INFLUENCE OF THE FIFTH-WHEEL LOCATION ON HEAVY ARTICULATED VEHICLE HANDLING

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ABSTRACT

Dynamic stability is crucial for vehicle safety, not least for heavy commercial vehicle safety. An articulated vehicle such as a tractor-semitrailer may if not correctly designed, handled or rebuilt fall into instability when driving. This is affected by several factors of which the geometric location of the fifth wheel is one. For many reasons the location of the coupling on heavy trucks is not always optimal. The vehicle may e.g. have been rebuilt, with new equipment behind the rear cab panel resulting in a backward relocated fifth wheel. This could negatively influence the directional stability and response. The effect on stability of the fifth wheel location along the longitudinal axis is therefore analysed using a simplified model and a more detailed multi-body model. The study implies that, as the fifth wheel is moved rearwards, the vehicle response becomes highly non-linear and unpredictable. This occurs already at fifth wheel locations close to the rear axle. Moreover, results indicate that as the coupling is moved further rearwards, the tractor-trailer may fall into instability, as it becomes over-steered.

INTRODUCTION

Commercial vehicle dynamic instability has grave implications: the resulting accident types contribute to injuries and environmental damage. Several vehicle occupants are seriously injured or killed every year and vehicles carrying hazardous goods may negatively impact the environment if it leaks or spills after an accident. When a driver detects the vehicle approaching or exceeding the stability limit, the possibility of returning to a stable driving situation and thereby avoiding the accident is often small. A vehicle with a high basic level of stability is however less prone to end up in such situations.

An articulated vehicle such as a tractor-semitrailer behaves in a different manner than a rigid. It is well known that a two-axle vehicle is over-steered, under-steered or neutral-steered. However an articulated vehicle has six different cases of stability. The cases entail divergent instability for the tractor (jack-knifing due to over-steer) and oscillating (tail-snaking) and divergent (trailer swing) instability induced by the trailer.

Several factors are known to influence the stability, e.g. centre of gravity location, tire properties, chassis and suspension stiffnesses, cf. e.g. Dahlberg (2001). One important factor for the stability that is not so well covered in the literature although Jindra (1965) derived the basic equations already 1965 is the property of the fifth wheel. Kaneko and Kageyama (2002) described the stability of a tractor-semitrailer combination while braking under the influence of varying fifth wheel properties. The roll stiffness of the coupling was shown influencing the jack-knifing risk. But certainly also the geometric location of the fifth wheel has an influence on stability. This parameter is not directly controlled by the truck manufacturers even though most manufacturers issue recommendations for mounting of bodywork such as guidelines for fifth wheel location, e.g. SCANIA (2004). By following these recommendations the vehicle will have good handling, ride and manoeuvrability, and correct length and axle load, but they are still only recommendations. In real life, other factors may give rise to a geometric location of the fifth wheel differing from that described in the recommendations. Stein and Hedrick (1980) showed that the ride quality of a tractor-semitrailer deteriorates as the fifth wheel moves forward. That fact could attract to locate it in a more rearward position. But more

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likely, mounting of new equipment behind the rear cab panel, e.g. a crane, will result in a backward relocated fifth wheel. This could negatively influence stability.

This article is therefore going to scrutinise the effects of the fifth wheel location along the longitudinal axis and its impact on directional response and dynamic stability.

ANALYSIS

Simulation models

Two different simulation models are used. The first is an extended bicycle model for parameter identification and comparison with the second, a more complex model. The second model is described as a multi-body system (MBS), using the commercial program ADAMS.

The simple model is a one-dimensional bicycle with two rigid bodies and 6 DOF, although the complex non-linear magic formula is used. Compared to a full three-dimensional model it offers considerably shorter simulation run-time and it is simpler and faster to change any parameter. The model allows an insight into the dynamic behaviour of the vehicle and particularly the consequence of the alterations of the parameters of the vehicle. The limitation is well known, the model is only suitable for a limited range of slip angles and lateral accelerations below 0,4 g approximately, Segel (1956). In other words, the simplified model is well suited for normal driving conditions but not for extreme conditions such as when the fifth wheel is displaced too much rearward.

Steady state simulation model

Linear

A steady state calculation is performed to investigate the influence of the location of the fifth wheel. The equations (1) to (3) are adopted from the book by Wong (2001).

The under-steer coefficient is defined for the vehicle units respectively as:

$$K_{us,i} = \frac{F_{zi}}{C_{\alpha i}} - \frac{F_{z(i+1)}}{C_{\alpha(i+1)}} \tag{1}$$

If $K_{us,i}$ is negative the vehicle is over-steered and if it is positive it is under-steered. Here, small slip angles are assumed and hence the cornering stiffness $C_{\alpha i}$ is a constant. However, it will vary with the axle load as it is depicted in Figure 1.

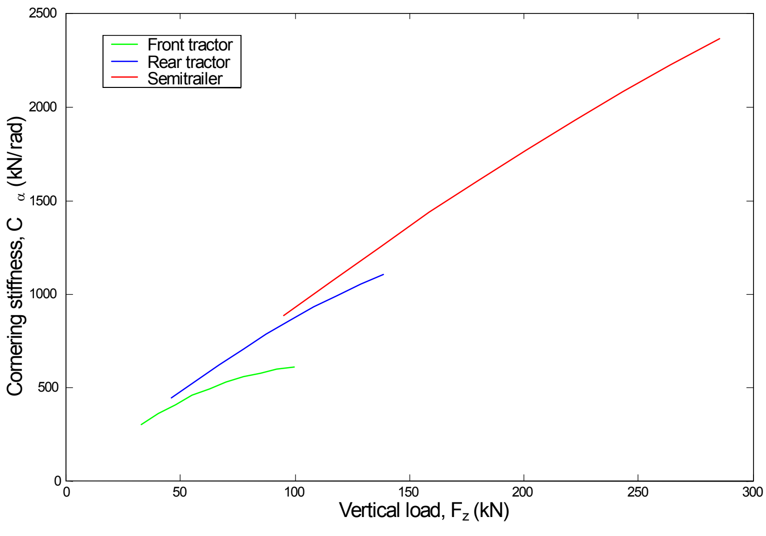

Figure 1. Steady state cornering stiffness versus vertical axle load.

A very interesting property in the steady state case is the gain of the steering angle (δ) in comparison with the articulation angle (ψ), equation (2).

$$G_\delta^\psi = \frac{g\left(L_2 + e\right)v_x^{-2} + K_{us,2}}{gL_1 v_x^{-2} + K_{us,1}} \tag{2}$$

By plotting gain versus time it becomes obvious under which conditions divergent instability occur as can be seen in Figure 2.

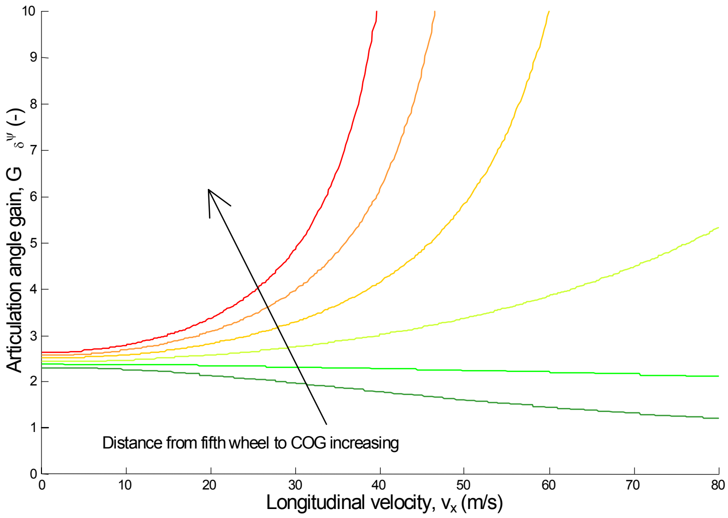

Figure 2. Steering angle to articulation angle gain versus velocity, fifth wheel position varied.

The critical velocity is meaningful if $K_{us,1} < 0$. In that case the gain (equation (2)) will have an asymptote at that velocity, which is calculated by the following formula.

$$v_{crit} = \sqrt{-\frac{gL_1}{K_{us,1}}} \tag{3}$$

Non-linear

One of the most important parameters in a dynamic simulation is the lateral force. If the slip angle is large the forces generated in the road-tire interface is not longer linear but will increase proportionally less than the increase of the slip angle. This non-linearity is commonly modelled with the magic formula, Pacejka (2002). One way to evaluate the lateral dynamics is to derive a handling diagram where the tire non-linearity is taken into account, cf. Winkler (1998) or Pacejka (2002).

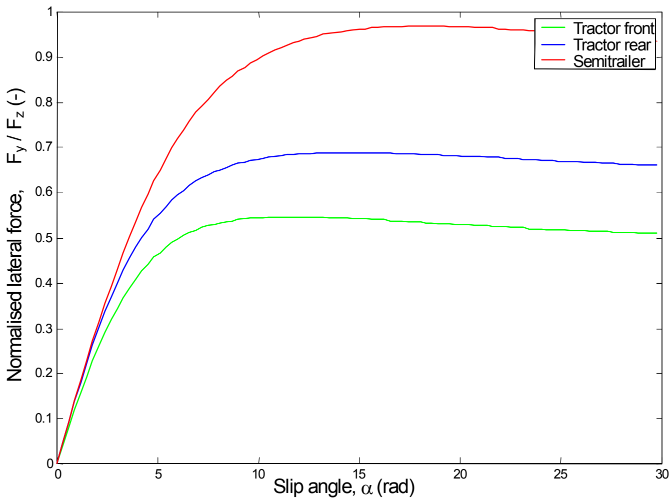

Figure 3. Normalised lateral force versus slip angle for all axles.

Transient simulation model

In this paper the equations are expressed as scalar equations. It is desirable to express these without introducing the heading angle therefore “vehicle fix co-ordinates” are used, otherwise an extra integration would be needed when solving to keep track of heading angle.

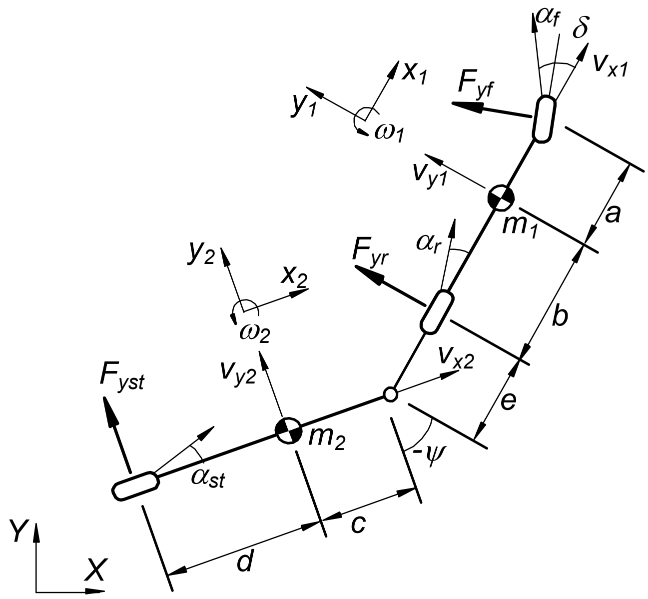

Figure 4. Bicycle model for an articulated vehicle.

The road is considered to be flat and hence, the motion will be planar. Equilibrium of forces gives:

Equilibrium of the tractor

$$m_1\left(v_x \omega_1(t) + \dot{v}_y(t)\right) = -F_{y5} + F_{yf} + F_{yr} \tag{4}$$

$$I_{z1}\dot{\omega}_1(t) = aF_{yf} - bF_{yr} + (b+e)F_{y5} \tag{5}$$

Equilibrium of the semitrailer

$$m_2\left(v_{xst}(t)\omega_2(t) + \dot{v}_{yst}(t)\right) = F_{y5} + F_{yst} \tag{6}$$

$$I_{z2}\dot{\omega}_2(t) = cF_{y5} - dF_{yst} \tag{7}$$

The co-ordinates in the trailer co-ordinate system can be expressed as

$$\begin{cases} v_{xst} = v_x \cos(\psi) + \left(\omega_1(b+e) - v_y\right)\sin(\psi) \\ v_{yst} = v_x \sin(\psi) + \left(v_y - \omega_1(b+e)\right)\cos(\psi) - c\omega_2 \end{cases} \tag{8}$$

in the co-ordinate system of the tractor. The forces F_{yf}, F_{yr} and F_{yst} depend on the slip angles, being highly non-linear for large angles.

Extended model with roll motion

In order to permit roll motion in a simulation with the non-linear bicycle model, it will now be extended by a suspended mass represented by a second rigid body attached to the chassis through the suspension. The roll axis is assumed to be at a constant distance, parallel to the x-axis of the co-ordinate system. The centre of gravity is at a height h above the rolling axis (cf. Figure 5). ϕ_s is the roll angle of the suspended mass. The roll damping and the roll stiffness coefficients of the spring-damper-system are D_S and C_S respectively.

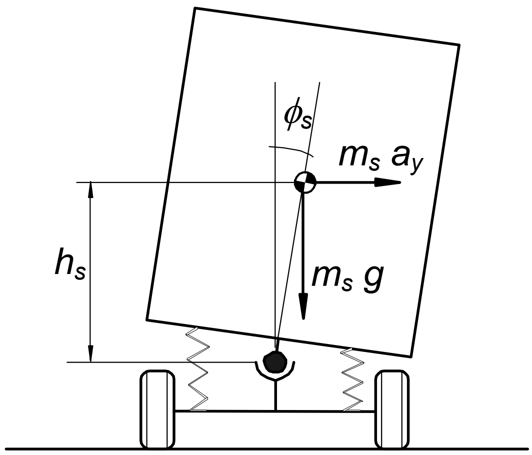

Figure 5. Additional suspended body.

The additional differential equations for the roll motion can be expressed as:

$$I_{x1}\ddot{\phi}_1 + D_{s1}\dot{\phi}_1 + C_{s1}\phi_1 = -m_{s1}a_{y1}h_{s1}\cos(\phi_1) - m_{s1}gh_{s1}\sin(\phi_1) \tag{9}$$

for the tractor, and for the trailer as:

$$I_{x2}\ddot{\phi}_2 + D_{s2}\dot{\phi}_1 + C_{s2}\phi_1 = -m_{s2}a_{y2}h_{s2}\cos(\phi_2) - m_{s2}gh_{s2}\sin(\phi_2) \tag{10}$$

By extending the bicycle model with a roll DOF, the limitation of the before mentioned model, with relevancy for the intended use, is omitted. There are two aspects that are not modelled using this simplified model. These are: the coupling of roll motions between tractor and trailer and also that there is only one DOF of roll per body. I.e. the suspensions of the tractor are lumped into one, and the same is valid for the trailer.

Non-linear tire model

All forces influencing vehicle motion are caused by vehicle-road interaction, with the exception of aerodynamic forces. Tire modelling is crucial for the simulation model. Hence the ability to mimic measured wheel forces is fundamental. In this simulation the tires are modelled according to the magic formula. The tire model requires a series of empirical values, the vertical axle load F_{zi} and the slip angle α_i of each axle as input. From these variables it calculates the lateral force F_{yi} for the axle. The tire model must be evaluated separately for each axle.

The geometry and the fact that plane motion is assumed imply the following expressions for the slip angles:

$$\alpha_f = \delta - \arctan\left(\frac{v_{y1} + b\omega_1}{v_{x1}}\right) \tag{11}$$

$$\alpha_r = -\arctan\left(\frac{v_{y1} - c\omega_1}{v_{x1}}\right) \tag{12}$$

$$\alpha_{st} = -\psi - \arctan\left(\frac{v_{y1} - (b+e)\omega_1 - (c+d)\omega_2}{v_{x1}}\right) \tag{13}$$

MBS model

The ADAMS software is used to develop the necessary MBS models. It is essential to understand what the underlying software is undertaking, as well as how the models are formulated and numerically solved. This perceptive will help to construct and understand efficient MBS models. The complexity of the software is such that the user needs to be very familiarised to the code to be able to construct an efficient and useful model. The model in this analysis has a total of 520 DOF.

RESULTS, COMPARISON AND DISCUSSION

The parameters used along this study are presented in Table 1. These values come from commercial specifications from a standard European tractor with semitrailer.

Table 1. Vehicle parameters.

a	1,3 m	m_1	7500 kg
b	2,4 m	m_2	29800 kg
c	5,62 m	I_{x1}	2800 kgm^2
d	2,93 m	I_{x2}	28000 kgm^2
e	*Variable*	I_{z1}	20000 kgm^2
Roll stiffness truck	2000 kNm/rad	I_{z2}	270000 kgm^2
Roll stiffness trailer	4000 kNm/rad		

All simulations assume ideal conditions. This means, that the vehicle is assumed to be equipped with a power steering that is adequately swift and precise to regulate the steering angle (δ) with no delay. Furthermore, a dry, clean road is assumed (friction coefficient μ = 0,9) and the road is level, i.e. a flat turn. External forces such as wind gusts or aerodynamic forces are not taken into account. Moreover the road conditions do not vary, for instance split-μ is not regarded.

Steady state

Figure 6 illustrates the causes and effects of moving the coupling too far aftward. The figure is divided into two smaller figures. The left part is the static vertical force at each axle as a function of the distance between the tractor rear axle and the fifth wheel (denoted e). The right hand part of the figure shows the critical velocity (equation (3)) versus the distance e. It is effortless to conclude that the vertical force on the front axle and rear axle of the tractor varies with e but the forces on the semitrailer axle and on the fifth wheel are constant. Furthermore the force on the front axle decreases whereas the force on the rear axle increases as the fifth wheel is displaced backward. This effect would in an extreme case lead to that the rear axle will support the whole weight of the tractor and that the front axle would leave the ground. But without going to this severe case it can be interred from the definition of the under-steer coefficient (equation (1)) that this will decrease as the distance e decreases. Not only will the vertical forces decrease for the front axle and increase for the rear axle correspondingly but the tires' cornering stiffness will also change (cf. Figure 1) leading to a double effect of the decrease for the under-steer coefficient. This leads to the almost exponential decrease of the critical velocity (equation (3)) on the right hand side of Figure 6.

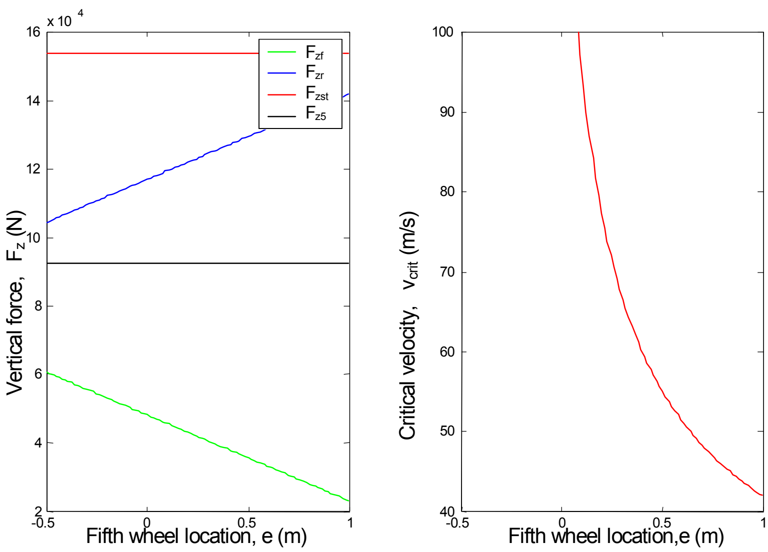

Figure 6. Vertical forces and critical velocity versus distance from fifth wheel to rear axle.

The implication of this was shown in Figure 2 where the gain of the articulation angle to the steering angle versus the fifth wheel position (cf. equation (2)) is calculated for six different cases. Three of these are with the fifth wheel in front of the rear axle and the other three with the fifth wheel behind. The three first cases give values that are stable, i.e. they converge to an asymptote, while in the other three cases, when the fifth wheel is located behind the rear axle, the gain grows to infinity. This indicates that the tractor-trailer would fall into to divergent instability, i.e. jack-knifing due to over-steering.

Steady state – non-linear case

The use of handling diagram is a practice very common in order to investigate the steady state properties taking into account the non-linearity of the tires. This is described in Pacejka (2002) and Winkler (1998).

With this diagram it is possible to see in one glance if the vehicle is inherently unstable for a certain lateral acceleration. This is well illustrated in Figure 7, which is the handling diagram for the original configuration of the tractor-trailer. Here the hitch point is well in front of the rear axle, at a distance of 0.68 metres. Both tractor and semitrailer are stable. This is also confirmed by the linear model. In Figure 8 the response is depicted for the same vehicle but with the hitch point located 0.4 metres behind the rear axle. Here the tractor would be over-steered at lateral acceleration lower than 0.3 g and under-steered for values higher than that.

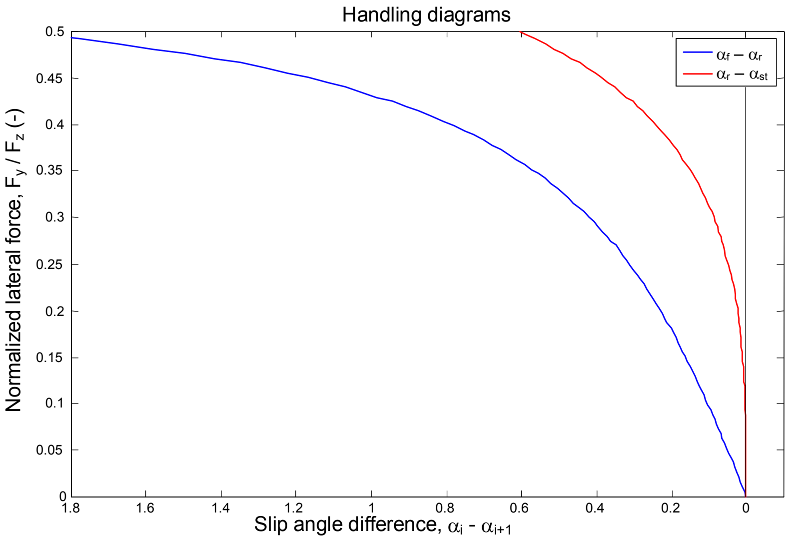

Figure 7. Handling diagram with fifth wheel 0.7 metres in front of the rear axle.

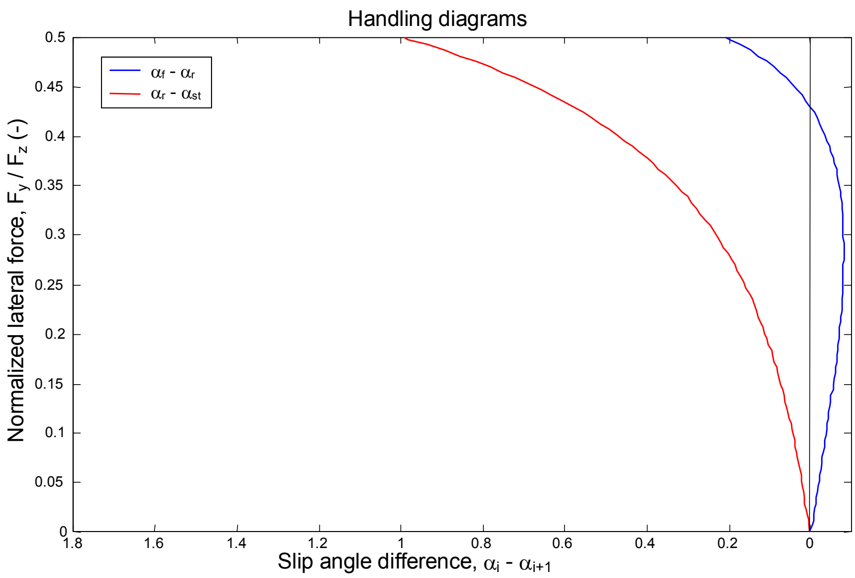

Figure 8. Handling diagram with fifth wheel 0.4 metres behind the rear axle.

Transient solutions

In Figure 9 to Figure 11 comparisons are made for the ADAMS and the transient bicycle model. These calculations are made for three different cases: with the hitch point at 0.7 metres before the rear axle, on top of it and 0.7 metres behind the rear axle. All calculations are made for sinusoidal steering angle with a frequency of two Hertz and with amplitude of one degree.

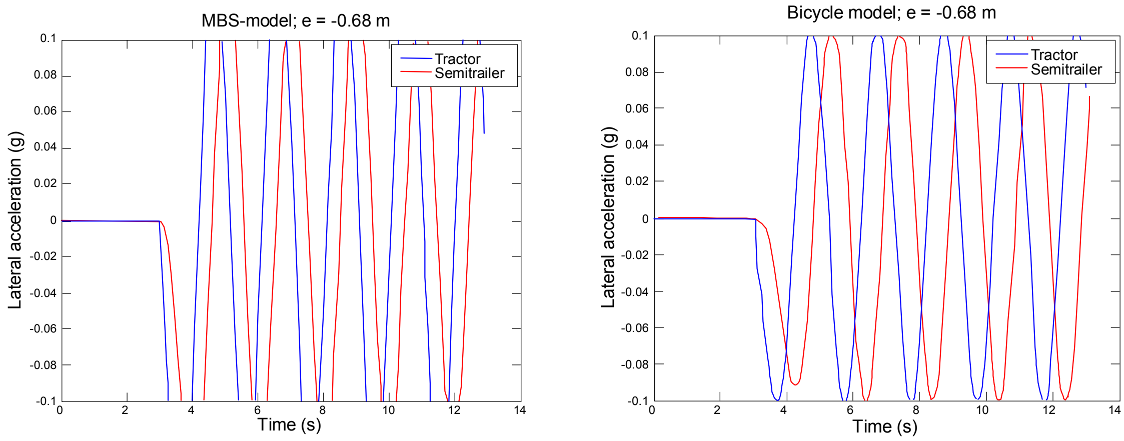

Figure 9. Lateral acceleration of tractor and semitrailer, with fifth wheel in front of rear axle.

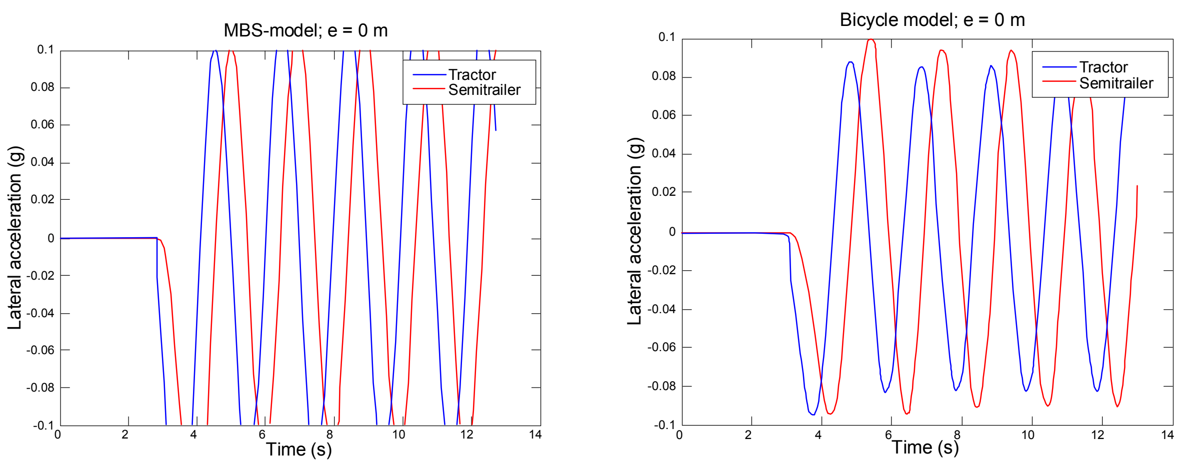

Figure 10. Lateral acceleration of tractor and semitrailer, with fifth wheel on top of rear axle.

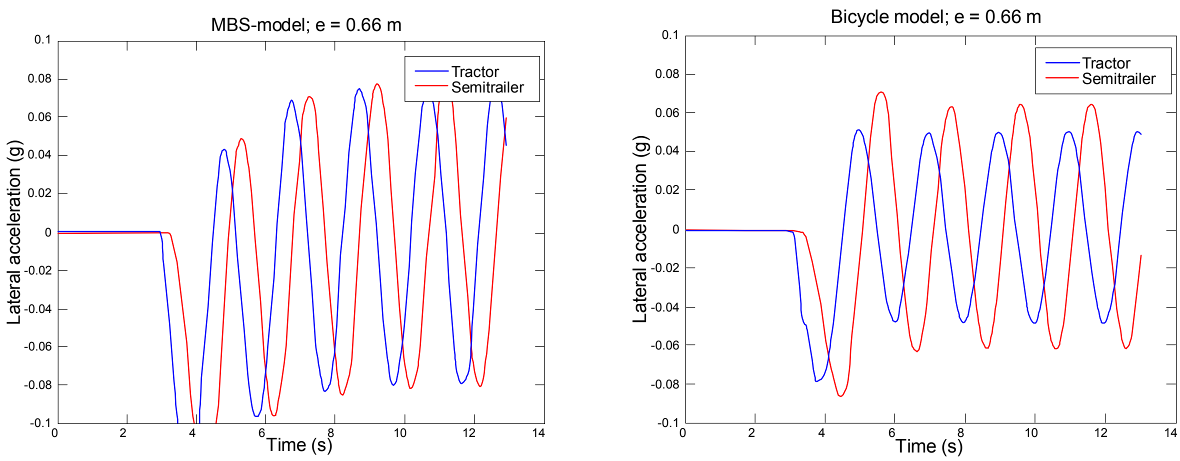

Figure 11. Lateral acceleration of tractor and semitrailer, with fifth wheel behind rear axle.

The simpler 6 DOF analytical model is shown at the left in each graph and the MBS model at the right. The results are remarkably similar considering that the MBS model has about one hundred times more DOF. This also makes the MBS model much slower to execute on a computer, a comparison of the CPU times is not meaningful due to the fast development of computers. However to give an idea: the equations of the simpler model take mere seconds to solve and the ADAMS model takes about one hour to execute. The amplitude is, within less than ten percent, similar. Yet, the difference in the phase is much bigger. This is due mainly to two different phenomena that are not included in the simpler model, the tire relaxation lengths and the stiffness of the hitch point.

Both models show that by moving the fifth wheel rearward the lateral acceleration of the trailer increases proportionally more than that of the tractor. Hence the rearward amplification ratio, RWA, will increase. This again confirms the findings from the quasi-static analysis that the fifth wheel should not be placed too far aftward. The RWA is also an important measure for the rollover propensity. In the case which is analysed no really catastrophic effects are detected because as it is shown in Figure 6 the vehicle is well below the critical velocity. Still the tendency is clear.

SUMMARY AND CONCLUSIONS

The most important result in this study was visualised in Figure 2 where the articulation angle gain was plotted for different fifth wheel positions. As the fifth wheel is moved rearwards, the gain becomes non-linear with velocity indicating that the response of the vehicle becomes unpredictable. This occurs already at fifth wheel locations close to the rear axle. Even though this by itself does not lead to instability, it is an unwanted behaviour since active safety goes hand in hand with a well predictable vehicle response. As the coupling is moved further rearwards the gain may even grow to infinity. This indicates that the tractor-trailer may fall into jack-knifing if the fifth wheel is located in a too far rearward position.

The critical velocity plot in Figure 6 and the handling diagrams in Figure 7 and Figure 8 support these results. As the coupling is located in a position somewhat behind the rear axle, the towing unit becomes over-steering, risking loss of stability.

From these discussions and the result from the analyses, several conclusions can be deducted, the most important being:

- Increased aftward placement of the fifth wheel will negatively influence the stability of a tractor-semitrailer.

Although not discussed explicitly in the paper, the analyses also indicated that the following parameters influence the stability of a tractor-semitrailer negatively:

- Increased height of the load.
- Decreased roll stiffness of the suspension.
- Too high fifth wheel stiffness.
- Tire combination such that lowest cornering stiffness is highest at the rear (of tractor or trailer).

NOMENCLATURE

COG	Centre of Gravity
DOF	Degree of Freedom
MBS	Multi-Body System
RWA	Rearward Amplification Ratio
1,2,5	Vehicle index, *1* indicate tractor, *2* indicate semitrailer, *5* indicate fifth wheel
f, r, st	Axle index, *f* indicate front, *r* indicate rear, *st* indicate semitrailer
s	Index indicating suspended body
a	Longitudinal distance from the COG of the truck to the front axle (m)
a_y	Lateral acceleration (m/s^2)
b	Longitudinal distance from the COG of the truck to the rear axle (m)
c	Longitudinal distance from the fifth wheel to the COG of the trailer (m)
C	Roll stiffness (Nm/rad)

C_α	Cornering stiffness (N/rad)
d	Longitudinal distance from the COG of the trailer to the rear axle of the trailer (m)
D	Roll damping (Nm/rad s)
e	Longitudinal distance from rear axle to the fifth wheel (m)
F_y	Lateral force (N)
F_z	Vertical force (N)
g	Gravity constant (m/s^2)
G	Gain (-)
h	Vertical distance between COG and roll axis (m)
I_x	Roll moment of inertia (kg m^2)
I_z	Yaw moment of inertia (kg m^2)
K_{us}	Under-steer coefficient (rad^{-1})
L	Axle distance (m)
m	Total mass (kg)
t	Time (s)
v_{crit}	Critical velocity (m/s)
v_x	Velocity in forward direction (m/s)
v_y	Velocity in lateral direction (m/s)
α	Slip angle (rad)
δ	Wheel steer angle (rad)
ϕ	Roll angle (rad)
ω	Yaw velocity (rad/s)
ψ	Articulation angle (rad)

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