CLOSED LOOP PERFORMANCE OF HEAVY GOODS VEHICLES

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Abstract

It is not uncommon that a driver of a heavy goods vehicle (HGV) is faced with an unexpected manoeuvre of another road user, resulting in an abrupt steering correction to avoid a collision, which may lead to loss of stability (roll-over, jack-knifing). In this closed loop handling and manoeuvring task, the driver is constantly adjusting the lateral and longitudinal vehicle control. To carry out such closed-loop analyses, a driver model is required. A well-known model is the path tracking model where the (driver) model looks a preview distance ahead of the vehicle to observe the deviation to an intended path, with steering being proportional to the path deviation with a certain gain. For passenger cars, we know the driver model parameters to be mutually dependent, to correlate with mental workload, and to define the closed loop yaw stability boundaries. This paper extends these results to HGV's, with emphasis on the design parameters of (articulated) HGV's. It will be shown that, similar to passenger cars, closed loop performance requires a hyperbolic relationship between preview length and steering gain, depending on the vehicle design (with axle characteristics being very relevant), loading and speed. Closed loop yaw stability will be examined, also in relationship to tractor and semitrailer understeer characteristics, and with consequences such as jack-knifing and trailer swing.

Keywords: driver support and platooning, yaw stability, closed loop performance

1. Introduction

Vehicles are driven by people. The driver is in charge of maintaining road safety under sometimes stressful conditions. High traffic density leads to increased workload, which may result in reduced driver performance. This is especially true in mixed traffic conditions where drivers of Heavy Goods Vehicles (HVG) are facing unexpected behaviour of other road users such as passenger cars, bicyclists, etc. With the high inertia and therefore limited vehicle response time, this is not an easy task.

Closed loop performance deals with the analysis of the performance of HGV with the driver response included. In this handling and manoeuvring task, the driver is constantly adjusting the lateral and longitudinal vehicle control (i.e. steering, braking, accelerating), including the avoidance of obstacles such as other road users, under potentially dangerous conditions.

Closed loop analysis requires a driver model. The most simple model is the path tracking model with three model parameters, a preview distance L_p ahead of the vehicle to observe the deviation D_p to the intended path, a steering gain K_p with respect to this path deviation, and a lag time τ. See [1] for more information, and figure 1. This model has been thoroughly investigated for passenger cars, see for example Abe [5], and applied (see also Genta and Morello, [6])

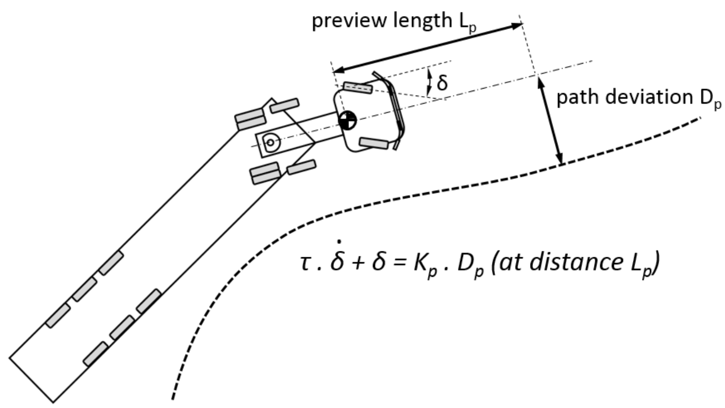

Figure 1.: Path tracking model for tractor-semitrailer

For passenger cars, a number of properties for this path tracking model and the parameters is discussed in [1], based on theoretical and experimental analysis:

- Model parameters (L_p, K_p, τ) are mutually dependent. The lag time is usually set at a fixed value (order 0.1 sec.). The preview length can be replaced by the preview time $T_p = L_p/V$ for vehicle velocity V). Preview time and steering gain are related through an hyperbolic relationship, not depending on the radius of path curvature. For passenger cars, this relationship is almost independent of the velocity as well which is a great advantage for describing closed-loop behaviour on a public road under real traffic conditions, with varying velocity. A

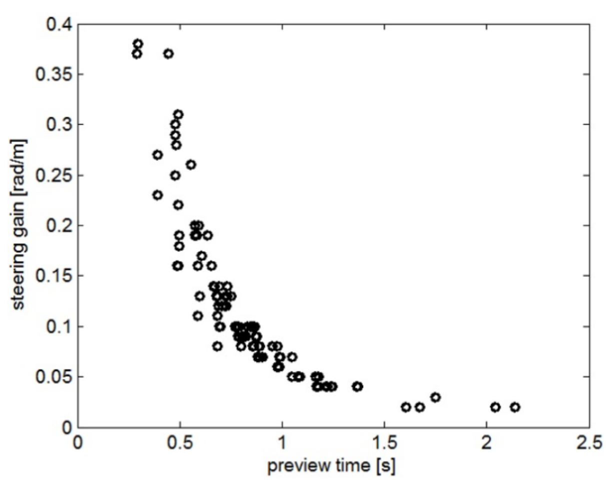

Figure 2.:Driver parameter dependency ([1])

typical example of (T_p, K_p) combinations during a route on a public road is shown in figure 2 (taken from [1]). The parameters (found from matching the model to the real vehicle-driver history) vary during the manoeuvre but follow this hyperbolic relationship.

- Describing a single vehicle with the so-called bicycle model, the hyperbolic relationship can be expressed analytically as follows (see Pauwelussen [1]):

$$K_p = \frac{A_2}{L_p.(A_1+\frac{1}{2}L_p)} \qquad (1)$$

  with parameters A_1 and A_2 depending on the vehicle design characteristics (including understeer gradient, axle positions) and the vehicle velocity, but not on the curve radius. Expression (1) is exact under steady state conditions, and corresponds to the situation where the deviation between intended path and vehicle path is minimal. It appears to be very accurate under dynamic handling conditions too, as discussed by Pauwelussen in [4]
- Parameters K_p (steering gain) and L_p (preview distance) are correlated with **mental workload**. High workload corresponds to high K_p and low L_p which means far upper-left in figure 2, whereas low workload corresponds to low K_p and high L_p,i.e. lower-right in figure 2. This means that the driver model can be used as a virtual sensor to estimate the workload level of the driver.
- **Closed loop stability** can be expressed in terms of these model parameters. This is important to understand safety risk under conditions being potentially dangerous. Closed loop stability appears to be lost for small or very large preview time, and for high steering gain. Compare this with figure 2 where reducing the preview time along the hyperbolic curve of preferred (T_p, K_p) combinations corresponds to higher workload but finally also to closed loop instability.

Hence, the simple model expressed in figure 1 is not only a model to carry out simulation studies with the driver controlling path following, but it gives us an understanding of preferred combinations of preview time and steering response, of the assessment of mental workload and of closed loop stability in relationship to driver performance.

Now let us go back to HGV's avoiding other road users at potentially critical conditions. When the other road user is just in front of the HGV, our previous discussion for single vehicles indicates an increase of workload for the HGV-driver as well as an approach of the closed loop stability boundaries of the driver-HGV system, where the higher inertia will definitely affect the handling performance of the HGV. In this paper, we shall discuss the driver model parameters for articulated HGV's, i.e. tractor-semitrailer combinations. In the next section we shall find a same relationship as (1) for articulated vehicles, where the parameters A_1 and A_2 depend on the vehicle design characteristics, now being a tractor semitrailer combination. Hence, A_1 and A_2 depend on the dimensions and loading conditions of both tractor and semitrailer. In section 3, we shall discuss the general open-loop and closed-loop yaw stability properties of tractor-semitrailer combinations for different vehicle speeds. It is of interest to consider the performance of the tractor-semitrailer with the driver parameters getting close to values corresponding to loss of closed loop stability. It helps us to understand the challenges of the driver when the conditions force the driver to choose a too low preview time or a too large steering gain. We shall also discuss the accuracy of the path following performance in these situations. In section 4, we shall change the axle characteristics to analyse the situation when either the tractor becomes less understeered or the trailer becomes less understeered. Articulated commercial vehicles may have poor open-loop handling properties with risk of jack-knifing or trailer swing. This may be caused by the axle cornering stiffness properties and/or the loading conditions. By varying the axle characteristics (especially of the rear tractor axle) we are able to determine the impact on the closed-loop (vehicle-driver) system, and interpret that. Section 5 includes conclusions and further discussion.

2. Research approach

The research is based on simulation analysis, using well validated vehicle models. In this paper, these mathematical models are simplified as linear single-track models, where the lateral vehicle characteristics are averaged over the vehicle width in a way such that relevant phenomena are maintained. We have based our model on the set of equations of motion, originally derived by Pacejka [3].

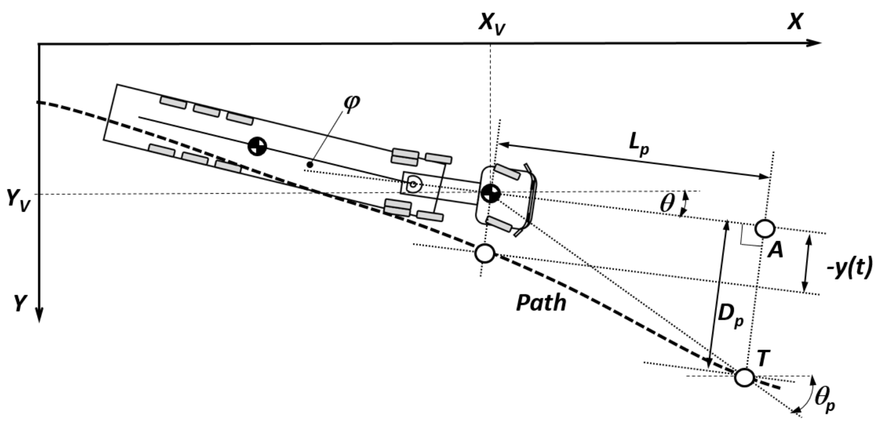

Figure 3.: An articulated vehicle, following a path

Detailing the path deviation D_p in figure 3 in terms of vehicle yaw angle θ, path yaw angle θ_p, vehicle path deviation $y(t)$, we can rewrite the equation for the vehicle steering angle as follows:

$$\tau.\dot{\delta}(\mathrm{t}) + \delta(t) = K_p.\left[L_p.\left(\theta_p - \theta\right) - y(t)\right] \tag{2}$$

Under steady state conditions (i.e. when the vehicle is driving a circular path with curve radius R, and constant speed V), this equation simplifies to:

$$\delta = K_p.\left[L_p.\left(\theta_p - \theta\right) - y\right] \tag{3}$$

When the vehicle is exactly following the desired path, the vehicle path deviation $y(t)$ vanishes. It can be shown (see also [1]) that the difference in yaw angles θ_p and θ can be expressed in body slip angle β, preview length L_p and curve radius R as follows:

$$\theta_p - \theta = \beta + arcsin\left(\frac{L_p}{2.R}\right) \approx \beta + \frac{L_p}{2.R} \tag{4}$$

When we substitute this expression in equation (3), using also expressions (A1.1) and (A1.3) from the annex to this paper, one is able to eliminate the dependent variables in (3), resulting in a similar relationship between preview length L_p and steering gain K_p as given in (1) but now the parameters A_1, A_2 depending on speed and the vehicle parameters of the tractor-semitrailer combination:

$$A_2 = L_1 + \eta_1 . \frac{V^2}{g}; \quad A_1 = g . A_2 . \frac{b - B . V^2}{L_1 . g + \eta_1 . V^2}$$

with all parameters indicated in the annex, see also figure A1.1 and where B is given by:

$$B = \frac{1}{L_1 . C_2} . \left[a . m_1 + \frac{d . m_2}{c + d} . (a + b + e) \right]$$

In figure 4, we have depicted for different speeds the $L_p - K_p$ combinations according to (1) and using the vehicle data from the annex. These are the parameter values which the driver is expected to apply, i.e. being able to follow an intended path with high accuracy. If we consider the curve for 70 km/h, this curve is typically related to a preview time (preview length divided by speed) exceeding 1 [sec]. If we compare this with the data for a passenger car (figure 2), one observes the desired preview time for an articulated commercial vehicle to be much higher. Apparently, a truck driver needs more preview time to follow a path accurately, compared to the passenger car driver, and the question is whether that preview time is available.

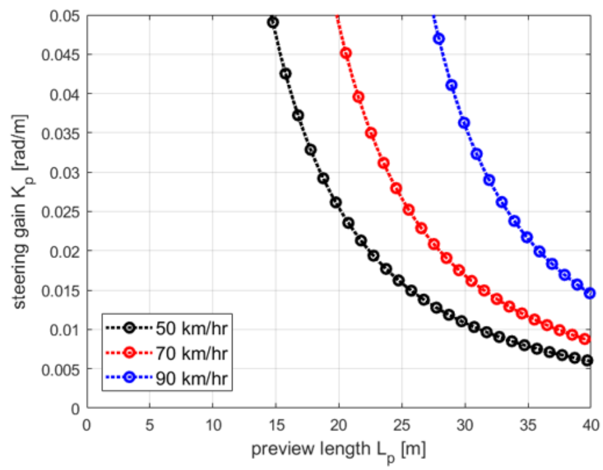

Figure 4.: Hyperbolic steady state L_p-K_p relationships

3. Open- and closed loop yaw stability of articulated commercial vehicles

The **open-loop yaw stability** of a tractor semitrailer is related to the understeer gradients η_1 and η_2, introduced in the annex. The ratio of articulation angle φ and the front axle steering angle δ under steady state conditions is given in (A1.4) in the annex:

$$\frac{\varphi}{\delta} = \frac{L_2 + \eta_2 . V^2 / g}{L_1 + \eta_1 . V^2 / g} \qquad (5)$$

We call the tractor (semitrailer) understeered if $\eta_1 > 0$ ($\eta_2 > 0$), and oversteered if the relevant gradients are negative. For an oversteered tractor, a critical speed exists for which the denominator vanishes in (5) meaning that the gain becomes infinite. Small steering angles with finite articulation angles means that the combination becomes unstable. If the understeer gradient of the trailer becomes negative, the articulation angle changes sign for a sufficiently high velocity, with the semitrailer therefore moving out of the circular path for increasing speed.

The analysis of the **closed loop stability** is based on the combined linear vehicle-driver equations of motion:

$$\dot{\underline{x}} = A.\underline{x} + \underline{f}(t) \tag{6}$$

for matrix A, vector function $\underline{f}(t)$ and state vector $\underline{x} = \left(\beta, \theta, \dot{\theta}, \varphi, \dot{\varphi}, \delta, y\right)^T$. The system becomes unstable if at least one of the eigenvalues of the matrix A has positive real part. We have varied the driver parameters L_p and K_p for fixed lag time $\tau = 0.1$ [sec] where we selected three different speeds, V = 50, 70 and 90 km/hr. The resulting stability areas are shown in figures 5 and 6 (being an enlargement of figure 5), lying below the indicated curves.

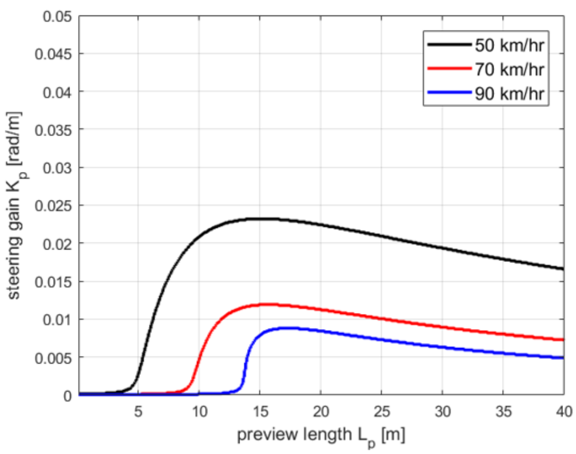

Figure 5.: Closed loop stability area for three different speeds

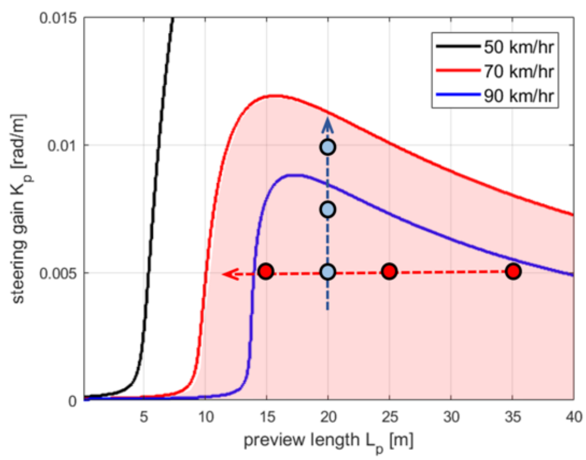

Figure 6.: Same plot as figure 5, but with stability area for V = 70 km/hr highlighted

This means that the closed loop vehicle – driver combination becomes unstable for small preview length (or time, how far is the driver looking ahead of the vehicle?) or for large steering gain (larger corrections in steering for a certain path deviation). For large preview length, the boundary of the stability area drops to lower steering gain values with increasing L_p. This figure also shows that stability is significantly reduced with increasing speed. Increasing speed means that the driver has to compensate by further looking ahead or reducing the steering gain. In more general terms, avoidance of obstacles at short distance ahead of the tractor is only possible at the costs of reduced closed-loop stability or even loss of it. Too severe steering corrections (which may be expected at emergency conditions) lead to a similar effect.

To understand this loss of stability, we have determined the closed loop response of the tractor-semitrailer (linear, single track model), following a single lane change path. We have selected the parameter combinations, indicated in figure 6, for a vehicle speed of 70 km/hr:

$$K_p = 0.005 \text{ [rad/m]}, L_p = 15, 25, 35 \text{ [m]}$$
$$L_p = 20 \text{ [m]}, K_p = 0.005, 0.0075, 0.01 \text{ [rad/m]}$$

The time histories of the tractor centre of gravity are shown in figures 7 and 8. Approaching the stability boundary for low preview length leads to extreme oscillations in the lateral vehicle position (i.e. poor path accuracy).

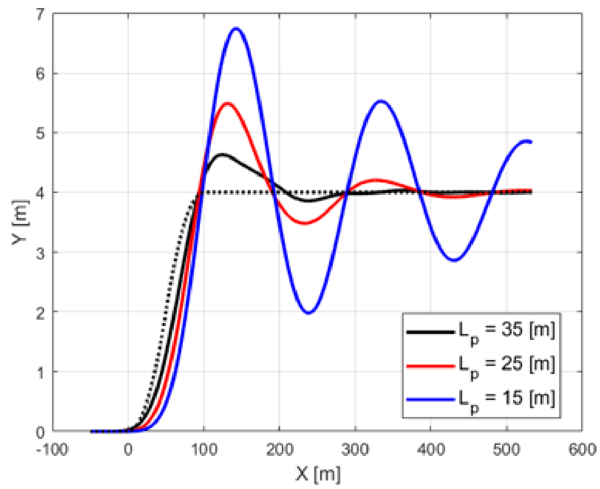

Figure 7.:Single lane change behaviour for K_p = 0.005 [rad/m], 70 km/hr, and different preview lengths

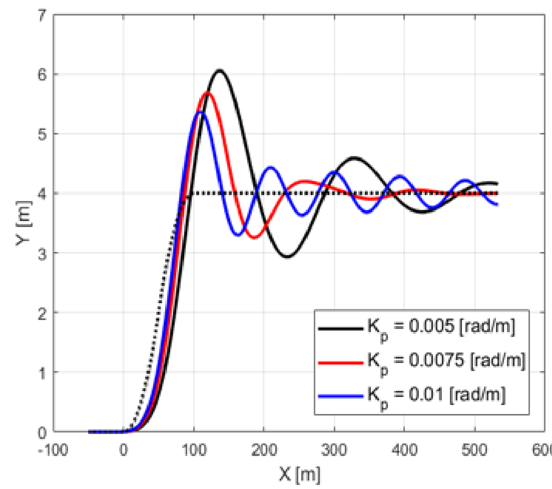

Figure 8.:Single lane change behaviour for L_p = 20 [m], 70 km/hr, and different preview lengths

Increasing the steering gain for fixed preview length (figure 8) leads to higher frequencies, not decaying in time. Increasing the steering gain beyond 0.01 [rad/m] will show the oscillations to increase in time.

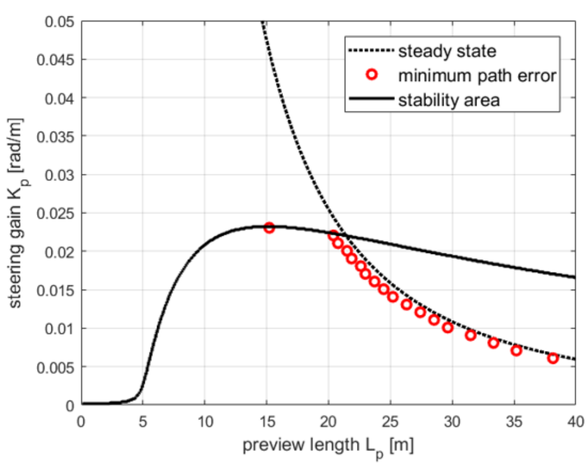

Figure 9.:Steady state behaviour and closed loop stability area (50 km/hr)

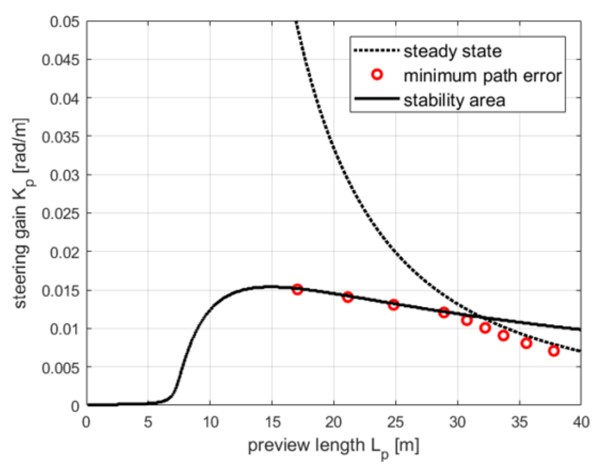

Figure 10:Steady state behaviour and closed loop stability area (60 km/hr)

We now can combine the stability plots with the preferred $L_p - K_p$ combinations, corresponding to the exact steady state behaviour. Results are shown in figures 9 and 10 for 50 and 60 km/hr. The red dots show the parameter combinations for minimum path error for given preview length, within the closed loop stability area, for the single lane change. This confirms that the hyperbolic relationship cf. (1) describes the preferred $L_p - K_p$ combinations very well, even under highly dynamic conditions. The path error increases significantly moving away from this hyperbolic curve. Hence, increasing the speed from 50 to 60 km/hr. means that closed stability can only be maintained at the cost of larger path deviation, i.e. reduced path accuracy.

4. Closed-loop stability for varying axle characteristics.

Some vehicle combination may behave less stable compared to other vehicles combinations. This has to do with the design, payload variation (magnitude, weight distribution on the semitrailer), and axle characteristics. We observed in the preceding section that the understeer gradients for tractor and semitrailer play an important role in that. In this section, we will analyse the impact of changing understeer gradients on closed loop stability. For that purpose, we varied the normalized slip stiffness c_2 of the tractor rear axle, with the resulting understeer gradients shown in figure 11. For small values of c_2 (low cornering stiffness of tractor rear axle), the tractor becomes oversteered (i.e. less open-loop stable), whereas for large value, the tractor becomes more understeered (i.e. more open-loop stable). For the semitrailer, we see the opposite effect where a very stiff axle (large c_2) may lead to a small articulation angle, being possibly opposite in sign compared to the tractor steering angle. In figures 12 and 13, we have plotted the stability areas for the three speeds of 50, 70 and 90 km/hr. again, for two extreme cases, $c_2 = 4.3$ (with η_1 slightly negative) and $c_2 = 7.5$ ($\eta_2 < 0$).

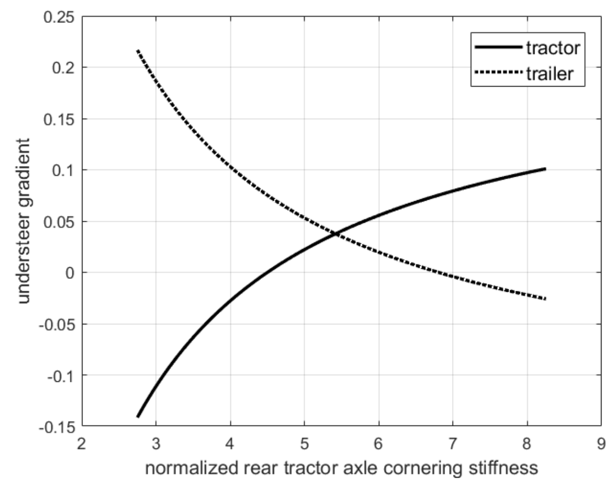

Figure 11.: Understeer gradients η_1 and η_2 for varying normalized axle stiffness c_2

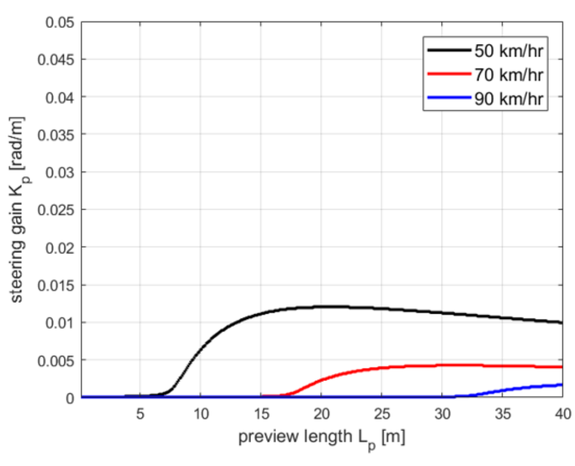

Figure 12.: Closed loop stability, rear tractor normalized axle stiffness $c_2 = 4.3$

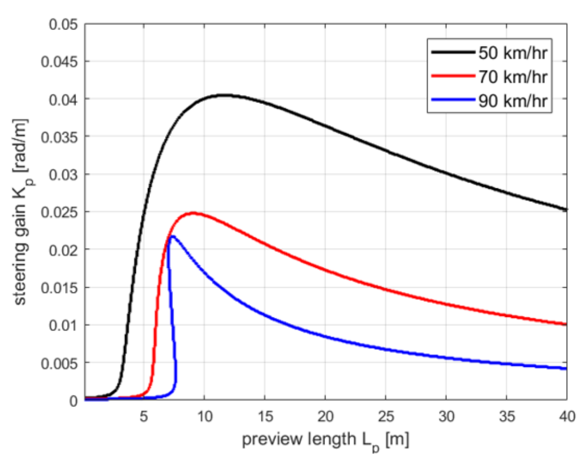

Figure 13.: Closed loop stability, rear axle tractor normalized axle slip stiffness $c_2 = 7.5$

In figure 12, the stability areas have reduced considerably, with for 90 km/hr this area being almost absent. In figure 13, these areas are enlarged, especially for small preview length (relevant for obstacle avoidance). The curve of minimal path deviation according to relationship (1) will slightly change when we vary c_2 but the major impact is that a larger part of that curve corresponds to (closed-loop) stable performance for large tractor understeer gradients, and hardly no closed-loop stable performance for negative tractor understeer gradient.

5. Conclusions and discussion.

The research, presented in the paper, discusses closed loop performance of articulated HGV's. That means that we examined the interactive behaviour of a tractor-semitrailer and the truck driver, in following an intended path. This is important for normal cornering and manoeuvring with finite speed, but especially relevant when the driver is faced with a sudden obstacle (which may be another rad user) which has to be avoided. The driver model, being included, assumes the driver to look some distance ahead of the vehicle (the preview length), and to correct the observed path deviation with a certain steering gain.
The following conclusions can be made:

- There exist combinations of preview length and steering gain, being exact for steady state cornering, and showing a minimum path error compared to closed-loop behaviour for other driver parameter combinations. These preferred parameter combinations appear to give small path deviations, also under dynamic handling conditions, and can therefore be considered as preferred parameter combinations for non-steady state conditions too. The preview length related to these combinations is relatively large, compared to the closed-loop situation for a passenger car. In other words, a truck driver facing an obstacle in front of his vehicle and also close to his vehicle will have difficulties in following an intended avoidance path accurately. It may even lead to loss of closed loop stability.
- In addition, for too severe steering corrections, the closed loop behaviour shows high frequent non-damped and possibly increasing oscillatory response of the driver-vehicle combination. It is clear that (stability induced) restrictions to steering corrections under potentially dangerous traffic circumstances doesn't help the driver to avoid an accident successfully.

These conclusions indicate that avoiding an obstacle by a driver of a tractor-semitrailer is not an easy task. The vehicle properties, especially those affecting the understeer characteristics of tractor and semitrailer, appear to have a strong impact on the closed-loop performance. For low understeer of the tractor, the situation gets worse whereas for higher understeer of the tractor and lower understeer of the semitrailer, the situation improves. These properties depend on the axle characteristics but also on the loading configuration. Axle configurations are related to the tyres, number of axles and wheels, suspension design and stiffnesses. Hence, there are many possibilities for improved closed loop response, which need to be further investigated.

This is just one vehicle combination, described by a single track model with linear axle characteristics. More work has to be done. Other combinations need to be evaluated, taking into account the more detailed vehicle behaviour, also including possibly more articulations, and with variation of loading conditions. It makes sense to link the observed closed-loop performance to existing performance based standards, and possible introduce a new measure related to the closed-loop obstacle avoidance situation.

This will give us a better understanding of the closed-loop performance of driver and (articulated) HGV, helping us to contribute to improved road-safety with HGV's involved.

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Annex.: A linear, single track, tractor-semitrailer model

This paper uses expressions and gains in analytical form, derived from a linear single track model for a tractor semitrailer. The derivation of these expressions, being given in this annex and based on the equations for an articulated vehicle as given by Pacejka in [3], will be treated in a more extensive paper. The notation used is indicated in figure A1, including the wheelbases L_1 and L_2 for tractor and semitrailer, respectively. Steering the vehicle leads to slip angles α_i, side forces F_{yi} ($= C_i.\alpha_i$for axle slip stiffness C_i) for i = 1, 2, 3, lateral acceleration a_y, lateral speed v_y and body

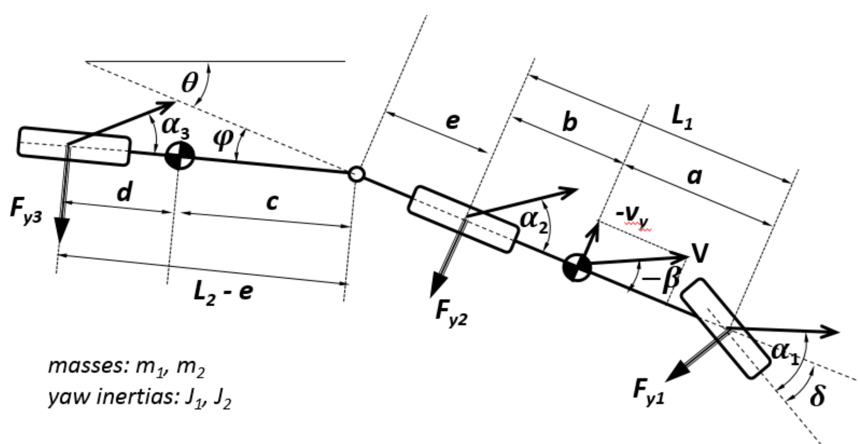

Figure A1.1.: The single track articulated vehicle, notations

slip angle β at the tractor, and the articulation angle φ. The masses and yaw inertias of the tractor and the semitrailer are denoted as m_1, m_2 and as J_1, J_2, respectively.

The steering angle and the articulation angle follow from the lateral acceleration and can be expressed as:

$$\delta = \frac{L_1}{R} + \eta_1 \frac{a_y}{g} \tag{A1.1}$$

$$\varphi = \frac{L_2}{R} + \eta_2 . \frac{a_y}{g} \tag{A1.2}$$

for acceleration of gravity g. The coefficients $\eta_1$and η_2 are the so-called understeer gradients for the tractor and the semitrailer, depending on the vehicle properties. For linear axle behaviour, these coefficients can be written as:

$$\eta_1 = -g . \left[\frac{a.C_1 - b.C_2}{L_1.C_1.C_2} . m_1 + \frac{d}{c+d} . \frac{(a+b+e).C_1 + e.C_2}{L_1.C_1.C_2} . m_2 \right]$$

$$\eta_2 = g . \left[\frac{a}{L_1.C_2} . m_1 + \frac{d.(a+b+e)}{(c+d).L_1.C_2} . m_2 - \frac{c}{(c+d).C_3} . m_2 \right]$$

Hence, these understeer gradients depend strongly on the axle slip stiffnesses C_i. They describe the effect of increasing lateral acceleration on steering angle and articulation angle. Considering expressions (A1.1) and (A1.2), one observes that positive understeer gradients lead to increasing angles with increasing lateral acceleration (tractor and semitrailer are understeered), whereas for negative understeer gradients the angles decrease with increasing lateral acceleration. In that case, tractor and semitrailer are oversteered. Combinations are also possible where for example the tractor is understeered and the semitrailer is oversteered. For an oversteered tractor, the vehicle becomes (open-loop) unstable beyond a critical speed, where jack-knifing occurs, see for example [2]. Clearly, with an understeered tractor and an oversteered semitrailer (which is a realistic situation), the articulation angle may change sign to become negative beyond a certain speed, but still with a positive steering angle. In that case, under steady state conditions, the trailer moves out of the curve (often denoted as trailer swing).

We close this annex with the body slip gain and the articulation gain, being used in the paper.

$$\frac{\beta}{\delta} = \frac{b - \frac{V^2}{L_1.C_1} . \left[a.m_1 + \frac{d}{c+d} . (a+b+e).m_2 \right]}{L_1 + \eta_1 . {V^2}/{g}} \tag{A1.3}$$

$$\frac{\varphi}{\delta} = \frac{L_2 + \eta_2 . {V^2}/{g}}{L_1 + \eta_1 . {V^2}/{g}} \tag{A1.4}$$

For the paper, the detailed description is not necessary to understand the research done and the conclusions. However, from a scientific point of view, the reader should be able to reproduce the results, which is the reason why we have included the above expressions.

We have used the (reference) parameter values, listed in table A1.1. The axle stiffnesses have been specified in terms of the normalized stiffnesses $c_i = C_i / F_{zi}$ for axle loads F_{zi}, $i = 1,2,3$.

Tabel A1.1.: Reference parameter values tractor-semitrailer

parameter	**value**	**parameter**	**value**	**parameter**	**value**
m_1 [kg]	8800	m_2 [kg]	31080	a [m]	1.3
b [m]	2.4	c [m]	5.5	d [m]	2.0
e [m]	-0.6	g [m^2/s]	9.81	τ [sec.]	0.1
c_1 [.]	4.5	c_2 [.]	5.5	c_3 [.]	6.8
J_1 [kg.m^2]	27000	J_2 [kg.m^2]	285000		