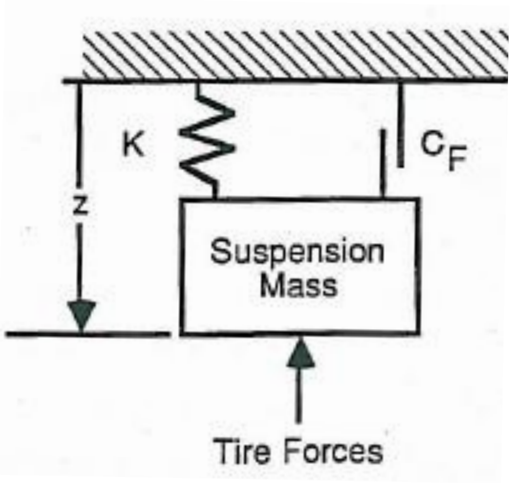


Figure 3.14: Basic leaf spring representation [23]

Where:

$K$ is the spring element

$C_f$ is the coulomb friction

$z$ is the spring deflection

In order to accurately model the suspension system as a simplified spring and damper, one needs to determine spring rates and coulomb friction constants. The development of 'average Coulomb damping force, $(C_f)$' and 'effective spring rate, $(K_e)$' accurately resemble the effect of amplitude and nominal load.

The average Coulomb damping force is based on the energy dissipated in a cycle of the spring stroke. The total energy dissipated is equal to the area, A, enclosed within the hysteresis loop, as illustrated in Figure 3.15 below.

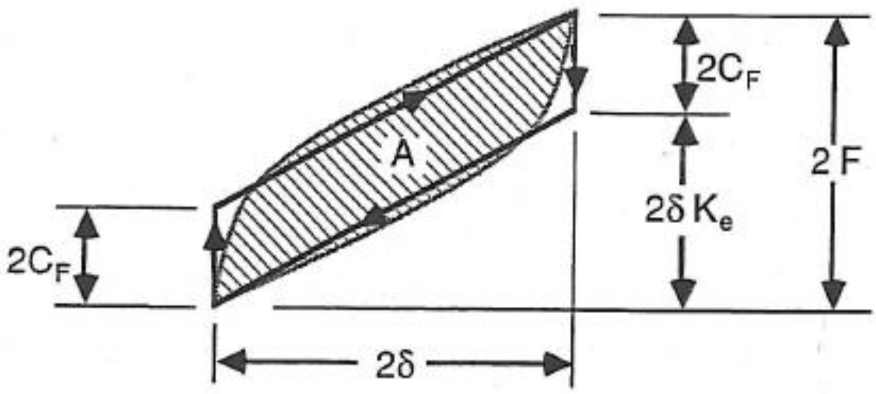


Figure 3.15: Average Coulomb damping force and effective spring rate representation [23]

$$A = (2C_F)(2\delta) \qquad or \qquad C_F = \frac{A}{4\delta}$$

Equation 16

and $$2F = 2C_F + 2\delta K_e \qquad or \qquad K_e = \left(\frac{2F - C_F}{2\delta}\right)$$

Equation 17

This representation of the average Coulomb damping force is fairly limited as it is only a representation of the conditions (load and stroke) under which they were derived [23]. An empirical model has been developed which models suspension characteristics over a wide range of parameters [26]. Equation 18 below is a representation of this empirical model, Figure 3.16 below illustrates the effective force-deflection curve for this approach.

$$F_i = F_{ENV_i} + \left(F_{i-1} - F_{ENV_i}\right)e^{\left(-|\delta_i - \delta_{i-1}|/\beta\right)}$$

Equation 18

Where:

$F_i$ is the suspension force at the current simulation time step

$F_{i-1}$ is the suspension force at the last simulation time step

$\delta_i$ is the suspension deflection at the current simulation time step

$\delta_{i-1}$ is the suspension deflection at the last simulation time step

$F_{ENV_i}$ is the force corresponding to the upper and lower boundaries of the envelope of the measured spring characteristics at the deflection, $\delta_i$

$\beta$ is an input parameter, exponential factor, used for describing the rate at which the suspension force within a hysteresis loop approaches the outer boundary of the envelope.

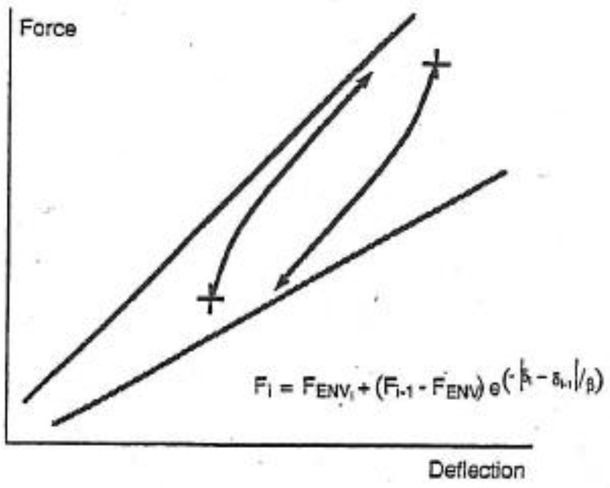


Figure 3.16: Empirical leaf spring model [23]

## *b) Air Spring Suspension*

Air spring suspensions are naturally non-linear, in that their spring rate increases with an increase in load. Figure 3.17 below is a representation of a typical air spring force-deflection curve, the figure illustrates the behaviour of an air spring tested under three differing loads; the non-linearity of the air spring is clearly evident.

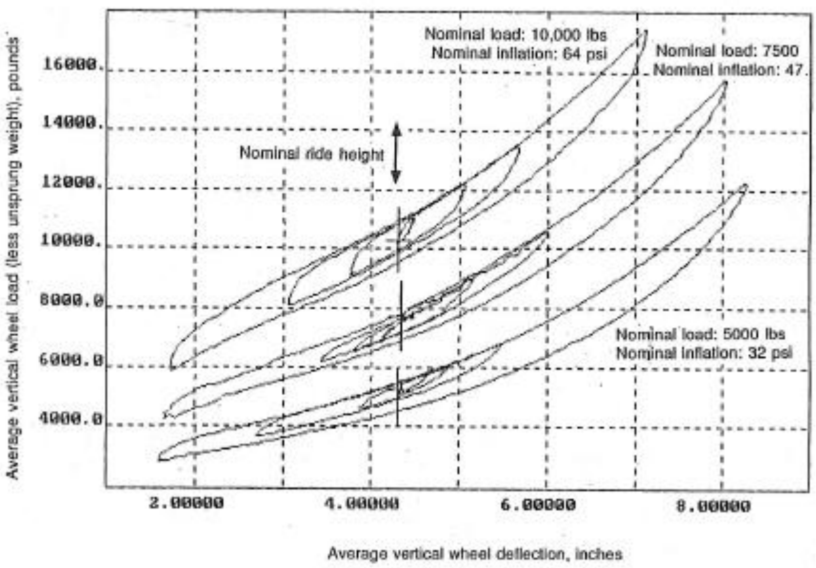


Figure 3.17: Typical force-deflection behaviour for air spring suspension, analysed under three differing loads [23]

Equation 19 below is a linear representation of the of the air spring suspension.

$$K = \frac{1.38\ A_v A_L (P_o + P_{at})}{V_o} + K_P$$

Equation 19

Where:

1.38 is the effective gas constant

$A_L$ is the effective area with respect to load

$A_V$ is the effective area with respect to volume

$K_P$ is the constant pressure spring rate

$h$ is the height of the spring

$h_0$ is the height of the spring at operating point

$L$ is the air spring load

$P_{at}$ is atmospheric pressure

$P$ is air spring gauge pressure

$P_0$ is air spring pressure at the operating point

$V_0$ is the air spring internal volume at the operating point

From this equation, one can approximate $A_L$ and $A_V$ to equal the nominal diametric area of the spring, and assuming $K_P$ as small. $A_L$, $A_V$ and $V_0$ are constant over a pressure range thus we can therefore consider them fixed. From this it can be seen that the spring rate $K,$ is a linear function of operating pressure, $P_{0,}$ and assuming, $L_0 = P_0 A_L$, the spring rate is proportional to the load.

### 3.2.7 Steering systems

The main purpose of the steering system is to allow the driver to steer the front wheels of the vehicle unit in response to the drivers steering input [24], thus in turn influence the directional response behaviour of the vehicle. The steering system is further influenced by the selection of the axles and suspension system used [23], the geometry of the steering linkage arrangement, and the drive-train characteristics of the vehicle.

Computational simulation models make use of two steering control systems, namely: open loop and closed loop control systems, the selection of which is dependent on the manoeuvre simulated.

An open loop control system simulates the reaction of a vehicle in response to a specified steer input, whilst for a closed loop control system a path trajectory is specified, which the vehicle must follow, the driver model continuously assess the vehicles directional response and adjusts the steering system accordingly.

### *a) Steering Linkages*

Heavy vehicle steering systems are designed in such a way that the frame mounted steering gearbox, which translates the rotational motion from the steering wheel to translational motion [23] in order to steer the front wheel, steers the right wheel of the steer axle. The left wheel is then steered by the right wheel via a tie rod connection.

### *b) Ackerman Steering Geometry*

The Ackerman steering geometry enables the correct turning angle of the steered wheels to be achieved when performing a tight turn. Ackerman steering states that the geometric layout of the steering linkage systems is not a parallelogram but in fact trapezoidal in design, as can be seen by Figure 3.18 below.

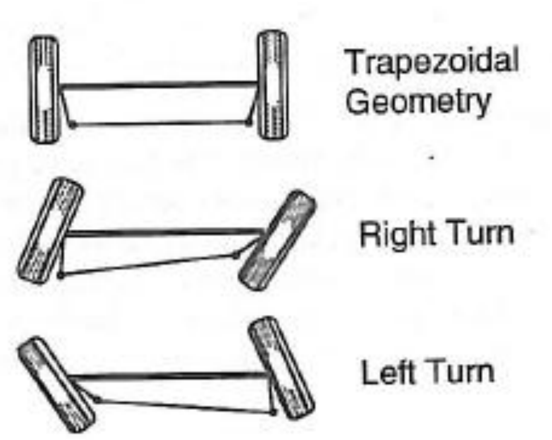


**Figure 3.18: Illustration of the trapezoidal linkage layout [24]**

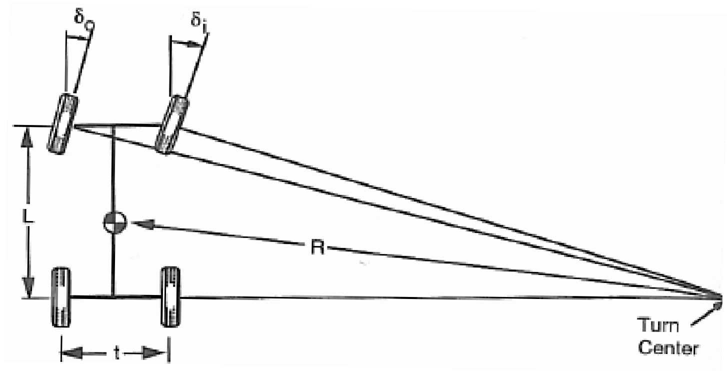


Figure 3.19: Ackerman steering geometry [23]

Figure 3.19 above is an illustration of a simplified Ackerman turning geometry, from this figure it can be seen that in order to achieve a correct Ackerman steering geometry, the following must be satisfied:

$$\delta_o = \tan^{-1}\frac{L}{\left(R + {}^{t}\!/_{2}\right)} \approx \frac{L}{\left(R + {}^{t}\!/_{2}\right)}$$

$$\delta_i = \tan^{-1}\frac{L}{\left(R - {}^{t}\!/_{2}\right)} \approx \frac{L}{\left(R - {}^{t}\!/_{2}\right)}$$

For small turning angles the arctangent is equal to the angle itself.

Further more from Figure 3.18 as well as the equations above it can be seen that during a tight turn the inside wheels to have a greater steer angle than that of the outside wheels.

The steering axis, or kingpin axis, as illustrated in Figure 3.20 below, is defined by its location with respect to the wheel centre, and its inclination in side view (the caster angle) and front view (the kingpin inclination angle). When the wheel is steered, the wheel and the steered portion of the suspension rotate about the kingpin axis [25].

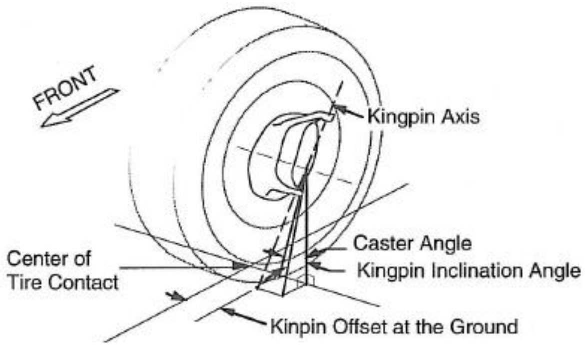


Figure 3.20: Illustration of the steering, kingpin, axis [23]

### c) Forces and Moments

The forces and moments acting on the steering system are generated from the tyre/road surface interface, as discussed in section 3.2.3 above, these forces and moments are illustrated in Figure 3.21. These tyre forces act about the kingpin axis of each wheel, and the sign convention used is based on the SAE standards.

The following subsections are a description of the forces and moments acting on the wheel at the tyre/road surface interface.

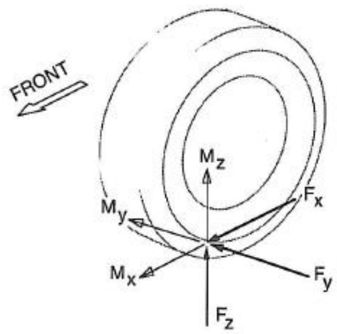


Figure 3.21: Forces and moments acting on the wheel [23]

#### i. Vertical Force

In relation to the SAE sign convention, the vertical force, $F_z$, acting upwards on the wheel is positive. The total vertical moment acting on the wheel ($M_v$)in order to steer the vehicle, is acted upon by the caster and inclination angles ($v$ $and$ $\delta$, respectively).

$$M_V = \underbrace{-(F_{zl} + F_{zr})d \sin\lambda \sin\delta}_{\text{Lateral inclination angle}} + \underbrace{(F_{zl} - F_{zr})d \sin v \cos\delta}_{\text{Caster angle}}$$

Where:

$F_{zl}, F_{zr}$ is the vertical load on the left and right wheels

$d$ is the lateral offset from the ground

$\lambda$ is the lateral inclination angle

Figures 3.22 and 3.23 below illustrate the moment produced by the vertical force acting on the lateral inclination angle and the caster angle, respectively.

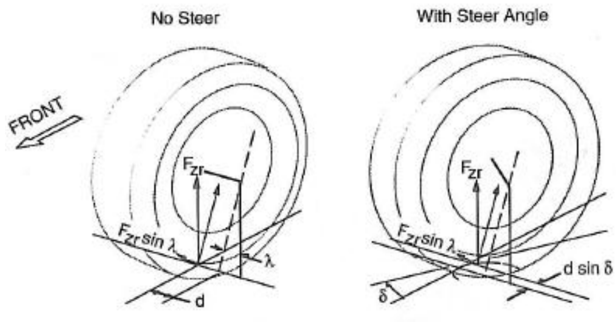


**Figure 3.22: Illustration of the moments produced by the vertical force acting on lateral inclination angle [23]**

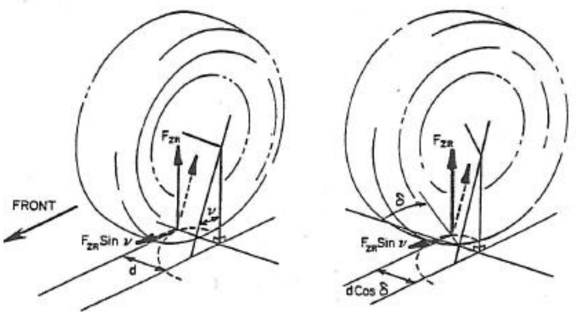


**Figure 3.23: Illustration of the moments produced by the vertical force acting on caster angle [23]**

### *ii. Lateral Force*

The lateral force $F_y$, acting on the wheel, at the centre of the tyre, produces a lateral moment ($M_L$) through the longitudinal offset.

$$M_L = (F_{yl} + F_{yr}) r \tan v$$

Where:

$F_{yl}, F_{yr}$ is the lateral force at the left and right wheels

$r$ is the tyre radius

Figure 3.24 below illustrates the moment produced by lateral force acting on the wheel, this lateral force is dependent on the steer angle and cornering conditions, positive caster produces a moment attempting to steer the vehicle out of the turn.

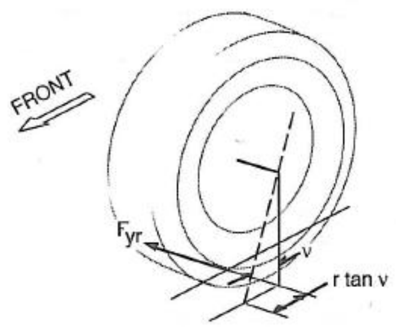


**Figure 3.24: Illustration of the moment produced by the lateral force [23]**

### *iii. Tractive Force*

The tractive force, $F_x$ acts on the kingpin offset to produce a net tractive moment,

$$M_T = (F_{xl} - F_{xr})d$$

Where:

$F_{xl}, F_{xr}$ is the tractive forces on the left and right wheels

Figure 3.25 below is an illustration of the tractive force moment at the kingpin offset.

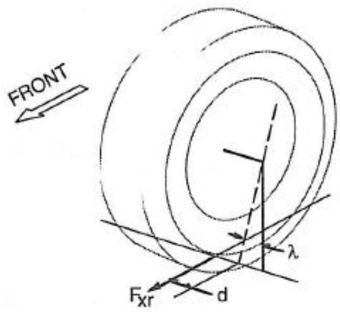


**Figure 3.25: Illustration of the moment produced by the tractive force [23]**

### *iv. Aligning Torque*

The aligning torque, $M_z$, acts vertically and acts to resist any turning motion.

$$M_{AT} = (M_{zl} + M_{zr}) \cos\sqrt{(\lambda^2 + v^2)}$$

Where:

$M_{zl}, M_{zr}$ is the aligning torques on the left and right wheels

Figure 3.26 below is an illustration of the aligning torque moment produced at the kingpin.

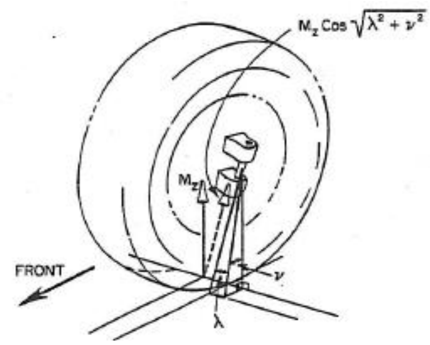


**Figure 3.26: Illustration of the aligning torque moment [23]**

### v. *Rolling resistance and Overturning moment*

These moments are not significant and are usually neglected in steer system models, thus have been excluded for the purpose of this report.

## 3.3 Conclusion

The previous sections provided a brief overview of the software packages utilised during the simulation process, as well as an overview of vehicle dynamics, the relevant axis systems utilised during the modelling process, and the forces and moments which influence various components and subsystems.

Chapter 4 provides an overview of the selection process utilised in order to determine the vehicle configurations as well as the number of vehicles selected in order to be assessed according to the PBS performance measures listed in Chapter 2.

# Chapter 4 - Vehicle Fleet Selection

This section provides a short overview of the process taken to determine the type of vehicle configurations as well as the number of vehicles selected in order to represent a sample of the South African heavy vehicle fleet. A short description into the key parameters and vehicle components selected is discussed, as well as any relevant assumptions that were made during the computational modelling process.

## 4.1 Vehicles

For the purpose of this report a survey was conducted to determine the main configurations of heavy vehicles used on South African roads. From this survey it was evident that there are four main heavy vehicle configurations, namely: rigid truck, semi-trailers, rigid draw bar and B-double (interlink).

The dynamic stability and handling of the rigid truck is relativity good in comparison to that of other heavy vehicle configurations in South Africa, and as such is not of great concern to the outcome of this project, as it does not pose any serious danger to other traffic users, it was therefore excluded for the purpose of this analysis.

The rigid drawbar configuration was also excluded as the computational mathematical model used to analyse this configuration had not yet been developed, and thus could not be simulated using the relevant software packages. Inquiries into the development of this maths model were investigated; however, the cost of development fell outside the budget allocated for the purpose of this research.

Therefore the two vehicle configurations selected for analysis were the semi-trailer and the B-double.

The data collected for each vehicle was sourced from various truck-trailer manufacturers and retailers, Hellberg Transport Management (HTM), Council for Scientific and Industrial Research (CSIR), Organisation for Economic Co-operation and Development (OECD), as well as other publicly accessible information.

## 4.2 Number of vehicles selected

From the two vehicle configurations investigated during this research study, the semi-trailer and B-double (interlink) configurations, five vehicles were selected from each vehicle class, thus providing a total of ten vehicles that were computationally modelled and analysed during the PBS evaluation process.

Each of the five vehicles selected from the relevant vehicle class, were chosen from different transportation sectors. This took into consideration the varying types of freight products transported on South African road network, thus allowing for the variation in CG heights according to the vehicles specific freight task.

One of the five vehicles from each vehicle class was selected from a previous international OECD studying into the performance of heavy vehicles. South Africa submitted four of their most common vehicles in order to be analysed according to selected Australian Performance Based Standards. It was therefore determined that this would be a control vehicle in order to analytically validate the results of the remaining four vehicle simulations.

Figure 4.1 – 4.10 below are 2-dimensional illustrations of the ten, semi-trailer and B-double, vehicles that were modelled and simulated. A fully dimensonalised detailed description of these ten vehicles can be found in Appendix B.2 and Appendix B.3, respectively.

### 4.2.1 Semi-Trailers

**a) OECD 1**

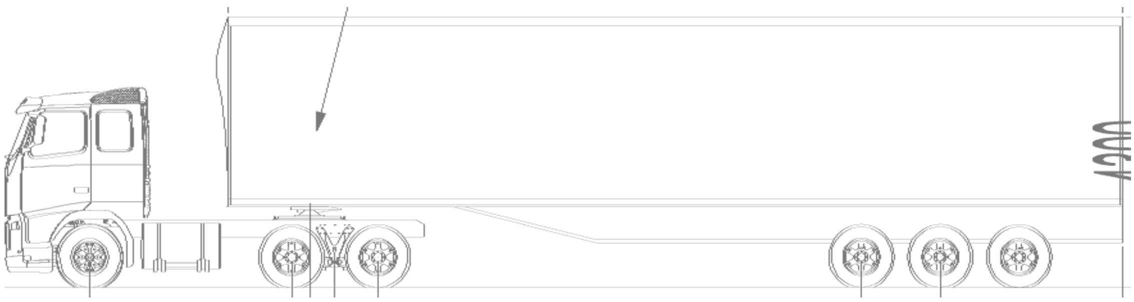

Figure 4.1: 2-dimensional side view of the OECD 1 semi-trailer

**b) Skeletal**

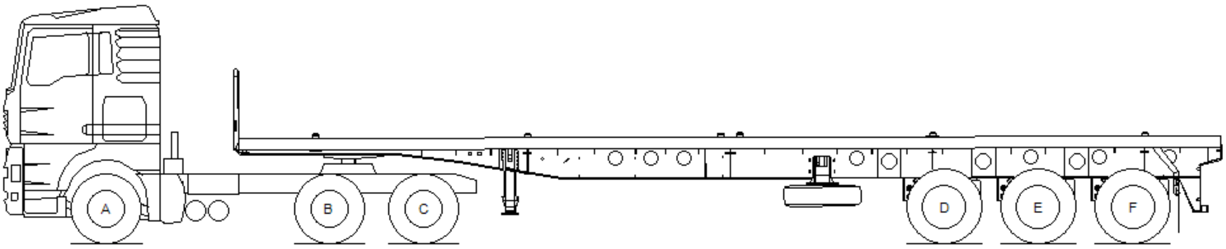


Figure 4.2: 2-dimensional side view of the skeletal semi-trailer

**c) Refrigeration**

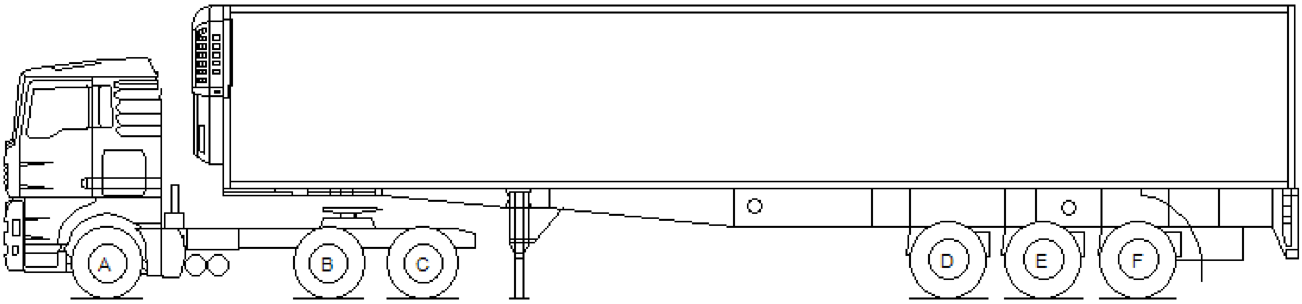


Figure 4.3: 2-dimensional side view of the refrigeration semi-trailer

**d) Side Curtain**

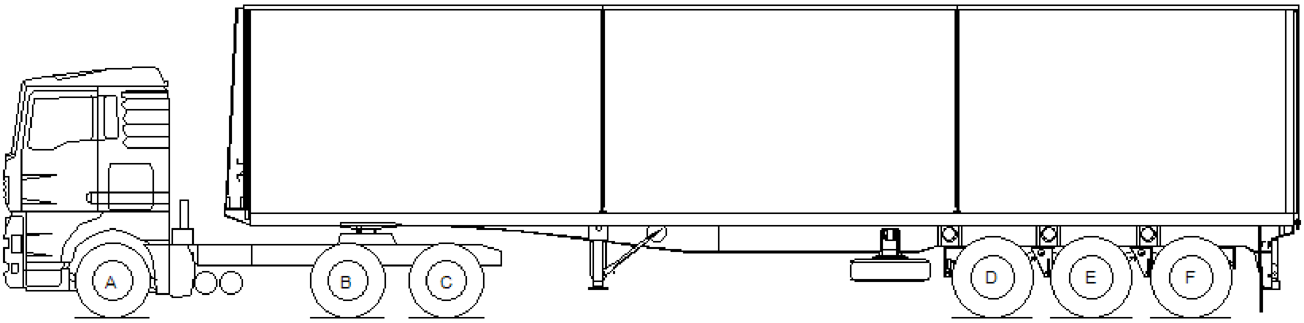


Figure 4.4: 2-dimensional side view of the side curtain semi-trailer

**e) Tipper**

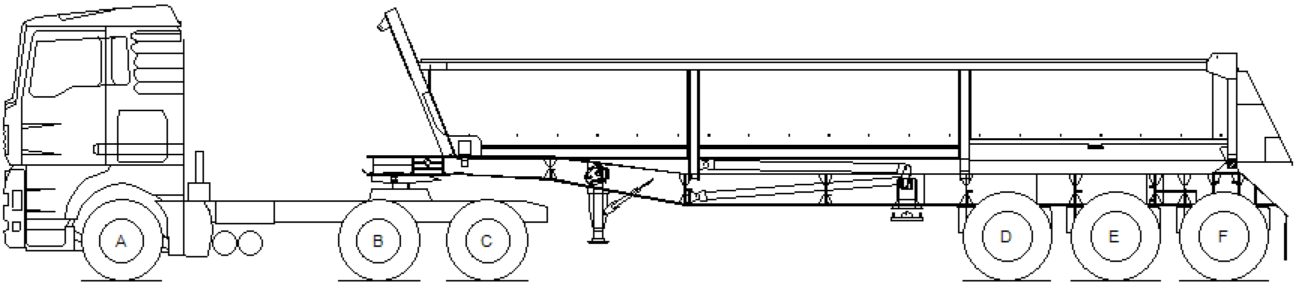


Figure 4.5: 2-dimensional side view of the tipper semi-trailer

## 4.2.2 B-double

**a) OECD 2**

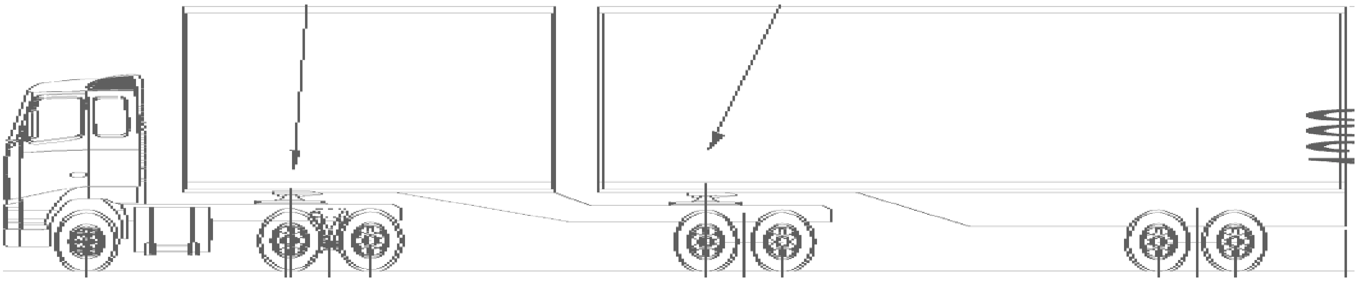

Figure 4.6: 2-dimensional side view of the OECD 2 B-double

**b) Skeletal**

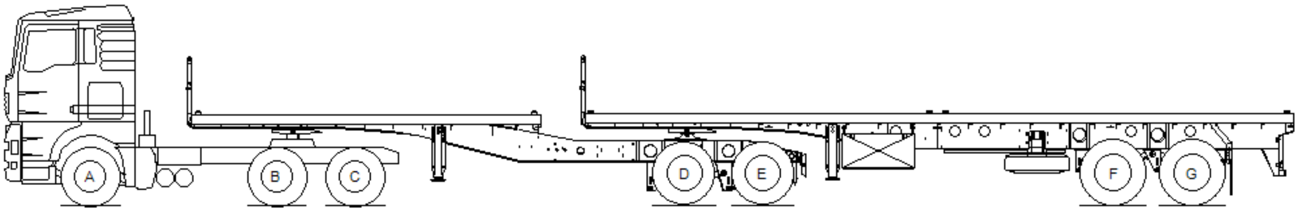


Figure 4.7: 2-dimsional side view of the skeletal B-double

**c) Cane**

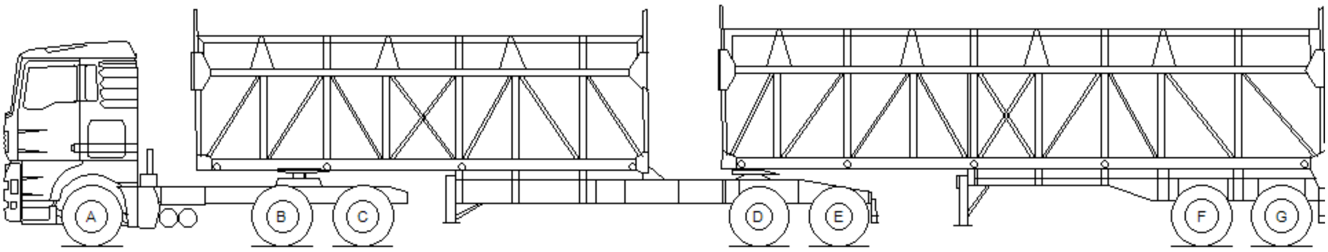


Figure 4.8: 2-dimenional side view of the cane B-double

**d) Side Curtain**

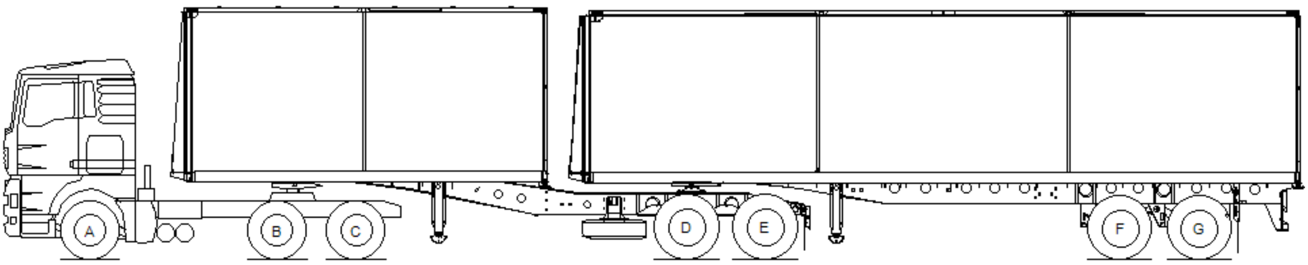


Figure 4.9: 2-dimensional side view of the side curtain B-double

**e) Tipper**

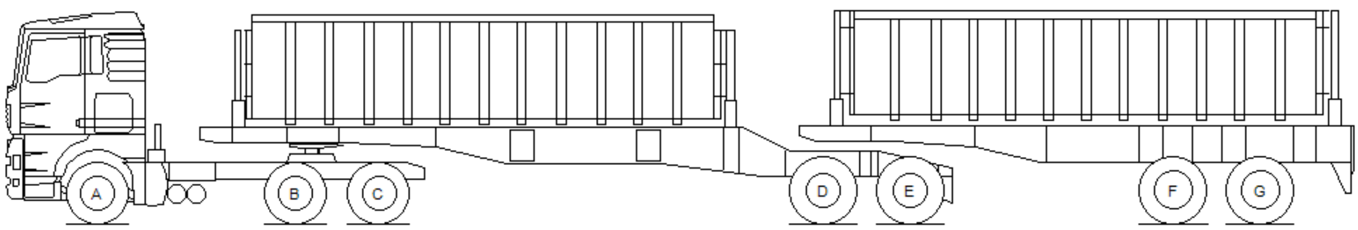


Figure 4.10: 2-dimensional side view of the tipper B-double

## 4.3 Key Dimensions

When developing 3-dimensional computer based models there are numerous parameters that are required. These parameters are not only those of the vehicle components (suspension, tyres, driveline characteristics etc.), but also that of the vehicle front and rear overhang, couplings, and container dimensions.

Some of the key parameters include: centre of gravity heights above the ground, wheelbases of all vehicle units, track widths, location of axles and axle groups, and the location of hitch points.

The information collected for the input of these key dimensions was sourced from vehicle and component manufacturers and suppliers, Trucksim, Hellberg Transport Management, OECD report, as well as publically accessible information.

The key dimensions for the prime movers, five semi-trailer and five B-double configurations can be found in Appendix B.1.1, Appendix B.2.1 and Appendix B.3.1, respectively.

## 4.4 Mass Properties

The tare and laden masses of each vehicle and trailer unit, the relevant component masses (suspension, axles, tyres etc.), mass moments of inertia, and the mass centre of gravity of each load above ground were obtained from vehicle manufacturers and suppliers, Hellberg Transport Management, as well as Trucksim.

The location, offset and height of each vehicle unit and vehicle load has a significant effect on the dynamic performance, stability and handling of each vehicle configuration.

The mass properties for the prime mover, five semi-trailer and five B-double configurations can be found in Appendix B.1.2, Appendix B.2.2 and Appendix B.3.2, respectively.

## 4.5 Axles

Information obtained in order to determine axle masses, mass moments of inertia, track width, centre of gravity (CG) heights above ground, loads and axle spacing's were sourced from component manufacturers, Hellberg Transport Management, Trucksim, government legislation as well as documentation available in the public domain.

The axle loads associated for each trailer were determined from an in-house software package developed by HTM. This allowed users to select various trailer configurations and input varying payloads and generic densities in order to determine the load applied on to each axle group, however, it was often noted that the axle group load is limited not to its product capacity, but rather to the national maximum legal load limit.

The axle parameters for the prime mover, five semi-trailers and five B-double configurations can be found in Appendix B.1.4, B.2.3 and B.3.5 respectively, as well as from the HTM dimensional drawings.

## 4.6 Couplings

The main purpose of a coupling is to connect one vehicle unit with another in a multi-combination arrangement, and to permit articulation between adjacent units. There are two main types of couplings used, namely: a turntable (fifth wheel) and a pin-coupling. Due to the selection of vehicle class a turntable was the only coupling used for the analyses of the vehicles selected in this report.

The data collected for the computational modelling of the turntable, degrees of freedom – translation as well and rotational motion (roll, pitch and yaw), was collected from manufacturers, suppliers, National Road Transport Commission reports, as well as from publically accessible data tables.

The coupling mechanical properties for the prime mover and semi-trailers, as well as the B-double configurations can be found in Appendix B.1.3 and Appendix B.3.3, respectively.

## 4.7 Suspension

Suspension information proved to be the most difficult component to obtain a complete list of data, therefore numerous sources were used and a “generic” data set was compiled.

The suspension data collected was sourced from various suppliers and manufacturers, Trucksim data base and University of Michigan Transportation Research Institute (UMTRI). All the data collected was compiled to obtain an overall representation of South African suspension systems. Two types of suspension were used during the modelling process, air suspension for the drive axles, and leaf spring (mechanical) suspension was utilised for the steer and trailing axles.

The suspension properties for the prime mover, semi-trailer and B-double configurations can be found in Appendices B.1.6, B.2.5 and B.3.6, respectively.

## 4.8 Tyres

Numerous discussions with individuals in the transportation industry indicated that the most common heavy vehicle truck tyres used on South African roads are a 315/80 R22.5 (radial ply) and a 12 R22.5

(radial ply) tyre. The 315/80 R22.5 was seen to be the most widely used across all vehicle transportation sectors, and thus the same tyre characteristics were used for all ten vehicle configurations models analysed during  the purpose of this research study.

Tyre characteristics were obtained from manufactures and suppliers, Trucksim and publically accessible brochures. The tyre characteristics used for the prime mover, semi-trailers and B-double configurations can be found in Appendix B.1.5, B.2.5 and B.3.5, respectively.

## 4.9 Driveline Characteristics

Driveline characteristics for the prime mover were obtained from manufacturers, suppliers, Trucksim, Hellberg Transport Management, as well as Mantec, an in-house software developed by the vehicle manufacturer.

The prime mover driveline characteristics used for all ten of the vehicles assessed were based on the MAN TGA 26.480 6x4 BLS front over cab. The prime mover is a conventional unit which is widely used throughout the transportation industry in South Africa. It has an output power of 480Hp (352 KW), and makes use of both leaf spring (steer axle) and air (drive axle) suspension. The driveline characteristics used in the modelling process can be seen in Appendices B.1and C.1.

## 4.10 Road Surface Unevenness

The road profiles upon which each vehicle safety performance manoeuvre was simulated, was stipulated by the National Transport Commission, and documented in 'The Standards and Vehicle Assessment Rules – July 2007' [13].

12 of the 13 safety performance manoeuvres were simulated on flat surfaces, whilst the tracking-ability on a straight path performance manoeuvre was simulated on an uneven road surface. This road profile was supplied to the assessors by the National Transport Commission; this road profile was taken from the work performed by Hans Prem for Austroads. This profile was then used to construct a 3-dimensional road surface, upon which all vehicles were simulated.

## 4.11 Assumptions

In order to accurately compare the performance results of the ten vehicle configurations according to their vehicle class, mass and dimensions, various assumptions had to be made during the modelling process. These assumptions would limit the effect various individual components (axles, suspension, tyres etc.) would pose on the outcome of the results, thus ensuring the results are based purely on the individual description of each vehicle, (mass, dimension and vehicle configuration).

Some of the assumptions made during the modelling process include:

- The payload mass was assumed to symmetrical and cover the entire base of the trailer
- The payload centre of gravity height was assumed to be 40 % of the available load space height of the trailer
- Least Favourable Load Condition (LFLC) was assumed to be the maximum dimension of the available load space of the trailer
- The tare mass, dimensions and payload were specified by the OECD report, as well as the HTM vehicle selection analysis
- A maximum width of 2.6 m was used for all ten vehicles
- The centre of gravity height of the prime mover was set as 1,1 m above the ground
- A single tyre selection of 315/80 R22.5 was used for axles: steer, drive and trailing.
- All axles were assumed to have dual tyres, except steer axles.
- Each axle steer, drive and trailing was assumed to have a mass of 527 kg, 735 kg and 800 kg, respectively, unless otherwise stated by OECD report.
- A single set of generic suspension data was used for all vehicle suspensions.
  - Leaf / steel spring suspension was used for the steer and trailer axles
  - Air suspension was used for the drive axles
- The same coupling (fifth wheel) was used throughout the modelling process for all ten vehicles
- The MAN TGA 26.480 prime mover was used for eight of the ten vehicles, whilst the remaining two OECD prime movers were specified by the OECD report.

## 4.12 Conclusion

This chapter provided a short overview of the number of heavy vehicles selected from each vehicle classification, semi-trailer and B-double, in order to represent a small sample size of the current heavy vehicles used in the South African transportation system.

It provided an overview of the components and key vehicles parameters, as well as where the data of each component was sourced, and a description of any assumptions made throughout the modelling process.

Chapter 5 provides the results for the semi-trailer and B-double fleet selected in Chapter 4, according to the performance measures discussed in Chapter 2, making use of the software packages that were discussed in Chapter 3.

# Chapter 5 - Results and Discussions

This section of the report provides the results for both the semi-trailer and B-double vehicle configurations. Due to the same prime mover being used for majority of the simulations the startability, gradeability and acceleration capability performance measures are grouped together in their own section. The remaining ten safety performance measures are grouped together according to vehicle configuration. Further investigation into major influences of the simulation results was undertaken, and their effect on each performance measure can be seen below.

This section also provides a means for analytical validation of simulated results through a comparison of two vehicles (semi-trailer, OECD 1, and B-double, OECD 2) from a published OECD report, as well as a discussion on the performance results achieved.

## 5.1 Startability, Gradeability and Acceleration Capability

The three performance measures in this section, Startability, Gradeability and Acceleration capability, are used to determine the ability of heavy vehicles to start on a grade, climb on a grade and accelerate from rest on zero grades, respectively.

They have therefore been grouped together due to the fact that they all depend on the driveline and engine characteristics of the prime mover, such as engine torque-speed characteristics, clutch engagement torque, gearbox and final drive ratios, time durations and delays associated with gear changes. They have also been grouped together due to the fact that the same prime mover has been used to asses each of the ten vehicles. For this reason they have not been associated according to vehicle configuration, semi-trailer or B-double, as with the other ten safety performance measures.

Various forms of data were collected from the manufacturer as well as from HTM; this data was then used to calculate the startability, gradeability and acceleration capability of a vehicle with a gross combination mass of 56 tons. The necessary data obtained can be found in Appendix C.1.

Due to the sensitive nature of the driveline and engine data, all necessary information was not available from the vehicle manufacturers, and these performance measures were thus not capable of being computationally simulated. However, data collected from various sources allowed for startability and gradeability to be analytically calculated, whilst field tests had to be undertaken in order to determine the vehicles acceleration capability.

### 5.1.1 Startability

Startability of a heavy vehicle is directly related to critical parameters such as gross combination mass and the overall length of the vehicle [5], an increase in either parameter would result in a negative effect on the vehicles startability.

Other factors such as driveline gear ratios and axle loads also influence this performance measure, an increase and decrease respectively, have a positive effect on the vehicle startability. Engine power does not influence the startability of heavy vehicles, as startability is mainly concerned with the clutch engagement torque at low speeds.

The startability of a vehicle is directly related to that of its gradeability; however, the vehicles startability is not frequently requested, in comparison to that of the vehicles gradeability, thus the data was not available from vehicle manufacturers. However, a general much utilised industrial rule suggests that the startability of a vehicle is equal to the gradeability of that vehicle less the tractive slip at the commencement of forward motion.

This tractive slip, or skid point, for a MAN TGA 26.480 BLS prime mover loaded to 56 tons travelling on an asphalt road, at an engine speed of approximately 900 rpm, in low gears is 21 %. This condition is satisfied by the gradeability of the MAN prime mover in first gear at an engine speed of 1000 rpm, illustrated by Figure C.5 in Appendix C.1, providing a gradeability of 39.6 %.

This gradeability less the tractive slip provides a vehicle startability of 30%, thus surpassing the minimum percentage requirement of 15% for Level 1 road classification. Thus ensuring that the prime mover has the capability to start and commence motion on an inclined grade for a vehicle loaded to a maximum combination mass of 56tons, minimising the safety risk to other road users.

### 5.1.2 Gradeability

Vehicle gradeability, similar to that of startability, is affected by gross combination mass and overall length, whilst other factors such as an increase in engine power / torque speed, an increase driveline gear ratios and a decrease in axle loads have a positive influence on the vehicles ability to climb on a graded surface.

#### *(a) Maintain forward motion on maximum grade*

Figure C.3 from the manufacturer provides a gradeability in first gear of 38.82 %, whilst Figure C.5 from HTM provides a gradeability of 39.6%. Due to this 2% variation in results, the lower more conservative value of 38.8% was therefore selected.

This result of 38% gradeability is greater than the required minimum performance requirement of at least 15%, thus qualifying for Level 1 road classification.

*(b) Maintain minimum speed on a 1% grade*

Figures C.4 and C.5 provided by HTM allow one to calculate the maximum speed obtainable on a 1 % grade. The tabular data provided indicates that a maximum speed of 87.4 km/h is obtainable at a grade of 1.1%, whilst a maximum speed of 98.4 km/h is obtainable on a 0.7% grade, a linear iterative process was therefore utilised in order to determine that a maximum speed of 90.1 km/h is obtainable on a 1.0% grade.

This result of 90km/h gradeability is greater than the minimum stipulated performance requirement of at least 80 km/h, thus satisfying the Level 1 road classification.

The results of the gradeability performance measure ensure that the vehicle has the ability to maintain forward motion on a grade of 38%, and also to maintain a minimum speed of 90 km/h on a grade of 1%, therefore limiting the risks of other road users.

### 5.1.3 Acceleration capability

Acceleration capability, similar to both the previous performance standards, is negatively affected by gross combination mass and overall length of the vehicle. Whilst other factors such as an increase in engine torque and a decrease in driveline gear ratios have a positive effect on the vehicle acceleration capability. Unlike startability and gradeability, acceleration capability is concerned with engine and gearbox characteristics across the entire speed range. Time delays and duration associated with automatic and manual gear changes also have a critical effect on the acceleration capability of heavy vehicles.

Due to the sensitive nature of various engine characteristics, critical data was not obtainable thus computer modelling was not possible, and as such field tests had to be undertaken. Timber24, a freight logistics company, allowed for the use of a vehicle, of length 22m and loaded to maximum allowable gross combination mass of 56 tons, to undertake an acceleration capability test. However, the prime mover used in the field tests was not the same as the MAN TGA 26.480 that was modelled throughout the other performance manoeuvres, but rather a Mercedes Benz Actros 33.50; however, this did not pose a concern as acceleration capability is not influenced by the engine power, and both the MAN and the Mercedes Benz vehicles have similar 12 gear gearboxes, with similar gear ratios.

Two sets of simulations were undertaken; firstly acceleration from a standing start changing through gears automatically, and secondly changing through gears manually.

**Table 5.1: Acceleration capability performance results for 56 ton heavy vehicle with automatic gear changes**

| Distance travelled | Time Run 1 | Run 2 | Run 3 | Run 4 | Average |
|---|---|---|---|---|---|
| 20 m | 7.82 | 8.49 | 7.98 | 8.08 | 8.1 |
| 40 m | 12.14 | 12.94 | 12.34 | 12.81 | 12.6 |
| 60 m | 15.48 | 16.13 | 15.48 | 16.16 | 15.9 |
| 80 m | 18.3 | 18.99 | 18.52 | 19.06 | 18.8 |
| 100 m | 20.97 | 21.78 | 21.22 | 21.79 | 21.5 |

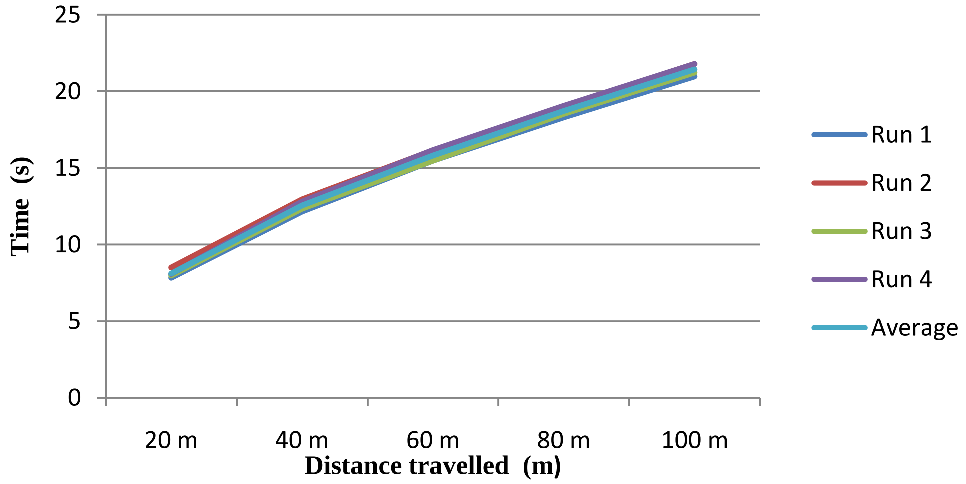


**Figure 5.1: Acceleration capability of a 56 ton heavy vehicle with automatic gear changes**

**Table 5.2: Acceleration capability performance results for 56 ton heavy vehicle with manual gear changes**

| Distance travelled | Time Run 1 | Run 2 | Run 3 | Run 4 | Average |
|---|---|---|---|---|---|
| 20 m | 8.11 | 7.9 | 7.8 | 7.78 | 7.9 |
| 40 m | 12.59 | 12.41 | 12.24 | 11.91 | 12.3 |
| 60 m | 16.09 | 16.7 | 15.45 | 15.51 | 16.0 |
| 80 m | 19.23 | 19.72 | 18.55 | 18.28 | 19.0 |
| 100 m | 21.87 | 22.42 | 20.94 | 20.89 | 21.6 |

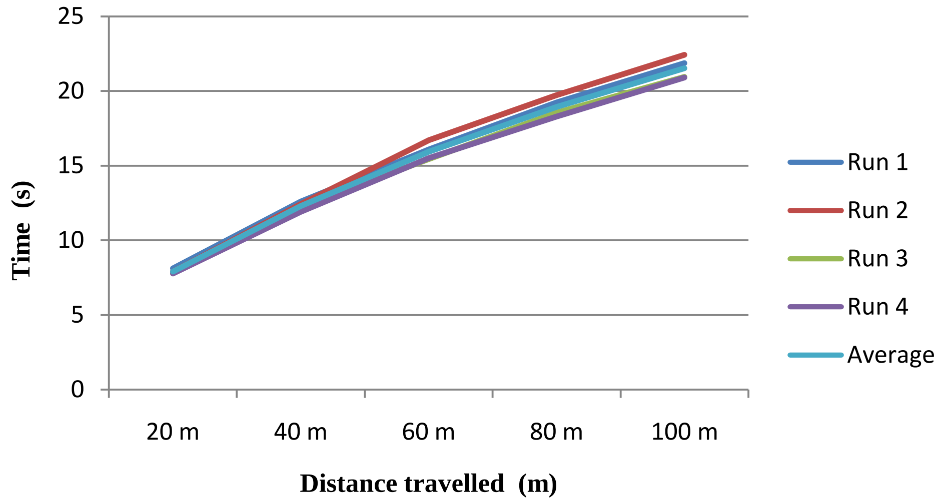


**Figure 5.2: Acceleration capability of 56 ton heavy vehicle with manual gear changes**

Tables 5.1 and 5.2 represent the time a fully laden heavy vehicle requires to travel a distance of 100 m with automatic and manual gear changes, respectively. Figures 5.2 and 5.3 illustrates the time the vehicle required to travel a distance of 20, 40, 60, 80, and 100m, for each of the four runs as well as a calculated average.

The results from Tables 5.1 and 5.2 range from 20.89 to 22.42 seconds, thus ensuring that this vehicle satisfies the Level 2 performance requirement of less than 23 seconds for a vehicle to accelerate from rest and travel a distance of 100m, for both automatic and manual gear changes. This ensures that the vehicle has the capability to clear intersections and over take additional vehicles in an acceptable period of time, thus reducing congestion and safety risk posed to other road users.

According to personnel at MAN a recent study of a similar vehicle was undertaken by Hans Prem at Mechanical Simulation Dynamics (MSD) Pty Ltd, this also provided the vehicle with a Level 2 performance classification

**Table 5.3: Startability, Gradeability and Acceleration Capability performance results of a MAN TGA 26.480 BLS prime mover**

| Performance Measure | Result | Level Passed |
|---|---|---|
| Startability | 30% | Level 1 |
| Gradeability | | |
| Part A - maintain forward motion | 38% | Level 1 |
| Part B - maintain minimum speed | 90 km/h | Level 1 |
| Acceleration Capability | Pass | Level 2 |

From these results in Table 5.3 it is evident that the MAN prime mover has the ability to start on a grade, climb on a grade and accelerate from rest on a zero grade, with results exceeding that stipulated by the PBS guidelines, and as such does not pose any concern.

## 5.2 Semi-Trailer

### 5.2.1 Tracking Ability on a Straight Path

The tracking ability of a heavy vehicle on a straight path is an important performance measure designed to determine the lateral deviation a vehicle experiences from the desired path when travelling at high speeds, under a worst case scenario.

Tracking ability on a straight path measures the lateral deviations, swept path, in the ground plane of a vehicle in response to road surface unevenness, cross-fall and other external disturbance in order to determine its lane width requirements. These external disturbances, road surface unevenness and cross-fall, increase the vehicles swept path, thus ensuring that the vehicle will perform better under a normal working environment.

The following section provides a description of the tracking ability performance results for the five semi-trailer combinations, the various reference points required by the computational model for verification, as well as the factors which influence the performance results.

Numerous points on the vehicle of concern include: the centre of steer axle – in order to ensure the vehicle follows the desired path within the prescribed limitations, the outside edges of the steer tyre, as well as the outside edges of the trailer. (The placement of these reference points for each specific vehicle can be found in Table C.2 of Appendix C.2.1.1)

**Table 5.4: Tracking Ability on a Straight Path performance results for the five Semi-trailer combinations**

| Vehicle | Min | Max | Swept Path | Result | Level Passed |
|---|---|---|---|---|---|
| OECD 1 | -1.254 | 1.703 | 2.957 | 3.0 | Level 2 |
| Skeletal | -1.233 | 1.636 | 2.870 | 2.9 | Level 1 |
| Refrigeration | -1.219 | 1.702 | 2.922 | 3.0 | Level 2 |
| Side Curtain | -1.222 | 1.711 | 2.933 | 3.0 | Level 2 |
| Tipper | -1.228 | 1.632 | 2.860 | 2.9 | Level 1 |

Table 5.4 is a representation of performance results for the five semi-trailer combinations, indicating the minimum and maximum lateral deviations of the references points (corresponding to that particular vehicle), the swept path (summation of the absolute values of the maximum and minimum lateral deviations), the performance result as well as the road classification level passed. The performance results for TASP for the semi-trailer combinations can be found in Appendix C.2.1.

From this table it can be seen that the values of TASP range from 2.860 m to 2.957 m. Two of the five vehicles (Skeletal and Tipper) qualify for the Level 1 road classification of not greater than 2.9 m, whilst the remaining three vehicles (OECD 1, Refrigeration and Side Curtain) qualify for the Level 2 road classification by not exceeding the 3.0 m limit stipulated.

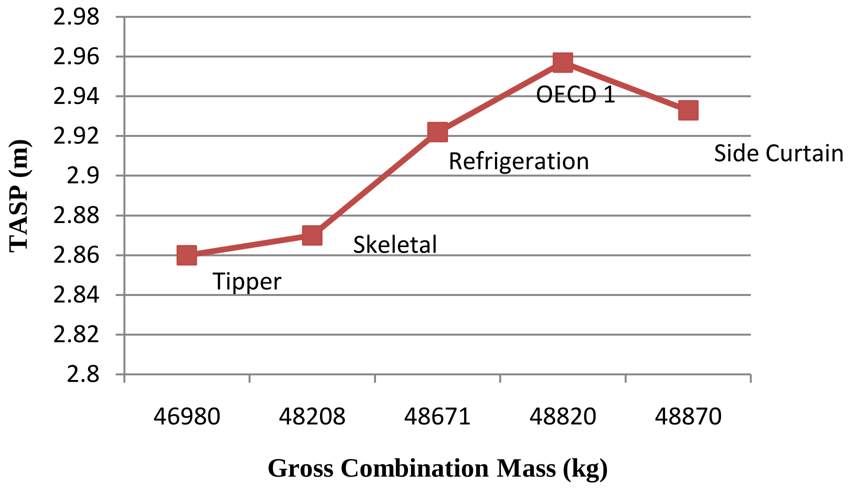


Figure 5.3: The influence of gross combination mass on the tracking ability of semi-trailer combinations

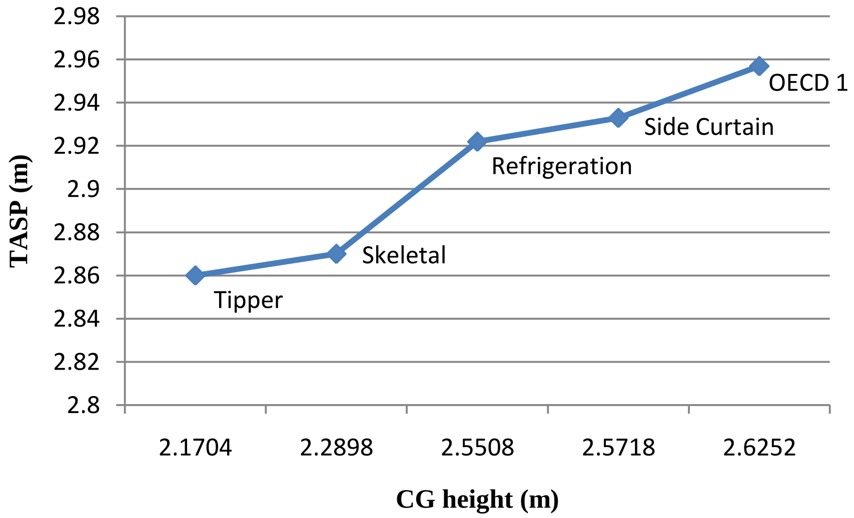


Figure 5.4: The influence of Centre of Gravity (CG) height on the tracking ability of semi-trailer combinations

Figures 5.3 and 5.4 above illustrate the influence of gross combination mass and centre of gravity height on the tracking ability of the semi-trailers combinations, each of which indicate a strong relationship between gross combination mass and centre of gravity height on the vehicles tracking ability. An increase in the gross combination mass or an increase in the centre of gravity height results in a negative effect on the vehicles tracking ability performance. Centre of gravity height is the most dominant factor concerned with tracking ability; this is evident between Side Curtain and OECD1 vehicles.

Other factors that influence the tracking ability include: number of trailers, the locations and type of coupling between vehicle units, tyre cornering stiffness, vehicle speed, and road surface unevenness.

From these results it is evident that the tracking ability of the five semi-trailer combinations does not pose a concern as all five vehicles track well within the minimum South African lane width of 3.25 m, therefore imposing no risk to other road users or the road side infrastructure.

### 5.2.2 Low Speed Swept Path

The low-speed swept path performance measure is designed to measure the lateral inward tracking of a vehicle when performing a tight turn at low speed. This section provides a description of the results for the low-speed swept path manoeuvre for the five semi-trailer combinations, as well as the factors which influence its performance.

**Table 5.5: Low speed swept path performance results for five semi-trailer combinations**

| Vehicle | Unladen | Laden | Result | Level Passed |
|---|---|---|---|---|
| OECD 1 | 6.5445 | 6.5052 | 6.6 | Level 1 |
| Skeletal | 6.5221 | 6.4996 | 6.6 | Level 1 |
| Refrigeration | 6.5459 | 6.5063 | 6.6 | Level 1 |
| Side Curtain | 6.5290 | 6.5006 | 6.6 | Level 1 |
| Tipper | 6.0239 | 5.9975 | 6.1 | Level 1 |

Table 5.5 provides a summary of the low-speed performance results for the five semi-trailer combinations, under both laden and unladen conditions, as well as the road classification level achieved.

The performance results for the five semi-trailer combinations range from 5.9975 m to 6.5459 m; all five of the vehicles achieved the less than 7.4 m requirement in order to qualify for Level 1 road classification. The resultant plots of this performance measure can be seen in Appendix C.2.2, however, it must be noted that these performance plots do not incorporate the overall vehicle width of 2.6 m.

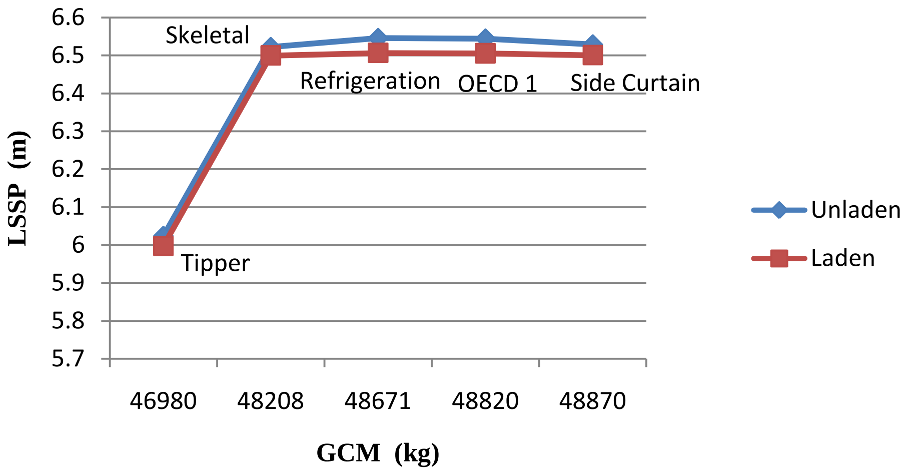


**Figure 5.5: The influence Gross Combination Mass (GCM) has on the Low Speed Swept Path of five semi-trailer combinations**

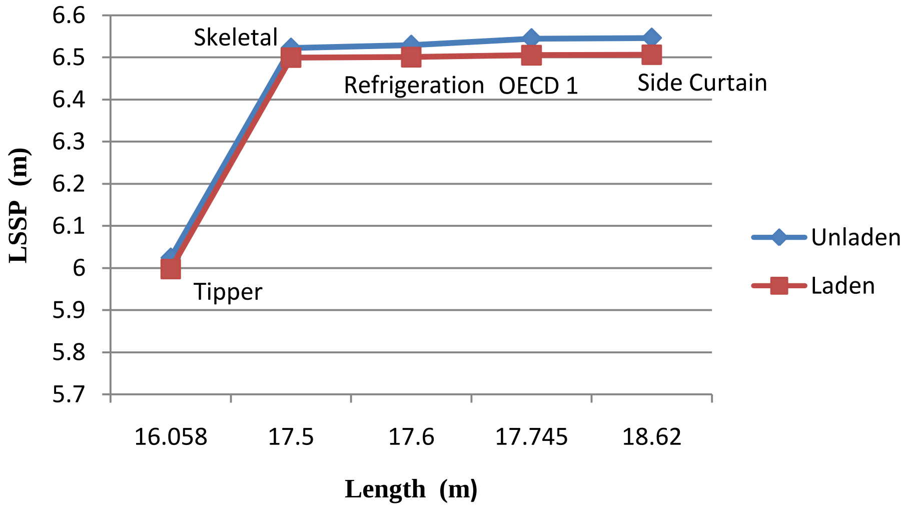


**Figure 5.6: The influence vehicle length has on the Low Speed Swept Path of five semi-trailer combinations**

Figure 5.5 and 5.6 above illustrate the influence of gross combination mass and vehicle length has on the low-speed swept path of the five semi-trailer combinations, under both laden and unladen conditions. Figure 5.5 shows a slight tendency of increased swept path with an increase in gross combination mass; Figure 5.6 illustrates a similar effect with an increase in vehicle length.

Other factors that influence low-speed swept path include: wheelbase of all vehicle units, frontal overhang of the hauling unit and coupling rear overhang, an increase in each of these parameters has a negative influence on the vehicles tracking capability.

From these results it is evident that low-speed swept path does not pose any concern for the five semi-trailer combination vehicles assessed, ensuring that all five vehicles have the ability to remain in their required lane widths when performing a tight turn at low speeds.

### 5.2.3 Frontal Swing

Frontal swing is designed to measure the amount of road space a vehicle requires when performing a low speed turn. Below are the results for Part A (hauling unit), Part B (Maximum Difference) and Part C (Difference of Maxima) for the five semi-trailer combination vehicles, a description of the results and the various factors which influence their performance, under both laden and unladen conditions.

**Table 5.6: Frontal swing Part A, Hauling unit, performance results for five semi-trailer combinations**

| Vehicle | Result | | Pass / Fail |
|---|---|---|---|
| | Laden | Unladen | |
| OECD 1 | 0.43 | 0.37 | Pass |
| Skeletal | 0.46 | 0.38 | Pass |
| Refrigeration | 0.45 | 0.39 | Pass |
| Side Curtain | 0.46 | 0.38 | Pass |
| Tipper | 0.45 | 0.39 | Pass |

Table 5.6 above is a representation of the results for Part A (hauling unit) frontal swing for the five semi-trailer combination vehicles. The results range from 0.37 m to 0.46 m, thus ensuring that all five vehicles achieved a performance result of less than the maximum stipulated 0.7 m performance requirement, this therefore ensure that the prime mover of the vehicle combination will remain within its own lane throughout the low speed manoeuvre.  The resultant plots for Part A Frontal swing can be found in Appendix C.2.3.1.

Due to the same prime mover being used the five semi-trailer combinations, there was little variation in prime mover frontal overhang, although frontal overhang being the predominant factor for Part A frontal swing; however, it must be noted that an increase in prime mover frontal overhang will result in an increase in frontal swing. From the results of Table 5.6 it can be seen that an increase in mass results unfavourably to an increase in frontal swing.

**Table 5.7: Frontal swing Part B, Maximum of Difference (MoD), performance results for five semi-trailer combinations**

| Vehicles | Results | | Frontal overhang (mm) | Pass / Fail |
|---|---|---|---|---|
| | Laden | Unladen | | |
| OECD 1 | 0.33 | 0.31 | 1300 | Pass |
| Skeletal | 0.41 | 0.39 | 1550 | Fail |
| Refrigeration | 0.52 | 0.49 | 1800 | Fail |
| Side Curtain | 0.39 | 0.38 | 1500 | Pass |
| Tipper | 0.09 | 0.09 | 385 | Pass |

Table 5.7 above is a representation of the results for Part B (Maximum of Difference, MoD) frontal swing for the five semi-trailer combination vehicles. The MoD results range from 0.09 m to 0.52 m, thus resulting in three of the five vehicles (OECD1, Side Curtain and Tipper) achieving the required performance requirement. The remaining two vehicles (Skeletal and Refrigeration) exceeded the 0.4m maximum requirement, and were therefore not deemed to comply. The failure to meet this standard indicates that the forward most outside point of the first semi trailer, when performing a tight turn at low speed, will tend to track outside it specified lane width, which may result in collisions with road side objects as well as other vehicle users. The resultant plots for the Part B Frontal swing can be found in Appendix C.2.3.2.

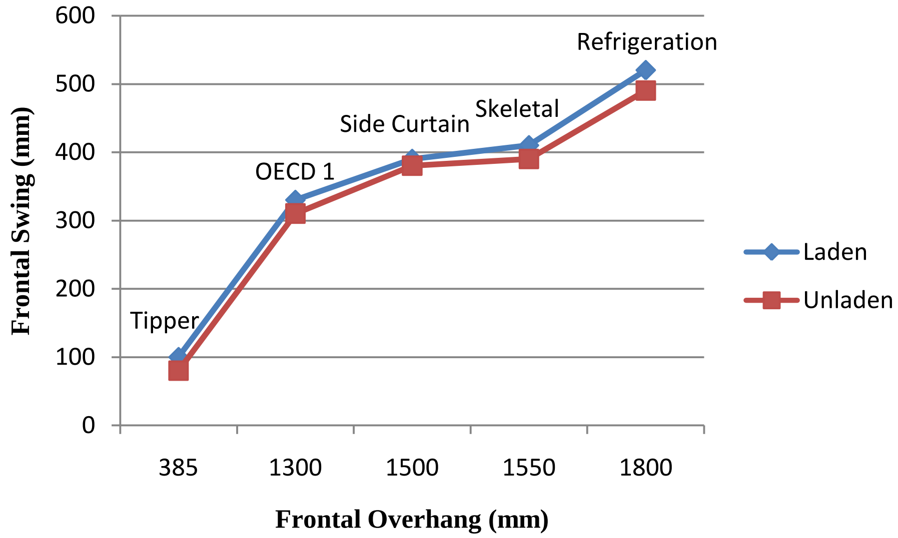


**Figure 5.7: The influence frontal overhang has on frontal swing MoD, for both laden and unladen conditions.**

Figure 5.7 above is an illustration of the relationship between frontal swing MoD and frontal overhang of the semi-trailer, under both laden and unladen conditions. It illustrates that in increase in frontal overhang of the semi-trailer results in a direct increase in the frontal swing, for both laden and unladen conditions. It also illustrates that mass also has an effect on the frontal swing of a vehicles, an increase in mass results in an increase of MoD frontal swing.

**Table 5.8: Frontal swing Part C, Difference of Maxima (DoM), performance results for five semi-trailer combinations**

| Vehicles | Results | | Frontal overhang (mm) | Pass / Fail |
|---|---|---|---|---|
| | Laden | Unladen | | |
| OECD 1 | 0.04 | 0.07 | 1300 | Pass |
| Skeletal | 0.12 | 0.16 | 1550 | Pass |
| Refrigeration | 0.25 | 0.27 | 1800 | Fail |
| Side Curtain | 0.10 | 0.14 | 1500 | Pass |
| Tipper | -0.28 | -0.23 | 385 | Pass |

Table 5.8 represents the results for Part C (Difference of Maxima, DoM) frontal swing for the five semi-trailer combination vehicles. The DoM results range from -0.28 m to 0.27 m, thus resulting in the four of the five vehicles (OECD1, Skeletal, Side Curtain and Tipper) achieving the 0.20 m required performance requirement, whilst the remaining vehicle (Refrigeration) did not satisfy this requirement and was therefore deemed not to comply. The failure of compliance increases the risk towards other road users, as the vehicle will require more lane width in order to perform a tight turn at low speeds. The resultant plots for the Part C Frontal swing can be found in Appendix C.2.3.2.

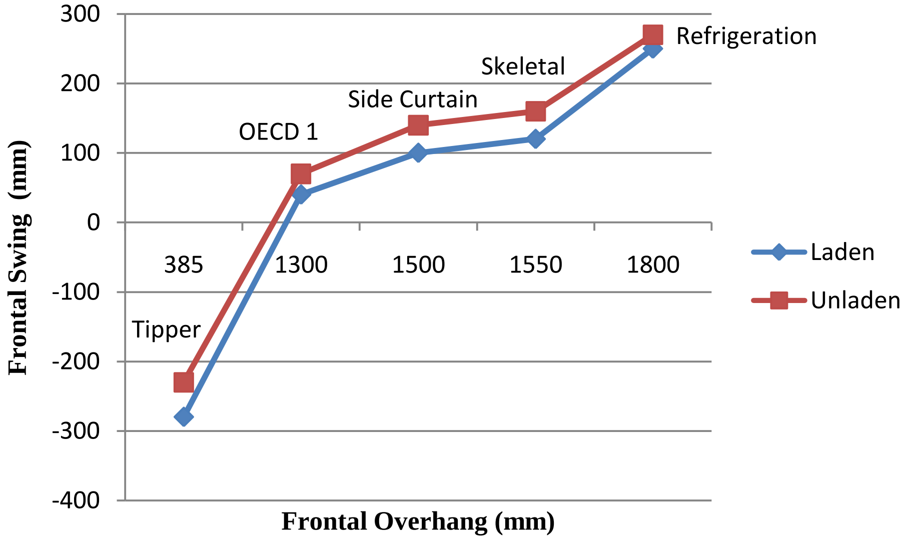


**Figure 5.8: The influence frontal overhang has on the frontal swing DoM, for both laden and unladen conditions**

Figure 5.8 is an illustration of the effect frontal overhang of the semi-trailer has on the frontal swing DoM, for both laden and unladen conditions. Similarly as with Figure 5.7 an increase in the frontal overhang results in an increase of frontal swing, it can also be seen that an increase in mass results positively on the DoM frontal swing of the vehicles.

From the above results it is evident that frontal overhang is the single most signification factor that influences frontal swing, other factors such as prime mover and trailer wheelbase, vehicle width, and mass, have a much lesser influence on the performance result. An increase in each factor would result in an increase in frontal swing.

Two of the five semi-trailer combination vehicles that were assessed did not achieve the required performance levels stipulated in the PBS guidelines, thus imposing a safety concern to other road users as well as road infrastructure, further research needs to be under taken in order to improve the frontal swing of these vehicles.

### 5.2.4 Tail Swing

The tail swing performance measure has been designed in order to limit the amount of road space a vehicle requires when performing a tight turn at low speed. The following section is a representation of the tail swing results for the five semi-trailer combination vehicles, at the entry and exit sections of the manoeuvre, under both laden and unladen conditions, as well as what factors influence this performance measure.

**Table 5.9: Tail swing performance results for five semi-trailer combinations**

| **Vehicle** | **Entry** | | **Exit** | | **Rear overhang (mm)** | **Level Passed** |
|---|---|---|---|---|---|---|
| | **Laden** | **Unladen** | **Laden** | **Unladen** | | |
| OECD 1 | 0.07 | 0.06 | No Swing Out | | 1540 | Level 1 |
| Skeletal | 0.04 | 0.03 | No Swing Out | | 1300 | Level 1 |
| Refrigeration | 0.12 | 0.10 | No Swing Out | | 2310 | Level 1 |
| Side Curtain | 0.05 | 0.04 | No Swing Out | | 1450 | Level 1 |
| Tipper | 0.03 | 0.02 | No Swing Out | | 858 | Level 1 |

Table 5.9 is a representation of the results for the tail swing of the five semi-trailer combination vehicles, illustrating the swing out under both laden and unladen conditions, for both the entry and exit section of the turn. The tail swing results range from 0 m (‘no swing out’ at the exit section of turn) to 0.12 m, thus all five vehicles, under both laden and unladen conditions, achieved the required performance requirement of less than 0.3 m, thus achieving Level 1 road classification. This ensures that when the vehicles assessed perform a tight turn at low speeds that the furthest rear most outside point of the last trailer does not track outside its specified lane, and as such does not pose a concern to other road users. The results of the tail swing performance measure can be seen in Appendix C.2.4.

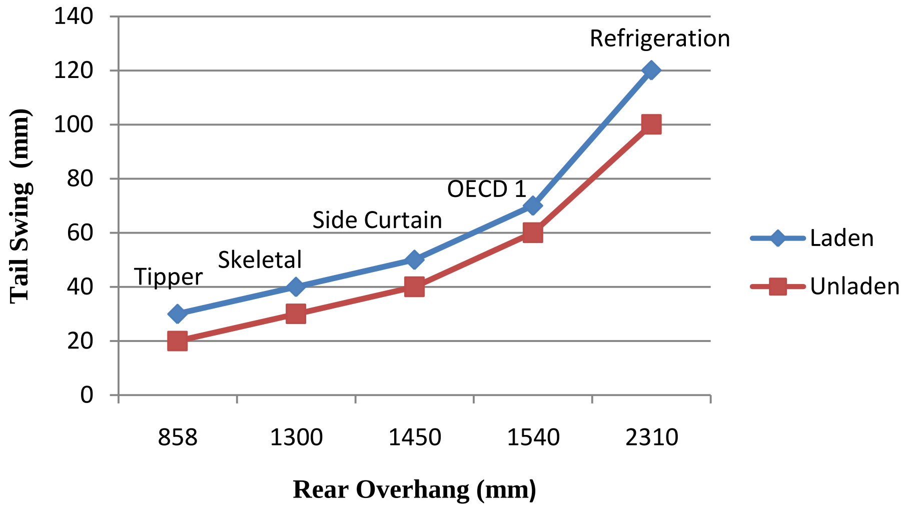


**Figure 5.9: The influence rear overhang has on the tail swing, for both laden and unladen conditions**

Figure 5.9 above is an illustration of the relationship between tail swing and rear overhang, for both laden and unladen conditions. It can be seen that an increase in rear overhang, which is the dominant influencing factor, results in an increase in vehicle tail swing, as well as the fact that an increase in vehicle mass has a negative effect on the vehicles tail swing performance.

Other factors which influence this performance measure include, width of the vehicle (an increase in vehicle width increases tail swing), and wheelbase of semi-trailer (an increase in vehicle wheelbase reduces the vehicles tail swing).

From these results it is evident that these five semi-trailer combination vehicles do not pose any safety concern for other traffic users or road side objects.

### 5.2.5 Steer Tyre Friction Demand

Steer tyre friction demand is designed to measure the possibility of a vehicle losing steering control when performing a tight turn at low speeds. The following section describes the results for the five semi-trailer combination vehicles, under both laden and unladen conditions, as well as the factors which influence this performance measure.

**Table 5.10: Steer tyre friction demand performance results for five semi-trailer combinations**

| Vehicle | Laden | | Unladen | | Pass / Fail |
|---|---|---|---|---|---|
| | LHS | RHS | LHS | RHS | |
| OECD 1 | 33.8 | 38.7 | 13.1 | 29.6 | Pass |
| Skeletal | 33.8 | 38.8 | 19.2 | 14.2 | Pass |
| Refrigeration | 33.5 | 37.7 | 21.1 | 16.7 | Pass |
| Side Curtain | 33.8 | 38.7 | 19.8 | 15.0 | Pass |
| Tipper | 33.6 | 37.8 | 20.7 | 16.2 | Pass |

Table 5.10 is a representation of the steer tyre friction demand results for the five semi-trailer combination vehicles; it illustrates the percentage steer tyre friction requirement for the left hand side (LHS) and right hand side (RHS) of each vehicle, under both laden and unladen conditions.

The results of this performance manoeuvre range from 13.1 % to 38.8 %, and as such all five vehicles assessed in this section achieved the required performance requirement of less than 80% of the available friction limit. The resultant plot for this performance measure can be seen in Appendix C.2.5

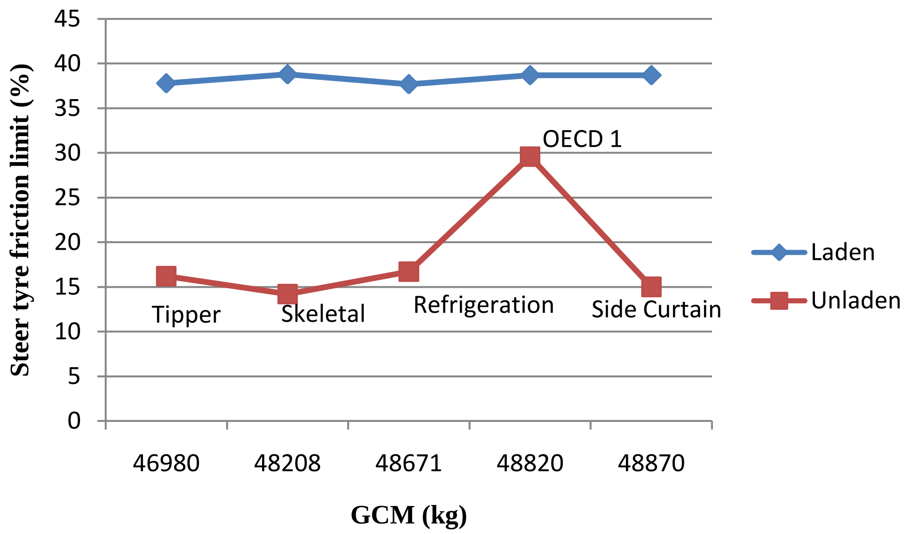


**Figure 5.10: The influence GCM has on the RHS steer tyre friction limit for both laden and unladen conditions**

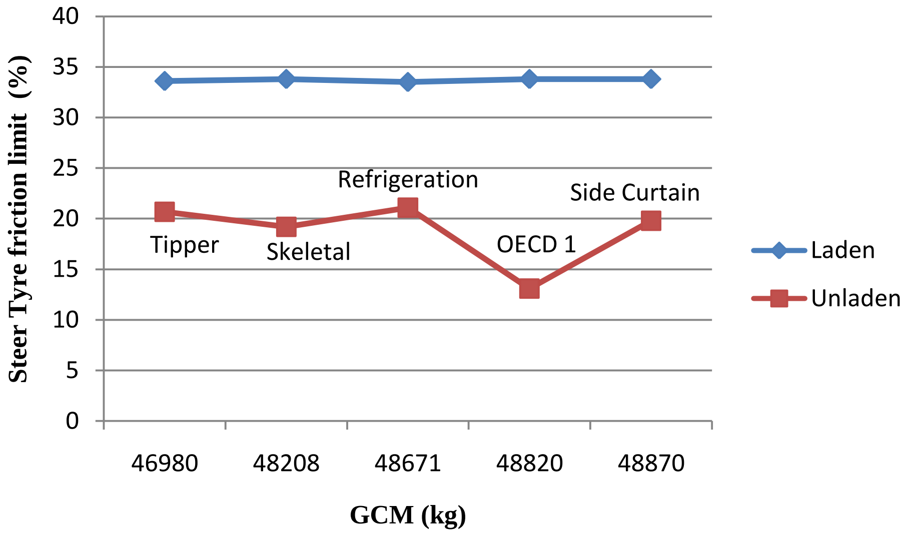


**Figure 5.11: The influence GCM has on the LHS steer tyre friction limit for both laden and unladen conditions**

Figures 5.10 and 5.11 illustrate the influence gross combination mass has on the steer tyre friction limit for the right and left hand side of the prime mover respectively, for the five semi-trailer combination vehicles, under both laden and unladen conditions.

These figures, although they do not show a significant change between each vehicle, they do illustrate the major increase in steer tyre friction requirement between the laden and unladen conditions. One of the factors which have a major influence on steer tyre friction demand is that of mass, as illustrated in Figure 5.10 and 5.11; an increase in mass has a dramatic increase in the required steer tyre friction demand.

The reason for the variation in trend for the OECD 1 vehicle in comparison to the other four vehicles, under both the laden and unladen condition, is due to the other influencing factors such as, increase wheelbase of prime mover, increase in drive axle group spread and as such a reduction in the steer axle load, all of these factors jointly influence the required OECD 1 vehicle steer tyre friction.

It is therefore deemed that none of these vehicles pose any concern with regard to a loss of steering.

This performance measure is generally of concern for vehicles with tri-axle drive units, however, due to the fact that this performance measure is determined from the same manoeuvre that analyses low speed swept path, frontal and tail swing, it has therefore has been included in the analysis.

### 5.2.6 Static Rollover Threshold

This performance measure is arguably the most important as it is strongly linked to rollover incidents, it is designed to measure the lateral acceleration a vehicle is capable of withstanding before rollover occurs. This section describes the results for two rollover tests, namely, circular and tilt table test, the percent deviation between them, as well as the various factors that influence this performance measure.

**Table 5.11: Static rollover threshold, Circular test, performance results for five semi-trailer combinations**

| Vehicle | Height of CG | Track width | Ratio TW/H | SRT | Pass / Fail |
|---|---|---|---|---|---|
| OECD 1 | 2.63 | 1.975 | 0.75 | 0.36 | Pass |
| Skeletal | 2.48 | 1.910 | 0.77 | 0.36 | Pass |
| Refrigeration | 2.82 | 1.910 | 0.68 | 0.33 | Fail |
| Side Curtain | 2.86 | 1.910 | 0.67 | 0.31 | Fail |
| Tipper | 2.38 | 1.910 | 0.80 | 0.40 | Pass |

Table 5.11 is a summary of the performance results for the static rollover circular test, it indicates the most important parameters concerned when analysing static rollover threshold, namely: height of CG above ground, track width and the ratio between them, the vehicles respective SRT values and their pass or fail compliance.

From this table it can be seen that the results from this performance measure range from 0.31g to 0.4g, thus indicating that three of the five vehicles (OECD 1, Skeletal and Tipper) achieved the minimum required performance measure, whilst the remaining two vehicles (Refrigeration and Side Curtain) did not achieve the minimum performance requirement of 0.35 g. The Tipper semi-trailer combination vehicle obtained a reading of 0.4 g due to its low centre of gravity. The results for the static rollover threshold circular test can be found in Appendix C.2.6.1.

A plot of static rollover threshold versus mass was developed, as with the previous performance measures, this however showed no significant correlation between the two parameters, and as such has not been included. However, numerous other factors influence the outcome of this performance measure namely: height of centre of gravity above ground, tyre track width of each vehicle unit, as well as suspension and tyre characteristics.

The two most important factors which influence SRT are the height of centre of gravity and the tyre track width, a reduction in the centre of gravity height or an increase in the track width results favourably to the roll stability of the vehicle. Due to the fact than many of the heavy vehicles make use of the maximum available track width; the CG height is the most significant parameter when looking at static rollover threshold. Whilst suspension and tyre characteristics do influence the vehicle stability it is negligible in comparison to that of CG height and track width.

Figure 5.12 illustrates the influence the ratio of overall track width to the height of CG above the ground has on the static rollover threshold for the five semi-trailer combination vehicles concerned. From this figure it can be seen that the ratio of overall track width to CG height is directly related to a vehicle SRT. An increase in this ratio has a positive influence on the Stability of all five semi-trailer combinations.

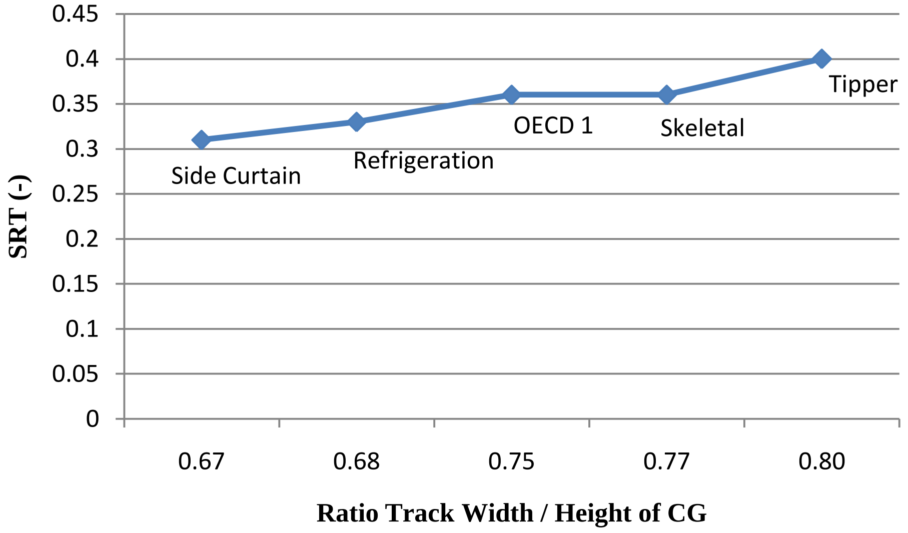


**Figure 5.12: The influence track width/ height of CG has on the SRT of five semi-trailer combinations**

**Table 5.12: Static rollover threshold, Tilt table test, performance results for five semi-trailer combinations**

| Vehicle | Time | Roll angle | SRT | Level Passed |
|---|---|---|---|---|
| OECD 1 | 10.48 | 25.9 | 0.36 | Pass |
| Skeletal | 10.40 | 25.2 | 0.35 | Pass |
| Refrigeration | 9.72 | 24.0 | 0.33 | Fail |
| Side Curtain | 9.28 | 23.0 | 0.31 | Fail |
| Tipper | 11.40 | 26.7 | 0.39 | Pass |

Table 5.12 is a summary of the performance results for the static rollover tilt table test, it indicates the time and roll angle each of the five vehicles were able to withstand during the simulation before rollover occurred, their respective static rollover threshold values and their pass or fail compliance.

The results of the tilt table test performance manoeuvre range from 0.31 g to 0.39 g, three of the five vehicles (OECD 1, Skeletal and Tipper) achieved the minimum required performance measure, whilst the remaining two vehicles (Refrigeration and Side Curtain) did not achieve the minimum performance requirement of 0.35 g. The results for the static rollover threshold tilt table test can be found in Appendix C.2.6.2.

These performance results closely mirror those of the circular test, indicating that the same three of the five vehicles passed the performance measure, whilst the remaining two vehicles failed. Table 5.13 below illustrates the percentage deviations between the two static rollover threshold tests for each of the five semi-trailer combination vehicles, these percentages range from 0.77 % to 3.47 %.

The variation in results between the two tests is due to the engine performance characteristics as well as the drive tyre slip at the commencement of rollover.

**Table 5.13: Percentage deviation between the Circular test and Tilt table test for the five semi-trailer combinations**

| Vehicle | % deviation |
|---|---|
| OECD 1 | 0.77 |
| Skeletal | 1.69 |
| Refrigeration | 1.61 |
| Side Curtain | 1.52 |
| Tipper | 3.47 |

Other factor which influence the static rollover of heavy vehicles include: chassis torsional flexibility, suspension and tyre characteristics, gross combination mass and length, however, these factors have a minor influence in comparison to those factors discussed above.

SRT is considered the most important performance measure with regard to PBS, failure to comply with this performance measure, indicates that the vehicle is unstable, and would require less lateral force to result in a roll over, in comparison to vehicles which achieved the 0.35g minimum limit. This would therefore pose a major safety concern to the driver and occupants of the vehicle, as well as road side objects and other road users. From these results it is evident that there is a need for further research and development, in order to improve heavy vehicle safety to acceptable international standards.

### 5.2.7 Rearward Amplification

The rearward amplification performance measure was developed in order to determine the lateral acceleration experienced by multi-articulated vehicles, when performing evasive manoeuvres at high speed. The following section described the results obtained for the five semi-trailer combination vehicles, as well as the parameters which influence this performance measure.

Table 5.14: Rearward amplification performance results for five semi-trailer combinations

| Vehicle | 5.7 x SRT | LA steer axle | LA rear unit | RA | Level Passed |
|---|---|---|---|---|---|
| OECD 1 | 2.05 | 0.16765 | 0.17964 | 1.07 | Pass |
| Skeletal | 2.00 | 0.16531 | 0.17167 | 1.04 | Pass |
| Refrigeration | 1.88 | 0.16519 | 0.1756 | 1.06 | Pass |
| Side Curtain | 1.77 | 0.16164 | 0.17771 | 1.10 | Pass |
| Tipper | 2.22 | 0.15805 | 0.17743 | 1.12 | Pass |

Table 5.14 above is a summary of the rearward amplification performance results for the five semi-trailer combination vehicles. This table includes the performance level required for each vehicle (5.7 x SRT), the lateral acceleration of the steer axle, the lateral acceleration of the centre of gravity of the rear-most vehicle unit, the rearward amplification result, as well as the vehicle pass / fail compliance.

The performance results for the five semi-trailer combination vehicles range from 1.04 – 1.12. From this table it can be seen that none of the five semi-trailer combination vehicle results exceeded the maximum performance requirement of 5.7 x SRT, thus were deemed to pass with this performance measure. The rearward amplification performance results can be found in Appendix C.2.7.

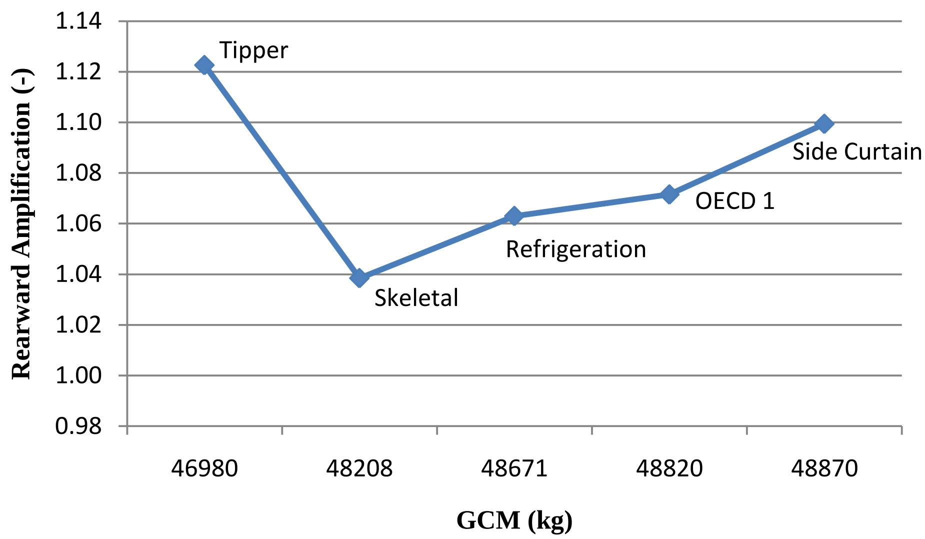


Figure 5.13: The influence GCM has on the RA of five semi-trailer combinations

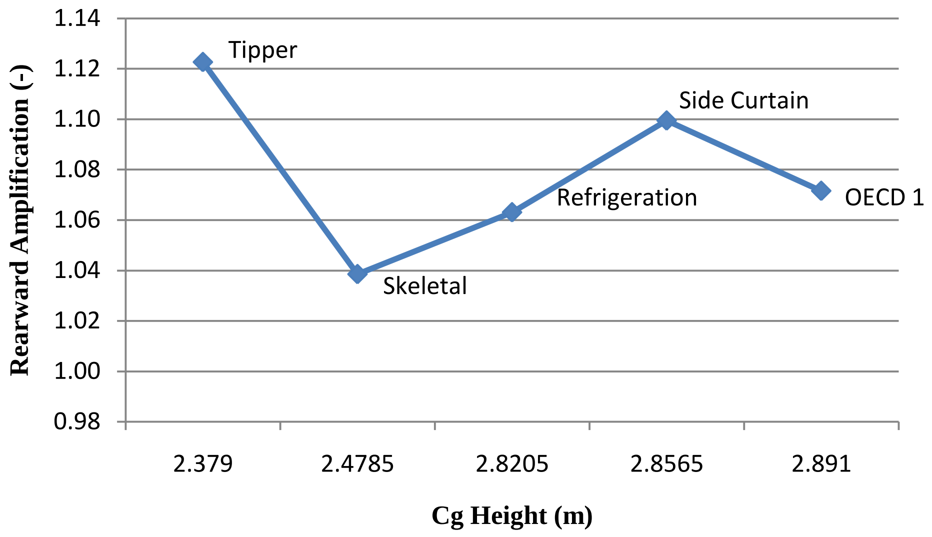


**Figure 5.14: The influence GC height has on the RA of five semi-trailer combinations**

Figures 5.13 and 5.14 above illustrate the influence gross combination mass and centre of gravity height have on the rearward amplification of the five semi-trailer combination vehicles, respectively. Figure 5.13 indicates that an increase in gross combination mass has a negative effect on the rearward amplification. However, the reason for the Tipper vehicle to deviate from this norm has to do with another significant influencing factor, that of semi-trailer wheelbase, a decrease in semi-trailer wheelbase increases the vehicles rearward amplification.

Although not clearly illustrated in Figure 5.14, an increase in vehicle centre of gravity height also impacts negatively on a vehicles rearward amplification. The reason for the variation of the Tipper semi-trailer is due to the semi-trailer wheelbase, as mentioned above, whilst the variation in OECD 1 vehicle is due to the increase in prime mover wheelbase; an increase in prime mover wheel base has a positive effect on a vehicle rearward amplification.

Other factors that influence rearward amplification include: chassis torsional flexibility (an increase in rigidity proves to positively improve rearward amplification), tyre cornering characteristics, as well as coupling lead or coupling rear overhang.

From the above results it is evident that all of the five semi-trailer combinations do not pose any concern for the rearward amplification performance manoeuvre.

### 5.2.8 High Speed Transient Off-tracking

This performance measure is designed to limit the lateral deviation of the last trailer axle of the last vehicle unit from the desired path, when performing an evasive manoeuvre at high speeds. This section provides a summary of the high speed transient off-tracking performance results for the five semi-trailer combination vehicles, as well as the parameters which influence this performance measure.

**Table 5.15: High speed transient off-tracking performance results for five semi-trailer combinations**

| Vehicle | GCM | CG height | Max Overshoot | Max target path | HSTO | Level Passed |
|---|---|---|---|---|---|---|
| OECD 1 | 48820 | 2.6252 | 1.66 | 1.46 | 0.2 | Level 1 |
| Skeletal | 48208 | 2.2898 | 1.65 | 1.46 | 0.19 | Level 1 |
| Refrigeration | 48671 | 2.5508 | 1.66 | 1.46 | 0.2 | Level 1 |
| Side Curtain | 48870 | 2.5718 | 1.67 | 1.46 | 0.21 | Level 1 |
| Tipper | 46980 | 2.1704 | 1.64 | 1.46 | 0.18 | Level 1 |

Table 5.15 provides a summary of the high speed transient off-tracking performance results for the five semi-trailer combination vehicles; it provides the gross combination mass and the centre of gravity height for each of the five semi-trailer combinations, as well as the high speed transient off-tracking result (Max overshoot less the Max target path) and the corresponding road classification level achieved by each vehicle.

The performance results for the five semi-trailer combination vehicles range from 0.18 m to 0.21 m, and as such all five of the semi-trailer combinations achieved the necessary performance measure of less than 0.6 m in order to classify for the Level 1 road classification. The high speed transient off-tracking performance results can be found in Appendix C.2.8.

Figures 5.15 and 5.16 below provide an illustration into the effect various parameters have on the outcome of high speed transient off-tracking. Figure 5.15 plots the high speed transient off-tracking versus gross combination mass for the five vehicles, from this figure it is evident that an increase in the gross combination mass has negative effect on the vehicle tracking capability.

Figure 5.16 plots the centre of gravity height versus high speed transient off-tracking, this similar to Figure 5.15, illustrates that an increase in the centre of gravity height has a negative effect on the vehicles tracking ability. The reason for the OECD 1 vehicle to be less than Side Curtain, even though it has a higher centre of gravity height, is due to the increase wheel base of the OECD prime mover in comparison to the MAN 26.480.

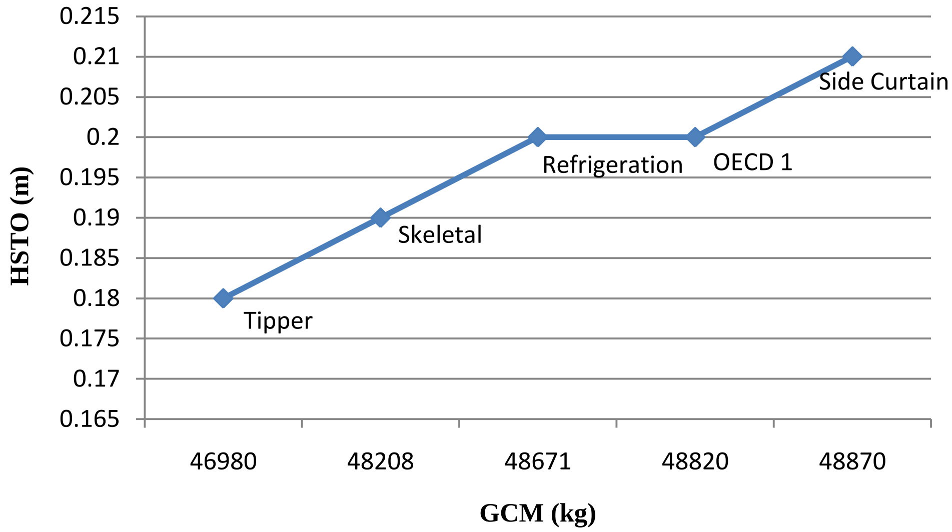


Figure 5.15: The influence GCM has on the HSTO of five semi-trailer combinations

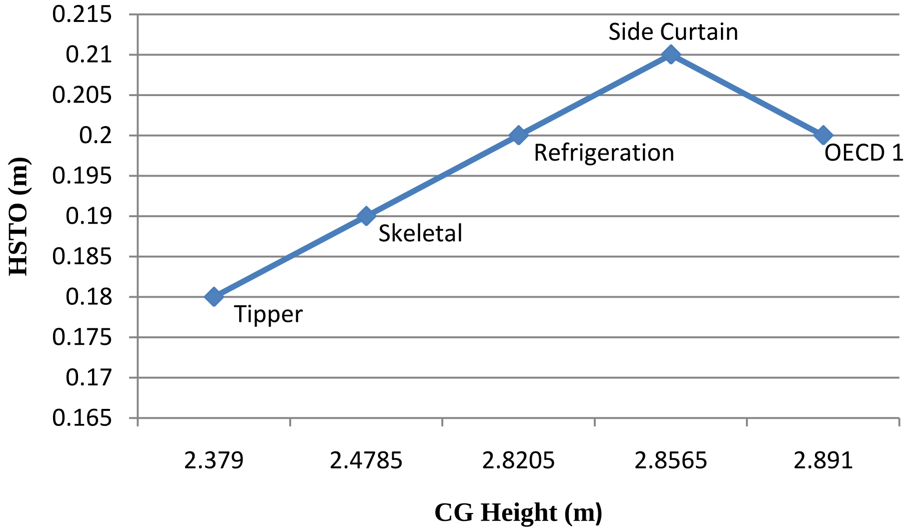


Figure 5.16: The influence CG height has on the HSTO of five semi-trailer combinations

Other factors that influence high speed transient off tracking include: tyre cornering stiffness, trailer wheel base and coupling lead or coupling rear overhang, an increase in each will have a positive effect on the vehicle tracking ability.

High speed transient off-tracking does not seem to pose any concern for this group of semi-trailer vehicle configurations analysed.

### 5.2.9 Yaw Damping Co-Efficient

This performance measure was designed in order to limit the time taken for oscillation to decay after a severe manoeuvre has been performed at high speed. The following section provides an overview of the yaw damping results for the five semi-trailer combination vehicles, as well as the factors which influence this performance measure.

**Table 5.16: Yaw damping performance results for five semi-trailer combinations**

| Vehicle | CG Height | Mass | Length | Wheelbase | Result | Level Passed |
|---|---|---|---|---|---|---|
| OECD 1 | 2.891 | 48820 | 17.745 | 10.0 | 0.379 | Pass |
| Skeletal | 2.4785 | 48208 | 17.5 | 10.0 | 0.279 | Pass |
| Refrigeration | 2.8205 | 48671 | 18.62 | 10.0 | 0.389 | Pass |
| Side Curtain | 2.8565 | 48870 | 17.5 | 10.0 | 0.402 | Pass |
| Tipper | 2.379 | 46980 | 16.058 | 9.0 | 0.408 | Pass |

Table 5.16 above provides a summary of the yaw damping performance measure for the five semi-trailer combination vehicles; it illustrates various parameters which influence this performance measure, the performance result, as well as the vehicles pass / fail compliance.

The performance results for the five semi-trailer combination vehicles range from 0.279 – 0.408 and as such all of the five semi-trailer vehicles assessed satisfy the necessary performance level of not less than 0.15. The yaw damping performance results for the semi-trailer combinations can be found in Appendix C.2.9.

Various factors that influence this performance measure include: centre of gravity height, gross combination mass, overall length, wheelbase, and tyre cornering characteristics, however no single parameter was seen to have a significant direct effect to the output results, thus a relationship between the performance result and influence parameter has not been plotted.

From these results it can be seen that all of the five semi-trailer combination vehicles passed this performance measure, and as such this performance measure does not pose any concern.

### 5.2.10 Directional Stability under Braking

This performance measure was designed in order to minimise the instability of a vehicle when braking in a turn or on a slope.

However, in accordance to Section C16.3 (b) ‘Deemed to comply provision’ of the PBS guidelines it states that ‘a vehicle that has a functioning anti-lock brake system that effectively prevents gross wheel lock-up on each axle group is defined to comply with this standard’.

All of the semi-trailer combination vehicles therefore comply with this standard, as under South African legislation all new trailers, tractors and trucks to have to have anti-lock brake systems in place.

## 5.3 B-Double

### 5.3.1 Tracking Ability on a Straight Path

The tracking ability of a heavy vehicle on a straight path is an important performance measure designed to determine the lateral deviation a vehicle experiences, from the desired path, when travelling at high speeds, under a worst case scenario.

Similar to the of TASP for semi-trailers (Section 5.2.1) various reference points were marked on the vehicle, these points were of concern for the validation of the simulation process as well as to measure the maximum and minimum lateral deviations. The reference points for each specific vehicle can be found in Table C.3 of Appendix C.3.1.

**Table 5.17: Tracking Ability on a Straight Path performance results for five B-double combinations**

| Vehicle | Min | Max | Swept Path | Result | Level Passed |
|---|---|---|---|---|---|
| OECD 2 | 1.2210 | 1.7780 | 2.9990 | 3.0 | Level 2 |
| Skeletal | 1.2094 | 1.7146 | 2.9240 | 3.0 | Level 2 |
| Cane | 1.2065 | 1.7581 | 2.9646 | 3.0 | Level 2 |
| Side Curtain | 1.2083 | 1.8129 | 3.0212 | 3.1 | Level 3 |
| Tipper | 1.2162 | 1.7110 | 2.9272 | 3.0 | Level 2 |

Table 5.17 is a summarised representation of the tracking-ability on a straight path performance results for the five B-double combinations; this table indicates the minimum and maximum lateral deviations of the reference points for each vehicle, the swept path (summation of the absolute values of the minimum and maximum lateral deviation), the results as well as the road classification level passed. The performance results for tracking-ability on a straight path B-double can be found in Appendix C.3.1.

The performance results from this table range between 2.924 m to 3.0212 m, for the 90 km/h vehicle simulation test speed. Four of the five vehicles (OECD 2, Skeletal, Cane and Tipper) achieved Level 2 the classification of not greater than 3.0 m, whilst the remaining vehicle (Side Tipper) achieved the Level 3 classification of not exceeding a swept width of 3.1 m. Failure to comply with Levels 1 and 2, the Side Tipper, therefore requires more lane width when travelling at high speeds down a straight even surfaced road. This increased tracking poses an increased safety risk to other road users and well as other road side objects.

As with the semi-trailer combinations, gross combination mass is one of the main factors that influence the vehicle tracking-ability, however, for the B-double combinations each of the five vehicles were restricted by mass to the maximum legal GCM limit of 56, 000 kg, therefore this relationship was not plotted.

Figure 5.17 below is an illustration of the influence centre of gravity height has on the tracking-ability of B-double combinations, similar to semi-trailer combinations an increase in centre of gravity height has a negative impact of the vehicles tracking-ability.

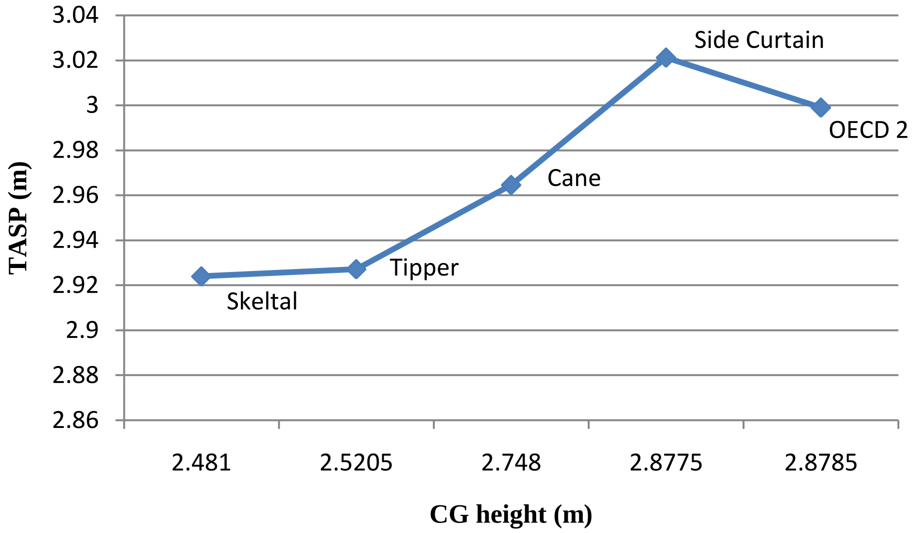


**Figure 5.17: The influence of Centre of Gravity (CG) height on the tracking-ability of B-double combinations**

The reason for the slight variation in trend between the Side Curtain and OECD 2 vehicles is due to the fact that the Side Curtain vehicle is slightly longer than that of the OECD 2 vehicle, hence the greater tracking result.

As stated earlier in the TASP of semi-trailer (Section 5.2.1) it is evident that the length of the vehicle and the number of articulation point do impose a negative effect on the vehicles tracking-ability.

Tracking-ability on a straight path for the five B-double configuration vehicles does not pose an immediate concern, as it tracks within the South African minimum lane width of 3.25m. However, it should be taken into consideration for longer vehicles operating at high mass limits under an abnormal load permit.

### 5.3.2 Low Speed Swept Path

This section provides a description of the results for the low-speed swept path performance manoeuvre for the five B-double configurations, as well as the factors which influence their performance measure.

Table 5.18: Low speed swept path performance results for five B-double combinations

| Vehicle | Unladen | Laden | Combined WB | Result | Passed |
|---|---|---|---|---|---|
| OECD 2 | 7.3074 | 7.1400 | 14.75 | 7.4 | Level 1 |
| Skeletal | 7.0996 | 7.1101 | 14.10 | 7.2 | Level 1 |
| Cane | 7.5431 | 7.5630 | 15.24 | 7.6 | Level 2 |
| Side Curtain | 7.0990 | 7.1095 | 14.05 | 7.2 | Level 1 |
| Tipper | 6.9532 | 6.9902 | 13.96 | 7.0 | Level 1 |

Table 5.18 above provides a summary of the low-speed performance results for the five B-double configurations, under both laden and unladen conditions, as well as the road classification level achieved.

The performance results for the five B-double configurations range from 6.9532 m to 7.563 m; Four of the five vehicles (OECD2, Skeletal, Side Curtain and Tipper) achieved a performance requirement of less than 7.4 m and thus qualify for the Level 1 road classification, whilst the remaining vehicle (Cane) qualified for the Level 2 road classification, by achieving a swept path of less than 8.7m

The resultant plots of the performance measure can be seen in Appendix C.3.2, however, it must be noted that these performance plots do not incorporate the overall vehicle width of 2.6 m.

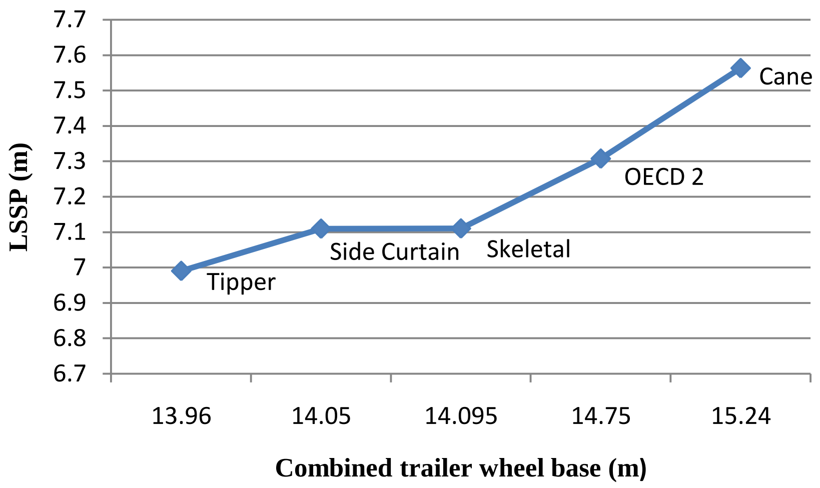


Figure 5.18: The influence of wheelbase on LSSP for the five B-double configurations

Figure 5.18 above illustrates the influence wheelbase has on the low-speed swept path performance results for the five B-double vehicle configurations. Trailer wheelbase is the single most significant influence with regard to vehicle tracking-ability at low speed, an increase in trailer wheelbase has a negative influence on the vehicle swept path.

Other factors that influence low-speed swept path include: overall length, gross combination mass, frontal overhang of the hauling unit and coupling rear overhang, an increase in each of these parameters has a negative influence on the tracking capability of a vehicle.

From these results it is evident that low-speed swept path does not pose any concern for these five B-double configuration vehicles assessed.

### 5.3.3 Frontal Swing

This section provides a description of the frontal swing results for Part A (hauling unit), Part B (Maximum Difference) and Part C (Difference of Maxima) for the five B-double configuration vehicles, under both laden and unladen conditions, as well as the various factors which influence this performance measure.

**Table 5.19: Frontal swing Part A, Hauling unit, performance results for five B-double combinations**

| Vehicle | Result | | Level Passed |
|---|---|---|---|
| | Laden | Unladen | |
| OECD 2 | 0.42 | 0.37 | Pass |
| Skeletal | 0.45 | 0.39 | Pass |
| Cane | 0.46 | 0.41 | Pass |
| Side Curtain | 0.45 | 0.40 | Pass |
| Tipper | 0.45 | 0.40 | Pass |

Table 5.19 above is a representation of the results for Part A (hauling unit) frontal swing for the five B-double configurations, under both laden and unladen conditions. The results range from 0.37 m to 0.46 m, thus ensuring that all five vehicles satisfy the 0.7 m performance requirement. The resultant plots for this performance measure can be seen in Appendix C.3.3.1.

Due to the same prime mover being used for the semi-trailer and B-double configurations, similar conclusions can be drawn. An increase in the frontal overhang or an increase in mass will have a negative impact on the frontal swing of the prime mover. Similarly as in Section 5.2.3, it can be seen from Table 5.19 that an increase in mass results in an increase in Part A frontal swing.

**Table 5.20: Frontal swing Part B, Maximum of Difference (MoD), performance results for B-double combinations**

| Vehicles | Results | | Frontal overhang (mm) | Level Passed |
|---|---|---|---|---|
| | Laden | Unladen | | |
| OECD 2 | 0.33 | 0.32 | 1755 | Pass |
| Skeletal | 0.31 | 0.29 | 1755 | Pass |
| Cane | 0.39 | 0.37 | 1800 | Pass |
| Side Curtain | 0.34 | 0.33 | 1855 | Pass |
| Tipper | 0.18 | 0.17 | 972 | Pass |

Table 5.20 is a representation of the frontal swing performance results for Part B (Maximum of Difference, MoD) frontal swing for the B-double configurations, under both laden and unladen conditions. The MoD frontal swing results range from 0.17 m to 0.39 m, thus resulting in all of the five vehicles achieving the required performance requirement of less than the 0.4 m. The resultant plot for this performance measure can be seen in Appendix C.3.3.2.

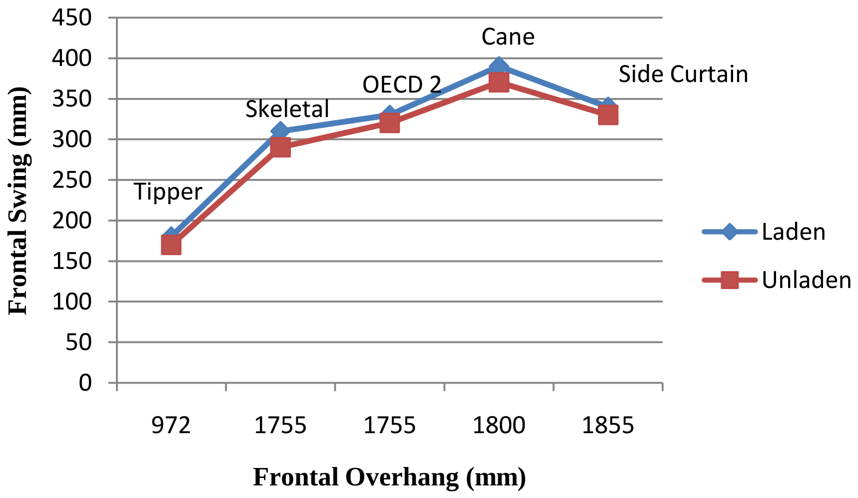


**Figure 5.19: The influence frontal overhang has on the frontal swing, for both laden and unladen conditions**

Figure 5.19 above is an illustration of the relationship between frontal swing and frontal overhang of the first trailer, under both laden and unladen conditions. It illustrates that in increase in frontal overhang of the first trailer results in a direct increase in the frontal swing. Figure 5.19 also illustrates that mass has a negative effect on the frontal swing of a vehicles, an increase in mass results in an increase of MoD frontal swing.

The reason for the variation in expected frontal swing for the Side Curtain vehicle is due to the secondary contributing factor concerned with frontal overhang, that of vehicle wheelbase. The reduction in wheelbase between the Cane and Side Curtain vehicles resulted in a lower frontal swing out for the Side Curtain B-double configuration vehicle.

**Table 5.21: Frontal swing Part C, Difference of Maxima (DoM), performance results for five B-double combinations**

| Vehicles | Results | | Frontal overhang (mm) | Level Passed |
|---|---|---|---|---|
| | Laden | Unladen | | |
| OECD 2 | 0.07 | 0.11 | 1755 | Pass |
| Skeletal | 0.06 | 0.08 | 1755 | Pass |
| Cane | 0.13 | 0.15 | 1800 | Pass |
| Side Curtain | 0.10 | 0.11 | 1855 | Pass |
| Tipper | -0.16 | -0.12 | 972 | Pass |

Table 5.21 represents the results for Part C (Difference of Maxima, DoM) frontal swing for the five B-double configurations, under both laden and unladen conditions. The DoM results range from -0.16 m to 0.15 m, thus resulting in the all of the five vehicles achieving the required performance measure of less than 0.2 m. The resultant plot for this performance measure can be seen in Appendix C.3.3.2.

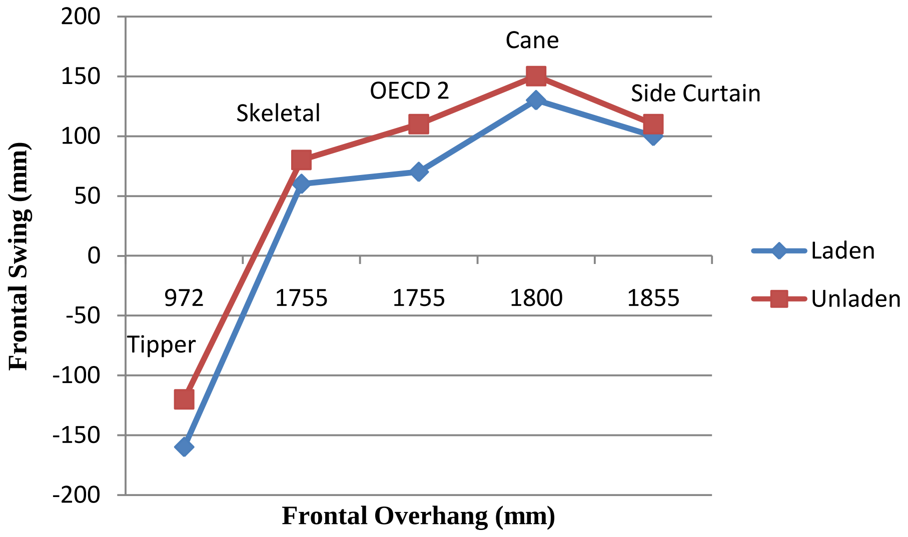


**Figure 5.20: The influence frontal overhang has on the frontal swing, for both laden and unladen conditions**

Figure 5.20 is an illustration of the effect frontal overhang of the first trailer has on the frontal swing, for both laden and unladen conditions. Similarly as with Figure 5.19, an increase in the DoM frontal overhang of the first trailer results in an increase of frontal swing, it can also be seen that an increase in mass results positively on the DoM frontal swing of the vehicles

As with frontal swing MoD, the increase in wheelbase for the Side Curtain B-double resulted in a variation in trend relating to a decrease in frontal swing. From results it is evident that frontal swing does not pose a concern for the five B-double configuration vehicles analysed.

### 5.3.4 Tail Swing

The following section describes the tail swing results obtained from the five B-double configuration vehicles, at both the entry and exit sections of the manoeuvre, under both laden and unladen conditions, as well as the various factors which influence this performance measure.

**Table 5.22: Tail swing performance results for five B-double combinations**

| Vehicle | Entry | | Exit | | Rear overhang (mm) | Level Passed |
|---|---|---|---|---|---|---|
| | Laden | Unladen | Laden | Unladen | | |
| OECD 2 | 0.02 | 0.02 | No Swing Out | | 1773 | Level 1 |
| Skeletal | 0.02 | 0.02 | No Swing Out | | 1773 | Level 1 |
| Cane | 0.00 | 0.00 | No Swing Out | | 735 | Level 1 |
| Side Curtain | 0.02 | 0.02 | No Swing Out | | 1805 | Level 1 |
| Tipper | 0.02 | 0.01 | No Swing Out | | 1520 | Level 1 |

Table 5.22 is a summarised representation of the tail swing performance results for the five B-double configurations analysed, illustrating the vehicle swing out for both the entry and exit sections of the manoeuvre, under laden and unladen conditions, the rear overhang, as well as the level passed.

The results range from 0 m (‘no swing out’ at the exit section of turn) to 0.02 m, thus indicating that all of the five B-double vehicle configurations achieved the required performance requirement of less than 0.3 m, and as such qualify for Level 1 road classification. The resultant plot for this performance measure can be seen in Appendix C.3.4.

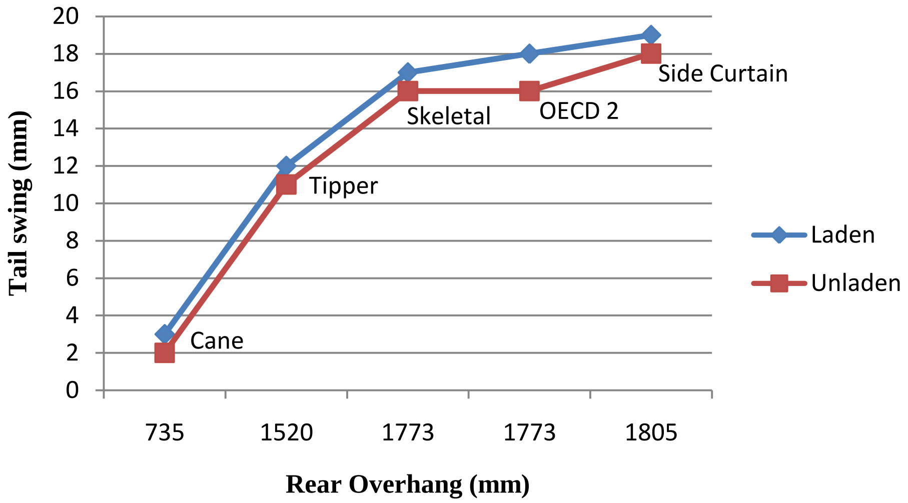


Figure 5.21: The influence rear overhang has on the tail swing, for both laden and unladen conditions

Figure 5.21 above illustrates the influence rearward overhang has on the tail swing for the five B-double vehicle configurations, under both laden and unladen conditions. An increase in rear overhang has a negative effect on the tail swing of the five B-double configurations.

Other factors which influence the tail swing performance measure include; width of the vehicle (an increase in vehicle width increases tail swing), and wheelbase of the unit with the critical rear overhang dimension (an increase in vehicle wheelbase reduces the vehicles tail swing).

From these results it can be seen that tail swing does not pose any concern for the five B-double configuration vehicles assessed.

### 5.3.5 Steer Tyre Friction Demand

This section describes the steer tyre friction demand results for the five B-double configuration vehicles, under both laden and unladen conditions, as well the factor which influence this performance manoeuvre.

Table 5.23: Steer tyre friction demand performance results for five B-double combinations

| Vehicle | Laden | | Unladen | | Level Passed |
|---|---|---|---|---|---|
| | LHS (%) | RHS (%) | LHS (%) | RHS (%) | |
| OECD 2 | 35.0 | 40.0 | 23.4 | 20.3 | Pass |
| Skeletal | 31.6 | 35.2 | 21.0 | 17.2 | Pass |
| Cane | 32.2 | 36.8 | 23.4 | 29.3 | Pass |
| Side Curtain | 34.7 | 40.2 | 21.7 | 18.1 | Pass |
| Tipper | 31.4 | 35.2 | 22.5 | 19.3 | Pass |

Table 5.23 is a summarised representation of the steer tyre friction demand performance results for the five B-double vehicle configurations; it illustrates the steer tyre friction percentages for the left hand side (LHS) and the right hand side (RHS) of each vehicle, under both laden and unladen conditions, as well as the pass or fail compliance.

The results of this performance measure range from 17.2 % to 40.2 %, thus all of the five B-double configuration vehicles analysed achieved the required performance level of less than 80% of the available steer tyre friction limit. The resultant plots for this performance measure can be seen in Appendix C.3.5.

Unfortunately plots similar to those of Section 5.2.5 could not be developed, as all five B-double combination vehicles were restricted by mass to the maximum legal gross combination mass limit of 56 tons, however, it can be clearly see that the steer tyre friction limit for both the left and right hand side of the vehicle increase dramatically under laden conditions compared to that of unladen.

Other factors that influence steer tyre friction demand performance measure include: wheelbase and steer axle load of the prime mover (an increase in each would result in a decrease in friction demand), and drive group axle spread (increase in this parameter would result in an increase in required friction limit).

From these results it can be seen that steer tyre friction demand performance measure is of no concern for dual axle drive units, as all five vehicles are well below the 80% steer tyre friction demand limit.

### 5.3.6 Static Rollover Threshold

This section describes the results for two rollover tests, namely: circular and tilt table test, the percent deviation between them, as well as the various factors that influence this performance measure.

Table 5.24: Static rollover threshold, Circular test, performance results for five B-double configurations

| Vehicle | Height of CG | Track width | Ratio TW/H | SRT | Level Passed |
|---|---|---|---|---|---|
| OECD 2 | 2.88 | 1.975 | 0.69 | 0.35 | Pass |
| Skeletal | 2.49 | 1.91 | 0.77 | 0.35 | Pass |
| Cane | 2.75 | 1.91 | 0.70 | 0.37 | Pass |
| Side Curtain | 2.88 | 1.91 | 0.66 | 0.32 | Fail |
| Tipper | 2.53 | 1.91 | 0.76 | 0.37 | Pass |

Table 5.24 is a summary of the performance results for the static rollover circular test for the five B-double configurations, it indicates the most important parameters concerned when analysing static rollover threshold, namely: height of CG above ground, track width and the ratio between them, the vehicles respective static rollover threshold values and their pass or fail compliance.

The results from this performance measure range from 0.32 g to 0.37 g, thus indicating that four of the five vehicles (OECD 2, Skeletal, Cane and Tipper) achieved the required minimum performance requirement, whilst the remaining vehicle (Side Curtain) did not achieve the minimum performance requirement of 0.35 g. The results for the static rollover threshold circular test can be found in Appendix C.3.6.1.

Similarly to that of the semi-trailer the two most important factors are the height of centre of gravity and the tyre track width. Figure 5.22 illustrates the influence the ratio of overall track width to the height of CG above the ground has on the static rollover threshold for the five B-double configurations concerned.

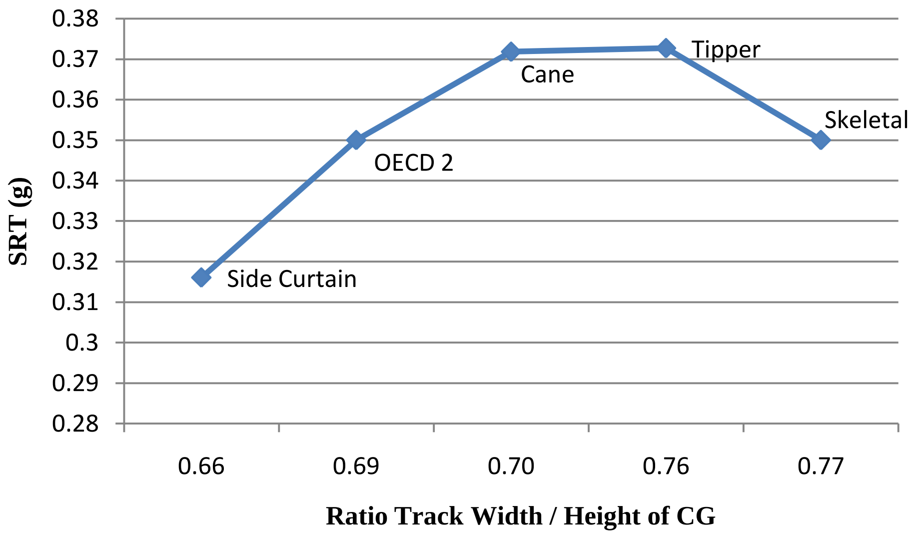


Figure 5.22: The influence track width/ height of CG has on the SRT of five B-double configurations

**Table 5.25: Static rollover threshold, Tilt table test, performance results for five B-double configurations**

| Vehicle | Time | Roll angle | SRT | Level Passed |
|---|---|---|---|---|
| OECD 2 | 10.6 | 26.48 | 0.36 | Pass |
| Skeletal | 10.64 | 27.63 | 0.36 | Pass |
| Cane | 11.28 | 28.95 | 0.38 | Pass |
| Side Curtain | 9.72 | 26.33 | 0.33 | Fail |
| Tipper | 11.36 | 28.49 | 0.38 | Pass |

Table 5.25 is a summary of the performance results for the static rollover tilt table test for the five B-double configurations, it indicates the time and roll angle each of the five vehicle were able to withstand during the simulation before rollover occurred, their respective static rollover threshold values and their pass or fail compliance.

The results from Table 5.25 range from 0.33 g to 0.38 g, four of the five vehicles (OECD 1, Skeletal, Cane and Tipper) achieved the required minimum performance requirement, whilst the remaining vehicle (Side Curtain) did not achieve the minimum performance requirement of 0.35 g. The results for the static rollover threshold tilt table test for the five B-double configurations can be found in Appendix C.3.6.2.

These results closely mirror those of the circular test, indicating that the same four of the five vehicles passed the performance measure, whilst the remaining vehicle did not achieve the required performance level.

Table 5.26 below illustrates the percentage deviations between the two static rollover threshold tests for each of the five B-double combination vehicles, these percentages range from 3.1 % to 5.0 %. The reason for the deviation between the two performance tests is due to the drive-train characteristics, for example tractive forces slow of drive during rollover due to loss of tyre contact surface area etc.

**Table 5.26: Percentage deviation between the Circular test and Tilt table test for the five B-double configurations**

| Vehicle | % deviation |
|---|---|
| OECD 2 | 3.1 |
| Skeletal | 3.6 |
| Cane | 3.1 |
| Side curtain | 5.0 |
| Tipper | 3.4 |

Other factors which influence the static rollover of B-double configuration vehicles are similar to those mentioned in Section 5.2.6.

SRT is considered the most important performance measure with regard to PBS, failure to comply with this performance measure, indicates that the vehicle is dynamically unstable, and would require less lateral force to result in a roll over, in comparison to vehicles which achieved the 0.35g minimum limit. This would therefore pose a major concern to the safety of the vehicle driver and occupants, as well as road side objects and other road users. From these results it can be seen that there is a need for further research into the rollover stability of heavy vehicles in South Africa.

### 5.3.7 Rearward Amplification

The following section provides a summary of the rearward amplification performance results for the five B-double configuration vehicles, as well as the factors which influence this performance manoeuvre.

**Table 5.27: Rearward amplification performance results for five B-double combinations**

| Vehicle | 5.7 x SRT | LA steer axle | LA rear unit | RA | Level Passed |
|---|---|---|---|---|---|
| OECD 2 | 2.06 | 0.1729 | 0.1873 | 1.08 | Pass |
| Skeletal | 2.07 | 0.1670 | 0.1823 | 1.09 | Pass |
| Cane | 2.19 | 0.1593 | 0.1693 | 1.06 | Pass |
| Side Curtain | 1.90 | 0.1661 | 0.1841 | 1.11 | Pass |
| Tipper | 2.20 | 0.1560 | 0.2083 | 1.34 | Pass |

Table 5. above is a summary of the rearward amplification performance results for the five B-double configuration vehicles; it illustrates the lateral acceleration of the steer axle and the rear unit centre of gravity, the performance level required for each vehicle (5.7 x SRT), rearward amplification performance results and the pass or fail compliance.

The performance results for the five B-double configuration vehicles range from 1.06 – 1.34. From this table it can be seen that all of the five B-double configuration vehicle results did not exceed the maximum performance requirement of 5.7 x SRT, thus all five vehicles were deemed to pass with this performance measure.

The performance output plots of the lateral acceleration for the steering axle, and centre of gravity for vehicle units 2 and 3 from Trucksim, as can be seen in Appendix C.3.7. These results were discontinuous and fluctuated significantly due to the lack of capability of the vehicle driver model, and the lash in the vehicle hitches.

The output data was then used to generate a 'best fit' polynomial plot in Matlab, this provided a much smoother output data plot, as can be seen in Figure C.59. From these plots the maximum lateral acceleration could be determined and the rearward amplification calculated.

Figures 5.23 and 5.24 below illustrate the influence centre of gravity height and gross combination mass has on the rearward amplification of the five B-double configurations, respectively. Although the centre of gravity generally negatively influences the rearward amplification, this is not evident in this instance due to the variation in prime mover and trailer wheelbase.

However, Figure 5.24 illustrates a slight tendency that an increase in the combined trailer wheelbase has a positive influence on the rearward amplification of the five B-double combination vehicles.

From these results it is therefore evident that rearward amplification does not pose any concern for the five B-double configurations analysed.

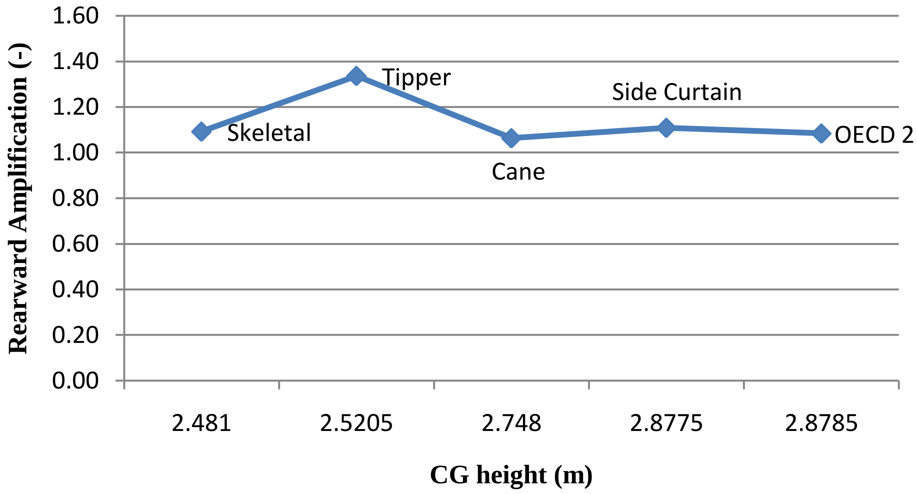


**Figure 5.23: The influence CG height has on the RA of the five B-double combinations**

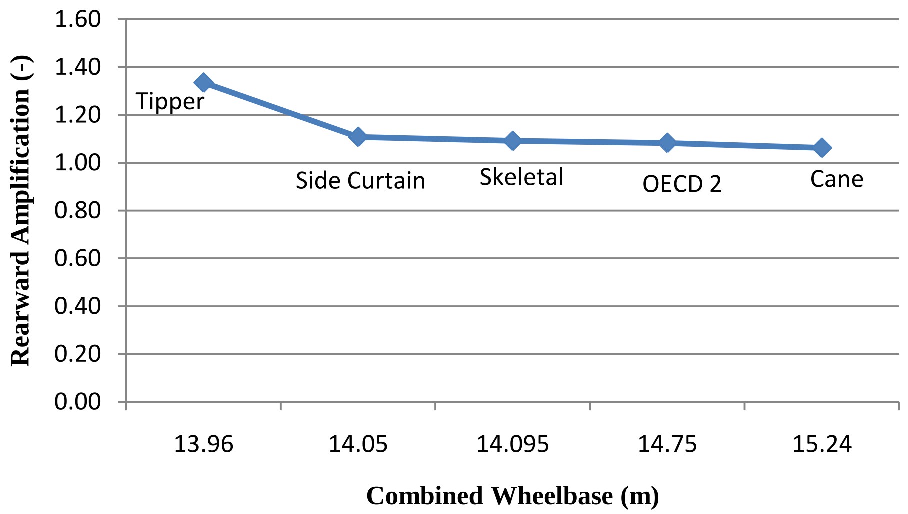


**Figure 5.24: The influence combined trailer wheelbase has on the RA of the five B-double combinations**

### 5.3.8 High Speed Transient Off-tracking

This section provides a summary of the high-speed transient off-tracking performance results for the five B-double vehicle configurations, as well as the parameters which influence this performance measure.

**Table 5.28: High speed transient off-tracking performance results for five B-double combinations**

| Vehicle | CG Height | Max Overshoot | Max target path | HSTO | Level Passed |
|---|---|---|---|---|---|
| OECD 2 | 2.8785 | 1.80 | 1.46 | 0.34 | Level 1 |
| Skeletal | 2.481 | 1.77 | 1.46 | 0.31 | Level 1 |
| Cane | 2.748 | 1.73 | 1.46 | 0.27 | Level 1 |
| Side Curtain | 2.8775 | 1.78 | 1.46 | 0.32 | Level 1 |
| Tipper | 2.5205 | 1.78 | 1.46 | 0.32 | Level 1 |

Table 5.28 provides a summary of the five B-double vehicle configurations performance results; it provides centre of gravity height for each of the five B-double configurations, as well as the high-speed transient off-tracking performance results (which is Max overshoot less the Max target path) and the corresponding road classification level which each vehicle achieved.

The performance results for the five B-double configurations range from 0.27 m to 0.34 m, and as such all five of the vehicle configurations achieved the necessary performance measure of less than

0.6 m in order to classify for the Level 1 road classification. The resultant plot for this performance measure can be seen in Appendix C.3.8.

Figure 5.25 plots the centre of gravity height versus high-speed transient off-tracking, this illustrates that an increase in the centre of gravity height has a slight tendency to cause a negative effect on the vehicles tracking-ability.

The reason for the variation in trend of the Cane vehicle from the remaining four B-double configurations is because this vehicle configuration has a larger first trailer wheelbase, a high unsprung centre of gravity, a low payload and a small rear overhang in comparison to the other four configurations, all of these factors combined influence the vehicles improved tracking capability.

Other factors which influence high speed transient off-tracking are similar to those mentioned in rearward amplification section 5.2.8 above.

From these results it can seen that the five B-double configuration vehicles assessed do not pose concern for the high-speed transient off tracking performance manoeuvre.

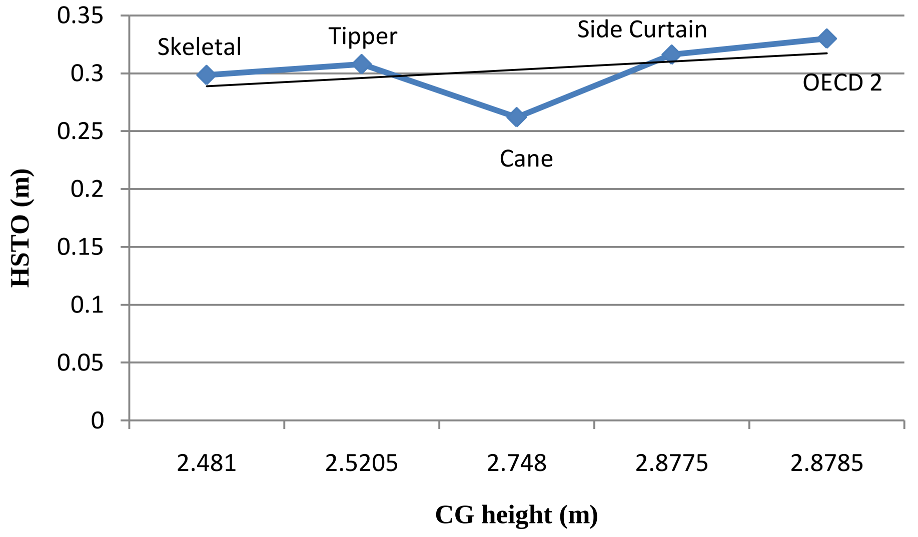


**Figure 5.25: The influence centre of gravity has on the HSTO of the five B-double combinations**

### 5.3.9 Yaw Damping Co-Efficient

This performance measure was designed in order to limit the time taken for oscillation to decay after a severe manoeuvre has been performed. The following section provides an overview of the yaw damping performance results for the five B-double configuration vehicles, as well as the factors which influence this performance measure.

**Table 5.29: Yaw damping performance result for five B-double configurations**

| Vehicle | CG Height | Mass | Length | Wheelbase | Result | Level Passed |
|---|---|---|---|---|---|---|
| OECD 2 | 2.8785 | 56000 | 21.983 | 14.75 | 0.256 | Pass |
| Skeletal | 2.481 | 56000 | 22.168 | 14.095 | 0.284 | Pass |
| Cane | 2.748 | 56000 | 22.193 | 15.24 | 0.199 | Pass |
| Side Curtain | 2.8775 | 56000 | 22.235 | 14.05 | 0.188 | Pass |
| Tipper | 2.5205 | 56000 | 21.805 | 13.96 | 0.214 | Pass |

Table 5.29 above provides a summary of the yaw damping performance results for the five B-double configuration vehicles; it illustrates various parameters which influence the performance measure, the performance result, as well as the vehicles pass or fail compliance.

The performance results for the five B-double configuration vehicles range from 0.188 – 0.284 and as such all of the five vehicles assessed satisfy the necessary performance level of not less than 0.15. The yaw damping performance results for the B-double configurations can be found in Appendix C.3.9.

Various factors that influence this performance measure are similar to those mentioned in Section 5.2.9 above.

From these results it can be seen that all of the five B-double configuration vehicles passed this performance measure, and as such this performance measure does not pose any concern.

### 5.3.10 Directional Stability under Braking

This performance measure was designed in order to minimise the instability of a vehicle when braking in a turn or on a slope.

However, in accordance to Section C16.3 (b) 'Deemed to comply provision' of the PBS guidelines it states that 'a vehicle that has a functioning anti-lock brake system that effectively prevents gross wheel lock-up on each axle group is defined to comply with this standard'.

All of the B-double configurations therefore comply with this standard, as under South African legislation all new trailers, tractors and trucks to have to have anti-lock brake systems in place.

## 5.4 Validation

This section of the report provides a means for validating the results obtained through computational simulation. Two forms of validation were considered during the course of this research, namely: physical testing and an analytical approach. Unfortunately due to financial constraints the comprehensive physical testing of ten 56 ton heavy vehicles was not seen as a feasible option, thus an analytical approach was undertaken in order to validate the output results obtained.

Validation is a necessity, in order to ensure that the computational simulation models that were developed accurately resemble ‘real world’ circumstances.

The following section proves to validate the simulation results achieved through computational simulation by comparing them to the desired simulation outputs from an OECD research study previously undertaken by PBS certified assessors.

### 5.4.1 Organisation for Economic Co-operation and Development

OECD is an organisation aimed to unify international governments, with a joint interest in development and economic growth, by comparing, analysing and forecasting development and research in various fields.

The OECD committee is made up of representatives from each of the 30 member countries, establishing working and expert groups, who discuss, contribute, review and implement various studies undertaken worldwide.

A recent study undertaken by OECD and the International Transport Forum (ITF) was to benchmark the current international heavy vehicles fleet from member countries, according to their vehicle dynamic safety performance characteristics.

The OECD report was conducted in order to accurately and informatively compare the performance of heavy vehicles on a broad based international scale from 11 different countries.

Significant variations in heavy vehicle design exist across the world due to differing legislation with regard to vehicle dimensions, axle load and vehicle gross combination mass. A benchmarking study was undertaken in order to analyse these heavy vehicles on a comparative level. The study required each participating country to submit a minimum of three different vehicles according to specified guidelines. A total of 39 heavy vehicles, from rigid truck and draw bar vehicles to 12 axles articulated road trains, were dynamically analysed and their results published. A full list of all the analysed vehicles can be found in Table D.1 of Appendix D.

The study analysed the 39 vehicles according to the Performance Based Standards (PBS) scheme developed by the Australian National Transport Commission (NTC).

Of the 16 safety standards developed by the performance based standards scheme, six were selected, these standards include:

- Tracking-ability on a straight path
- Low speed swept path
- Steer tyre friction demand
- Static rollover threshold
- Rearward amplification
- High speed transient off-tracking

The remaining ten standards were not selected for the purpose of this study for various reasons, as they were difficult to compare and they were believed not too add much to the outcome of the benchmarking study. The Australian Road Research Board, or ARRB group, were contracted to undertake the modelling and vehicle dynamics simulation assessments. ARRB made use of their own in-house software, developed from Autosim 2.80, the same generic software that was used to develop Trucksim in order to undertake the computational simulations.

### 5.4.2 Vehicles

South Africa submitted four of their main workhorse vehicles for comparative analysis, namely ZA 1, ZA 2, ZA 3, and ZA 4 corresponding to a six axle articulated semi-trailer, 7 axle articulated B-double, 5 axle articulated semi-trailer and a 8 axle articulated B-double, respectively.

Of these four vehicles, ZA 1 and ZA 2 were selected for analysis and verification process, and are referred to in this report as OECD 1 and OECD 2, respectively. The computational models developed were constructed from data supplied to the OECD group by the member organisation, these design parameters can be found in Appendix D.2.

### 5.4.3 Assumptions

In order to obtain an accurate statistical comparison between the performance results of each of the vehicle 39 vehicles, without various subsystems or subcomponents (such as suspensions systems, tyre characteristics, axles and centre of gravity height) skewing results, various assumptions had to be made during the modelling process. These assumptions would therefore allow the results of the simulations to be based on vehicle configuration, load carried by each vehicle unit and geometric design of each vehicle unit specific to that particular vehicle.

Below is a list of some of the assumptions made by OECD during their modelling process:

- The payload centre of gravity height was located at 40% of the load space height.
- The maximum allowable mass and heights are specified by each country.
- The centre of gravity height for the prime mover / tractor unit was taken to be 1.1 metres above the ground.
- The same generic suspension parameters were used for all vehicle as follows:
  - Parabolic springs used for the steer axle
  - Standard air suspension for the trailers and drive axles
- The same tyre type (11R22.5 tyres) was used on each axle, through whether dual tyre or single tyre axles were used were specified by each member country.

### 5.4.4 Results

The results of the study were documented according to the safety standard performance manoeuvres simulated, within which the outcome performance of each vehicle was plotted and analysed according to their region and vehicle specified classification (workhorse, higher capacity vehicle or very high capacity vehicles operating under special permit). The performance results of this study can be found in Tables D.1 – D.5 in Appendix D.3.

**Table 5.30: Comparative study between OECD report and simulation results for OECD 1, ZA1, semi-trailer**

| Performance Measure | OECD 1 | Semi Trailer Simulated Results | % Deviation |
|---|---|---|---|
| Tracking-ability on a straight path | 2.89 | 2.96 | 2.4 |
| Low speed swept path | 7.23 | 6.6 | 8.7 |
| Steer tyre friction demand | 4 - 47 | 38.7 | - |
| Static rollover threshold | 0.36 | 0.36 | 0.0 |
| Rearward amplification | 1.125 | 1.07 | 4.9 |
| High speed transient off-tracking | 0.25 | 0.24 | 4.0 |

Table 5.30 above illustrates the OECD report output results, the results obtained through computational simulation and the percentage deviation for the ZA1 semi-trailer combination. From these results it can be seen that the percentage deviation varies from 0 % to 8.7 %. This measure of output deviation is within the 10 % error band width and is therefore considered acceptable.

**Table 5.31: Comparative study between OECD report and simulation results for OECD 2, ZA2, B-double**

| Performance Measure | OECD 2 | B-double Simulated Results | % Deviation |
|---|---|---|---|
| Tracking-ability on a straight path | 2.93 | 2.99 | 2.0 |
| Low speed swept path | 7.85 | 7.31 | 6.8 |
| Steer tyre friction demand | 4 - 47 | 40 | - |
| Static rollover threshold | 0.37 | 0.36 | 2.7 |
| Rearward amplification | 0.97 | 1.08 | 11.3 |
| High speed transient off-tracking | 0.31 | 0.3 | 3.2 |

Table 5.31 above illustrates the OECD report output results, the results obtained through computational simulation and the percentage deviation for the ZA 2 B-double configuration. From this table it is evident that the percentage deviation varies from 2 % to 11.3 %. Five of the performance measures output deviations are within the 10 % error band width, excluding that of rearward amplification, and is therefore considered acceptable. The reason for the marginal variation in output deviation results for the rearward amplification performance manoeuvre is due to the discontinuities in the vehicle driver model and lash experienced by the vehicles hitches, as mentioned in Section 5.3.7 above, as well as the statistical errors incurred during the best fit polynomial plots in Matlab.

## 5.5 Tabular Summary of results

**Table 5.32: Summary of the performance levels achieved by each of the 10 vehicles assessed**

| | Semi-trailer | | | | | B-double | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| **Performance Measures** | **OECD 1** | **Skeletal** | **Refrigeration** | **Side Curtain** | **Tipper** | **OECD 2** | **Skeletal** | **Cane** | **Side Curtain** | **Tipper** |
| Startability | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 |
| Gradeability | | | | | | | | | | |
| a) Maximum Grade | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 |
| b) Speed on 1% Grade | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 |
| Acceleration Capability | Level 2 | Level 2 | Level 2 | Level 2 | Level 2 | Level 2 | Level 2 | Level 2 | Level 2 | Level2 |
| Tracking Ability on Straight Path | Level 2 | Level 1 | Level 2 | Level 2 | Level 1 | Level 2 | Level 2 | Level 2 | Level 3 | Level 2 |
| Low Speed Off-tracking | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 2 | Level 1 | Level 1 |
| Frontal Swing | | | | | | | | | | |
| a) Part A | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass |
| b) Part B | Pass | Fail | Fail | Pass | Pass | Pass | Pass | Pass | Pass | Pass |
| c) Part C | Pass | Pass | Fail | Pass | Pass | Pass | Pass | Pass | Pass | Pass |
| Tail Swing | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 |
| Steer Tyre Friction Demand | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass |
| Static Rollover Threshold | Pass | Pass | Fail | Fail | Pass | Pass | Pass | Pass | Fail | Pass |
| Rearward Amplification | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass |
| High Speed Transient Off-tracking | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 | Level 1 |
| Yaw Damping Co-efficient | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass | Pass |
| Direction Stability Under Braking | All Vehicles Pass | | | | | | | | | |
| **PBS Level Achieved** | Level 2 | Fail | Fail | Fail | Level 2 | Level 2 | Level 2 | Level 2 | Fail | Level 2 |

## 5.6 Conclusion

This chapter provided the results for the semi-trailer and B-double vehicle configurations, which were analysed according to the 13 PBS performance measures developed by Australia NTC. For ease of reading and in order to accurately compare each vehicle, the chapter was divided into four sections namely; Startability, Gradeability and Acceleration Capability, Semi-trailer, B-double and Validation.

Due to the fact that the same driveline characteristics were used for all ten of the analysed vehicles, it was decided that, Startability, Gradeability and Acceleration Capability would be grouped together in one section, thus reducing unnecessary repetition.

The following two sections 5.2 and 5.3, Semi-trailer and B-double respectively, where grouped into their own individual sections according to vehicle configuration, thus allowing a basis for comparative analysis between similar vehicles with in a specific vehicle classification.

The final Validation section was included in order to ensure that the computational simulation models that were developed accurately resemble the output plot published in the OECD report.

Table 5.32 provides a summary of the safety performance outcomes of the ten heavy vehicles assessed according to each of the 13 performance measures developed, their corresponding relevant performance level achieved, as well as each vehicles overall PBS level achieved.

Figures 5.26 - 5.35 provide a summarised graphical representation of ten of the thirteen performance results achieved for all ten, semi-trailer and B-double, vehicle configurations analysed, as well as their respective performance levels achieved and their pass or fail compliance.

The startability of the ten vehicles assessed was calculated from a much utilised industrial rule, where by the startability is determined from the gradeability less the tractive slip. Due to the fact that a single prime mover was utilised for the low speed longitudinal performance measure, a startability of 30% was calculated. This resulted in all ten vehicles satisfying the Level 1 performance result of not less than 15%.

The gradeability for the ten vehicles assessed was determined from data collected from the prime mover manufacturer as well as from HTM software. The results of which had a 2% variation, thus in order to be conservative the lower value was selected. This proved that the vehicle had the ability to obtain a minimum specified speed on a 38.8% gradient, satisfying a Level 1 performance result for the first aspect of gradeability (maintain forward motion on a maximum grade). The second aspect of gradeability (maintain a minimum speed on a 1% grade) was calculated through a linear iterative process, which stipulated that the vehicle has the capability to achieve a speed of 90km/h, this surpassed the 80km/h limit specified, and as such all ten vehicle achieved a Level 1 performance result.

The acceleration capability study was not capable of being computationally modelled, and as such field tests were undertaken. The results of which indicated that a fully laden vehicle (GCM 56 tons) has the capability to accelerate from rest and travel a distance of 100m in less than 23 seconds, changing through gears both automatically and manually. Thus the ten vehicles achieved a Level 2 performance result, of not more than 23 seconds.

However, it must be noted that the five semi-trailer combinations were not loaded to a maximum of 56 tons, and as such may have achieved a better performance level.

The Tracking-Ability on a Straight Path (TASP) performance manoeuvre was simulated in order to determine the amount of road space a vehicle requires when travelling on a straight road, at high speed, on an uneven road surface. The performance results for the five semi-trailer combinations illustrated that two of the five vehicles (Skeletal and Tipper) achieved performance results within Level 1 classification, whilst the three remaining vehicles (OECD 1, Refrigeration and Side Curtain) obtained Level 2 classification by exceeding the maximum 2.9 m swept path stipulated by Level 1.

The results for the five B-double configurations that were analysed showed that four of the five vehicles (OECD 2, Skeletal, Cane and Tipper) achieved Level 2 classification, whilst the remaining vehicle (Side Curtain) achieved a Level 3 classification, by exceeding the maximum swept path of 3.0 m specified by the Level 2 classification. Failure to comply with Levels 1 and 2, the Side Tipper, therefore requires more lane width when travelling at high speeds down a straight even surfaced road. This increased tracking poses an increased safety risk to other road users and well as other road side objects.

The results of this performance measure are illustrated graphically in Figure 5.26.

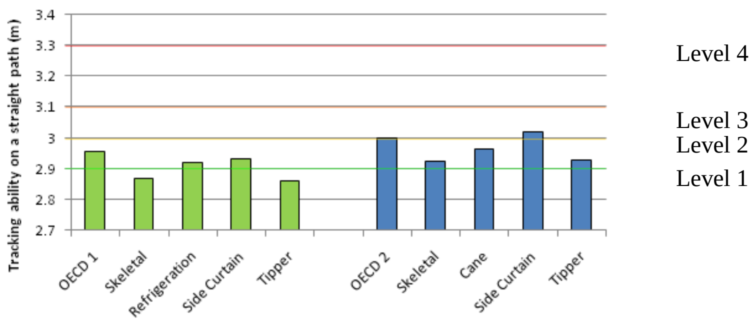


**Figure 5.26: Summary of the Tracking ability on a straight path performance result for the ten simulated vehicles**

Low-Speed Swept Path (LSSP) performance manoeuvre allows one to determine the amount of road space a vehicle requires when performing a tight turn at low speed. The semi-trailer performance results for this measure illustrated that all five vehicle combinations achieved Level 1 classification by not exceeding a maximum swept path of 7.4m.

The results for the five B-double configurations indicated that four of the five vehicles (OECD 2, Skeletal, Side Curtain and Tipper) achieved Level 1 classification, whilst the remaining vehicle (Cane) achieved a performance result of 7.6 m thus satisfying Level 2 classification.

The results for this performance measure are illustrated graphically in Figure 5.27.

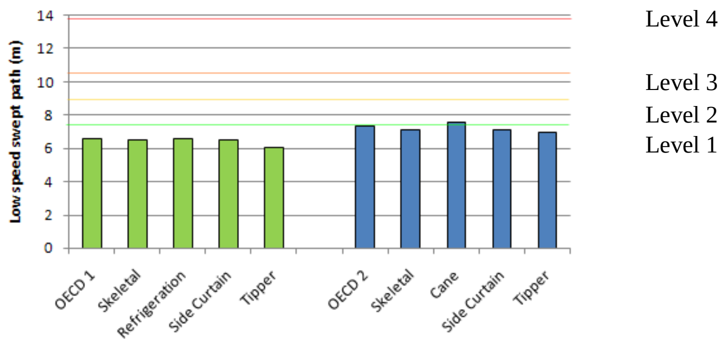


**Figure 5.27: Summary of the Low speed swept path performance results for the ten simulated vehicles**

The Frontal Swing (FS) performance manoeuvre was separated into three sub-sections, Part A (Prime mover), Part B (MoD) and Part C (DoM). The performance results for the five semi-trailer vehicle combinations illustrated that three of the five vehicles (OECD 1, Side Curtain and Tipper) passed this manoeuvre, whilst the remaining two vehicles (Skeletal and Refrigeration) failed, both of which exceeded the maximum 0.4 m limitation for Part B. The failure to meet this standard indicates that the forward most outside point of the first semi trailer, when performing a tight turn at low speed, will tend to track outside it specified lane width, which may result in collisions with road side objects as well as other vehicle users.

The performance results for the five B-double configurations satisfied all three frontal swing criteria, and thus all passed this performance measure.

The results for this performance measure are illustrated graphically in Figure 5.28 (Part A), 5.29 (Part B) and 5.30 (Part C).

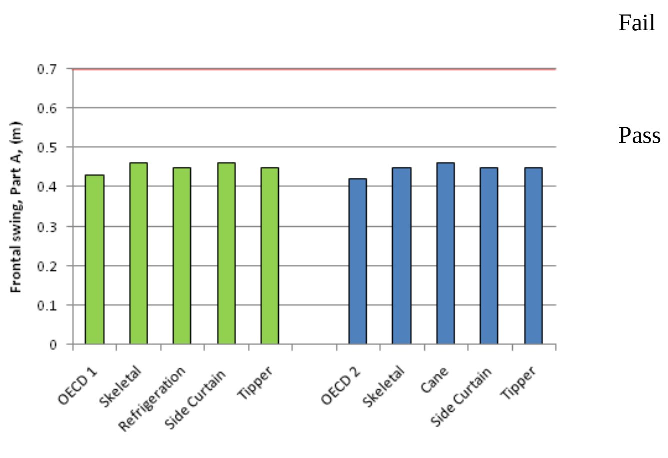


**Figure 5.28: Summary of the Frontal swing, Part A, performance results for the ten simulated vehicles**

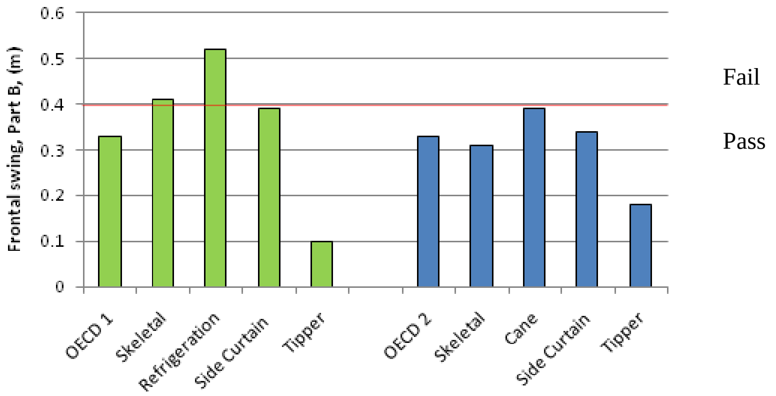


**Figure 5.29: Summary of the Frontal swing, Part B, performance results for the ten simulated vehicles**

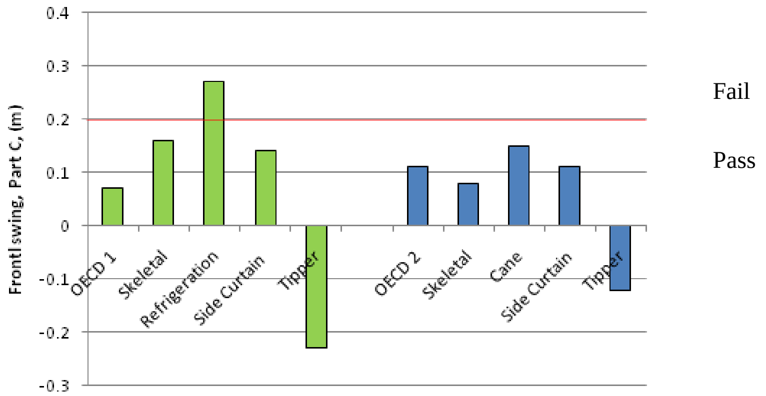


**Figure 5.30: Summary of the Frontal swing, Part C, performance results for the ten simulated vehicles**

The Tail Swing (TS) performance measure, similar to that of LSSP and FS, determines the amount of road space required when a vehicle performs a tight turn at low speed. This performance measure is also separated into two sections, TSentry and TSexit, the tail swing at the entry of the turn and the tail swing at the exit of the turn, respectively. None of the ten vehicles analysed, semi-trailer and B-double, exhibited a tail swing at the exit of the turn, whilst all vehicles experienced a TSentry achieved a Level 1 classification.

The results for this performance measure are illustrated graphically in Figure 5.31.

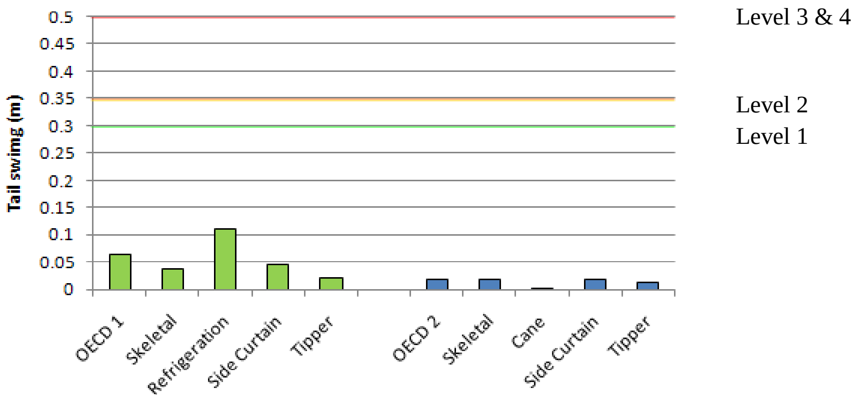


**Figure 5.31: Summary of the Tail swing performance results for the ten simulated vehicles**

Steer Tyre Friction Demand (STFD) performance manoeuvre measures the possibility of a vehicle losing steering control when performing a tight turn at low speed. All ten vehicles assessed, semi-trailer and B-double, did not exhibit a loss of steering throughout the manoeuvre, thus all ten vehicles passed this performance measure.

Static Rollover Threshold (SRT), arguably the most important performance measure, is designed to determine the amount of lateral acceleration a vehicle can withstand before rolling over. This performance measure may be calculated in two ways, namely: circular test and tilt table test, both of which were simulated during this assessment. The performance results for the five semi-trailer combinations illustrated for both manoeuvres, that three vehicles (OECD 1, Skeletal and Tipper) satisfied the minimum requirement of 0.35g, whilst the remaining two vehicles (Skeletal and Refrigeration) achieved a SRT value of 0.33g and 0.31g respectively, for both manoeuvres.

The performance results for the five B-double configurations illustrated that four of the five vehicles (OECD 2, Skeletal, Cane and Tipper) obtained the minimum performance level of 0.35g for both performance manoeuvres, whilst the remaining vehicle (Side Curtain) obtained a performance result of 0.32 and 0.33 for each of the respective testing method.

The percentage deviation between the two manoeuvres, circular test and tilt table, varied between 0.77 -5.0 %, this variation is due to the driveline characteristics of which were only utilised in the circular test method.

Failure to comply with this performance measure indicates that the vehicles are not dynamically stable, and would require less lateral force to cause rollover, in comparison to vehicle who do achieve the minimum 0.35g limit. This lack of compliance is of great concern not only for the driver and occupants of the vehicle, but also to other road users, and road side infrastructure.

The results for this performance measure, (an average of both circular test and tilt table test), are illustrated graphically in Figure 5.32.

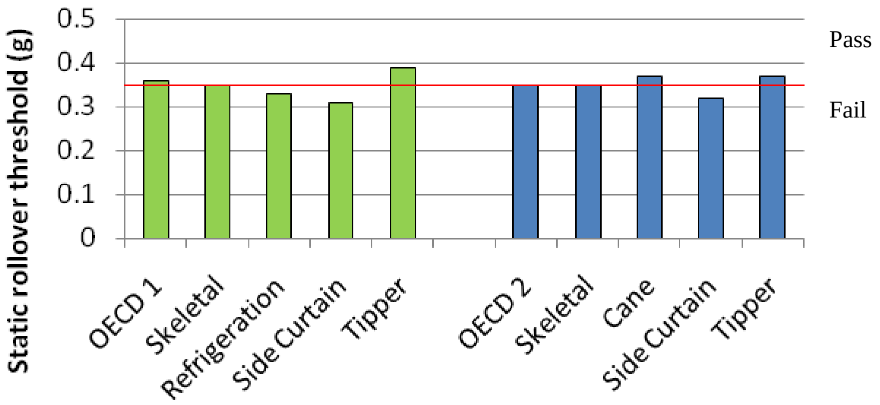


Figure 5.32: Summary of the Static rollover threshold performance results for the ten simulated vehicles

The Rearward Amplification (RA) performance measure was developed in order to determine the amount of lateral acceleration experienced between vehicle units. All ten vehicles assessed, semi-trailer and B-double, achieved performance results that were less than the maximum specified limit, thus all ten vehicles passed this performance manoeuvre.

The results of this performance manoeuvre are illustrated graphically in Figure 5.33.

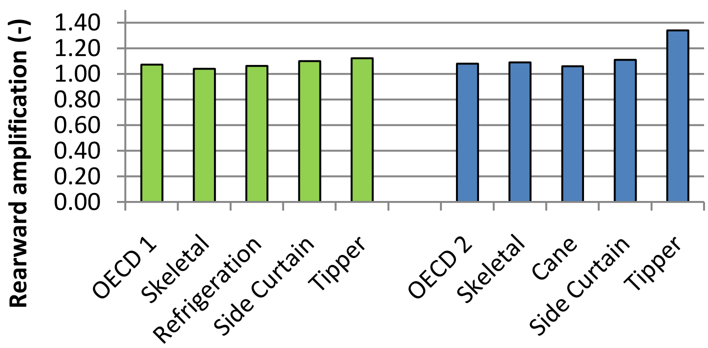


Figure 5.33: Summary of the Rearward amplification performance results for the ten simulated vehicles

High-Speed Transient Off-tracking (HSTO) is design to limit the lateral deviation of the last trailing axle of the last vehicle unit from the desired path. All ten vehicles assessed, semi-trailer and B-double, achieved Level 1 classification by not exceeding the maximum deviation of 0.6 m.

The results from this performance manoeuvre are illustrated graphically in Figure 5.34.

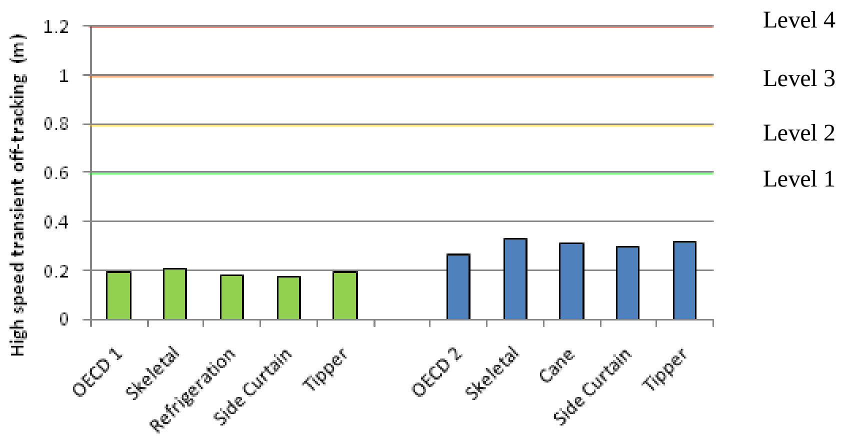


**Figure 5.34: Summary of the High speed transient off-tracking performance results for the ten simulated vehicles**

The Yaw Damping performance measure is designed to limit the time taken for a vehicle's oscillations to settle down after performing a manoeuvre at high speed. The results for this performance measure illustrated that all ten vehicles assessed, semi-trailer and B-double, achieved acceptable results, not less than 0.15 for the specified speed. Thus all ten vehicles passed this performance measure.

The results from this performance manoeuvre are illustrated graphically in Figure 5.35.

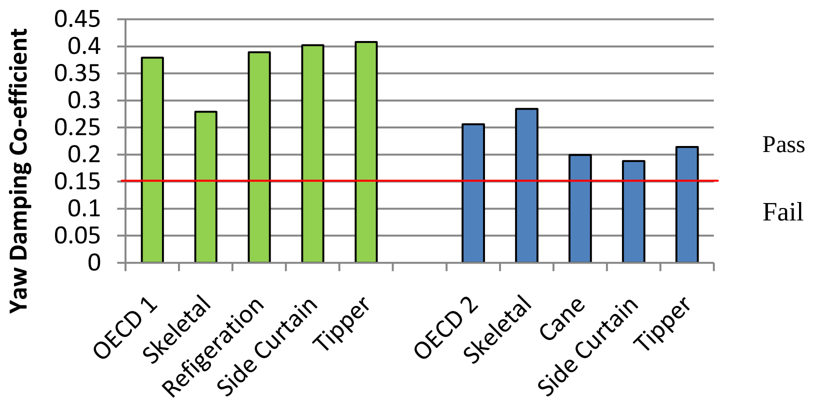


**Figure 5.35: Summary of the Yaw damping performance results for the ten simulated vehicles**

The Directional Stability under Braking performance measure was not simulated as all ten vehicles were deemed to comply in accordance with Section C16.3 (b) stipulated in the PBS 'Standards and Vehicle Assessment Rules' [15].

From Table 5.32 it can be seen that six of the ten vehicles (Semi-trailer: OECD 1, Tipper. B-double: OECD 2, Skeletal, Cane and Tipper) achieved Level 2 classification. Whilst the remaining four vehicles (Semi-trailer: Skeletal, Refrigeration, Side Curtain. B-double: Side Curtain) failed the PBS assessment for varying reasons.

The OECD 2 and Tipper semi-trailers achieved Level 1 classification for 12 of the 13 performance measures, however, the acceleration capability classification of Level 2, reduced theses vehicles to an overall Level 2 classification. As discussed previously the acceleration capability result is based on a 56 ton vehicle, which negatively influences the result of these two vehicle combinations.

The Skeletal semi-trailer achieved Level 1 classification for 11 of the 13 performance measures, it achieved a Level 2 for the acceleration capability, however, it failed the Frontal Swing Part B, MoD, performance requirement.

The Refrigeration semi-trailer achieved Level 1 classification for nine of the 13 performance measures; it achieved Level 2 classification for acceleration capability and TASP, whilst it failed Frontal Swing Part B (MoD) and Part C (DoM) and also SRT.

Side Curtain semi-trailer achieved Level 1 classification for 10 of the 13 performance measures; it achieved Level 2 classification for acceleration capability and TASP, whilst it failed the SRT performance measure.

OECD 2, Skeletal and Tipper B-double configurations achieved Level 1 classification for 11 of the 13 performance measures; however, it achieved Level 2 classification for acceleration capability and TASP, thus providing an overall PBS result of Level 2.

The Cane B-double configuration achieved Level 1 classification for 10 of the 13 performance measures; however it achieved Level 2 classification for acceleration capability, TASP and LSSP, thus having an overall PBS result of Level 2.

The Side Curtain B-double configuration achieved Level 1 classification for 10 of the 13 performance measures, it achieve Level 2 classification for acceleration capability; however, it achieved Level 3 classification for TASP, which is considered a fail, as Level 1 and Level 2 were the only level classification considered for South African conditions, and it also failed the SRT performance measure. Thus this vehicle did not achieve the minimum PBS criteria.

However, it must be noted that these performance requirements are based on Australian conditions, and only provide an indication of the vehicle performance that it would meet if the vehicle were to operate on the Australia road network. Thus the performance levels achieved by each vehicle can only be used as a comparison between vehicles operating in different countries.

An analytical approach was used in order to validate the computational simulation output results obtained during the modelling and simulation process, these output plots were compared to performance results in a recent OECD study, which aimed to benchmark the current heavy vehicle fleet of member countries according to Australian Performance Based Standards.

The validation results varied between 0 – 11.3 % for the OECD 1 semi-trailer and the OECD 2 B-double, nine of the ten vehicle were within the 10% acceptable error band width, whilst the OECD 2 B-double vehicle has a percentage deviation of 11.3% for the rearward amplification performance manoeuvre.

# Chapter 6 - Conclusion

A literature survey into the South African transportation industry illustrated a drastic need for the introduction of a modernised legislative regulatory system. Investigation into various implementations of Performance–Based Standards (PBS) schemes around the world was conducted, and from which it was decided that the 16 Safety Performance Standards developed by Australia would be utilised in order to benchmark a sample of the current South African fleet.

A survey was conducted in order to determine the main heavy vehicle configurations used on the South African transportation network. From this study four main vehicle configurations were identified, namely: rigid truck, semi-trailer, rigid-drawbar and B-double. The two main configuration, semi-trailer and B-double, were selected for the purpose of this research.

A total of ten vehicles were selected, five from each vehicle class. One vehicle from each class, OECD 1 and OECD 2, respectively, was selected from a previous OECD study. These two vehicles were then utilised in order to analytically validate the simulated performance results of the remaining eight vehicles.

13 of the 16 Australian Safety Performance Standards developed were modelled in Trucksim, a vehicle dynamic simulation software package, whilst the remaining three safety standards were excluded as they had not yet been fully developed.

All ten vehicles were then modelled and simulated according to these 13 performance standards, the results of which are tabulated and discussed in Sections 5.5 and 5.6, respectively. These results illustrated that six of the ten vehicles analysed achieved Level 2 road classification, whilst the remaining four vehicles did not achieve PBS status for various reasons, as discussed in Chapter 5.

The validation process illustrated the results found in Section 5.4. A percentage deviation ranging from 0.0 – 11.3 % was achieved; the reasons for these variations were discussed in Section 5.4.5.

The objective of the project was to develop a benchmark of current South African heavy vehicle configurations according to the Australian PBS initiative, which has been achieved through the use of computational modelling software. The results of which illustrate that 40% of the vehicles analysed did not meet the minimum Australian requirements, for various reasons as discussed in Section 5.6 above. This therefore illustrates that almost half of the vehicles on the South African road network do not measure up to international vehicle dynamic performance and safety criteria, and are deemed to be unsafe.

The introduction of PBS in South Africa will have a positive impact on the stability and safety of our heavy vehicles, increasing vehicle productivity, reducing road infrastructure damage and costs, and reducing the safety risk imposed to other road users.

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# Appendix A – Performance Based Standards

## C1. STARTABILITY

### C1.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk associated with starting on grade by ensuring that a vehicle participating in the Scheme has adequate starting capability on grades.

#### (b) Intent

A vehicle participating in the Scheme must be capable of starting on the steepest grade it has to negotiate on the nominated route when operating at its maximum allowed gross mass, otherwise it can become a safety risk and an inconvenience to other road users.

A combination vehicle that is stopped on a grade beyond its capability will either require that its units be separated and moved or require the use of heavy haulage equipment to move it to a location where it can restart.

### C1.2 Definition

#### (a) Summary statement

The ability to commence forward motion on specified upgrade.

#### (b) Detailed statement

When operating at maximum laden mass, a vehicle participating in the Scheme must be able to commence and maintain steady forward motion from a standing start on a pavement section of specified upgrade. Momentary reverse motion (downhill) at commencement, associated with the release of brakes and clutch engagement, or similar, is acceptable provided uphill forward motion is subsequently achieved and maintained.

### C1.3 Measure

#### (a) Performance value

The maximum upgrade on which forward motion is commenced and maintained must be measured and reported as the achieved performance value, in units of percentage grade[10], rounded down to the nearest whole number.

#### (b) Performance levels

**Table 4. Startability performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | At least 15% |
| Level 2 | At least 12% |
| Level 3 | At least 10% |
| Level 4 | At least 5% |

### C1.4 Test specification

#### (a) Test load

The vehicle being assessed must be loaded to its maximum laden mass. Each tyre on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyre. Each tyre must be inflated to a pressure within the range as specified by the vehicle and/or tyre manufacturer.

#### (b) Test conditions

The full length of the vehicle being assessed must be on an upgrade appropriate to the road classification level. The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

#### (c) Test procedure

From a standing start on a slope having an upgrade not less than the specified grade, the vehicle being assessed must commence and maintain steady forward motion. Steady forward motion on the specified grade is achieved when the vehicle's speed is either constant or increasing for a distance of at least 5 metres.

#### (d) Test method

Numerical modelling (computer-based simulation) or field-testing.

## C2: GRADEABILITY

### C2.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk associated with travel on grade by ensuring that a vehicle participating in the Scheme has the capability to maintain acceptable speeds on upgrades.

#### (b) Intent

When operating at the maximum allowable gross mass, vehicles participating in the Scheme must be able to maintain a specified minimum speed on upgrades. This is desirable in order to minimise traffic congestion or delays to other vehicles travelling in the same direction. Further, heavy vehicles travelling on grade that impede traffic are known to increase accident rates on two lane rural highways (Khan et al, 1990).

Gradeability is applicable to all heavy vehicle operations – in urban, rural/regional and remote areas – and to all vehicles participating in the Scheme. In addition to safety considerations and concerns, gradeability also influences vehicle productivity, route selection and access.

The vehicle-related factors that determine startability also influence gradeability. However, a vehicle may be designed to maximise startability at the expense of its gradeability performance (geared low). Similarly, if optimised for gradeability it may not meet the startability requirement. For this reason both startability and gradeability must be within acceptable limits for a vehicle participating in the Scheme.

## PART (A) MAINTAIN MOTION

### C2.2 Definition

#### (a) Summary statement

The ability to maintain forward motion on specified upgrade.

#### (b) Detailed statement

When operating at maximum laden mass, a vehicle participating in the Scheme must be able to maintain steady forward motion on a pavement section of specified upgrade. An initial change in speed associated with the transition from the approach to the upgrade is acceptable, provided steady forward motion can be maintained on the upgrade.

### C2.3 Measure

#### (a) Performance value

The maximum upgrade on which steady forward motion is maintained must be measured and reported as the achieved performance value, in units of percentage grade rounded down to the nearest whole number.

#### (b) Performance levels

**Table 5. Gradeability - Part (a) Maintain Motion performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | At least 20% |
| Level 2 | At least 15% |
| Level 3 | At least 12% |
| Level 4 | At least 8% |

### C2.4 Test specification

#### (a) Test load

The vehicle being assessed must be loaded to its maximum laden mass. Each tyre on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyre. Each tyre must be inflated to the pressure as specified by the vehicle and/or tyre manufacturer.

#### (b) Test conditions

Same as for startability. Additionally, the upgrade must be of sufficient length to allow steady forward motion to be established. The full length of the vehicle being assessed must be on the upgrade. The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

#### (c) Test procedure

With the vehicle being assessed in forward motion on a slope having an *upgrade* not less than the specified grade, it must maintain steady forward motion. Steady forward motion is achieved when the vehicle's forward speed on the upgrade is either constant or increasing for a distance of at least 5 metres.

#### (d) Test method

Numerical modelling (computer-based simulation) or field-testing.

## PART (B) MAINTAIN SPEED

### C2.5 Definition

#### (a) Summary statement

The ability to maintain a minimum speed on a specified upgrade.

#### (b) Detailed statement

When operating at maximum laden mass, a vehicle participating in the Scheme must be able to maintain a specified minimum speed on a pavement section having an upgrade of not less than 1%. An initial change in speed associated with the transition from the approach to the upgrade is acceptable, provided the specified minimum speed is maintained on the upgrade.

### C2.6 Measure

#### (a) Performance value

The minimum sustained steady speed must be measured and recorded as the achieved performance value, in units of km/h rounded down to the nearest whole number.

#### (b) Performance levels

**Table 6. Gradeability - Part (b) Maintain Speed performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | At least 80 km/h |
| Level 2 | At least 70 km/h |
| Level 3 | At least 70 km/h |
| Level 4 | At least 60 km/h |

### C2.7 Test specification

#### (a) Test load

The vehicle being assessed must be loaded to its *maximum laden mass*. Each tyre on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyre. Each tyre must be inflated to the pressure as specified by the vehicle and/or tyre manufacturer.

#### (b) Test conditions

The full length of the vehicle being assessed must be on an upgrade. The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

#### (c) Test procedure

With the vehicle being assessed in forward motion on a slope having an upgrade of not less than 1%, steady forward motion must be maintained at a speed at least equal to the specified minimum speed. Steady forward motion is when the forward speed of the vehicle on the upgrade is either constant or increasing for a period of at least 5 seconds.

#### (d) Test method

Numerical modelling (computer-based simulation) or field-testing.

## C3: ACCELERATION CAPABILITY

### C3.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk associated with travel through intersections and rail crossings by specifying minimum times for a vehicle participating in the Scheme to accelerate from rest, to increase speed, and travel specified distances.

#### (b) Intent

Acceleration capability is of primary concern to long or slow vehicles and addresses issues associated with intersection clearance times and rail level crossings. Heavy vehicles that require long times to accelerate to speed will take too long to clear intersections or railway level crossings, causing congestion and delays to through traffic, as well as posing a threat to safety if sight distances are inadequate. At an unsignalised intersection the probability of finding a gap in opposing traffic decreases as the size of the gap required increases. Signalised intersections operate on phase times that, where possible, should permit most heavy vehicles to clear the intersection in the allocated time.

Acceleration capability has an effect on the productivity of heavy vehicles in urban traffic, the capacity of the intersection and traffic congestion.

### C3.2 Definition

#### (a) Summary statement

The ability to accelerate either from rest or to increase speed on a road with no grade.

#### (b) Detailed statement

When operating at maximum laden mass, a vehicle participating in the Scheme must be able to accelerate from rest, and travel 100 m on a road with no grade within a specified time.

### C3.3 Measure

#### (a) Performance value

The time taken to travel the distance of 100 metres must be reported as the achieved performance, to the nearest 0.1 second.

#### (b) Performance levels

**Table 7. Acceleration capability performance levels**

| Performance Based Standards Road Class | Time To Travel 100m Free Rest (secs) |
|---|---|
| 1 | 20 |
| 2 | 23 |
| 3 | 26 |
| 4 | 29 |

### C3.4 Test specifications

#### (a) Test load

The vehicle being assessed must be loaded to its maximum laden mass. Each tyre on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyre. Each tyre must be inflated to the pressure as specified by the vehicle and/or tyre manufacturer.

#### (b) Test conditions

The full length of the vehicle being assessed must be on a site with zero grade throughout the test (except for assessment by testing – see Appendix F). The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

#### (c) Test procedure

From a standing start the vehicle being assessed must accelerate, changing through the gears as required, over a distance of at least 100 metres.

The point of commencement of acceleration should be taken as the moment forward motion starts.

#### (d) Test method

Numerical modelling (computer-based simulation) or field-testing.

## C4: OVERTAKING PROVISION

The requirements for Overtaking Provision have been moved to the *Network Classification Guidelines* as a key component of classifying the heavy vehicle freight network. This allows the standard to be applied to the Performance Based Standards network without diminishing the intent of the standard, whilst improving the process for the assessment of individual applications.

To assist in the application of the *Network Classification Guidelines*, the maximum vehicle lengths for each road class are as set out below. For heavy vehicles participating in the Scheme that wish to gain access to a specific network level but exceed the maximum permitted length for that level, an individual route assessment will need to be carried out.

**Table 8. Network classification by vehicle length**

| Vehicle Performance Level | Network Access by Vehicle Length, L (m) | |
|---|---|---|
| | Access Class 'A' | Access Class 'B' |
| Level 1 | L ≤ 20<br>(General Access*) | |
| Level 2 | L ≤ 26 | 26 < L ≤ 30 |
| Level 3 | L ≤ 36.5 | 36.5 < L ≤ 42 |
| Level 4 | L ≤ 53.5 | 53.5 < L ≤ 60 |

* General Access is subject to a 50 tonne gross mass limit, posted local restrictions and restrictions or limitations specified by the jurisdiction.

### C5: TRACKING ABILITY ON A STRAIGHT PATH

#### C5.1 Purpose and intent

**(a) Purpose**

The primary purpose of this standard is to manage safety risk associated with lane width and lateral clearance by ensuring that a vehicle participating in the Scheme remains within its traffic lane when travelling at high speed on straight roads with uneven surfaces.

**(b) Intent**

When the hauling unit of a heavy vehicle follows a straight path along a section of road, it is both a practical requirement and necessary for safe operation that the rear of the vehicle follows with adequate fidelity and tracks within a specified lane width. Vehicles that require more lane width than is available present a risk to safety and the infrastructure when crossing the centre-line when being overtaken or passed, and when leaving the pavement seal causing edge break-off and aggravating shoulder drop. If large enough and the conditions sufficiently adverse, shoulder drop can initiate a rollover.

In practice, each trailer in a combination vehicle will undergo small lateral excursions from the path of its lead unit as it responds to the driver's steering actions, road surface unevenness and other external disturbances, such as cross winds. The ability of the trailing units to faithfully track along the same path as the hauling unit is referred to as tracking ability. Tracking ability depends on a range of vehicle-related factors, including: number of trailers and the location and type of coupling between them (turntable or pintle hitch); alignment of axles; suspension geometry (roll and bump steer effects); tyre cornering stiffness; vehicle length; and speed.

#### C5.2 Definition

**(a) Summary statement**

The total swept width while travelling on a straight path, including the influence of variations due to crossfall, road surface unevenness and driver steering activity.

**(b) Detailed statement**

When operating at least favourable load condition and travelling along the specified straight path on a section of pavement having the specified unevenness and cross-slope characteristics, the total swept width of the vehicle being assessed in the specified test must be no greater than the specified value.

#### C5.3 Measure

**(a) Performance value**

The total swept width of the vehicle being assessed must be measured and reported over the specified test section as the achieved performance value, expressed in units of metres, rounded up to the nearest 0.1metre The swept width at any point along the straight path is the length of the straight-line segment intersecting all the path trajectories, measured in the ground plane and perpendicular to the direction of travel, as shown in Figures 1 to 3. For single unit and combination vehicles, the swept width along the path must be determined from the path trajectories of the outermost paths scribed in the ground plane by the vertical projections of the outermost reference point, or points of all vehicle units. The number of reference points must ensure that the swept width is measured to the defined accuracy. The reference points must be selected when the vehicle is on a flat level surface. Where there is a choice of more than one reference point at any particular location, such as al the vertical outside corner of the front or rear of a vehicle unit, the outermost point nea to the ground must be selected.

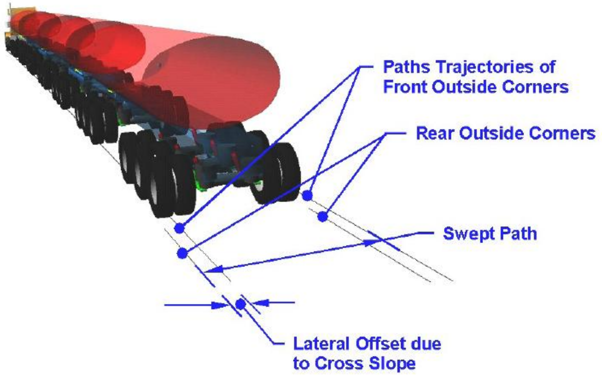



**Figure 1.** **Perspective view illustration of front and rear outside-corner path trajectories and swept width in the tracking ability on a straight path test.**

* Note that the path trajectories of only 4 points are shown in the above example.)

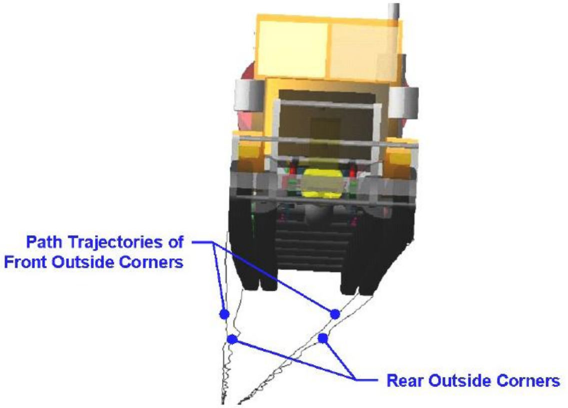



**Figure 2.** **Underside perspective view illustration showing typical characteristics of the path trajectories and offsets due to cross slope.**

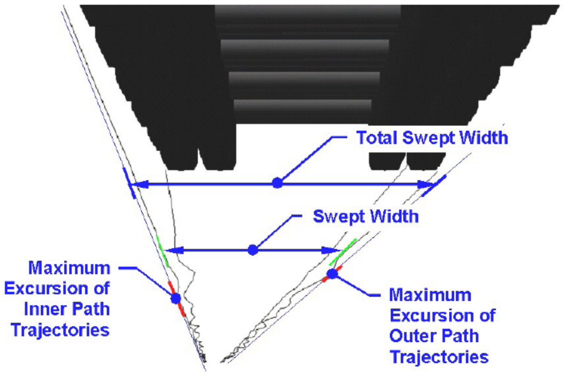



**Figure 3.** Close-up underside perspective view illustration showing both the swept width envelope and Total Swept Width from the 4 path trajectories defined in Figures 1 and 2.

#### (b) Performance levels

**Table 9.** Tracking ability performance levels

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | Not greater than 2.9 m |
| Level 2 | Not greater than 3.0 m |
| Level 3 | Not greater than 3.1 m |
| Level 4 | Not greater than 3.3 m |

If the assessment is undertaken by numerical modelling, a test road conforming to the test conditions will be supplied to the assessor so that all assessments are undertaken under equal conditions. However, to allow for variations in computer models, the 99th percentile of the measured swept width will be accepted as meeting the performance level. With assessment by testing, a risk exists that one or two major bumps on the test section will not provide a reasonable test of the ability of the vehicle being assessed to meet the standard and again the 99th percentile of the measured swept width will be accepted as meeting the performance level.

### C5.4 Test specification

#### (a) Test load

The vehicle being assessed must be loaded to the least favourable load condition. The maximum laden mass and the corresponding maximum axle group loads must not be exceeded. For standard test conditions, the tyres on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyres, and have no less than 100 km of running. Tyres must be inflated to pressures as specified by the vehicle and/or tyre manufacturer.

#### (b) Test conditions

The road pavement test section must be at least 1000 metres long and the surface must have an overall unevenness level in each wheel path of not less that 3.8 m/km IRI (International Roughness Index). The unevenness level in each wheel path reported every 100 m must be not less than 3.0 m/km IRI. The entire test section must have an average crossfall, falling to the left when viewed in the direction of travel, of not less than 3.0%. The average crossfall must have a crossfall standard deviation of not less than 1.0%.

For the numerical modelling, a standard set of road profiles will be supplied to the assessor. The standard profiles are taken from the work performed for Austroads described in Prem et al (1999). The profiles were used in the Performance Based Standards fleet analysis project that developed this and other standards (NRTC, 2002).

#### (c) Test procedure

The vehicle being assessed must traverse a road segment not less than 1000 metres long at a travel speed not less than 90 km/h. The vehicle must be driven in a normal manner at the specified speed while as closely as possible following a straight path that is either defined specifically for the task, such as a contrasting line, or by existing pavement edge line or centreline markings.

#### (d) Test method

Numerical modelling or field-testing. If performed by numerical modelling, as noted above a test road conforming to the test conditions specified above will be supplied by the Panel to the assessor.

## C6: RIDE QUALITY (DRIVER COMFORT)

### C6.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting driver whole-body vibration especially on uneven roads where travel speeds are high and vibration levels are expected to be significant.

#### (b) Intent

The effect of whole-body vibration on heavy vehicle drivers is an important road safety and occupational health and safety issue for the road transport industry. Short-term exposure to high-intensity vibration can have adverse effects on motor processes and the sensory system (limb movements, sensing and response to motion, vision, and hearing) that can impair the driver's ability to control the vehicle. This type of vibration can also lead to acute injuries. Further, long-term exposure to occupational whole-body vibration poses a health risk that can lead to chronic injuries associated with the back and abdominal regions of the human body.

While there appears to be a connection between whole-body vibration and driver comfort and health, vibration has been observed both to improve and to reduce proficiency. This may be because it fatigues or arouses or, because of increased task difficulty, motivates. These effects cannot be reliably predicted at present.

Ride vibration is influenced by a variety of factors, including vehicle load, suspension and tyre characteristics, prime mover wheelbase, seat location (fore-aft and height), seat transmissibility characteristics, road surface unevenness, speed, kingpin lead and trailer characteristics.

### C6.2 Definition, measure and test specification

This standard has yet to be defined. Its three main components (a performance measure, a test procedure from which to obtain the performance measure and a performance level, or levels, to be satisfied) are not able to collectively be defined to an acceptable level of robustness at this time based on current research.

There are three (very similar) recognised standards[11] for assessing whole-body vibration that could form the basis of defining the performance measure and performance level(s), but at this point in time there has yet to be developed a suitable test procedure from which to obtain measurements. It is important that the severity of the test procedure is at a level that allows good-performing vehicles to pass the standard and disallows poor-performing vehicles from passing the standard. Setting the severity at too low a level will potentially allow all vehicles to pass the standard, regardless of their performance. Conversely, setting the severity at too high a level may potentially disallow all vehicles.

The NTC has initiated a project to finalise this standard in its 2007/2008 work programme. A proposal will be submitted to the Australian Transport Council for approval in 2008 following consultation with industry and Transport Agency Chief Executives.

## C7: LOW-SPEED SWEPT PATH

### C7.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk associated with turns at intersections by limiting the road space required by a vehicle participating in the Scheme when making low-speed turns.

#### (b) Intent

When a long vehicle makes a low-speed turn at an intersection, the rear of the vehicle will follow a path that is inside the path taken by the front of the vehicle. This is known as low-speed offtracking. A high value of offtracking is undesirable because the vehicle, sweeping a wider path, will require more road space for turning than may be available. This may cause the vehicle to encroach into adjacent or opposing lanes, collide with parked or stopped vehicles, damage roadside furniture, endanger pedestrians, or the rear wheels may climb the kerb or fall off the edge of the pavement.

### C7.2 Definition

#### (a) Summary statement

The maximum width of the swept path in a prescribed 90º low speed turn.

#### (b) Detailed statement

When operating at maximum laden mass and unladen, the maximum width of the swept path of a vehicle participating in the Scheme in the prescribed 90º turn performed at a speed of no more than 5 km/h must be no greater than the specified value.

### C7.3 Measure

#### (a) Performance value

The maximum width of the swept path must be measured and reported as the achieved performance value, in units of metres rounded up to the nearest 0.1 metre. The maximum width of the swept path is the maximum distance, $SPW_{max}$, between the outer and inner path trajectories of the swept path envelope of the vehicle being assessed in the specified low-speed turn, shown in Figures 5 and 6. The maximum distance, $SPW_{max}$, is the straight-line segment intersecting both trajectories perpendicularly to their respective tangents at the intersection points. The swept path must be determined from the path trajectories of: 1) the outermost path scribed in the ground plane by the vertical projection of the furthest forward or outside point, or points, on the vehicle on the outside of the turn; and 2) the innermost path scribed in the ground plane by the vertical projection of the point, or points, on the vehicle on the inside of the turn. The above is summarised in Figures 4 to 6.

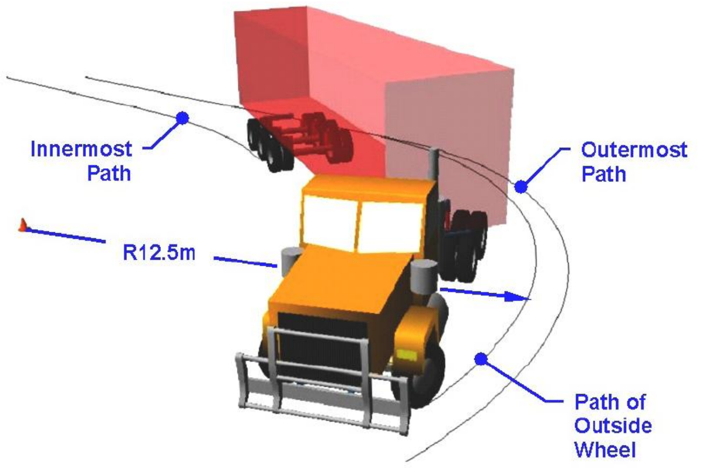



**Figure 4.** Perspective view illustration of vehicle partway through the Performance Based Standards low-speed turn showing path trajectories.

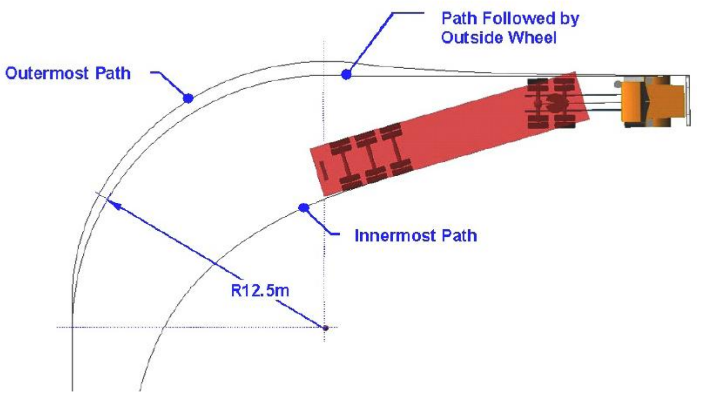



**Figure 5.** Plan view illustration of path trajectories that define the vehicle's swept path in the Performance Based Standards low-speed turn.

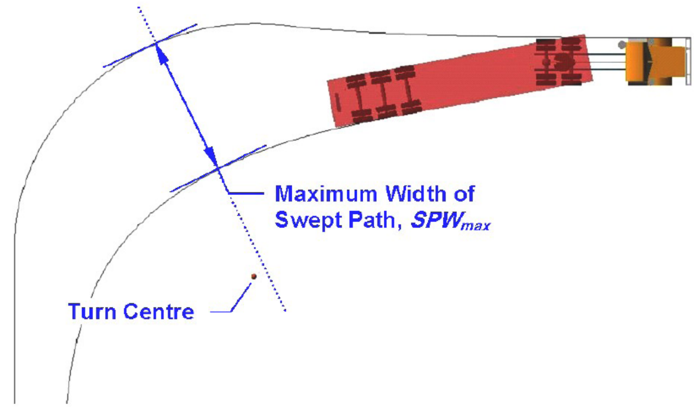



**Figure 6.** Plan view illustration of Maximum Width of Swept Path, $SPW_{max}$. Note that the line perpendicular to both path trajectories (shown dotted in the illustration) does not necessarily pass through the turn centre.

**(b) Performance levels**

**Table 10. Low speed swept path performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | No greater than 7.4 m |
| Level 2 | No greater than 8.7 m |
| Level 3 | No greater than 10.6 m |
| Level 4 | No greater than 13.7 m |

### C7.4 Test specification

**(a) Test load**

The vehicle being assessed must be tested fully laden and unladen. When fully laden it must be loaded to its maximum allowed gross mass and the corresponding maximum allowed axle group loads must not be exceeded. For the purposes of measuring swept path, mirrors and signalling devices are ignored.

**(b) Test conditions**

The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

### (c) Test procedure

The vehicle being assessed must be driven through the specified turn, unladen and laden, at a speed no greater than 5 km/h. The path of the specified turn that the driver will use to guide the vehicle must comprise straight tangent approaches to a 90° circular arc of 12.5 metre radius. The approaches to the turn must be of sufficient length to ensure:

(i) the entire vehicle is straight at the point where the 90° turn is commenced; and

(ii) at the conclusion of the turn the vehicle travels far enough into the straight exit segment to record the maximum width of the swept path.

The driver must ensure the entire vehicle is straight at the commencement of the turn (within 0.1 metre of the entry approach tangent). In the turn the driver must steer the vehicle along the specified path. The vehicle must be steered such that the vertical projection in the ground plane of the outer most point on the outer tyre sidewall nearest to the ground[12], on the forward most outside steered-wheel, follows the specified path as illustrated in Figure 5. Using the above point as a reference, the driver must steer the vehicle along the specified path and maintain a lateral distance error between the reference point and the specified path not greater than 50 millimetres.

### (d) Test method

Numerical modelling (computer-based simulation) or field-testing.

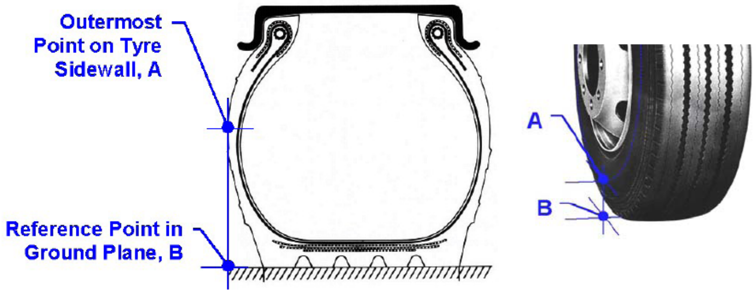



**Figure 7. Illustration of outside wheel reference point.**

## C8: FRONTAL SWING

### C8.1 Purpose and intent

### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting the road space requirement of a vehicle participating in the Scheme when making a tight turn at low speed.

### (b) Intent

In a low-speed turn the front overhang of the hauling unit (rigid truck, prime mover, bus and coach) will generally cause the path of the front outside corner to track outboard the path of the front outside steered wheel. This behaviour is known as frontal swing. A large amount of frontal swing is undesirable because the vehicle will require more road (and/or kerbside) space for turning than may be available. In some situations this may cause the vehicle to encroach into adjacent or opposing lanes, interfere with roadside objects, collide with parked or stopped vehicles, endanger pedestrians, or require reversing the vehicle in the middle of a turn.

In addition to the above, on the exit side of the turn, the path of the front outside corner of a semi-trailer with large front overhang may track outboard of the path of the front outside corner of the hauling unit. For these vehicles, the road space and safety implications are similar to those for the hauling unit described above.

Both of these aspects of frontal swing are controlled by this standard, and they are important in situations where a vehicle operates in an environment and traffic situations where tight turns are frequently required to be performed.

Part A of frontal swing tested by this performance measure specifically addresses issues related to the hauling unit. The second and third components (Parts B and C) of the frontal swing performance measure, which deal with the semi-trailer(s), are both measured on the exit side of the turn and perpendicular to the exit path. These relate to the distance the semi-trailer front outside corner tracks outside the path of the front outside corner of the unit ahead of it (prime mover or converter dolly); on a prime mover the front outside corner is the primary reference that the driver sees, knows and will use (assisted by side mirrors) to establish adequate clearance between the vehicle and road and/or kerbside objects adjacent to the path of the turn, to prevent collisions or conflicts. Part B controls the maximum of the difference between the two paths. It is referred to as the Maximum of Difference between frontal swing-out values, or frontal swing MoD. Part C of the measure, referred to as the Difference between the Maximum frontal swing-out values, or frontal swing DoM, places an upper limit on maximum swing-out of the front outside corner of the semi-trailer measured relative to the maximum swing-out of the prime mover.

The key parameters that influence frontal swing are the amount of front overhang forward of the steer axle, and for semi-trailers the amount of overhang forward of the kingpin. For a fixed amount of front-overhang, longer-wheelbases will generally increase frontal swing.

**PART (A) RIGID TRUCKS, PRIME MOVERS, BUSES AND COACHES**

**C8.2 Definition**

**(a) Summary statement**

The maximum width of the frontal swing swept path in a prescribed 90° low-speed turn.

**(b) Detailed statement**

When operating at the maximum laden mass and unladen, the maximum width of th frontal swing swept path of a vehicle participating in the Scheme in a prescribed 90° low speed turn performed at the specified speed must be no greater than the specified value.

**C8.3 Measure**

**(a) Performance value**

The maximum width of the frontal swing swept path must be measured and reported as th achieved performance value, in units of metres rounded up to the nearest 0.1 metre Th maximum width of the frontal swing swept path is the maximum distance, $FS_{max}$, betwee the outer and inner path trajectories of the frontal swing swept path envelope of the vehicl being assessed in the specified low-speed turn. The maximum distance, $FS_{max}$, is th straight-line segment intersecting both trajectories perpendicularly to their respectiv tangents at the intersection points. The swept path must be determined from the patl trajectories of:

(i) the outermost path scribed in the ground plane by the vertical projection of the furthes forward or outside point, or points, on the vehicle on the outside of the turn; and

(ii) the path scribed in the ground plane of the outer most point on the outer tyre sidewal nearest to the ground, on the forward most outside steered-wheel.

The above is summarised in Figure 8.

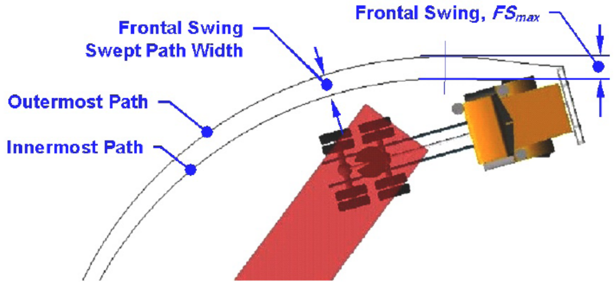



**Figure 8. Illustration of path trajectories in the Performance Based Standards low-speed turn of Part A of the frontal swing performance measure. For this measure the innermost path is the path followed by the outside front wheel.**

**(b) Performance levels**

**Table 11. Frontal swing performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1<br>Level 2<br>Level 3<br>Level 4 | For rigid trucks and prime movers no greater than 0.7 m, for buses and coaches no greater than 1.5 metres. |

**C8.4 Test specification**

**(a) Test load**

Same as for low-speed swept path.

**(b) Test conditions**

Same as for low-speed swept path.

**(c) Test procedure**

Same as for low-speed swept path.

**(d) Test method**

Same as for low-speed swept path.

**PART B: SEMI-TRAILERS, MAXIMUM OF DIFFERENCE (*MoD*)**

**C8.5 Definition**

**(a) Summary statement**

The maximum of the difference between the frontal swing-out distances between adjacent vehicle units in a prescribed 90° low-speed turn.

**(b) Detailed statement**

When operating at the maximum laden mass and unladen, the maximum of the difference between the frontal swing-out distances of adjacent vehicle units, one of which is a semi-trailer of a vehicle participating in the Scheme, when measured relative to the exit tangent of the prescribed 90° low-speed turn performed at the specified speed, must be no greater than the specified value.

**C8.6 Measure**

**(a) Performance value**

The maximum of the difference between the frontal swing-out distances of adjacent vehicle units, referred to as frontal swing MoD, must be measured and reported as the achieved performance value, in units of metres rounded up to the nearest 0.01 metre The difference between the frontal swing-out distances must be determined from the path trajectories of the outermost path scribed in the ground plane by the vertical projection of the furthest forward or outside point, or points, on each of two adjacent vehicle units, one of which is a semi-trailer. Frontal swing MoD is the maximum value of the straight-line segment intersecting both trajectories perpendicular to the low-speed turn exit tangent, as shown in Figures 9 to 12. If the frontal swing-out of the second vehicle unit is less than the frontal swing-out of the first vehicle unit throughout the entire turn then frontal swing MoD is not applicable and must be recorded as "not applicable".

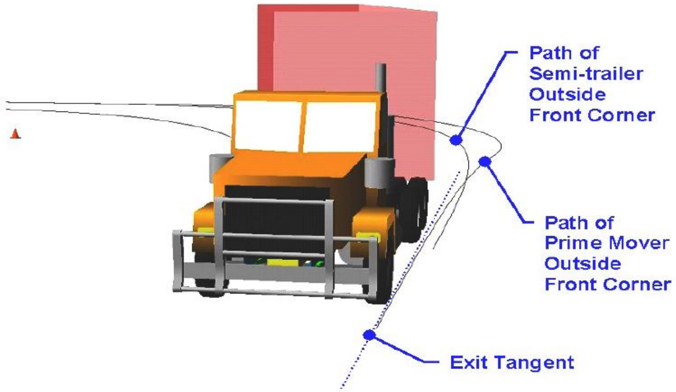



**Figure 9.** **Perspective view illustration of path trajectories partway through the Performance Based Standards low-speed turn for Parts B and C of frontal swing MoD and DoM performance measures.**

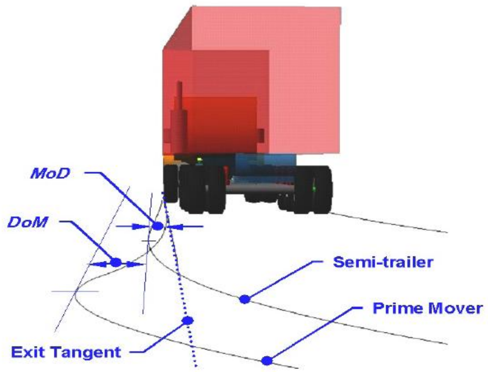



**Figure 10.** **Perspective view illustration of Parts B and C of the frontal swing MoD and DoM performance measures looking along the exit tangent during the final stages of the low-speed turn.**

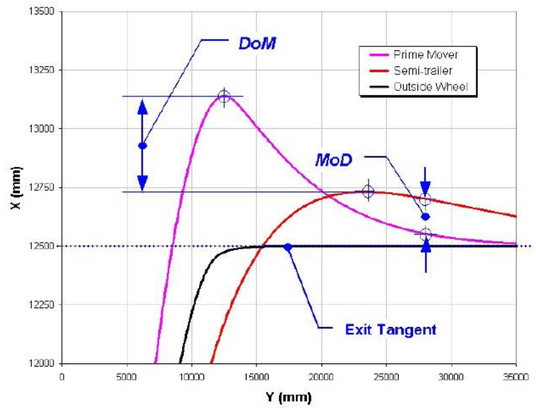



**Figure 11.** **Further detail on the Parts B and C frontal swing DoM and MoD performance measures. Note that the vertical (X) axis scale is exaggerated.**

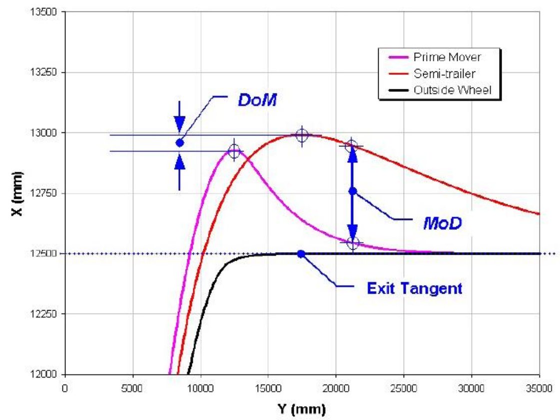



**Figure 12.** **A contrasting example to the one shown in Figure 11 of the Parts B and C frontal swing DoM and MoD performance measures. Note that the vertical (X) axis scale is exaggerated.**

**(b) Performance levels**

**Table 12. Maximum difference of frontal swing path performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | |
| Level 2 | |
| Level 3 | No greater than 0.40 metre |
| Level 4 | |

### C8.7 Test specification

**(a) Test load**

Same as for low-speed swept path.

**(b) Test conditions**

Same as for low-speed swept path.

**(c) Test procedure**

Same as for low-speed swept path.

**(d) Test method**

Same as for low-speed swept path.

## PART C: SEMI-TRAILERS, DIFFERENCE OF MAXIMA (*DoM*)

### C8.8 Definition

**(a) Summary statement**

The difference between the maximum frontal swing-out distances between adjacent vehicle units in a prescribed 90° low-speed turn.

**(b) Detailed statement**

When operating at the maximum laden mass and unladen, the difference between the maximum frontal swing-out distances of adjacent vehicle units of a vehicle participating in the Scheme, when measured relative to the exit tangent of the prescribed 90° low-speed turn performed at the specified speed, must be no greater than the specified value.

### C8.9 Measure

**(a) Performance value**

The difference between the maximum frontal swing-out distances of adjacent vehicle units, referred to as frontal swing DoM, must be measured and reported as the achieved performance value, in units of metres rounded up to the nearest 0.01 metre The difference between the maximum values of frontal swing-out distances must be determined from the path trajectories of the outermost paths scribed in the ground plane by the vertical projection of the furthest forward or outside point, or points, on each of two adjacent vehicle units, one of which is a semi-trailer. Frontal swing DoM is the length difference between the two longest line segments perpendicular to the low-speed turn exit tangent intersecting it and one of the path trajectories, as shown in Figures 9 to 12. A negative value of DoM must be reported when the frontal swing-out of the second vehicle unit is less than the frontal swing-out of the first unit, examples are shown in Figures 10 and 11. If MoD is recorded as “not applicable”, *DoM* must also be recorded as “not applicable”.

**(b) Performance levels**

**Table 13. Difference of maximum frontal swing out for performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | |
| Level 2 | |
| Level 3 | No greater than 0.20 metre |
| Level 4 | |

### C8.10 Test specifications

**(a) Test load**

Same as for low-speed swept path.

**(b) Test conditions**

Same as for low-speed swept path.

**(c) Test procedure**

Same as for low-speed swept path.

**(d) Test method**

Same as for low-speed swept path.

## C9: TAIL SWING

### C9.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting the road space requirement of a vehicle participating in the Scheme when making a tight turn at low speed.

#### (b) Intent

Tail swing is important in situations where vehicle units with a large amount of rear overhang operate in an environment where tight turns are frequently required. Where tail swing is large, the rear outside corner of a rigid truck, bus or coach, prime mover or semi-trailer may swing-out a significant distance at the commencement of a turn. For conventional vehicles tail swing is only significant during commencement of a turn, but it must be tested on the entry approach and exit to the turn when a vehicle is towing trailers with steerable axles.

In urban operations, vehicles with significant rear overhang (such as route buses or semi-trailers), and/or coupling rear overhangs (such as car carriers with the turntable located behind the drive axle) will exhibit significant amounts of tail swing when negotiating tight manoeuvres (such as buses and coaches exiting kerbside pickup areas). Collisions with vehicles in adjacent lanes (including cyclists) and roadside objects may result.

### C9.2 Definition

#### (a) Summary statement

The maximum outward lateral displacement of the outer rearmost point on a vehicle unit during the initial and final stages of a prescribed 90° low speed turn.

#### (b) Detailed statement

When operating at the maximum laden mass and unladen, the maximum outward lateral displacement of the outer rearmost point on a vehicle unit of a vehicle participating in the Scheme during the initial and final stages of a prescribed 90° low-speed turn performed at the specified speed must be no greater than the specified value.

### C9.3 Measure

#### (a) Performance value

The maximum tail swing during the initial and final stages of the prescribed turn, referred to as $TS_{entry}$ and $TS_{exit}$, respectively, must be measured and reported as the achieved performance value, in units of metres rounded up to the nearest 0.01 metre. Tail swing must be determined from the path trajectory of the outermost path scribed in the ground plane by the vertical projection of the furthest rearward or outside point, or points, on the vehicle unit having the greatest tail swing.

On the entry side of the turn, tail swing is the length of the longest line segment perpendicular to the low-speed turn entry tangent intersecting it and the path trajectory, similar to that shown in Figure 13.

On the exit side of the turn, tail swing is the length of the longest line segment perpendicular to the low-speed turn exit tangent intersecting it and the path trajectory, the same as that shown in Figure C9 referred to above. If there is no tail swing-out on the exi side of the turn "no swing-out" should be recorded.

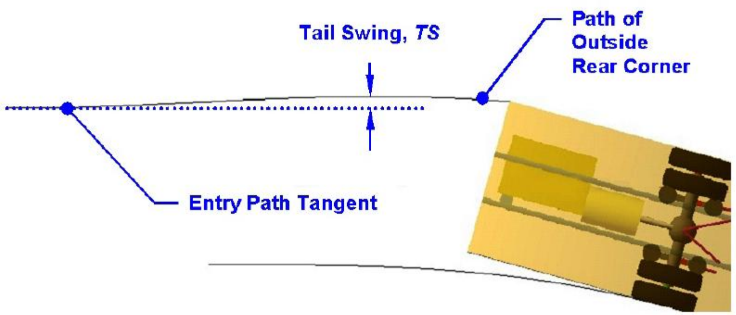



Figure 13. Illustration of tail swing performance measure at commencement of th Performance Based Standards low-speed turn.

#### (b) Performance levels

Table 14. Tail swing performance levels

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | No greater than 0.30 m |
| Level 2 | No greater than 0.35 m |
| Level 3 | No greater than 0.35 m |
| Level 4 | No greater than 0.50 m |

### C9.4 Test specification

#### (a) Test load

Same as for low-speed swept path.

#### (b) Test conditions

Same as for low-speed swept path.

#### (c) Test procedure

Same as for low-speed swept path.

#### (d) Test method

Same as for low-speed swept path.

## C10: STEER-TYRE FRICTION DEMAND

### C10.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting the likelihood of a vehicle participating in the Scheme losing steering control when making a tight turn at low speed.

#### (b) Intent

During a small-radius turn at low-speed, loss of steering will occur when the available tyre/road friction limit at the steer-tyres is exceeded. In this situation the vehicle will tend to "plough straight ahead" exhibiting significant heavy understeer risking low-speed collisions with other vehicles or roadside objects. This phenomenon has been observed to occur on the hauling units of multi-combination vehicles (road trains) featuring tri-axle drive systems that have a widely spread axle layout. This is generally not an issue for prime movers with single-axle (or tandem-axle) drive systems, and less of an issue for prime movers equipped with twin-steer axles.

The problem of friction demand and "steerability" in a low-speed turn is greatest when the axle spread on the drive group is large and the prime mover wheelbase is small. Fore-aft and lateral forces from the towed trailers that are imposed on the prime mover kingpin, and kingpin lead will also influence friction demand on the steer tyres. Increasing the vertical load on the steer tyres, or decreasing the drive group load also serves to decrease the friction demand on the steer tyres.

### C10.2 Definition

#### (a) Summary statement

The maximum friction level demanded of the steer tyres of the hauling unit in a prescribed 90º low speed turn.

#### (b) Detailed statement

When operating at the maximum allowed gross mass and unladen, the maximum friction level demanded of the steer tyres of the hauling in a prescribed 90º low-speed turn performed at the specified speed must be no greater than the specified value.

### C10.3 Measure

#### (a) Performance value

The maximum value of steer tyre friction demand must be measured and reported as the achieved performance value, in percentage units, rounded up to the nearest 1%. The friction demand of an axle or axle group is given by the following expression:

$$\text{friction demand (\%)} = 100\left(\frac{\text{friction required}}{\text{friction available}}\right)$$

$$= 100\frac{\left|\dfrac{\sum_{n=1}^{N}\sqrt{F_{xn}^2 + F_{yn}^2}}{\sum_{n=1}^{N} F_{zn}}\right|}{\mu_{peak}} \qquad \text{(C10)}$$

where:

$F_{xn}$ = longitudinal tyre force at $n$th tyre (N)
$F_{yn}$ = lateral tyre force at $n$th tyre (N)
$F_{zn}$ = vertical tyre force at $n$th tyre (N)
$N$ = number of tyres on the steer axle or axle group (-)
$\mu_{peak}$ = peak value of prevailing tyre/road friction (-)

Lateral, longitudinal and vertical tyres forces must be consistent with Society of Automotive Engineers (1976) as shown in Figure 14.

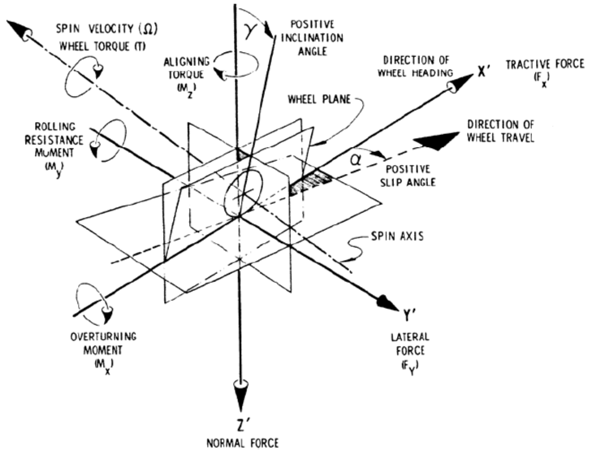



**Figure 14. SAE Tyre Axis System (Society of Automotive Engineers, 1976)**

The numerator in equation (C10) represents the absolute value of the ratio of the sum of horizontal tyre forces (all tyres on the axle, or axle group) to the vertical load supported by the axle, or axle group. Equation (C10) must be used to calculate steer tyre friction demand and its maximum value in the prescribed low-speed turn.

### (b) Performance levels

**Table 15. Steer tyre friction demand performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| All levels | Not greater than 80% of the maximum available tyre/road friction limit. |

## C10.4 Test specification

### (a) Test load

Same as for low-speed swept path.

### (b) Test conditions

Same as for low-speed swept path.

### (c) Test procedure

Same as for low-speed swept path.

### (d) Test method

Same as for low-speed swept path.

## C11: STATIC ROLLOVER THRESHOLD

## C11.1 Purpose and intent

### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting the rollover tendency of a vehicle participating in the Scheme during steady turns.

### (b) Intent

A vehicle travelling along a curved path is subjected to an outward force and an overturning moment that is proportional to the lateral (or sideways) acceleration. Rollover occurs when the lateral acceleration that causes the overturning moment is sufficient to exceed the vehicle's rollover stability threshold.

Rollover stability is the most significant safety issue and arguably the most important performance measure for heavy vehicles because it has been strongly linked to rollover crashes. Crashes that involve heavy vehicle rollover are strongly associated with severe injury and fatalities (Winkler et al, 2000).

The basic measure of rollover stability is static rollover threshold, usually expressed as a fraction of the acceleration due to gravity in units of 'g', where 1g is an acceleration of $9.807 m/s^2$ corresponding to the force exerted by the earth's gravitational field. High values of static rollover threshold imply better resistance to rollover.

Rollover stability is very sensitive to the ratio of the overall track width to the height above ground of the centre of gravity of the vehicle. Rollover stability increases either by increasing this width or by decreasing centre of gravity height. Suspension properties influence static rollover stability but they are generally of lesser importance when compared with the ratio of track width to centre of gravity height.

Rollover stability for multiple trailer combinations is much more complex than for single, rigid units and depends on the type of coupling between trailers. Trailers that are connected through a turntable are said to be "roll-coupled" and will rollover together as connected units, whereas full trailers (comprising a dolly and semi-trailer) connected by a pin coupling, can both roll and rollover independently of the other units in the combination. This means that any full trailer in a combination reaching its own stability limit first would rollover before other trailers in the combination. This also applies to entire roll-coupled units within combinations, such as B-double trailer combinations in triple- or quad-trailer configurations (AB-triple, AAB-quad, or BAB-quad).

## C11.2 Definition

### (a) Summary statement

The steady state level of lateral acceleration that a vehicle can sustain without rolling over during turning.

### (b) Detailed statement

When operating up to the maximum laden mass and least favourable load conditions, the highest steady state level of lateral acceleration that a vehicle participating in the Scheme can sustain without rolling over must be no less than the specified value.

### C11.3 Measure

#### (a) Performance value

The rollover threshold of the vehicle, or vehicle unit with the lowest rollover stability, must be measured and reported as the achieved performance value, expressed as a fraction of the acceleration due to gravity in units of 'g', rounded down to the nearest 0.01g. For single unit vehicles, such as rigid trucks, buses and coaches, the rollover threshold is the lateral acceleration[13] of the sprung mass centre of gravity measured at the point of rollover instability. For combination and multi-combination vehicles, the rollover threshold is the resultant lateral acceleration of any unit or roll-coupled set of units, $AY_{rcu}$, as defined by equation C11.2a of this report, measured at the point of rollover instability.

Rollover instability is achieved when the lateral acceleration, or resultant lateral acceleration, starts to decrease with increasing roll angle, as illustrated in Figure 15 for a prime mover and semi-trailer vehicle or roll-coupled set. In general for heavy vehicles, the point of roll instability is also achieved when, or immediately after, the vertical load on all tyres along one side of the vehicle, excluding the tyres on the lightly loaded side of a steer axle(s) with soft springs, have reduced to zero.

Further, for combination and multi-combination vehicles, if the roll coupling between units in any set of units is either weak or is not continuous (such as found in double oscillating turntables or similar), and the roll angle[14] of any unit within the set is greater than 30º, rollover instability must be assumed to have been achieved.

Vehicles that reach or exceed the tyre/road friction limits before rollover occurs, and achieve a steady state lateral acceleration that is not less than the required performance level are deemed to have acceptable rollover stability.

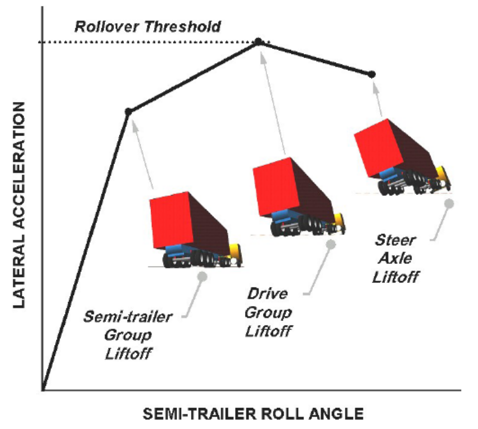



**Figure 15. Typical axle lift-off sequence and rollover threshold for a prime mover and semi-trailer combination.**

Vehicles that reach or exceed the tyre/road friction limits before rollover occurs, and achieve a steady state lateral acceleration that is not less than the required performance level are deemed to have acceptable rollover stability.

#### (b) Performance levels

**Table 16. Static rollover threshold performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| All levels | Road tankers hauling dangerous goods in bulk and buses and coaches not less than 0.40g. All other vehicles not less than 0.35g |

#### (c) Calculation of resultant lateral acceleration of roll-coupled units

This sub-section specifies the method that must be used to calculate the resultant lateral acceleration of a roll-coupled set of vehicle units in a multi-combination vehicle.

The parameter of prime interest is the resultant overturning moment and the key consideration being whether or not this moment is sufficient to cause the roll-coupled units to rollover. When the proximity of the roll-coupled units to rollover is expressed in terms of the resultant lateral acceleration, the static rollover threshold performance measure can be applied directly.

For the example of the 2 trailer roll-coupled rear units illustrated in Figure 16, the resultant lateral acceleration (ignoring the contribution from the dolly) when expressed in terms of roll moments is given by the following:

$$AY_{rcu} = \frac{m_1 h_1 AY_1 + m_2 h_2 AY_2}{m_1 h_1 + m_2 h_2} \qquad \text{(C11.1)}$$

where:

$AY_{rcu}$ = resultant lateral acceleration of the roll-coupled units (m/s$^2$)
$m_{1,2}$ = semi-trailer sprung mass (kg)
$h_{1,2}$ = height of sprung mass centre of gravity (m)
$AY_{1,2}$ = lateral acceleration[15] of sprung mass centre of gravity (m/s$^2$)

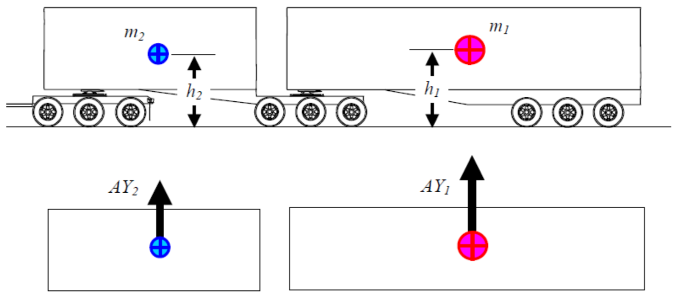



**Figure 16. Side and plan view illustration of 2 roll-coupled trailers showing sprung masses, sprung mass centre of gravity heights, and lateral accelerations, *AY*.**

Equation (C11.2a) shown below is the generalised form of equation (C11.1), which can be used to determine the resultant lateral acceleration for vehicles having any number (*N*) of roll-coupled units.

$$AY_{rcu} = \frac{\sum_{n=1}^{N} m_n h_n AY_n}{\sum_{n=1}^{N} m_n h_n} \qquad \text{(C11.2a)}$$

As the last trailer of a roll-coupled trailer set will generally have both a greater mass and slightly higher centre of gravity than the trailers ahead of it, as illustrated above in Figure 16, the average lateral acceleration will be weighted more heavily to the larger value of the rear trailer. However, it is useful to note that when the masses and centre of gravity heights of all the roll-coupled units are identical, equation (C11.2a) reduces to equation (C11.2b), which is simply the average value of lateral accelerations at any instant during the manoeuvre.

$$AY_{rcu} = \frac{\sum_{n=1}^{N} AY_n}{N} \qquad \text{(C11.2b)}$$

### C11.4 Test specification

#### (a) Test load

The vehicle being assessed must be tested at the *maximum laden mass* and in both turn directions at the least favourable load conditions. Each tyre on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyre. Each tyre must be inflated to the pressure as specified by the vehicle and/or tyre manufacturer.

The tread depth of each tyre must not decrease by more than 2 mm during field testing.

#### (b) Test conditions

The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

#### (c) Test procedure

One of the following two test procedures must be used to measure static rollover threshold:

(i) Constant radius quasi-steady turn – The vehicle being assessed must be driven along the specified path at an initial speed that is at least 10 km/h slower than the speed at which the rollover instability will occur. The path of the specified turn that the driver will use to guide the vehicle must be circular and of radius not less than 100 metres. In the turn the driver must steer the vehicle along the specified path. The vehicle must be steered such that the vertical projection in the ground plane of a point at the centre of the forward-most steer axle follows the specified path. Using the above point as a reference, the driver must steer the vehicle along the specified path and maintain a lateral distance error between the reference point and the specified path not greater than 1.5 metres. From the initial speed, which must be maintained for at least 15 seconds on the specified path, the driver must increase the speed of the vehicle to the point of rollover instability at:

(a) an average rate, measured over any 5-second period, not greater than 0.5 km/h per second; or

(b) in increments of 2 km/h per lap.

This procedure is particularly relevant to long multi-combination vehicles that take much longer to reach steady turn conditions than short vehicles.

(ii) Tilt table – In accordance with recommended practice SAE J2180 (Society of Automotive Engineers, 1998).

#### (d) Test method

Numerical modelling (computer-based simulation) or field-testing.

## C12: REARWARD AMPLIFICATION

### C12.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting the lateral-directional response of multi-articulated vehicles participating in the Scheme in avoidance manoeuvres performed at highway speeds without braking.

#### (b) Intent

Rearward amplification generally pertains to heavy vehicles with more than one articulation point, such as truck-trailers and road train combinations. It shows as a tendency for the following or trailing unit(s) to experience higher levels of lateral acceleration than the towing unit. It is a serious safety issue in rapid path-change manoeuvres as it can lead to rear-trailer rollover.

As the name rearward amplification suggests, each unit in the combination experiences lateral acceleration that is an amplification of that experienced by the unit immediately ahead of it, and thus amplification of lateral acceleration increases toward the rear of the vehicle. Lower values of rearward amplification indicate better performance. High values of rearward amplification imply high probabilities of rear-trailer rollover. In some cases rearward amplification is less than unity (i.e. lateral acceleration is attenuated).

Rearward amplification improves with fewer articulation points, a shorter distance from the centre of gravity of the hauling unit to the hitch point, roll-coupling through turntables at articulation points, shorter coupling rear overhang, longer drawbar lengths on dollies, longer trailer wheelbase, and tyres with higher cornering-stiffness.

### C12.2 Definition

#### (a) Summary statement

The degree to which the trailing unit(s) amplify the lateral acceleration of the hauling unit.

#### (b) Detailed statement

When operating up to the maximum allowed gross mass and least favourable load conditions, the ratio of the maximum value of the specified lateral acceleration response of the rearmost unit, or rearmost roll-coupled units, to the lateral acceleration input measured at the steer axle of the vehicle being assessed in the specified test must be no greater than the specified value.

### C12.3 Measure

#### (a) Performance value

The maximum value of the ratio of the specified lateral acceleration[16] response of the rearmost unit or roll-coupled set of units (referred to as the lateral acceleration output) to the specified lateral acceleration input, measured at the steer axle of the vehicle being assessed (referred to as the lateral acceleration input), must be measured and reported as the achieved performance value, expressed as a non-dimensional quantity, rounded up to the nearest 0.01. The specified lateral acceleration input must be measured at the centre of the forward-most steer axle of the vehicle in the specified single lane-change manoeuvre defined in Section C12.4(c). The specified lateral acceleration output must be measured, and rearward amplification calculated, in accordance with Section C12.3 (c).

#### (b) Performance levels

**Table 17. Rearward amplification performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| All levels | Not greater than 5.7 times the static rollover threshold of the rearmost unit or roll-coupled set of units taking account of the stabilising influence of the roll coupling. |

#### (c) Measurement of rearward amplification

The general definition of rearward amplification presented in terms of lateral acceleration that must be used is given by the following:

$$RA = \frac{|AY|_{\max} \; of \; following \; vehicle \; unit}{|AY|_{\max} \; of \; first \; vehicle \; unit} \qquad \text{(C12.1)}$$

The following is ascribed to the numerator and the denominator terms, respectively:

$|AY|_{\max}$ *of following vehicle unit* = maximum absolute value of the lateral acceleration of the centre of mass of the sprung mass of the last vehicle unit ($m/s^2$)

$|AY|_{\max}$ *of first vehicle unit* = maximum absolute value of the lateral acceleration of the centre of the front axle ($m/s^2$)

#### (i) Rigid vehicles and combination vehicles with a single rear unit

For rigid vehicles the lateral acceleration of the "following vehicle unit" described above in equation (12.1) is simply that of the centre of gravity of the sprung mass. That is, for rigid vehicles the "following vehicle unit" and "first vehicle unit" is the same unit.

Where the last unit comprises a single full trailer, namely, a dolly and a single semi-trailer, the lateral acceleration of the "following vehicle unit" described above is simply that of the centre of gravity of the sprung mass of the last semi-trailer.

#### (ii) Combination Vehicles with Roll-Coupled Rear Units

All units in a vehicle combination that are connected to each other either by turntables or other means of transmitting overturning-moments are said to be roll-coupled. The rearmost roll-coupled units (rrcu) in a vehicle combination are all the rear units that are connected by mechanical components or devices able to transmit overturning moments. The rrcu in 3 example vehicles is illustrated below in Figure 17.

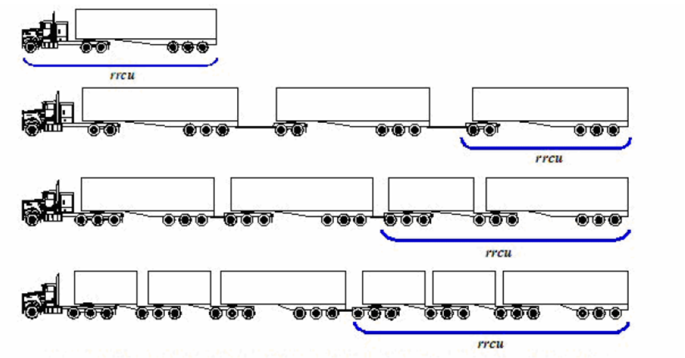



**Figure 17. Illustration of rearmost roll-coupled units (rrcu) in 3 example vehicles; a prime mover and semi-trailer combination, an A-triple, an AAB-quad and a Double B-Triple.**

Combination vehicles having rear units that are roll-coupled are generally able to perform better in a lane change manoeuvre than a vehicle having the same number of units but where roll coupling is not present. This is because where roll-coupling is present, overturning moments acting on one semi-trailer are transmitted through the points of connection, such as turntables, to the adjacent units thereby providing some additional roll support. Adjacent units connected by a turntable will only overturn when both units rollover.

The following method must be used to calculate rearward amplification for vehicles with rear units that are roll-coupled. The method is better able to quantify the proximity of the rearmost trailer set to rollover in the lane change manoeuvre, and, therefore, is consistent with setting a performance level linking rearward amplification to the static rollover threshold of the rearmost roll-coupled units.

### (iii) Lateral acceleration of roll-coupled rear units

Where the rearmost vehicle units comprise, say, two or more semi-trailers that are roll-coupled, the lateral acceleration of the "following vehicle unit" described above in equation (12.1) must be calculated using the following scheme.

The parameter of prime interest is the resultant overturning moment and the key consideration being whether or not this moment is sufficient to cause the rear roll-coupled units to rollover. When the proximity of the rear roll-coupled units to rollover is expressed in terms of the lateral acceleration it can be compared directly to the static rollover threshold.

For the example 2 trailer roll-coupled rear units illustrated in Figure 17 the resultant lateral acceleration when expressed in terms of roll moments at each instant in the lane change is given by following (which is the same as equation C11.1):

$$AY_{rcu} = \frac{m_1 h_1 AY_1 + m_2 h_2 AY_2}{m_1 h_1 + m_2 h_2} \qquad \text{(C12.2)}$$

where:

- $AY_{rcu}$ = resultant lateral acceleration of the roll-coupled units (m/s$^2$)
- $m_{1,2}$ = semi-trailer sprung mass (kg)
- $h_{1,2}$ = height of sprung mass centre of gravity (m)
- $AY_{1,2}$ = lateral acceleration of sprung mass centre of gravity (m/s$^2$)

Equation (C12.3a) shown below is the generalised form of equation (C12.2), which can be used to determine the resultant lateral acceleration for vehicles having any number ($N$) of roll-coupled rear units.

$$AY_{rcu} = \frac{\sum_{n=1}^{N} m_n h_n AY_n}{\sum_{n=1}^{N} m_n h_n} \qquad \text{(C12.3a)}$$

As the last trailer of a roll-coupled trailer set will generally have both a greater mass and slightly higher centre of gravity than the trailers ahead of it, as illustrated in Figure 16, the average lateral acceleration will be weighted more heavily to the larger value of the rear trailer. However, it is useful to note that when the masses and centre of gravity heights of all the roll-coupled rear units are identical, equation (C12.3a) reduces to equation (C12.3b), which is the mean of the lateral accelerations at any instant during the manoeuvre.

$$AY_{rcu} = \frac{\sum_{n=1}^{N} AY_n}{N} \qquad \text{(C12.3b)}$$

For a two-unit roll-coupled rear trailer set, with each unit having identical masses and centre of gravity heights, equation (C12.3b) shows that if the lateral accelerations are equal in magnitude and opposite in sign (making the average of $AY_{rcu}$ equal to zero), the motion of the roll-coupled rear trailers are exactly out of phase and the roll moments exactly balance each other, one trailer trying to roll to the left while the other is trying to roll to the right. While this is unlikely to occur in practice, it is a simple illustration of the positive influence of roll coupling.

The generalised version of equation (C12.1), which must be applied as part of this specification to vehicles with rear units that are roll-coupled is given below in equation (C12.4).

$$RA_{rcu} = \frac{|AY_{rcu}|_{max} \text{ of last vehicle unit}}{|AY|_{max} \text{ of steer axle}} \qquad \text{(C12.4)}$$

where:

$|AY_{rcu}|_{max}$ of last vehicle unit = maximum absolute value of the resultant lateral acceleration of the roll-coupled rear units determined from equation (C12.3a) (m/s$^2$)

### (iv) When all units are roll-coupled

As the input to the manoeuvre and excitation to the vehicle occurs at the hauling unit, the output/input relationship must clearly distinguish between the hauling unit (the steer input, or the source of the excitation) and the roll-coupled trailer set (the location where the response occurs). Therefore, when all units in a vehicle combination are roll-coupled, examples being a prime-mover and semi-trailer combination, B-doubles, B-triples and B-quad combinations, equation (C12.3a) should be applied to the semi-trailers. That is, the numerator of equation (C12.4) must include only a single semi-trailer for a prime mover and semi-trailer combination, and double, triple and quad semi-trailer sets for B-double, B-triple and B-quad combinations, respectively.

### C12.4 Test specification

The vehicle must execute a single lane change manoeuvre in accordance with the "Single Lane-Change", "Single Sine-Wave Lateral Acceleration Input", specified in ISO 14791:2000(E) (International Standards Organisation, 2000)[17]. The basic course layout must be used. The manoeuvre must have a maximum lateral acceleration of not less than 0.15g and a steer frequency equal to 0.40 Hz. The test must be conducted at 88 km/h.

The driver must steer the vehicle along the specified path and maintain a lateral distance error between the reference point – taken to be the vertical projection in the ground plane of the centre of the forward most steer axle – and the specified path that is either:

- not greater than 30 mm; or,
- as specified in ISO 14791 such that the lateral acceleration and frequency of the input is not less than the value for the manoeuvre specified in the paragraph above.

If the specifications given in ISO 14791 differ to those described here, this specification takes precedence.

### (a) Test load

In accordance with the "Single Lane-Change", "Single Sine-Wave Lateral Acceleration Input", specified in ISO 14791:2000(E) (International Standards Organisation, 2000).

Additionally, the vehicle being assessed must be tested laden at the least favourable load condition. If load asymmetries constitute the least favourable load condition then the vehicle must also be tested in both turn directions.

### (b) Test conditions

In accord with the "Single Lane-Change", "Single Sine-Wave Lateral Acceleration Input", specified in ISO 14791:2000(E) (International Standards Organisation, 2000).

### (c) Test procedure

In accord with the "Single Lane-Change", "Single Sine-Wave Lateral Acceleration Input", specified in ISO 14791:2000(E) (International Standards Organisation, 2000).

### (d) Test method

Numerical modelling or field-testing.

## C13: HIGH-SPEED TRANSIENT OFFTRACKING

### C13.1 Purpose and intent

### (a) Purpose

The primary purpose of this standard is to manage safety risk by limiting the sway of the rearmost trailers of multi-articulated vehicles participating in the Scheme in avoidance manoeuvres performed without braking, at highway speeds.

### (b) Intent

In an abrupt evasive manoeuvre, the lateral displacement of the rear end of the last trailer of an articulated vehicle may "overshoot" the final path of the front axle of the hauling unit; the path achieved after the hauling unit has completed the manoeuvre and stabilised in its new straight ahead path parallel to its original path. The amount of overshoot, referred to as high-speed transient offtracking (and sometimes also referred to as trailer overshoot), can be viewed as an indication of the severity of intrusion into an adjacent or opposing lane, striking a kerb or dropping off the road seal (thus precipitating rollover) or collision with a roadside objects.

Crash studies suggest there is a trend of crash rate increasing with increased high-speed transient off tracking. The crash consequences of heavy vehicles performing avoidance manoeuvres will depend on the road environment and factors such as lane width and traffic volume. Where lane widths are narrow and traffic volumes are high, it is desirable for heavy vehicles to have lower levels of high-speed transient offtracking.

The parameters that influence rearward amplification have similar strong influences on High-Speed Transient Offtracking.

### C13.2 Definition

### (a) Summary statement

The lateral distance that the last-axle on the rearmost trailer tracks outside the path of the steer axle in a sudden evasive manoeuvre.

### (b) Detailed statement

When operating up to the maximum laden mass and least favourable load conditions, the maximum lateral displacement between the specified point on the rearmost axle of the rearmost vehicle unit of a vehicle participating in the Scheme and the exit tangent in the specified test must be no greater than the specified value.

### C13.3 Measure

### (a) Performance value

The maximum lateral distance between the path trajectory of the specified point on the vehicle being assessed, measured in the ground plane and perpendicular to the exit tangent of the single lane change, single sine-wave lateral acceleration input, test course must be measured and reported as the achieved performance value, expressed in units of metres and rounded up to the nearest 0.1 metre. The specified point on the vehicle is the vertical projection in the ground plane of a point at the centre of the rearmost axle of the rearmost vehicle unit, as illustrated in Figure 18. When the maximum distance corresponds to an overshoot situation, as shown in Figure 18 and defined in Figure 19, the performance value must be recorded as a positive distance. When the maximum distance corresponds to an undershoot situation, as illustrated in Fig. C13.2, the performance value must be recorded as a negative distance.

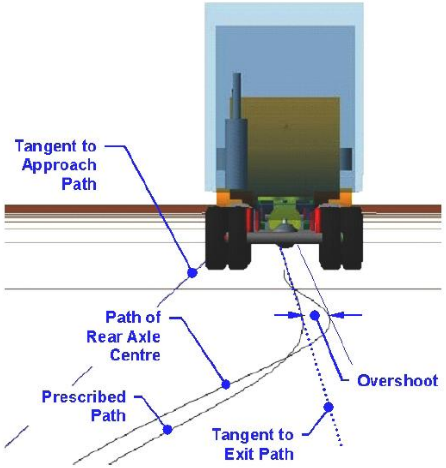



Figure 18. Perspective view illustration of final stages of the single lane change manoeuvre showing overshoot dimension in the ground plane of the rear axle centre for high-speed transient offtracking.

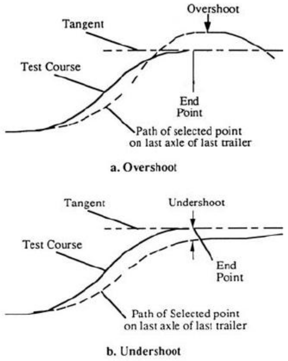



Figure 19. Illustration of high-speed transient offtracking overshoot and undershoot scenarios from Society of Automotive Engineers (1993a).

### (b) Performance levels

**Table 18. High speed transient offtracking performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| Level 1 | No greater than 0.6 m |
| Level 2 | No greater than 0.8 m |
| Level 3 | No greater than 1.0 m |
| Level 4 | No greater than 1.2 m |

## C13.4 Test specification

### (a) Test load

Same as for rearward amplification.

### (b) Test conditions

Same as for rearward amplification.

### (c) Test procedure

Same as for rearward amplification.

### (d) Test method

Same as for rearward amplification.

### (e) Further Notes

If the specification in ISO 14791:200(E) differs to those described here, this specification takes precedence.

## C14: YAW DAMPING COEFFICIENT

### C14.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk by requiring acceptable attenuation of any sway oscillations of rigid vehicles participating in the Scheme or between the trailers of multi-articulated vehicles participating in the Scheme.

#### (b) Intent

An important consideration in the stability and handling of heavy vehicles is how quickly swing or sway oscillations take to "settle down" or decay after a severe manoeuvre has been performed. Vehicles that take a long time to settle increase the driver's workload[18] and represent a higher safety risk to other road users and to the driver. The yaw damping coefficient performance measure quantifies how quickly "sway", or yaw oscillations take to settle after application of a short duration steer input at the hauling unit.

Yaw damping decreases with speed, and at higher speeds the oscillations may take longer to decay or they may become divergent (increase in amplitude) and lead to rollover.

The parameters that influence rearward amplification have similar strong influences on yaw damping coefficient.

### C14.2 Definition

#### (a) Summary statement

The rate at which "sway" or yaw oscillations decay after a short duration steer input at the hauling unit.

#### (b) Detailed statement

When operating up to the maximum allowed gross mass and least favourable load conditions, the maximum rate at which yaw oscillations decay in the specified test must be no less than the specified value.

### C14.3 Measure

#### (a) Performance value

The damping ratio calculated from the specified motion variable in the specified test must be measured and reported as the achieved performance value, expressed as a dimensionless quantity and rounded down to the nearest 0.01. The specified motion variable is 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 20. The mean value of the amplitude ratios, $\bar{A}$, must be calculated separately for each articulation joint, or unit, using the following equation:

$$\bar{A} = \frac{1}{n-2}\left(\frac{A_1}{A_3} + \frac{A_2}{A_4} + \frac{A_3}{A_5} + \ldots + \frac{A_{n-2}}{A_n}\right) \qquad \text{(C14a)}$$

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

$$D = \frac{\ln(\bar{A})}{\sqrt{(2\pi)^2 + [\ln(\bar{A})]^2}}, \qquad \text{(C14b)}$$

If the 5% limit referred to above is reached before the 6th amplitude, then the following formulae may be used in place of equations (C14a) and (C14b), respectively:

$$\bar{A} = \frac{1}{n-1}\left(\frac{A_1}{A_2} + \frac{A_2}{A_3} + \frac{A_3}{A_4} + \ldots + \frac{A_{n-1}}{A_n}\right) \qquad \text{(C15a)}$$

$$D = \frac{\ln(\bar{A})}{\sqrt{(\pi)^2 + [\ln(\bar{A})]^2}}, \qquad \text{(C15b)}$$

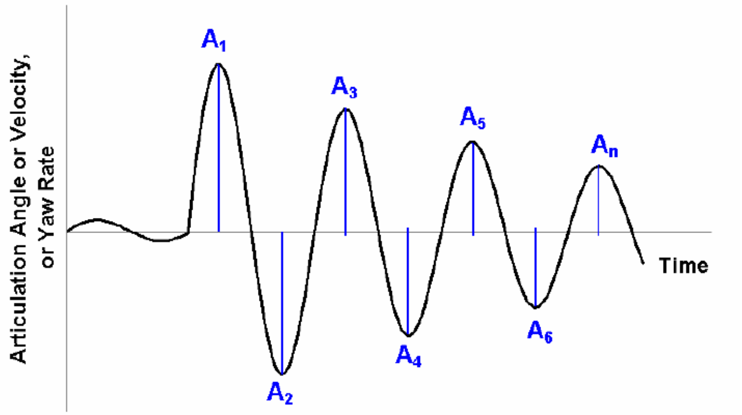



Figure 20. Determination of amplitudes for damping ratio calculation.

### (b) Performance levels

**Table 19. Yaw damping performance levels**

| Performance Based Standards Road Class | Performance Level Required |
|---|---|
| All levels | Not less than 0.15 at the certified vehicle speed. |

## C14.4 Test specifications

### (a) Test load

In accordance with the "Pulse Input", "Steer Impulse", method specified in ISO 14791:2000(E) (International Standards Organisation, 2000).

Additionally, the vehicle being assessed must be tested laden at the least favourable load conditions and include load asymmetries. If load asymmetries constitute the least favourable load condition then the vehicle must also be tested in the left and right steer directions.

### (b) Test conditions

In accord with the "*Pulse Input*", "*Steer Impulse*", method specified in ISO 14791:2000(E) (International Standards Organisation, 2000).

### (c) Test procedure

In accord with the "*Pulse Input*", "*Steer Impulse*", method specified in ISO 14791:2000(E) (International Standards Organisation, 2000).

### (d) Test method

Numerical modelling or field-testing.

### (e) Further notes

If the specification in ISO 14791:2000(E) differs to those described here, this specification takes precedence.

# C15: HANDLING QUALITY (UNDERSTEER/OVERSTEER)

## C15.1 Purpose and intent

### (a) Purpose

The primary purpose of this standard is to manage safety risk by ensuring adequate steering control over a wide range of turn conditions.

### (b) Intent

For reasons of practicality and safety, a heavy vehicle should be controllable and stable enough to follow a desired path in response to steering. Handling quality, expressed in terms of the "understeer/oversteer" coefficient, in simple terms refers to the responsiveness and "feel" of the vehicle to driver steering control. The understeer/oversteer behaviour of heavy vehicles has been found to vary significantly with lateral acceleration, and in some situations this marked change in steering response may make the vehicle difficult to control or unstable.

Handling quality will be primarily influenced by the mechanical properties of the prime mover, including hauling unit wheelbase, king-pin lead, tyre cornering stiffness, steer-axle roll centre height, roll steer coefficient and total roll stiffness.

## C15.2 Definition, measure and test specification

This standard has yet to be defined. Its three main components (a performance measure, a test procedure from which to obtain the performance measure and a performance level, or levels, to be satisfied) are not able to be defined to an acceptable level of robustness at this time based on current research.

There is one recognised method[19] for assessing heavy vehicle handling quality. The method was developed as a research tool and is presently not considered to be sufficiently robust to be incorporated as a performance standard. One known deficiency with the method is that its results are very sensitive to minor changes in vehicle design parameters. Further research is required in order to determine exactly what constitutes satisfactory (or unsatisfactory) heavy vehicle handling. This will enable the setting of a performance level, or levels, for this standard to be completed in a robust manner. Setting the performance level(s) such that they are too easy to satisfy will potentially allow all vehicles to pass the standard, regardless of their performance. Conversely, setting the performance level(s) such that they are too difficult to satisfy may potentially disallow all vehicles.

The NTC has initiated a project to finalise this standard in its 2007/2008 work programme. A proposal will be submitted to the Australian Transport Council for approval in 2008 following consultation with industry and Transport Agency Chief Executives.

## C16: DIRECTIONAL STABILITY UNDER BRAKING

### C16.1 Purpose and intent

#### (a) Purpose

The primary purpose of this standard is to manage safety risk of vehicle instability when braking in a turn or on pavement cross slopes.

#### (b) Intent

The ability of a vehicle to remain stable, controllable and kept within its lane during heavy braking is a key safety consideration in all road transport tasks and in all areas of heavy vehicle operation – urban, regional and remote. Rollover or loss of control (such as if a jack-knife occurs), present high safety risks to the driver and to other road users, which can lead to injury and fatalities.

Heavy braking in a turn is a challenging manoeuvre that subjects the vehicle to a complex combination of longitudinal and lateral acceleration placing severe demands on both driver skill and vehicle performance. A high level of stability reduces the likelihood of a crash and is therefore desirable, particularly in environments where traffic volumes and/or travel speeds are high, and the probability of a crash having a severe outcome is great.

### C16.2 Definition

#### (a) Summary statement

The ability to maintain directional stability under braking.

### C16.3 Measure

A vehicle must meet the performance level specified in paragraph (a) below, or one of the arrangements specified in paragraph (b) below must exist.

#### (a) Performance Level

- A vehicle must not exhibit gross wheel lock-up behaviour in any loading condition and must remain in a straight lane of width equal to that specified in the standard 'Tracking ability on a straight path' for the corresponding level of operation when it is braked from 60 km/h to achieve the assessment deceleration level on a high-friction surface roadway. The proposed assessment deceleration levels are shown in Table 20.

Table 20. Deceleration levels for vehicles participating in the Scheme

| Performance Based Standards Network Access Level | Typical vehicle configuration | Average Deceleration from 60 km/h |
|---|---|---|
| Single motor vehicles (All access levels) | Rigid trucks and buses | 0.40 g |
| 1 | Semi-trailers | 0.35 g |
| 2 | B-double combinations | 0.30 g |
| 3 | Road-train A-doubles and B-triples | 0.25 g |
| 4 | Road-Train A-triples | 0.20 g |

- A vehicle (part) that relies upon an antilock brake system to comply with the standard must have automatic brake adjusters on each axle that are controlled by the anti-lock brake system.
- A (motor) vehicle that has an auxiliary brake system fitted that could produce an average deceleration of 0.1g or higher on the unladen (motor) vehicle must not be capable of applying automatically. This requirement is deemed to be met if the (motor) vehicle has an antilock brake system that both controls all the drive axle group wheels and has veto control over the auxiliary brakes.
- All parts of the (combination) vehicle must comply with the applicable Australian Design Rule for braking at the time of manufacture. If a load-proportion brake system is fitted it must comply with the unladen compatibility requirements in Australian Design Rule 35/02 (motor vehicles) or Australian Design Rule 38/03 (trailers). Concessions against the Australian Design Rule requirements in individual cases that have been agreed to by the Administrator of Motor Vehicle Standards are to be recognised when assessing compliance with this item.

Definitions:

The absence of gross wheel lock-up is defined as follows:

- Single and tandem axle groups do not exhibit sustained wheel lock-up on any axles at the assessment deceleration levels.
- Tri-axle and quad axle groups can exhibit sustained wheel lock-up on one axle only in the group at the assessment deceleration level.
- An axle group with more than four axles can exhibit wheel lock-up on any two axles in the group at the assessment deceleration level.
- Wheel lock-up that occurs on any wheel at speeds below 10 km/h can be ignored because the potential for a vehicle to deviate seriously from the preferred path is minor.

Axle groups are defined as follows:

- The axle-group definitions applicable to Australian Design Rules 35 and 38 are pertinent.
- The axles on a dolly trailer should be considered as forming one axle group.

#### (b) Deemed-to-comply provisions

Three deemed-to-comply arrangements are available as alternatives to independently finding means of meeting the proposed standard:

- a vehicle that has a functioning anti-lock brake system that effectively prevents gross wheel lock-up on each axle group (as defined in the definitions) is deemed to comply with the standard; or
- a motor vehicle in a combination vehicle that has a functioning anti-lock brake system that effectively prevents gross wheel lock-up behaviour on the motor vehicle, can be ignored when the test or simulation assessment is made. That is, the motor vehicle is deemed-to-comply and only the performance of the trailer(s) against the performance standard needs be addressed; or

- a combination vehicle that has a load proportioning brake system on each part that has been set to meet the lightly laden compatibility limits in the pending revisions to Australian Design Rules 35 and 38 (Australian Design Rule 35/02 and 38/03) is deemed to comply with this standard. Note that a motor vehicle that has an antilock brake system as described in the preceding paragraph and trailer(s) that meet the lightly laden compatibility limits are deemed to comply with this standard.

#### C16.4 Test specifications

#### (a) Test load

The vehicle being assessed should be tested in the unladen condition. If the vehicle complies in the unladen condition, it is deemed to comply in the laden condition.

Each tyre on the vehicle must have a tread depth of at least 90% of the original value over the whole tread width and circumference of the tyre. Each tyre must be inflated to the pressure as specified by the vehicle and/or tyre manufacturer. The tread depth of each tyre must not decrease by more than 2 mm during field testing.

#### (b) Test conditions

The test site must have uniform, smooth, dry, hard pavement, which is free from contaminants. The surface must have a coefficient of friction value, $\mu_{peak}$, at the tyre/road contact surface of not more than 0.80.

#### (c) Test procedure

The test (initial) speed should be in the range 59 – 65 km/h.

The point where deceleration starts and the point where the vehicle stops should be marked on the test roadway. The computed average deceleration is then:

$$\text{Average deceleration} = \tfrac{1}{2}\,(\text{initial speed in m/s})^2 \,/\, (\text{g x stopping distance in m})$$

$$\text{where } g = 9.81 \text{ m/s}^2$$

The wheel lock-up behaviour can be assessed by video recording the wheels on each side of the vehicle. Temporary marking of the exposed tyre walls will assist in assessing whether the wheels are locked or not.

Acceptable test conditions exists when the average deceleration is at least equal to the applicable assessment deceleration level in Table 20.

#### (d) Test method

Numerical modelling or field-testing.

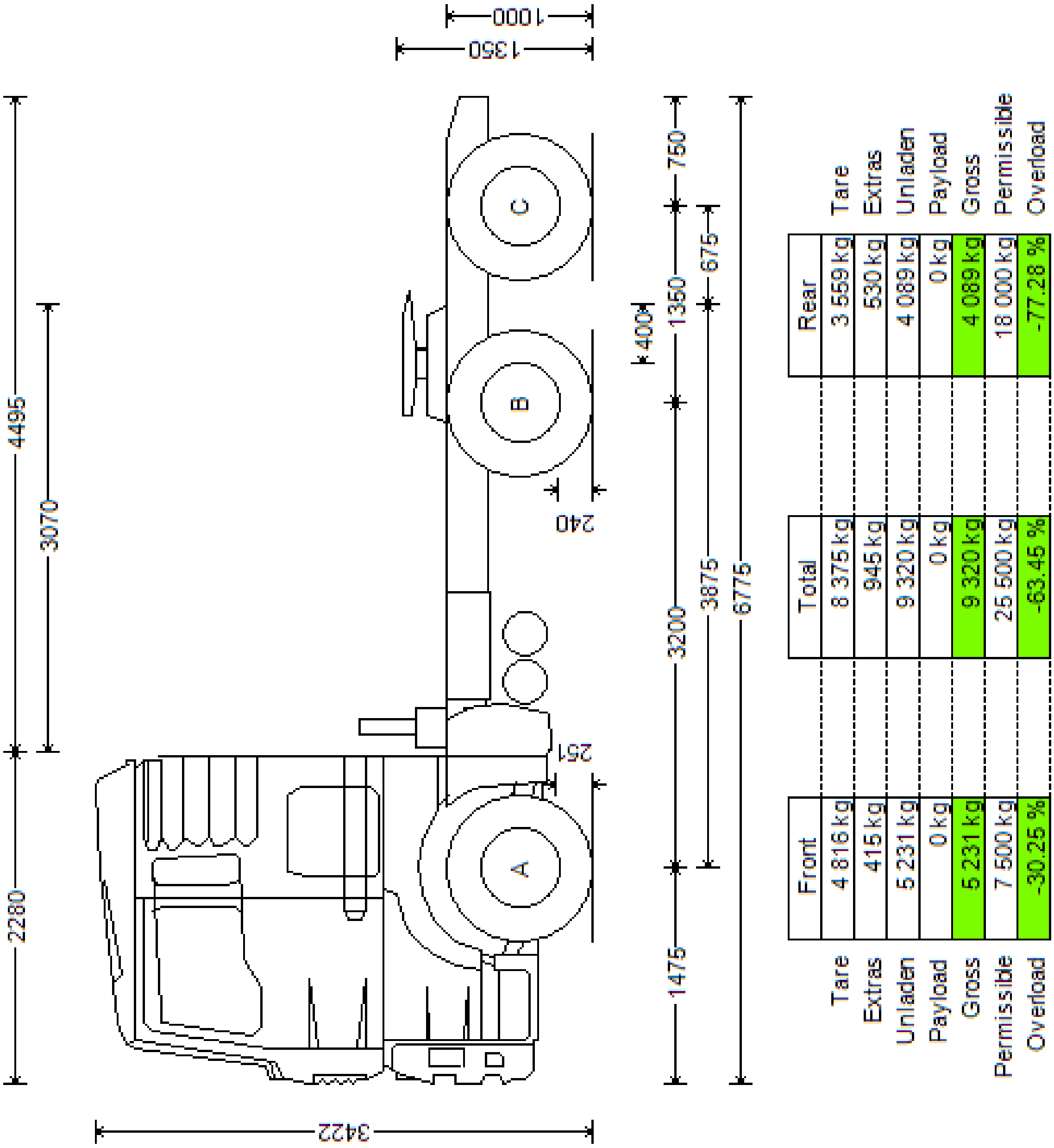
#### (e) Further notes

All parts of the (combination) vehicle must comply with the applicable Australian Design Rule for braking at the time of manufacture.

If a load-proportion brake system is fitted it must comply with the unladen compatibility requirements in Australian Design Rule 35/02 (motor vehicles) or Australian Design Rule 38/03 (trailers).

# Appendix B – Vehicle Fleet Selection

## B.1 Prime movers




| | Front | Total | Rear |
|---|---|---|---|
| Tare | 4 816 kg | 8 375 kg | 3 559 kg |
| Extras | 415 kg | 945 kg | 530 kg |
| Unladen | 5 231 kg | 9 320 kg | 4 089 kg |
| Payload | 0 kg | 0 kg | 0 kg |
| Gross | 5 231 kg | 9 320 kg | 4 089 kg |
| Permissible | 7 500 kg | 25 500 kg | 18 000 kg |
| Overload | -30.25 % | -63.45 % | -77.28 % |

Figure B.1: Layout dimensional drawing of MAN Prime mover

### B.1.1 Key dimensions

**Table B.1: Key dimension for the three prime movers (MAN, OECD 1 and OECD 2)**

| Dimensions | MAN | OECD 1 | OECD 2 |
|---|---|---|---|
| Prime Mover | | | |
| Length | 6.775 | - | - |
| Width | 2.5 | 2.5 | 2.5 |
| Height | 3.422 | - | - |
| Wheel Base | | | |
| Axle 1 - 2 | 3.2 | 3.21 | 3.3 |
| Axle 2 - 3 | 1.35 | 4.56 | 4.65 |
| Front Overhang | 1.475 | 1.36 | 1.44 |
| Rear Overhang | 0.75 | - | - |

### B.1.2 Mass properties

**Table B.2: Mass properties for the three prime movers (MAN, OECD 1 and OECD 2)**

| Mass Properties | MAN | OECD 1 | OECD 2 |
|---|---|---|---|
| Prime mover | 7523 | 7523 | 7523 |
| Roll Inertia, Ixx | 6879 | 6879 | 6879 |
| Pitch Inertia, Iyy | 21711 | 21711 | 21711 |
| Yaw Inertia, Izz | 19665 | 19665 | 19665 |
| Product of Inertia, Ixy | 0 | 0 | 0 |
| Product of Inertia, Ixz | 130 | 130 | 130 |
| Product of Inertia, Iyz | 0 | 0 | 0 |
| CG height above ground | 1.019 | 1.019 | 1.019 |

### B.1.3 Couplings (Fifth wheel)

**Table B.3: Mechanical Properties of couplings for the tree Prime movers (MAN, OECD 1 and OECD 2)**

| Design Parameter | MAN | OECD 1 | OECD 2 |
|---|---|---|---|
| Roll stiffness | 56,000 | 56,000 | 56,000 |
| Pitch stiffness | 0 | 0 | 0 |
| Yaw stiffness | 0 | 0 | 0 |
| Longitudinal compliance | 0 | 0 | 0 |
| Lateral compliance | 0 | 0 | 0 |
| Vertical compliance | 0 | 0 | 0 |

### B.1.4 Axles

#### *B.1.4.1 MAN*

**Table B.4: Axles parameter for MAN prime mover**

| Mass parameter | Steer | Drive |
|---|---|---|
| Number of axles | 1 | 2 |
| Mass per axle | 527 | 735 |
| Roll and Yaw inertia | 600 | 600 |
| Axle track width | 2030 | 1863 |
| CG height above ground | 0.51 | 0.51 |

#### *B.1.4.2 OECD 1 and OECD 2*

**Table B.5: Axle parameters for OECD 1 and OECD 2 prime movers**

| Mass parameter | Steer | Drive |
|---|---|---|
| Number of axles | 1 | 2 |
| Mass per axle | 600 | 1000 |
| Roll and Yaw inertia | 600 | 600 |
| Axle track width | 2030 | 1870 |
| CG height above ground | 0.51 | 0.51 |

### B.1.5 Tyres

Table B.6: Tyre parameters for the three prime movers (MAN, OECD 1 and OECD 2)

| Dimension | Steer | Drive |
|---|---|---|
| Number of tyres per axle | 2 | 8 |
| Tyre width | 0.315 | 0.315 |
| Dual tyre spacing | | 0.31 |
| Tyre track width | 2.048 | 1.804 |
| Rolling resistance | See below | |
| Effective rolling radius | 0.51 | 0.51 |
| Spring rate (N/mm) | 980 | 980 |
| Peak friction value | 0.8 | 0.8 |

***Rolling resistance:***

Fx_rr = Fz * Rr_surf * (Rr_c + Rr_v * Vx)
Rolling Resistance surface (Rr_surf) = 1.5
Rolling radius co-efficeint (Rr_c) = 0.0041
Rolling resistance speed co-efficient (Rr_v) = 0.0000256

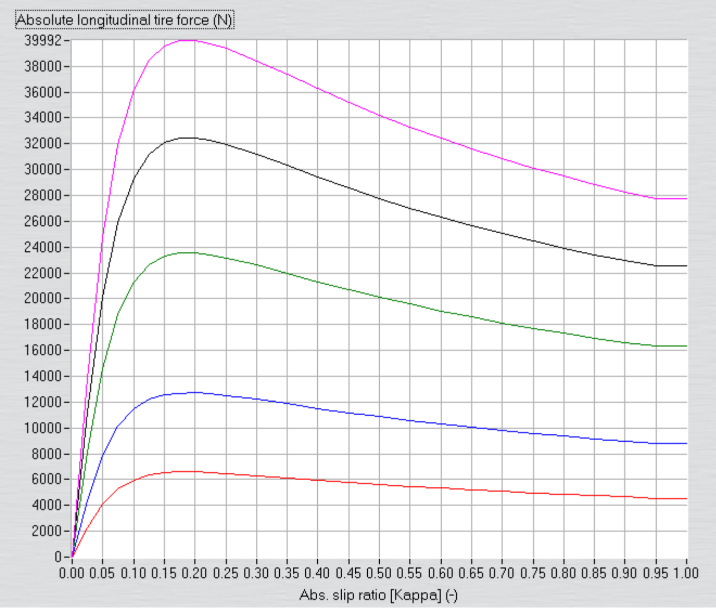



Figure B.2: Vertical tyre force, Fx, due to absolute slip ratio

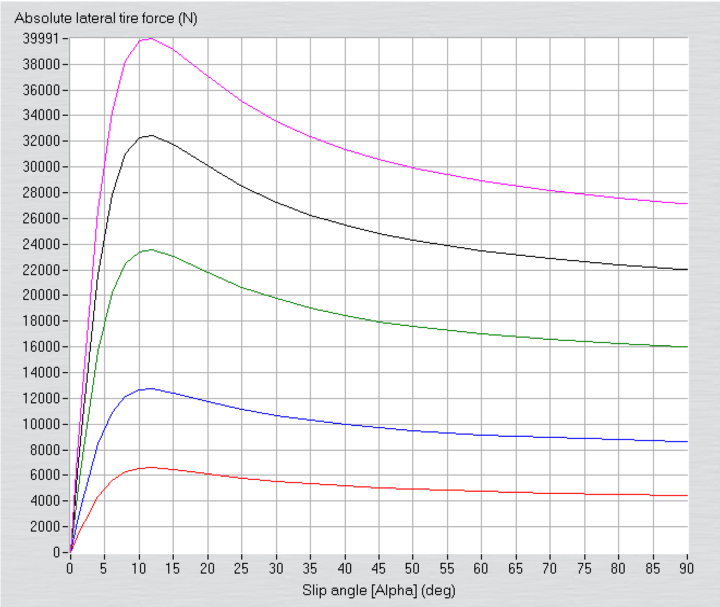



**Figure B.3: Lateral tyre force, Fy, due to slip angle**

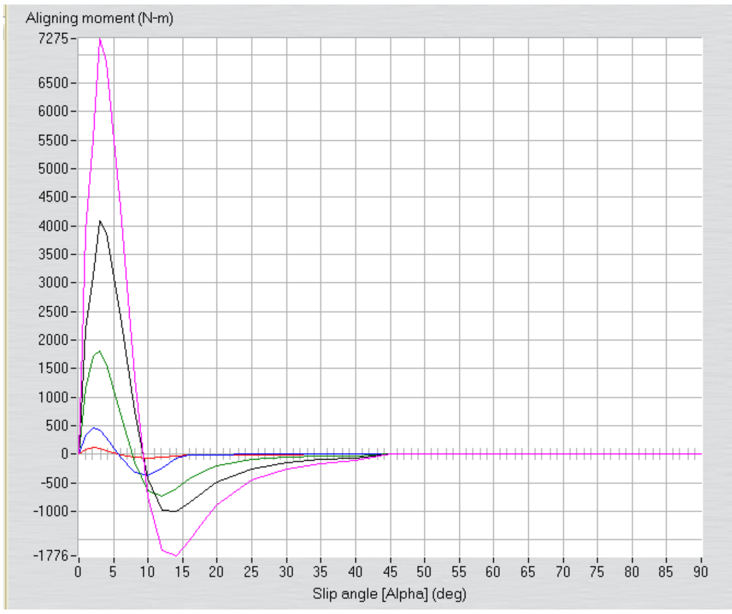



**Figure B.4: Aligning moment, Mz, due to slip angle**

### B.1.6 Suspension

**Table B.7: Suspension parameters for prime mover**

| Design parameter | Steer | Drive |
|---|---|---|
| Spring force data set | See below | See below |
| Roll centre height | 53 mm below axle | 195 mm above axle |
| Shock absorbers (kN-s/m) | 15 | 50 |
| Jounce / Rebound stops | 150 / -60 | 60 / -60 |
| Auxiliary roll / suspension roll (N-m/deg) | 500 | - |

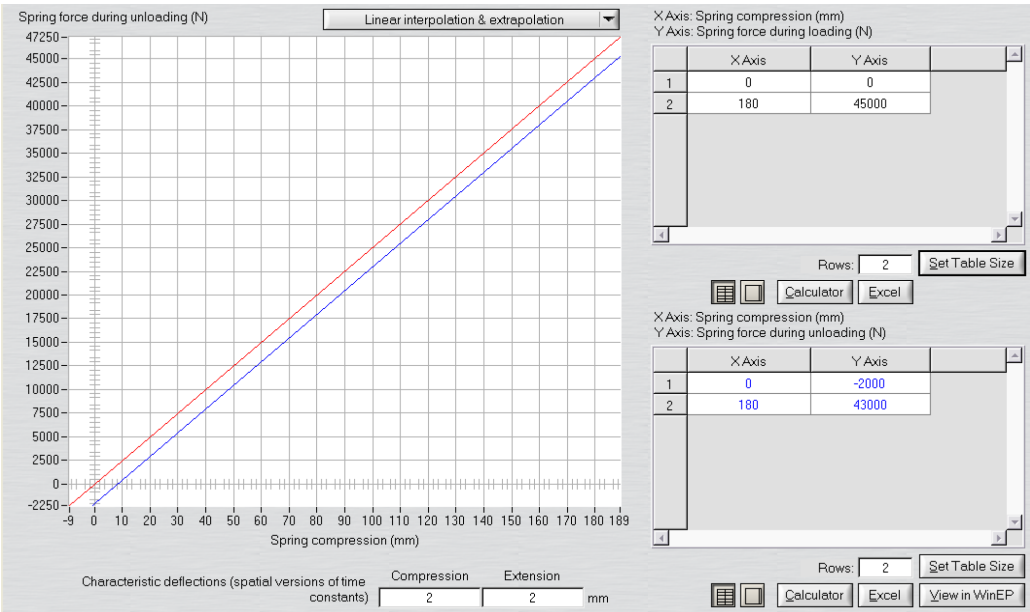



**Figure B.5: Spring force data set – Steer axle**

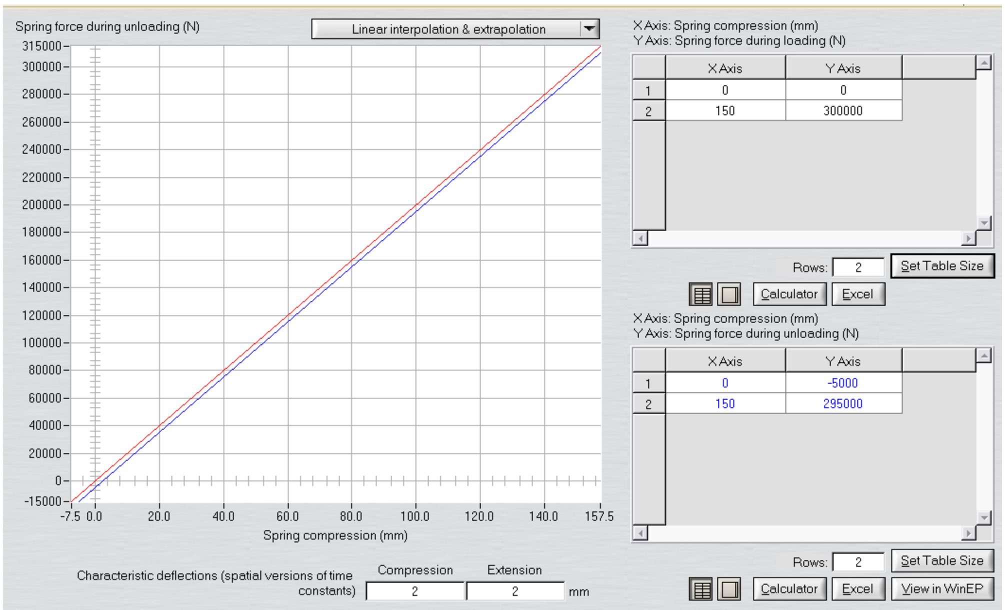



Figure B.6: Spring force data set – Drive axle

### B.1.7 Steering

Table B.8: Steering wheel alignment for the three prime movers (MAN, OECD 1 and OECD 2)

| Design parameter | Steer |
|---|---|
| Camber | 0 |
| Caster | 0 |
| Toe-in | 0 |

## B.2 Semi Trailers

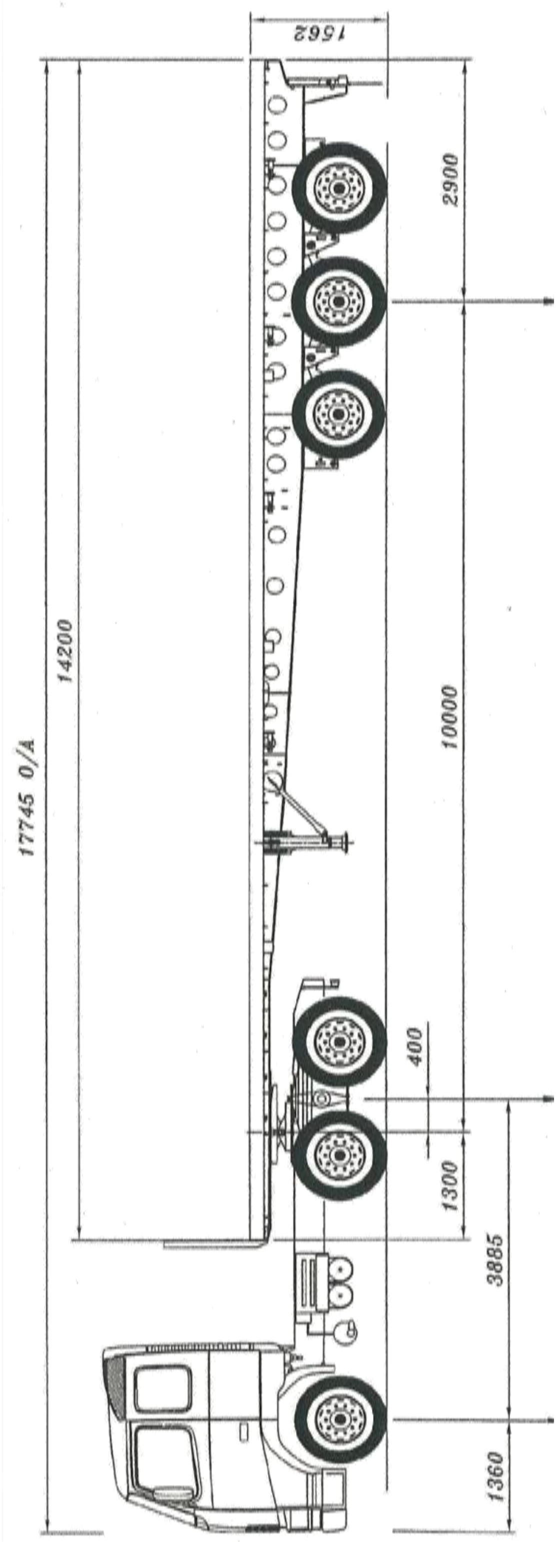
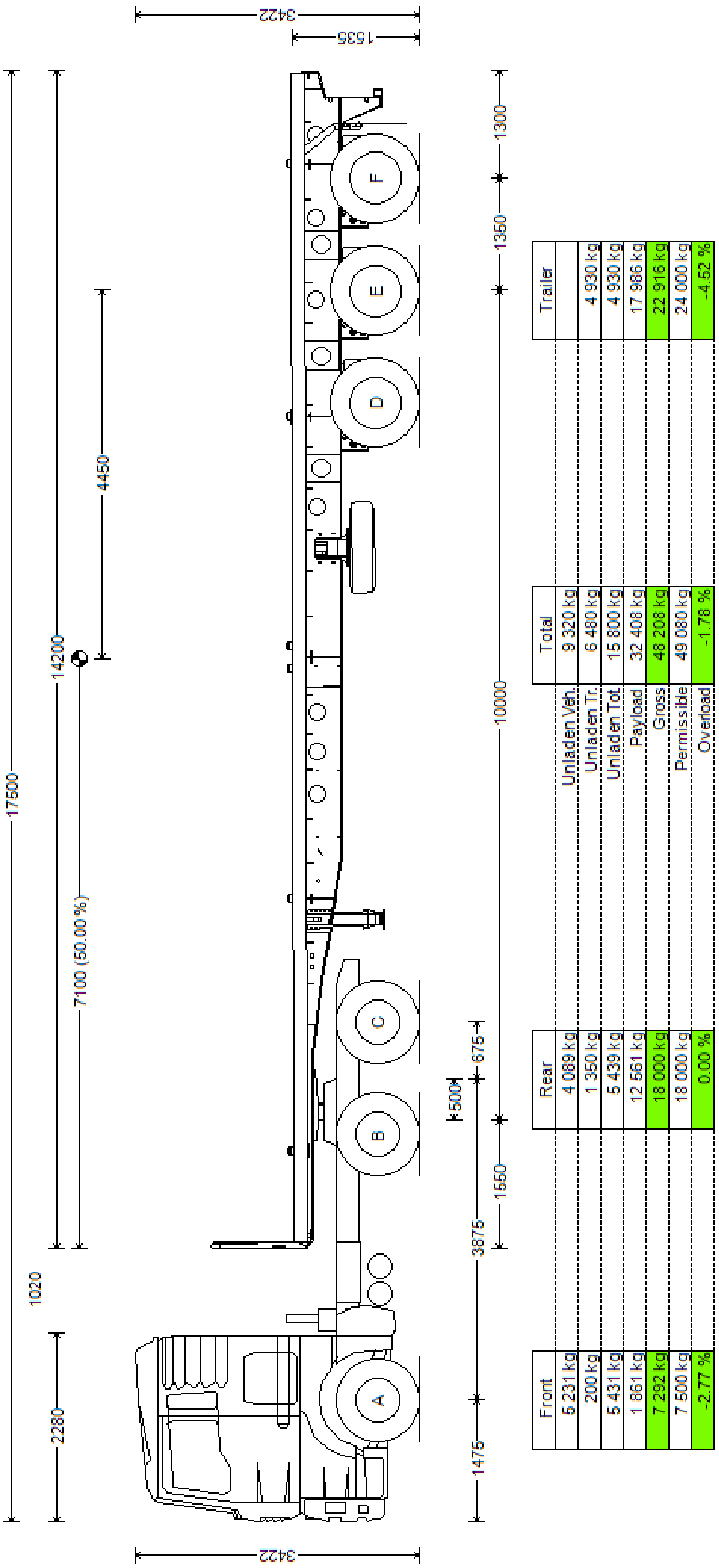
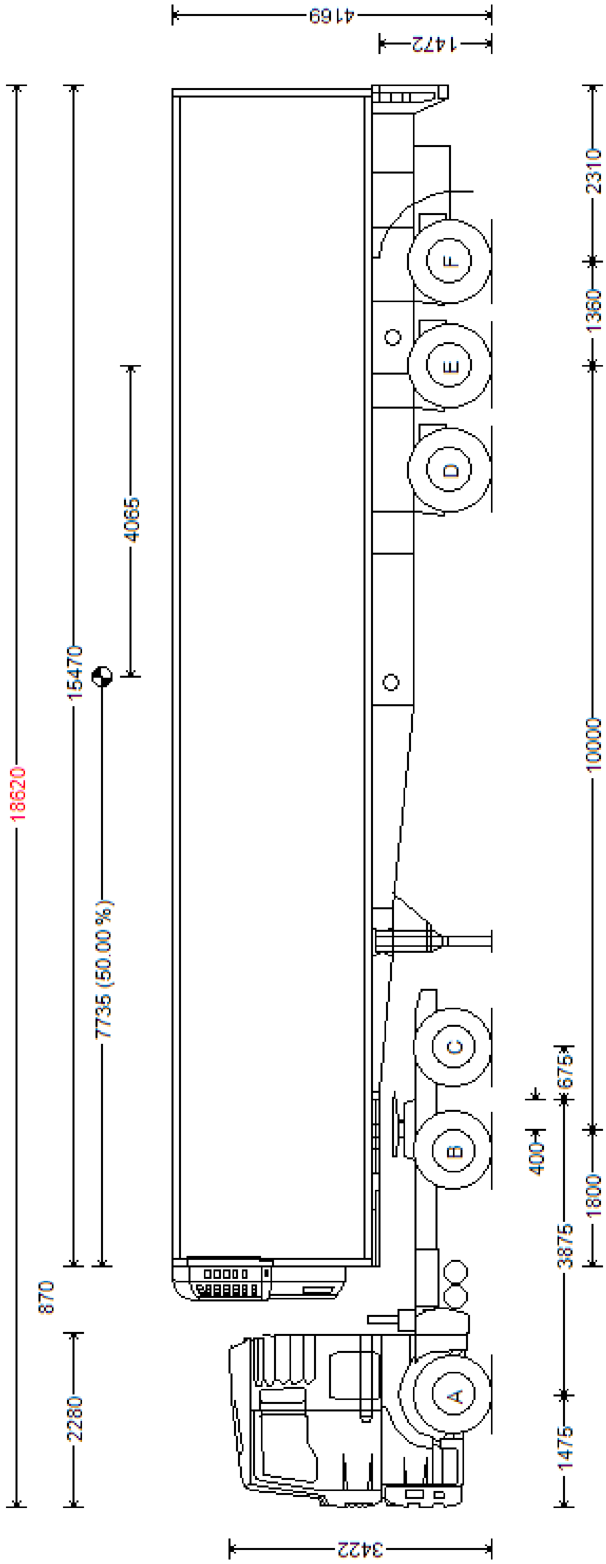



Figure B.7: Layout dimensional drawing of OECD 1 semi-trailer




| | Front | Rear | Total | Trailer |
|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | 9 320 kg | |
| Unladen Tr. | 200 kg | 1 350 kg | 6 480 kg | 4 930 kg |
| Unladen Tot | 5 431 kg | 5 439 kg | 15 800 kg | 4 930 kg |
| Payload | 1 861 kg | 12 561 kg | 32 408 kg | 17 986 kg |
| Gross | 7 292 kg | 18 000 kg | 48 208 kg | 22 916 kg |
| Permissible | 7 500 kg | 18 000 kg | 49 080 kg | 24 000 kg |
| Overload | -2.77 % | 0.00 % | -1.78 % | -4.52 % |

**Figure B.8: Layout dimensional drawing of Skeletal semi-trailer**




| | Front | Rear | Total | Trailer |
|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | 9 320 kg | |
| Unladen Tr. | 345 kg | 2 994 kg | 9 800 kg | 6 461 kg |
| Unladen Tot | 5 576 kg | 7 083 kg | 19 120 kg | 6 461 kg |
| Payload | 1 240 kg | 10 772 kg | 29 551 kg | 17 539 kg |
| Gross | 6 816 kg | 17 855 kg | 48 671 kg | 24 000 kg |
| Permissible | 7 500 kg | 18 000 kg | 49 290 kg | 24 000 kg |
| Overload | -9.12 % | -0.81 % | -1.26 % | 0.00 % |

**Figure B.9: Layout dimensional drawing of Refrigeration semi-trailer**

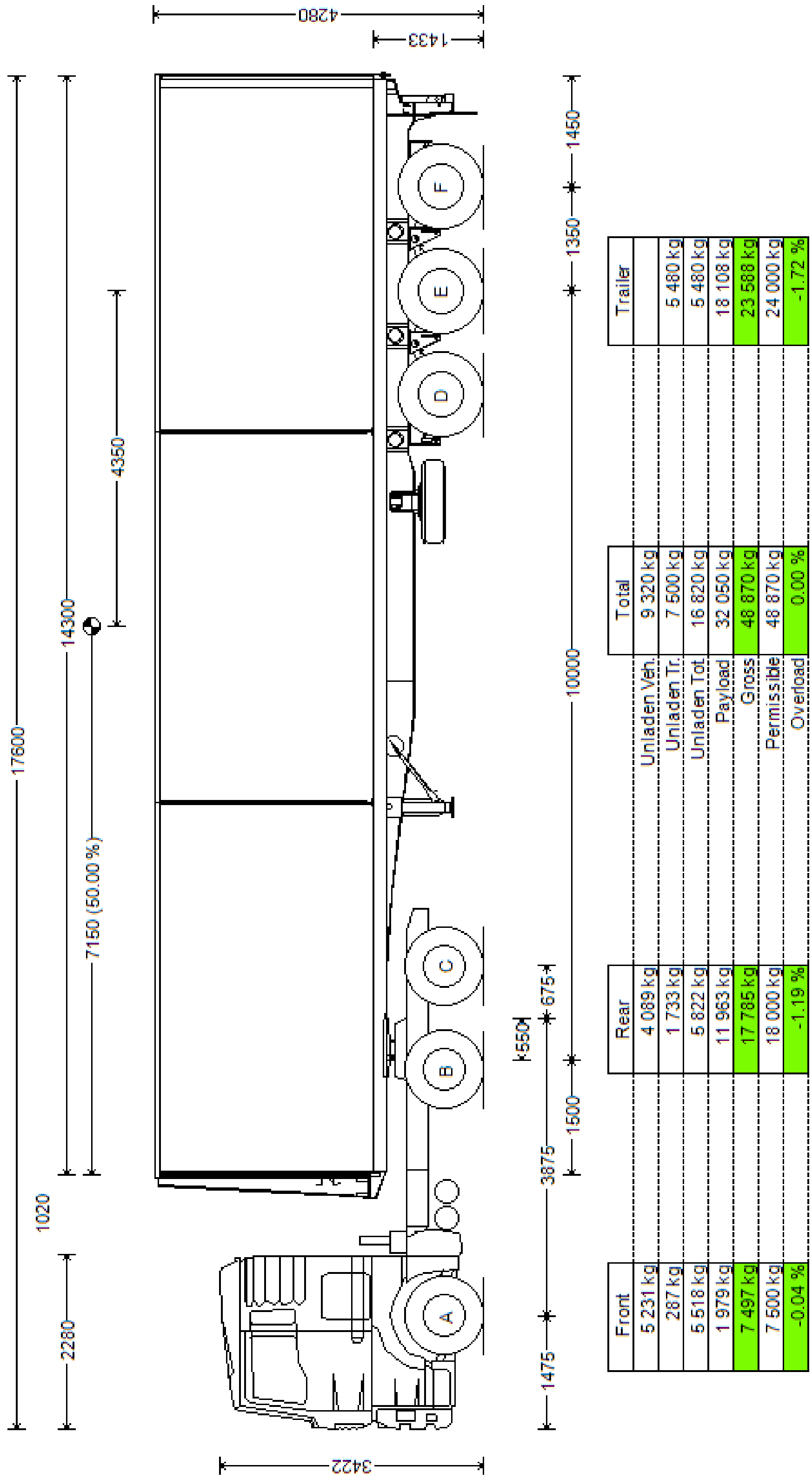
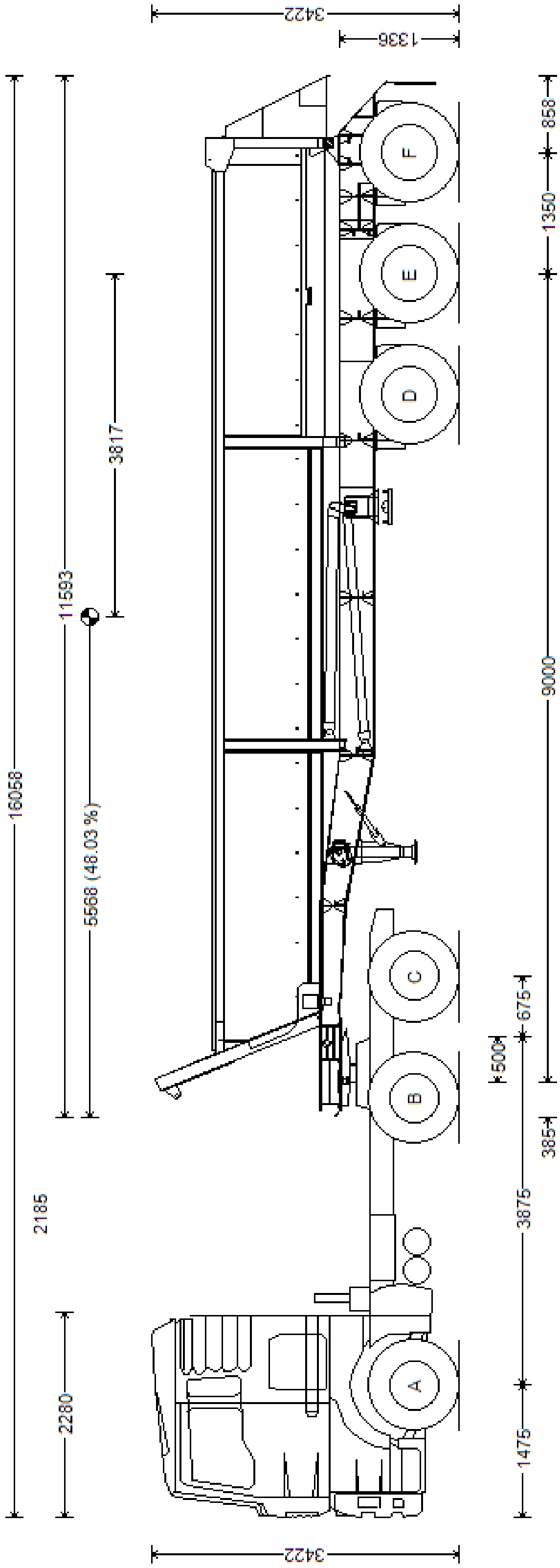



Figure B.10: Layout dimensional drawing of Side curtain semi-trailer




| | Front | Rear | Total | Trailer |
|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | 9 320 kg | |
| Unladen Tr. | 382 kg | 2 578 kg | 8 740 kg | 5 780 kg |
| Unladen Tot | 5 613 kg | 6 667 kg | 18 060 kg | 5 780 kg |
| Payload | 1 583 kg | 10 682 kg | 28 920 kg | 16 655 kg |
| Gross | 7 196 kg | 17 349 kg | 46 980 kg | 22 435 kg |
| Permissible | 7 500 kg | 18 000 kg | 46 980 kg | 24 000 kg |
| Overload | -4.05 % | -3.62 % | 0.00 % | -6.52 % |

Figure B.11: Layout dimensional drawing of Tipper semi-trailer

### B.2.1 Key dimensions

**Table B.9: Key dimensions of the five semi-trailers**

| Dimensions | OECD 1 | Skeletal | Refrigeration | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Overall dimensions | | | | | |
| Length | 17.745 | 17.5 | 18.62 | 17.6 | 16.058 |
| Width | 2.6 | 2.6 | 2.6 | 2.6 | 2.6 |
| Height | 4.22 | 3.422 | 4.169 | 4.28 | 3.422 |
| Trailer Dimensions | | | | | |
| Length | 14.2 | 14.2 | 15.47 | 14.3 | 11.593 |
| Wheelbase | 10.0 | 10.0 | 10.0 | 10.0 | 9.0 |
| Axle spacing | 1.36 | 1.35 | 1.36 | 1.35 | 1.35 |
| Front overhang | 1.3 | 1.55 | 1.8 | 1.5 | 0.382 |
| Rear overhang | 1.54 | 1.3 | 2.31 | 1.45 | 0.858 |
| Height of cargo floor above ground | 1.562 | 1.535 | 1.472 | 1.433 | 1.336 |

### B.2.2 Mass properties

**Table B.10: Mass properties of the five semi-trailers**

| Mass Properties | OECD 1 | Skeletal | Refrigeration | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Unladen | - | 15 800 | 19 120 | 16 820 | 18 060 |
| Payload | 30000 | 32 408 | 29 551 | 32 050 | 28 920 |
| GCM | - | 48 208 | 48 671 | 48 870 | 46 980 |
| Roll Inertia, Ixx | 28203.94 | 24411.03 | 28110.98 | 31909.68 | 23003.19 |
| Pitch Inertia, Iyy | 515403.9 | 550717 | 600811.6 | 560013.6 | 330609.9 |
| Yaw Inertia, Izz | 521000 | 562818.9 | 605994.7 | 564213.6 | 340189.9 |
| CG height above ground | 2.6252 | 2.2898 | 2.5508 | 2.5718 | 2.1704 |

### B.2.3 Axles

**Table B.11: Axle parameters of the five semi-trailers**

| Mass parameter | OECD 1 | Skeletal | Refrigeration | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Number of axles | 3 | 3 | 3 | 3 | 3 |
| Mass per axle | 800 | 800 | 800 | 800 | 800 |
| Roll and Yaw inertia | 600 | 600 | 600 | 600 | 600 |
| | 0.51 | | | | |
| CG height above ground | 1975 | 0.51 | 0.51 | 0.51 | 0.51 |
| Axle track width | | 1910 | 1910 | 1910 | 1910 |
| CG position aft of king pin | 5.8 | 5.55 | 5.935 | 5.65 | 5.183 |

### B.2.4 Tyres

**Table B.12: Tyre properties of the five semi-trailers**

| Dimension | OECD 1 | Skeletal | Refrigeration | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Number of tyres per axle | 4 | 4 | 4 | 4 | 4 |
| Tyre width | 0.315 | 0.315 | 0.315 | 0.315 | 0.315 |
| Dual tyre spacing | 0.31 | 0.31 | 0.31 | 0.31 | 0.31 |
| Tyre track width | 1.975 | 1.91 | 1.91 | 1.91 | 1.91 |
| Rolling resistance | | | See below | | |
| Effective rolling radius | 0.51 | 0.51 | 0.51 | 0.51 | 0.51 |
| Spring rate (N/mm) | 980 | 980 | 980 | 980 | 980 |
| Peak friction value | 0.8 | 0.8 | 0.8 | 0.8 | 0.8 |

### Rolling resistance:

Fx_rr = Fz * Rr_surf * (Rr_c + Rr_v * Vx)
Rolling Resistance surface (Rr_surf) = 1.5
Rolling radius co-efficeint (Rr_c) = 0.0041
Rolling resistance speed co-efficient (Rr_v) = 0.0000256

The tyre longitudinal, lateral and aligning moments for the semi-trailers are the same as those in sections B.1.5 above.

### B.2.5 Suspension

Table B.13: Suspension design Parameters for the five semi-trailers

| Design parameter | Trailer |
|---|---|
| Spring force data set | See below |
| Roll centre height | 195 mm above axle |
| Shock absorbers (kN-s/m) | 30 |
| Jounce / Rebound stops | 100 / -60 |
| Auxiliary roll / suspension roll (N-m/deg) | 3000 |

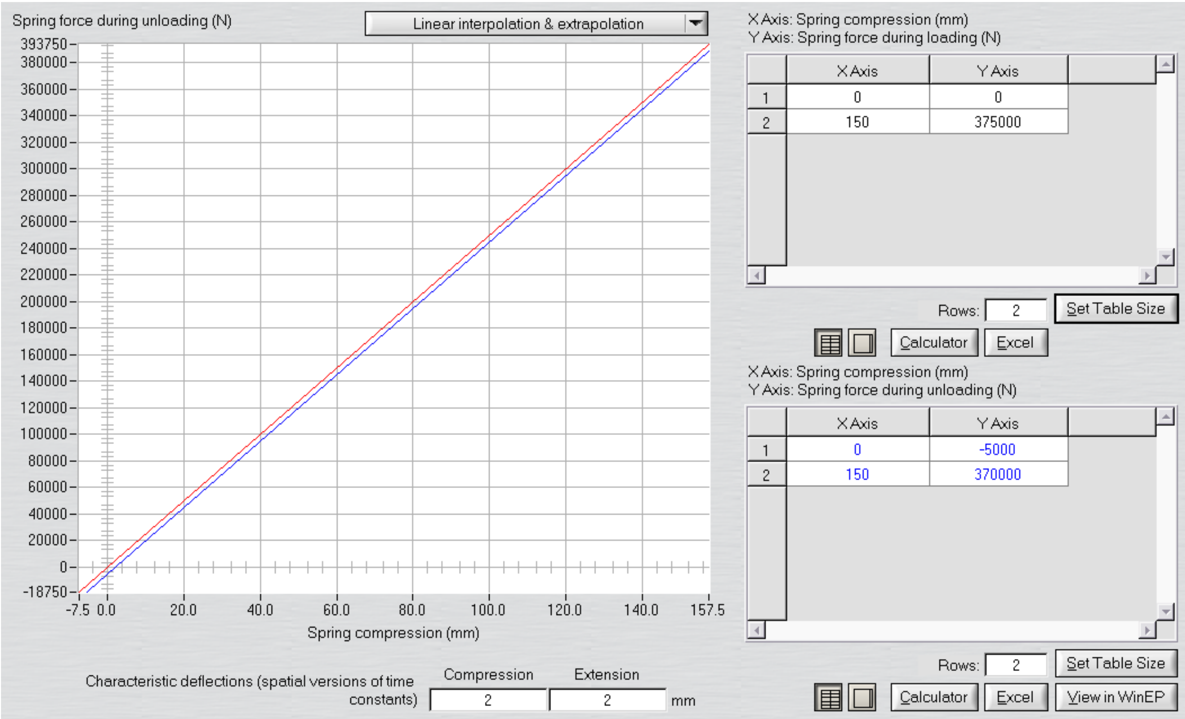



Figure B.12: Spring force dataset – Trailing axle semi-trailer

## B.3 B-Double

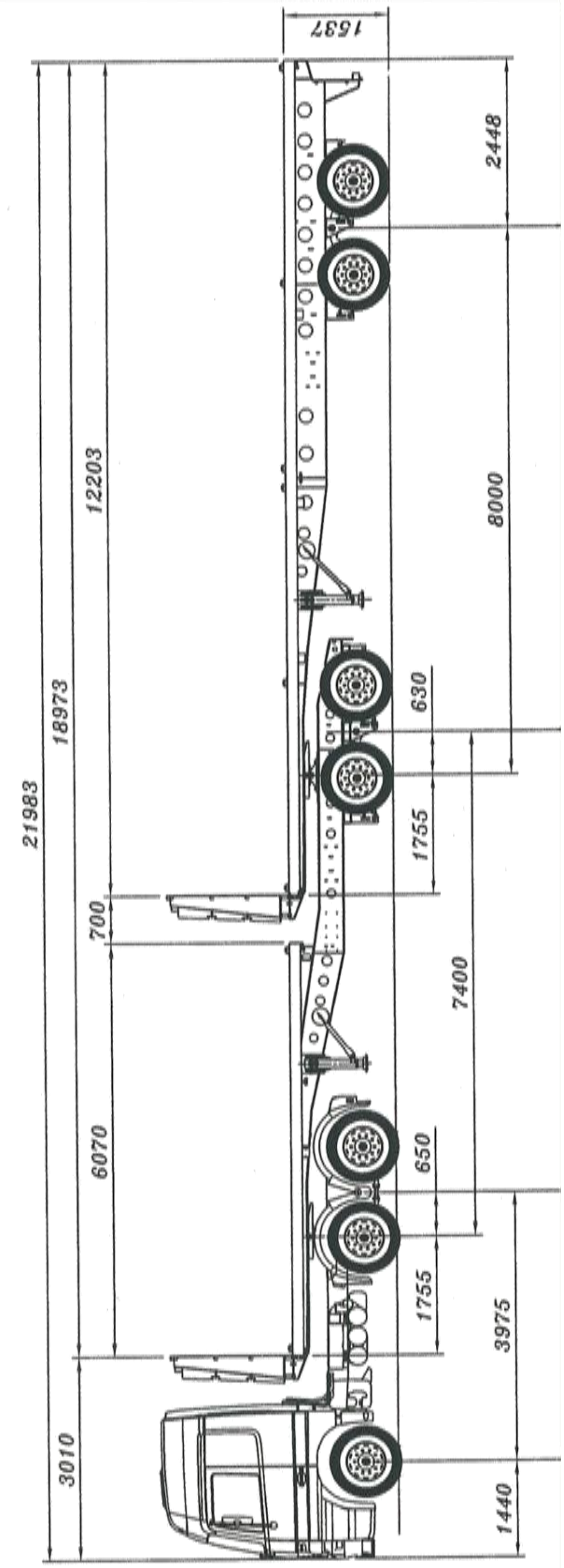
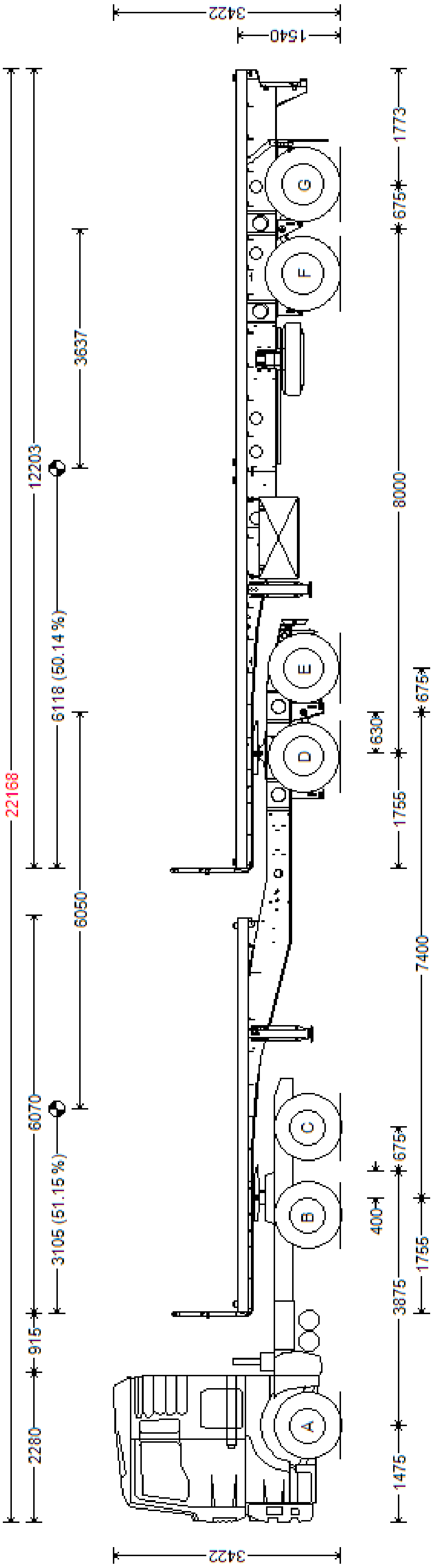
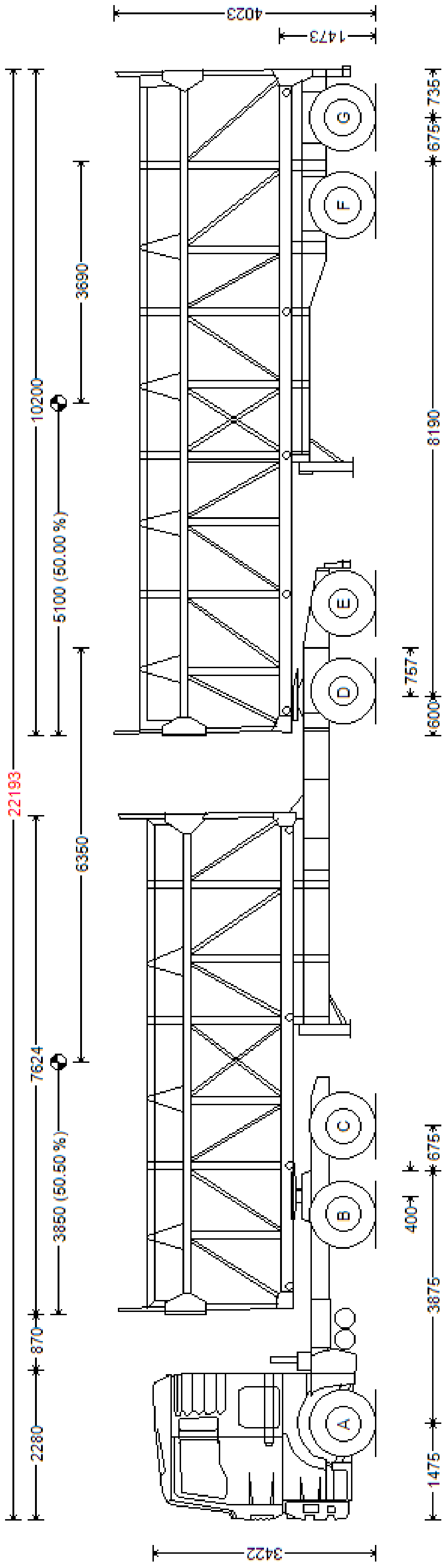



Figure B.13: Layout dimensional drawing of the OECD 2 B-double configuration



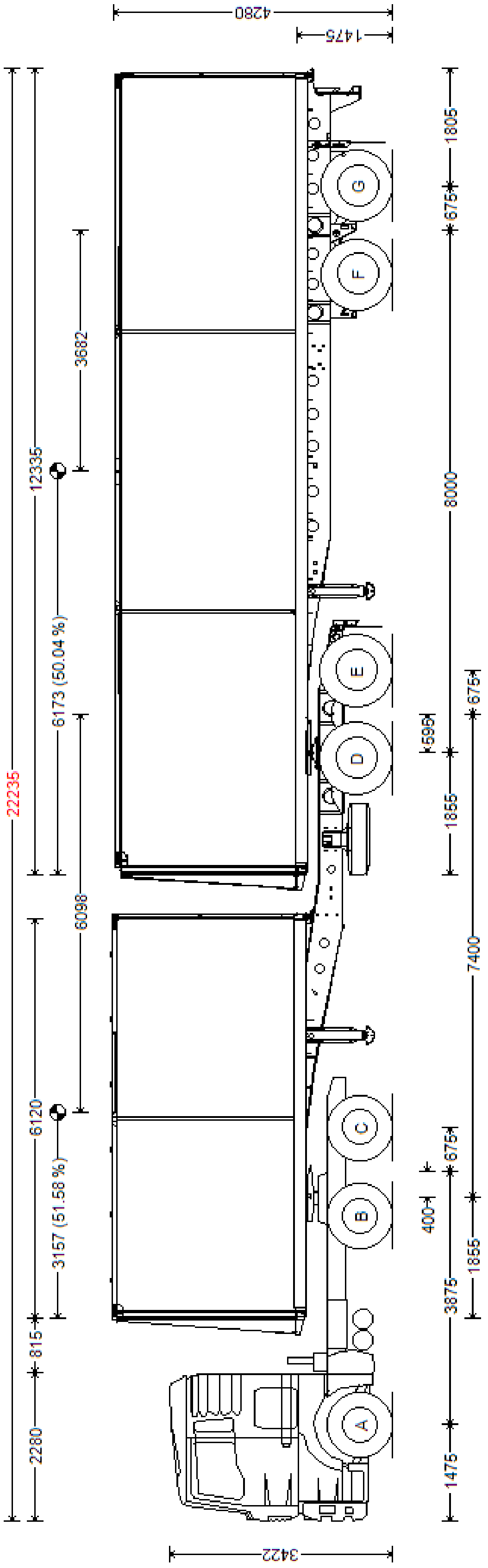

| | Veh. Front | Veh. Rear | Trailer 1 | Total | Trailer 2 |
|---|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | | 9 320 kg | |
| Unladen Tr. 1 | 134 kg | 1 166 kg | 2 920 kg | 4 220 kg | |
| Unladen Tr. 2 | 11 kg | 100 kg | 1 189 kg | 4 800 kg | 3 500 kg |
| Unladen Tot. | 5 376 kg | 5 355 kg | 4 109 kg | 18 340 kg | 3 500 kg |
| Payload Tr. 1 | 1 056 kg | 9 172 kg | 2 282 kg | 12 510 kg | |
| Payload Tr. 2 | 100 kg | 873 kg | 10 461 kg | 25 150 kg | 13 716 kg |
| Payload Tot. | 1 156 kg | 10 045 kg | 12 743 kg | 37 660 kg | 13 716 kg |
| Gross | 6 532 kg | 15 400 kg | 16 852 kg | 56 000 kg | 17 216 kg |
| Permissible | 7 500 kg | 18 000 kg | 18 000 kg | 56 000 kg | 18 000 kg |
| Overload | -12.91 % | -14.44 % | -6.38 % | 0.00 % | -4.36 % |

**Figure B.14: Layout dimensional drawing of the Skeletal B-double configuration**



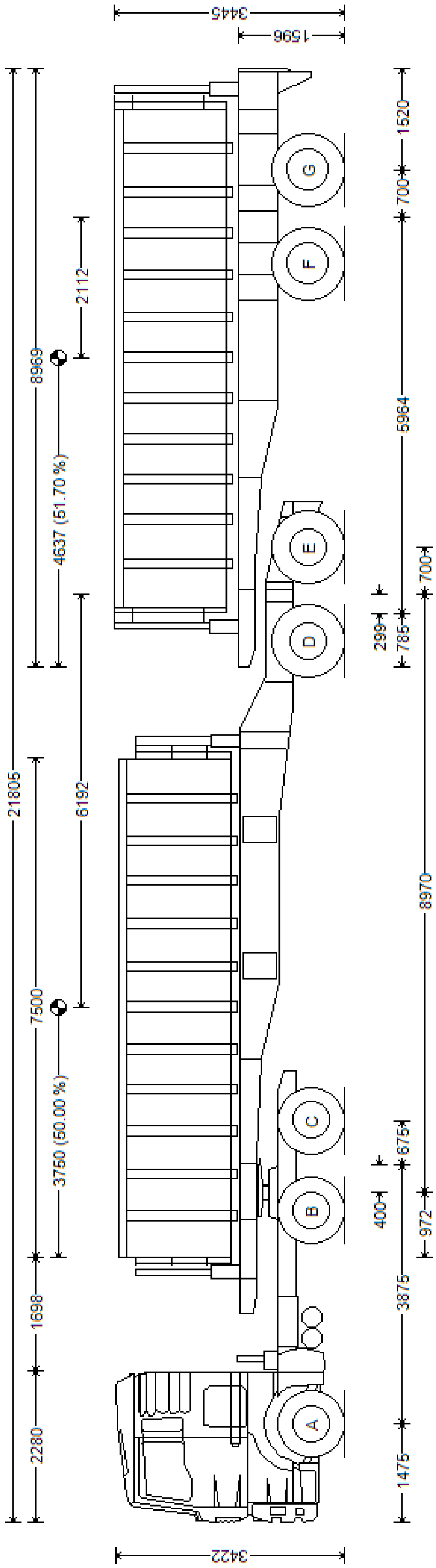

| | Veh. Front | Veh. Rear | Trailer 1 | Total | Trailer 2 |
|---|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | | 9 320 kg | |
| Unladen Tr.1 | 328 kg | 2 849 kg | 3 863 kg | 7 040 kg | |
| Unladen Tr.2 | 22 kg | 189 kg | 2 130 kg | 7 480 kg | 5 139 kg |
| Unladen Tot | 5 581 kg | 7 127 kg | 5 993 kg | 23 840 kg | 5 139 kg |
| Payload Tr.1 | 1 073 kg | 9 326 kg | 3 357 kg | 13 756 kg | |
| Payload Tr.2 | 77 kg | 670 kg | 7 545 kg | 18 404 kg | 10 112 kg |
| Payload Tot | 1 150 kg | 9 996 kg | 10 902 kg | 32 160 kg | 10 112 kg |
| Gross | 6 731 kg | 17 123 kg | 16 895 kg | 56 000 kg | 15 251 kg |
| Permissible | 7 500 kg | 18 000 kg | 18 000 kg | 56 000 kg | 18 000 kg |
| Overload | -10.25 % | -4.87 % | -6.14 % | 0.00 % | -15.27 % |

Figure B.95: Layout dimensional drawing of the Cane B-double configuration




| | Veh. Front | Veh. Rear | Trailer 1 | Total | Trailer 2 |
|---|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | | 9 320 kg | |
| Unladen Tr. 1 | 212 kg | 1 838 kg | 2 700 kg | 4 750 kg | |
| Unladen Tr. 2 | 14 kg | 119 kg | 1 517 kg | 5 400 kg | 3 750 kg |
| Unladen Tot | 5 457 kg | 6 046 kg | 4 217 kg | 19 470 kg | 3 750 kg |
| Payload Tr. 1 | 1 031 kg | 8 952 kg | 2 131 kg | 12 114 kg | |
| Payload Tr. 2 | 93 kg | 811 kg | 10 333 kg | 24 416 kg | 13 179 kg |
| Payload Tot | 1 124 kg | 9 763 kg | 12 464 kg | 36 530 kg | 13 179 kg |
| Gross | 6 581 kg | 15 809 kg | 16 681 kg | 56 000 kg | 16 929 kg |
| Permissible | 7 500 kg | 18 000 kg | 18 000 kg | 56 000 kg | 18 000 kg |
| Overload | -12.25 % | -12.17 % | -7.33 % | 0.00 % | -5.95 % |

**Figure B.16: Layout dimensional drawing of the Side curtain B-double configuration**




| | Veh. Front | Veh. Rear | Trailer 1 | Total | Trailer 2 |
|---|---|---|---|---|---|
| Unladen Veh. | 5 231 kg | 4 089 kg | | 9 320 kg | |
| Unladen Tr. 1 | 279 kg | 2 421 kg | 4 200 kg | 6 900 kg | |
| Unladen Tr. 2 | 7 kg | 60 kg | 1 933 kg | 7 046 kg | 5 046 kg |
| Unladen Tot | 5 517 kg | 6 570 kg | 6 133 kg | 23 266 kg | 5 046 kg |
| Payload Tr. 1 | 1 062 kg | 9 228 kg | 4 617 kg | 14 907 kg | |
| Payload Tr. 2 | 22 kg | 188 kg | 6 103 kg | 17 827 kg | 11 514 kg |
| Payload Tot | 1 084 kg | 9 416 kg | 10 720 kg | 32 734 kg | 11 514 kg |
| Gross | 6 601 kg | 15 986 kg | 16 853 kg | 56 000 kg | 16 560 kg |
| Permissible | 7 500 kg | 18 000 kg | 18 000 kg | 56 000 kg | 18 000 kg |
| Overload | -11.99 % | -11.19 % | -6.37 % | 0.00 % | -8.00 % |

**Figure B.17: Layout dimensional drawing of the Tipper B-double configuration**

### B.3.1 Key dimension

Table B.14: Key dimensions for five B-double configurations

| Dimensions | OECD 2 | Skeletal | Cane | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Overall dimensions | | | | | |
| Length | 21.983 | 22.168 | 22.193 | 22.235 | 21.805 |
| Width | 2.6 | 2.6 | 2.6 | 2.6 | 2.6 |
| Height | 4.22 | 3.422 | 4.023 | 4.28 | 3.445 |
| 1st Trailer Dimensions | | | | | |
| Length | 6.07 | 6.07 | 7.624 | 6.12 | 7.5 |
| Wheelbase | 7.4 | 7.4 | 8.4 | 7.4 | 8.97 |
| Front overhang | 1.755 | 1.755 | 1.8 | 1.855 | 0.972 |
| Rear overhang | - | 0.74 | 0.653 | 0.621 | 0.695 |
| 2nd Trailer Dimensions | | | | | |
| Length | 12.203 | 12.203 | 10.2 | 12.335 | 8.969 |
| Wheelbase | 8.0 | 8.0 | 8.19 | 8.0 | 5.964 |
| Front overhang | 1.755 | 1.755 | 0.6 | 1.855 | 0.785 |
| Rear overhang | 1.188 | 1.773 | 0.735 | 1.805 | 1.52 |
| Axle spacing | 1.26 | 1.35 | 1.35 | 1.35 | 1.4 |
| Height of cargo floor above ground | 1.537 | 1.54 | 1.473 | 1.475 | 1.596 |

### B.3.2 Mass properties

Table B.15: Mass properties for Five B-double configurations

| Mass Properties | OECD 2 | Skeletal | Cane | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Unladen | 24000 | 18 340 | 23 840 | 19 470 | 23 266 |
| Payload | 30000 | 37 660 | 32 160 | 36 530 | 32 734 |
| GCM | 56000 | 56 000 | 56 000 | 56 000 | 56 000 |
| Roll Inertia, Ixx | 9959.7 | 9959.7 | 9959.7 | 9959.7 | 9959.7 |
| Pitch Inertia, Iyy | 171336 | 171336 | 171336 | 171336 | 171336 |
| Yaw Inertia, Izz | 179992 | 179992 | 179992 | 179992 | 179992 |
| CG Height above ground | 2642.2 | 2292.8 | 2493 | 2597 | 2335.6 |

### B.3.3 Couplings (Fifth wheel)

**Table B.16: Coupling parameter for five B-double configurations**

| Design Parameter | MAN | OECD 1 | OECD 2 |
|---|---|---|---|
| Roll stiffness | 56,000 | 56,000 | 56,000 |
| Pitch stiffness | 0 | 0 | 0 |
| Yaw stiffness | 0 | 0 | 0 |
| Longitudinal compliance | 0 | 0 | 0 |
| Lateral compliance | 0 | 0 | 0 |
| Vertical compliance | 0 | 0 | 0 |

### B.3.4 Tyres

**Table B.17: Tyre parameter for five B-double configurations**

| Dimension | OECD 2 | Skeletal | Cane | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Number of tyres per axle | 4 | 4 | 4 | 4 | 4 |
| Tyre width | 0.315 | 0.315 | 0.315 | 0.315 | 0.315 |
| Dual tyre spacing | 0.31 | 0.31 | 0.31 | 0.31 | 0.31 |
| Tyre track width | 1.975 | 1.91 | 1.91 | 1.91 | 1.91 |
| Rolling resistance | | | See below | | |
| Effective rolling radius | 0.51 | 0.51 | 0.51 | 0.51 | 0.51 |
| Spring rate (N/mm) | 980 | 980 | 980 | 980 | 980 |
| Peak friction value | 0.8 | 0.8 | 0.8 | 0.8 | 0.8 |

**Rolling resistance:**

Fx_rr = Fz * Rr_surf * (Rr_c + Rr_v * Vx)
Rolling Resistance surface (Rr_surf) = 1.5
Rolling radius co-efficeint (Rr_c) = 0.0041
Rolling resistance speed co-efficient (Rr_v) = 0.0000256

The tyre longitudinal, lateral and aligning moments for the B-double are the same as those in sections B.1.5 above.

### B.3.5 Axles

Table B.18: Axle parameters for five B-double configurations

| Mass parameter | OECD 2 | Skeletal | Cane | Side Curtain | Tipper |
|---|---|---|---|---|---|
| Number of axles | 4 | 4 | 4 | 4 | 4 |
| Mass per axle | 800 | 735 | 735 | 735 | 735 |
| Roll and Yaw inertia | 600 | 600 | 600 | 600 | 600 |
| CG height above ground | 0.51 | 0.51 | 0.51 | 0.51 | 0.51 |
| Axle track width | 1975 | 1910 | 1910 | 1910 | 1910 |
| CG position aft 1st king pin | 1.28 | 1.35 | 2.05 | 1.302 | 2.778 |
| CG position aft 2nd king pin | 4.3465 | 4.363 | 4.5 | 4.318 | 3.852 |

### B.3.6 Suspension

Table B.19: Suspension parameters for five B-double configurations

| Design parameter | Trailer |
|---|---|
| Spring force data set | See below |
| Roll centre height | 195 mm above axle |
| Shock absorbers (kN-s/m) | 30 |
| Jounce / Rebound stops | 100 / -60 |
| Auxiliary roll / suspension roll (N-m/deg) | 3000 |

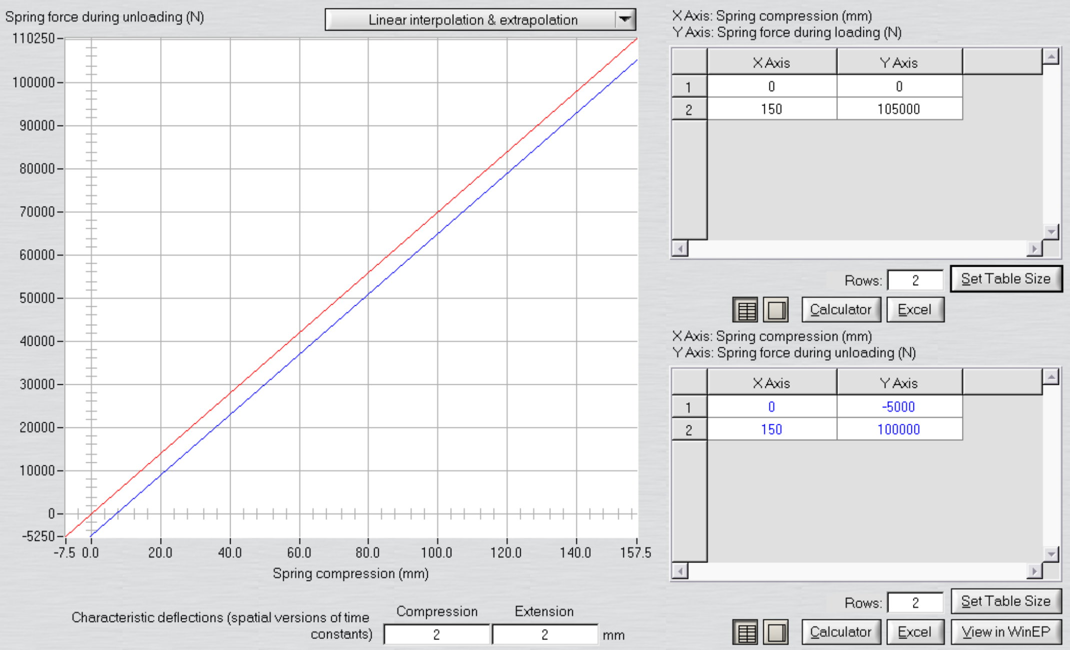



Figure B.18: Spring force dataset – Trailing axle B-double

# Appendix C – Results

## C.1 Startability, Gradeability and Acceleration Capability

### C.1.1 Gearbox and Engine characteristics

Table C.1: Transmission gear ratios, efficiencies and change time for a 12 gear ZF 12 AS 2301 OD Tipmatic Gearbox

| Transmission |
|---|

| | |
|---|---|
| Model | ZF 12 AS 2301 OD TipMatic |
| Type | Constant mesh |
| Shift | Tipmatic |

| Gear | Ratio | Efficiency | Change Time |
|---|---|---|---|
| 1 | 12.33 | 0.93 | 1.2 |
| 2 | 9.59 | 0.93 | 1.2 |
| 3 | 7.44 | 0.93 | 1.2 |
| 4 | 5.78 | 0.93 | 1.2 |
| 5 | 4.57 | 0.95 | 1.2 |
| 6 | 3.55 | 0.95 | 1.2 |
| 7 | 2.7 | 0.95 | 0.8 |
| 8 | 2.1 | 0.95 | 0.8 |
| 9 | 1.63 | 0.95 | 0.8 |
| 10 | 1.27 | 0.95 | 0.8 |
| 11 | 1 | 0.97 | 0.8 |
| 12 | 0.78 | 0.95 | 0.8 |

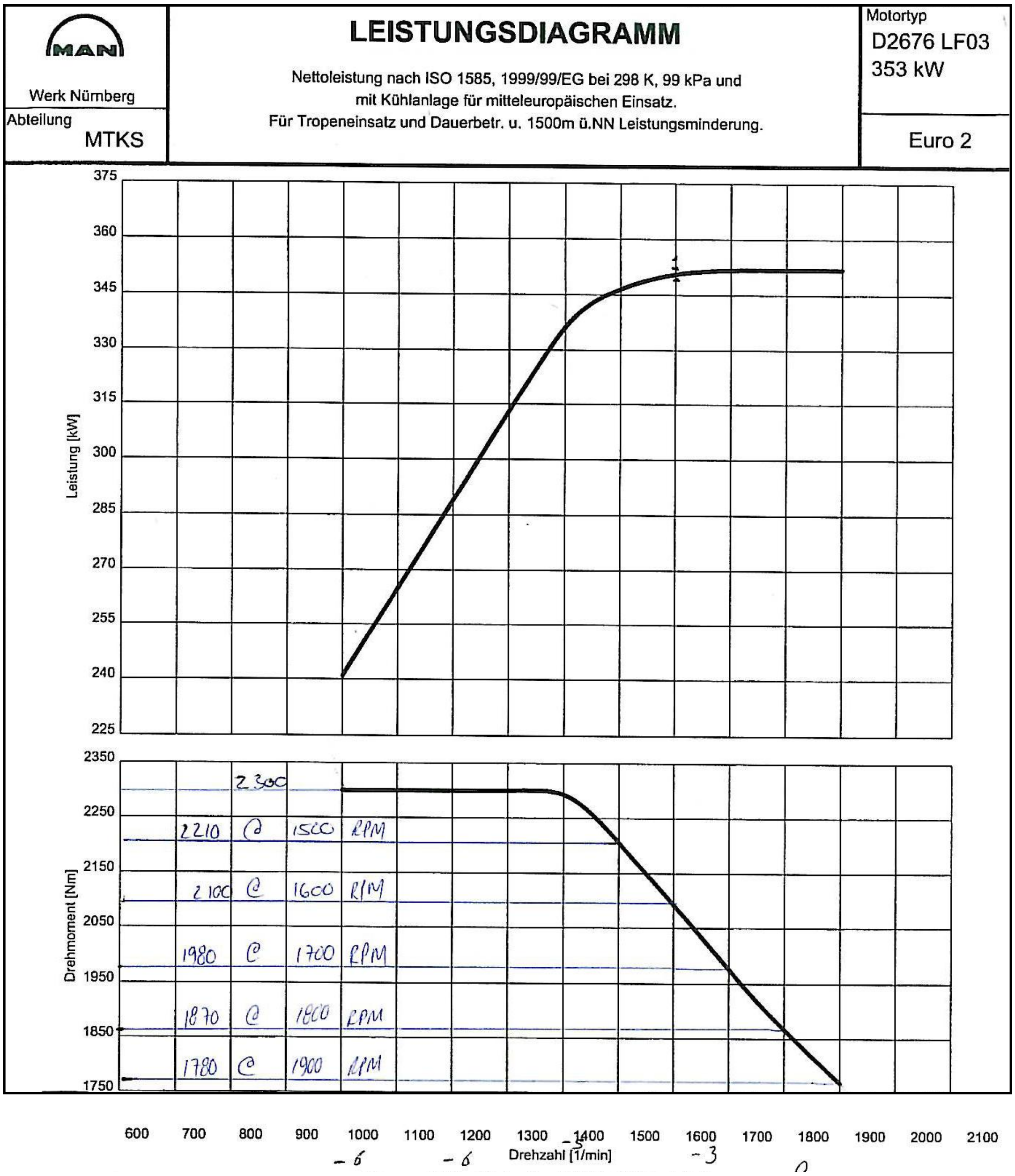



Figure C.1: Power torque curves for MAN TGA 26.480 BLS

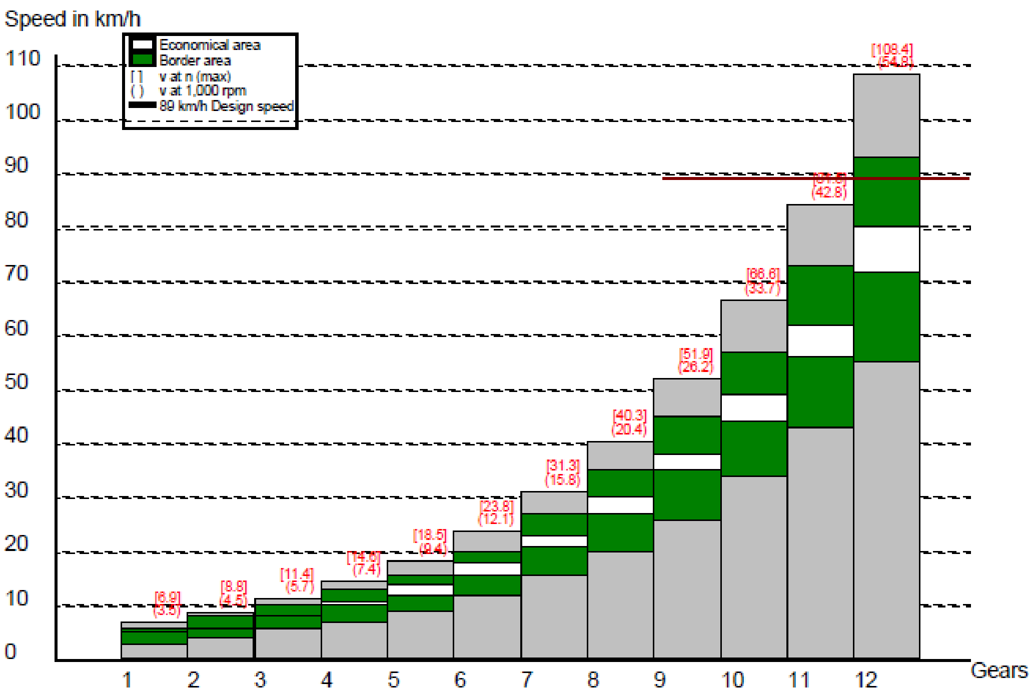



**Figure C.2: Speed vs. Gears illustration for a MAN TGA 46.480 BLS prime mover on dry road from MAN**

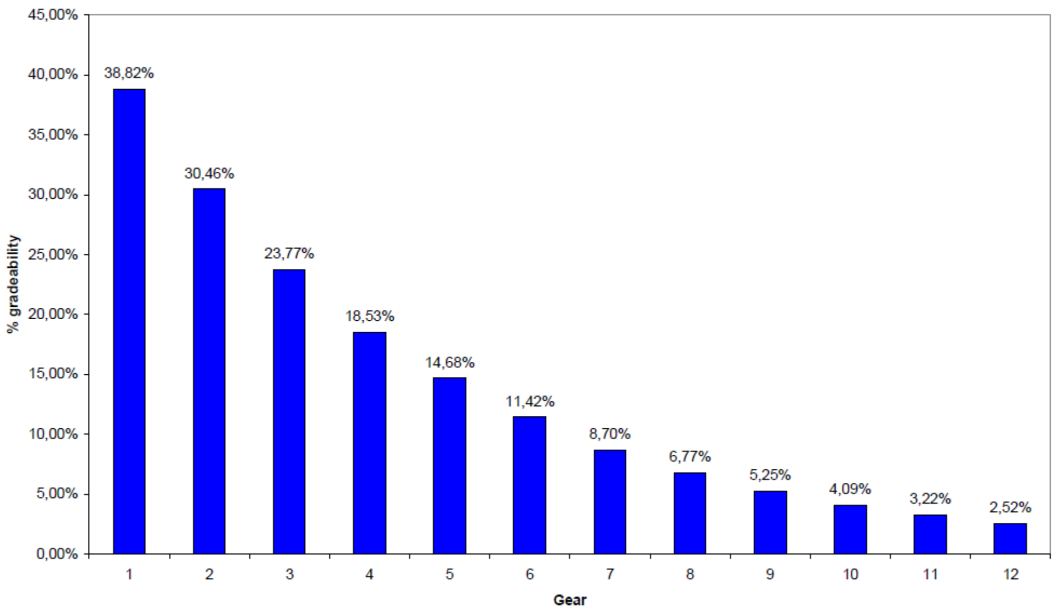



**Figure C.3: Gradeability of a 56 ton GCM for a MAN TGA 26.480 6x4 BLS prime mover from MAN**

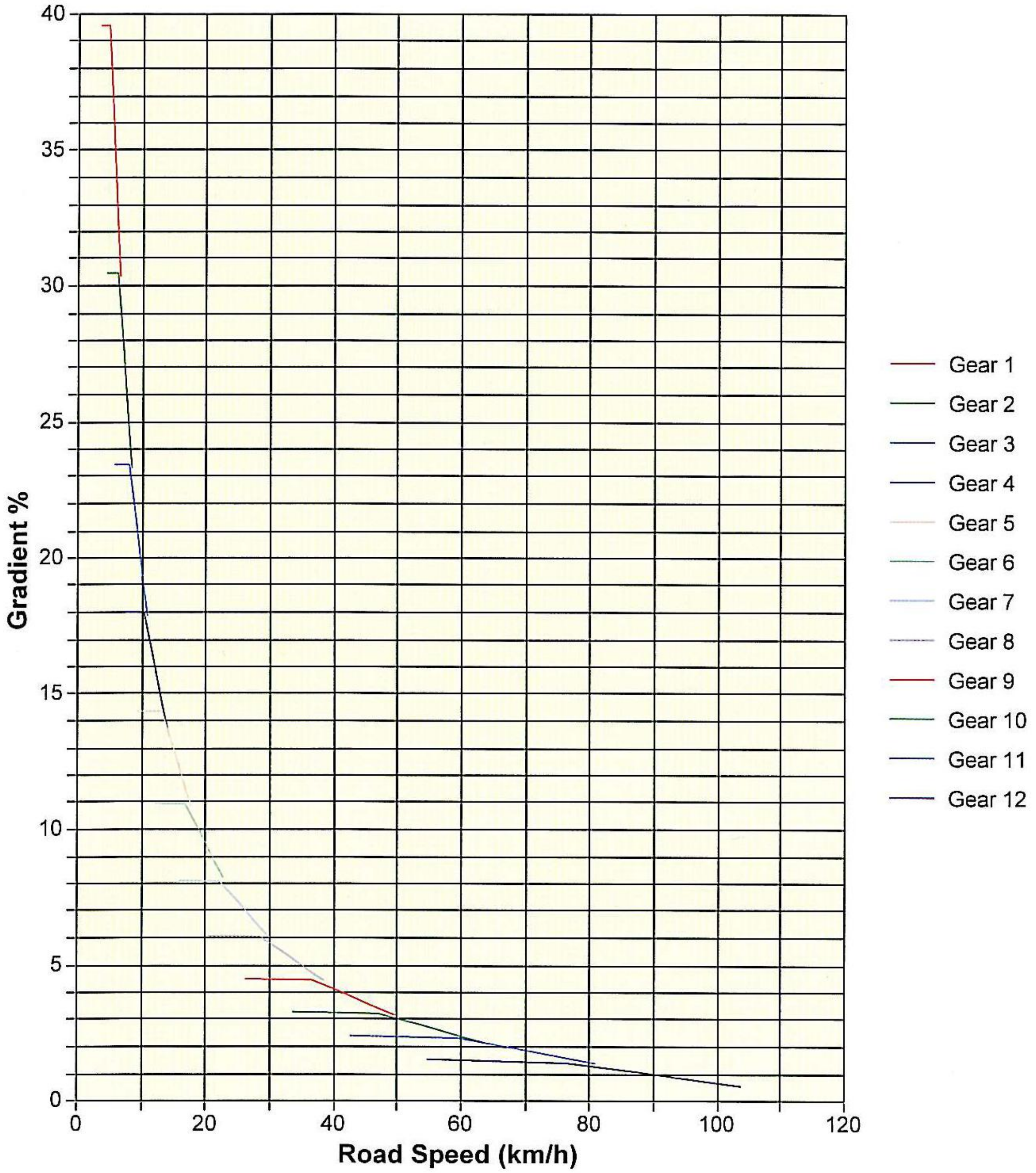



**Figure C.4: Gradient vs. Speed plot of the MAN TGA 26.480 BLS prime mover form HTM**

## Results

| Engine | Gear 1 | | Gear 2 | | Gear 3 | | Gear 4 | | Gear 5 | |
|---|---|---|---|---|---|---|---|---|---|---|
| RPM | km/h | grd % | km/h | grd % | km/h | grd % | km/h | grd % | km/h | grd % |
| 1000 | 3.4 | 39.6 | 4.4 | 30.5 | 5.7 | 23.4 | 7.4 | 18 | 9.3 | 14.4 |
| 1200 | 4.1 | 39.6 | 5.3 | 30.5 | 6.9 | 23.4 | 8.9 | 18 | 11.2 | 14.4 |
| 1400 | 4.8 | 39.6 | 6.2 | 30.5 | 8 | 23.4 | 10.3 | 18 | 13.1 | 14.4 |
| 1600 | 5.5 | 36 | 7.1 | 27.7 | 9.2 | 21.3 | 11.8 | 16.3 | 14.9 | 13 |
| 1800 | 6.2 | 32.1 | 8 | 24.6 | 10.3 | 18.9 | 13.3 | 14.5 | 16.8 | 11.5 |
| 1900 | 6.5 | 30.3 | 8.4 | 23.3 | 10.9 | 17.9 | 14 | 13.7 | 17.7 | 10.9 |

| Engine | Gear 6 | | Gear 7 | | Gear 8 | | Gear 9 | | Gear 10 | |
|---|---|---|---|---|---|---|---|---|---|---|
| RPM | km/h | grd % | km/h | grd % | km/h | grd % | km/h | grd % | km/h | grd % |
| 1000 | 12 | 10.9 | 15.8 | 8.1 | 20.3 | 6.1 | 26.2 | 4.5 | 33.6 | 3.3 |
| 1200 | 14.4 | 10.9 | 18.9 | 8.1 | 24.4 | 6.1 | 31.4 | 4.5 | 40.3 | 3.3 |
| 1400 | 16.8 | 10.9 | 22.1 | 8.1 | 28.4 | 6.1 | 36.6 | 4.5 | 47 | 3.2 |
| 1600 | 19.2 | 9.9 | 25.3 | 7.3 | 32.5 | 5.4 | 41.8 | 4 | 53.7 | 2.8 |
| 1800 | 21.6 | 8.7 | 28.4 | 6.4 | 36.5 | 4.7 | 47.1 | 3.4 | 60.4 | 2.4 |
| 1900 | 22.8 | 8.2 | 30 | 6 | 38.6 | 4.4 | 49.7 | 3.2 | 63.8 | 2.2 |

| Engine | Gear 11 | | Gear 12 | |
|---|---|---|---|---|
| RPM | km/h | grd % | km/h | grd % |
| 1000 | 42.6 | 2.4 | 54.7 | 1.5 |
| 1200 | 51.2 | 2.4 | 65.6 | 1.5 |
| 1400 | 59.7 | 2.3 | 76.5 | 1.4 |
| 1600 | 68.2 | 2 | 87.4 | 1.1 |
| 1800 | 76.7 | 1.6 | 98.4 | 0.7 |
| 1900 | 81 | 1.4 | 103.8 | 0.6 |

**Figure C.5: A tabular illustration of the Gradeability of a MAN TGA 26.480 BLS prime mover at different gear selection and different RPM from HTM**

## C.2 Semi Trailer

### C.2.1 Tracking Ability on a Straight Path

#### *C.2.1.1 Reference Points*

**Table C.2: Reference points for OECD and MAN prime movers, and for five semi trailer combinations**

Unit 1 – Prime mover – OECD 1

| | x | y | z |
|---|---|---|---|
| 1 | 1360 | 1200 | 0 |
| 2 | 1360 | -1200 | 0 |
| 3 | 1360 | 0 | 0 |

OECD 1

| | x | y | z |
|---|---|---|---|
| 1 | 1300 | 1300 | 3120 |
| 2 | 1300 | -1300 | 3120 |
| 3 | -12900 | 1300 | 3120 |
| 4 | -12900 | -1300 | 3120 |
| 5 | -12900 | 1300 | 462 |
| 6 | -12900 | -1300 | 462 |

Refrigeration

| | x | y | z |
|---|---|---|---|
| 1 | 1800 | 1300 | 3069 |
| 2 | 1800 | -1300 | 3069 |
| 3 | -13670 | 1300 | 3069 |
| 4 | -13670 | -1300 | 3069 |
| 5 | -13670 | 1300 | 372 |
| 6 | -13670 | -1300 | 372 |

Tipper

| | x | y | z |
|---|---|---|---|
| 1 | -500 | 1300 | 2322 |
| 2 | -500 | -1300 | 2322 |
| 3 | -11593 | 1300 | 2322 |
| 4 | -11593 | -1300 | 2322 |
| 5 | -11593 | 1300 | 236 |
| 6 | -11593 | -1300 | 236 |

Unit 1 - Prime mover - MAN

| | x | y | z |
|---|---|---|---|
| 1 | 1475 | 1250 | 0 |
| 2 | 1475 | -1250 | 0 |
| 3 | 1475 | 0 | 0 |

Skeletal

| | x | y | z |
|---|---|---|---|
| 1 | 1550 | 1300 | 2322 |
| 2 | 1550 | -1300 | 2322 |
| 3 | -12650 | 1300 | 2322 |
| 4 | -12650 | -1300 | 2322 |
| 5 | -12650 | 1300 | 435 |
| 6 | -12650 | -1300 | 435 |

Side Curtain

| | x | y | z |
|---|---|---|---|
| 1 | 1500 | 1300 | 3180 |
| 2 | 1500 | -1300 | 3180 |
| 3 | -12800 | 1300 | 3180 |
| 4 | -12800 | -1300 | 3180 |
| 5 | -12800 | 1300 | 333 |
| 6 | -12800 | -1300 | 333 |

## *a. OECD 1*

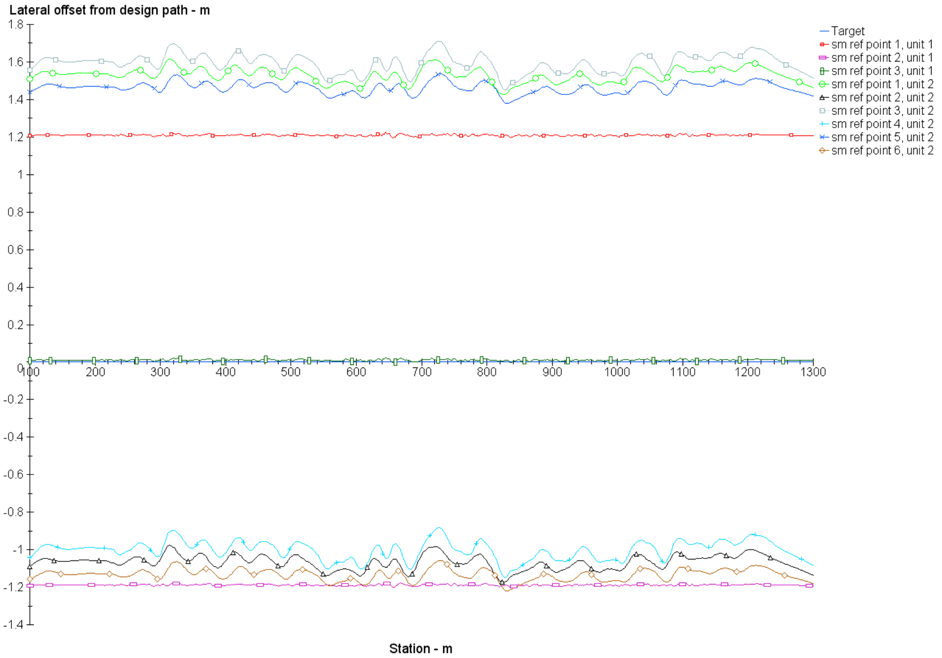



**Figure C.6: Tracking ability on a straight path performance results for OECD 1 semi-trailer**

Please note due to the extensive amount of output data plots for each vehicle, only OECD 1 semi-trailer and OECD 2 B-double have been included in this appendix. The output data plots for the remaining eight vehicles can be found on the CD attached.

### C.2.2 Low speed Swept Path

#### C.2.2.1 Unladen

*a. OECD 1*

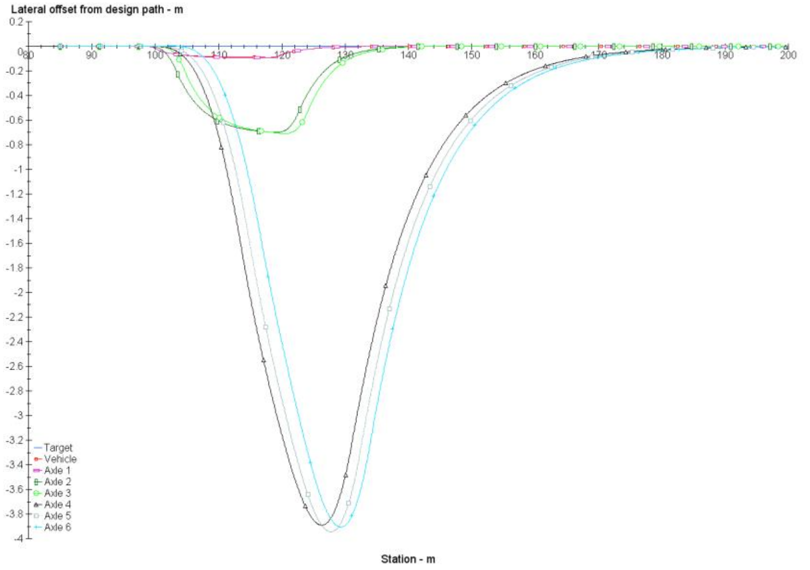



**Figure C.7: Low speed swept path performance result for unladen OECD 1 semi-trailer**

#### C.2.2.2 Laden

*a. OECD 1*

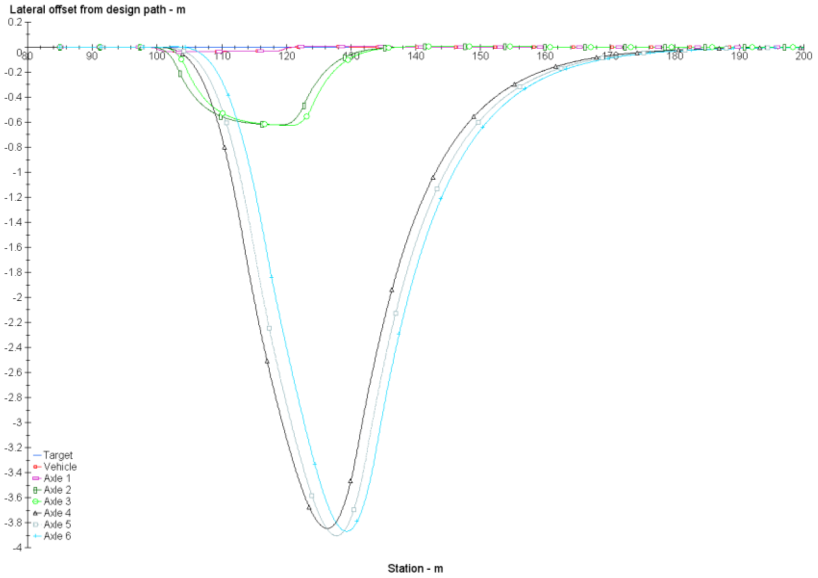



**Figure C. 8: Low speed swept path performance result for laden OECD 1 semi-trailer**

### C.2.3 Frontal Swing

### C.2.3.1 Part A – Hauling unit

#### a. Unladen

##### *i. OECD 1*

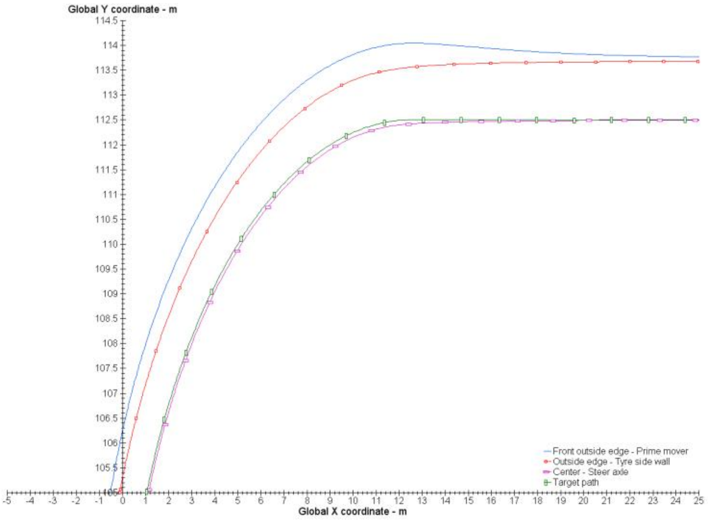



**Figure C.9: Frontal swing, Part A, performance result for unladen OECD 1 semi-trailer**

#### b. Laden

##### *i. OECD 1*

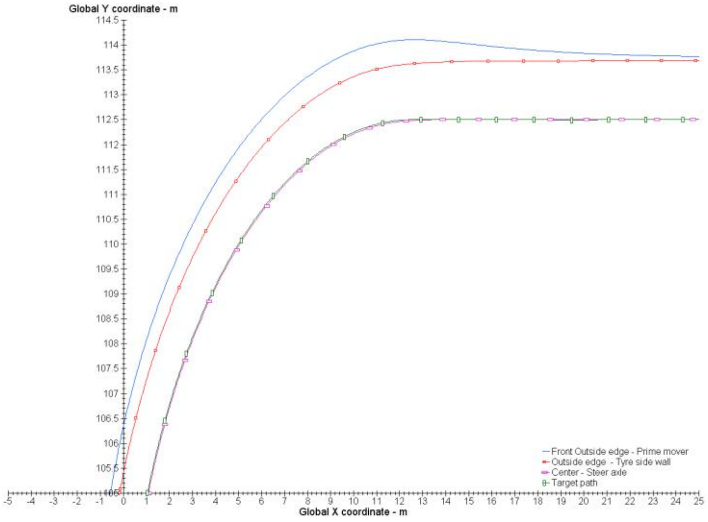



**Figure C.10: Frontal swing, Part A, performance result for laden OECD 1 semi-trailer**

### C.2.3.2 PART B and C

#### a. Unladen

#### *i. OECD 1*

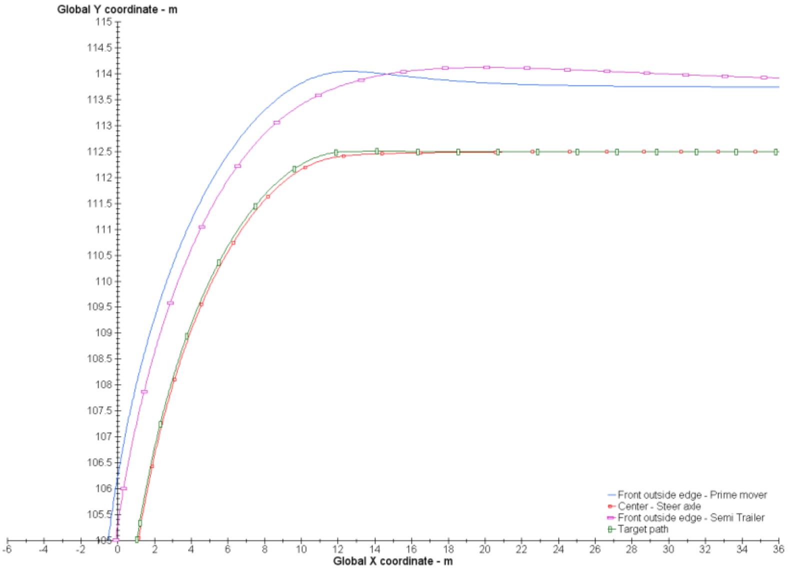



**Figure C.11: Frontal swing, PART B and C, performance results for unladen OECD 1 semi-trailer**

#### b. Laden

#### *i. OECD 1*

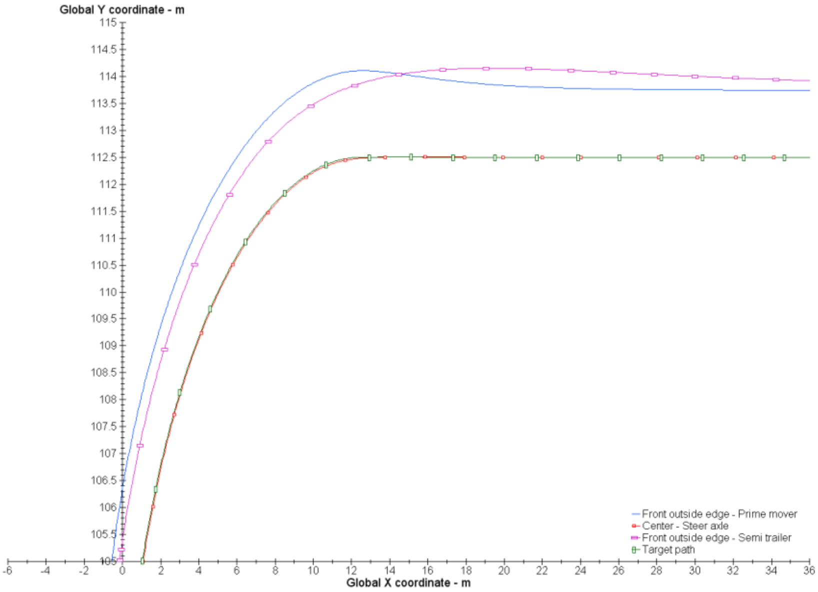



**Figure C.12: Frontal swing, Part B and C, performance result for laden OECD 1 semi-trailer**

## C.2.4 Tail Swing

### C.2.4.1 Unladen

#### a. OECD 1

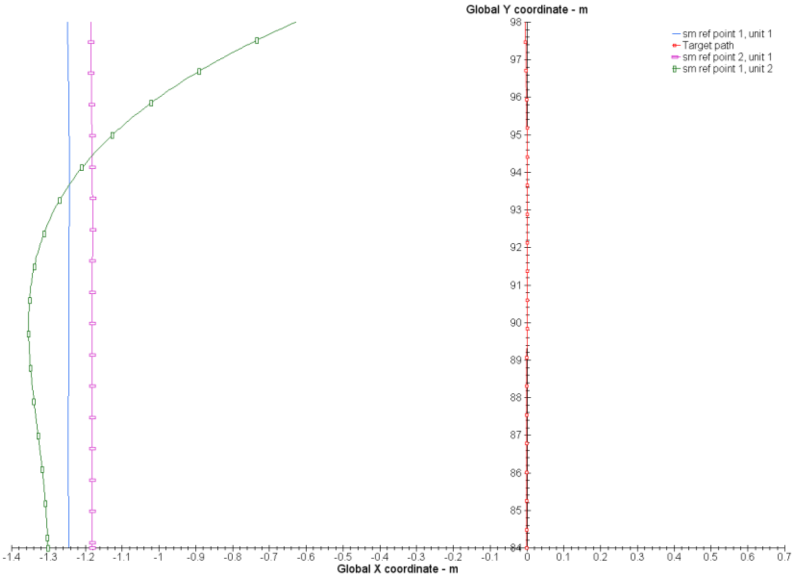



**Figure C.13: Tail swing, entry to turn, performance result for unladen OECD 1 semi-trailer**

### C.2.4.2 Laden

#### a. OECD

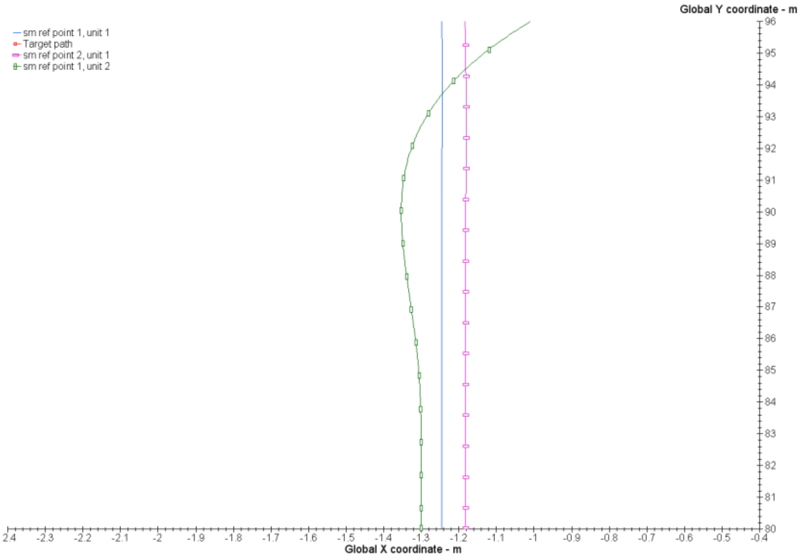



**Figure C.14: Tail swing, entry to turn, performance result for laden OECD 1 semi-trailer**

## C.2.5 Steer Tyre Friction Demand

### C.2.5.1 Unladen

#### a. OECD 1

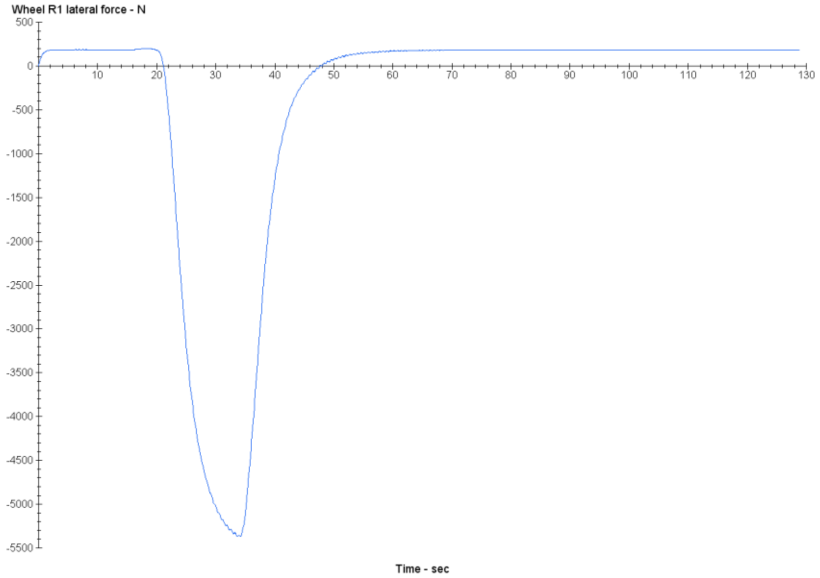



**Figure C.15: Steer tyre friction demand performance result, right hand side lateral tyre force, for OECD 1 semi-trailer**

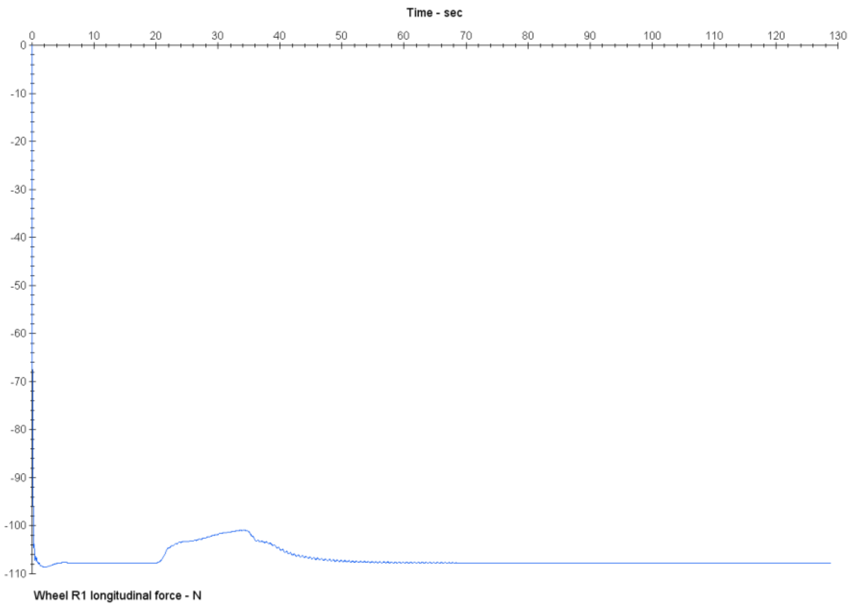



**Figure C.16: Steer tyre friction demand performance result, right hand side longitudinal tyre force, for OECD 1 semi-trailer**

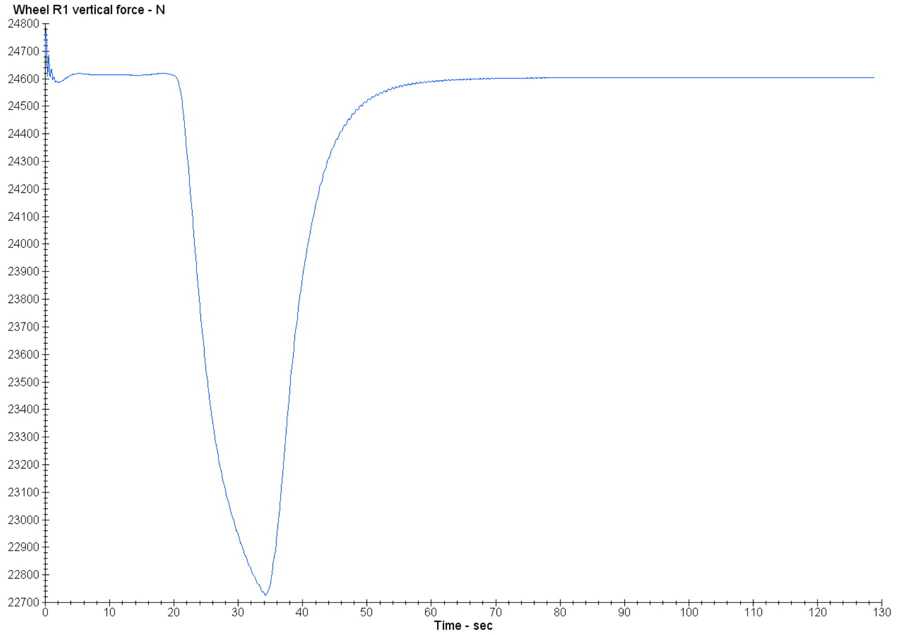



Figure C.17: Steer tyre friction demand performance result, right hand side vertical tyre force, for OECD 1 semi-trailer

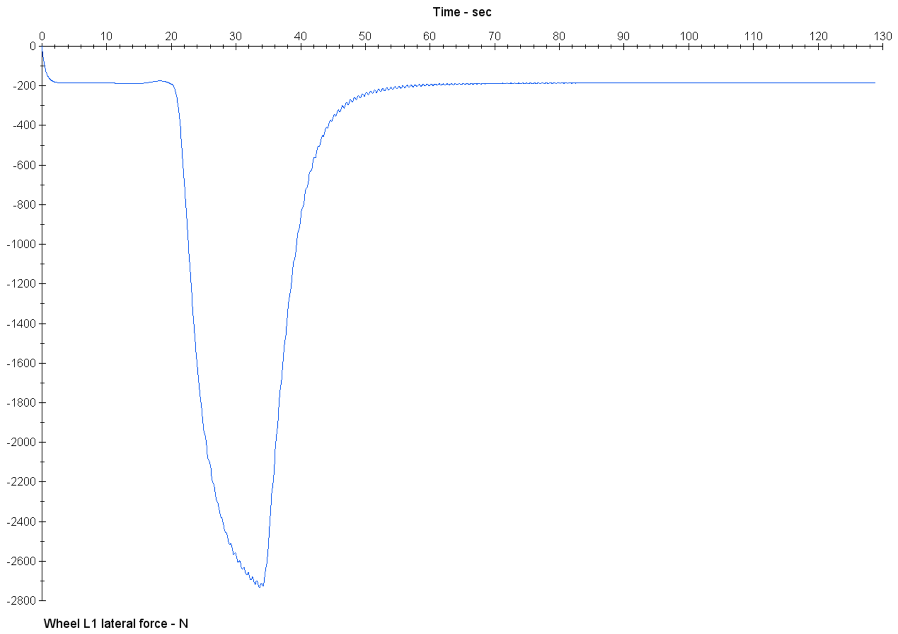



Figure C.18: Steer tyre friction demand performance result, left hand side lateral tyre force, for OECD 1 semi-trailer

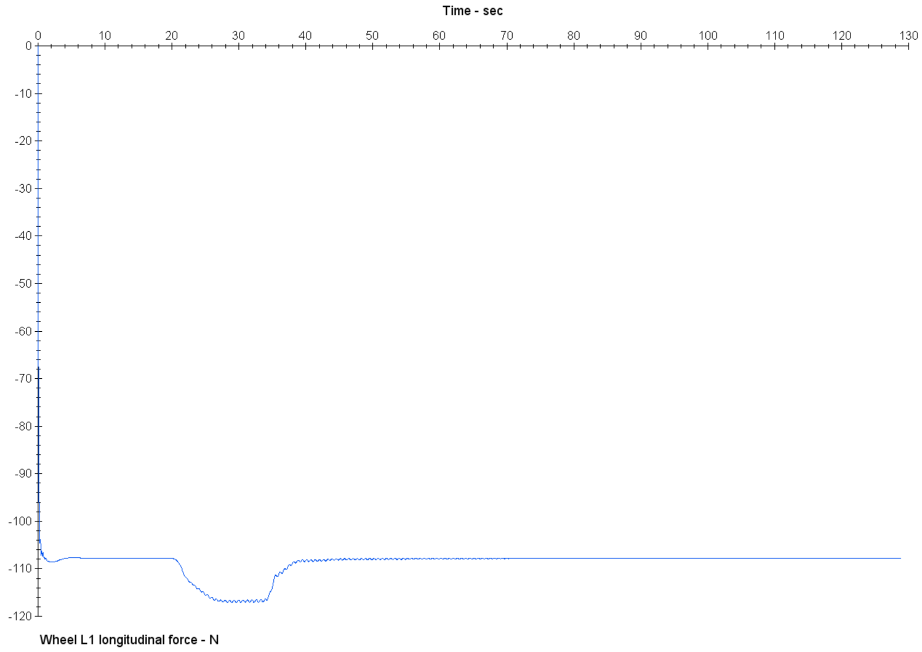



**Figure C.19: Steer tyre friction demand performance result, left hand side longitudinal tyre force, for OECD 1 semi-trailer**

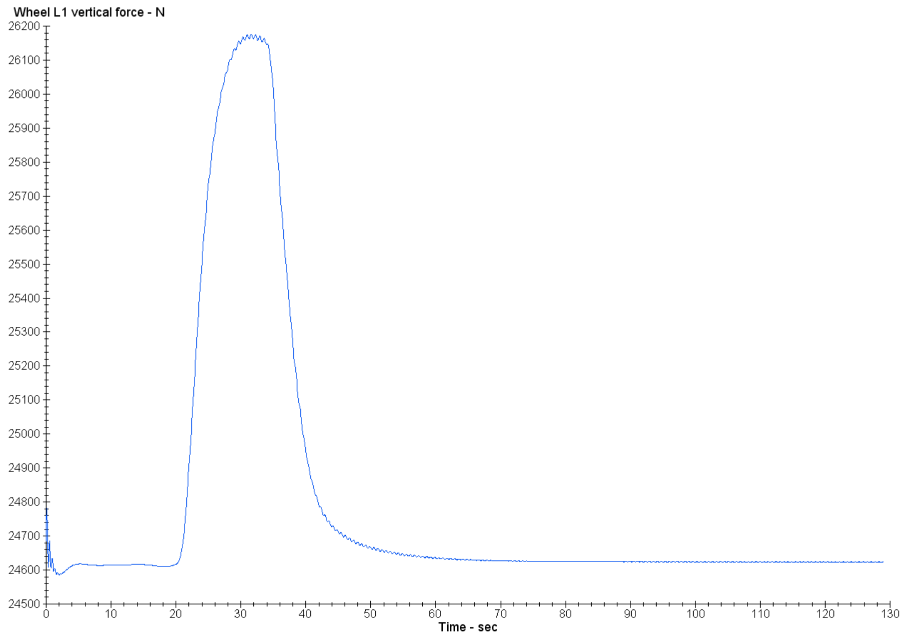



**Figure C.20: Steer tyre friction demand performance result, left hand side vertical tyre force, for OECD 1 semi-trailer**

### C.2.5.2 Laden

#### a. OECD 1

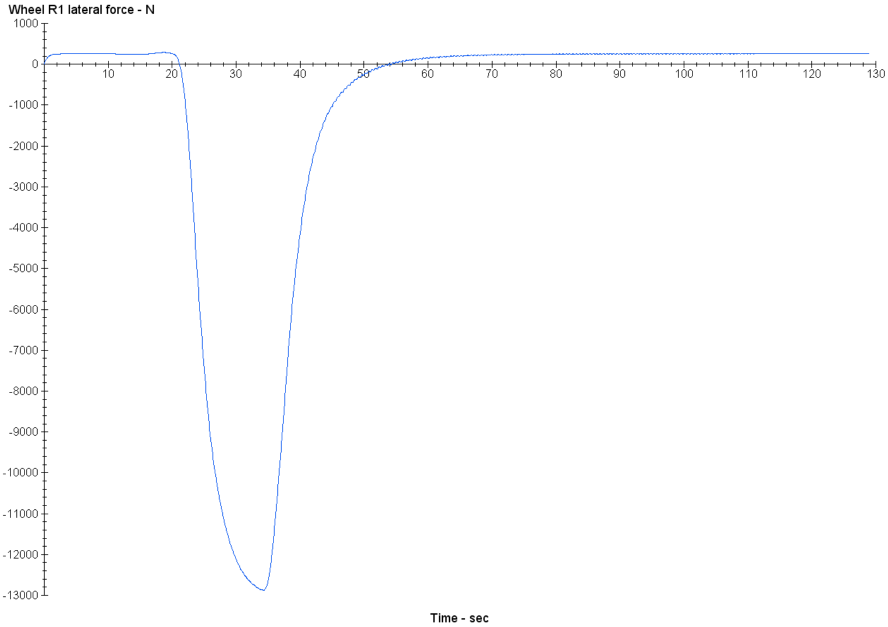



Figure C.21: Steer tyre friction demand performance result, right hand side lateral tyre force, for OECD 1 semi-trailer

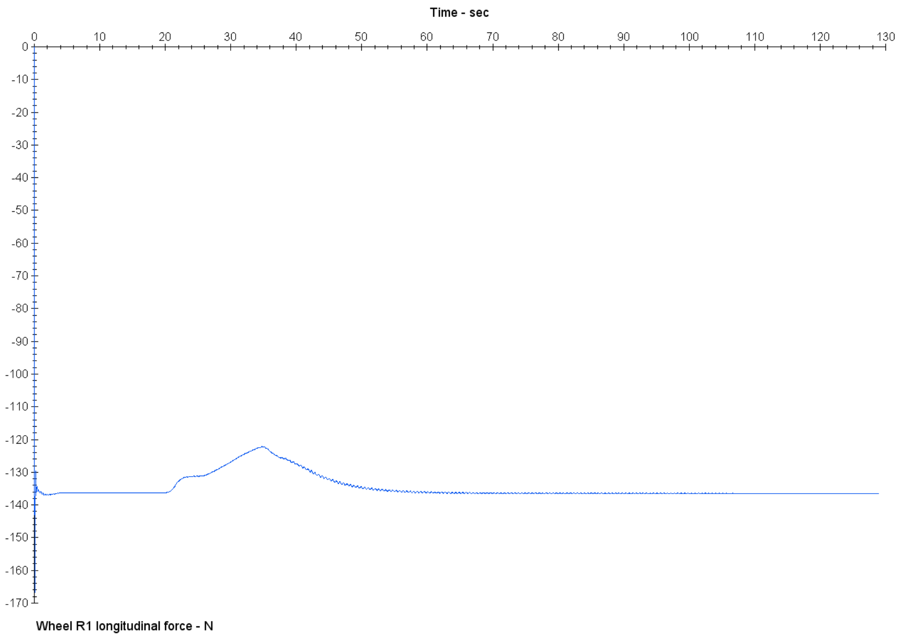



Figure C.22: Steer tyre friction demand performance result, right hand side longitudinal tyre force, for OECD 1 semi-trailer

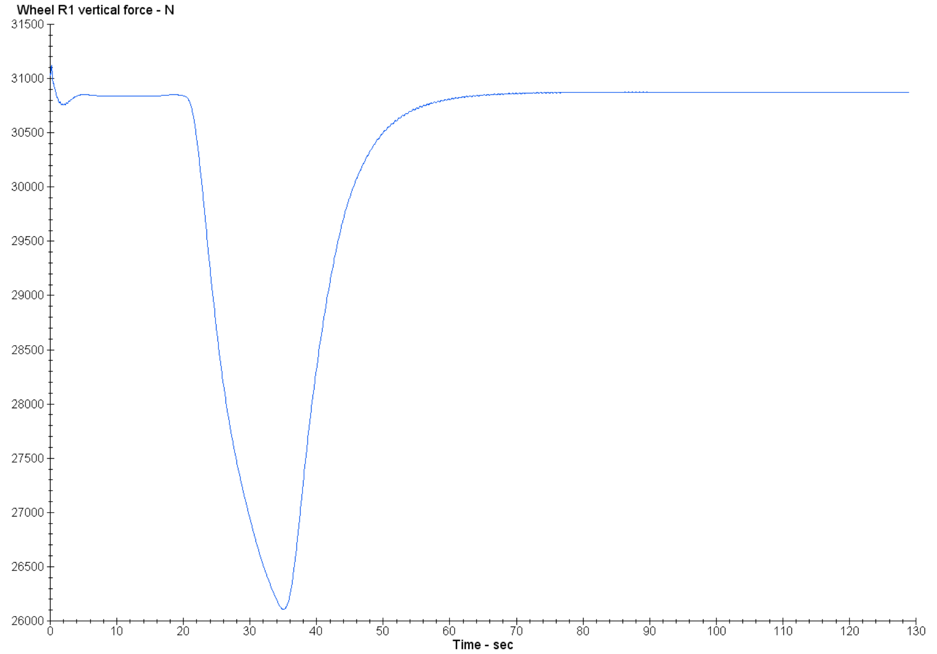



Figure C.23: Steer tyre friction demand performance result, right hand side vertical tyre force, for OECD 1 semi-trailer

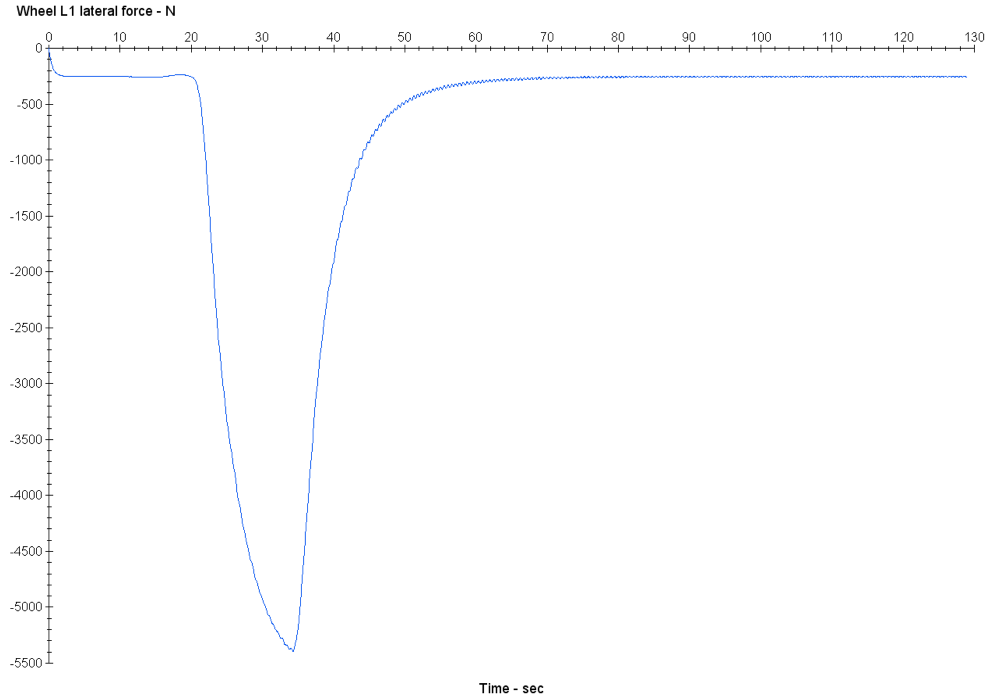



Figure C.24: Steer tyre friction demand performance result, left hand side lateral tyre force, for OECD 1 semi-trailer

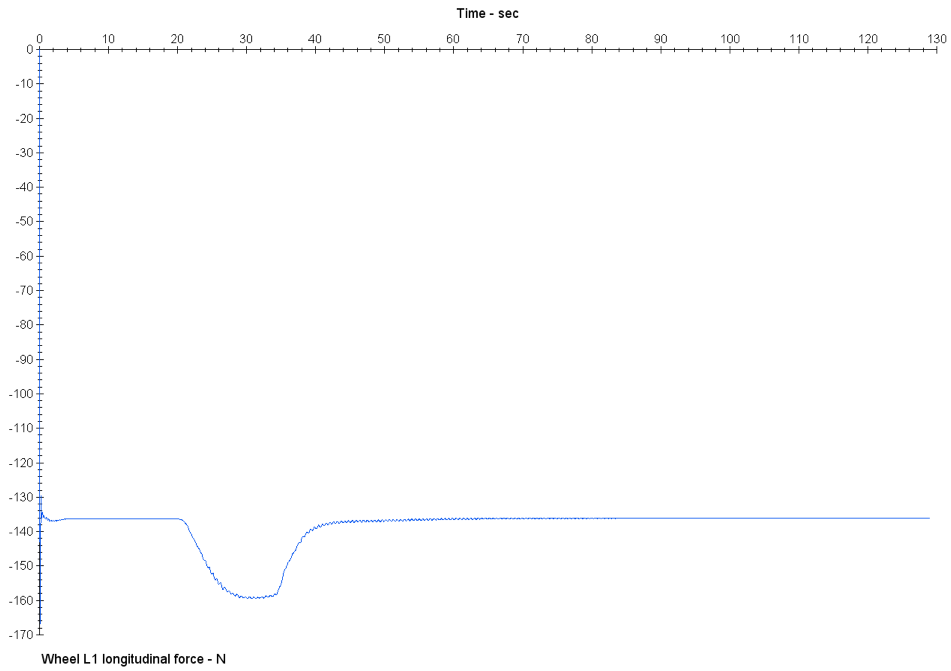



**Figure C.25: Steer tyre friction demand performance result, left hand side longitudinal tyre force, for OECD 1 semi-trailer**

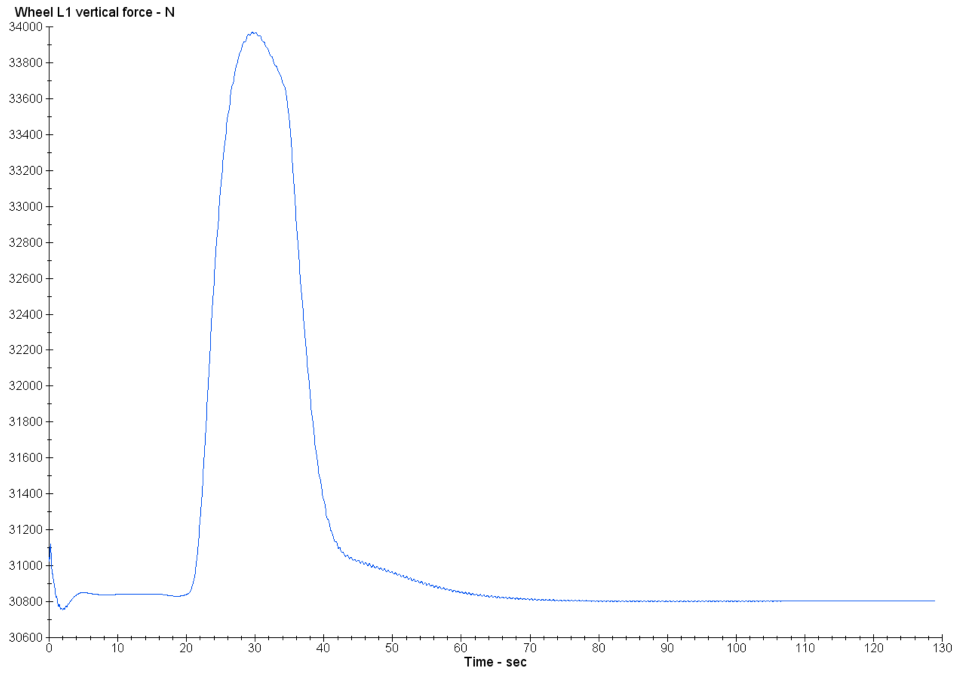



**Figure C.26: Steer tyre friction demand performance result, left hand side vertical tyre force, for OECD 1 semi-trailer**

## C.2.6 Static Rollover Threshold

### C.2.6.1 Circular test

#### a. OECD1

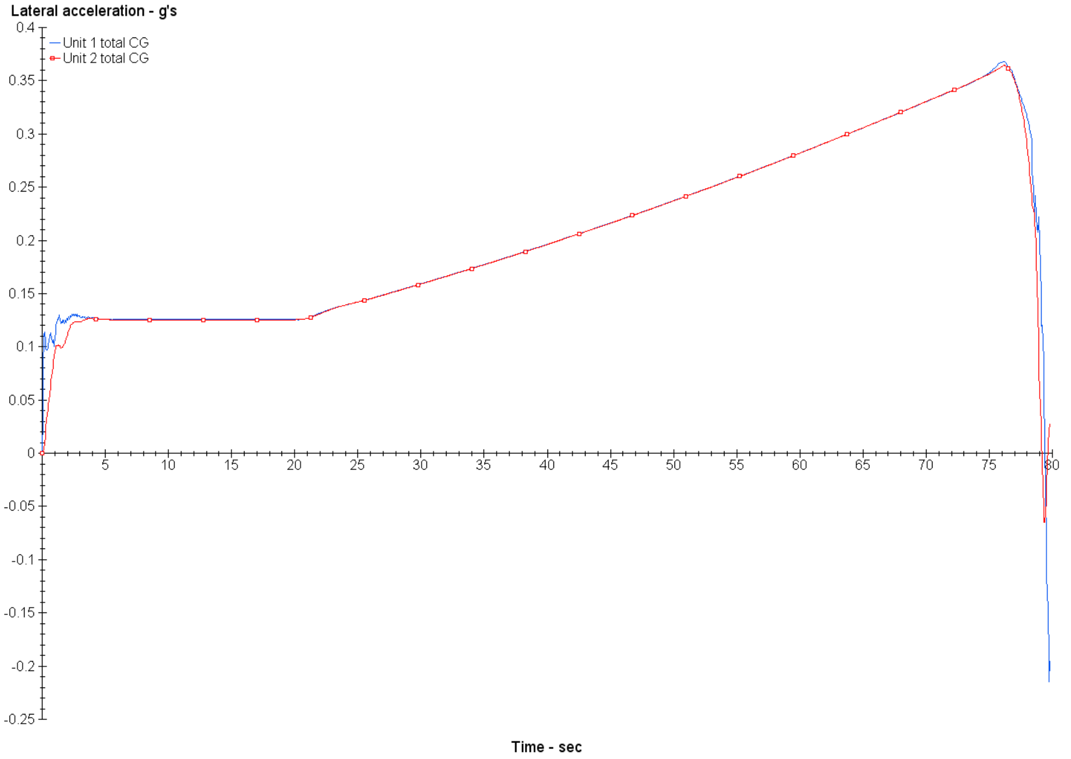



Figure C.27: Static rollover threshold, circular test, performance result for OECD 1 semi-trailer

### C.2.6.2 Tilt table test

#### a. OECD 1

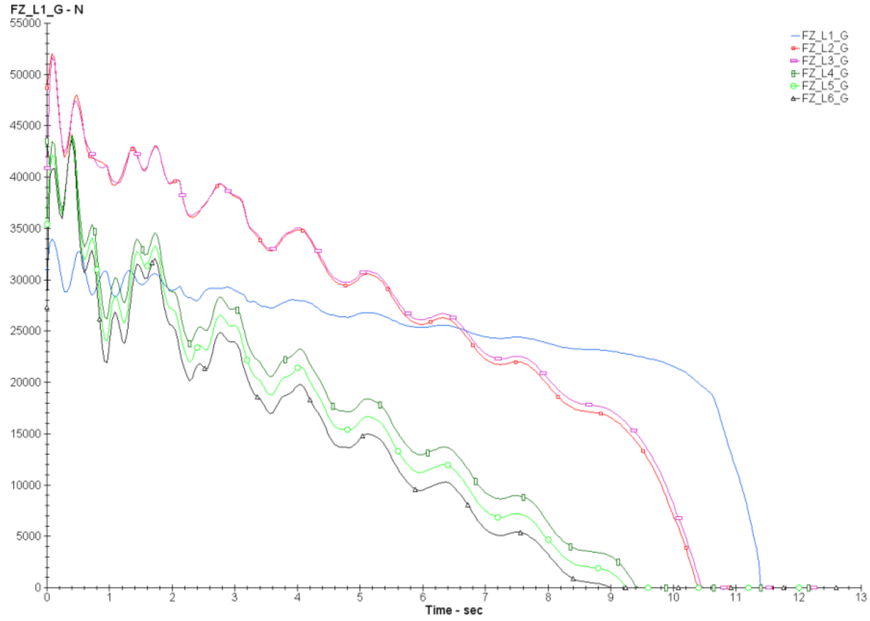



**Figure C.28: Vertical tyre forces for static rollover threshold, tilt table test, of OECD 1 semi-trailer**

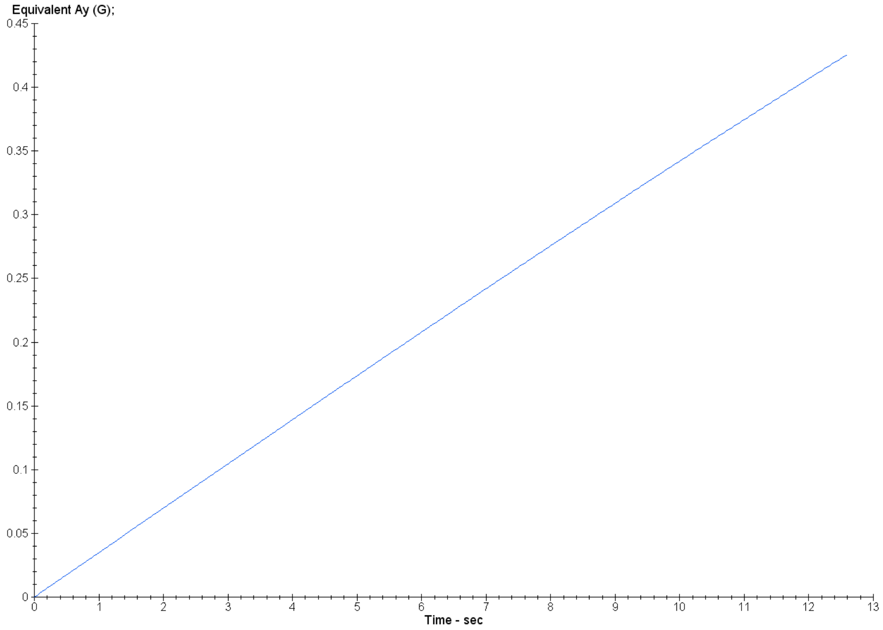



**Figure C.29: Equivalent lateral acceleration plot for static rollover threshold, tilt table test, of OECD 1 semi-trailer**

## C.2.7 Rearward Amplification

### *C.2.7.1 OECD 1*

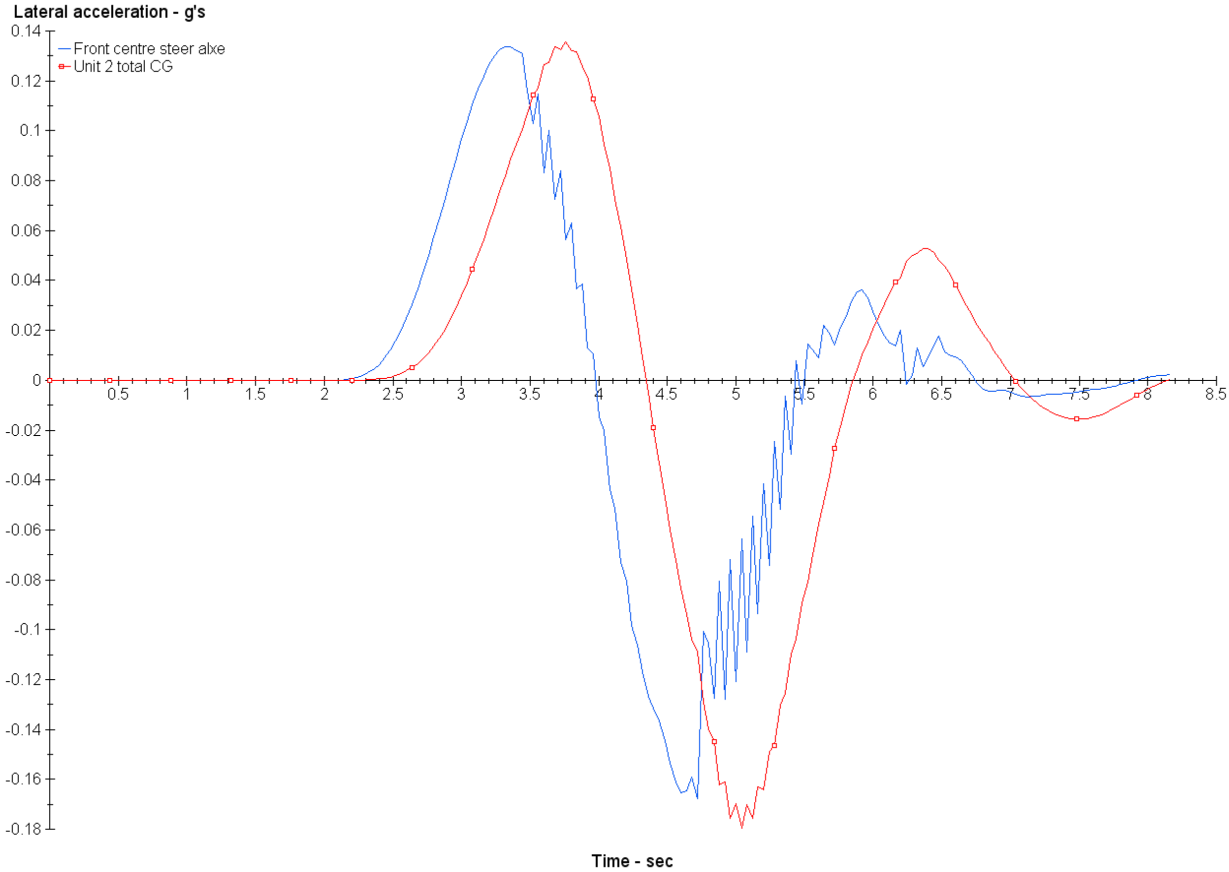



**Figure C.30: Rearward Amplification performance result for OECD 1 semi-trailer**

## C.2.8 High Speed Transient Off-tracking

### C.2.8.1 OECD 1

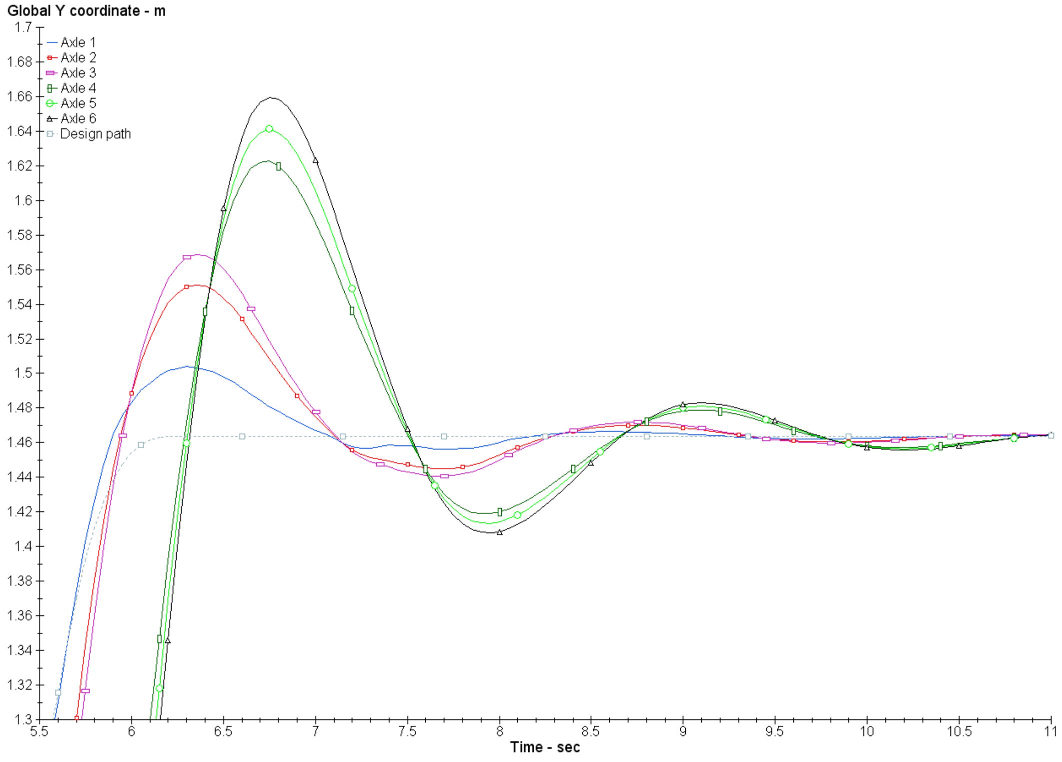



Figure C.31: High speed transient off-tracking result for OECD 1 semi-trailer

## C.2.9 Yaw Damping Co-efficient

### C.2.9.1 OECD 1

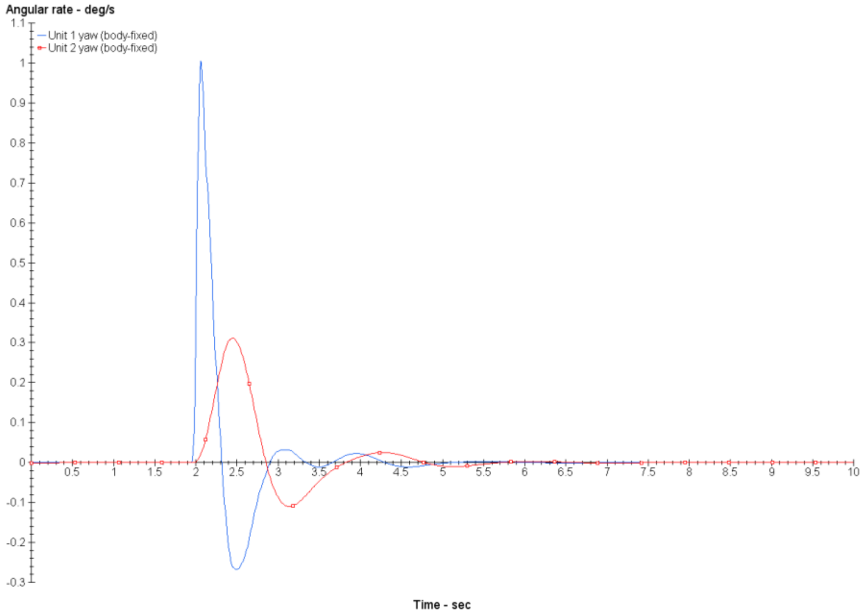



**Figure C.32: Yaw damping, unit 1 and unit 2, result for OECD 1 semi-trailer**

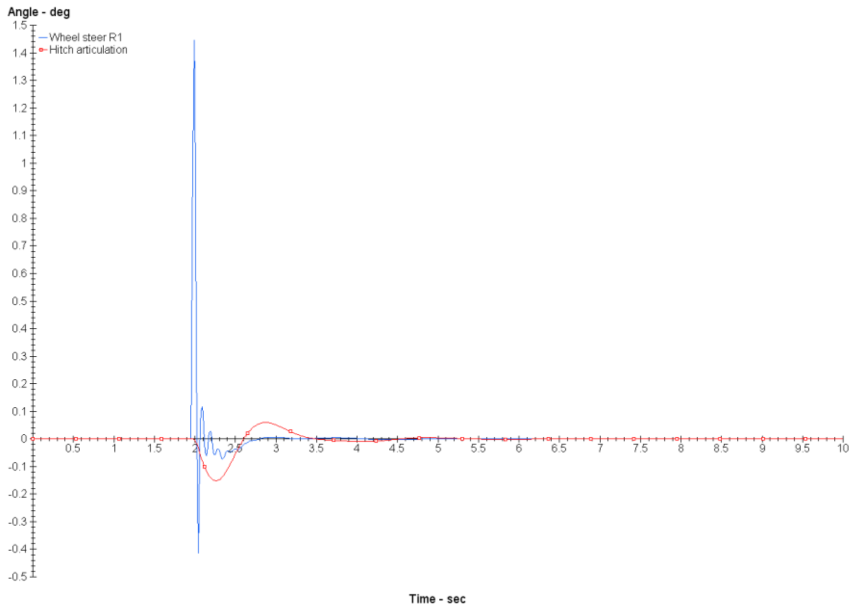



**Figure C.33: Yaw damping, hitch 1, result for OECD 1 semi-trailer**

## C.3 B-double

### C.3.1 Tracking Ability on a Straight Path

#### C.3.1.1 Reference Points

**Table C.3: Reference points for OECD and MAN prime movers, and for five B-double trailer combinations**

Unit 1 - Prime mover – OECD 2

| | x | y | z |
|---|---|---|---|
| 1 | 1360 | 1200 | 0 |
| 2 | 1360 | -1200 | 0 |
| 3 | 1360 | 0 | 0 |

Unit 1 – Prime mover - MAN

| | x | y | z |
|---|---|---|---|
| 1 | 1475 | 1250 | 0 |
| 2 | 1475 | -1250 | 0 |
| 3 | 1475 | 0 | 0 |

OECD 2

| | x | y | z |
|---|---|---|---|
| 1 | -10448 | 1300 | 3120 |
| 2 | -10448 | -1300 | 3120 |
| 3 | -10448 | 1300 | 437 |
| 4 | -10448 | -1300 | 437 |

Skeletal

| | x | y | z |
|---|---|---|---|
| 1 | -10448 | 1300 | 2322 |
| 2 | -10448 | -1300 | 2322 |
| 3 | -10448 | 1300 | 440 |
| 4 | -10448 | -1300 | 440 |

Cane

| | x | y | z |
|---|---|---|---|
| 1 | -9600 | 1300 | 2923 |
| 2 | -9600 | -1300 | 2923 |
| 3 | -9600 | 1300 | 373 |
| 4 | -9600 | -1300 | 373 |

Side Curtain

| | x | y | z |
|---|---|---|---|
| 1 | -10480 | 1300 | 3180 |
| 2 | -10480 | -1300 | 3180 |
| 3 | -10480 | 1300 | 375 |
| 4 | -10480 | -1300 | 375 |

Tipper

| | x | y | z |
|---|---|---|---|
| 1 | -8184 | 1300 | 2345 |
| 2 | -8184 | -1300 | 2345 |
| 3 | -8184 | 1300 | 496 |
| 4 | -8184 | -1300 | 496 |

## a. OECD 1

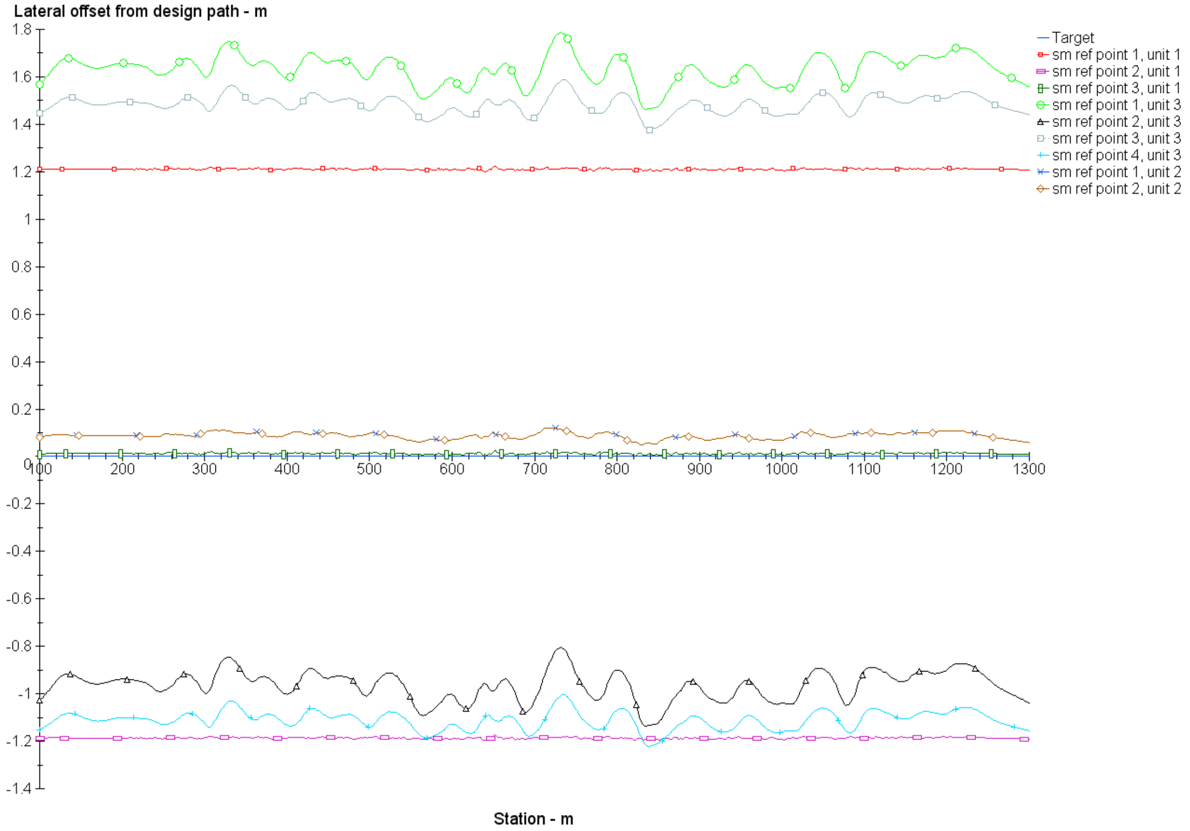



Figure C.34: Tracking ability on a straight path performance results for OECD 2 B-double configuration

### C.3.2 Low Speed Swept Path

#### C.3.2.1 Unladen

##### a. OECD 1

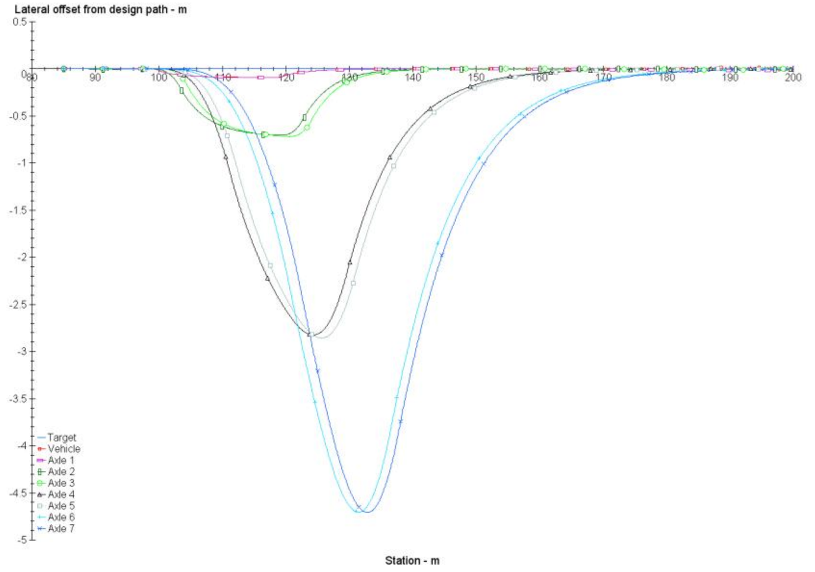



**Figure C.35: Low speed swept path performance result for unladen OECD 2 B-double**

#### C.3.2.2 Laden

##### a. OECD 1

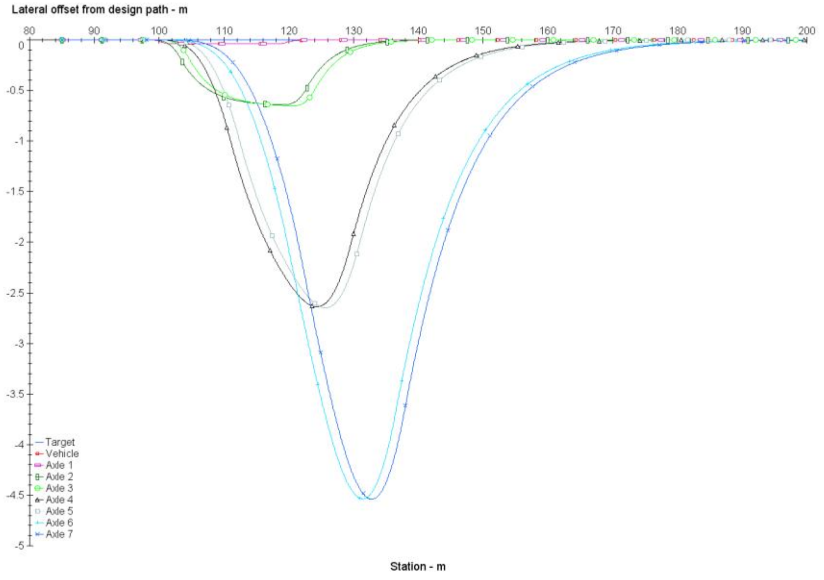



**Figure C.36: Low speed swept path performance result for laden OECD 2 B-double**

### C.3.3 Frontal Swing

#### C.3.3.1 Part A – Hauling unit

**a. Unladen**

**i. OECD 1**

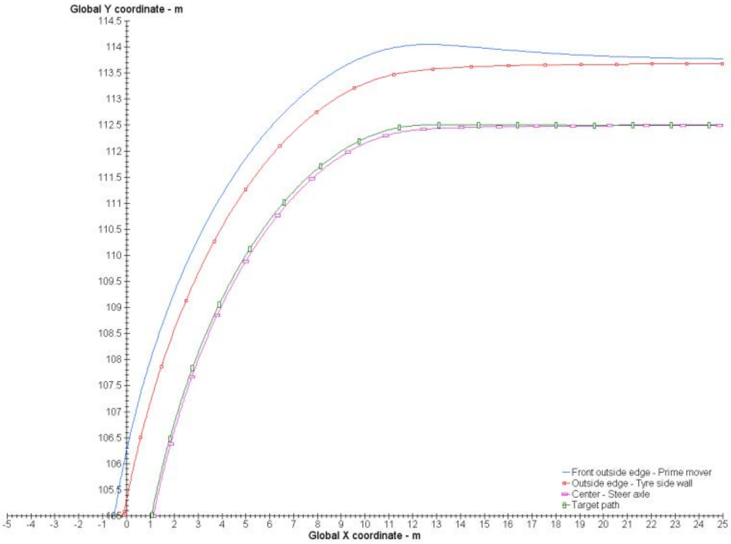



**Figure C.37: Frontal swing, Part A, performance result for unladen OECD 2 B-double**

**b. Laden**

**i. OECD 1**

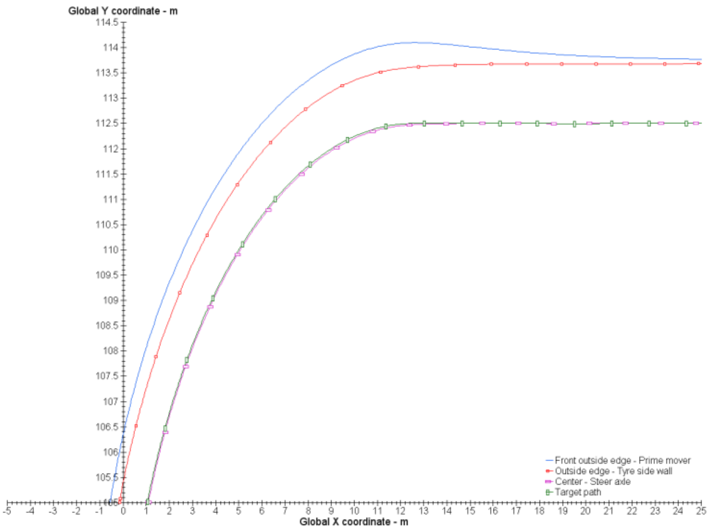



**Figure C.38: Frontal swing, Part A, performance result for laden OECD 2 B-double**

### C.3.3.2 Part B and C

#### a. Unladen

#### i. OECD 1

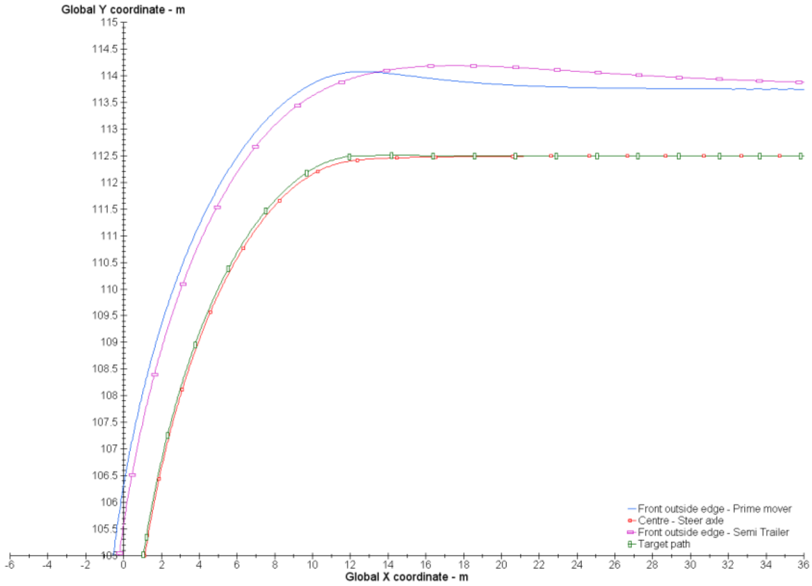



Figure C.39: Frontal swing, PART B and C, performance results for unladen OECD 2 B-double

#### b. Laden

#### i. OECD 1

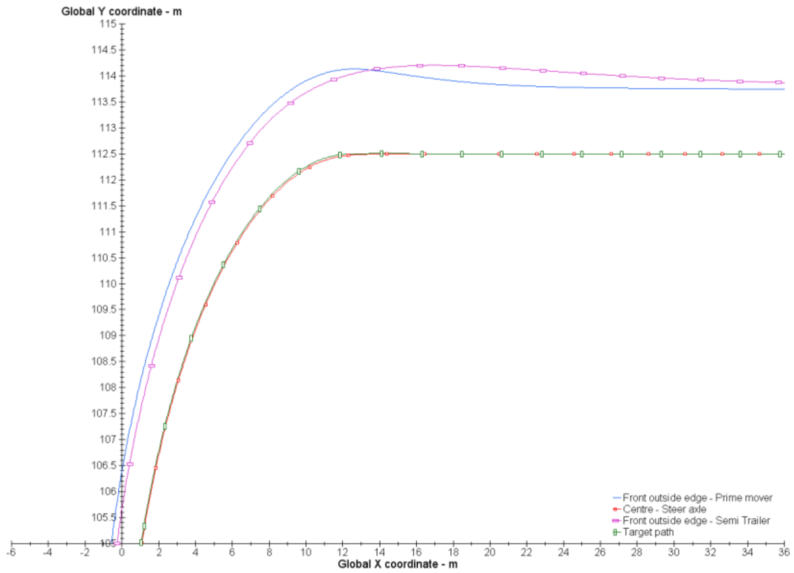



Figure C 40: Frontal swing, PART B and C, performance results for laden OECD 2 B-double

### C.3.4 Tail Swing

#### C.3.4.1 Entry

##### a. Unladen

##### i. OECD 1

The deviations from the target path was so small, as such these plots have been omitted

##### b. Laden

##### i. OECD 1

The deviations from the target path was so small, as such these plots have been omitted

### C.3.5 Steer Tyre Friction Demand

#### C.3.5.1 Unladen

a. **OECD 1**

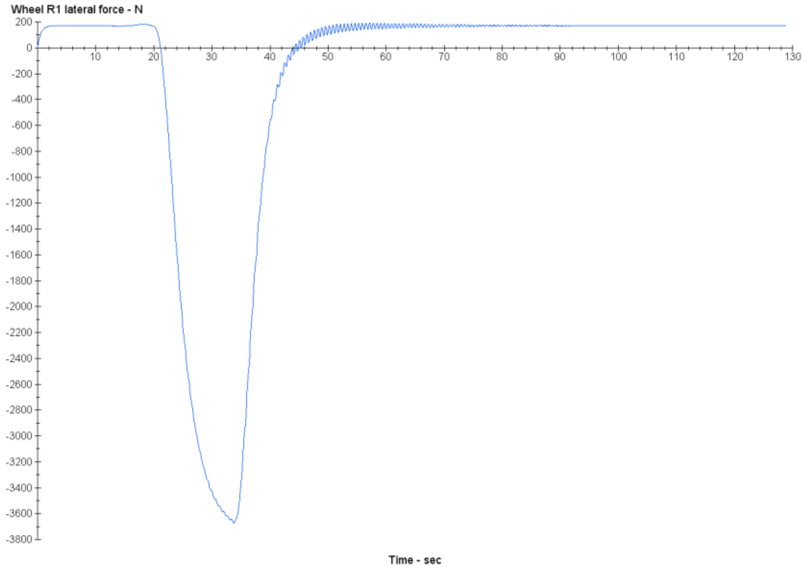



Figure C.41: Steer tyre friction demand performance result, right hand side lateral tyre force, for OECD 2 B-double

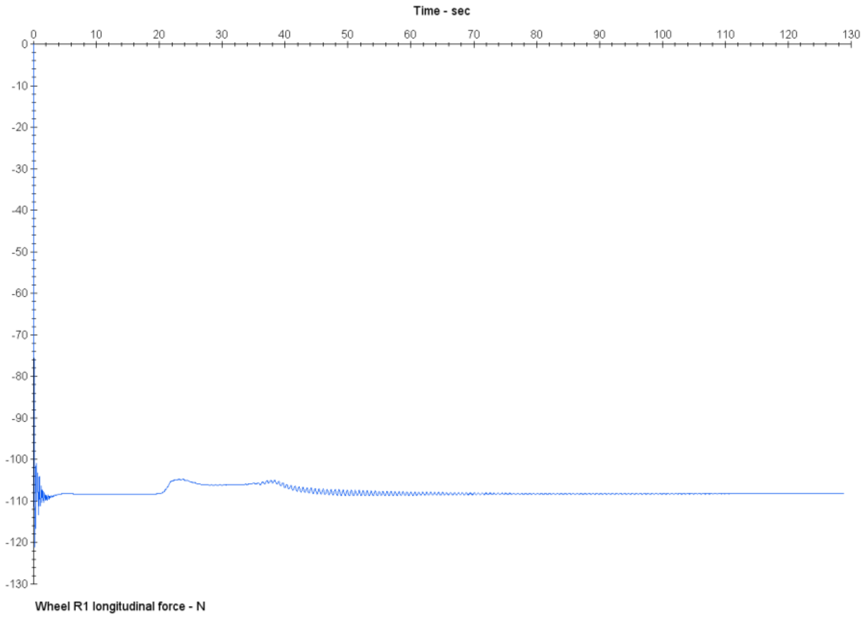



Figure C.42: Steer tyre friction demand performance result, right hand side longitudinal tyre force, for OECD 2 B-double

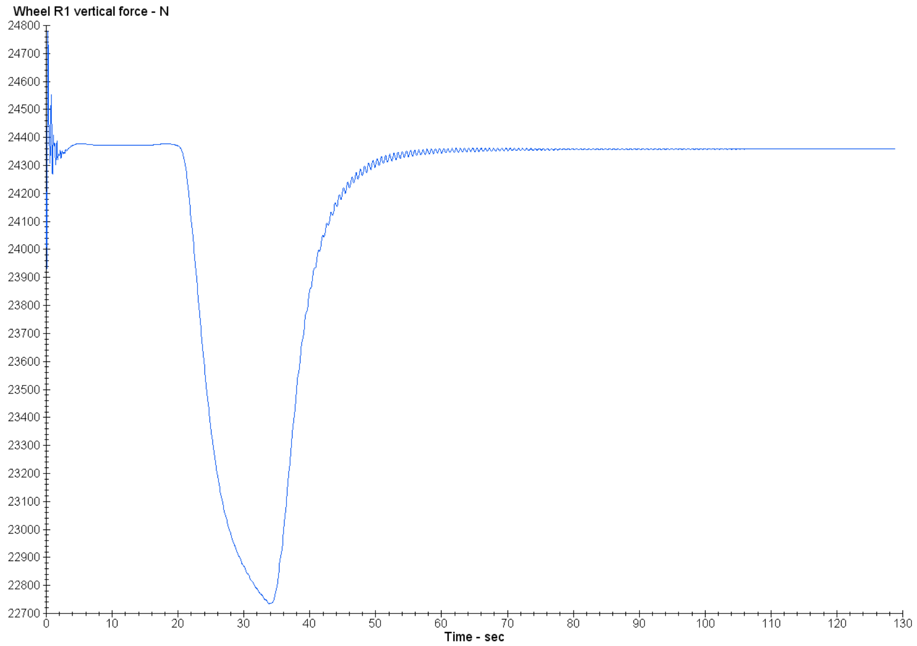



**Figure C.43: Steer tyre friction demand performance result, right hand side vertical tyre force, for OECD 2 B-double**

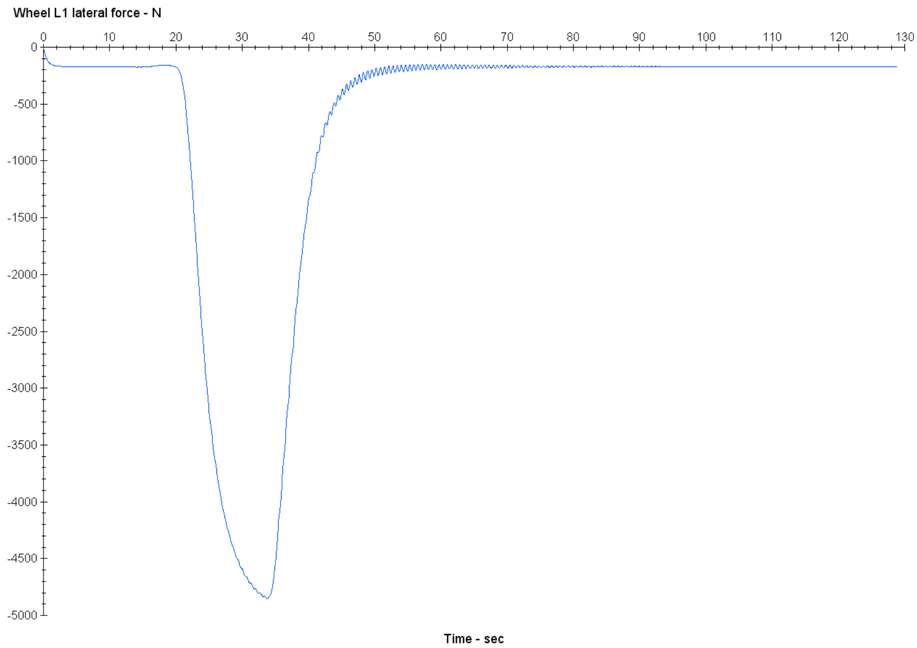



**Figure C.44: Steer tyre friction demand performance result, left hand side lateral tyre force, for OECD 2 B-double**

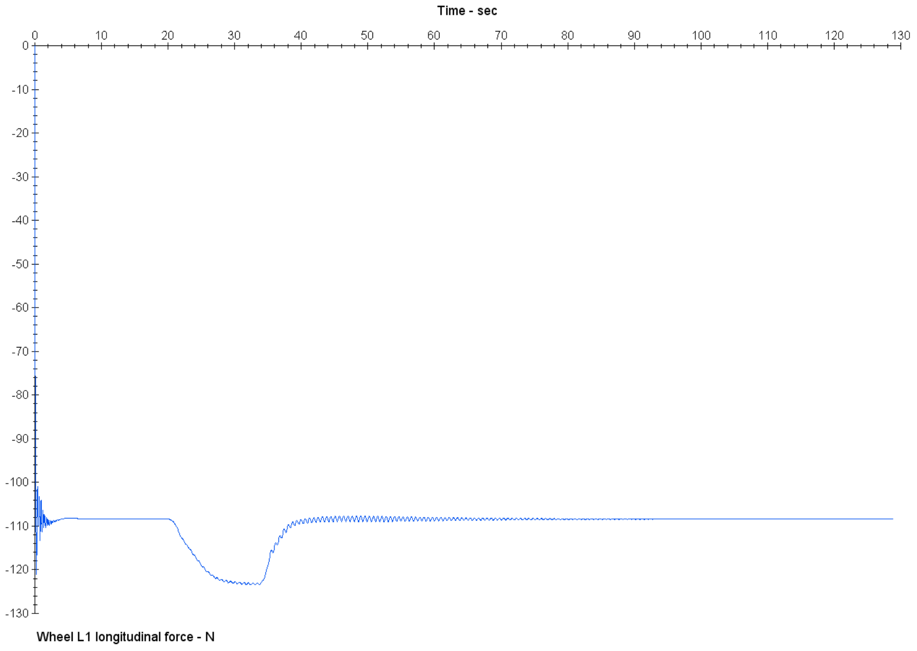



**Figure C.45: Steer tyre friction demand performance result, left hand side longitudinal tyre force, for OECD 2 B-double**

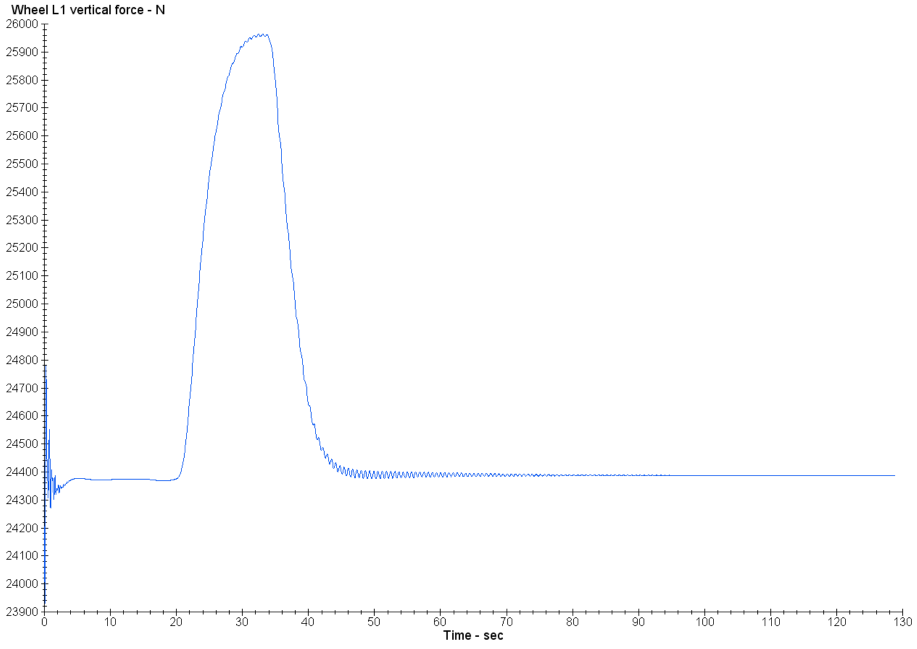



**Figure C.46: Steer tyre friction demand performance result, left hand side vertical tyre force, for OECD 2 B-double**

### C.3.5.2 Laden

#### a. OECD 1

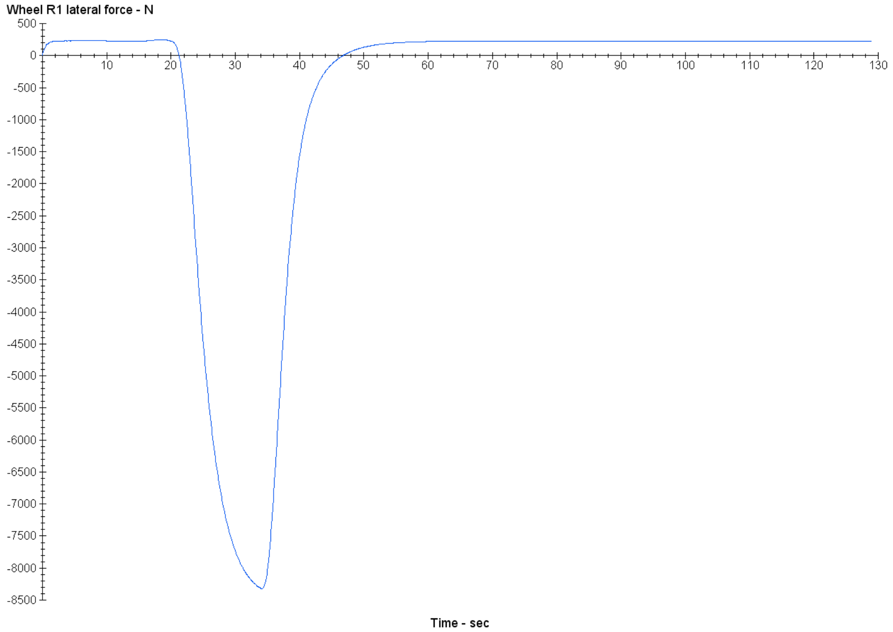



Figure C.47: Steer tyre friction demand performance result, right hand side lateral tyre force, for OECD 2 B-double

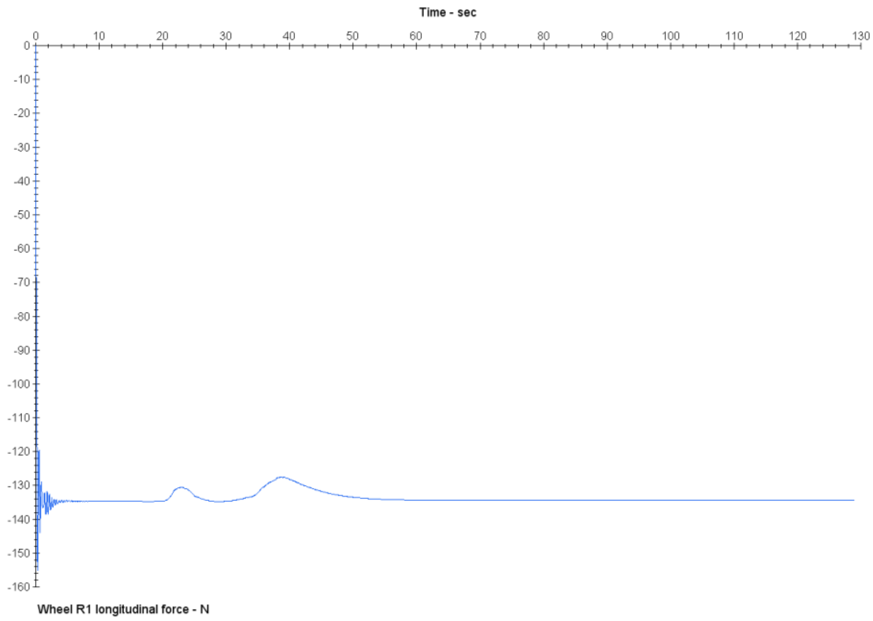



Figure C.48: Steer tyre friction demand performance result, right hand side longitudinal tyre force, for OECD 2 B-double

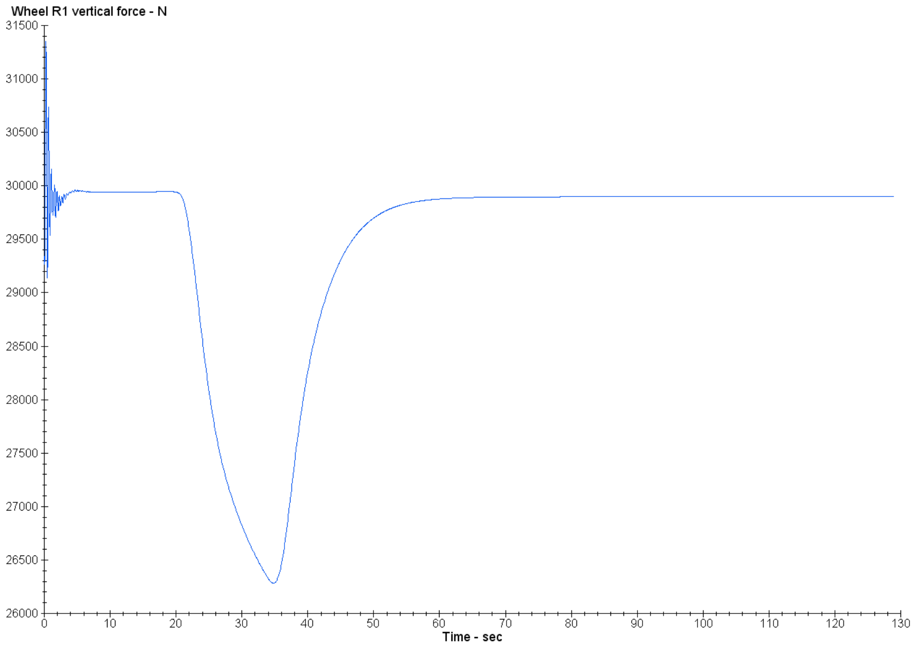



Figure C.49: Steer tyre friction demand performance result, right hand side vertical tyre force, for OECD 2 B-double

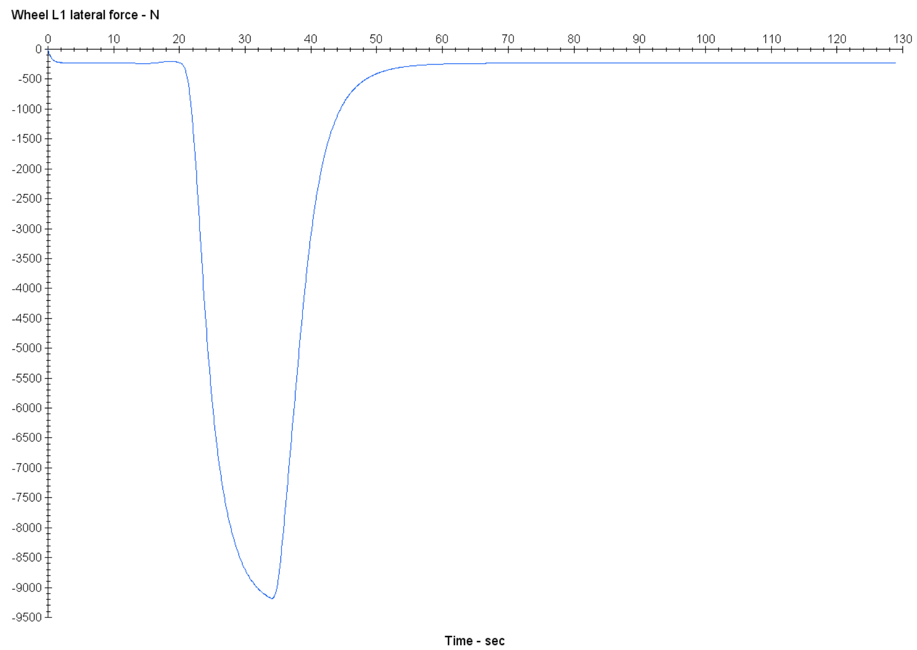



Figure C.50: Steer tyre friction demand performance result, left hand side lateral tyre force, for OECD 2 B-double

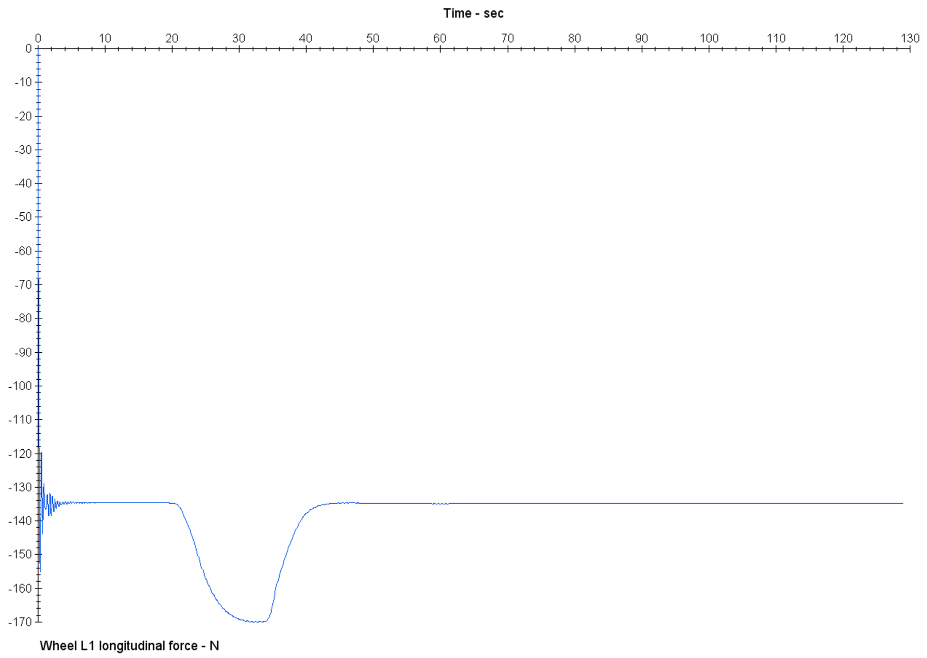



**Figure C.51: Steer tyre friction demand performance result, left hand side longitudinal tyre force, for OECD 2 B-double**

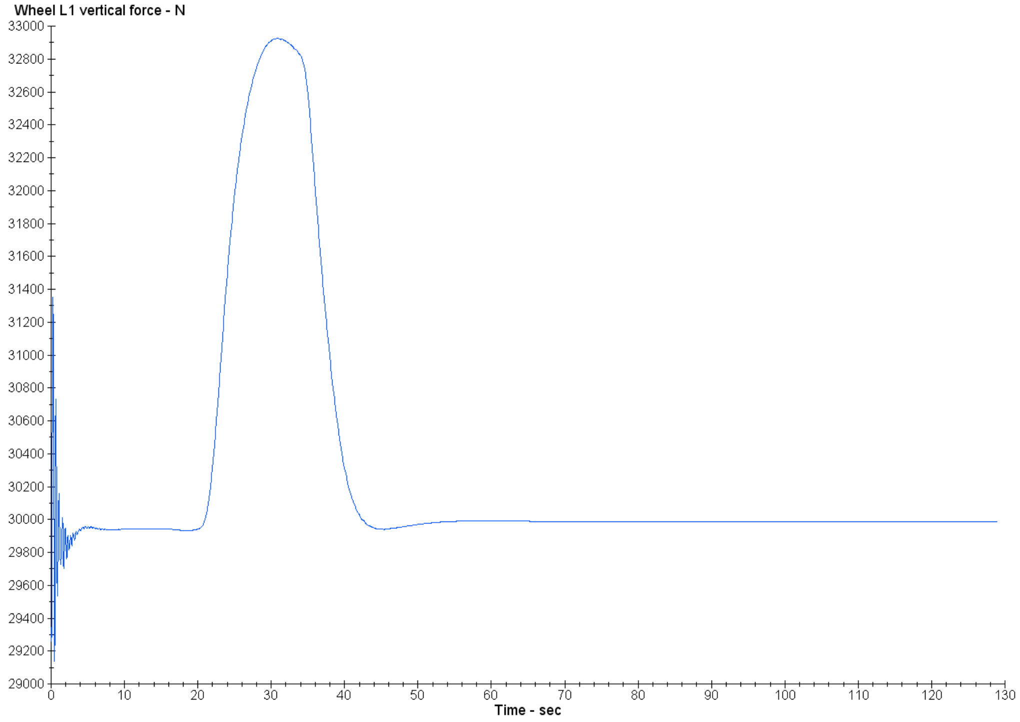



**Figure C.52: Steer tyre friction demand performance result, left hand side vertical tyre force, for OECD 2 B-double**

## C.3.6 Static Rollover Threshold

### C.3.6.1 Circular test

#### a. OECD 1

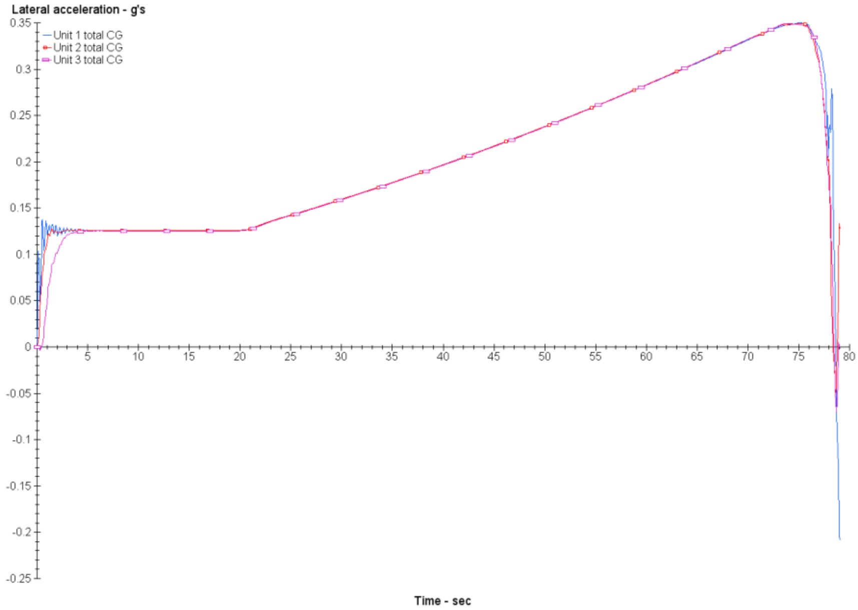



Figure C.53: Static rollover threshold, circular test, performance result for OECD 2 B-double

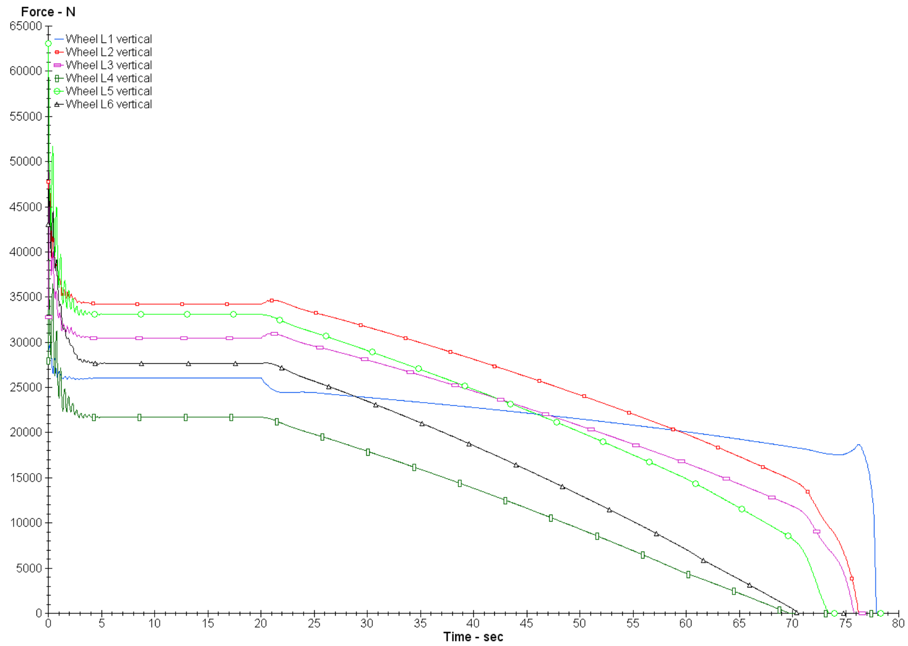



Figure C.54: Left hand side, vertical tyre forces, for OECD 2 B-double (circular test)

### D.3.6.2 Tilt table test

#### a. OECD 1

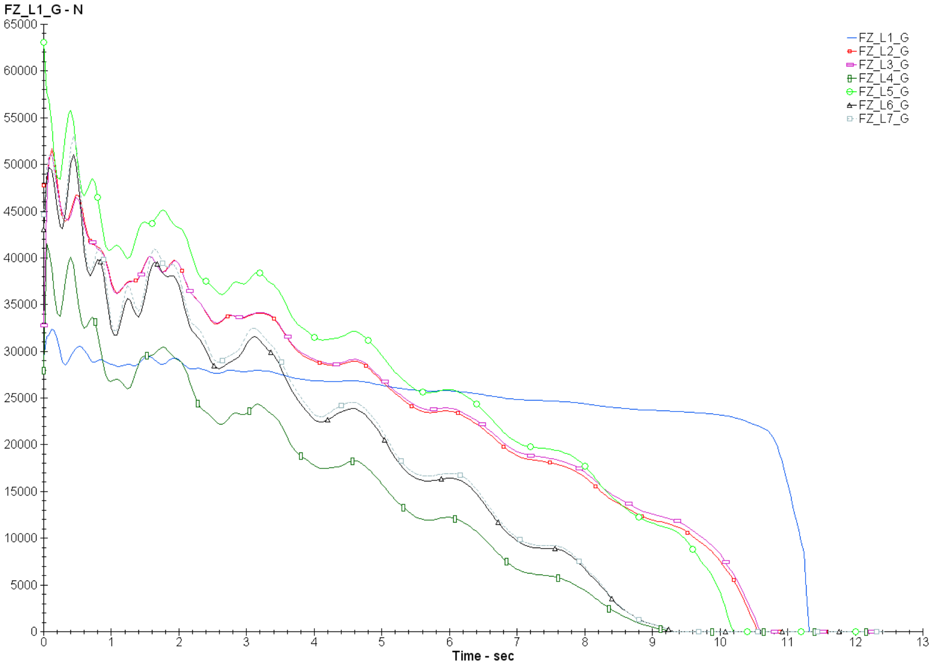



Figure C.55: Left hand side, vertical, tyre forces, for OECD 2 B-double (tilt test)

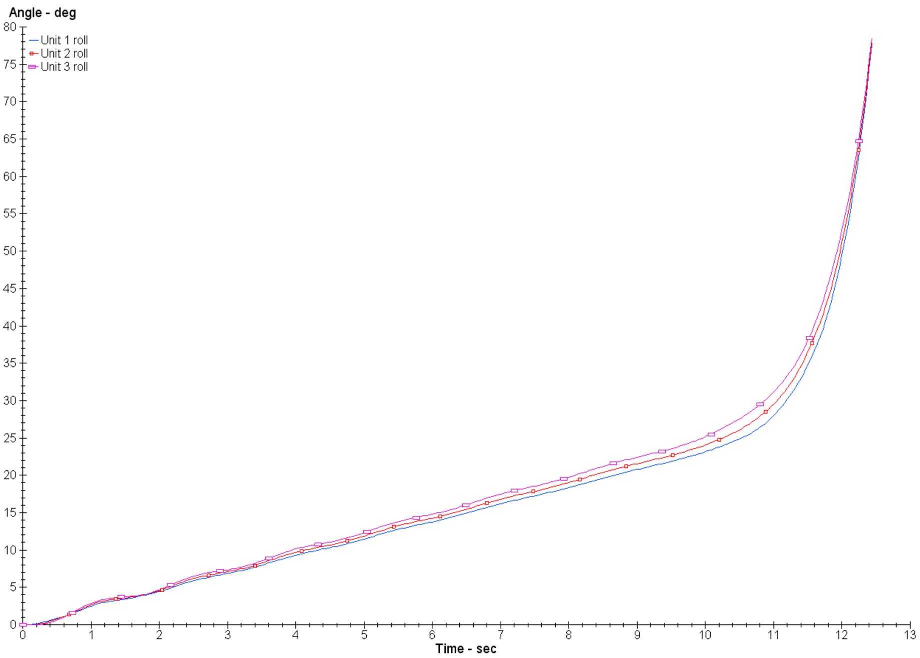



Figure C.56: Time versus angle relationship for OECD 2 B-double (tilt test)

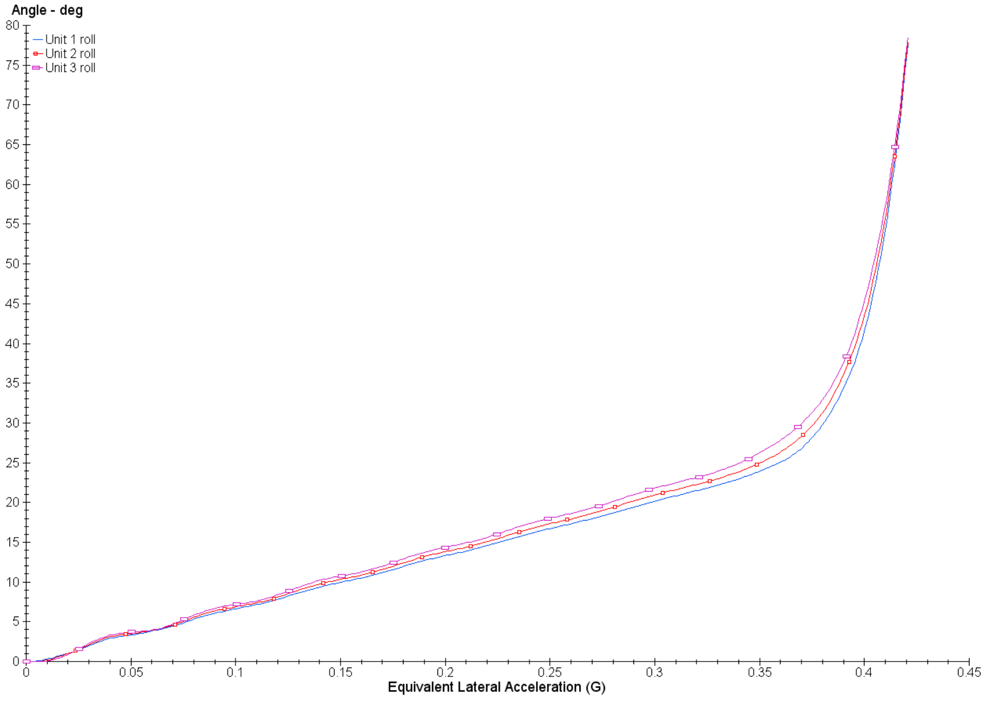



**Figure C.57: Angle versus equivalent roll angle for OECD 2 B-double (tilt test)**

## C.3.7 Rearward Amplification

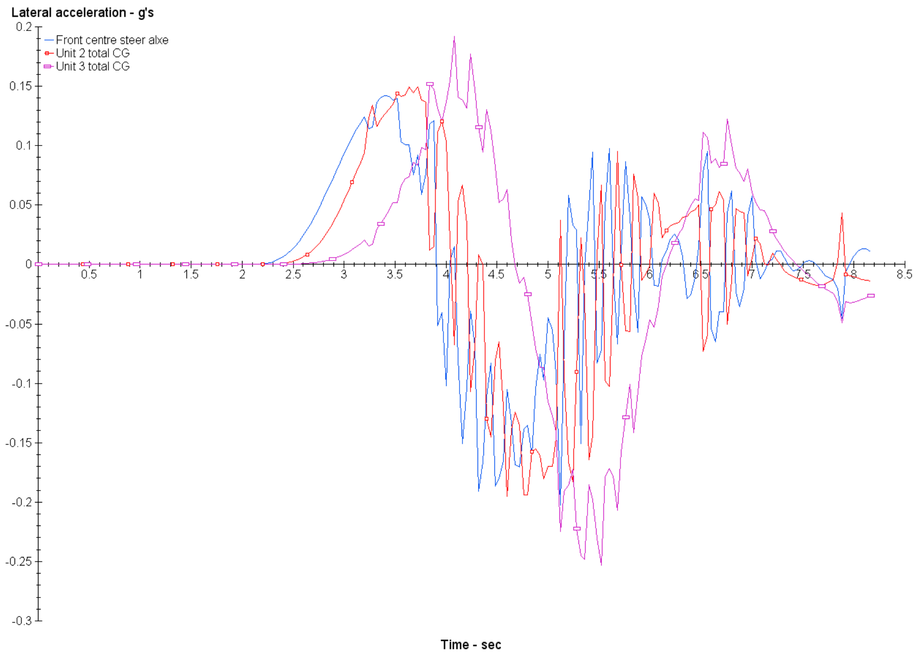



Figure C.58: Rearward amplification performance result for OECD 2 B-double

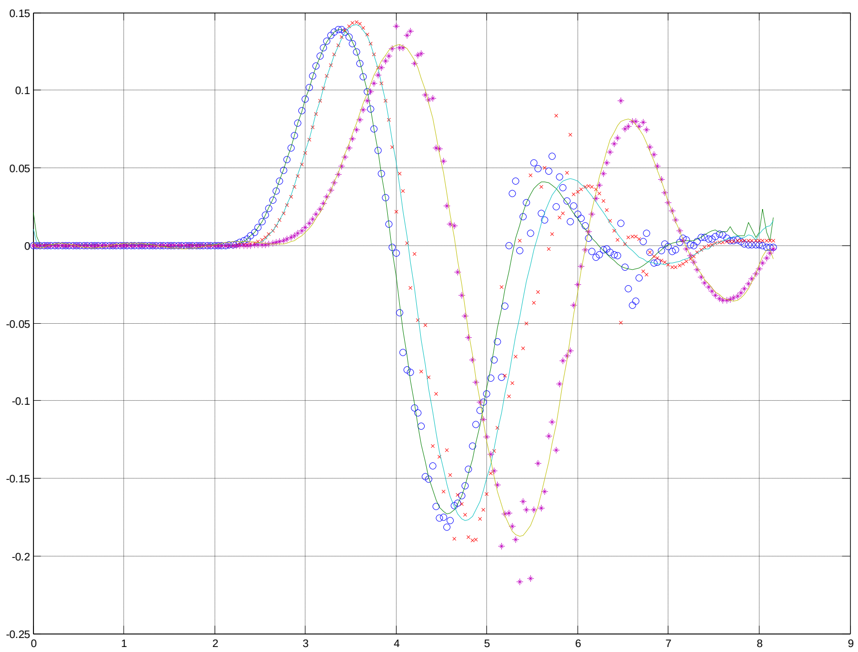


Figure C.59: Matlab polyfit results for OECD 2 B-double

## Matlab Polyfit Code

```
load C:\RA\OECD2\data.txt    %(this is just a paste of your xls file in
                              notepad)
x=data(:,1);                 %(this is 1st column of the txt file - time)
y=data(:,2);                 %(this is 2nd column of the txt file - acc)
%plot(x,y)                  %(plot of x and y NB % means invisible- you can
                              turn on/off as you please without getting
                              rid of the code)
p=polyfit(x,y,40);           %(this fits a n th order poly to your data)
xp=0:0.04:8.16               %(Generating time data starting at 0 ending at
8.16 in steps of 0.04)
yp=polyval(p,xp);            %(evaluating the polynomial at xp)

load C:\RA\OECD2\data1.txt
x1=data1(:,1);
y1=data1(:,2);
%plot (x1,y1)
p1=polyfit(x1,y1,25);
xp1=0:0.04:8.16
yp1=polyval(p1,xp1);

load C:\RA\OECD2\data2.txt
x2=data2(:,1);
y2=data2(:,2);
%plot (x1,y1)
p2=polyfit(x2,y2,15);
xp2=0:0.04:8.16
yp2=polyval(p2,xp2);

plot(x,y,'o',xp,yp,x1,y1,'x',xp1,yp1,x2,y2,'*',xp2,yp2)
grid on
yp_min = min(yp)
yp1_min = min(yp1)
yp2_min = min(yp2)
```

## C.3.8 High Speed Transient Off-tracking

### C.3.8.1 OECD 1

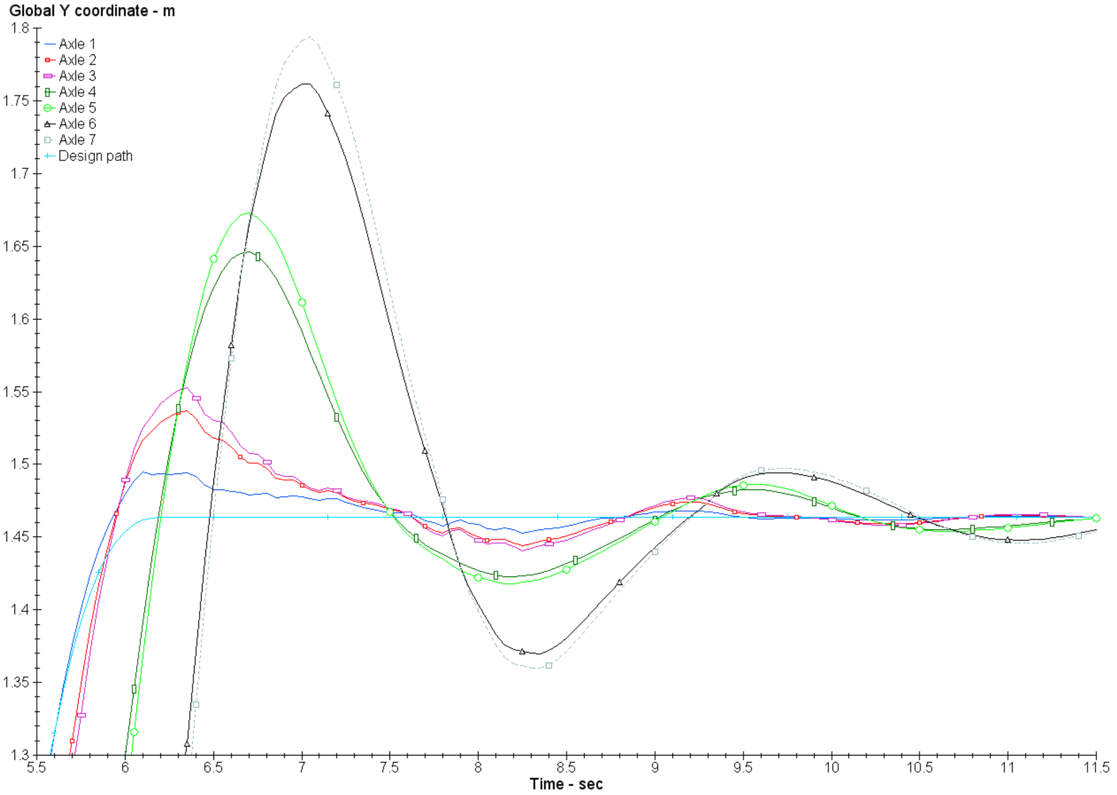



Figure C.60: High speed transient off-tracking result for OECD 2 B-double

## C.3.9 Yaw Damping

### C.3.9.1 OECD 2

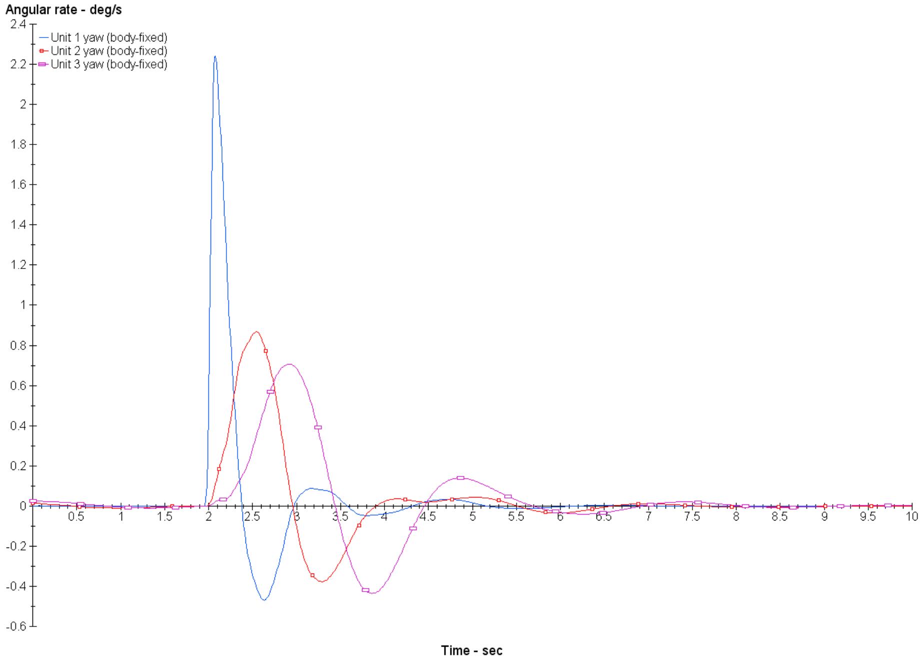



Figure C.61: Yaw damping, unit 1, 2 and 3, results for OECD 2 B-double

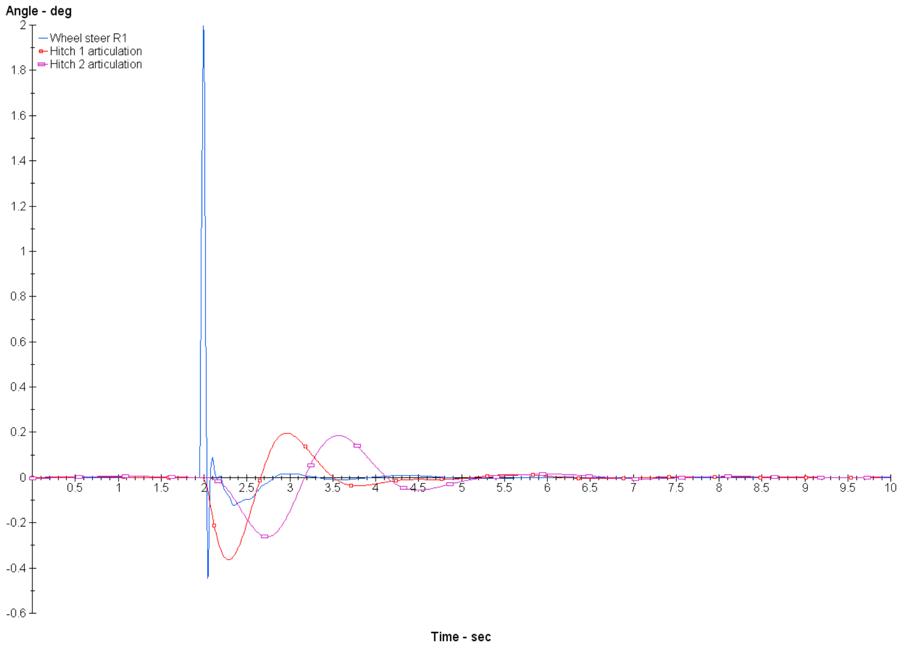
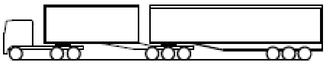
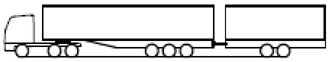
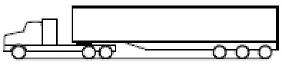
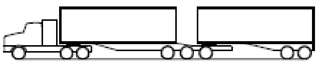
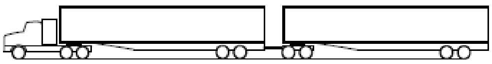
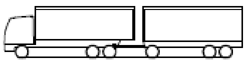
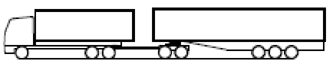


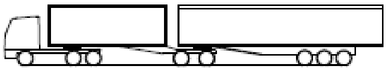
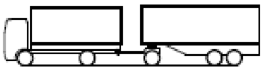
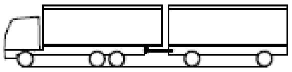
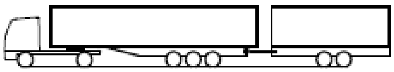
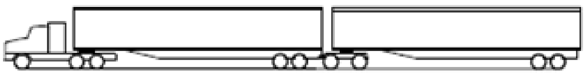
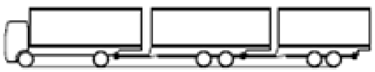
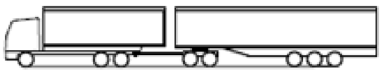
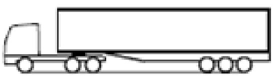
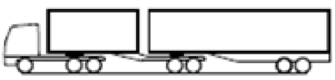

Figure C.62: Yaw damping, hitch 1 and hitch 2, result for OECD 2 B-double

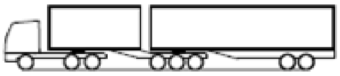
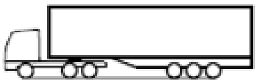
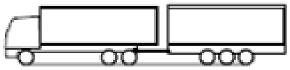
# Appendix D – Validation

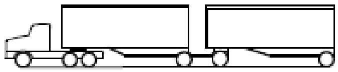
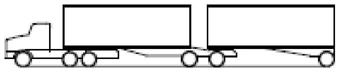
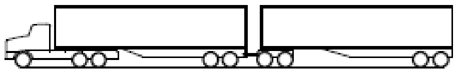
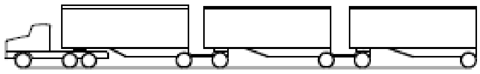
## D.1. List of the 39 international vehicles assessed in the OECD study

**Table D.1: List of vehicles as modelled during the OECD benchmarking study**

| Vehicle origin & identification number | GCM (kg) | Vehicle Classification | Schematic |
|---|---|---|---|
| Australia 1 (AU1) | 68,000 | Higher capacity |  |
| Australia 2 (AU2) | 90,500 | Very high capacity |  |
| Australia 3 (AU3) | 45,500 | Workhorse |  |
| Belgium 2 (BE2) | 39,000 | Workhorse<br>Base European vehicle |  |
| Belgium 3 (BE3) | 60,000 | Very high capacity<br>European modular vehicle |  |
| Canada 1 (CA1) | 39,500 | Workhorse |  |
| Canada 2 (CA2) | 46,500 | Workhorse |  |
| Canada 3 (CA3) | 62,500 | Higher capacity |  |
| Canada 4 (CA4) | 62,500 | Very high capacity |  |
| Denmark 1 (DK1) | 40,000<br>44,000 | Workhorse<br>Base European vehicle |  |
| Denmark 2 (DK2) | 48,000 | Workhorse<br>European modular vehicle |  |
| Denmark 3 (DK3) | 48,000 | Workhorse<br>Base European vehicle |  |
| Denmark 4 (DK4) | 60,000 | Higher capacity<br>European modular vehicle |  |

| | | | |
|---|---|---|---|
| Denmark 6 (DK6) | 60,000 | Higher capacity<br>European modular vehicle |  |
| France 1 (FR1) | 38,000 | Workhorse<br>Base European vehicle |  |
| France 2 (FR2) | 40,000 | Workhorse |  |
| Germany 2 (DE2) | 40,000 | Workhorse<br>European modular vehicle |  |
| Germany 4 (DE4) | 40,000 | Higher capacity<br>European modular vehicle |  |
| Mexico 1 (MX1) | 44,000 | Higher capacity |  |
| Mexico 2 (MX2) | 48,500 | Higher capacity |  |
| Mexico 3 (MX3) | 66,500 | Higher capacity |  |
| Mexico 4 (MX4) | 44,000 | Higher capacity |  |
| Netherlands 1 (NL1) | 50,000 | Very high capacity |  |
| Netherlands 2 (NL2) | 60,000 | Very high capacity<br>European modular vehicle |  |
| Netherlands 3 (NL3) | 60,000 | Very high capacity |  |
| South Africa 1 (ZA1) | 49,300 | Workhorse |  |
| South Africa 2 (ZA2) | 56,000 | Workhorse |  |

| | | | |
|---|---|---|---|
| South Africa 3 (ZA3) | 43,500 | Workhorse |  |
| South Africa 4 (ZA4) | 56,000 | Workhorse |  |
| United Kingdom 2 (UK2) | 44,000 | Workhorse<br>Base European vehicle |  |
| United Kingdom 4 (UK4) | 44,000 | Workhorse<br>European modular vehicle |  |
| United States 1 (US1) | 36,350 | Workhorse |  |
| United States 2 (US2) | 41,900 | Workhorse |  |
| United States 3 (US3) | 44,100 | Workhorse |  |

| | | | |
|---|---|---|---|
| United States 4 (US4) | 36,360 | Higher capacity |  |
| United States 5 (US5) | 36,360 | Higher capacity |  |
| United States 6 (US6) | 57,040 | Very high capacity |  |
| United States 7 (US7) | 53,752 | Very high capacity |  |

## D.2. Vehicle parameters for the four South African vehicles assessed

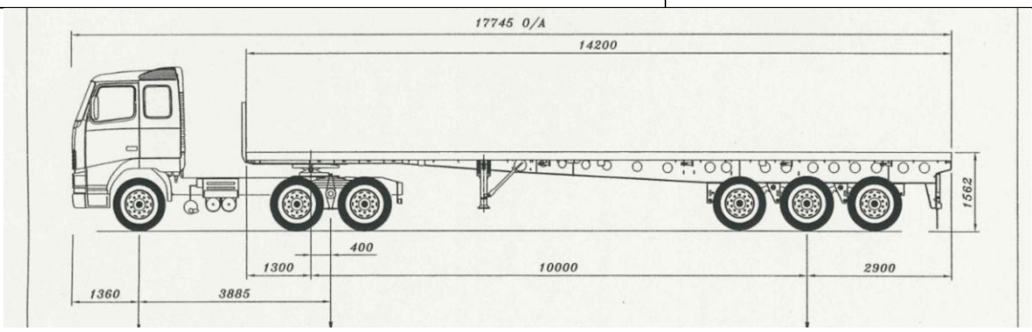
# Vehicle Performance Benchmark Task Force

## List of vehicle parameters: South Africa

| Parameter | | Vehicle 1 | Vehicle 2 | Vehicle 3 | Vehicle 4 |
|---|---|---|---|---|---|
| 1. Line drawing with axle loads (laden and unladen) and dimensions | | D104-187 | D112-165 | D102-58 | D214-05 |
| 2. Self steering | | N/A | N/A | N/A | N/A |
| 3. Axle widths (outer edge of tyre treads) | | | | | |
| Steering | | 2,350 | 2,350 | 2,350 | 2,350 |
| Drive axle unit | | 2,500 | 2,500 | 2,500 | 2,500 |
| Non-steering | | 2,560 | 2,560 | 2,560 | 2,560 |
| 4. Tyre sizes | | | | | |
| Steering | | 315/80R22.5 | 385/65R22.5 | 385/65R22.5 | 385/65R22.5 |
| Drive axle unit | | 315/80R22.5 | 315/80R22.5 | 315/80R22.5 | 315/80R22.5 |
| Semi-trailer | | 12R22.5 | 12R22.5 | 12R22.5 | 12R22.5 |
| Tridem semi-trailer | | | | | 385/65R22.5 |
| 5. Hitch height | | N/A | N/A | N/A | N/A |
| Fifth wheel height | | 1,320 | 1,320 | 1,320 | 1,320 |
| 6. Tare weight centre of mass height | | | | | |
| Truck tractor | | Est. range from 1 000 to 1 200 | | | |
| Semi-trailer 1 | | Usually approx. 1 300 | | | |
| Semi-trailer 2 | | Usually approx. 1 300 | | | |
| 7. Inside cargo box dimensions | Length | 14,200 | 6,100<br>12,200 | 12,200 | 6,100<br>12,200 |
| | Width | 2,600 | 2,600 | 2,600 | 2,600 |
| | Height | 2,658 | 2,683 | 2,683 | 2,655 |
| 8. Height of cargo floor above ground | | 1,562 | 1,537 | 1,537 | 1,565 |

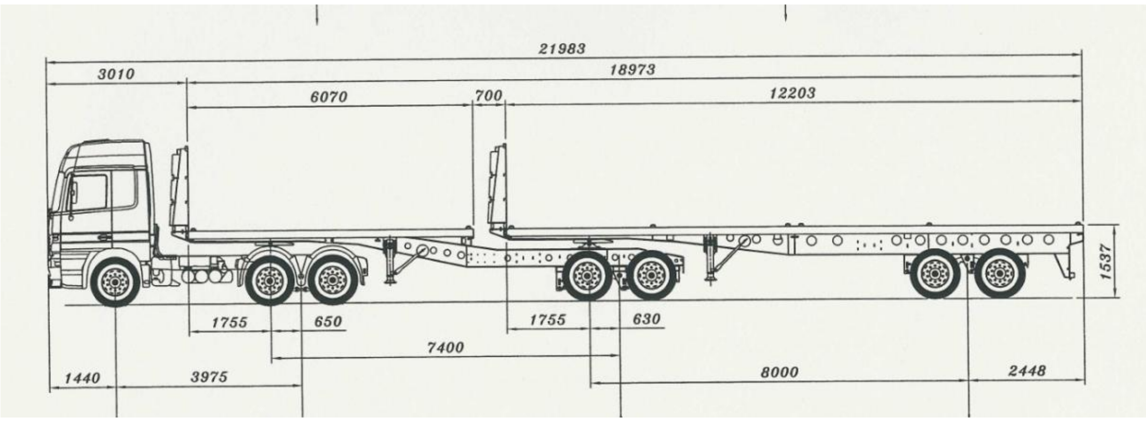
South African Truck Description No. 1

| | | |
|---|---|---|
| 1. | Vehicle classification (*tick as appropriate*):<br>Workhorse vehicle ☒ higher capacity vehicle ❑ very high capacity vehicle ❑ | |
| 2. | How many trailers does this vehicle have? | One |
| 3. | How many axles does this vehicle have? | Six |
| 4. | Does it generally operate with dual tyre axles or wide-based single tyre axles? | Drive axle unit: Dual tyres<br>Tridem: Dual tyres or wide-based single tyres |
| 5. | How many points of articulation does this vehicle have (*e.g.* A tractor semi-trailer has one)? | One |
| 6. | What is the maximum legal gross vehicle weight of this vehicle with respect to size and weight regulation? | 49 300 kg |
| 7. | What is the normal axle load for each axle (beginning with the front axle) at the legal gross vehicle weight. Please note that the GVW in national operations may differ from the GVW in international transport. | 7 300 (depends on make of truck tractor); 9 000; 9 000; 8 000; 8 000; 8 000 kg. |
| 8 | What is the overall length of the vehicle ? | 17.7 m (up to 18.5 m) |
| 9 | Where does this vehicle generally operate ( 1. Within a province, state or territory, nationally, internationally? 2. On all truck routes, under special permit, on specified roads?) | Nationally & internationally. |



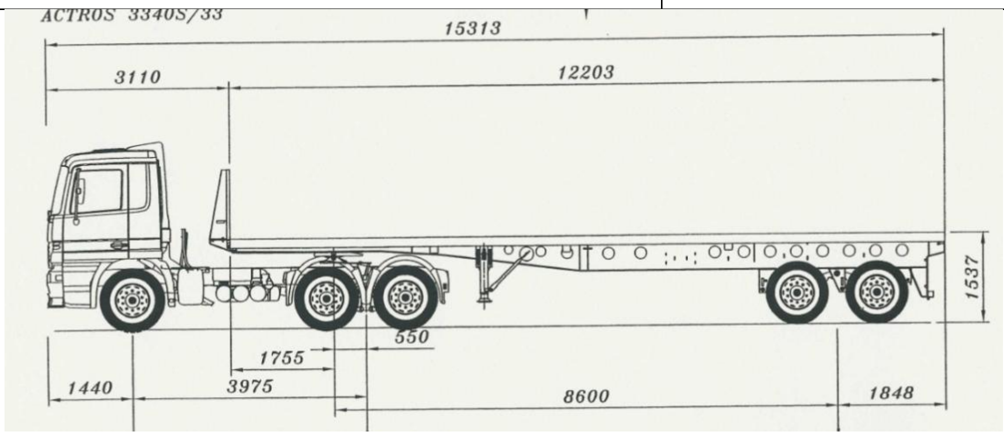
South African Truck Description No. 2

| | | |
|---|---|---|
| 1. | Vehicle classification (tick as appropriate): | |
| | Workhorse vehicle ☒ higher capacity vehicle ❑ very high capacity vehicle ❑ | |
| 2. | How many trailers does this vehicle have? | Two |
| 3. | How many axles does this vehicle have? | Seven |
| 4. | Does it generally operate with dual tyre axles or wide-based single tyre axles? | Mostly dual tyres |
| 5. | How many points of articulation does this vehicle have (*e.g.* A tractor semi-trailer has one)? | Two |
| 6. | What is the maximum legal gross vehicle weight of this vehicle with respect to size and weight regulation? | 56 000 kg |
| 7. | What is the normal axle load for each axle (beginning with the front axle) at the legal gross vehicle weight. Please note that the GVW in national operations may differ from the GVW in international transport. | 6 500 (depends on make of truck tractor); 8 250; 8 250; 8 250; 8 250; 8 250; 8 250 kg. |
| 8 | What is the overall length of the vehicle ? | 22 m |
| 9 | Where does this vehicle generally operate ( 1. Within a province, state or territory, nationally, internationally? 2. On all truck routes, under special permit, on specified roads?) | Nationally & internationally. |



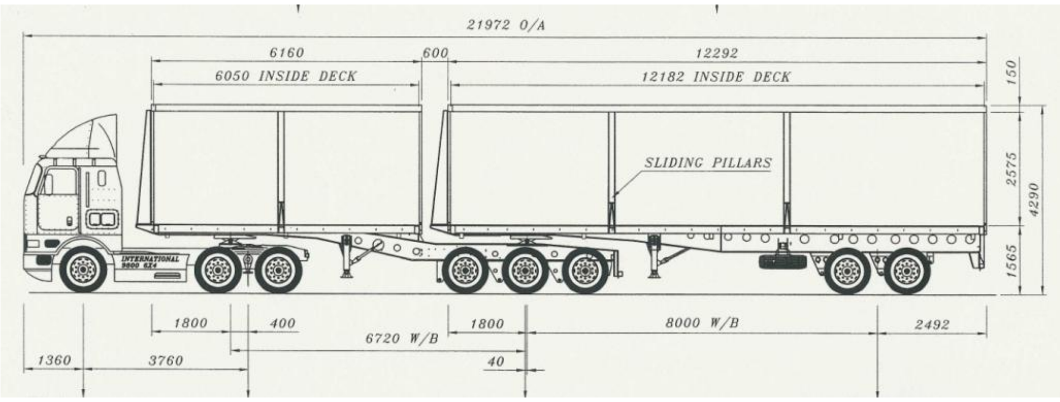
South African Truck Description No. 3

| | | |
|---|---|---|
| 1. | Vehicle classification (tick as appropriate): | |
| | Workhorse vehicle ☒ higher capacity vehicle ❑ very high capacity vehicle ❑ | |
| 2. | How many trailers does this vehicle have? | One |
| 3. | How many axles does this vehicle have? | Five |
| 4. | Does it generally operate with dual tyre axles or wide-based single tyre axles? | Mostly dual tyres |
| 5. | How many points of articulation does this vehicle have (*e.g.* A tractor semi-trailer has one)? | One |
| 6. | What is the maximum legal gross vehicle weight of this vehicle with respect to size and weight regulation? | 43 500 kg |
| 7. | What is the normal axle load for each axle (beginning with the front axle) at the legal gross vehicle weight. Please note that the GVW in national operations may differ from the GVW in international transport. | 7 500 (depends on make of truck tractor); 9 000; 9 000; 9 000; 9 000 kg. |
| 8 | What is the overall length of the vehicle ? | 15.3 m (up to 18.5) |
| 9 | Where does this vehicle generally operate ( 1. Within a province, state or territory, nationally, internationally? 2. On all truck routes, under special permit, on specified roads?) | Nationally & internationally. |



South African Truck Description No.4

| | | |
|---|---|---|
| 1. | Vehicle classification (tick as appropriate): | |
| | Workhorse vehicle ☒ higher capacity vehicle ❑ | very high capacity vehicle ❑ |
| 2. | How many trailers does this vehicle have? | Two |
| 3. | How many axles does this vehicle have? | Eight |
| 4. | Does it generally operate with dual tyre axles or wide-based single tyre axles? | Mostly dual |
| 5. | How many points of articulation does this vehicle have (*e.g.* A tractor semi-trailer has one)? | Two |
| 6. | What is the maximum legal gross vehicle weight of this vehicle with respect to size and weight regulation? | 56 000 kg |
| 7. | What is the normal axle load for each axle (beginning with the front axle) at the legal gross vehicle weight. Please note that the GVW in national operations may differ from the GVW in international transport. | 6 500 (depends on make of truck tractor); 7 500; 7 500; 6 500; 6 500; 6 500; 7 500; 7 500 kg. |
| 8 | What is the overall length of the vehicle ? | 22 m |
| 9 | Where does this vehicle generally operate (1. within a province, state or territory, nationally, internationally? 2. On all truck routes, under special permit, on specified roads?) | Nationally & internationally. |



## D.3. OECD resultant plots of the 39 internationally assessed vehicles

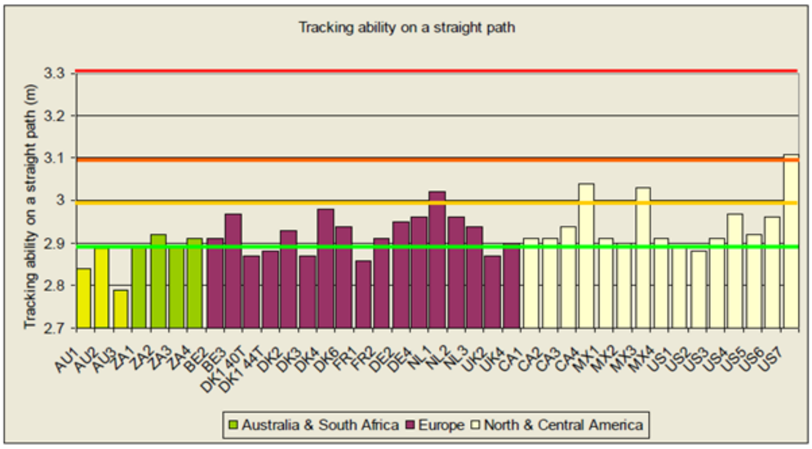



**Figure D.1: Tracking ability on a straight path performance by region**

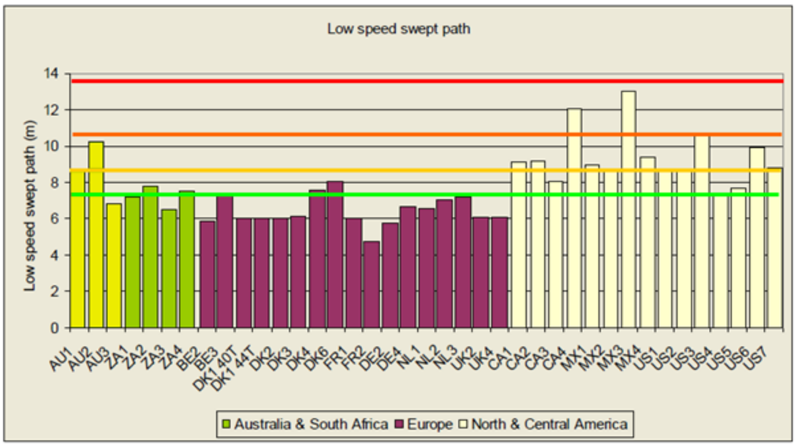



**Figure D.2: Low speed swept path performance by region**

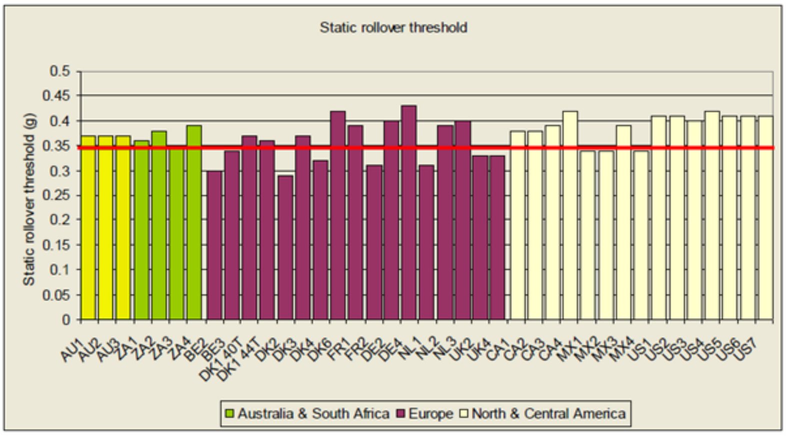



Figure D.3: Static rollover threshold performance by region

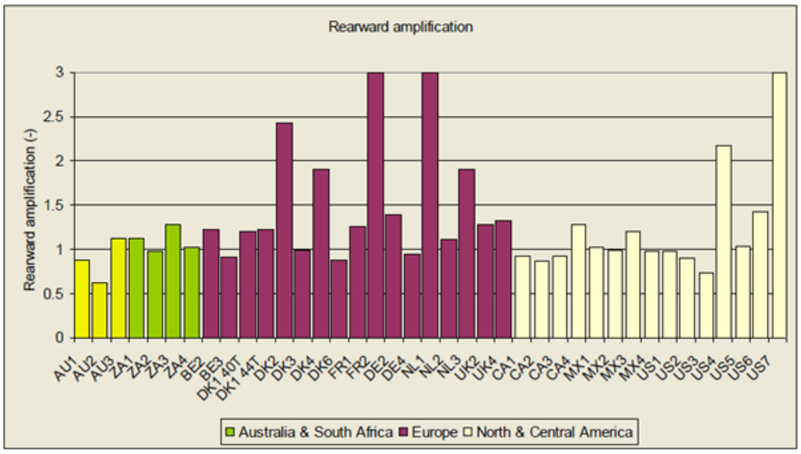



Figure D.4: Rearward amplification performance by region

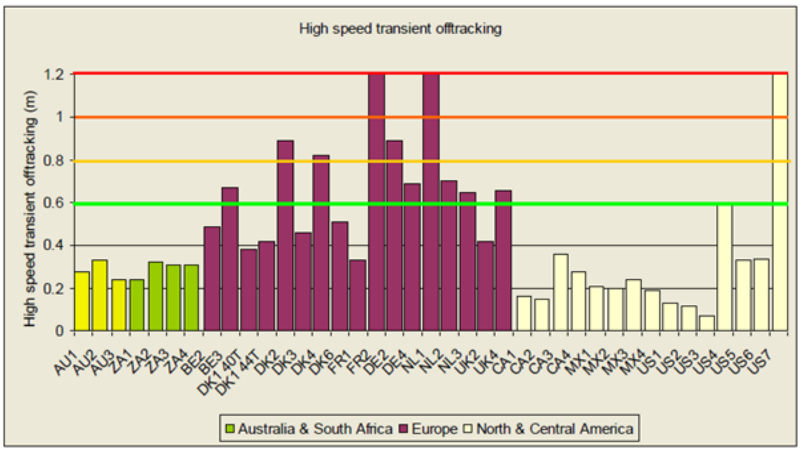



Figure D.5: High speed transient off-tracking performance by region