THE DYNAMIC STABILITY OF
HEAVY VEHICLE COMBINATIONS

by

Olle Nordström and Lennart Strandberg

A paper presented at the
Third International Conference on Vehicle Systems Dynamics
August 12—15, 1974. Blacksburg, Virginia, USA

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ABSTRACT

The object of the investigation was to develop criteria suitable for regulations on the dynamic stability of heavy vehicle combinations. Demands for type approval are proposed in this paper.

A double lane change manoeuvre was chosen as critical for evaluating the dynamic behaviour. It was carried out on a seven metres wide road with the lane off-set being three metres.

The investigation included digital computer simulations and full scale field tests. The field test recordings are used for validation of the simulation model. The vehicle model has nonlinear tyre characteristics and includes roll but not pitch motions. By an inverse method the appropriate steering angle is computed in order to give the tractor a prescribed lateral acceleration time history.

For the evaluation the following risk variables are chosen: 1) The slip angle maximum for each axle, 2) The lateral deviation maximum for each axle, and 3) The overturning risk maximum for each axle or vehicle unit. The simulation results show the influence from speed and from a number of vehicle parameters. They are also used for comparing different types of combinations.

The dynamic behaviour of the tractor + semitrailer + full trailer combination is found to be speed dependent and inferior to comparable truck + full trailer and tractor + semitrailer combinations.

The double lane change is proposed as a test procedure for the dynamic stability of heavy vehicle combinations. The manoeuvre should be performed at 70 km/h with full load, and maximum allowed center of gravity height. Certain maximum values on the mentioned risk variables must not be exceeded for approval.

The problem of off-tracking towards the outside of a curve due to the side slip angles is also discussed, and a steady state test is proposed to assure that this off-tracking is kept within safe limits.
In addition to these studies a full scale investigation was made of the static overturning stability of some heavy trucks, tractor + semitrailers and full trailers at maximum allowed weight and varying centre of gravity height. As a result of this investigation a full scale static overturning stability test is proposed.
THE DYNAMIC STABILITY OF HEAVY VEHICLE COMBINATIONS

By Olle Nordström 1) and Lennart Strandberg 2)

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INTRODUCTION

At the moment no regulations concerning the dynamic behaviour of heavy vehicle combinations exist in Sweden. With increasing traffic density and the development of longer and heavier vehicle combinations the need of such regulations has become more and more obvious.

In 1970 the Swedish Department of Transportation contracted the National Swedish Road and Traffic Research Institute to make the necessary investigations to develop a test procedure for control of the dynamic stability of heavy vehicle combinations and to propose demands which have to be fulfilled for type approval. Furthermore the dynamics of the vehicle combination tractor-semitrailer-full trailer was to be studied with respect to speed dependence and to be compared to truck-full trailer combinations.

The scope of the investigation was mainly restricted to manoeuvres not involving braking and only three types of vehicle combinations were taken into account, namely

1. Tractor-semitrailer
2. Truck-full trailer
3. Tractor-semitrailer-full trailer (double bottom)

Some general discussions concerning braking problems based on literature studies were included in order to complete the picture of the problem area.

1) M Sc, Chief Engineer
2) M Sc, Senior Engineer
TEST METHOD

Full scale field tests were performed for validation purpose. Digital computer simulation was used as the primary test method because of following reasons.

1. This method offers easy variation and control of vehicle and environment data compared to full scale or physical model test.

2. Analytical stability analysis based upon linearizing was not considered appropriate as the nonlinearities are important in severe manoeuvres.

3. Apart from local or global stability, also response time and handling qualities, etc. should be investigated. Thus, simulation of some severe manoeuvres with close connection to real traffic situations was selected.

TEST MANOEUVRE

A double lane change manoeuvre was chosen as most suitable for the investigation. Other possible manoeuvres under consideration were

J-turn
single lane change
slalom course

J-turn and single lane change were considered less severe and the slalom course to far from a real traffic situation.

DYNAMIC STABILITY CRITERIA

In this investigation it has been the aim to study the dynamic stability of the combination in terms which have as close relation to the accident risk in a real traffic situation as possible. Typical accidents caused by the dynamic behaviour of a vehicle combination may be defined as

1. Skidding accidents such as jack-knifing and trailer swing accidents with excessive side slip angles.

2. Overturning accidents due to low overturning stability often in combination with acceleration or side slip angle amplification towards the rear end of the combination.

3. Accidents caused by excessive space demand due to the geometrical dimensions in combination with moderate side slip angles.
The following variables have close connection to these accident definitions and they are generally referred to as risk factors.

**Lateral acceleration (SA)** for the centre of gravity (c.g.) of each vehicle unit is a measure of the manoeuvre violence and of the tyre-road friction necessary to perform the manoeuvre.

**Side slip angle (δ)** for each axle is a measure of the use of available friction. Above a certain side slip angle the maximum cornering capacity is reached and the vehicle goes more or less out of control. In other words the vehicle is skidding.

**Overtuming risk (RV)** describes the overturning risk in terms of relative wheel load change on individual axles or all axles of a vehicle unit. This risk variable is defined in chapter 6.3.

**Lateral deviation** of the vehicle axles was used for description of the space demand.

**Yaw angle** and angles between the vehicle units have been used to measure jack-knifing tendency and oscillatory stability of the units in the combination.

The wheel steer angle of the leading vehicle indicates the difficulty for the driver to perform the manoeuvre. However, easy steering combined with high rearward risk factor amplification (see below) was considered as dangerous.

**Rearward risk factor amplification** defined as the ratio between the risk factor maxima of the semitrailer or trailer axles and the corresponding mean value of the risk factor maxima for the leading vehicle axles. High amplification is judged as dangerous because the feed back to the driver of the behaviour from the rear units is very weak. This is especially true for the full trailer.

The double lane change manoeuvre can be separated in sections, the entry section A, the middle section B and the departure section C. Most risk factors have one major extrem value in each of these sections. The dynamic behaviour of many vehicle combinations have been summarized only by these values. Of course this was preceded by a thorough examination of the time histories for every variable in the computer output.
The mathematical vehicle model permitted the study of three different heavy vehicle combinations:

1. Tractor and semitrailer. One articulation.
2. Truck and full trailer. Two articulations.

The moving coordinate system was fixed to the sprung mass of the tractor (truck) but did not roll. The system was right hand Cartesian with vertex in the centre of gravity of the total mass of the tractor. The x-axis was horizontal in the plane of symmetry of the tractor with positive direction forward. The z-axis was positive upwards.

The most important of the simplifying assumptions in the mathematical model are listed below:

1. Roll axes were horizontal and the centres of gravity of the unsprung masses were located on the roll axes. Thus the centre of gravity of the total mass was fixed to the sprung mass.
2. Pitch motion was neglected.
3. Roll angles were considered small.
4. Camber angles were neglected.
5. Roll and compliance steering effects were neglected.
6. The road was considered flat and horizontal, and no vertical movement was included.
7. The principal axes of inertia of the tractor coincided with the vehicle fixed coordinate system. The principal axes of inertia of the other units in the combination were located analogously.
8. The tyres were considered to be rigid. Consequently the unsprung masses had no freedom to roll.

9. The side slip angle for all wheels on one axle were regarded as equal and calculated at the axle centre. The lateral movement of the axle in the centre of gravity system, due to roll, was neglected in these calculations.

10. Bogie axles were substituted by single axles.

For the vehicle combination tractor-semi trailer-full trailer the program was based upon 17 basic equations. For the truck-full trailer the number of equations were 13. The tractor-semi trailer combination required 8 basic equations. The following variables were solved from these equations by matrix inversion.

1. Longitudinal and lateral acceleration of the leading vehicle c.g.
2. Yaw angle acceleration for each unit in the vehicle combination
3. Roll angle acceleration for each unit in the vehicle combination
4. Horizontal coupling forces in each articulation point
5. Roll torque transmission in the full trailer turntable

(In order to avoid numerical oscillations, caused by small inertia and large forces, the dolly roll movement was attached to the rear trailer movement. So, the original number of equations - 16, 12 and 8 respectively - might have been reduced - to 15, 11 and 8 respectively. However, programming efforts were less with the solution mentioned above - i.e. 17, 13 and 8 equations respectively).

Subsequent to the matrix inversion, integration and a number of subroutines were performed for evaluation of sideslip angles tyre forces (chapter 6), steering angle (chapter 7 and 8), output variables etc.

The program structure was according to the IBM 360 CSMP translator but most statements were written in FORTRAN IV. The computer cost was approximately 6 US dollars per real time second when about 50 variables were printed (5 times per real time second) and when about 30 diagrams were plotted.
6. TYRE-ROAD CONTACT FORCES

6.1 Side forces

Tyre side forces for different wheel loads and side slip angles were entered into the simulation program in tabular form. Table data originated from measurements according to appendix.

Linear interpolation between table data was used when side force and wheel loads were computed by an iterative procedure for each axle. The side slip angle for each time step was solved in advance outside the iterative loop.

6.2 Traction forces

The only traction forces involved were the rolling resistance and the forces on the rear wheels of the tractor necessary to keep the speed constant. These forces were comparatively small, and their effect on the side forces was not considered.

6.3 Wheel loads and overturning risk

The dynamic wheel loads (iteratively computed according to section 6.1) were used for calculation of the overturning risk (RV) defined by

\[ RV = \left| \frac{\text{Dynamic wheel load on left side}}{\text{Static wheel load on left side}} - 1 \right| \]

RV was always calculated for each axle. When appropriate, RV was also evaluated for all axles together on a truck, on tractor + semitrailer, and on full trailer.

7. INVERSE STEERING PROCEDURE

An inverse steering procedure was developed in order to get comparable simulation results for the investigated vehicle combinations. The procedure made the leading vehicle in every vehicle combination follow the same trajectory and lateral acceleration time history, independent of vehicle parameters, load or load distribution.
Of course, certain conditions concerning tyre characteristics and trajectory design (see next chapter) had to be satisfied. In these simulation runs the steering angle output from the inverse steering procedure subroutine was reasonable and similar to the full scale test. See Figure 9.3.

The steering angle demand was calculated at each time step from the equations of motion for the leading vehicle. Inputs to the subroutine in question were:

- predetermined lateral acceleration for the centre of gravity of the leading vehicle,
- current values of state variables for the leading vehicle necessary in calculation of steering angle demand.

The output from the subroutine - i.e. the steering angle necessary to achieve the desired lateral acceleration - acted as input to the vehicle model outlined in chapter 5. In this way no "railway-effects" were introduced by the inverse steering procedure.

Other methods of steering along a predetermined path have been found in the literature before and after this investigation. However, this procedure seems promising and the development will continue.

MATHEMATICALLY DEFINED TEST COURSE

Preinvestigations of the inverse steering procedure in an analogue computer indicated that some specific constraints had to be put on the test course. Unreasonable steering movements appeared if the third order derivative (\( \dddot{y} \)) of the lateral deviation was discontinuous.

The test course illustrated in Figure 8.1 was selected as input for the inverse steering and has been used in all simulations except some validation runs. The acceleration peak values were 1.75 m/s² and its time history was composed by harmonic and linear functions of time. The lateral deviation peak value was 3.0 m.

At none of these simulations the difference between desired test course and simulated trajectory exceeded 0.05 m. This error was regarded as negligible compared to the desired lateral deviation peak value (3.0 m). The comparison indicates a satisfactory effectiveness of the test course and inverse steering subroutines.
Figure 8.1: Lateral road position (Y) as function of time for the desired test course.

Figure 8.2: Road oriented lateral acceleration (\ddot{Y}) as function of time in the desired test course.
VALIDATION OF THE MATHEMATICAL MODEL

9.1 Full scale tests

In order to validate the mathematical model full scale field tests have been performed. The following configurations were tested

<table>
<thead>
<tr>
<th>Vehicle combination</th>
<th>Load condition</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Truck</td>
</tr>
<tr>
<td>Tractor-semi trailer</td>
<td>Full load</td>
</tr>
<tr>
<td>Truck-full trailer</td>
<td>Full load</td>
</tr>
<tr>
<td></td>
<td>Unloaded</td>
</tr>
<tr>
<td>Tractor-semi trailer-full trailer</td>
<td>Full load</td>
</tr>
<tr>
<td></td>
<td>Unloaded</td>
</tr>
</tbody>
</table>

The test manoeuvre represented a real traffic situation where the vehicle combination is driven on the right hand side of a road of seven metres width. Suddenly the driver discovers an obstacle of ten metres length blocking the right hand side of the road. In order to avoid a collision the driver starts steering over to the left hand side 40 m in front of the obstacle. Because of oncoming traffic the driver has to be back on the right hand side 40 metres behind the obstacle.

For the tests the road edges, road centre line and obstacle were marked by rubber cones.

The driver kept the speed as constant as possible. The test was carried out at several speeds between 40 and 80 km/h. The test course was not changed for different speeds like in chapter 13.

Recorded data were steering wheel angle, road distance, yaw angle of the tractor, lateral acceleration for all vehicle units, angles between the different vehicle units and the roll angle for one vehicle unit selected differently on repeated runs.

* Data according to "Low c.g. combination" in appendix, table A1.
9.2 Validation

At the validation simulations the mathematical vehicle model has been steered either according to recorded steering wheel angle time history or by the inverse steering routine along a test course closely corresponding to the course performed during the field test. See appendix, table A1 for vehicle data.

Figure 9.1 a-c shows a comparison between lateral accelerations achieved at field test and simulation. Only one of the field test curves was measured by a roll stabilized accelerometer. For the other curves a correction for the roll angle influence has been indicated for the peak values. As the simulation accelerations were horizontal this was necessary for a fair comparison. The result is considered as satisfactory.

In figure 9.1 d the front axle courses of the tractor from field tests and simulation are compared. The course achieved at the simulation deviates from the field test course. It must be noted however that a change of the initial direction as slight as 4 m rad (~0.23°) corresponds to the shown deviation at the end of the course. If this correction is made the two curves coincide quite well.

Figure 9.2 shows a comparison between courses from a field test and from a simulation where the inverse steering routine has been used. Figure 9.3 shows the corresponding wheel steer angles. The field test wheel steer angle has been calculated from recorded steering wheel angle. No correction for steering compliance, roll steering etc. has been made why the similarity probably is better than shown.

Figure 9.4 a-c show comparisons of lateral accelerations. The similarity is regarded as satisfactory.

The validation tests show that the mathematical vehicle model gives simulation results well coinciding with field test results.
Figure 9.1 a-c Comparison between simulation and field test results for a tractor-semi-trailer-full trailer combination. The steering function measured at the field test was used as steering input in the simulation.
Figure 9.2  Comparison between front and rear end trajectories of a tractor-semi-trailer-full trailer combination in field test and simulation. Inverse steering procedure and mathematically defined test course were used in the simulation.

Figure 9.3  Comparison between wheel steer angles in field test and simulation. Tractor-semi-trailer-full trailer combination. Inverse steering procedure and mathematically defined test course were used in the simulation.
Figure 9.4 a-c Comparison between simulation and field test results for a tractor-semi trailer-full trailer combination. Inverse steering procedure and mathematically defined test course were used in the simulation.
SINGLE VEHICLE PARAMETER VARIATION

The influence from changes in single vehicle-parameters was studied for a tractor-semitrailer-full trailer combination. The aim was to get an idea of the relative importance of different design parameters. The basic vehicle data conformed to the fully loaded field test combination except for the tyre-road study where the low c.g. truck-full trailer was used. See appendix for vehicle data. Each parameter was assigned three different values according to the table below. Basic data are in the middle column of test values. Best value is indicated by +.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Test values</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Tractor wheel base (m)</td>
<td>+ 2.5 3.4 6.0</td>
</tr>
<tr>
<td>2. Fifth wheel position in front of rear axle (m)</td>
<td>0 0.60 + 2.23</td>
</tr>
<tr>
<td>3. Roll torque stiffness of the fifth wheel - kingpin (Nm/rad)</td>
<td>3000 30000 300000</td>
</tr>
<tr>
<td>4. Roll stiffness of the semitrailer rear suspension (Nm/rad)</td>
<td>850000 1700000 3400000</td>
</tr>
<tr>
<td>5. Yaw moment of inertia of the semitrailer (kgm²)</td>
<td>120000 236000 350000</td>
</tr>
<tr>
<td>6. Distance between semitrailer axle and trailer drawbar tow pin (m)</td>
<td>++ 0 1.94 3.9</td>
</tr>
<tr>
<td>7. Drawbar length of the full trailer (m)</td>
<td>++ 1.72 3.34 4.5</td>
</tr>
<tr>
<td>8. Viscous yaw velocity damping of the articulation joints (Nm/rad)</td>
<td>8.1 Kingpin 0 + 400000</td>
</tr>
<tr>
<td>8.2 Tow pin 0 + 50000</td>
<td></td>
</tr>
<tr>
<td>8.3 Turntable 0 + 50000</td>
<td></td>
</tr>
<tr>
<td>9. Tyre road characteristics. standard cornering stiffness:S (Low c.g. truck-full trailer)</td>
<td>S +++ 1.5 S</td>
</tr>
</tbody>
</table>

+ = best value, small effect
++ = best value, medium effect
+++ = best value, large effect
The most important effect on the risk factors was achieved from the following changes

1. Distance between semitrailer axle and trailer drawbar tow pin. The shortest distance was most favourable (see fig. 10.1).

2. Drawbar length of the full trailer. Contrary to reported results from step steering tests the shortest drawbar was considered as the best alternative. Though the longer drawbars were superior in the first part of the manoeuvre the ranking order was reversed in the middle part which was the most critical (see fig. 10.2).

3. Tyre-road characteristics. Tyres with 50% higher cornering stiffness and friction compared to standard tyres had considerably lower risk factor values (see fig. 10.3).

Less important, but worth mentioning, are the following indications from the simulation results.

Short tractor wheel base was favourable.
Viscous kingpin and tow-pin damping were favourable.
Viscous turntable damping was unfavourable compared to no damping.

The rest of the parameter variations showed no clear effects.
Figure 10.1 Maximum absolute values of different risk factors. Field test combination with different distance between axle and tow pin \((q_{56} - 1_{56})\) for the semitrailer.
Figure 10.2

Overturning risk

Maximum absolute values of different risk factors. Field test

Overturning risk maxima for each axle

Lateral deviation for each axle

Sidetrip angle

Lateral acceleration

Vehicle unit

Course section A

Course section B

Course section C

TP: Center of gravity
Figure 10.3  Maximum absolute values of different risk factors. Low c.g. truck-trailer combination with different cornering stiffness.
COMPARATIVE STUDIES OF 18 M TRUCK-FULL TRAILER AND TRACTOR-SEMI-
TRAILER-FULL TRAILER COMBINATIONS WITH DIFFERENT LENGTH DISTRIBUTION

Three different ratios between the length of front and rear load carrier were investigated. In all cases the total length of 18 m and the Swedish weight limit were almost reached.

As the comparisons were primarily to be made between truck-full trailer and tractor-semitrailer-full trailer combinations the total mass of the loads has been equal for both combinations in each length distribution. However, the difference in total mass between the different length distribution variants of one type of combination was very small. Thus a comparison also within one type of combination is possible. See appendix fig A2 for further data.

Shortenings will be used in the following sections. For the tractor-semitrailer-full trailer (T-ST-FT) they are:

1 short semitrailer and long full trailer - short-long combination
2 semitrailer and full trailer of equal length - equal length combination
3 long semitrailer and short full trailer - long-short combination

The same shortenings will be used for the corresponding truck-full trailer (T-FT) combinations.

Comparison between truck-full trailer combinations

Every risk criterion based upon side slip angles, overturning risk, lateral side deviations, rearward amplification of risk factors etc, showed that the short-long combination was better than the others. See figure 11.1

Usually, the equal length combination had smaller risk factor values in course section C. However, these maximum values were smaller than in course section B where the short-long combination was more favourable. In addition the short-long combination has better oscillatory damping. This is also the case for rearward amplification of the risk factors. Similar results have been found from other investigations.

The short-long combination is therefore to be recommended.
11.2 Comparison between tractor-semi-trailer-full trailer combinations

The largest risk factor values were found for the long-short combination (see figure 11.1). The comparison between the short-long and the equal length combination showed that the short-long combination generally seen was more favourable in terms of risk factor maxima and rearward amplification of the risk factors. However, the overturning risk of the semitrailer axle was larger on the short-long combination to such an extent that the equal length combination is judged to be safer.

11.3 Comparison between truck-full trailer and tractor-semi-trailer-full trailer combinations

In this comparison (see figure 11.1) it can be noted that the tractor-semi-trailer-full trailer combinations show the largest risk factors in the critical middle part (B) of the manoeuvre and at least in case of the side slip angles have a larger rearward amplification.

It is noteworthy that the time histories of lateral accelerations, side slip angles, overturning risk factors, side deviations and in some cases even the angles between the vehicle units are very similar for the rear units and axles with the same index number independent of the type of combination. This supports the hypothesis that the number of foregoing articulations is of significant importance for the behaviour of the vehicle unit. See chapter 14 for further discussion.
Figure 11.1 Maximum absolute values of sideslip angle and lateral deviation for each axle in different 18 m combinations. T-FT is short for truck-full trailer and T-ST-FT means tractor-semi-trailer-full trailer.
12 INFLUENCE OF LOAD DISTRIBUTION AND WEIGHT

Vehicle and load data concerning this chapter are summarized in appendix, table A1 and figure A1.

12.1 Axle load distribution of the trailer in a 24 m truck-full trailer

In order to study the influence of load magnitude and distribution, three load configurations were compared.

1. Truck and trailer fully loaded
2. Truck fully loaded and trailer with full load on the front axle and empty axle load on the rear axle
3. Truck fully loaded and trailer with full load on the rear axle and empty axle load on the front axle

The greatest side slip angle appeared for the front loaded combination at the trailer front axle. The skid motion was similar to the start of a front end trailer swing.

The overturning risk was smallest for the front loaded trailer and greatest for the fully loaded trailer. It has to be observed however that the rear end load was much larger than the front end load. Thus, the size of the loads was possibly more important to this result than the position of the loads.

It should be noted that the trailer frame has been considered as rigid. Thus, the effective roll stiffness was higher than what can be expected in reality.

12.2 Load location on the semitrailer of a 24 m tractor-semitrailer-full trailer

To study the influence of load location on the dynamics of the vehicle, a specific load was located in three different positions on the semitrailer while the trailer was fully loaded.
The three load positions were

1. Load centre of gravity at the fifth wheel kingpin
2. Load centre of gravity between kingpin and semitrailer axle with the same load distribution as when fully loaded
3. Load centre of gravity at the semitrailer axle

The rear loaded combination showed the poorest course tracking, jerky steering and the highest risk factor values.

The middle loaded combination generally had the best performance except in the initial section A of the test course. As the highest risk factor values occur in the middle section B the middle load location can still be said to be the best one. It should be noted that the judgements were based upon the full trailer behaviour as it had the largest risk factors of the combination.

Load location on the full trailer of a tractor-semitrailer-full trailer

Simulations were also made with a fully loaded semitrailer and a partly loaded full trailer with a specific load which was located in three positions:

1. Load centre of gravity at the front axle
2. Load centre of gravity between front and rear axle with the same load distribution as when fully loaded
3. Load centre of gravity at the rear axle

The behaviour of the tractor and the semitrailer was not affected by the load distribution on the trailer. Thus the behaviour of the trailer was closely related to the load location.

The rear loaded combination behaved worse than the front loaded and the middle loaded combination was the most favourable of the three configurations.
12.4 Load weight and load distribution between semitrailer and full trailer in a 24 m combination

Fig 12.1 shows some results from four simulations that were carried out with the following load configurations.

1. Semitrailer partly loaded
   Full trailer partly loaded symbol DD
2. Semitrailer fully loaded
   Full trailer partly loaded symbol FD
3. Semitrailer partly loaded
   Full trailer partly loaded symbol DF
4. Semitrailer fully loaded
   Full trailer fully loaded symbol FF

It is noteworthy that the FD combination behaved better than the DF combination although the FD load was clearly heavier than the DF load. Therefore, unloading should start at the rear trailer.
(see text)

<table>
<thead>
<tr>
<th>Combination symbol</th>
<th>DD</th>
<th>FD</th>
<th>DF</th>
<th>FF</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load weight (kg)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Totally</td>
<td>15200</td>
<td>26510</td>
<td>20850</td>
<td>32160</td>
</tr>
<tr>
<td>Semitrailer</td>
<td>8360</td>
<td>19670</td>
<td>8360</td>
<td>19670</td>
</tr>
<tr>
<td>Full trailer</td>
<td>6840</td>
<td>6840</td>
<td>12490</td>
<td>12490</td>
</tr>
<tr>
<td>Weight of complete combination (kg)</td>
<td>34100</td>
<td>45410</td>
<td>39750</td>
<td>51060</td>
</tr>
<tr>
<td>Loads (N)</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fifth wheel</td>
<td>54550</td>
<td>96900</td>
<td>54550</td>
<td>96900</td>
</tr>
<tr>
<td>Axle no 12</td>
<td>53800</td>
<td>63480</td>
<td>53800</td>
<td>63480</td>
</tr>
<tr>
<td>34</td>
<td>64390</td>
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<tr>
<td>910</td>
<td>63950</td>
<td>63950</td>
<td>92110</td>
<td>92110</td>
</tr>
</tbody>
</table>

a) Load configuration and axle loads

b) Sidestep angle

c) Overturning risk

d) Lateral deviation

Figure 12.1 Maximum absolute values of different risk factors. Tractor-semi-trailer-full trailer 24 m combination with different loads according to the table (a).
INFLUENCE FROM SPEED

In this study the lateral movement as a function of time was kept unchanged and independent of speed. This means that the centre of gravity in the leading vehicle was exposed to the same road oriented lateral acceleration - time function at all speeds. It also means that the test course was expanded longitudinally with increasing speed.

The following combinations were studied. Vehicle data appear in appendix table A1.

1. The field test combination i.e. the tractor-semitrailer-full trailer combination used at the validation simulations.

2. A typical Swedish 24 m truck-full trailer combination, fully loaded (fig A1).

3. A typical Swedish 24 m tractor-semitrailer-full trailer combination, fully loaded (fig A1).

Compared to combination 1, combination 2 and 3 have higher centre of gravity and slightly higher tyre cornering stiffness.

For combination 1 simulations were made at 40, 70, 90 and 110 km/h. With combinations 2 and 3 simulations were made at 40, 70 and 90 km/h.

Some of the simulation results for combination 1 are shown in figure 13.1. The risk factor amplification at the rear end of the combination was negligible at 40 km/h, quite important at 70 km/h and then successively increasing at higher speeds. At 110 km/h, wheel lift and severe rear end skid occurred for the full trailer at the end of the manoeuvre.

The comparative simulations with vehicle combination 2 and 3 are illustrated by figure 13.2 and 13.3. Figure 13.4 shows the speed dependency of the steer angle for the full trailer in combination 3. Noteworthy is the great increase in amplitude between 70 km/h and 90 km/h.

The simulations showed that increased speed has a strong negative effect on the dynamic behaviour of vehicle combinations with more than one articulation point.
Figure 13.1 Maximum absolute values of different risk factors. Field test combination at four different speeds. Lack of values for 110 km/h in course section C is due to overturning of the full trailer. Graphical interpolation between speeds is indicated in a.
Maximum absolute values of sideslip angles for each axle and overturning risk maxima for each vehicle unit. Fully loaded 24m combinations at different speeds. Graphical interpolations is possible like in figure 13.1a. T-FT is short for truck-full trailer and T-ST-FT means tractor-semi-trailer-full trailer.
Figure 13.3  Speed influence on lateral deviation. Fully loaded 24 m combinations. Graphical interpolation possible like in figure 13.1.a. T-FT is short for truck-full trailer and T-ST-FT means tractor-semitrailer-full trailer.

Figure 13.4  Steer angle of the full trailer (AFLO) as function of time. Fully loaded 24 m tractor-semitrailer-full trailer combination at different speeds.
INFLUENCE FROM NUMBER OF ARTICULATIONS

The possibility for vehicle movements that are uncontrollable for the driver, will increase with the number of articulations and degrees of freedom. In addition, all simulations at high speed showed the largest risk factor values for the highest articulation number among comparable vehicle combinations. See figures 11.1, 13.2 and 13.3.

Another argument against articulation concerns the driver's perception of the rear vehicle unit dynamic behaviour. Although the rear unit appears to be the most critical unit at high speed, it offers the poorest sensory feedback to the driver.

Rearward risk factor amplification due to added articulations may occur even in steady-state cornering. This phenomenon is indicated by fig 14.1 where the outwards off-tracking increases when one articulation is added. If the side forces and the sideslip angles ($\delta$) are unchanged the outer track radius ($R_y$) will increase when the rigid trailer (a) is substituted by an articulated trailer (b).

Thus, it must be emphasized that the number of free articulations should be minimized for vehicle combinations operating at high speed.

Figure 14.1 Outwards off-tracking increase when a rigid vehicle unit (a) is substituted by an articulated unit (b). See text.
PROPOSED DEMANDS ON DYNAMIC STABILITY OF HEAVY VEHICLE COMBINATIONS

The following test procedure and demands are proposed on basis of the theoretical and experimental investigations. These demands are considered to be minimum performance regarding dynamic stability during non-braking conditions. The vehicle combination has to carry out a double lane change manoeuvre with specifications according to the simulations of this investigation. The test is to be done with maximum load and maximum load centre of gravity height at a constant speed of at least 70 km/h. For acceptable performance the following limit values must not be exceeded.

1. The side slip angles of any axle except the front axle of the leading vehicle must not exceed 150 m rad (8.6 degrees)

2. The side slip angle maxima for all axles must not exceed 20 m rad 75 metres after the point where the first axle of the combination has returned to straight course

3. The amplification of the axle side slip angle maxima relative to the mean value of the side slip maxima of the axles of the leading vehicle must not exceed 2.

4. The maximum space requirement during the manoeuvre must not exceed the limits defined by side deviation limits for the axle centres according to figure 14.1. This corresponds to the requirement that the combination must stay within the road limits of a seven meter wide road, typical for Swedish conditions, and not touch an obstacle, ten metres long and occupying the right hand side of the road.

5. During the manoeuvre the overturning risk (RV, see chapter 6.3) for any axle must not exceed 1.0.
MEASUREMENTS OF STATIC OVERTURNING LIMITS

In order to get empirical data on the overturning stability for heavy vehicle combinations, static tests were carried out. The full scale, loaded vehicles were inclined until the gravity force \((m \cdot g)\) caused overturning. An inclination angle \(\alpha\) corresponds to steady state cornering at the lateral acceleration \(g \cdot \tan \alpha\) with the vehicle real mass reduced to \(m \cdot \cos \alpha\). The measurements were performed on a tilting device consisting of a number of wheel support beams, one for each axle. The beams are tilted by hydraulic jacks.

Subject for investigation were two tractor-semitrailer combinations, two full trailers and one truck. All vehicles were maximum loaded at different centre of gravity heights. The tested vehicles were considered as representative for modern heavy vehicles in Sweden. However, the roll stiffness was comparatively high especially for the semitrailers and trailers. Results are shown in figure 16.1.
A demand on minimum static overturning stability has been proposed. It is regarded as a complement to requirements at the dynamic computer simulation test. It is also simple to supervise. The proposed limit value was assessed with the aid of experimental results according to Fig 16.1.
HIGH SPEED OFF-TRACKING

Low speed off-tracking is a well-known problem for long vehicle combinations in sharp curves.

At high speed and large sideslip angles off-tracking towards the outside of the curve will occur. This phenomenon is probably less known to the drivers than the "classic", low speed off-tracking. Furthermore, it is often impossible for the driver to observe the outer track of the rear vehicle. Accidents have occurred in Sweden that seem to be caused by this kind of off-tracking.

The outside off-tracking can be reduced by shortening the combination, reducing the number of free articulations, using tyres with high cornering stiffness etc. Unfortunately, the common method to reduce low speed off-tracking (i.e. articulation) will also increase the high speed off-tracking. See fig 14.1.

PROPOSED COMPLEMENTARY DEMANDS ON OVERTURNING STABILITY AND HIGH SPEED OFF-TRACKING

Overtuming stability

As a complement to the dynamic overturning stability requirement, $4 \text{ m/s}^2$ was proposed as the minimum static overturning stability. In fact, this means a limitation of the load and its c.g. height, to be specified for each vehicle.

High speed off-tracking

The proposed limitation of dynamic space requirements in the double lane change manoeuvre (fig 15.1) was completed by following demand. Off-tracking towards the outside of the curve must not exceed 0.5 m in a curve defined by a speed of 70 km/h and a lateral acceleration $2 \text{ m/s}^2$. The lateral acceleration should be maintained during five seconds. During the test, the vehicles should carry maximum load with c.g. at maximum height.
TEST METHODS

The following test methods were suggested for type approval

1. Mathematical simulation of the double lane change manoeuvre and the off-tracking test
2. Full scale static overturning test

DISCUSSION ON DEMANDS ON DYNAMIC STABILITY DURING BRAKING

Following demands on braking behaviour of heavy vehicle combinations have been considered desirable.

The vehicle combination must not lose stability and manoeuvrability when maximum pedal force is applied. At the same time good braking efficiency should be maintained. This must be fulfilled in empty and full load condition, on high and low friction surfaces and on even and uneven roads.

It can directly be stated that antilock braking systems will be needed to fulfill these demands. Thereby the longitudinal slip distribution between the axles during braking can be expected to be a major design problem for these systems. Further research in this area seems to be needed. However, important investigations have been performed.

SUGGESTIONS FOR FUTURE RESEARCH

The relatively poor overturning stability for heavy vehicles and the rearward risk factor amplification in articulated vehicles are the major problems, that are revealed in this paper, in the main report and its summary. Novel design principles have been suggested. The compromise between low speed and high speed handling requirements may be fulfilled by speed-dependent braking of relative motions in articulations and by unconventionally steered axles. In order to develop these ideas, following investigations are now active at the institute.
1. Handling characteristics of articulated heavy vehicles. The mathematical model mentioned in this paper will be expanded for steering axles. Low speed and high speed off-tracking as well as different steering strategies for the axles will be considered.

2. Overturning risk for heavy vehicles. Measurements in vehicles used in real traffic and driver interviews have been performed. Driver estimated overturning risk will be compared to the measurements.

3. Overturning risk due to sloshing in vehicles with liquid loading. The overturning risk for a simple vehicle model is calculated in an analogue computer. Liquid force input is fed to the computer from a laterally moving physical tank scale model.
NOTATION

a  Fifth wheel or tow pin distance behind c.g. of the leading vehicle

aij Forward articulation point distance in front of c.g. of indexed vehicle unit

b  Distance from rear axle to c.g. of the leading vehicle

c.g. Centre of gravity

f  Distance from c.g. to front axle of the leading vehicle

g  Acceleration of gravity (9.81 m/s²)

l  Distance from foremost articulation point to the axle for a trailer or a dolly

m  Mass of vehicle unit

q  Distance between articulation points for a semitrailer or dolly

RV Overturning risk

SA Lateral acceleration (with index according to vehicle unit). Direction is perpendicular to the longitudinal symmetry plane of the unit

T-FT Abbreviation for truck-full trailer

T-ST-FT Abbreviation for tractor-semitrailer-full trailer

Y Road fixed coordinate perpendicular to the initial velocity vector of the vehicle

δ Side slip angle

Index

12, 34, 56, 78, 910 Axle index or vehicle unit index

14, 16, 58, 710 Index for leading vehicle, tractor-semitrailer, full trailer
REFERENCES


APPENDIX A

A.1 General vehicle data

Every vehicle parameter necessary for simulation was measured on the vehicle combinations that were used in the field tests. Special apparatus were developed for measurement of moments of inertia, e.g. height, roll stiffness, roll damping, effective track width etc. Almost every parameter connected to roll dynamics was evaluated from experiments in the tilting device mentioned in chapter 16.

Experience from these measurements was valuable for the assessment of reasonable parameter values to the 18 m and 24 m combinations. Specific formulas were developed where all parameter values could be calculated from the desired design parameters. Table A1, figures A1 and A2 show some parameters for the investigated vehicle combinations.

A.2 Tyre side force measurement

The side forces as a function of wheel load and side slip angle have been measured for the tyres on the vehicles used in the field test. The measurements were carried out on wet asphalt at low speed (≈10 km/h) by a special technique. Two trucks were driven, laterally interconnected by a cable and a force transducer. The method is illustrated in fig A3. An example of measured tyre data is shown in fig A4.
Table A1. Vehicle combinations tested in simulations

<table>
<thead>
<tr>
<th>Refer to text chapter no.</th>
<th>Vehicle combination</th>
<th>Distances according to fig A1 (m)</th>
<th>Truck or semitrailer load configuration (fig A1) and load weight</th>
<th>Full trailer load configuration (fig A1) and load weight</th>
<th>Gross weight (kg)</th>
<th>Load c.g. above load platform (m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>9.1; 10</td>
<td>Low c.g. (T-FT)</td>
<td>21.32 3.40</td>
<td>0.60</td>
<td>9.09</td>
<td>1.94* 3.34* 4.15</td>
<td>Fully loaded 17680 kg</td>
</tr>
<tr>
<td>10</td>
<td>Low c.g. (T-FT)</td>
<td>15.70 5.26</td>
<td>+2.95</td>
<td>3.34</td>
<td>4.15</td>
<td>Fully loaded 11435 kg</td>
</tr>
</tbody>
</table>

11 18 m 15.80 Six different configurations according to fig A2

12.1 24 m T-FT

Front loaded 6600 kg
Rear loaded 10920 kg
Fully loaded 17540 kg

12.2 24 m T-St-FT

Front loaded 8360 kg
Rear loaded 12490 kg

12.3 24 m

Front loaded 13250 kg

12.4 (DD)

Front loaded 13250 kg

12.6 24 m

Front loaded 13250 kg

12.7 24 m (FD)

Front loaded 13250 kg

12.8 24 m (DF)

Front loaded 13250 kg

12.9 (FF)

Front loaded 13250 kg

* Different values in chapter 10

TRUCK - FULL TRAILER (T-FT)

TRACTOR - SEMI TRAILER - FULL TRAILER (T-ST-FT)

Fig A1. Scale drawings of the 24 m combinations with different load configurations.
General notation for distances in table A1.
Fig A2: Measures, weights and axle loads for 18 m combinations according to chapter 11. a: "Short-long" b: "Equal length" c: "Long-short"

The load masses are homogenous, width 2.4 m, height (above load platform): 1.0 m.
Figure A3. Tyre side force measurement

Figure A4. Example of measured side force characteristics. Radial tyre, Dimension 11.00-20. Tyre pressure 6 bar. Wet asphalt.