AUTONOMOUS REVERSING OF HEAVY GOODS VEHICLES

A.J. RIMMER†
A.M.C. ODHAMS#
D. CEBON†*

† Cambridge University Engineering Department, Cambridge, United Kingdom
# McLaren Group Ltd, Woking, United Kingdom
* Corresponding author: e-mail dc@eng.cam.ac.uk

Abstract

The problem of reversing heavy goods vehicles with two or more articulation joints could be solved with autonomous steering. This requires a controller designed to position the vehicle using steer angle alone. This paper presents an articulated vehicle model and a controller algorithm which is capable of reversing Long Combination Vehicles successfully. The closed loop stability of this controller is analysed using a linearisation technique and the effects of various parameters and vehicle configurations are investigated. It has been found that the controller is stable for vehicle configurations with up to three articulation joints and three semi-trailers, for which the controller is able to correct a starting offset in an acceptable settling time, with minimal overshoot.

Keywords: Reversing, Control, Stability, Steering, Long Combination Vehicles
1. Introduction

It has been shown that Long Combination Vehicles (LCVs) with two or more trailers have lower fuel consumption per freight task than the tractor semi-trailer combination (Odhams, Roebuck et al. 2010). LCVs are widely used in countries such as USA, Canada and Australia due to their long, straight roads. LCVs with two articulation joints include the B-double and the Nordic combination, which is popular in Scandinavia. See Figure 1.

Reversing an HGV with two or more articulation joints is challenging, particularly when reversing. The normal solution for this is to avoid the need to reverse and unhitch the trailers if necessary. Alternatively, some distribution centres employ highly skilled drivers specifically for manoeuvring B-doubles. This is a very complex task and requires the driver to concentrate on steering, while also checking the mirrors for hazards. It would therefore be beneficial to have autonomous steering for reversing these vehicles, allowing the driver to focus on the mirrors and positioning of the vehicle.

Attempts to solve this problem have been found in the literature, using a variety of approaches. Of these, Fuzzy Logic (Tanaka, Taniguchi et al. 1999) shows the most promise since it has been implemented on a miniature articulated vehicle in the lab. An input-output linearisation method has also been used (Bolzern, DeSantis et al. 2001), as well as a kinematic technique (Michalek 2011). These approaches, however, all have limitations for implementation on a full scale vehicle. Firstly, the hitch point is always assumed to be on-axle or behind the axle, whereas HGVs tend to have hitch points in front of the axle on the tractor units. The kinematic controller uses speed as a control input, in addition to steer angle. It is preferable in practice to have a speed invariant controller so that the driver can control the speed as necessary, for obvious safety reasons. The controllers also have many parameters that require tuning. This is undesirable for a controller which may be implemented on a variety of vehicle configurations, particularly if a ‘learning phase’ is required for this. These issues will all be addressed in this paper. Other gaps in the literature, such as vehicle implementation issues, will not be discussed but investigated later in the project.

A rigorous methodology is required for assessing and tuning the performance of any proposed controller is required. Monitoring the time response for various manoeuvres does not necessarily capture the characteristics of the system and is therefore insufficient. Stably reversing in a straight line, resisting any small perturbations, is the first requirement for any controller that may be used in this application. Therefore, this is a useful starting point for measuring the performance of a controller.

This paper investigates the stability of reversing the various vehicle combinations in a straight line. A simple vehicle model is shown and a controller for path tracking in reverse is presented. These are both linearised about a straight line and the stability characteristics of the closed loop system are analysed. Both the effect of changing controller parameter values and the stability of the various vehicle combinations are investigated.
2. Vehicle Model

2.1 Model Outline
A simple vehicle model with five degrees of freedom was derived in MATLAB, as shown in Figure 3. The parameters used represent those of the Cambridge Vehicle Dynamics Consortium (CVDC) B-Double test vehicle (Figure 2) (Roebuck, Odhams et al. 2010).

\[ M\ddot{z} + Nz + D\delta + O(z^2) = 0 \]  
(1)

Equations were derived in the form of Equation 1 where \( z \) is the state vector \([v_1 \ \Omega_i \ \Gamma_{i-n} \ \dot{\Gamma}_{i-n}]\), \( v_1 \) is the tractor unit lateral velocity at the centre of gravity, \( \Omega_i \) is the tractor unit angular velocity about the centre of gravity, \( \Gamma_i \) is the articulation angle at the \( i^{th} \) hitch point, \( \dot{\Gamma}_i \) is the articulation angle rate, \( n \) is the number of trailers and \( \delta \) is the front axle steer angle. \( M, N, \) and \( D \) are the linear matrices based on geometric, inertial and tyre properties and speed and \( O(z^2) \) represents the non-linear terms.

The vehicle model was generalised, such that it could simulate any LCV configuration by use of suitable matrices \( M, N \) and \( D \).

2.2 Model Assumptions
(i) Constant vehicle speed
(ii) The effects of vehicle roll are neglected
(iii) Linear tyre model of the form \( F=Ca \), where \( F \) is the lateral tyre force, \( C \) is the cornering stiffness and \( \alpha \) is the tyre slip angle (Cheng and Cebon 2008). Non-linear models were compared and found to make negligible difference because slip angles are small in this application.
(iv) No steering angle limits as it was assumed that these will not be exceeded for the straight line case.
(v) No steer angle rate limits for ease of linearisation.
(vi) Single axle trailers were assumed (Winkler 1998), although the tyre parameters were amended to account for this.
(vii) No lateral load transfer.

3. Controller Description
The controller was designed with the objective that the axle of the rearmost trailer (point A in Figure 3) follows a given path. The concept of a preview point (B in Figure 3) is used. This estimates the deviation of the path ahead, allowing time for the necessary steer angle to take effect on the vehicle positioning.

\[ \Gamma_{d2} = f(y, L_p) \]  
(2)

\[ \Gamma_{d1} = -K(\Gamma_{d2} - \Gamma_2) \]  
(3)
\[ \delta = K(G_d1 - G_1) \quad (4) \]

The algorithm is outlined in Equations 2-4 where \( y \) is the lateral preview point offset from the path and \( L_p \) is the preview distance, as shown in Figure 3. \( G_{di} \) is the desired articulation angle for the \( i^{th} \) trailer and \( K \) is the controller gain.

The lateral offset, \( y \), from the vehicle to the path is measured at the preview point. A desired articulation angle between the first and second trailers is calculated based on this lateral offset, using the known vehicle parameters. A desired articulation angle between the tractor unit and the first trailer can be calculated using a proportional controller for the articulation error between the first and second trailers. The steer angle is then calculated using a similar method with the articulation error between the tractor unit and the first trailer. This algorithm can be extended and applied to combinations with any number of trailers.

One advantage of this controller is that there are only two parameters that need tuning, the controller gain, \( K \), and the preview distance, \( L_p \). This controller can be applied to vehicles with more than one axle, using an effective wheelbase (Winkler 1998).

4. Closed Loop Stability Analysis

4.1 Analysis Methodology

The vehicle model and the controller were both linearised about a straight line and Equations 1 and 4 were simplified to Equations 5 and 6 respectively, where \( K_c \) is the matrix representing linearised controller.

\[ M\dot{z} + Nz + D\delta = 0 \quad (5) \]

\[ \delta = K_c z \quad (6) \]

The closed loop system is then defined by:

\[ \dot{z} = Az \quad \text{where} \quad A = -M^{-1}(P + D K_c) \quad (7) \]

The eigenvalues of \( A \) represent the poles of the system and can be analysed to understand the closed loop stability for various controller parameters and vehicle combinations. A root locus diagram can be used to monitor the position of the eigenvalues as a controller parameter varies.

A measure commonly employed as an indication of stability is the damping ratio, \( \zeta \), corresponding to each eigenvalue, as defined by (Franklin, Powell et al. 2001):

\[ \zeta = -\frac{\text{Re}\{eig}(A)\}}{|\text{eig}(A)|} \quad \text{for} \quad \text{Re}\{eig}(A)\} \leq 0 \quad (8) \]

If any eigenvalue has a positive real part (lies on the right half-plane of the root locus diagram), the system is unstable and the damping factor is not defined. The stability of the
system will be governed by the pole with the lowest damping ratio, so this was the value of most interest.

4.2 Stability of the B-Double for Various Controller Parameters

**Damping Ratio**

For the B-double, the lowest damping ratio was calculated for the closed loop system as the preview distance and controller gain varied. The variation of preview distance was constrained to be positive and less than 50m, which was thought to be a reasonable distance, given the overall vehicle length (25m). The results are shown in the form of a contour plot in Figure 4. The threshold of stability is shown by the \( \zeta = 0 \) contour. Values for which the closed loop system is unstable are shown by the grey shading.

The general trend shows that stability increases as the preview distance increases and a controller gain of 5 gives the highest damping. For a given preview distance, the system is stable over a range of controller gains. Within the parameter ranges in Figure 4, the maximum value of damping ratio is 0.54 at \((K = 5, L_p = 50m)\), Point A on Figure 4.

It may possible to increase the damping using a state feedback approach. However, this would require that either all states are observable or an appropriate state estimator can be designed to predict the un-measured states. The controller presented here only requires information on tractor positioning and articulation angles, which can be measured directly using appropriate instrumentation (Lahrech, Boucher et al. 2005). Given these constraints, a damping value of 0.54 is considered reasonable.

**Step Response**

The controller parameters which gave the maximum damping ratio \((K = 5, L_p = 50m)\) were used in a non-linear simulation of the vehicle reversing at a speed of 5km/h. The starting configuration of the vehicle was set to be 0.2m laterally offset from the desired path. Figure 5a shows the resultant offset of both the rear trailer axle and the front of the tractor unit from the path, as a function of distance travelled. This shows a long settling distance of approximately 100m, which is undesirable.

Alternative controller parameters were found to achieve a minimum settling distance \((K = 5, L_p = 10m)\), Point B on Figure 4. The same simulation was run with these parameters and the results are plotted in Figure 5b. Although the settling time is reduced, the tractor unit oscillates with a much larger amplitude than Figure 5a. As is often the case with control systems, there is a trade-off between settling time and damping.

4.3 Stability of Various Vehicle Configurations

A similar analysis was performed on other vehicle configurations and a comparison was made. The configurations of interest had one, two and three trailers, corresponding to the tractor semi-trailer, B-double and B-triple respectively. The B-triple is equivalent to the B-double with an extra B-link trailer. For these configurations, all parameters used were representative of the CVDC test vehicles, as discussed in section 2.

**Stability Threshold**

Figure 6 shows the stability thresholds \((\zeta = 0 \text{ contour})\) of the three vehicle combinations as the controller parameters vary. This shows that the area of stable behaviour is reduced as the
number of trailers increases. The tractor semi-trailer combination shows no upper bound on the controller gain within this range. (Note, however, that in practice, robustness considerations would likely cause a limit on the maximum gain.)

Analysis was carried out for vehicles with more than three trailers and the closed loop system was found to be unstable over all controller parameters. Intuition suggests there is a limit to the number of trailers which can be controlled in reverse using steer angle alone. For this controller, the limit was found to be three trailers.

**Step Response**

As Figures 5a and b show, a trade-off is required between damping and settling time. This is the case for all three vehicle configurations. Controller parameters were determined for responses corresponding to the mid-point between maximum damping and minimum settling time, for all three vehicle configurations. These values were then used in similar simulations to that described in Section 4.2, with the vehicle starting 0.2m offset from the desired path, for all three vehicle configurations. The resultant offsets of the rear trailer axle and the front of the tractor unit are shown in Figures 7a, b and c for the tractor semi-trailer, the B-double and the B-triple respectively.

The three time response graphs demonstrate the stability of the controller to perturbations. There is an increase in the settling distance as the number of trailers increases. There is also an increase in the number of oscillations present in the motion of the tractor unit as the number of trailers increases and the damping of the controller decreases.

The settling distances shown may seem large, particularly in the case of the tractor semi-trailer, where a professional driver could be comparable. However, for the case of the B-double and B-triple, even a long settling distance may be acceptable, given the complexities involved in reversing these vehicles. Also, final accuracy is not something a driver could easily achieve.

**Root Locus Analysis**

Using the preview distances from the mid-range controller parameters described in the previous section, root locus plots were created for all three vehicle combinations. This was done by plotting the eigenvalues of the closed loop system as the controller gain varied from 0 to 50. The plots are shown in Figures 7d, e and f.

Figure 7d demonstrates the stability of the tractor semi-trailer configuration. Although the system is unstable when $K < 0.7$, as shown by the zoomed-in plot, there is no upper limit to the gain within the range plotted. Figure 7e shows that the stability of the B-double depends on one mode, which lies on the positive real axis in the open loop ($K = 0$). As $K$ is increased, this mode has a bifurcation and the two complex eigenvalues cross the imaginary axis at $K = 1.4$. The closed loop system remains stable until these same eigenvalues loop round and cross back over the imaginary axis at $K = 29.8$. A similar trend is seen for the B-triple in Figure 7f but the bounds for $K$ are 2.9 and 7.8.

The root locus plot for the tractor semi-trailer (Figure 7d) is significantly different to the plots for the B-double and B-triple (Figures 7e and f), which have some similar features in common. This may be due to the recursive relationship between the desired articulation angle and the steer angle. For the tractor semi-trailer combination, there is a direct relationship
between steer angle and desired articulation angle but this is not the case for a vehicle with more than one trailer.

5. Conclusions

(i) Straight line stability is a good indication of controller performance. Although it does not capture steady state offset, it is a minimum requirement for any controller that may be implemented on a HGV.

(ii) Using the simple controller presented in this paper, it is theoretically possible to control a B-double in reverse, and a damping ratio of 0.54 can be achieved for the closed loop system.

(iii) There is a trade-off between settling time and damping ratio for the controller performance.

(iv) As the number of trailers is increased from one to three:
   (a) The range of controller parameters for which the closed loop system is stable decreases.
   (b) The settling distance of the step response increases.
   (c) The motion of the tractor unit becomes more oscillatory.

(v) Using the controller presented in this paper, the closed loop system of a vehicle with more than three trailers is always unstable.
6. Figures

**Figure 1 – Vehicle Combinations with Two Articulation Joints**

**Figure 2 – Cambridge Vehicle Dynamics Consortium B-Double Test Vehicle**

**Figure 3 – Diagram of Vehicle Model and Controller Parameters**
Figure 4 – Contours of Damping Factor $\zeta$ as $K$ and $L_p$ Vary for B-Double Combination

Figure 5 - Step Responses of Lateral Offset for B-Double Combination

(a) Controller tuned for damping ratio (Point A on Figure 4)

(b) Controller tuned for settling time (Point B on Figure 4)

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Figure 6 – Contours of Stability Threshold ($\zeta=0$) for Different Combinations

T-S-T = Tractor Semi-Trailer; B-D = B-Double; B-T = B-Triple
Figure 7 - Step Responses of Lateral Offset (a-c) and Root Locus Diagrams (d-f)

- **a, d** - Tractor Semi-Trailer
- **b, e** - B-Double
- **c, f** - B-Triple

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8. References