

COMPARATIVE PERFORMANCE OF SEMI-TRAILER STEERING SYSTEMS

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ABSTRACT

A yaw-roll model of a tractor semi-trailer is used to compare the performance of various trailer axle steering systems fitted to a standard length semi-trailer. The model is highly adaptable and incorporates effects such as non-linear tyres and lateral load transfer, which are important in heavy vehicle handling. Within the model a simple driver sub-model is used to steer the vehicle down the desired path at a desired velocity.

Results from simulations of the various steering systems are presented. Common manoeuvres are performed including low-speed corners, low-speed roundabouts, high-speed lane changes and high-speed circles. Performance is characterised by a set of performance measures, which include the vehicle's ability to follow the desired path, its handling characteristics, lateral tyre forces, load transfer and rearward amplification. Reasons for the differences in performance of the various systems are discussed and recommendations are made to improve the performance where necessary.

1. INTRODUCTION

In recent years a number of systems have been developed which allow the rear axles of semi-trailers to be steered. By steering the rear axles such systems aim to improve the low speed manoeuvrability of the vehicle as well as reduce tyre scrub. This is important for transporting goods in urban areas where vehicles need to negotiate sharp corners and small roundabouts.

Current semi-trailer steering systems can be grouped into one of three types; namely self-steering systems, command steer systems and pivotal bogie systems. Each type uses a different strategy to determine the angles of the axles. Self-steering systems steer the trailer axles in relation to the lateral tyre forces, command steer systems steer in relation to the articulation angle between the tractor and trailer and pivotal bogie systems steer in relation to angle of the rear bogie assembly. Hence although all can be classified as semi-trailer steering systems they are fundamentally different. Details of each type of steering system are presented along with the governing equations in Section 2.

To date, only a limited number of papers have been published on the performance of semi-trailer steering systems. In the 1980's LeBlanc, El-Gindy and Woodrooffe undertook research into self-steering axles and their use on rigid body trucks and C-type dollies [1] and [2]. Command steer systems have been studied by Jones and Wright [3] and Sankar, Rakhja and Piche [4]. Most recently, the performance of pivotal bogie systems have been investigated by Henderson [5] and Prem [6]. The studies above have generally concentrated on one particular type of steering system and how it performs under certain circumstances.

The aim of this paper is to compare the performance of the three types of semi-trailer steering systems. The comparison is made using an adaptable simulation model, capable of modelling different steering systems fitted to the same standard length tri-axle semi-trailer. The model and its main components are described in detail in Section 3.

To quantify the high and low speed performance of the steering systems a comprehensive set of performance measures were used. The measures are introduced in Section 4 and tabulated in Table 1. The measures are mainly based on those proposed for Performance Based Standards (PBS) legislation in Australia.

The results from the comparative study are presented and discussed Section 5. The discussion includes how the results from the steering semi-trailers compare to the fixed baseline trailer as well as how the results compare to the proposed PBS limits. Reasons for differences in performance are given.

Finally, conclusions are drawn in Section 6 regarding how the various semi-trailer steering systems compare along with their potential advantages and limitations. Future work required in this area is also described, including validation of the model and comparison of the simulation results with field tests data.

2. DESCRIPTION OF SEMI-TRAILER STEERING SYSTEMS

2.1. Self-Steering Systems

Self-steering systems are the most widely used type of semi-trailer steering system, mainly due to their simplicity and low cost. In typical systems the rearmost fixed axle in the tri-axle group is replaced with a self-steering axle. The most popular type of self-steering axle is an “automotive style” self-steering axle. This is similar in design to a conventional steering axle but with a positive trail. Instead of being controlled by a steering box, the steering of the axle is typically controlled by a preloaded spring and damper attached to the trailing arm. The purpose of the spring-damper is to help centre the axle, offset the effects of unbalanced braking and provide lateral forces at low angles of steer necessary to prevent instability. Mechanisms are usually incorporated into the designs to lock the axle when travelling in reverse.

When a semi-trailer fitted with a self-steering axle transverses a low-speed corner the wheels on the self-steering axle align with the direction of travel. This reduces the lateral forces acting on the self-steering axle. Lateral forces on the other axles are also reduced because the self-steering axle decreases the magnitude of any “locked-in” lateral forces associated with the axle group. Hence the self-steering axle is generally beneficial to tyre wear on all axles.

A self-steering axle also improves the low-speed cornering and manoeuvrability of a vehicle. Since the self-steering axle generates little lateral force when cornering it can be neglected. This reduces the effective wheelbase of the semi-trailer to that of an equivalent fixed tandem semi-trailer (i.e. a semi-trailer without the self-steering axle).

The degree to which a self-steering axle steers is determined by the balance of moments about its king pins. For normal running it can be assumed that longitudinal tyre forces are negligible and thus the tyres are subjected to lateral forces only. LeBlanc et al. [2] showed that self-steering axles generally have a non-linear relationship between the steer angle and the lateral tyre force, as shown in Figure 1. This relationship can be represented by the following equation;

$$\delta = \begin{cases} \frac{F_y}{K_1} & , |F_y| < F_{yc} \\ \frac{F_y}{K_2} + \frac{F_{yc}}{K_2} \left(1 - \frac{K_2}{K_1}\right) & , |F_y| > F_{yc} \end{cases} \quad \dots(1)$$

where

$$\begin{aligned} \delta &= \text{Steering angle [rad]} \\ F_y &= \text{Total lateral tyre force acting on axle [N]} \\ F_{yc} &= \text{Centring force [N]} \\ K_1, K_2 &= \text{Axle Cornering Stiffnesses [N/rad]} \end{aligned}$$

Hence a self-steering axle is relatively stiff in yaw at low steer angles but becomes relatively compliant at larger steer angles.

2.2 Command Steer Systems

In a typical command steer system the rear two fixed axles in the tri-axle semi-trailer group are replaced with conventional style steering axles. These axles are made to steer in relation to the articulation angle between the tractor and semi-trailer. The articulation angle is sensed using either electronics or a special ballrace attached to the trailer side of the fifth wheel. This angle then transmitted to the steering axles electronically or via mechanical or hydraulic linkages.

The equations that govern the behaviour of a command steer system can be obtained by considering the vehicle in a low speed turn, as shown in Figure 2. In order to have no sideslip the rearmost wheels need to be steered so that their normals pass through the same centre as the front trailer axle. This eliminates the lateral tyre forces on the rear axles and eliminates locked in forces in the group, both of which are beneficial to tyre wear.

The following relationships between the articulation angle and steer angles can be obtained from the geometry of the system:

$$\begin{aligned}\delta_m &= \tan^{-1}\left(\frac{d \cdot \sin \Gamma}{b \cdot \cos \Gamma - a}\right) \\ \delta_r &= \tan^{-1}(2 \cdot \tan(\delta_m))\end{aligned}\quad \dots(2)$$

where;

$$\begin{aligned}\Gamma &= \text{Articulation angle [rad]} \\ \delta_m, \delta_r &= \text{Middle axle and rear axle steer angles [rad]} \\ a &= 5^{\text{th}} \text{ wheel to rear tractor axle distance [m]} \\ b &= 5^{\text{th}} \text{ wheel pin to front trailer axle distance [m]} \\ d &= \text{Spacing between trailer axles [m]}\end{aligned}$$

Note that the relationships are non-linear. However, if small angles are assumed they can be linearized giving the following relationships:

$$\begin{aligned}\delta_m &= \frac{d}{b-a} \times \Gamma \\ \delta_r &= 2 \cdot \delta_m\end{aligned}\quad \dots(3)$$

Thus the steering angles are directly proportional to the articulation angle. The small angle approximation is valid for most vehicle operating conditions. However, on very tight radii turns the angles become significant and thus a system designed with a linear relationship does not quite achieve the desired steering angle. This results in small lateral forces being applied to the steering axles and a slight decrease in the cornering radius.

Like the self-steering system, the command steer system also effectively reduces the wheelbase of the semi-trailer. Since both rear axles generate no lateral forces in a turn they can be neglected and the wheelbase of the semi-trailer is reduced to that of an equivalent fixed single axle semi-trailer, as indicated on Figure 2.

2.3 Pivotal Bogie Systems

Pivotal bogie systems have been used for some time in the heavy haulage industry as a means of steering extremely long semi-trailers. It is only recently however that this principle has been applied to normal length semi-trailers.

In a pivotal bogie system a ballrace-mounted tri-axle bogie assembly replaces the fixed-axle trailer group. The ballrace allows this assembly to yaw freely relative to the trailer chassis. The bogie consists of a fixed front axle and two steered rear axles, which steer in relation to the angle between the bogie and the trailer chassis. Thus the axles progressively steer as the angle between the bogie and the trailer increases, bringing the bogie back inline with the trailer chassis.

The governing equations for a pivotal bogie system can be derived in a similar manner to the command steer equations. Figure 3 shows a simplified representation of a pivotal bogie system in a curve. To minimise the lateral tyre forces and turning radius the rearmost wheels must again steer such that their normals pass through the instantaneous turning centre. The geometry dictates the following relationships:

$$\delta_m = \tan^{-1} \left(\frac{2d \cdot \sin \beta \cdot \cos \beta}{b \cdot \cos \beta - a + 2(d-c) \sin^2 \beta} \right) \quad \dots(4)$$

$$\delta_r = \tan^{-1} (2 \cdot \tan(\delta_m))$$

where;

β = Bogie angle [rad]

c = Trailer centre axle to ballrace distance [m]

Again, the relationships are non-linear due to the bogie articulation and steer angles. Assuming small angles results in the following linearized equations:

$$\delta_m = \frac{2d}{b-a} \times \beta \quad \dots(5)$$

$$\delta_r = 2 \cdot \delta_m$$

Comparison of equation (5) with equation (3) reveals that the relationship is the same as a command steer system but with half the gain. Thus the small angle approximation and fifth wheel offset associated with the command steer system produce similar errors in a pivotal bogie system.

By steering all of the axles in the tri group a pivotal bogie system is able to greatly reduce the effective wheelbase of the semi-trailer. For such a system the wheelbase is approximately half of the distance from the fifth wheel to the bogie ballrace, as indicated on Figure 3. Thus the effective wheelbase is approximately half that of a standard fixed tri-axle semi-trailer.

3. SIMULATION MODEL

In order to compare the performance of the various steering systems a roll-yaw model was developed in Simulink. The model consists of a single drive tractor unit coupled to a triaxle semi-trailer. The equations of motion for the system are similar to those used in the UMTRI yaw-roll model [7]. Parameters for the model were chosen to represent a typical European articulated tanker vehicle that complies with current UK legislation [8]. Critical weights and dimensions are shown on Figure 4.

The simulation model has yet to be fully validated although preliminary results from the model were compared with those from other yaw-roll models. Such models have been found to quite accurately represent real vehicles in basic handling manoeuvres. Full validation of the model will be achieved in the near future by comparing simulated results to those from field tests. This is the subject of ongoing work.

The main difference between this model and a conventional articulated vehicle model is the way in which the semi-trailer axle group is modelled. In most vehicle models the axles are non-steerable and connected directly to the trailer chassis. However in this model the axles are steerable and connected to a ballrace mounted bogie assembly.

The steering input to the trailer axles can be varied to suit the type of steering system. For a fixed axle system the steering input is zero and the axles are locked. For a self-steering system the input is the axle's lateral force. For a command steer system the input is the semi-trailer articulation angle. Finally, for a pivotal bogie system the input is the bogie articulation angle.

In a similar manner the bogie can be locked or released to suit the type of steering system. For fixed, self-steering and command steer systems the bogie is locked to the trailer chassis. For a pivotal bogie system the bogie is

released. Thus by choosing the steering input and locking or releasing the bogie the steering systems investigated in this paper can be modelled.

A simplified block diagram of the model is given in Figure 5. Each of the main elements are discussed in the sections below.

3.1 Driver Sub-Model

The driver sub-model determines the front axle steering angle required to make the vehicle move along the desired path. The sub-model uses a preview controller that steers in proportion to the lateral offset between the desired path and the vehicle's current heading at a user defined preview distance from the front axle L , as shown in Figure 6. Thus;

$$\delta_1 = K_1 \cdot x \quad \dots(6)$$

where

$$\begin{aligned} \delta_1 &= \text{Front axle steer angle [rad]} \\ x &= \text{Lateral offset [m]} \\ K_1 &= \text{User selectable gain} \end{aligned}$$

The desired path can be selected from a list of standard manoeuvres or input directly by the user. Standard manoeuvres include constant radius circles, 90-degree bends and SAE lane changes [9].

3.2 Semi-Trailer Steer Angle Sub-Model

The semi-trailer steer angle sub-model outputs the steer angle of each of the steered semi-trailer axles. The block essentially applies equation (1), (2) or (4) depending on which type of steering is selected; self-steer, command steer or pivotal bogie.

3.3 Static Vertical Load Sub-Model

The static vertical load sub-model is used to determine the static vertical load on each axle based on the mass and geometry information for each body input by the user. In the model it is assumed that there is perfect loadsharing between the triaxle trailer group so that each axle has the same static vertical load.

3.4 Lateral Load Transfer Sub-Model

The lateral load transfer sub-model calculates the additional load transferred to each wheel when the vehicle is subjected to a lateral acceleration. The model initially determines the roll angle of the tractor and trailer body based on the roll stiffness of each axle and the compliance of the chassis' and fifth wheel. The lateral load transfer for each axle is then computed. The variation in roll stiffness due to the tractor and bogie articulation angle is taken into account.

The results from the lateral load transfer sub-model are added to the static vertical loads to yield the total vertical load on each wheel of the vehicle.

3.5 Tyre Force Sub-Model

The tyre force sub-model determines the lateral force generated by each wheel based on the vertical load and slip angle. The model is based on an empirical "brush" model with a parabolic pressure distribution and is similar to the models used by Billing [10] and Radt and Milliken [11]. The model, however, takes into account the non-linear relationship between vertical tyre load and lateral stiffness as described by Fancher et al. in [12]. In the model the lateral tyre force is described by the equation;

$$F_y = \begin{cases} F_z \cdot \bar{C} \cdot \delta - \frac{F_z \cdot \bar{C}^2}{3\mu} \delta^2 + \frac{F_z \cdot \bar{C}^3}{27\mu^2} \delta^3 & , \quad \delta < \frac{3\mu}{\bar{C}} \\ \mu \cdot F_z & , \quad \text{elsewhere} \end{cases} \quad \dots(7)$$

where

$$\begin{aligned}F_y &= \text{Lateral tyre force [N]} \\F_z &= \text{Vertical tyre force [N]} \\\mu &= \text{Coefficient of friction} \\C_1 &= \text{Tyre cornering coefficient [rad}^{-1}\text{]} \\C_2 &= \text{Tyre curvature coefficient [(N.rad)}^{-1}\text{]} \\\bar{C} &= C_1 + C_2 F_z \text{ [rad}^{-1}\text{]}\end{aligned}$$

3.6 Vehicle Dynamics Sub-Model

The vehicle dynamics sub-model calculates the displacements, velocities and accelerations of the vehicle's constituent bodies, namely the tractor, semi-trailer and bogie. Application of Newton's Laws of motion to the bodies yields nine differential equations. A further four differential equations are obtained by considering the force balance at the fifth wheel and bogie ballrace. These equations are solved simultaneously to determine the body accelerations and coupling forces. Body displacements and velocities are then calculated by integration of the body accelerations.

Four test vehicles were constructed using the model. The first represented a typical articulated vehicle with fixed semi-trailer axles, which was used as a baseline for comparison purposes. The other test vehicles contained one of each of the semi-trailer steering systems in question.

4. PERFORMANCE MEASURES

In order to compare the performance of the various steering systems a set of performance measures was defined. These measures were used to judge how well each semi-trailer steering system performed relative to a standard fixed semi-trailer and the other steering systems.

A comprehensive set of performance measures for heavy vehicles was compiled by Australia's National Road Transport Commission (NRTC) as part of their performance based standards project [13]. These have recently been reviewed and updated following a review of the characteristics of Australia's heavy vehicle fleet [14]. It was decided to use the NRTC measures as a basis for this study. Only measures deemed relevant to trailer steering systems were selected.

In addition to the NRTC measures, two measures that are applicable to tractor/semi-trailer units operating in the UK were defined. The measures were the maximum swept path width and peak trailer lateral tyre force associated with a vehicle travelling in a small radius circle at low speed. These measures were deemed important for vehicles that have to manoeuvre around tight roundabouts.

The complete set of performance measures used in this study is shown in Table 1.

5. RESULTS

For each test vehicle, simulations were conducted of the vehicle undergoing the six manoeuvres outlined in Table 1. Results from these simulations were used to determine the twelve performance measures for each vehicle. These are shown in absolute terms in Table 2 and relative to the fixed-axle trailer vehicle in Table 3. Each measure is discussed in detail below.

5.1 Low-Speed Corner Swept Path Width (SPW₉₀)

As expected, all steering systems reduced the width of the swept path of the vehicle when travelling around a low-speed corner. Hence an articulated vehicle with a semi-trailer steering system can better negotiate tighter corners and narrower roads compared to a conventional articulated vehicle. This is evident in Figure 7a, which shows the path followed by each test vehicle's trailer group. A narrower swept path is one of the major benefits of installing a trailer steering system.

The narrower swept path opens up the possibility of using the vehicle on narrower roads. An indication of the suitability of a vehicle to a particular road type can be gained by comparing its swept path to the NRTC proposed performance level in Table 1. For travel on arterial roads SPW_{90} must be less than 7.4 m. All vehicles, including the fixed semi-trailer, passed the arterial road criterion (Table 2). For travel on local roads SPW_{90} must be less than 5.0 m. Only those vehicles with semi-trailer steering systems passed this criterion. Thus fitting of semi-trailer steering systems could allow articulated vehicles to be used on local roads as well as arterials.

The relative ability of each of the semi-trailer steering systems to reduce SPW_{90} can be obtained by comparing the values in Table 3. The best performing system was found to be the pivotal bogie system, which reduced SPW_{90} by 27%. For the pivotal bogie vehicle the swept path was only 1.3m wider than the vehicle, indicating excellent tracking. The superior tracking performance of the pivotal bogie system is primarily a result of its effective wheelbase being approximately half that of the fixed semi-trailer.

The command steer system reduced SPW_{90} by 11%. In a corner the command steer system steers two of the axles to reduce the effective wheelbase of the trailer to that of an equivalent single. The least effective system was the self-steer system, which reduced SPW_{90} by only 6%. The reduction in performance is due to steering only one of the axles in the group and hence only reducing the wheelbase of the trailer to that of an equivalent tandem. To improve tracking two self-steering axles could be fitted to a trailer. However, this is seldom done in practice because it can lead to stability and tracking problems at high speed.

5.2 Tail Swing (TS)

All steering systems investigated increased the amount of entry tail swing compared to a fixed semi-trailer. The order of performance was found to be opposite to that of SPW_{90} . The best performing steering system was the self-steer system while the worst was the pivotal bogie, as shown in Table 3. Entry tail swing of the pivotal bogie was above the proposed NRTC limit as indicated in Table 2.

The increased entry tail swing associated with steering semi-trailers is due to their greater effective rear overhang. All steering systems reduce the effective wheelbase of the semi-trailer to reduce the amount the vehicle cuts in on a corner. However, reducing the effective wheelbase increases the effective rear overhang, producing more tail swing.

The fixed semi-trailer, self-steer and command steer systems exhibited no tail swing on exit. In contrast the exit tail swing of the pivotal bogie semi-trailer was above the proposed NRTC limit. Such tail swing may prove to be a problem since drivers are not used to taking it into account when exiting a corner.

The difference in exit tail swing performance is due to differences in the paths the vehicles follow through the corner. The fixed, self-steer and command steer semi-trailers remain inside of the path of the tractor throughout the turn. The pivotal bogie semi-trailer, however, overshoots the path of the tractor on exit, as highlighted in Figure 7a. This overshoot causes the tail to swing out.

Tail swing performance of any of the steering systems can be improved by moving the trailer group rearwards to reduce the amount of rear overhang. However, this will also lead to an increase in SPW_{90} and possible uneven loading distribution between axle groups. Thus the location of the trailer group has to be chosen to produce the best compromise between tail swing, SPW_{90} and load distribution. The ideal location will vary depending on the type of steering system used.

5.3 Steer Tyre Friction Demand (STFD)

Steer tyre friction demand was found to reduce with the addition of a semi-trailer steering system, as shown in Table 3. The order of performance was similar to SPW_{90} , with the best performing system the pivotal bogie and the worst the self-steer axle. All systems had STFD values well below the NRTC proposed level in Table 1 and therefore this area is not of real concern for the type of articulated vehicle simulated.

5.4 Low-Speed Corner Lateral Tyre Force (LTF₉₀)

All steering systems investigated were effective at reducing the lateral tyre forces generated by the drive and trailer wheels when travelling around a low-speed corner. LTF₉₀ gives a relative indication of the amount of tyre wear due to lateral sliding and associated surface damage to pavements due to shear. Lower lateral tyre forces reduce tyre wear and road damage, which are major benefits of installing a semi-trailer steering system.

The most effective systems at reducing lateral tyre forces were the command steer and pivotal bogie systems. Both produced similar results, reducing drive LTF₉₀ values by more than 60% and trailer group LTF₉₀ values by 50%, as shown in Table 3. This represents a substantial reduction in tyre and road wear. The reduction in lateral tyre forces on the trailer group is due to the steering systems aligning all trailer tyres with the direction of travel of the vehicle. The reduction in lateral forces on the drive axle is due to lower lateral forces being transferred from the semi-trailer to the tractor through the fifth wheel.

The self-steering axle was not as effective at reducing lateral tyre forces. Nonetheless, it still reduced drive and trailer LTR₉₀ values by more than 20%. The decrease in performance compared to other systems is due to only steering one of the trailer axles. Opposing “locked in” lateral forces are still developed by the two front trailer axles which leads to higher trailer and drive axle lateral forces.

5.5 Low-Speed Circle Swept Path Width (SPW₃₆₀)

The width of the swept path of the vehicle when travelling around a low-speed circle gives an indication of the vehicle’s ability to negotiate roundabouts. This measure differs from SPW₉₀ because the increased time in the circle allows the vehicle to obtain steady state or equilibrium conditions. A plot of the path followed by each of the trailers compared to the desired path is shown in Figure 7b.

As expected, most vehicles had a greater swept path width when travelling in a circle compared to a corner, as seen by comparing Figures 7a and 7b. Relative performance compared to the fixed-axle trailer, however, was similar, as shown in Table 3. The exception was the pivotal bogie system, which had the same SPW₉₀ and SPW₃₆₀. By contrast to the other systems, the pivotal bogie system’s swept path width reached a peak near the start of the turn and reduced the further it travelled. Thus the maximum swept path width is independent on the length of curve traversed in a circle.

The level proposed for the swept path width in a corner was used as the acceptable swept path width in a circle, namely 5.0m for local roads. Only the command steer and pivotal bogie systems were found to pass this criterion. Hence it may be necessary to fit articulated vehicles with such semi-trailer steering systems if they are to be used on local roads containing tight roundabouts.

5.6 Low-Speed Circle Lateral Tyre Force (LTF₃₆₀)

All vehicles had LTF₃₆₀ values that were slightly higher than their corresponding LTF₉₀ values, as shown in Table 2. This indicates that the steady state lateral tyre forces developed when traversing a circle are generally higher than the initial transient forces encountered on entry.

Although LTF₃₆₀ values were higher, their level relative to the fixed-axle trailer was similar, as shown in Table 3. The pivotal bogie and command steer systems still produced the lowest forces. The self-steer system did not reduce forces by the same extent but still showed marked improvement over the fixed-axle trailer.

5.7 Static Rollover Threshold (SRT)

All vehicles were found to have the same SRT. Hence SRT does not appear to be affected by the use of a semi-trailer steering system. Similar results would be expected if tilt table tests were conducted on representative vehicles since the steering systems do not significantly alter the location of the centre of gravity or the suspension roll characteristics. The SRT was well above the proposed NRTC level of 0.35g.

5.8 Rearward Amplification (RA)

The level of rearward amplification was found to vary depending on the type of semi-trailer steering system used. Two of the three systems increased the level of rearward amplification, as shown in Table 2. The self-steering system, however, was found to have the same rearward amplification as the fixed-axle trailer. This is due to the self-steering axle being relatively stiff in yaw at low steer angles, allowing both systems to generate similar lateral tyre forces in high-speed manoeuvres.

The command steer system was found to have 20% more rearward amplification than the self-steer system while the pivotal bogie system had 30% more, as shown in Table 3. Both of these systems significantly reduce the lateral tyre forces which react inertial loads, hence they increase the lateral acceleration of the trailer unit in high-speed manoeuvres.

Although the command steer and pivotal bogie systems produced higher RA values, the values were still well below the proposed NRTC criterion in Table 1. Thus the increased rearward amplification should be acceptable. Note however, that the SAE lane change manoeuvre used to assess the level of rearward amplification is a relatively mild manoeuvre to perform with a tractor/semi-trailer vehicle. If more drastic manoeuvres are performed with such vehicles the additional rearward amplification could lead to premature rollover.

One way of limiting excessive rearward amplification is to prevent the trailer axles from steering at high-speeds, so the vehicle behaves like a fixed-axle trailer. Some commercially available command steer and pivotal bogie systems employ this strategy.

5.9 High-Speed Transient Offtracking (TO)

All steering systems tested had higher levels of transient offtracking than the fixed-axle trailer. A good indication of the relative performance of each system can be seen in the high-speed lane change paths in Figure 8.

By far the worst performing system was the pivotal bogie. This showed twice the amount of high-speed offtracking as the fixed-axle trailer. Motion of the bogie caused the trailer to overshoot when coming back in line with the tractor after a high-speed lane change.

The command steer and self-steer systems were found to have similar levels of performance, with the command steer system being slightly superior. Transient offtracking levels were 20% and 30% of that of the fixed-axle trailer, as shown in Table 3.

In all cases the level of transient offtracking was well below the NRTC proposed criterion in Table 1. Hence although the transient offtracking is worse it should be acceptable. Like rearward amplification, the increase in transient offtracking can be removed by locking the steering systems at high-speed.

5.10 Load Transfer Ratio (LTR)

LTR was found to vary with the type of steering in a similar manner to rearward amplification. The self-steering system performed the same as the fixed-axle trailer whilst the command steer and pivotal bogies were progressively worse. Compared to the proposed criterion, however, the load transfer ratios were quite similar and relatively low. Hence the load transfer ratios should be acceptable.

5.11 High-Speed Steady State Offtracking (SSO)

As with transient offtracking, all steering systems produced higher levels of high-speed steady state offtracking than the fixed-axle trailer. The command steer system was found to perform best, tracking just 10 mm outside of the fixed-axle trailer as shown in Table 3. The self-steer and pivotal bogie systems had similar results, both tracking 70 mm outside of the fixed-axle trailer. In absolute terms the small increase in offtracking due to the trailer steering system is insignificant. Hence high-speed steady state offtracking of steered semi-trailers is acceptable.

Note that all vehicles, including the fixed-axle trailer, failed the proposed NRTC criterion. This indicates that the proposed criterion may be too stringent for the type of vehicle modelled.

5.12 Handling Quality (HQ)

The handling quality refers to the way a vehicle feels to the driver. A driver may find it difficult to control a vehicle with a significantly different handling quality to a conventional vehicle since it responds in an unexpected way to steering inputs.

Handling quality is usually evaluated using a tractor handling diagram, as shown in Figure 9. The plot shows the lateral acceleration vs. $\delta - L/R$ relationship for the vehicle travelling at constant speed in a diminishing radius circle. The yaw stability of the vehicle can be evaluated by comparing the slope of the handling line to the critical understeer gradient K_{ucr} (indicated by the slope of the black line on the diagram). When the slope of handling curve reaches the critical value the vehicle exhibits yaw instability and becomes directionally unstable. It is desirable for a heavy vehicle not to become directionally unstable before reaching its rollover threshold. Points of yaw instability and rollover have been marked on the diagram.

From Figure 9 it is clear that the fixed, command steer and pivotal bogie vehicles have similar handling qualities while the self-steer vehicle is quite different.

The slopes of the command steer and pivotal bogie curves indicate that both vehicles in turn are slightly more understeer at low lateral acceleration levels and more oversteer at high acceleration levels compared to the fixed-axle trailer. The understeer gradient of the pivotal bogie vehicle reaches the critical level K_{ucr} before reaching the SRT. Hence it becomes directionally unstable before rolling over in contrast to the fixed-axle trailer and command steer vehicles.

The self-steer vehicle starts out being more understeer than the fixed-axle trailer. The understeer gradient stays relatively constant until the lateral acceleration reaches the point where the stiffness of the self-steering axle changes. The vehicle then suddenly becomes more understeer. As lateral acceleration levels increase above this point it becomes progressively more oversteer. The vehicle understeer coefficient reaches the critical level long before the vehicle reaches the static rollover threshold. Hence not only does it become directionally unstable but it does so long before the other vehicles.

In order to quantify the handling performance of a vehicle the NRTC had proposed using a “three point” method. This proposal is currently under review. The method involves looking at the slope of the curve or lateral acceleration at three critical points as shown in Table 1. The results of applying this method are presented in Table 2. All vehicles passed the three point criterion, even the self-steering vehicle with its vastly different handling characteristics. Thus it appears as if the proposed method may not easily indicate if a vehicle handles differently. A better, somewhat qualitative, indication is gained by looking at the complete handling diagram.

6. CONCLUSIONS

Overall, semi-trailer steering systems were found to generally improve the low-speed performance of articulated vehicles. The main advantages of such systems are that they substantially reduce the vehicle’s swept path width and lateral tyre forces. As a result semi-trailers fitted with these systems are generally more manoeuvrable, able to access more of the road network, have lower tyre wear and do less damage to the road surface whilst turning compared to conventional fixed-axle semi-trailers. The only disadvantage found at low-speed was an increase in the amount of tail swing.

The advantages gained at low-speed by semi-trailer steering systems are partially offset by poorer high-speed performance. Such systems generally increased rearward amplification and transient offtracking, which in the worst case could lead to high-speed stability problems. These areas can be improved, however, by locking the steering mechanisms at high speeds.

The semi-trailer steering systems were found to have little effect on the remaining performance measures. These included the static rollover threshold, load transfer ratio and high-speed steady state offtracking. With the exception of the self-steering system, trailer steering also had little effect on the handling of the vehicle.

Comparatively, no semi-trailer steering system was found to perform better than the others in all areas. The pivotal bogie system generally showed the best low-speed performance due to its ability to steer all trailer axles. However, it also had the worst high-speed performance and tail swing. The self-steer system performed best at high-speeds but did not improve load speed performance by such an extent and greatly influenced vehicle handling. The command steer system generally performed at a level between the other two, showing that it may provide the optimal balance between high and low speed performance.

It is important to note that the comparison has been made using the same standard semi-trailer configuration, based on the type of articulated vehicle currently allowed under U.K legislation. As a result parameters such as the trailer length, axle group location and weight distribution were not varied. It is likely that changing these parameters, so that they are optimal for the type of steering system employed, could further improve both high and low speed performance and lead to safer and more economic articulated vehicles. The introduction of performance based vehicle legislation, such as that being proposed in Australia, would allow steering trailers optimised in this way to operate.

6.1 Future Work

The work described in this paper forms the first stage of a larger project concerned with semi-trailer steering systems. The next phase of the project is to validate and calibrate the model, as outlined in Section 3. Further field trials will then be conducted to verify that the main claims made in this paper are correct.

As described above, it may be possible to further improve the performance of steering trailers by changing parameters such as the trailer length, axle group location and weight distribution. More work is required to determine what effect these parameters have and to what extent the performance of the vehicle can be optimised.

The use of active steering systems may also improve performance, especially in those areas that are degraded or not affected by the current passive semi-trailer steering systems. One strategy may involve steering the trailer axles in the same direction as the tractor steer axle at high-speed, similar to four-wheel steering automobiles. It is proposed to investigate various active strategies and verify their performance on a full sized vehicle in the near future.

It is important to note that this study has just compared the performance of the various semi-trailer steering systems. A number of other factors need to be considered when deciding which system is 'best' for a certain application. Such factors include weight, capital and operating costs, durability and adaptability. Whilst this is outside the scope of this project it may be the subject of future work.

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9. TABLES AND FIGURES

Table 1- Summary of Performance Measures

| | PERFORMANCE MEASURE | SHORT NAME | MANOEUVRE | DEFINITION | PROPOSED NRTC PERFORMANCE LEVEL |
|-----|--|--------------------|---|---|--|
| 1* | Low-Speed Corner Swept Path Width | SPW ₉₀ | Centre of steer axle to follow path on straight approaches to a 11.25 m radius 90° circular arc. Vehicle speed is 10 km/h. | Maximum width of the path swept by the vehicle as it traverses the corner. | 5 m for local roads.. 7.4 m for arterial roads |
| 2* | Tail Swing | TS | Same as above | Maximum lateral distance the rear of the vehicle travels outside of the front wheel path. | < 0.35 m on both approaches. |
| 3* | Steer Tyre Friction Demand | STFD | Same as above | $F_{y1} / (F_{z1} \cdot \mu_{peak}) \times 100$ | < 80% of the maximum available tyre/road friction. |
| 4* | Low-Speed Corner Lateral Tyre Force | LTF ₉₀ | Same as above | Drive: Max (F_{y2}) Trailer: Max ((F_{y3}, F_{y4}, F_{y5})) | - |
| 5 | Low-Speed Circle Swept Path Width | SPW ₃₆₀ | Centre of steer axle to follow path on straight approaches to a 11.25 m radius 360° circular arc. Vehicle speed is 10 km/h. | Maximum width of the path swept by the vehicle as it traverses the circle. | - |
| 6 | Low-Speed Circle Lateral Tyre Force | LTF ₃₆₀ | Same as above | Drive: Max (F_{y2}) Trailer: Max ((F_{y3}, F_{y4}, F_{y5})) | - |
| 7* | Static Rollover Threshold | SRT | Centre of steer axle to follow a 100 m radius circular path, test speed slowly increased from 60 km/h until rollover occurs | Max (V^2 / Rg) | > 0.35 g |
| 8* | Rearward Amplification | RA | Prescribed-path lane-change manoeuvre as defined in SAE J2179 (Society of Automotive Engineers, 1993b). Vehicle speed is 88 km/h. | Max ($A_{y \text{ trailer}} / A_{y \text{ front axle}}$) | < 5.7 x SRT |
| 9* | High-Speed Transient Offtracking | TO | Same as above | Maximum lateral distance the rear axle tracks outside the path of the front axle | < 0.8 m |
| 10* | Load Transfer Ratio | LTR | Same as above | $\text{abs}(\text{sum}(F_{y1} - F_{yr})) / \text{sum}(F_{y1} + F_{yr})$ for the trailer unit | < 0.6 |
| 11* | High-Speed Steady State Offtracking | SSO | Centre of steer axle to follow a 393m radius circular path, test speed 100km/h. | Maximum lateral distance the rear axle tracks outside the path of the front axle | < 0.5 m |
| 12* | Handling Quality (Note: <i>Still Under Review</i>) | HQ | As specified in El-Gindy, Woodrooffe and White (1991). Vehicle speed of 100 km/h. | Defined by 3 points on handling curve: 1 st point @ 0.15 g 2 nd point @ over/understeer transition 3 rd point @ 0.3 g | Required values at 3 points: 1 st point: $0.5 < K_{tr} < 2 \text{ deg/g}$ 2 nd point: $A_y > 0.2g$ 3 rd point: $K_{tr} > K_{tr \text{ ter}}$ |

Note: * denotes performance measures obtained from [13] and [14].

Table 2- Absolute Performance Measures

| | PERFORMANCE MEASURE | FIXED-AXLE TRAILER | SELF-STEER SYSTEM | COMMAND STEER SYSTEM | PIVOTAL BOGIE SYSTEM | PROPOSED NRTC LEVEL |
|----|---|---------------------------------|---------------------------------|---------------------------------|---------------------------------|-------------------------------|
| 1 | Low-Speed Corner Swept Path Width (SPW ₉₀) | 5.2 m | 4.9 m | 4.6 m | 3.8 m | <5 m |
| 2 | Tail Swing (TS) | Ent. 0.08 m Exit none | Ent. 0.11 m Exit none | Ent. 0.27 m Exit none | Ent. 0.68 m Exit 0.39 m | Ent. < 0.35 m Exit <0.35 m |
| 3 | Steer Tyre Friction Demand (STFD) | 20.4 % | 18.4 % | 15.1% | 14.3 % | < 80% |
| 4 | Low-Speed Corner Lateral Tyre Force (LTF ₉₀) | Drive 8.4 kN Trailer 18.5 kN | Drive 6.4 kN Trailer 14.4 kN | Drive 3.4 kN Trailer 8.8 kN | Drive 2.7 kN Trailer 9.1 kN | - |
| 5 | Low-Speed Circle Swept Path Width (SPW ₃₆₀) | 6.3 m | 5.9 m | 5.0 m | 3.8 m | <5 m |
| 6 | Low-Speed Circle Lateral Tyre Force (LTF ₃₆₀) | Drive 8.5 kN Trailer 22.0 kN | Drive 6.6 kN Trailer 18.0 kN | Drive 3.4 kN Trailer 12.3 kN | Drive 3.0 kN Trailer 10.4 kN | - |
| 7 | Static Rollover Threshold (SRT) | 0.42 g | 0.42 g | 0.42 g | 0.42 g | > 0.35 g |
| 8 | Rearward Amplification (RA) | 1.07 | 1.07 | 1.24 | 1.38 | < 2.4 |
| 9 | High-Speed Transient Offtracking (TO) | 0.26 m | 0.34 m | 0.31 m | 0.53 m | < 0.8 m |
| 10 | Load Transfer Ratio (LTR) | 0.30 | 0.30 | 0.34 | 0.39 | < 0.6 |
| 11 | High-Speed Steady State Offtracking (SSO) | 0.53 m | 0.60 m | 0.54 m | 0.60 m | <0.5 m |
| 12 | Handling Quality (HQ) | Pass/Pass/Pass | Pass/Pass/Pass | Pass/Pass/Pass | Pass/Pass/Pass | - |

Note: Shaded cells show values that fail the proposed NRTC criterion from [13] and [14].

Table 3- Normalized Results Relative to Fixed Trailer

| | PERFORMANCE MEASURE | FIXED-AXLE TRAILER | SELF-STEER SYSTEM | COMMAND STEER SYSTEM | PIVOTAL BOGIE SYSTEM |
|----|---|----------------------------|----------------------------|-----------------------------|-----------------------------|
| 1 | Low-Speed Corner Swept Path Width (SPW ₉₀) | 1.00 | 0.94 | 0.89 | 0.73 |
| 2 | Tail Swing (TS) | Ent. 1.00 Exit none | Ent. 1.38 Exit none | Ent. 3.38 Exit none | Ent. 8.50 Exit >10 |
| 3 | Steer Tyre Friction Demand (STFD) | 1.00 | 0.90 | 0.74 | 0.70 |
| 4 | Low-Speed Corner Lateral Tyre Force (LTF ₉₀) | Drive 1.00 Trailer 1.00 | Drive 0.76 Trailer 0.94 | Drive 0.40 Trailer 0.48 | Drive 0.32 Trailer 0.49 |
| 5 | Low-Speed Circle Swept Path Width (SPW ₃₆₀) | 1.00 | 0.94 | 0.79 | 0.60 |
| 6 | Low-Speed Circle Lateral Tyre Force (LTF ₃₆₀) | Drive 1.00 Trailer 1.00 | Drive 0.78 Trailer 0.82 | Drive 0.40 Trailer 0.56 | Drive 0.35 Trailer 0.47 |
| 7 | Static Rollover Threshold (SRT) | 1.00 | 1.00 | 1.00 | 1.00 |
| 8 | Rearward Amplification (RA) | 1.00 | 1.00 | 1.16 | 1.29 |
| 9 | High-Speed Transient Offtracking (TO) | 1.00 | 1.31 | 1.19 | 2.04 |
| 10 | Load Transfer Ratio (LTR) | 1.00 | 1.00 | 1.13 | 1.30 |
| 11 | High-Speed Steady State Offtracking (SSO) | 1.00 | 1.13 | 1.02 | 1.13 |
| 12 | Handling Quality (HQ) | - | - | - | - |

Note: Shaded cells show values that are worse than the fixed-axle trailer

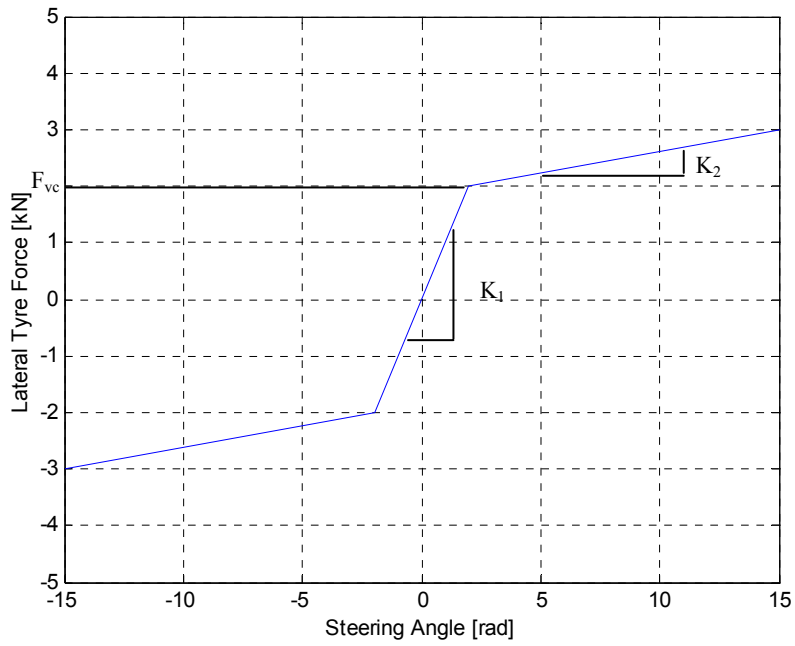


Figure 1 – Self Steering Axle Lateral Tyre Force vs Steer Angle Relationship [2]

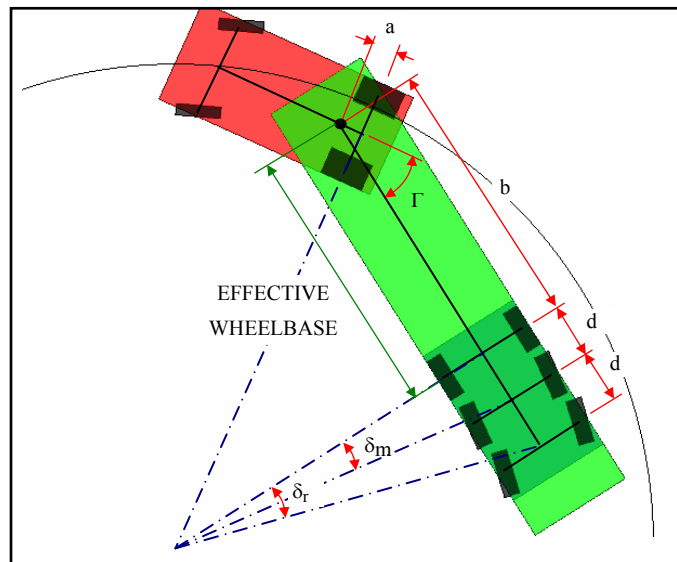


Figure 2– Command Steer Low Speed Geometry

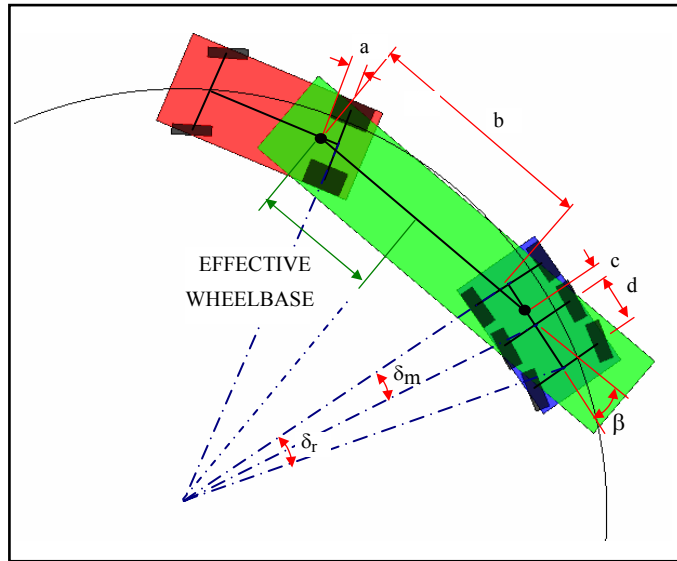


Figure 3– Pivotal Bogie Low Speed Geometry

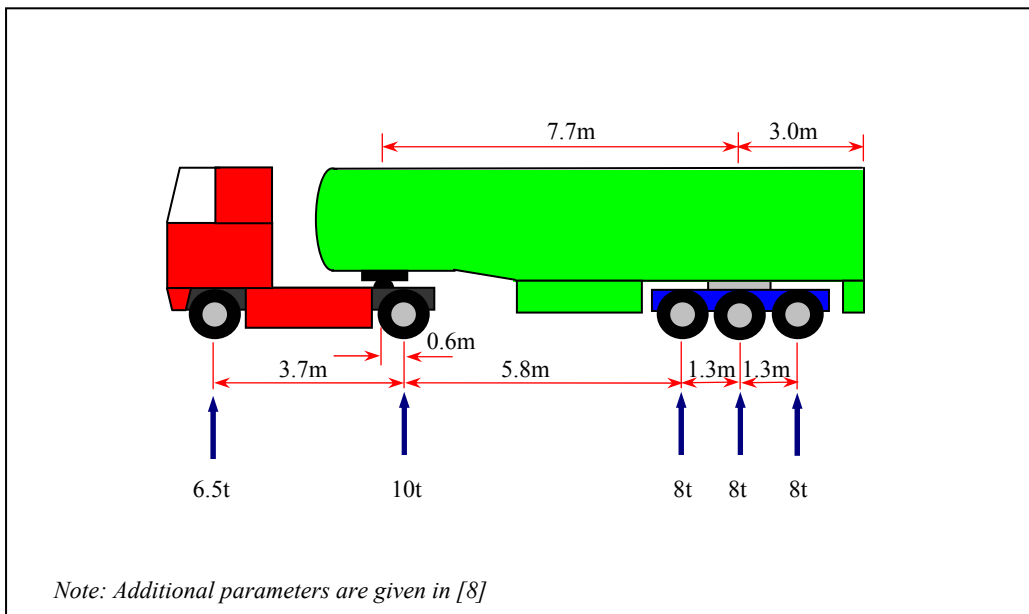


Figure 4– Simulation Model Weights and Dimensions

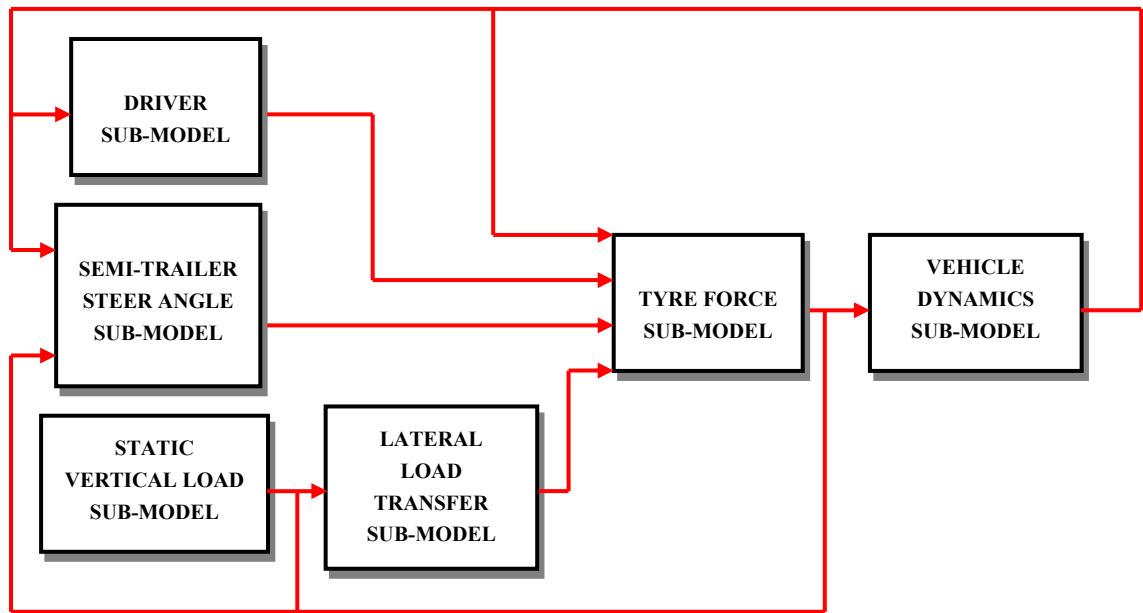


Figure 5- Simulation Program Block Diagram

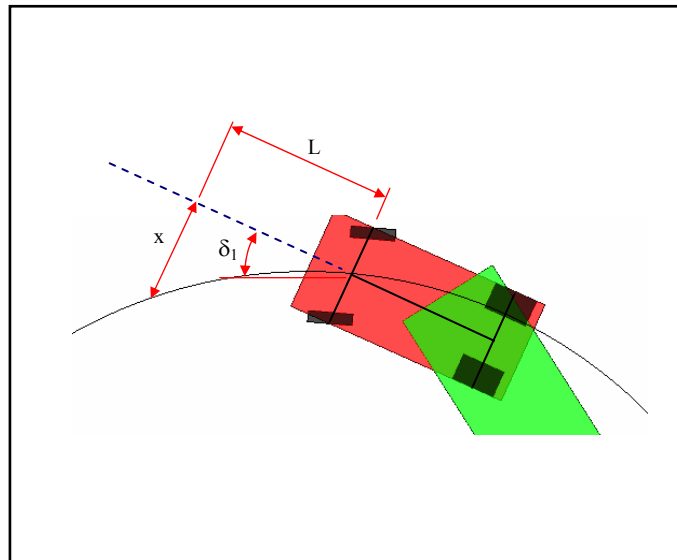
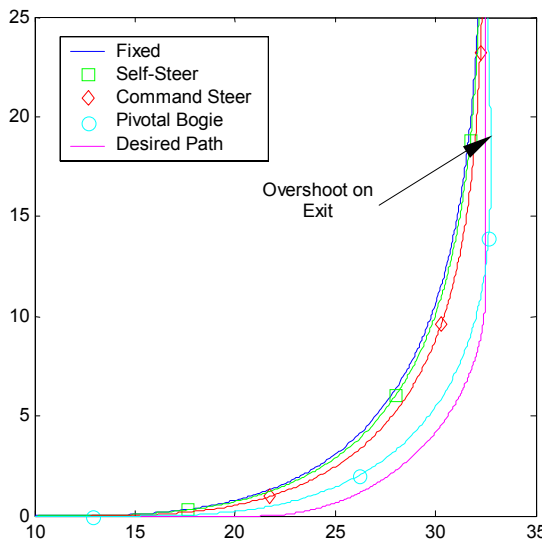
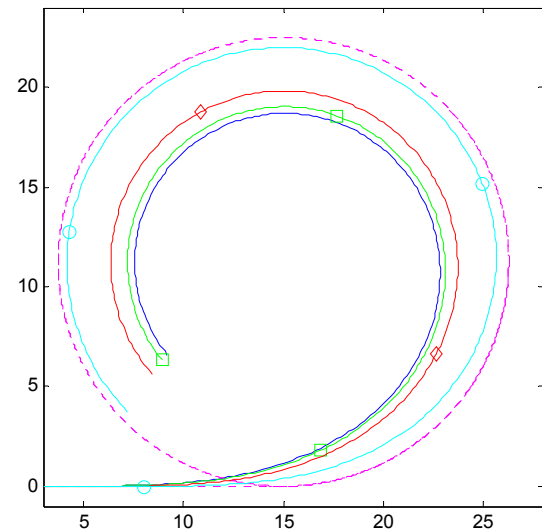


Figure 6- Driver Sub-Model



7a - Low-Speed Corner



7b - Low-Speed Circle

Figure 7 – Path Followed by Trailer Group in Low Speed Manoeuvres

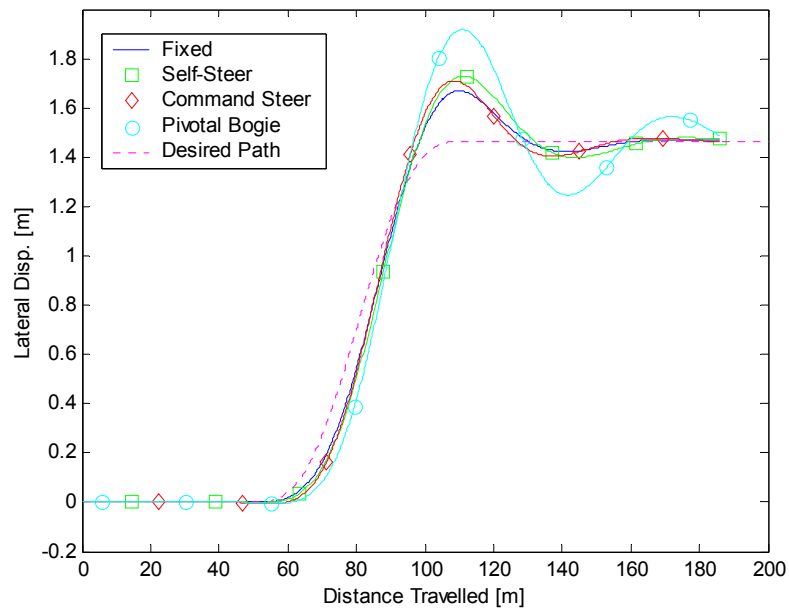


Figure 8 – Path Followed by Trailer Group in a High-Speed Lane Change

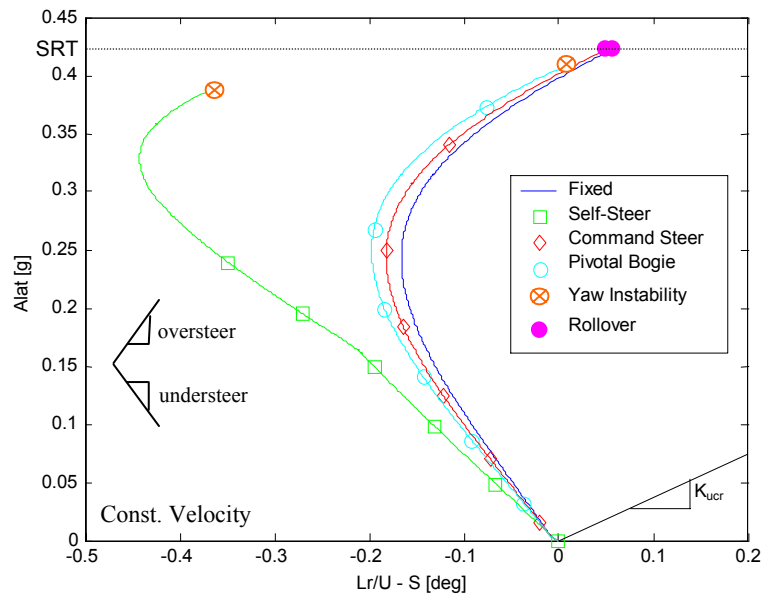


Figure 9 –Handling Diagram