Caravans or trailers with an electrically powered support axle

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by

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Publication of the Federal
Highway Research Institute

Imprint

Section:
Active Vehicle Safety and Driver Assistance Systems

Publisher:
Federal Highway Research Institute
Brüderstraße 53, D-51427 Bergisch Gladbach
Telefon: +49 (0 22 04) 43 - 0

Design concept:
MedienMélange:Kommunikation

English translation provided by:
Language Services Division of the Federal
Ministry for Digital and Transport (BMDV)

Bergisch Gladbach, January 2024

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Abstract

This project report describes an investigation into the influence powered trailers may have on the driving dynamics of vehicle combinations compared with non-powered trailers.

Theoretical reasoning suggests that a destabilizing effect on the towing vehicle around the yaw axis may result in particular from shear forces and longitudinal forces with a pushing effect, which are transferred from the trailer to the coupling while, in contrast, longitudinal forces with a pulling effect are generally expected to have a stabilizing effect. Due to the application of the same driving force to the wheels on both sides of the trailer as long as there is always a resulting tractive force, the drive system is not expected to influence the shear force at the trailer coupling.

Bearing this in mind, trailers with a symmetrical driving force that is lower than the driving resistance force of the trailer do not have a significant negative impact on the dynamics of the vehicle combination.

As the required tractive effort of the towing vehicle is lower when the trailer is powered, the driveability of the vehicle combination is expected to improve.

To verify these two assumptions, the stability at high speeds and the swerving behaviour were identified as decisive criteria and were assessed by performing a standard defined steering impulse test and a standard double lane change manoeuvre respectively. In addition, the expected lateral forces were quantified by means of simple equations for trailers with and without active drive.

The theoretical considerations were confirmed in driving tests. A slightly reduced sway control and characteristic speed were found in one of the two trailers when an axle was powered; no relevant differences were found in the other trailer. Driveability was not found to be worse for either of the trailers with active drive compared to the same trailers without active drive; in one trailer, driveability was found to have improved significantly when the drive was active. The simple equations to quantify the lateral forces were confirmed by measurements.

Based on the results, vehicle safety is not considered to be compromised if powered trailers are designed so that the driving forces are distributed evenly between both wheels and the trailers do not push the towing vehicle.

Uneven distribution of the driving forces could be used for active stabilization; however, as torque vectoring was not available in the vehicles, no such assumptions could be established and tested.

1 Introduction and explanation of the need for investigation

1.1 Background information: Why powered trailer axles?

By definition[1], trailers are non-powered vehicles designed to be attached to powered vehicles. This means that the traction required for a combination (up to three vehicles in Germany based on the regulations on longer goods vehicles) to overcome the driving resistance can only be provided by the frontmost vehicle, the towing vehicle.

The question therefore arises whether, from a scientific point of view, i.e. from a technical point of view and from the point of view of driving dynamics, it would make sense to power trailers. This could be achieved in principle by

- every trailer only powering itself (and partially, or ideally, fully compensating its driving resistance, e.g. drag, so that the coupling devices remain subject to tensile loads), or
- individual trailers providing more power than necessary to overcome their driving resistance (with coupling devices now also subject to compressive loads caused by individual trailers pushing).

The benefit of powered trailers in both cases would be increased traction of the combination and thus the ability to climb gradients even in adverse friction conditions, the ability to distribute and use energy storage more efficiently, and the ability to recuperate energy more efficiently, i.e. to make better use of braking energy. This could improve traffic flows (for example, by avoiding disruptions caused by combinations stranded on motorway gradients in winter) as well as energy efficiency (through better energy recovery). Installing additional energy storage units in the trailer could also increase the range of the combination.

However, this would require precisely controllable electric trailer drives.

1.1.1 Traction and theoretical gradeability

With the gradient angle α and the total mass of the combination m_{ges}, the required tractive force at a gradient is

$$F_{\mathrm{z,erf}} = m_{\mathrm{ges}} \cdot g \cdot \sin\alpha$$

For small angles, the angle α can be replaced by the dimensionless gradient measure[2] q:

[1] R.E.3 (ECE/TRANS/WP.29/78/RE3): 'Trailer' means any non-self propelled vehicle, which is designed and constructed to be towed by a power driven vehicle and includes semi-trailers.

[2] Corresponds to the gradient given as a percentage, but divided by 100%.

$$F_{\mathrm{z,erf}} = m_{\mathrm{ges}} \cdot g \cdot q.$$

The force that can be transmitted from the drive axle depends on the longitudinal friction coefficient μ and the drive axle mass m_{AA}:

$$F_{\mathrm{z,ü}} = m_{\mathrm{AA}} \cdot g \cdot \mu.$$

The maximum gradeability (maximum transmittable tractive force is exactly compensated by the downhill force) of a vehicle as a function of load distribution, friction coefficient and gradient thus becomes

$$q_{max} = \frac{m_{\mathrm{AA}}}{m_{\mathrm{ges}}} \cdot \mu.$$

This means that the maximum gradeability depends on the ratio of the mass on the drive axle to the total mass of the vehicle as well as on the friction coefficient.

By introducing the traction coefficient

$$\tau = \frac{m_{\mathrm{AA}}}{m},$$

which describes the percentage of the total load accounted for by the drive axle load, the maximum drivable gradient is simplified to

$$q_{max} = \tau \cdot \mu.$$

The drivable gradient is found to increase proportionally to the traction coefficient for a given friction coefficient. Another way of looking at it is that for a given gradient a decreasing friction coefficient can be compensated by increased traction.

Common traction coefficients of heavy goods vehicle combinations are in the range of around 0.25 (and even lower if not optimally loaded); motorway gradients are usually not greater than 7% (q=0.07). This means that a friction coefficient of 0.28 is required to drive up the gradient. If it were possible to increase the traction coefficient to about 0.75 by powering the trailer axles, a significantly lower friction coefficient of about 0.1 would be sufficient to drive up the gradient. The same applies accordingly to passenger car combinations: While passenger cars alone have a traction coefficient of approximately 0.5 (when one axle is driven), this value also decreases to 0.25 (mass of the trailer equals mass of the towing vehicle), depending on the mass of the trailer attached.

1.1.2 Increasing energy efficiency and recuperation capacity

Dividing the driving force between the individual units of a vehicle combination means that the towing vehicle does not have to carry oversized energy storage when driving solo or, conversely, prevents the energy storage from being undersized when driving with trailers. Like the concept that each trailer (category O2+) should be capable of slowing its own mass, each trailer in this case would be compensating for its own driving resistance or at least overcoming part of it.

Trailers with electric drives would also be able to recuperate their own kinetic energy in batteries. This energy would otherwise be converted to thermal energy in the existing trailer braking systems, which are installed anyway, and not used for other purposes.

1.2 State of the art

The advantages of trailers with active (electric) drives described in section 1.1 have led to the development of prototypes and demonstrators of different vehicle categories with different drivetrain concepts such as the following:

Vehicle category	Name
O2	Dethleffs eHome
O2	Weinsberg (KnausTabbert)
O3, O4	Electric trailer axle by Bosch (https://www.springerprofessional.de/schwere-lkw/elektrofahrzeuge/bosch-elektrifiziert-die-achse-von-lkw-sattelanhaengern/16077896)
Bicycle trailer	Pusher trailers, various (https://en.wikipedia.org/wiki/Pusher_trailer)

Table 1: Overview of concepts

As trailers have to date explicitly been defined as non-self-powered vehicles, no approved serial products exist. An initiative by CLCCR, the non-profit International Association of the Body and Trailer Building Industry, aims to adapt the UNECE regulatory framework to powered trailers, see for example Informal Document GRSG-121-34.

To our knowledge, apart from powered trailers (pusher trailers) for bicycles, there are no prototypes or concepts where trailer drives would overcompensate driving resistance – motor vehicles trailers therefore never push the towing vehicle.

1.3 Explanation of the need for investigation

The benefits of powered trailers for traffic flow and energy efficiency were described in section 1.1. In consequence, prototypes and initiatives were developed, which are described in excerpts in section 1.2. However, the boundary conditions which must be observed by powered trailers in order to not endanger road safety are unclear.

The aim of the underlying research for this report was therefore first of all to assess whether and, if so, in which operational areas, powered axles of trailers can negatively influence the driving dynamics of vehicle combinations. In a next step, specific requirements for the behaviour of the drive axles were to be derived from these findings. The project was also to identify other potential challenges posed by powered axles.

For the research work, BASt was able to obtain two prototypes (O2 trailers) for driving tests in a short time frame.

The work described was coordinated in advance with Division StV 22 of the Federal Ministry for Digital and Transport (BMDV). The vehicles were provided via the Caravan Industrieverband Deutschland, CIVD e.V.

2 Methodology and tools

The aim of the research, as explained in section 1.3, was to identify road safety risks which could arise from powered axles of trailers (for two-track vehicles) and, where appropriate, to determine what requirements should be made for trailers to prevent such risks.

For this purpose, it was necessary to analyse the driving dynamics of trailers, describe the dependencies of the relevant physical variables and derive the relevant criteria for road safety. Finally, the findings were verified and supplemented with driving tests.

2.1 Fundamentals of driving dynamics

2.1.1 Stability of the vehicle combination

The challenge in terms of driving dynamics when operating a vehicle with a trailer is a possible destabilization of the towing vehicle around the yaw axis, mainly due to shear forces introduced at the trailer coupling and longitudinal forces pushing the towing vehicle (by increasing the yaw angle of the towing vehicle). Pulling longitudinal forces around the yaw axis stabilize the yaw movement (by reducing the yaw angle of the towing vehicle).

> Note: In the case of single-track vehicles, destabilization around the rolling axle, which is already statically unstable, would also have be investigated; <u>single-track vehicles and their trailers are therefore not included in these investigations</u> (parallel work on this is underway at the Federal Highway Research Institute [BASt]).

First of all, it is assumed that trailers should not transmit any compressive forces to the towing vehicle during normal operation. This could be ensured by the requirement to always remain in traction mode, even when the trailer is powered, and by appropriately assuring functional safety.

Under ideal conditions, one can then assume that <u>additional</u> shear forces result from an active trailer drive in the following cases:

- during steady-state circular driving (where shear forces are expected to be identical with and without active drive),
- during curve braking (where no driving force is applied to the trailer axle, so that no influence of the drive is expected here either) as well as
- dynamically due to sway movements at higher speeds.

It is assumed that influences from powered trailers are most likely to be exhibited in sway movements at high speeds.

While shear forces will show during curve braking, cornering and curve acceleration, their impact will not be greater than in non-powered trailers (assuming correct system function without a pushing trailer, which is then, in turn, a functional safety issue).

Due to the large variety of trailer designs, the initial aim is to describe the influence of the driving force of non-steering trailers (rigid drawbar trailers) on the coupling forces, assuming that driving safety is not impaired when the shear forces resulting from the active drive are lower in the respective direct comparison (with/without active drive) over time.

For the theoretical considerations the unicycle model is used, in which the force ratios of the two wheels are projected onto a single wheel. Furthermore, the angles should be low enough so that the cosine of the articulation angle becomes 1; the sine of the articulation angle then corresponds to the angle itself (in radian measure).

The conditions during high-speed sway movements are shown in Figure 1.

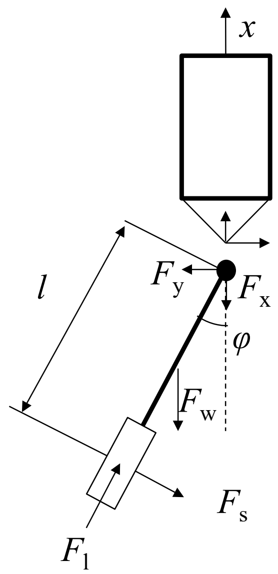

Figure 1: Forces at the trailer coupling for a trailer at an articulation angle, with longitudinal wheel force F_l, lateral wheel force F_S, driving resistance F_w, shear force at the trailer coupling Fy and longitudinal force at the trailer coupling F_x. The figure shows the aforementioned unicycle model which assumes that both wheels are subject to equal driving forces.

Neglecting the low (f≈0.02) mechanical friction in the wheel, tyre and axle bearings, as well as for unaccelerated travel, driving resistance is

$$F_w = \frac{\rho_{Luft}}{2} \cdot c_W \cdot A \cdot \dot{x}^2,$$

where ρ represents the air density, A the front face and c_w the drag coefficient. The lateral force at the wheel (calculated here for the unicycle model) corresponds to the shear force at the trailer coupling; as a function of the articulation angle φ and its derivation with the skew stiffness k_α and the gross vehicle weight (vertical load is neglected as it is minor in comparison with the weight of the trailer, e.g. 50 kg vs. 2000 kg at constant speed), it is

$$F_S = k_\alpha \cdot m \cdot g \cdot \left(\varphi + \frac{l}{\dot{x}} \cdot \dot{\varphi}\right) = F_y,$$

and the longitudinal force (positive for towed trailer) becomes

$$F_x = \frac{\rho_{Luft}}{2} \cdot c_W \cdot A \cdot \dot{x}^2 - F_l.$$

It is thus clear that when the <u>driving forces</u> on the wheels <u>are equal</u> and the trailer does not push, the drivetrain does not exert any influences lateral to the trailer coupling which could destabilize the towing vehicle.

The yaw axis of the slightly articulated towing vehicle could still be destabilized if compressive forces (F_x negative) are introduced at the trailer coupling. This always happens when the driving force exceeds the driving resistance.

An active influence on the stabilization is possible if the driving forces on the individual wheels can be controlled separately, as shown in Figure 2.

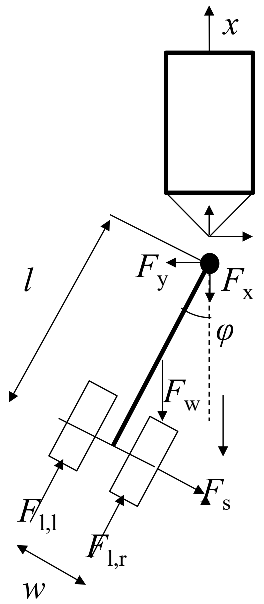

Figure 2: Forces acting on the trailer coupling of an articulated trailer. As above, with additional left longitudinal force on wheel $F_{l,l}$, right longitudinal force on wheel $F_{l,r}$ and track width w.

This requires a deviation from the unicycle model. The shear force on the trailer coupling then becomes

$$F_y = F_s + F_{l.links} \cdot \frac{w}{2l} - F_{l,rechts} \cdot \frac{w}{2l}.$$

Direct control of the shear force might therefore be accomplished by controlling the difference in longitudinal forces on the wheel.

The aim of the driving tests with the test vehicles was to confirm the above calculations as far as possible. If successful, the equations could be used to derive requirements.

Unfortunately, it is not possible to confirm the equations with different forces per wheel, also referred to as torque vectoring, because this is not implemented in the test vehicles.

Controllability of the vehicle combination by the driver

If the trailer is powered, this will change the driving behaviour of the vehicle combination. The impact of these changes for drivers is easy to assess under the given boundary conditions, i.e. short test time for the respective trailers, in the closed-loop test. For example, the driveability can be investigated using the double lane change test, see section 2.2.2.

2.2 Driving tests

2.2.1 High-speed stability

The standard test for the high-speed stability of a combination is the defined steering impulse at constant driving speed in accordance with ISO 9815:2010 “Road vehicles – Passenger-car and trailer combinations - Lateral stability test”. In this test, a defined steering impulse is given at constant driving speed in such a way that the maximum lateral acceleration of the trailer amounts to between 3 m/s² and 5 m/s² and the combination retains its original direction of movement after the defined steering impulse has been given. The extreme steering movement causes the trailer to oscillate. Depending on the level of self-damping, the internal oscillation abates faster or slower.

A simple assessment criterion is to determine whether there are differences between the swaying oscillations with and without a trailer. This requires the excitation and driving speed to be exactly the same. If successful, statistical methods can be applied to check whether the time curves differ significantly between trailers with and without active drive. At least 5 runs were carried out for each configuration (driving speed, mode, vehicle). A sample measurement is shown in Figure 3. From the data, the natural frequency and damping can be deduced based on the first two oscillation cycles after the end of the excitation, see Figure 4.

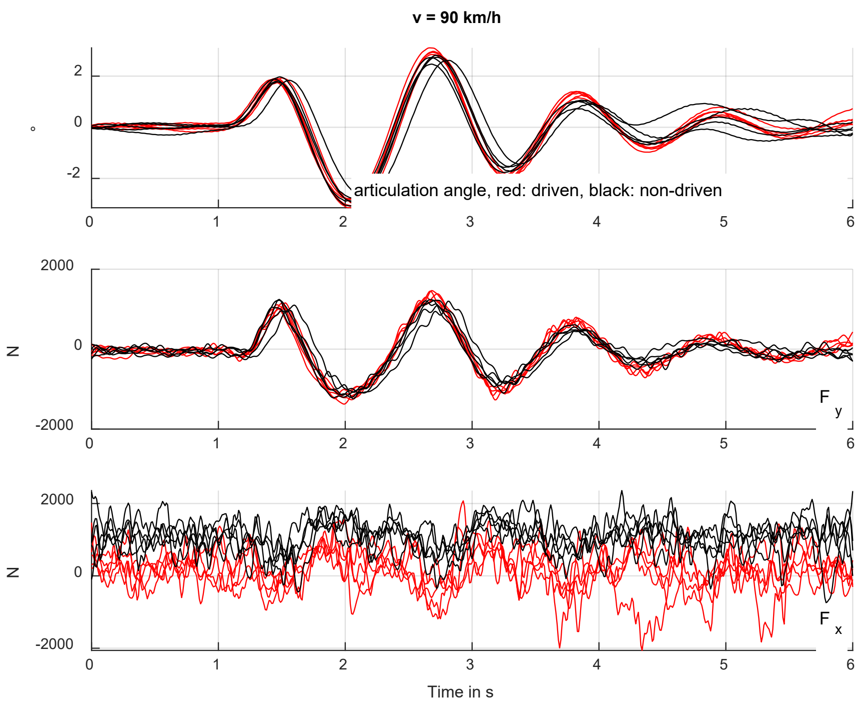

Figure 3: Sample measurements (Manufacturer A vehicle) of articulation angle, shear force and longitudinal force in the drawbar at a driving speed of 90 km/h after a defined steering impulse according to ISO 9815 has been given.

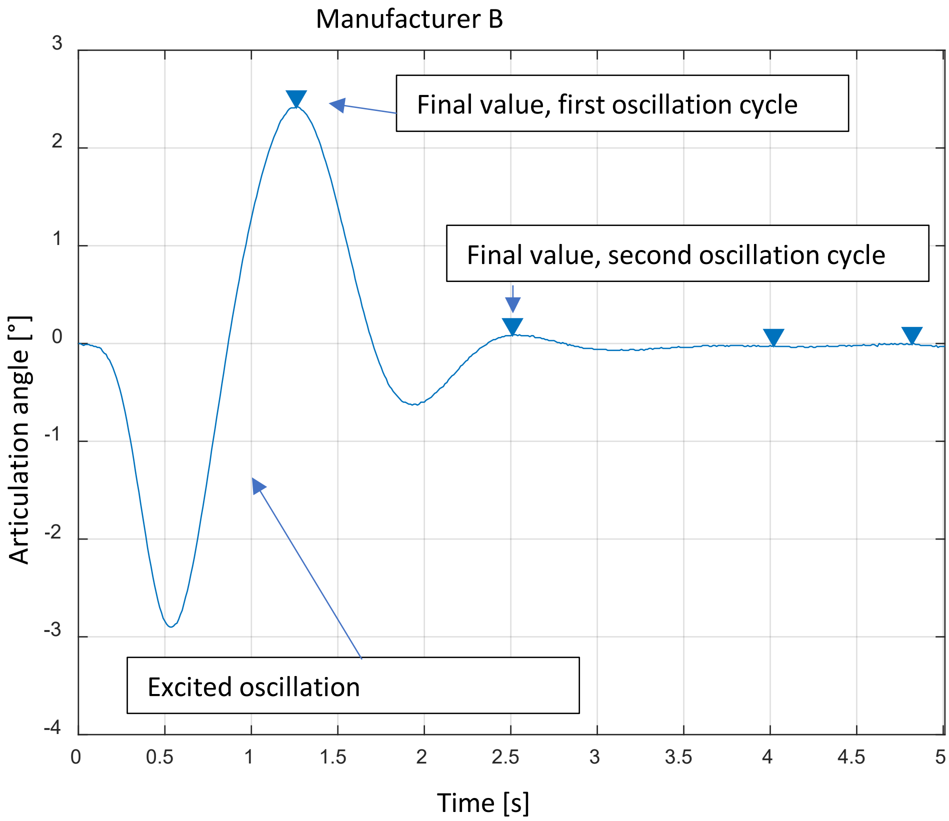

Figure 4: Oscillating cycles and articulation angles, Manufacturer B example

The evaluation is based on ISO 9815:2010, but only for these first two oscillation cycles. The frequency can be calculated as

$$f = \frac{1}{t_{max,2} - t_{max,1}},$$

the damping D requires the amplitude ratio

$$r = \frac{1}{2}\frac{\widehat{\varphi_1} + \widehat{\varphi_2}}{\widehat{\varphi_2}}$$

and is thus calculated as

$$D = \frac{\ln r}{\sqrt{\pi^2 + (\ln r)^2}}.$$

As a rule, the damping decreases with increasing driving speed; the parameter for stability is then the intersection between the regression line through the damping as well as the x-axis. For safety reasons and partly because it was requested by manufacturers, the tests ended at speeds above 85 km/h (Manufacturer B; no drive support at speeds above 85 km/h) and 95 km/h (Manufacturer A).

Controllability of the vehicle combination by the driver

The setup according to ISO 3888-1:2018 was selected for the test, see Figure 5. The input parameter is the driving speed. (To be able to better reproduce them, the tests described were carried out at a constant driving speed. However, the ISO standard also allows the entry speed to be defined in combination with the engine brake after passing the first lane of cones). The test is considered passed if the vehicle drives through the lane without knocking over one of the cones.

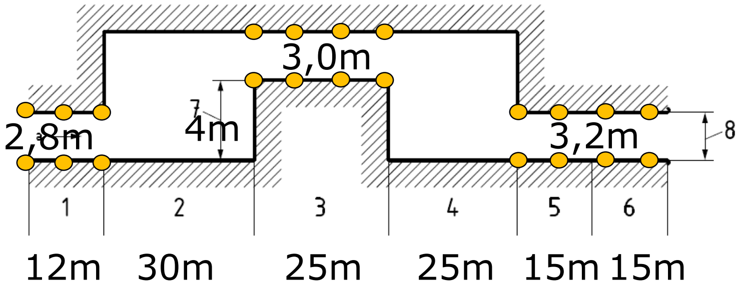

Figure 5: Double lane change test setup

For reasons of consistency and to take the learning effect into account, it was also defined here that three attempts are available for each speed, at least one of which must be passed. If the test for one speed is passed, the next speed is tested. The parameter for a specific configuration (vehicle, mode) then equals the highest speed for which at least one of the three runs has been passed.

Figure 6 gives an impression of the test setup.

Figure 6: Impression of the double lane change test setup, here: Knocking over cones when attempting to drive through the lane of cones.

2.3 Tools

2.3.1 Test vehicles

In all cases, the towing vehicle was a Mercedes-Benz GLC 250D fitted with measurement and control technology (but otherwise in standard condition), with 2-3 occupants; total mass in test configuration 1845 kg + approx. 300 kg = approx. 2.2 t, see Figure 7.

The following vehicles, neither of which are production vehicles, were used as test trailers:

Manufacturer A vehicle, mass in test configuration approx. 2.1 t, single central axle, length including drawbar approx. 6 m, distance from coupling point to axle approx. 4 m, see also Figure 7.

Manufacturer B vehicle, mass in test configuration 1.8 t, tandem axle, length including drawbar 9 m, distance from front axle to coupling point 5 m, see Figure 8.

The vehicles were powered by electric motors. The motors were controlled evenly on both sides (no torque vectoring). The software that controls the motors in both vehicles was not a final, approved version and is therefore probably still subject to changes after the tests. Since BASt did not have access to the software, no details on the software version used in the respective road tests can be given.

The Manufacturer A vehicle gradually reduced the drive support at between 100 and 110 km/h; the Manufacturer B vehicle switched off the support system from 85 km/h onwards.

Figure 7: Mercedes-Benz GLC towing vehicle and Manufacturer A test vehicle

Figure 8: Manufacturer B test vehicle

2.3.2 Test sites

The sites where the tests were conducted are shown in Figure 9 and Figure 10.

Figure 9: Vehicle dynamics area at ATC Aldenhoven (Image: ATC Aldenhoven)

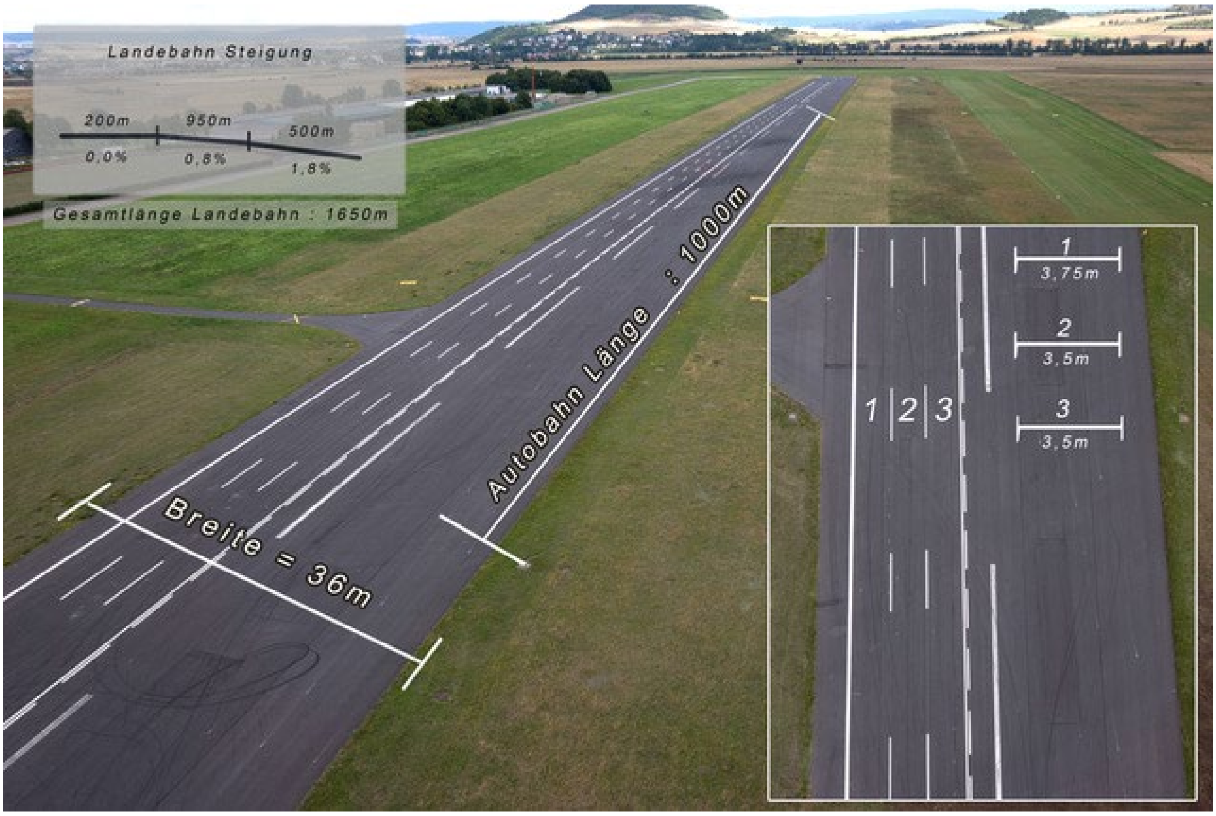

Figure 10: Runway at TEA Mendig (Image: TEA Mendig)

All driving tests were carried out at the Aldenhoven Testing Center[3] (all Manufacturer B tests; Manufacturer A: double lane change) and the Mendig Test Event Area[4] (Manufacturer A high-speed stability test).

In Aldenhoven, the circular vehicle dynamics platform with a diameter of 200 m and an acceleration lane of approx. 300 m were available (see Figure 9), while in Mendig a former runway with a length of approx. 1200 m was used (see Figure 10).

2.3.3 Measurement and control technology

All vehicles were equipped with ADMA-type DGPS inertial platforms (models G and Slim). The towing vehicle had a type-CS1200 Vehico steering robot, see Figure 11. This device consists of a gear wheel that is attached behind the steering wheel with cable ties and a motor with a pinion that is supported by suction cups on the windscreen. The steering movements are freely programmable.

The steering robot was operated in open loop mode. The parameters of the steering controller for the defined steering impulse were set so that the maximum lateral acceleration of the trailer was between 3 to 5 m/s², adjusted to the conditions of the first test trailer (Manufacturer A vehicle). For reasons of comparability, the programming was retained for the second vehicle (Manufacturer B vehicle), even though the lateral accelerations achieved were not exactly the same due to different length ratios.

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[3] www.atc-aldenhoven.de
[4] www.tea-mendig.de

An overview of the measured lateral acceleration for the first vehicle (Manufacturer A vehicle) for speeds of 70 to 100 km/h (from top left to bottom right) is shown in Figure 12.

Figure 11: Vehico CS1200 steering robot, fitted in another vehicle

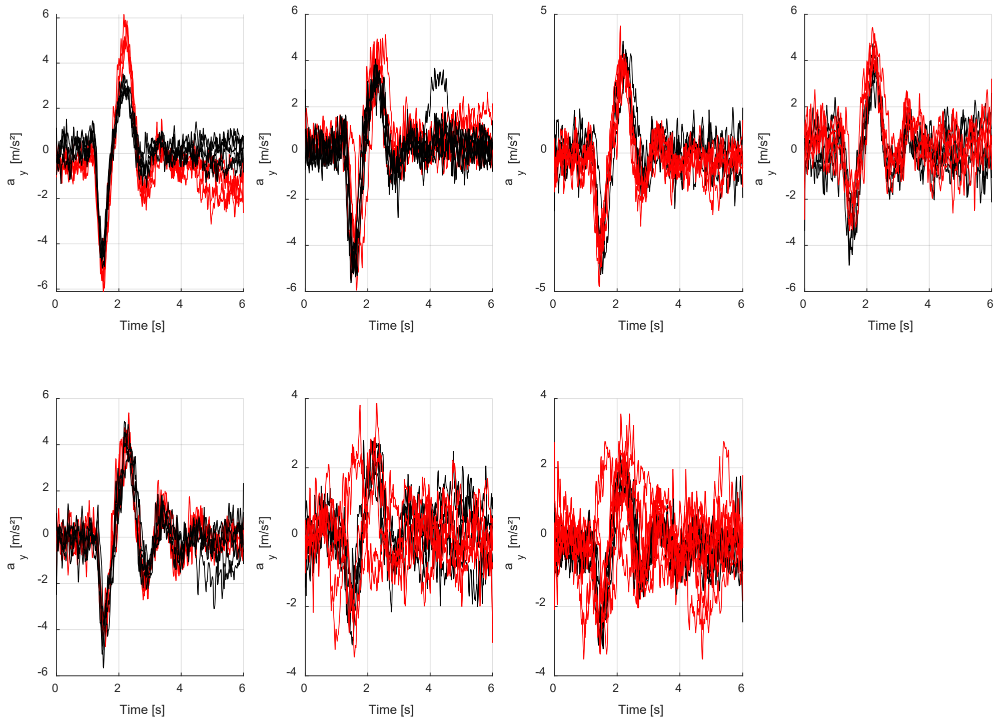

Figure 12: Lateral acceleration in the trailer as a result of the defined steering impulse, at all speeds driven, example: Manufacturer A vehicle

The main parameter for the test evaluation is the articulation angle, i.e. the angle between the vehicle and the trailer. It is calculated from the difference of the yaw angles of both vehicles (positive to the left).

Another important parameter is the speed of travel, which was ensured using a speed limiter. The speed of travel of the towing vehicle set by the limiter is very exactly 3 km/h above the real speed; this difference was taken into account when setting the speed limiter.

The shear force on the trailer coupling is also important, where it is measured.

There was an occasional partial failure of the GPS measurement technology. For these occasions, the articulation angle was determined by integrating the difference in yaw rate between the towing vehicle and trailer. As the rotation rate sensors in the measuring units have very low drift, this is no problem for short periods of a few seconds. The tests could not be repeated because the test vehicles were only available for a limited amount of time.

One of the vehicles (Manufacturer A) had a force measuring device to measure the three components of the force in the trailer coupling (longitudinal, lateral and vertical).

3 Results

3.1 High-speed stability

The test results are described below based on the evaluation of the measurements as defined in section 2.2.1.

3.1.1 Manufacturer A vehicle

For the Manufacturer A vehicle, the forces lateral to the drawbar (F_y) and longitudinal to the drawbar (F_x) as well as articulation angles were measured at excitations between 70 and 95 km/h, as shown in Figure 13 to Figure 18 .

It was found that at 70 km/h, the oscillation in the trailer with active drive abates more slowly after an initial, equal oscillation cycle than in a trailer without active drive, as can be seen in the shear force and articulation angle; this effect is not observed at all other speeds. There is no explanation for this, but the behaviour is not considered critical due to the low speed.

Furthermore, it is evident that, as expected, the longitudinal force is significantly lower when the vehicle is powered.

When evaluating the natural frequency and damping Figure 19, it is apparent that the characteristic speed v_{ch} when the trailer drive system is active slightly lower than the speed for a trailer without active drive support.

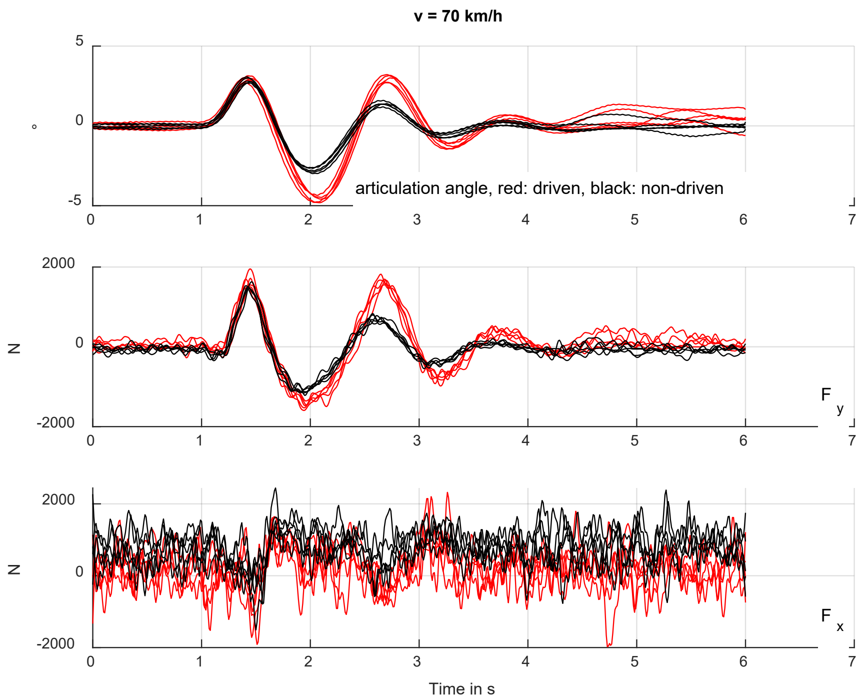

Figure 13: Manufacturer A, 70 km/h

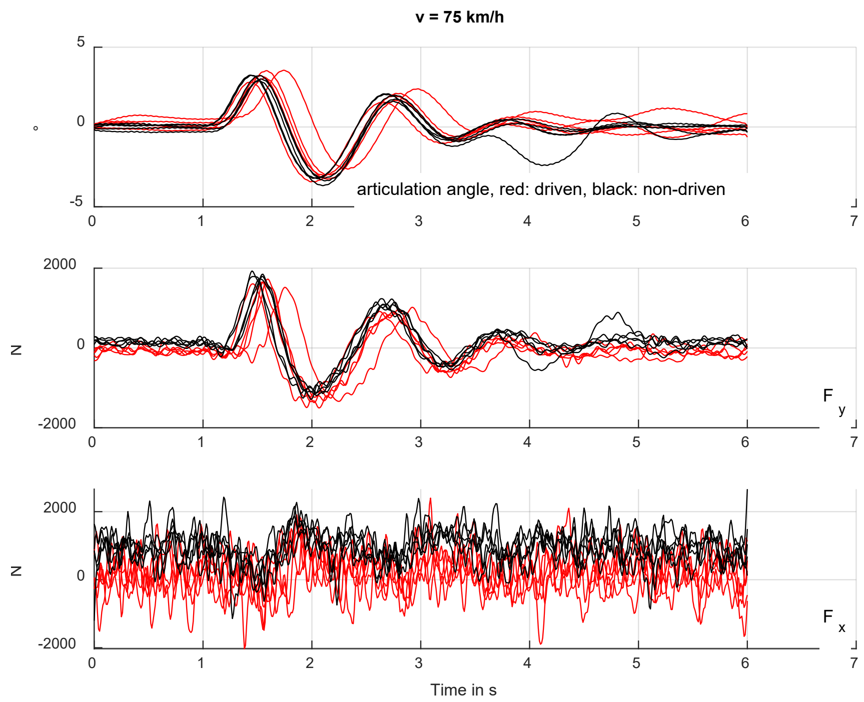

Figure 14: Manufacturer A, 75 km/h

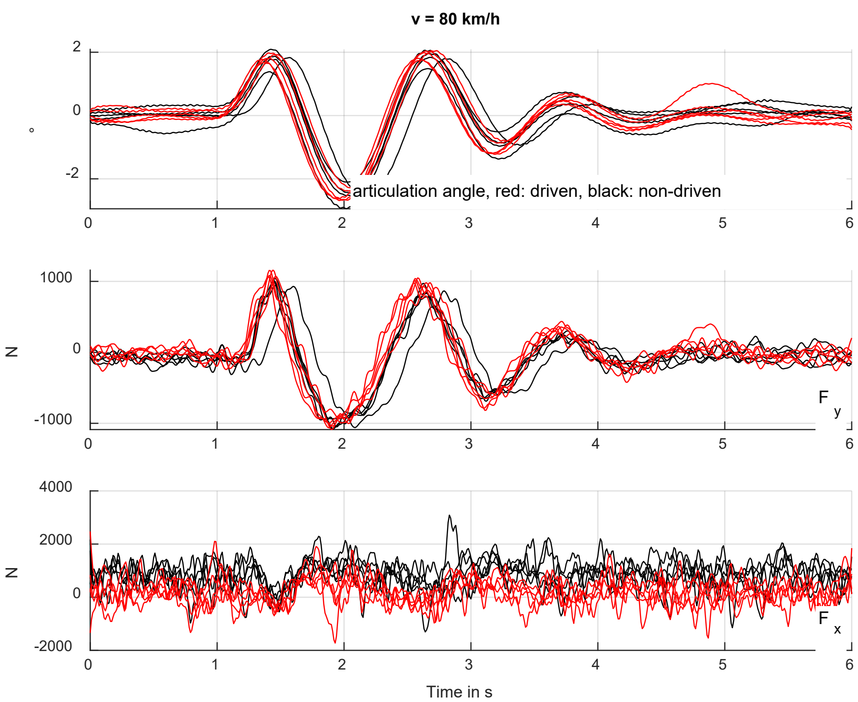

Figure 15: Manufacturer A, 80 km/h

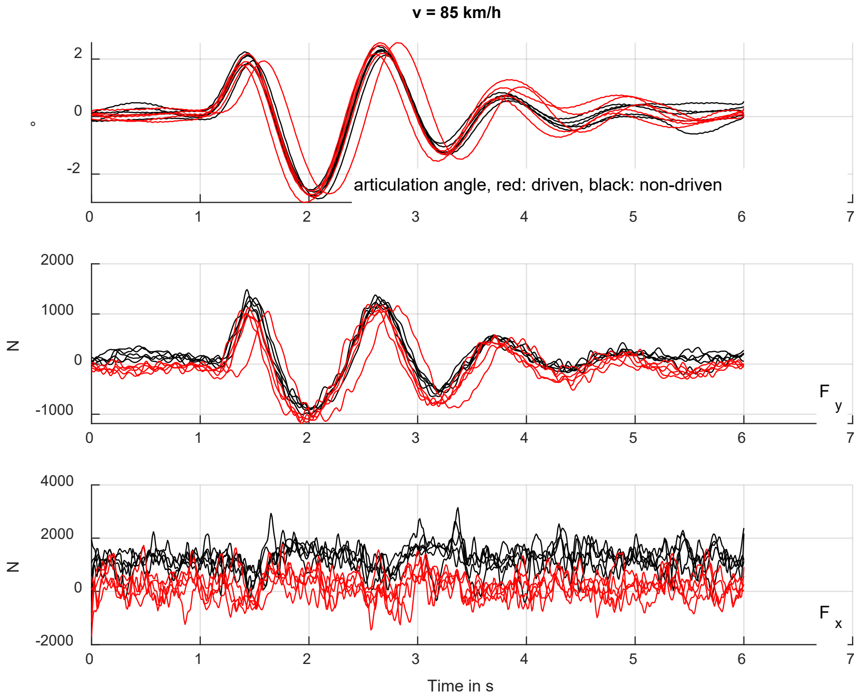

Figure 16: Manufacturer A, 85 km/h

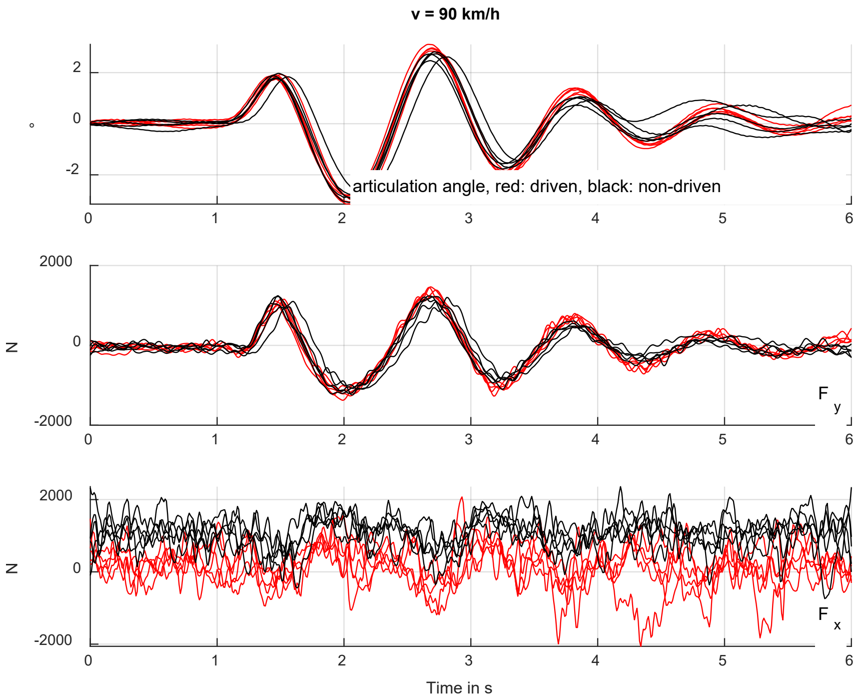

Figure 17: Manufacturer A, 90 km/h

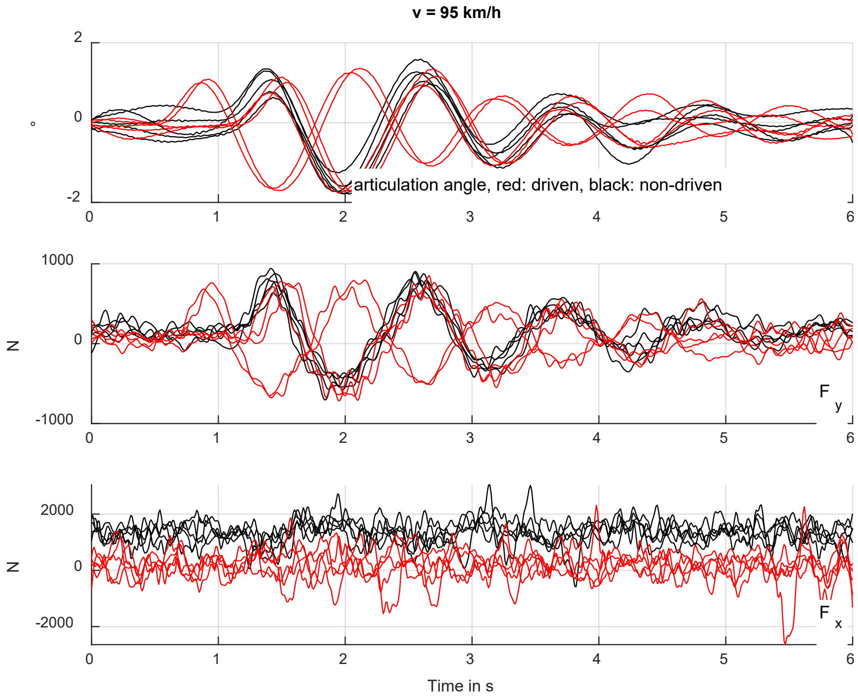

Figure 18: Manufacturer A, 95 km/h

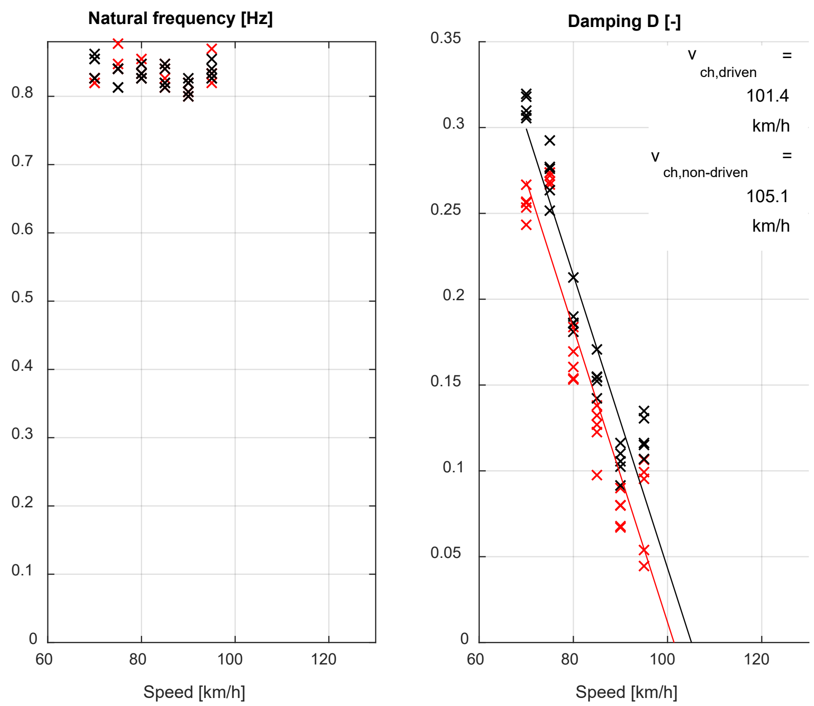

Figure 19: Manufacturer A vehicle, natural frequency and damping comparison of active (red) and passive operation, as well as the regression line for determining the characteristic speed v_{ch}.

3.1.2 Manufacturer B vehicle

For the Manufacturer B vehicle, the articulation angles shown in Figure 20 to Figure 24 were measured at excitations between 60 and 85 km/h. Measurement of the drawbar forces was not possible. The time curves of the articulation angle, natural frequency and damping, as well as the characteristic speed, shown in Figure 26, are almost identical for this vehicle when directly comparing operation with/without active drive support.

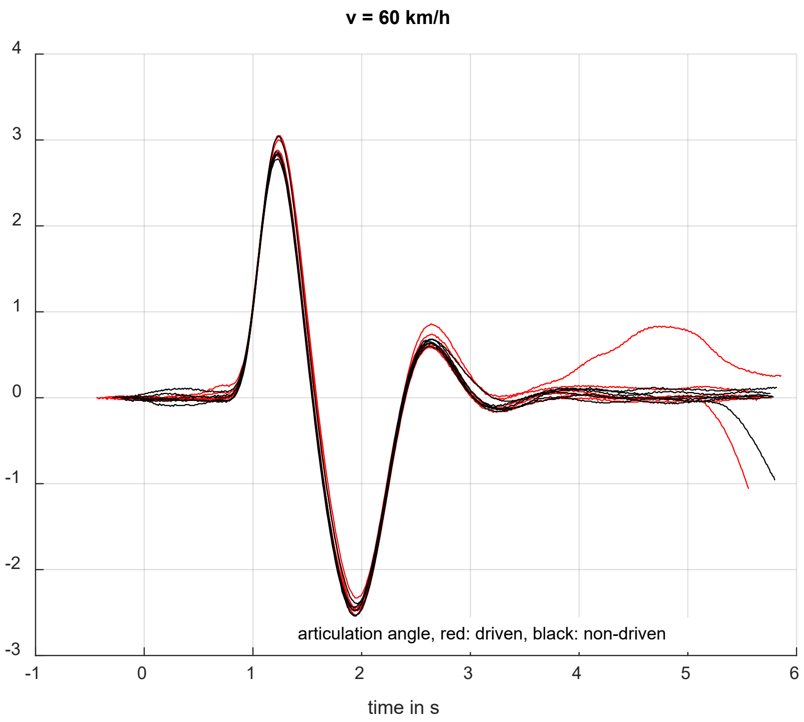

Figure 20: Manufacturer B, 60 km/h

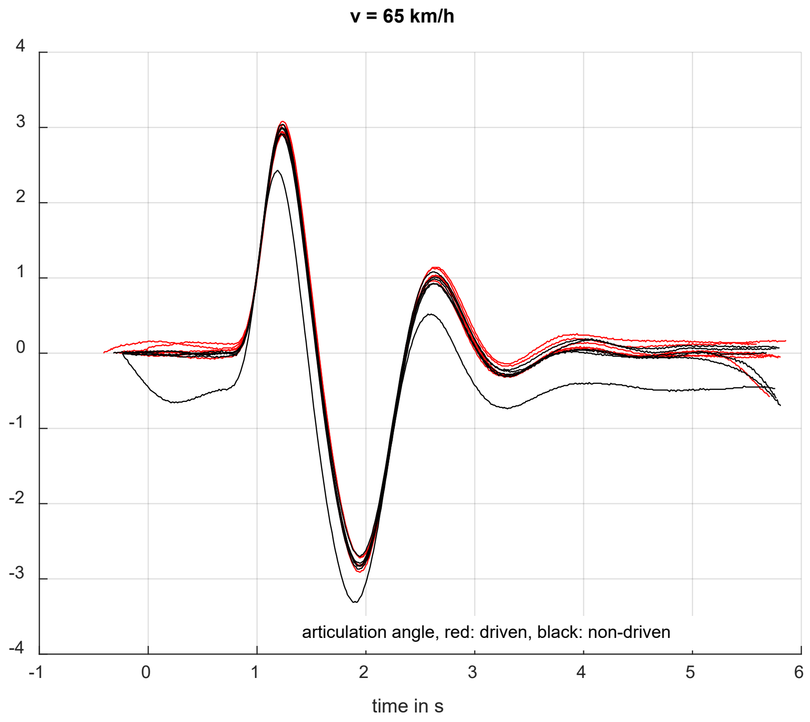

Figure 21: Manufacturer B, 65 km/h

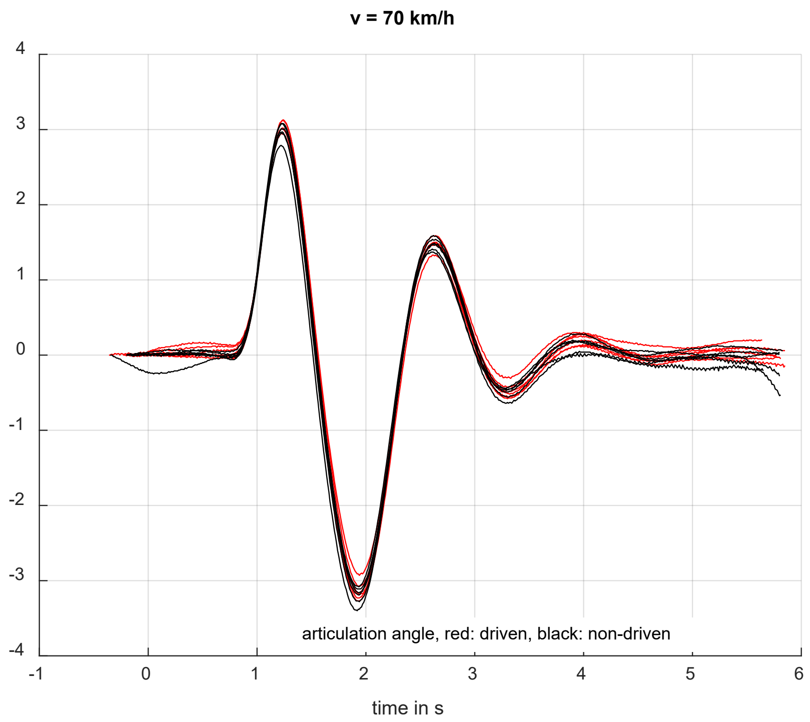

Figure 22: Manufacturer B, 70 km/h

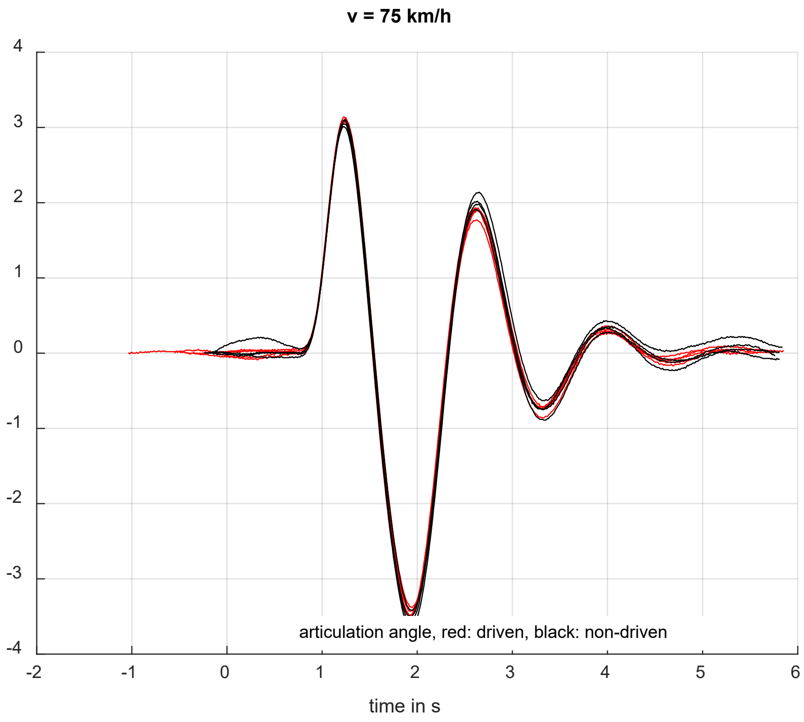

Figure 23: Manufacturer B, 75 km/h

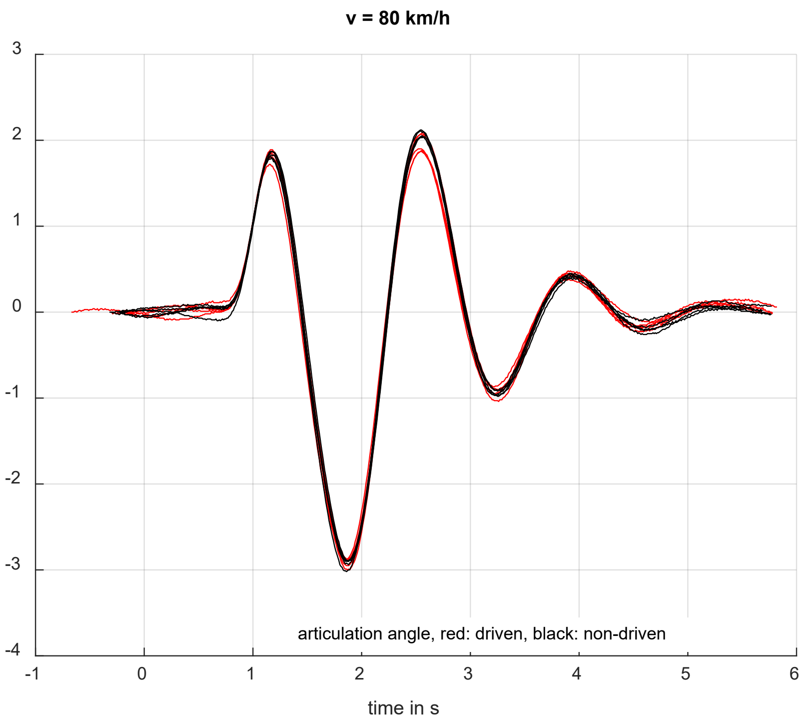

Figure 24: Manufacturer B, 80 km/h

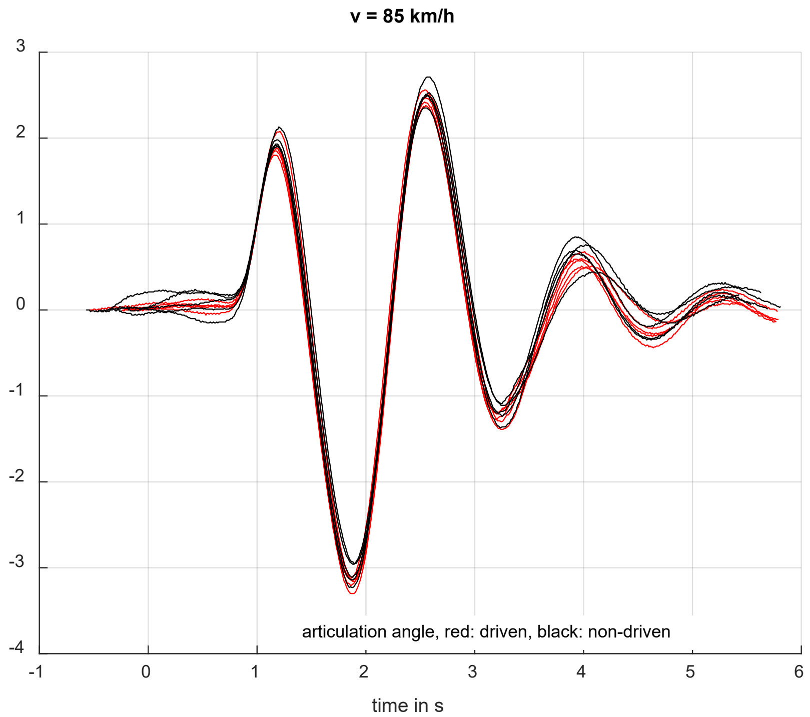

Figure 25: Manufacturer B, 85 km/h

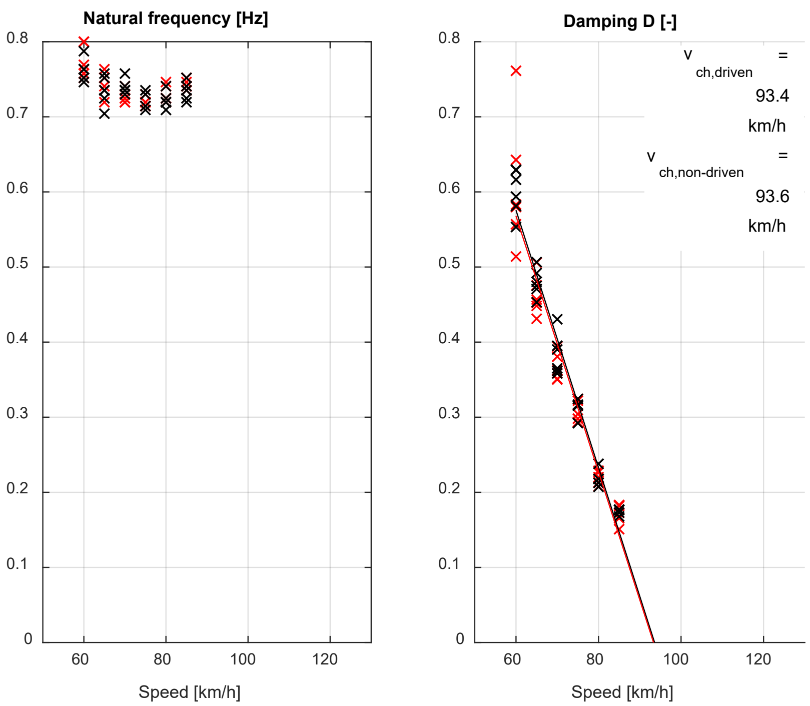

Figure 26: Manufacturer B vehicle, natural frequency and damping comparison of active (red) and passive operation, as well as the regression line for determining the characteristic speed v_{ch}.

3.1.3 Conclusions

The high-speed stability of two trailers was investigated in a direct comparison of active drive and passive drive (= drive support deactivated). Slightly reduced sway control and also a slightly lower characteristic speed for the case with active drive was identified for one of the vehicles (101.4 km/h to 105.1 km/h), while no differences were found for the second vehicle (93.4 km/h to 93.6 km/h). Even the difference found in the characteristic speed is less than 5% and is therefore considered negligible.

From the point of view of high-speed stability, there are therefore no misgivings about drives, at least for the vehicles investigated – section 3.3 discusses the transferability of the findings to other vehicles.

3.2 Double lane change

As shown in section 2.2.2, the double lane change is a test conducted in the closed loop driver-vehicle-environment. Each speed of travel was driven three times in order to largely rule out subjective influences due to individual driving ability. One pass was the requirement. The objective was to find the highest speed at which the vehicle completed at least one attempt without knocking over cones, i.e. passed. The higher the speed, the better the driveability of the combination. The results of the individual tests are shown in Table 1.

v [km/h]	**Manufacturer A, active**	**Manufacturer A, passive**	**Manufacturer B, active**	**Manufacturer B, passive**
50	1			
60	1			3
65	1			
70	2		1	**3**
75	1		1	0
80	1	1	**1**	
85	3	1	0	
90	2	1		
95	**3**	**3**		

Table 2: Test passed for each speed of travel, vehicle and mode (1: passed on the first attempt, 2: passed on the second attempt, 3: passed on the third attempt. 0: no pass in three attempts.)

The results were as follows:

- Manufacturer A 95 km/h active and passive,
- Manufacturer B 80 km/h active, 70 km/h passive.

The differences between the vehicles are due to the fact that the Manufacturer B trailer is significantly longer. It was found that active drives do not adversely affect the driveability (the fact that the Manufacturer A trailer tended to need more attempts to pass in the active case is the result of learning effects – it was driven actively first). The driveability of the very long Manufacturer B trailer actually even improved significantly.

A subjective statement by the test driver: When the trailer drive was active, the Manufacturer B vehicle could be driven more precisely through the cone lane. An explanation based on driving dynamics could be that the braking force resulting from the longitudinal force F_x of the trailer and the hindrance of the turning motion are then significantly lower. This effect is probably also present in the Manufacturer A vehicle, but is possibly less noticeable due to the lower moment of inertia around the yaw axis.

From the point of view of driveability, there are therefore no objections to drive systems, at least for the vehicles examined.

3.3 Verification of the equations and transferability

The measurements from the Manufacturer A vehicle can be used to verify the equations from section 2.1.1, in particular also to estimate the lateral force based on the articulation angle of the combination.

The following additional assumptions are made for this purpose:

- Tyre skew stiffness k_α = 0.75 1/rad,
- The time delay between angle measurement and force measurement was a constant 0.2 s (this would indicate that there is a 5 Hz low-pass filter in the inaccessible force measurement hardware and software).

With these assumptions, there seems to be a good correlation between measurement and calculation (Figure 27), though the calculation tends to overestimate the shear forces at higher speeds of travel (Figure 28).

Based on the fact that the vehicle dynamic calculations apparently show no fundamental differences to the measured lateral force, it is assumed that the findings are transferable at least to other trailer types that can be simplified as a unicycle model. This is not initially possible for trailer types with multiple degrees of articulation freedom, such as turntable trailers.

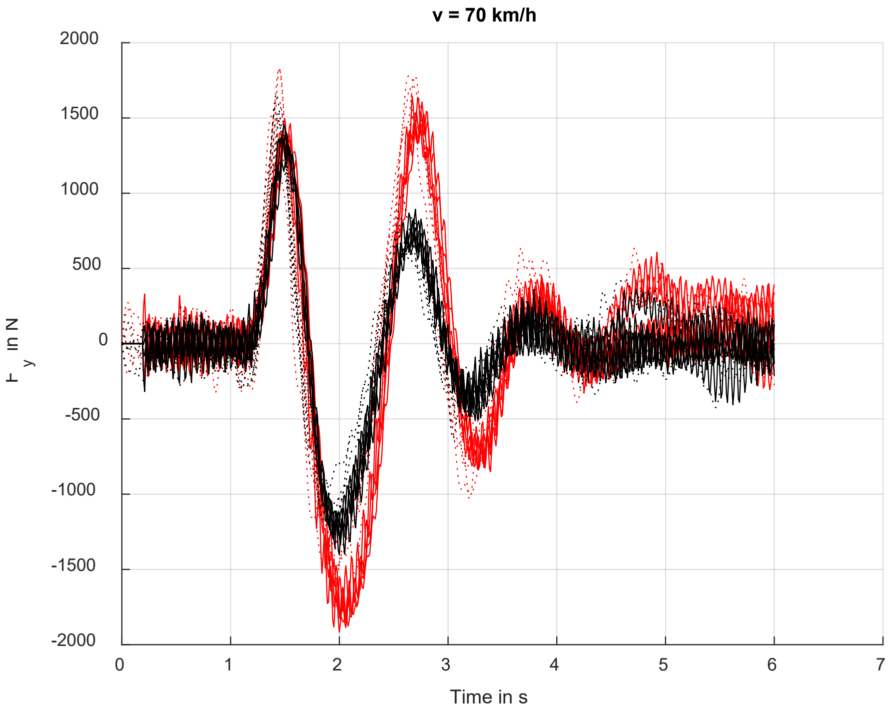

Figure 27: Comparison of calculations (dashed line) and measurements (solid line) for the Manufacturer A vehicle, 70 km/h (minimum speed of high-speed stability)

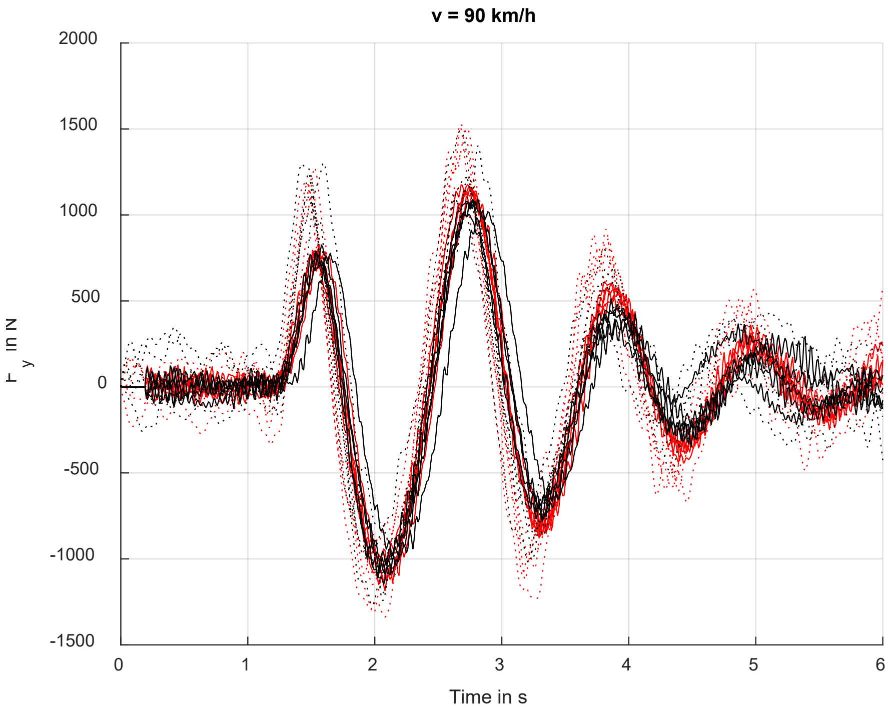

Figure 28: Comparison of calculations (dashed line) and measurements (solid line) for the Manufacturer A vehicle, 90 km/h

3.4 Other requirements derived from observations

The vehicles provided were prototypes that were not final production versions, especially their operator controls as well as their control strategies.

However, even with these vehicles, it is already evident that powered trailers significantly improve the handling of the combination and the subjective driving experience.

However, special attention must be paid to the fact that reversing must be possible at any time, and even when the system is not active, for example if the energy storage device is empty (this was not possible with both vehicles).

Long-term stability must also be guaranteed, especially if force measuring elements are part of the control loop in series operation (signal drift is possible). When using one vehicle, the drive system failed after one hour of driving during the transfer from the test site to the BASt. This could be due to signal drift.

In some cases, the vehicles had a hard cut-off of the drive support from a configured speed of travel. This can be perceptible – a gradual reduction or hysteresis would be better.

4 Conclusions

The project investigated the influence powered trailers may have on the vehicle dynamics of combinations compared with non-powered trailers.

Theory suggests that in particular shear forces and longitudinal forces with a pushing effect, which are transferred from the trailer to the coupling, may destabilize the towing vehicle around the yaw axis. Longitudinal forces with a pulling effect in contrast generally have a stabilizing effect. Due to the application of the same driving force to the wheels on both sides of the trailer as long as there is always a resulting tractive force, the drive system is not expected to influence the shear force at the trailer coupling.

Bearing this in mind, trailers with a symmetrical driving force that is lower than the driving resistance force of the trailer do not have a significant negative impact on the dynamics of the vehicle combination.

As the required tractive effort of the towing vehicle is lower when the trailer is powered, the driveability of the vehicle combination is expected to improve.

To verify these two assumptions, the stability at high speeds and the swerving behaviour were identified as decisive criteria and were assessed by conducting a standard defined steering impulse test and a standard double lane change manoeuvre respectively. In addition, the expected lateral forces were quantified by means of simple equations for trailers with and without active drives.

The theoretical assumptions established by means of calculations could be confirmed in driving tests. A slightly reduced sway control and characteristic speed were found in one of the two trailers when an axle was powered; no relevant differences were found in the other trailer. Driveability was not found to be worse for either of the trailers with active drive compared to the same trailers without active drive; in one trailer, driveability was found to have improved significantly when the drive was active. The simple equations to quantify the lateral forces were confirmed by measurements.

Based on the results, vehicle safety is not considered to be compromised if powered trailers are designed so that the driving forces are distributed evenly between both wheels and the trailers do not push the towing vehicle.

Uneven distribution of the driving forces could be used for active stabilization; however, as torque vectoring was not available in the vehicles, no such assumptions could be established and tested.

4.1 Derivation of recommendations

The following requirements for powered trailers can be derived from the equations:

- The driving force should not exceed the driving resistance so that the trailer is always pulled. When that is the case, the destabilizing shear forces on the trailer coupling are not expected to be greater than when the trailers drive system was not active.

- The driving force should be applied largely evenly to both wheels in terms of amount and phase. This should be demonstrable by considering the functional safety of the system.

The following steps, for example, would be necessary to enshrine such powered trailers in legislation (without having analysed all regulatory texts; in particular, an assessment of the further requirements for drives in general that would then also have to be transferred to drives installed in the trailer is required). In addition to the concept of putting symmetrically powered, non-pushing trailers on an equal footing with non-powered trailers, one could also create a category for 'self-propelled trailers' like that for 'towing trailers'.

- For example, R.E.3 (ECE/TRANS/WP.29/78/RE3) could be amended as follows: 'Trailer' means any non-self-propelled vehicle, which is designed and constructed to be towed by a power driven vehicle and includes semi-trailers. **Vehicles propelled with a force below their driving resistance at all times can be considered non-self-propelled if the force acts symmetrically.**
- The trailer definition in Regulation (EU) No 858/2018 could also be amended as follows: (17) 'trailer' means any non-self-propelled vehicle on wheels designed and constructed to be towed by a motor vehicle, that can articulate at least around a horizontal axis perpendicular to the longitudinal median plane and around a vertical axis parallel to the longitudinal median plane of the towing motor vehicle; **Vehicles propelled with a force below their driving resistance at all times can be considered non-self-propelled if the force acts symmetrically.**
- Requirements for functional safety, control quality and similar properties would have to be established in an additional document at UN or EU level, if necessary.
- In a second step, provisions to govern uneven distribution of forces to the wheels could be made. It has the potential to improve driving stability even further; in this case, however, special attention should be paid to ensure that the drive systems are not controlled incorrectly. For example, in the context of state-of-the-art functional safety systems (e.g. ISO 26262), the avoidance of destabilizing the towing vehicle could be explicitly defined as a safety objective. Besides hardware problems, this must explicitly take software problems into account.

Figures

Tables