

THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING

in

Machine and Vehicle Systems

Truck Steering System and Driver Interaction

by

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Göteborg, Sweden, 2014

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THESIS FOR LICENTIATE OF ENGINEERING no 2014:16
ISSN 1652-8565

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Chalmers Reproservice
Göteborg, Sweden 2014

*Tilägned Löfåhsen i Sunds Sⁿ
och Inhwånarna afh denna Bygd*

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Abstract

This thesis presents a compilation of methods to consider when mapping steering functions and results as vehicle dimensions change. It is concluded that some final tuning inevitably will be required as major changes are performed. It is however possible to create qualified starting points using a set of simple rules as presented. Considered properties are steering wheel size, wheelbase, steering ratio, and understeer gradient.

An investigation of driver behaviour when a sudden yawing disturbance is acting on the vehicle is also presented. Two examples, automatic braking on split friction and front tyre blow-out, are studied in detail. For automatic braking of a heavy truck it is concluded that current legal requirements and technology for split friction conditions are sufficient for an alert driver, but may create some problems for a driver being distracted. Most heavy trucks have positive steering-axis offset at ground, also known as kingpin offset at ground. This can induce a destabilising steering wheel torque when a front tyre is damaged. The effect from this is investigated using a qualitative approach. For an active driver it is found that elimination of the destabilising steering wheel torque has a small, yet statistically significant, effect on lateral deviation. And furthermore that the lateral deviation increases as the driver exhibits higher admittance.

A general conclusion from the analysis, on driver behaviour at yawing disturbances, is that lateral deviation will reduce substantially when driver reaction time is reduced. This can be achieved by warning the driver prior to the incident. Hence, the warning phase, that commonly precedes automatic brake activation, is of high importance. Another method is to use steering support in the initial phase of the incident.

Keywords: active safety, active steering, driver behaviour, heavy trucks, steering system, torque feedback

List of Included Papers

Paper 1: K. Tagesson, B. Jacobson, and L. Laine, “The influence of steering wheel size when tuning power assistance,” *International Journal of Heavy Vehicle Systems*, vol. 21, no. 4, 2014.

Contribution: The study was designed, run, analysed and authored by Tagesson. Laine and Jacobson contributed with good ideas and in reviewing.

Paper 2: K. Tagesson, B. Jacobson, and L. Laine, “Driver response to automatic braking under split friction conditions,” in *The 12th International Symposium on Advanced Vehicle Control*, 2014, pp. 666-671.

Contribution: The study was designed, run, analysed and authored by Tagesson. Laine and Jacobson contributed with good ideas and in reviewing.

Paper 3: K. Tagesson, B. Jacobson, and L. Laine, “Driver response at tyre blow-out in heavy vehicles & the importance of scrub radius,” in *The 2014 IEEE Intelligent Vehicle Symposium*, 2014, pp. 1157-1162.

Contribution: The study was designed, run, analysed and authored by Tagesson. Laine and Jacobson contributed with good ideas and in reviewing.

Acknowledgements

The research presented in this thesis has been financially supported by Volvo Group Trucks Technology and VINNOVA, Sweden's innovation agency. This support is gratefully acknowledged.

I would like to direct my first personal sincere gratitude to my supervisors Adjunct Prof. Leo Laine and Prof. Bengt Jacobson; always providing motivation, enthusiasm and insightful comments. Leo's relentless pursuit towards future visions is unparalleled. Bengt's complete eagerness to understand and question everything often triggers the best ideas. To my managers at Volvo GTT, Stefan Edlund and Inge Johansson. Your incredible belief in individual creativity is truly honourable. That trust and encouragement make me grow as a person and in research. Dr. David Cole, thank you for all the shared ideas and valuable advise you gave during your time as a guest researcher. David, we have some stuff left; I'm looking forward to it. Thank you Dr. Jochen Pohl for all the time you have spent to get this project started. My roommate, Peter Nilsson, you gave me inspiration to actually start doing a Phd. Today I am most grateful for that. And thank you for all the pleasant chats and cooperation we have had. Thank you Carl-Johan for helping me out with the trucks. Thank you Gustav and Jan-Inge for linking me into the real world of steering every now and then. The Vehicle Analysis group at Volvo, big thank you have all contributed. Adi, Anton, Fredrik, Derong, Gunnar, Artem, Manjurul and Mattias—surrounded by like-minded gurus what more can I say. The ladies who have helped me keep track of the oh-so-important administration, Sonja, Marianne and Britta big thanks. And thank you all others not mentioned for making my days at work enjoyable.

Last but not least, I would like to thank family and friends. You have all contributed with support, inspiration and joyful moments. Josefina my constant shining light—you make every day special. Our endless discussions help me navigate through life.

Kristoffer Tagesson
Göteborg, December 2014

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Chapter 1

Introduction

This chapter presents the background, motivation and objectives of the thesis. It also gives the limitations of scope and outline of the thesis.

1.1 Background

During an ordinary walk on a street our brains are able to perceive huge amount of information every second. Like the feel of a cool breeze against the skin, see a bird fly by or how you slide on a patch of ice. Each and every impression is composed from endless details. All impressions, coming from our senses, are put together and used as we take continuous decisions. Like lifting a foot, blinking with an eye or stretching out an arm. All of this is to us evident. It is only by comparison to other systems that our great capacity becomes apparent.

When seated in a vehicle as a driver, on the same street, continuous decisions will still have to be taken. Like adding more force on the steering wheel, braking or accelerating. Here, however, our connection to the surroundings has drastically been changed. The eyesight still works fairly in the same way. Same goes with the sense of balance. But everything that was perceived using the sense of touch when walking—is now shielded from the surroundings. Furthermore, the outcome of actions has now another meaning; turn the steering wheel to change the direction. For us to be able to choose as wise decisions when driving as when walking all of our senses need sufficient information and the outcome from action needs to be predictable.

The steering wheel is the instrument being used the most by a driver. It is used to decide on the direction of the vehicle, but also as an extension to the sense of touch. By holding the steering wheel the driver can feel the motion of the vehicle, the road surface and forces from the surroundings. We will never be able to sense as much holding a steering wheel as what we do with our tactile organs on a walk. However steering system technologies, developed in recent years, at least show that it is possible to add on to what can be felt in the steering wheel; thereby hopefully supporting the driver. In this the effectiveness of the design, which lies in being compliant with the human driver, is the real challenge. In my career I have spent several years in trying to understand what this means and come to a conclusion; it is only by hard work in trying to understand the behaviour of a human driver that we can design effective steering. An arrangement where

all senses are in place for taking important decisions. Continuing on the walk analogy it is also obvious that the consequences of our actions when driving are often far greater than when walking. Often we don't only carry the responsibility of our own lives, but also others: passengers, pedestrians, cyclists, and other vehicles. Consider driving a 60 ton truck combination. Then obviously all of our senses need sufficient information and the outcome from action needs to be predictable. Fig. 1.1 shows the enormous forces that are released in an accident involving a heavy truck.

For sure, road vehicles are and will remain a very important reason for global prosperity. Goods and people are transported nationally and internationally for many reasons. At some locations it is easier and cheaper to grow food. Others have natural resources available. The sum of only national goods transported on roads, counted in tonnes, within OECD in 2013 equals moving half of Sweden's population to the moon¹ [1]. As put in [2], "Transport is important for poverty eradication because it provides access to markets and supply chains of intermediate outputs." It was even committed by UN General Assembly in the Rio de Janeiro 2012 meeting [3] that "We recognize the importance of the efficient movement of people and goods and access to environmentally sound, safe and affordable transportation as a means to improve social equity, health, resilience of cities, urban-rural linkages and productivity of rural areas. In this regard, we take into account road safety as part of our efforts to achieve sustainable development." Transportation is evidently an important aspect of the global future development, but equally important is safety. The consequences of only one fatal accident are huge emotional and economical losses. According to [4] 1.24 million people were killed and another 20 to 50 million people got non-fatal injuries on the world's roads in 2010. Obviously there is a long way to go.



Figure 1.1: A snapshot from a roll-over test of a heavy truck.

¹Here considering equal payload distance (kilogram-kilometre).

1.2 Motivation

Global key risk factors, identified by WHO in [4], for traffic accidents are speed, drink-driving, motorcycle helmets, seat-belts, and child restraints. WHO recommends an introduction and enforcement of proper legislation as the primal countermeasure for these factors. Looking at EU only the situation is somewhat different. Even though the number of vehicles per capita is high the number of fatal accidents per capita is less than half that of the global level [4, 5]. The simple explanation to this is developed legislation, safer roads and safer vehicles [4]. Yet, 28,126 fatalities and 1.4 million people injured were reported in 2012 [6]. In this heavy trucks are involved in about 17% of the fatalities and 7% of all casualties [5]. A general verdict, about these accidents, is given in [5] "Human error is involved in as many as 90% of all accidents". Looking at heavy trucks only [5] continues "The two most common human factor related factors that contribute to heavy truck accidents are failure to look properly and failure to judge another person's path or speed. When the vehicle contributes to the accident, the most common cause is limited visibility due to blind spots."

Based on the details of the findings on accident cause and technology in reach [5] suggests heavy trucks prioritised areas, amongst active safety systems, as: headway support, lane keeping support, driver awareness support, vehicle stability, vehicle communication, and visibility support. In common for the first four suggestions is a need of understanding the interaction between the driver and the steering system. This is the focus of this thesis. At first headway support might look disconnected from driver steering interaction, but as will be shown with a recurrent use case it is not.

Two areas, within driver steering interaction, were identified as most valuable to be able to design effective active safety systems. The first was to find a map between results received in research on cars and research on heavy truck. The other was to better understand driver behaviour at a sudden lateral disturbance. These two important elements will be explained more in detail in the following subsections.

1.2.1 Steering Properties Independent on Vehicle Type

There are lane keeping support systems suggested, that alter the steering wheel torque, to give an indication of where the lane ends, e.g. [7]. There are systems suggested that give steering guidance to avoid a crash with other vehicles, e.g. [8]. There are systems suggested that provide steering guidance to avoid rolling over the vehicle, e.g. [9]. There are systems suggested that exaggerate the feel of a slippery road, e.g. [8]. All these systems are examples of active safety systems where driver steering interaction is on its limits, i.e. refining the information presented to the driver via the steering wheel to the sense of touch. Some developed for cars and some developed for heavy trucks. In order to draw full benefit of the research commissioned on cars when working with heavy trucks, as said, being able to translate results would be required. For the driver steering task both vehicle steering response and steering wheel feedback is of value. The obvious differences in this between a heavy truck and a car are steering wheel size, steering gear ratio and wheelbase. When searching for previous research in this field very little material is found. Only a few that can give parts of the answer.

In [10] a fixed base simulator was used to conclude that a driver perceives force rather

than torque, here the angular degree of freedom was held fix, known as isometric motion. When the angular degree of freedom was unlocked and torque set to zero it was further found that a driver perceives steering wheel, StW, angle rather than hand translation. It has not been shown how force feedback should scale with StW size in a real vehicle where isometric motion no longer holds and where the driver subjectively decides on optimal balance between handling and comfort. Also no such test has been performed where the subjects are free to choose seating and StW position which could be different when changing StW. *This is the first gap identified.*

When working with a more urgent situation a sudden StW torque input would be tempting to apply, i.e. discontinuous StW torque. For this no in vehicle measurements have been performed to show differences of driver response when StW size is changed and where the tension level of the driver is a consequence of the main mission of following a real road, or similar. *This is the second gap identified.*

Regarding driver behaviour when changing steering gear ratio or wheelbase there is already relevant work carried out. For instance Neukum and Ukem [11] tested steering angle failures on four different cars and gathered subjective ratings on the severity experienced. They concluded that the subjective rating was independent of vehicle type if the magnitude of the failure was measured in terms of lateral accelerations or yaw rate. This is strongly supported in [12] where steering gear ration was altered. It was found that "yaw rate sensitivity is the actual vehicle steady-state gain characteristic which is noticed by drivers". By taking account for these and other studies it is possible to suggest how steering functions should be scaled as steering gear ratio and wheelbase changes.

1.2.2 Driver Behaviour at Sudden Vehicle Disturbance

Driver reaction to sudden lateral disturbance has been studied in e.g. [13]. It is concluded that the state of mind and body of the driver has large impact on the outcome. Sudden lateral disturbance in combination with an inattentive driver can be assumed in real traffic at shifting side wind, side collision, tyre blow-out, or heavy braking on a road section with significantly different level of friction between left and right vehicle sides. The later example is commonly known as split friction braking or split- μ braking. Sudden lateral disturbance in combination with an inattentive driver has been seen to cause severe accidents in history. For instance tyre failures are involved in many fatal accidents every year [14, 15]. Again to be able to create effective active safety systems preventing these accidents detailed understanding of driver reaction is crucial.

For split friction braking a hypothesis is that when the brake action is activated by an advanced emergency braking system, AEBS, automatically the driver would be less attentive compared to when the driver activates the brakes. An illustration of the use case is shown in Fig. 1.2, where an oncoming accident is created after lateral deviation of a truck combination. The danger experienced and created could therefore increase. As AEBS, is soon mandatory on heavy trucks in Europe [16] there is a strong need to gain more knowledge about driver reaction in this particular case. *This is the third gap identified.*

The axle installation arrangement used on heavy trucks in general creates a destabilising steering wheel torque upon front tyre blow-out. This effect is not present on modern passenger cars. In [17] Pettersson et.al. present the only study found on behaviour among

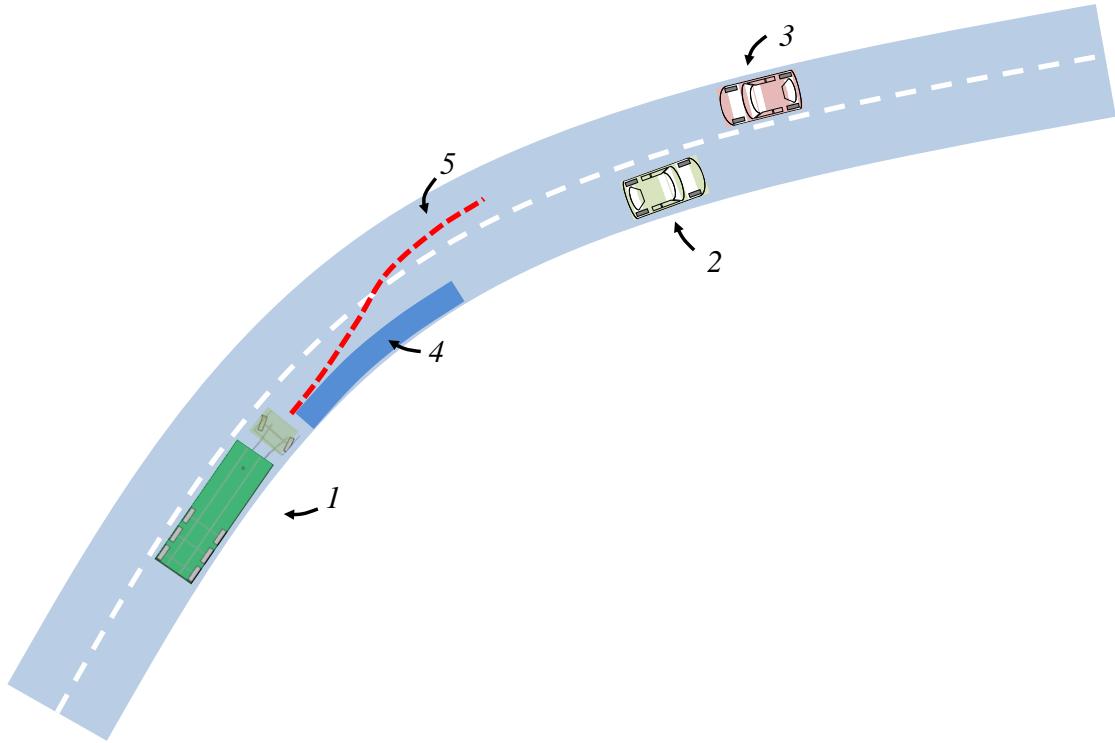


Figure 1.2: A truck (1) that brakes because of a stationary car (2). A patch of one-sided low friction (4) causes the truck to yaw and move sideways (5). When adding an oncoming car (3) into the scene an accident is imminent. The picture is taken from Paper 2.

different drivers at tyre blow-out. The level of StW torque induced was however not varied. *This is the fourth gap identified.*

1.3 Objectives

The overall objective of this work is to strengthen the knowledge about driver and steering interaction that can be used in the design of effective active safety functions, developed to reduce number of traffic accidents. First of all being able to map results and steering functions from cars to heavy trucks, in this field, constitutes the first main objective of this thesis. This involves taking account for previous work and carrying out additional studies to get comprehensive understanding.

The second objective of this thesis is to better understand how drivers react when brakes are automatically triggered and a yawing disturbance is present, due to e.g. split friction conditions. This understanding is crucial in order to design robust future headway support systems.

The third and final objective is to gain knowledge on what consequences a destabilising StW torque has when a front tyre suddenly explodes.

1.4 Limitations

Material, methods and applications that is not handled in this thesis are here listed.

- A destabilising StW torque is also present in a heavy truck during automatic braking on split friction. This effect is not analysed in here, but might be the subject of future studies.
- The primal applications of the research presented in this thesis are heavy trucks and heavy truck combinations. Results will therefore not be discussed for other vehicle types.
- A one degree of freedom mechanical link in the steering system is assumed if not otherwise stated. A steer-by-wire installation which is missing this link often provides more freedom to the design of steering functionality. Today it is however costly and therefore not treated in here.
- No trailer is included in the experiments performed. Some comments are however made regarding the inclusion of a trailer, but only from a theoretical point of view.

1.5 Thesis Outline

The thesis is structured as follows. Chapter 2 gathers basic theory about heavy truck steering systems. Then in Chapter 3 methods on how to translate findings on steering functions between different vehicle types are described. In chapter 4 driver reaction to lateral disturbance is treated. Finally, chapter 5 concludes the work and suggest future priorities. Notations used follow ISO 8855, [18], and units are SI unless otherwise stated.

Chapter 2

Truck Steering System

A steering system provides directional control of the vehicle. This chapter gives an overview on how the steering system works on a heavy truck and what consequences this has.

2.1 Conventional Steering System

The power required when steering a truck is very high compared to a car. This becomes obvious when considering the relative difference in front axle load, where a standard car carry 750 kg and most heavy trucks up to 7,500 kg. The most common front axle steering arrangement for a heavy truck includes a hydraulic steering gear. A hydraulic steering gear provides high power in comparison to its volume. Rear drive axles are in general not steered. However on other rear axles steering is often seen. The steering principles of these axles are often of simple nature, having the purpose to avoid tyre wear or shorten the effective wheelbase. Rear axle steering has an effect on vehicle response and manoeuvrability but not directly on steering wheel forces and will therefore not be described further.

The different parts of a conventionally steered front axle on a truck are shown and explained in Fig. 2.1. A steering wheel angle movement essentially results in a movement of the steer angle down at the wheels. The left and the right wheel steer angles are tied together with the track rod. The geometry of how the track rod connects to the steering knuckles should be chosen to produce a proper level of Ackermann, i.e. more steering on the inner wheel in corners.

The geometry of the steering knuckle, or more specifically how the kingpin bolt is oriented, creates the basis for how forces acting on the wheels propagate into the steering system. In Fig. 2.2 a real installation taken from a Volvo FMX is shown. Of particular importance are two angles, kingpin inclination and caster. These will be defined in the following sections. In accordance with [18], X , Y , Z is used to denote the intermediate axis system, where X is directed horizontally forward on the vehicle, Y pointing horizontally left, and Z pointing upwards. Furthermore the vehicle axis system, X_V , Y_V , Z_V , is introduced. It is fixed on the vehicle sprung mass so that X_V is directed forward on the vehicle, Y_V pointing left, and Z_V pointing upwards. Note that the vehicle axis system follows e.g. roll and pitch motion of the sprung mass, whereas the intermediate axis system does not.

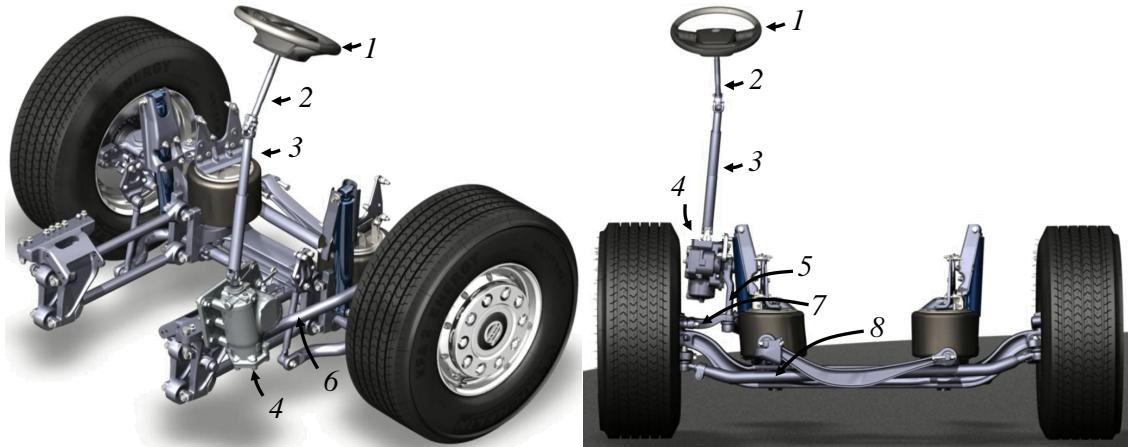


Figure 2.1: A conventional steering system from a left hand drive Volvo is shown from front left hand side, in left subfigure, and from rear, in right subfigure. The steering wheel (1) is connected via the steering column (2) and the steering shaft (3) to the hydraulic steering gear (4). The steering gear amplifies the steering wheel torque and produces a downshift from the incoming shaft angle to the angle of the Pitman arm (5), also known as drop arm. The Pitman arm is connected via the drag link (6) to the upper steering arm (7) which is controlling the angle of the steering knuckle around the kingpin bolt. The left and the right wheel steer angles are made dependent via the track rod (8).



Figure 2.2: The kingpin bolt, also known as the spindle bolt, is angled to produce proper steering characteristics. A red line is included to visualise the slant.

2.1.1 Kingpin Geometry

The steering axis, also known as kingpin axis, is the axis about which the wheel rotates relative to the vehicle structure when steered. For a truck with conventional steering this axis runs through the kingpin bolt. The kingpin inclination angle, σ , is the angle between the Z_V -axis and the steering axis, projected onto the Y_VZ_V -plane, see left part of Fig. 2.3. The kingpin inclination angle on trucks is normally around 5 degrees, and normally higher on cars [19]. The lateral component of the distance between road wheel contact centre and the steering axis, see left part of Fig. 2.3, is known as the steering-axis offset at ground¹, r_k . On heavy trucks the steering-axis offset at ground ranges from 5 cm to 15 cm, depending on the exact tyre and rim being used. On cars this value is closer to zero or slightly negative. But again, depending on the exact tyre and rim being used.

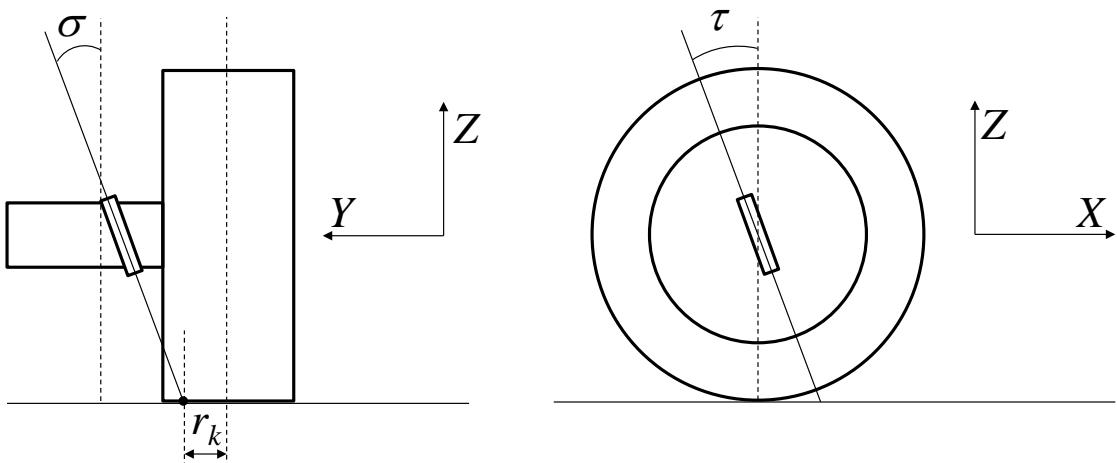


Figure 2.3: The tilt of the kingpin bolt can be decomposed to kingpin inclination and caster.

2.1.2 Caster Geometry

The caster angle, τ , also known as castor angle is the angle between the Z_V -axis and the steering axis, being projected onto the X_VZ_V -plane, see right part of Fig. 2.3. For heavy trucks a typical caster angle is 5 degrees at standstill. Note that during e.g. heavy braking when the vehicle pitches forward the caster angle will reduce and can even become negative.

2.1.3 Steering Wheel

Steering wheels can be seen on old cars dating back to around 1900. Before that tiller steering was the state of art². The steering wheel provides two dimensions, steering wheel

¹In ISO 8855 [18] r_k is referred to as steering-axis offset at ground or kingpin offset at ground. In [19] it is referred to as kingpin offset at ground or scrub and in [20] as scrub radius. The term scrub radius is differently defined in ISO 8855 [18] as the distance from wheel contact centre to the point where the steering axis intersect ground, i.e. also affected by caster. In this thesis steering-axis offset at ground will be used to denote r_k . However in Paper 2 and Paper 3 r_k is referred to as scrub radius.

²A tiller is a lever which on cars was attached to the steering mechanism. It was directed backwards, as opposed to what is often seen on boats.

angle, δ_h , and steering wheel torque, M_h . The relation between these two is here referred to as steering characteristics. The diameter of the steering wheel is important from two perspectives. It acts as a lever arm for the driver and it also strongly influences the total inertia of the steering system. In heavy trucks it is larger as a consequence of legislation. As stated by [21] the driver should be able to manoeuvre with limited steering forces also in the case of an assistance failure. This is achieved by designing the steering system so that the required force to steer the vehicle is limited even without assistance. With common wheelbase and steering ratio this typically means a steering wheel diameter of 45-50 cm on modern heavy trucks. Moving away from conventional steering system arrangements may well change these constraints.

2.1.4 Steering Gear

A hydraulic steering gear is shown in Fig. 2.4. On the incoming axle, from the steering shaft, a torsion bar is located. This torsion bar causes the opening and closing of valves for hydraulic high pressure fluid. The steering shaft also turns a ballscrew, known as the worm. The high pressure fluid is also acting on the worm to amplify the torque applied by the driver. The other member of the ballscrew causes the outgoing axle to turn the Pitman arm. The principle is used on most heavy trucks [9]. The design of the valves within the hydraulics has large influence on the amplification characteristics and therefore also on the steering characteristics. The amplification characteristics is often visualised with hydraulic assistance pressure as a function of torsion bar torque, e.g. see [9]. This curve is known as boost curve. In a heavy truck a common ratio between incoming and outgoing shaft angle is 16:1 to 27:1. The ratio is often nonlinear with higher ratio closer to end stops, with the purpose to make the truck manoeuvrable also at loss of hydraulic assistance [21].



Figure 2.4: A steering gear provides high power in relation to its volume.

The steering gear together with the linkage geometry produce the overall steering ratio, i_s , between the steering wheel angle and the average of the two wheel steer angles, δ . Where δ is formed by the X direction of the vehicle and the horizontal direction of the respective wheel. The steering ratio i_s is defined when no load is applied to the steering

system. The ratio is as mentioned before depending on the absolute angle. In a heavy truck i_s is in general close to the ratio provided by the steering gear for small steer angles, but will show a deviation for large steer angles when relay linkages induce nonlinearities.

When loading is added to the steering system, e.g. forces from wheel road interaction, the actual ratio can deviate substantially from i_s . This is due to compliance in the steering system. Here the torsion bar, within the steering gear, dominates [9]. Some trucks might even produce a steering ratio of double that defined as i_s when normal load is added [19]. This phenomenon adds understeer as experienced by the driver, since more steering is required when negotiating a curve at increased speed. Just like the steering ratio down shifts the steering wheel angle; it up shifts the torque applied on the steering wheel.

2.1.5 Equivalent Wheelbase

Wheelbase, l , is defined for a conventional two-axle vehicle, with a steered front axle and an unsteered rear axle, as the longitudinal distance between the front and rear axle wheel contact centre. For vehicles having more than one rear axle the equivalent wheelbase, l_{eq} , is instead introduced. It describes the wheelbase of a two axle vehicle with similar steady state turning behaviour as the multi-axle vehicle [22, 18, 23]. When assuming linear tyre forces the equivalent wheelbase can be calculated as

$$l_{eq} = L \left(1 + \frac{T}{L^2} \left(1 + \frac{C_{\alpha R}}{C_{\alpha F}} \right) \right) \quad (2.1)$$

where L is the wheelbase of the real vehicle calculated as the distance from the front axle to the point where the moments generated by vertical loads of the rear axles add up to zero. $C_{\alpha F}$ and $C_{\alpha R}$ are front cornering stiffness and sum of rear cornering stiffnesses respectively. T is the tandem factor which is calculated as

$$T = \frac{\sum_{i=1}^N \Delta_i^2}{N} \quad (2.2)$$

where N is the number of rear axles and Δ_i is the longitudinal distance from axle i to the rear end of L . From Eq. (2.1) it is seen that a multi-axle vehicle will behave as longer than its geometrical wheelbase, L . Most linear theory on ground vehicles can be used when substituting the wheelbase, l , for the equivalent wheelbase, l_{eq} , [22].

2.1.6 Steering Response

Steady state steering response of a vehicle is commonly measured in terms of lateral acceleration gain or yaw velocity gain. Lateral acceleration gain, $\frac{\partial a_Y}{\partial \delta_H}$, is the relation between change in lateral acceleration and change in steering wheel angle input, where lateral acceleration is denoted \vec{a}_Y . In the steady state linear region it holds that

$$\frac{\partial a_Y}{\partial \delta_H} = \frac{\vec{a}_Y}{\delta_H} = \frac{\vec{v}_X^2}{l_{eq} + K_u \vec{v}_X^2 / g} \frac{1}{i_s} \quad (2.3)$$

where K_u is the understeer gradient having the unit rad, \vec{v}_X vehicle longitudinal velocity and g the gravitational constant. Yaw velocity gain $\frac{\partial \omega_Z}{\partial \delta_H}$, is the relation between change in

yaw velocity and change in steering wheel angle input. In the steady state linear region it holds that

$$\frac{\partial \omega_Z}{\partial \delta_H} = \frac{\vec{\omega}_Z}{\delta_H} = \frac{\vec{v}_X}{l_{eq} + K_u \vec{v}_X^2 / g i_s} \quad (2.4)$$

Fig. 2.5 shows typical steering response for a semi-trailer tractor and a rigid truck as the longitudinal velocity varies. The relative difference in steering response is obvious between the tractor and the truck.

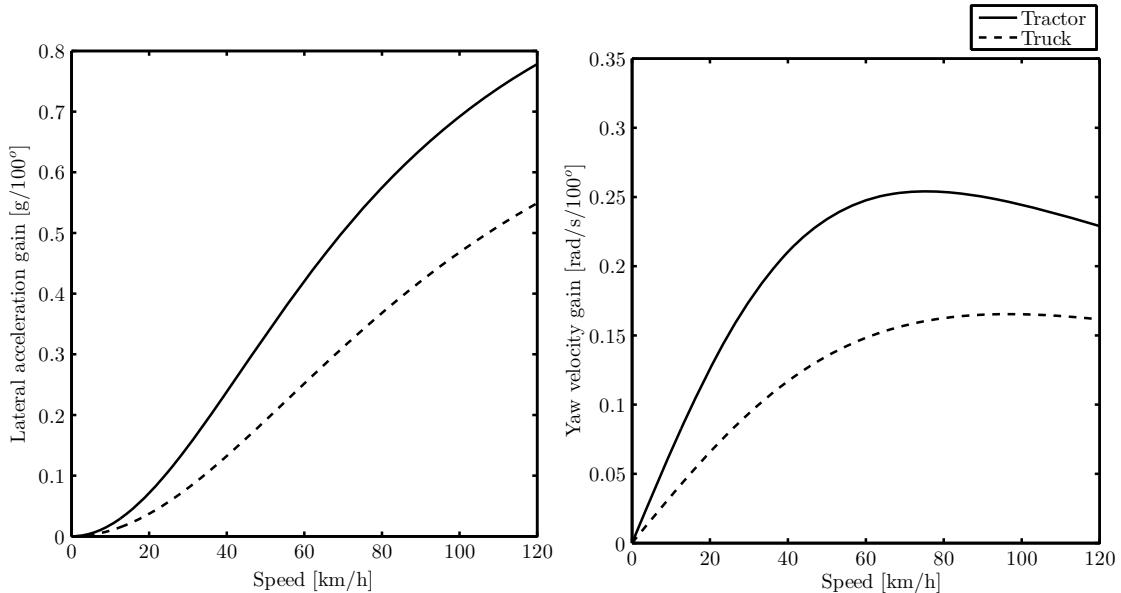


Figure 2.5: Typical steering response shown for a semi-trailer tractor unit and a rigid truck.

2.1.7 Ackermann Geometry

In theory left and right wheel steering angles should be chosen so that the rotation axes always intersect in one point. A point which around all wheels rotate. This would provide highest degree of manoeuvrability and lowest tyre wear. At low speeds this can be derived purely from vehicle geometry and is known as Ackermann geometry. The relation between left wheel steer angle, δ_L , and right wheel steer angle, δ_R then becomes

$$\frac{1}{\tan \delta_R} = \frac{1}{\tan \delta_L} + \frac{b}{l_{eq}} \quad (2.5)$$

where b is the lateral distance between left and right tyre contact patch, known as track. An alternative to Ackermann geometry is to have parallel steering, i.e. $\delta_L = \delta_R$. At high speeds, where wheels are subjected to high side slip, this can in fact provide improved manoeuvrability and lowered tyre wear compared to Ackermann.

The steering geometry is in general closer to Ackermann than parallel on heavy trucks. This is because of the importance of low speed manoeuvrability and that the average speed is low, c.f. cars. Fig. 2.6 provides an example of the steering relation between left and right wheels, taken from a heavy truck.

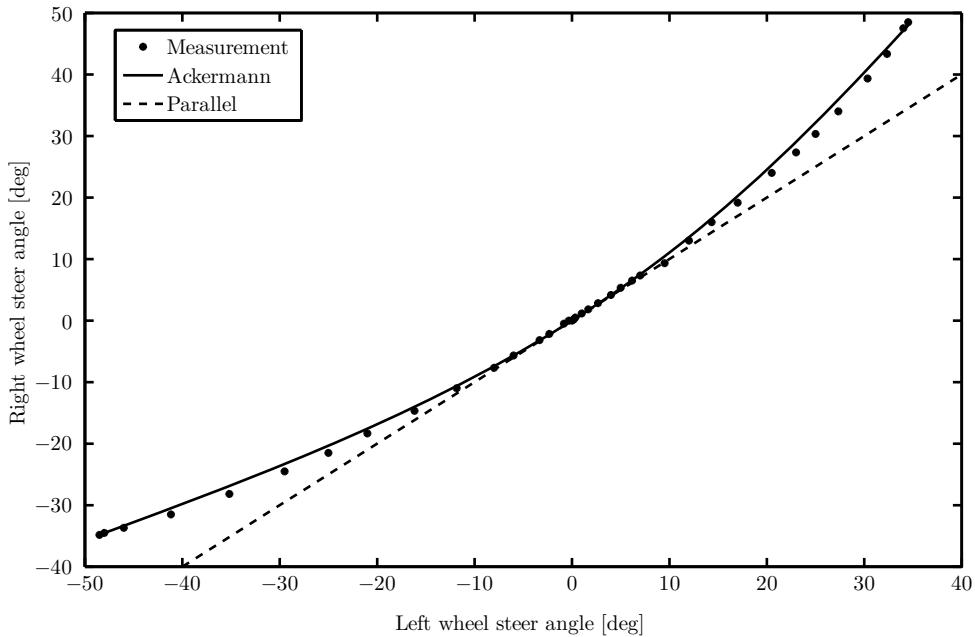


Figure 2.6: The relation between left and right wheel steer angles measured on a specific Volvo truck. It is obvious that it corresponds a lot closer to Ackermann geometry than the parallel geometry.

2.1.8 Induced Steering Error

The relay linkages within the steering system will move as the suspension of the vehicle travels up and down or roll. This will induce a wheel steer angle, disconnected from steering wheel movement. The relation between the joint, connecting the drag link and the upper steering arm, and the geometry of the suspension will be the main way of controlling this effect. The coupling to roll motion, known as roll-steer, in particular is high on some heavy trucks. This adds understeer (or theoretically oversteer with opposite sign) as the vehicle rolls in corners. It can also make the vehicle sensitive to vertical one-sided disturbances.

2.1.9 Steering Forces and Moments

Forces and moments acting on the wheels interact with the steering system. This is well described in [19]. The most significant terms in this will now be described together with other torque components acting on the steering system. This gives an overview on all terms that contribute to the final steering wheel torque, as experienced by the driver. Left and right wheel steer angles are assumed equal, i.e. small steer angles. Furthermore, caster and king pin inclination angles are assumed small and symmetric.

To start with the tyre axis system X_T, Y_T, Z_T is defined. This system coincides with the intermediate axis system X, Y, Z , but X_T, Y_T is rotated around the Z -axis so that X_T coincides with the wheel plane. The wheel is subjected to forces and moments in the X_T , Y_T and the Z_T directions. Forces are denoted as in order \vec{F}_{XT} , \vec{F}_{YT} and \vec{F}_{ZT} . Moments are denoted as in order \vec{M}_{XT} , \vec{M}_{YT} and \vec{M}_{ZT} . The later, \vec{M}_{ZT} , is known as the aligning

moment which has a large impact on the steering system as will be shown.

Resulting Moment from Vertical Force

A vertical force \vec{F}_{ZT} is acting on both left and right front wheels, denoted \vec{F}_{ZTL} and \vec{F}_{ZTR} . The indices L and R will be used in the remainder as left and right. The resulting moment acting on the upper steering arm is

$$M_V = -(\vec{F}_{ZTL} + \vec{F}_{ZTR}) \cdot r_k \sin \sigma \sin \delta + (\vec{F}_{ZTR} - \vec{F}_{ZTL}) \cdot r_k \sin \tau \cos \delta \quad (2.6)$$

Here the first term, which includes kingpin inclination, dominates what is experienced at low speeds in a heavy vehicle. When steering both wheels the vehicle is lifted which causes a returning moment. The second term, including the caster angle, may cause steering pull.

Resulting Moment from Lateral Force

The lateral forces \vec{F}_{YTL} and \vec{F}_{YTR} build up with speed when cornering. Road disturbances can also cause lateral tyre forces. The resulting moment here is

$$M_L = -(\vec{F}_{YTL} + \vec{F}_{YTR}) \cdot r_{stat} \tan \tau \quad (2.7)$$

where r_{stat} denotes wheel radius measured from ground to wheel centre.

Resulting Moment from Longitudinal Force

The tractive forces \vec{F}_{XTL} and \vec{F}_{XTR} caused by e.g. front wheel drive or more likely brake activation acts thorough the steering-axis offset at ground and produce a resulting moment as

$$M_T = (\vec{F}_{XTR} - \vec{F}_{XTL}) \cdot r_k \quad (2.8)$$

As r_k is positive on heavy trucks it causes a destabilising steering wheel torque during split friction braking.

Aligning Moment

The resulting lateral force is in general not acting at the centre of the tyre, as assumed in Eq. (2.7), but further backwards. This distance is known as the pneumatic trail. As stated by [24], the pneumatic trail will reduce as tyre road friction drops. This makes it possible to experience a change in friction level even before reaching the actual friction limit. It should however be stressed that this really requires both a skilled driver and a steering system free from high friction and damping etc. The resulting moment acting on the upper steering arm caused by aligning moment, \vec{M}_{ZT} , is thus

$$M_{AT} = -(\underbrace{\vec{F}_{YTL} \cdot t_L}_{=-\vec{M}_{ZTL}} + \underbrace{\vec{F}_{YTR} \cdot t_R}_{=-\vec{M}_{ZTR}}) \cos \sqrt{\sigma^2 + \tau^2} \quad (2.9)$$

where t_L and t_R denote the pneumatic trail length on left and right wheel respectively, positive backwards from wheel centre. E.g. the Brush tyre model provides an explanation of why the pneumatic trail depends on the current friction level and also lateral slip angle [25]. The pneumatic trail also depends on wheel pressure [24].

Friction Acting on Steering System

The steering system contains several joints, sealing and bearings. All these contribute with a small amount of friction, i.e. elements slide against each other. Together these contributions sum up to a total amount of friction within the steering system. Friction can suppress disturbances, but will also make it impossible for a driver to perceive small force changes between road and wheel. An example of a model for friction is given in [24]. As described a simple coulomb friction model is not representative. Therefore [24] suggests other alternatives, e.g. a spring coupled in series with coulomb friction.

Friction is also present between wheels and road surface at low speeds. This effect often even produces the highest contribution of moment and can therefore be dimensioning for the entire system. It is sometimes argued that steering-axis offset at ground would reduce wheel friction moment. This is shown not to be true in [26] when the wheel is free rolling. The relation between offset and wheel friction level is very weak. When the wheel is locked the friction moment increases as offset is introduced.

Damping Acting on Steering System

Damping, which is a speed dependent torque is acting within the system in several places. Damping is stabilising the steering wheel movement. However too much damping will make the vehicle heavy and slow to steer.

Inertia in Steering System

Steering system inertia mainly comes from the steering wheel itself. This is because of the ratio acting between the lower and the upper side of the system.

2.2 Rack and Pinion Steering System

Rack and Pinion is a mechanism used on most passenger cars. A pinion is connected to a linearly moving rack. It contains fewer joints than the steering gear arrangement. This has the benefit of less compliance and backlash. Heavy truck rack and pinion steering was introduced by Volvo Trucks in 2012 as they launched the individual front suspension, shown in Fig. 2.7. The principle of forces, as presented above, acting on the steering system in Fig. 2.7 still remains however.

2.3 Electric Power Steering System

Electric power steering, EPS, systems consume less energy in general and are easier to control than hydraulic power steering, HPS, systems [27]. This is the reason why EPS

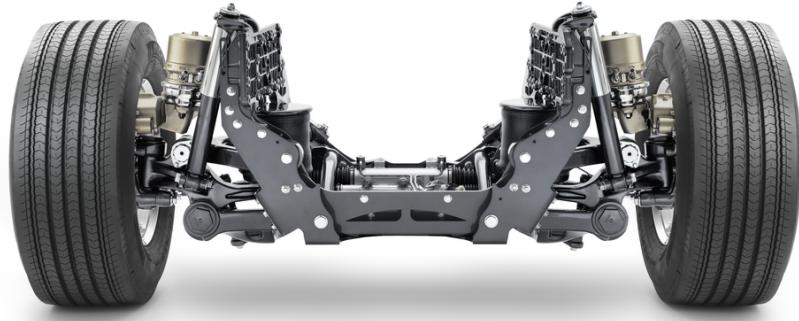


Figure 2.7: Rack and pinion steering is here used on Volvo's individually suspended axle. It has a linear hydraulic piston acting on the rack to provide power steering. The piston can be seen in the middle of the picture, in silver colour. The axle installation is of type double-wishbone.

more or less has wiped out the usage of HPS, for high-end passenger cars. For heavy trucks the story is a bit different. The requirement on power density has made hydraulics retain its position. The difficulty in controlling hydraulic power steering in an exact and quick way has however made room for a compromise. A mixture of the two has been introduced by e.g. Volvo Trucks [28]. A sketch of the system is provided in Fig. 2.8. An electrical motor is placed on top of a hydraulic steering gear. Both add torque on top of what the driver does. This is known as torque overlay.

When introducing electronics into the system it is possible to fundamentally change what is felt in the steering wheel. A pure HPS system can in general only act upon input from the driver. With electronics introduced it is possible to control the system independently of driver input. This is called active steering. With active steering it is possible to have progressive power steering amplification, reduce impact from road disturbances [28], support the driver with lane-keeping aid functions, and a lot more. It is however not possible to turn the wheels independently of the steering wheel.

2.4 Angle Overlay System

In [9] an Harmonic Drive gearbox is installed into a heavy vehicle. This makes it possible to overlay a steer angle on top of the steering wheel angle. Similar systems are used in some high-end cars. The system, as used in research, in [9] is controlled to induce artificial understeer and to change the yaw velocity gain. Angle overlay systems provide an opportunity for changing vehicle response and adding active safety functionality where



Figure 2.8: A combination of EPS and HPS has been introduced on heavy trucks. This picture shows the system offered by Volvo Trucks. An electric motor (highlighted in blue) is placed on top of a conventional hydraulic steering gear.

the driver can be taken out of the loop to a larger extent than what is possible with a torque overlay system. Also an angle overlay system can be called active, when it is controlled independently of driver input.

2.5 Steer-by-Wire System

When removing the mechanical linkage between the steering wheel and the wheels it is possible to control vehicle response completely independent on driver interaction. It is also possible to apply any torque onto the steering wheel, completely independent on road wheel interaction. This is known as steer-by-wire, SbW. Nissan Motor recently introduced the first commercially available car with SbW. It is characterised with "quick response, a high disturbance suppression and straight-line capability as well as wide range of steering ratio settings" [29]. When the mechanical link has been removed very high requirements on redundancy in electronics are needed. This creates a costly system, which previously has been the main reason for not having SbW in production cars or trucks.

Chapter 3

Mapping Results Between Different Vehicle Types

This chapter gives an overview on how steering functions should adapt as vehicle dimensions change.

3.1 Vehicle Differences

In here the fundamental differences between cars and heavy trucks are primarily treated. Also, when comparing two different trucks great differences are often observed, in length and number of axles etc. This relation is also considered. Measures from a typical car and two typical heavy trucks are shown in Tab. 3.1. The table discloses fundamental differences. The lateral acceleration gains show that a car is a lot more responsive than a truck. This is mainly due to shorter wheelbase and a more direct steering ratio. The understeer gradient is also often higher on a truck than on a car. On reason for this is higher compliance in the steering system. The steering wheel torque gradient explains how much torque that is required to achieve a certain change in lateral acceleration. It should be noted that the relation between steering wheel torque and lateral acceleration is highly non-linear, even at low lateral acceleration levels. The gradient is therefore very dependent on the working point being used. It can be said that the value for trucks is higher than what it is for cars, but comparing the exact values should be avoided. In summary several properties differ between a car and a truck. This is also true when comparing different trucks.

Now to the question, how steering functions and results should adapt as physical dimensions change? Here considering one dimension at a time. The discussion is divided depending on whether the function is applying a torque onto the steering system (torque overlay), or if it is applying an angle (angle overlay). For both of these a distinction is made between subjective and objective mapping. *Subjective conservation* is defined as scaling of a steering function such that the experienced balance between handling and comfort is preserved after a change in vehicle configuration. And, *objective conservation* defined as scaling of a steering function such that the measured vehicle response is preserved after a change in vehicle configuration. The following sections will describe scaling of torque feedback where it is considered that one property change at a time.

Table 3.1: Specification of a BMW 116i (denoted **car**), an unloaded Volvo FH 6×2 **tractor** unit (truck with a fifth wheel plate), and a fully loaded solo Volvo FH 8×4 rigid timber **truck** (truck that can carry payload). All data calculated at 50 km/h. Car data found in [8] and from supplier specification. The steering response for the two trucks was previously presented in Fig. 2.5.

Property	Car	Tractor	Truck	Unit	Description
l_{eq}	2.69	3.6	5.9	m	Equivalent wheelbase
m	1330	7500	30000	kg	Total weight
i_s	15	20	24	-	Overall steering ratio
r_{StW}	0.19	0.225	0.25	m	StW radius, measured from centre to rim edge
K_u	0.015	0.08	0.08	rad	Understeer gradient
$\frac{\partial a_Y}{\partial \delta_H}$	0.78	0.3265	0.19	g/100°	Lateral acceleration gain
$\frac{\partial M_H}{\partial a_Y}$	5.86	16	16	Nm/g	Steering wheel torque gradient at $\vec{a}_Y = 0$

3.2 Driver Torque Feedback Adaptation

The torque feedback that is present in the steering wheel is composed from all torque components as described in chapter 2. These are shaped by the power steering system which e.g. results in a certain level of steering wheel torque gradient, as listed in Tab. 3.1. There are however a lot more properties that are of importance when describing steering characteristics. The importance of steering torque has been analysed in several studies [30, 31, 24]. Different drivers often have different opinions about the optimal level of feedback [32, 33] when considering subjective rating.

3.2.1 Steering Wheel Size

In [10] it was concluded that a driver perceives force rather than torque. This was based on a test where the StW angular degree of freedom was locked, known as isometric test. When the angular degree of freedom was unlocked and torque set to zero it was further found that a driver perceives StW angle rather than hand translation. In Paper 1 a complementary study is performed analysing how force feedback should vary with StW size in a real vehicle where isometric motion no longer holds and where the driver subjectively decides on optimal balance between handling and comfort. Also analysed is objective conservation of a steering pulse when StW size changes.

Subjective Tuning of Base Characteristics

A method was developed to scale the complete steering wheel torque in a truck. See Fig. 3.1 for a visual illustration. A single scaling parameters k_g is used to scale all torque components contributing to the steering characteristics according to

$$M_H(\delta_H, \dot{\delta}_H, \vec{v}_X) = k_g \cdot M_{H,0}(\delta_H, \dot{\delta}_H, \vec{v}_X) \quad (3.1)$$

where $\dot{\delta}_H$ denotes steering wheel angular rate, $M_{H,0}$ baseline steering torque characteristics (solid lines in Fig. 3.1) and k_g a scaling parameter.

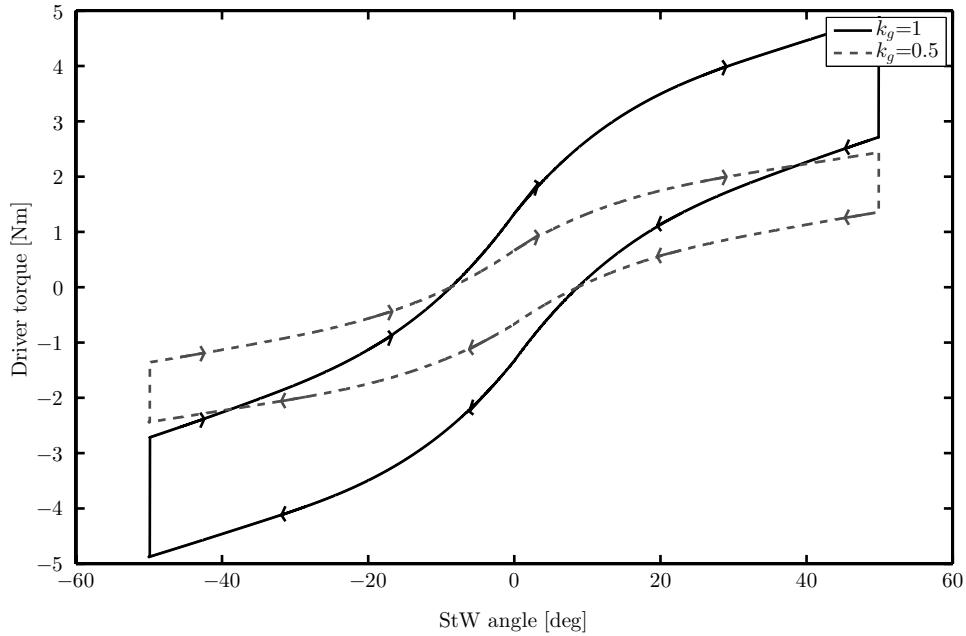


Figure 3.1: A method was developed making it possible to scale the complete steering wheel torque as experienced by the driver, using only one single parameter k_g . The characteristics shown is measured at 80 km/h on the truck used in the experiment. From Paper 1.

A test was run where 17 subjects decided on their preferred value of k_g . Each subject ran with totally three differently sized steering wheels. The steering wheels are denoted as *large*, *medium* and *small*. This corresponds to a steering wheel radius, r_{StW} , of 0.225 m, 0.195 m and 0.165 m. The test took place on a handling track and subjects were told to stay between 45 km/h to 90 km/h.

The reported optimal level of k_g is shown in Fig. 3.2. The variance in trend between subjects is large. This is also expected from existing knowledge on the ability of humans to differentiate steering stiffness. The results indicate that torque feedback should be scaled when StW size changes. And suggested as a rule of thumb, to use linear scaling of total torque in order to accomplish maintained driver force level. It was however noted that further adjustment of damping, friction etc. might be needed to realise conservation of steering wheel free response return rate.

Objective Evaluation of Pulse Scaling

All 17 subjects also took part in an objective part. They were told to continue driving around the track with both hands on the StW. Then an operator fired off several StW torque square pulses. The pulses were 1 s in duration and the size was set to $-3 \cdot k_g$ Nm.

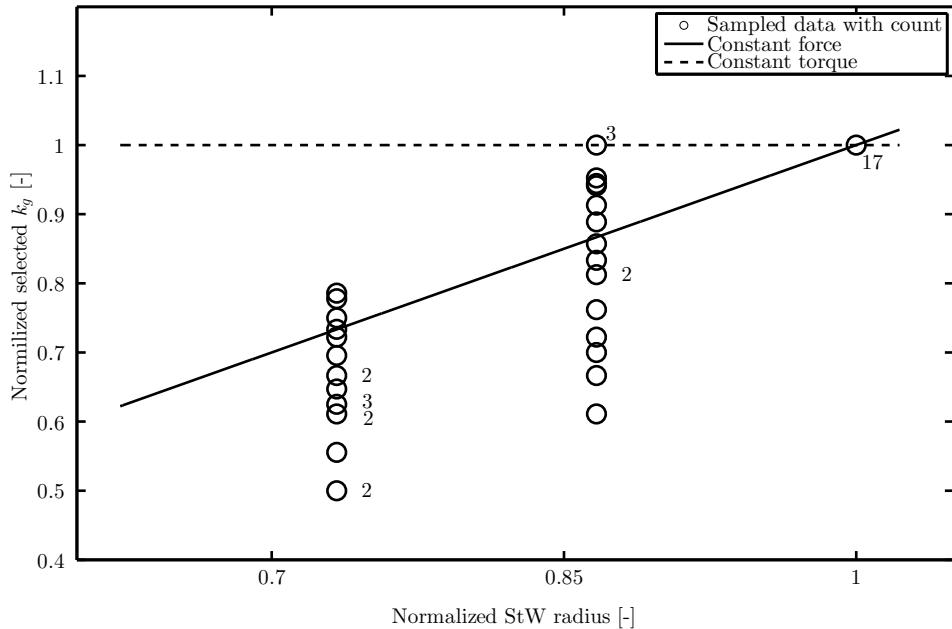


Figure 3.2: k_g values are divided with k_g from *large* StW per subject. r_{StW} is divided with r_{StW} from *large* StW. Numbers are used to denote multiple occurrence of data-point. Also included are two lines corresponding to constant force and constant torque respectively. From Paper 1.

Where k_g was set in random order to value 1.0, 0.85 or 0.7. These levels roughly correspond to StW radius in relation to the *large* StW radius. Eq. (3.1) was still applied, i.e. both the continuous characteristics and the pulse were scaled with k_g . This was again repeated for all three steering wheels.

In total 858 pulses were recorded above 50 km/h, and where no obvious steering motion was ongoing at the start of the pulse. The relative changes in steering wheel angle after 0.25 s from these are shown in Fig. 3.3. Not obvious from the figure, but when looking closer there is in fact a small variation in the response observed for the different steering wheels. The best match found for preserving driver angular response, and hence vehicle lateral response, was to scale the torque pulse in inverse relation to StW size, i.e. preserve StW force. In fact, StW angle change was very similar for all the three steering wheels when the equal force approach was applied. It is therefore suggested to use the same rule of thumb as in the subjective section, i.e. use linear scaling of total torque to accomplish maintained driver force level when StW size is changed.

Common Conclusion

In order to transfer steering functions and map results as the steering wheel size is changed the common conclusion is that StW force should be conserved. This will ensure both subjective and objective conservation. For the objective part it is important to note that the force that is applied to the hands of a driver not only originate from the added torque pulse itself. All other force components, as described in chapter 2, will also have to be accounted for.

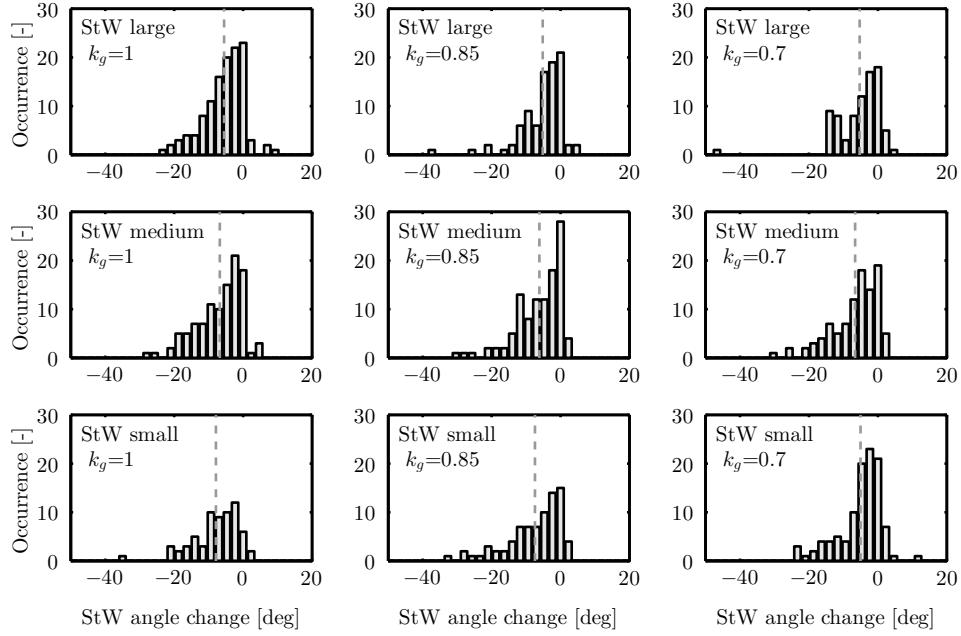


Figure 3.3: Histogram of StW angle delta change 0.25 s from the start of the pulse. Mean value of samples is shown with a vertical line. The y-axis show number of times that the delta change was observed. From Paper 1.

3.2.2 Steering Ratio, Wheelbase & Understeer Gradient

In [11] it was found that subjective rating was independent of vehicle type if the magnitude of disturbance was measured in terms of lateral accelerations or yaw rate¹. This is also supported by [12] where it was found that yaw velocity gain is the primary cue used by drivers when comparing the response between two vehicles. On the other hand in [9] nine dimensions are identified as important for subjective rating of a vehicle. It is therefore not practically possible to completely conserve subjective rating of a function when transferred between vehicle. However just like for StW size some rough rules can be developed. Starting from the findings in [11, 12] it can be assumed that conservation of lateral acceleration is consistent with subjective conservation. Objective and subjective conservation are thereby equal.

Lateral acceleration gain, that is exemplified in Tab. 3.1, can be calculated in the linear region with Eq. (2.3). It can be seen that a change in overall steering ratio, i_s , will leave the lateral acceleration \vec{a}_Y conserved if δ_h/i_s is constant, i.e. leaving road wheel steer angle unaffected. When changing wheelbase the following relation must hold to conserve lateral acceleration

$$\frac{\Delta\delta_H}{\delta_H} = \frac{\Delta l_{eq}}{l_{eq} + K_u \vec{v}_X^2/g} \quad (3.2)$$

where $\Delta\delta_H$ is the required change in StW angle to account for a change in equivalent wheelbase, denoted Δl_{eq} . As seen the adaptation is speed dependent.

¹Yaw rate and lateral acceleration are closely coupled for steady state cornering in the linear region. Conservation of these two is therefore treated as exchangeable.

A change in the understeer gradient, denoted ΔK_u , can be accounted for in a similar manner as wheelbase. It results in a required change in StW angle according to

$$\frac{\Delta \delta_H}{\delta_H} = \frac{\Delta K_u \cdot \vec{v}_X^2}{gl_{eq} + K_u \vec{v}_X^2} \quad (3.3)$$

From this it can be seen that a change in overall steering ratio, wheelbase or understeer gradient requires a change in StW angle. This is to be realised using a change in overlaid steering torque. When applying torque onto the StW the driver responds with hand force. This process is as seen from Fig. 3.3 coupled to large variance. When no hands are placed on the StW the required change in StW torque can be calculated from steering characteristics, see e.g. solid line in Fig. 3.1. When a driver is part of the loop, driver admittance should also be included, e.g. see [34].

3.2.3 Discussion

When comparing torque overlay results or when mapping a torque overlay function it is important to consider fundamental physical properties. For StW size this means that driver force should be conserved. It is here important to recall that all force components acting in the steering system should be considered.

Other physical properties of high importance are steering gear ratio, wheelbase and understeer gradient. When these are changed a conservation of lateral acceleration response should be pursued.

3.3 Angle Overlay Adaptation

An angle overlay function should be scaled for objective conservation. Subjective conservation is not applicable as there is no direct connection to the driver other than via vehicle response and small StW torque disturbances. Changing the wheelbase or the understeer gradient would call for adaptation as both wheelbase and understeer gradient influence vehicle response. As the overlaid angle is commonly applied closer to the wheels Eq. (3.2) is rewritten for convenience as

$$\frac{\Delta \delta}{\delta} = \frac{\Delta l_{eq}}{l_{eq} + K_u \vec{v}_X^2 / g} \quad (3.4)$$

where $\Delta \delta$ denotes the required change in road wheel steer angle to account for a change in wheelbase. When doing the same thing for a change in understeer gradient it becomes

$$\frac{\Delta \delta}{\delta} = \frac{\Delta K_u \cdot \vec{v}_X^2}{gl_{eq} + K_u \vec{v}_X^2} \quad (3.5)$$

When a steering function is working in a closed loop fashion, i.e. taking in a vehicle state and outputting an overlaid steer angle, it will by nature account for the relative change in vehicle properties. However control gains should be updated according to Eq. (3.4) and Eq. (3.5) in order to preserve performance.

Chapter 4

Driver Reaction in Selected Use Cases

In this chapter driver reaction is studied in two different but similar scenarios. To start with, automatic braking when triggered on a split friction surface. And secondly, front tyre blow-out.

4.1 Common Ground

As discussed in chapter 1 a driver of a heavy truck is really tested to extremes when a front tyre explodes. As the tyre is torn to shreds it can produce a lot higher rolling resistance than a normal tyre. At worst it even stops rolling and instead develops full slip, similar to a locked up tyre. This will first of all induce a yawing torque on the truck. It will also exert abnormal forces on the steering system. Furthermore, a truck towing one or more trailers will experience forces in the connection point because it is only the towing vehicle that is braked from the disturbance. If an angle has developed, between the units, this force will act destabilising on the towing vehicle. Combined, these effects can result in run off road, collision with oncoming vehicles, roll-over or jack-knife, unless the driver is able to balance the effects by steering or braking. When designing vehicles it is therefore important to know how a driver reacts at a tyre blow-out.

The other scenario of interest automatic braking on split friction or more general, automatic braking activated when a yaw disturbance is present. This scenario has a striking similarity to the blow-out case. It contains all the above elements, a yawing torque, a destabilising StW torque, and can induce forces in the trailer connection point. The main differences are the relative levels and that the truck will decelerate more heavily in the automatic braking case.

In order to better understand driver behaviour in these two cases a test was set up with a 9 ton solo tractor. All tests were performed on an even test-track with a group of volunteers. The brake system was first controlled to emulate automatic braking on split friction. After this blow-out was emulated by locking the front left wheel using the brake. Paper 2 and Paper 3 contain details on the set-up and outcomes from these two parts respectively. The following sections give a summary of the experiments and a common discussion.

4.2 Automatic Braking Activated on Split Friction

In this scenario 12 drivers were exposed to sudden automatic braking. They were not aware of the true purpose of the test in order to preserve the effect from surprise. Drivers were told that the intention of the test was to record normal positioning in lane and that they should run back and forth inside a 300 m straight lane. Cruise control was set to 50 km/h. After running back and forth for 5 minutes, without any intervention, an operator fired off automatic braking. After the first unexpected intervention two repeated runs were made at the same speed, followed by two more at 70 km/h.

No other vehicle was nearby; therefore cones were put in the adjacent lanes creating a sense of danger. Fig. 4.1 provides an illustration of the set-up.

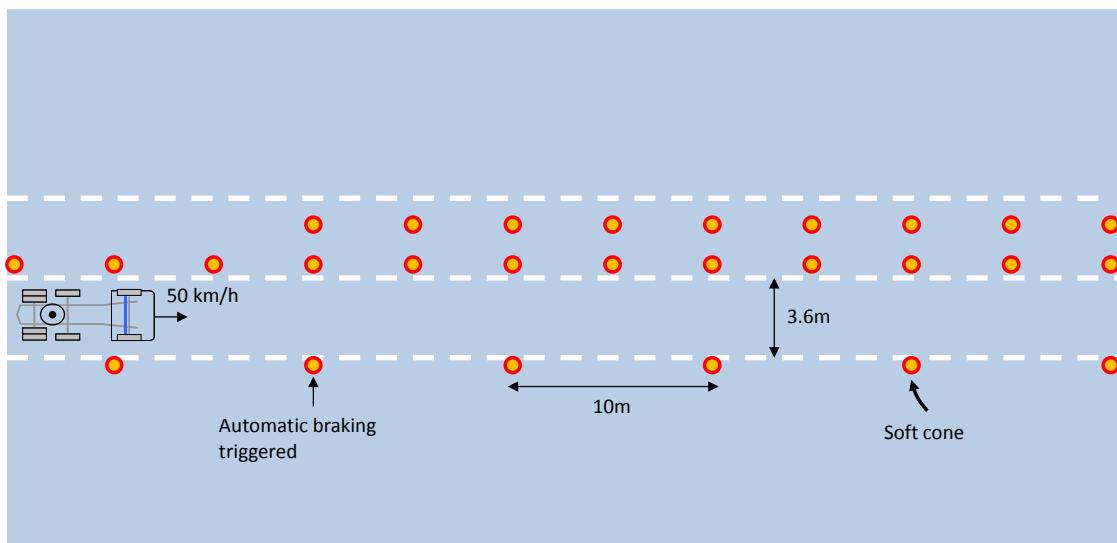


Figure 4.1: A sketch of the first exposure of automatic braking with split friction emulated using the brakes.

4.2.1 Controlling the Brakes

Regulations for AEBS [16] state that the system should be designed to "avoid autonomous braking in situations where the driver would not recognise an impending forward collision". This implies that AEBS at least should be capable of decelerating at 3.5 m/s^2 during the emergency braking phase. Here assuming that brake initiation is delayed until TTC 4.0 s, [35, 36], and that the brake system has a delay of 0.2 s from brake request until full deceleration is reached. Therefore as target deceleration 3.5 m/s^2 was used in a brake controller.

Left to right brake force distribution was determined by running on a real split-friction area with standard functionality enabled to comply with present legal requirements for split-friction braking [37]; and where the induced yaw velocity was recorded. The magnitude of yaw velocity was replicated, on a even road, when a fixed ratio of four times as much brake action was used on left side as on right.

The final brake controller used was a longitudinal acceleration feedback PI-controller with added feedforward.

4.2.2 Results

All 12 trajectories relating to the very first exposure of automatic braking interventions are shown in Fig. 4.2. The stopping distance, counting from brake onset, ranges from 30.2 m to 33.3 m. Two drivers instinctively deactivated the intervention by pressing the accelerator pedal. This corresponds to the two trajectories that continue to travel even after 35 m. The mean maximum lateral deviation was 0.25 ± 0.07 m, using 95% confidence level. Two drivers deviated by 0.5 m. All values given in the figure relate to the position of the drive axle. The mean maximum lateral deviation at the front axle is 4 cm higher than that of the drive axle, due to the heading of the vehicle. Also included in Fig. 4.2 is an open loop response produced by locking the steering wheel. It deviates by 2.2 m at standstill.

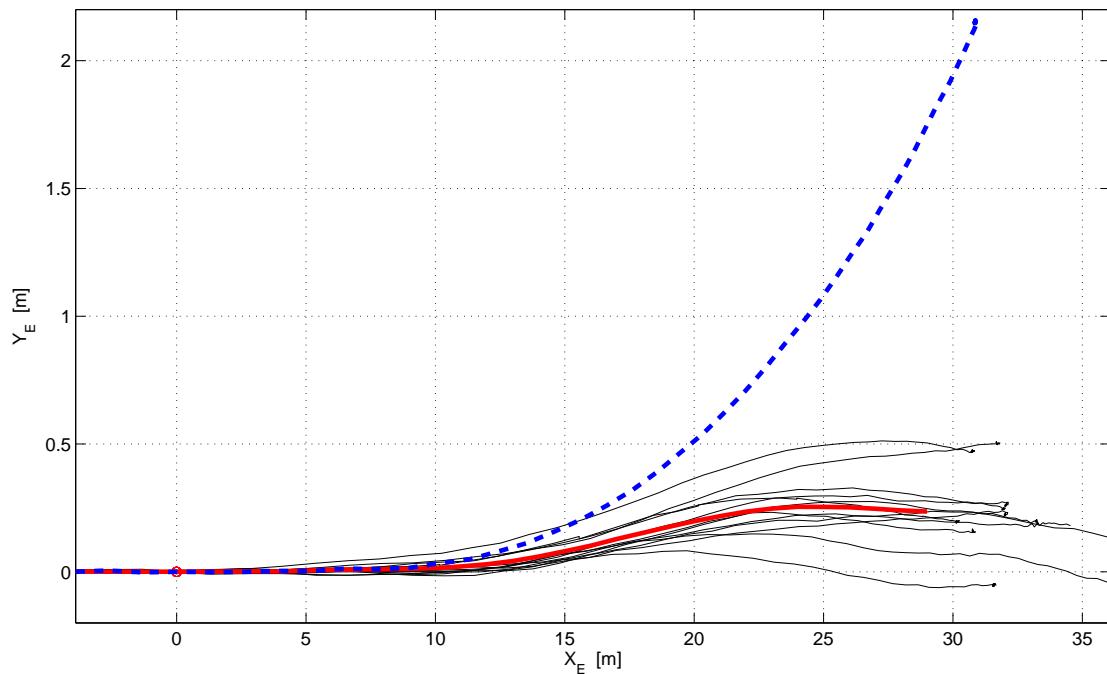


Figure 4.2: Position of tractor rear axle during unexpected brake intervention, initiated at (0, 0) m.

One black solid curve per driver, thick solid red is average of all drivers, dashed thick blue is reference run with fixed steering.

Corresponding time series are shown in Fig. 4.3. Looking at the speed curves it can again be seen that two drivers instinctively deactivated the intervention by pressing the accelerator pedal. The second subfigure displays change in StW angle, calculated as

$$\Delta\delta_H = \delta_H(t) - \delta_H(0) \quad (4.1)$$

where $\delta_H(t)$ is the StW angle at time t , and $t = 0$ correspond to brake activation. Some drivers responded with a smooth and steady movement of the steering wheel, whereas others oscillated widely. The positive steering-axis offset at ground which acts destabilizing, see Eq. (2.8), can be observed in the StW torque plot. Around -2.5 Nm of the disturbance reached the driver. As seen in the last subfigure yaw rate is in general shaped

as a one period sine wave. The corresponding frequency, 0.5 Hz, happens to match the resonance frequency of several truck combination types, see [38].

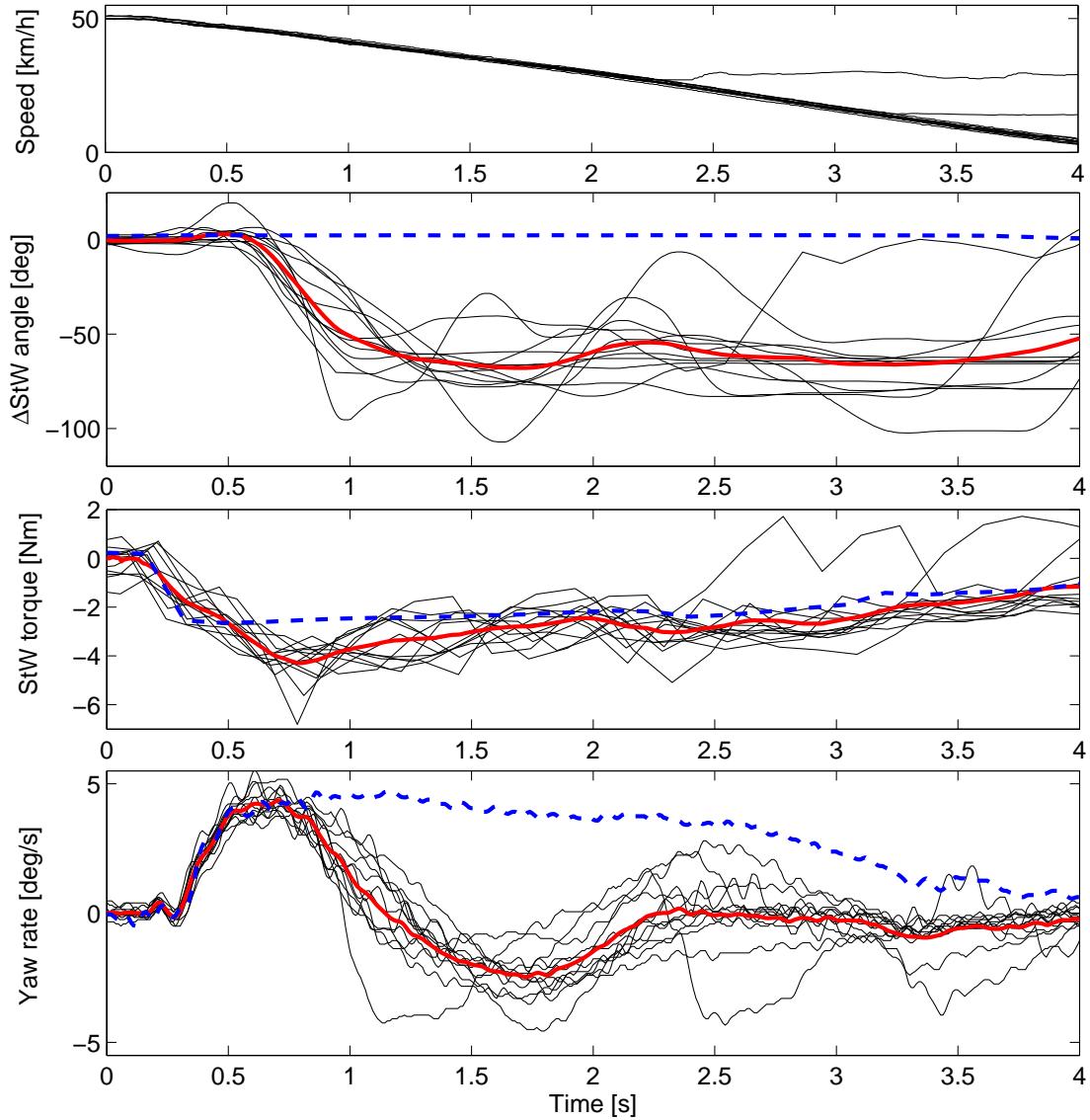


Figure 4.3: Response to unexpected brake intervention, starting at time 0 s with a short delay for brake activation. Line styles same as in Fig. 4.2.

When repeated runs were performed, at 50 km/h, the lateral deviation was nearly halved on average. The average maximum lateral deviation observed was 0.13 ± 0.03 m, again using 95% confidence level. The underlying reason for this is identified as shorter reaction time. In the repeated runs performed at 70 km/h the average maximum lateral deviation observed was 0.10 ± 0.04 m.

4.3 Tyre Blow-out & Steering Wheel Forces

The positive steering-axis offset at ground which acts destabilizing, see Eq. (2.8), is as stated of importance as a tyre blow-out occurs. A deflated tyre has a smaller radius than

a normal tyre. The resulting steering-axis offset at ground will hence increase and induce higher torque onto the steering system, see Eq. (2.8), [39]. In [17] a truck simulator study was run where it was concluded that the effect of surprise is the main factor to consider, in order to be able to replicate lateral deviation as observed in real accidents. This is not targeted in this experiment. Instead the role of steering-axis offset at ground is analysed.

Drivers taking part were not aware of the intention of the test, but had been exposed to the automatic braking scenario. After this several repetitions of emulated tyre blow-out was carried out. Cones were again used and put in adjacent lanes to create a sense of danger and a reason to maintain the intended lane. Data from totally 20 subjects is included in the analysis that follows.

By using a modified EPS system it was possible to change steering-axis offset at ground virtually. Two settings were configured 12 cm and 0 cm, where the order used was varied. Each driver was exposed to three blow-outs per steering-axis offset. The front left brake was applying 350 kPa. This level was selected just below tyre locking. The produced tyre force was thereby nearly maximised, but discontinuities relating to ABS control was eliminated. The relatively high level was selected to produce worst case blow-out forces, which is still not far above what has been observed during real blow-out, see e.g. [40]. In the case that the driver pressed the brake pedal a select high pressure routine was used. If the driver pressed the accelerator pedal the test was aborted.

4.3.1 Results

Fig. 4.4 contains all trajectories produced for front left blow-out runs. Black colour is used for runs with 12 cm steering-axis offset at ground. Red colour is used for runs with 0 cm offset. Bold lines are used for average. The produced average lateral deviation from the original direction is 23 cm, when steering-axis offset at ground is 12 cm, compared to 16 cm on average, when steering-axis offset at ground is 0 cm. There is however large variance in data, so a direct comparison will not prove a significant difference. Instead the relative improvement per subject was tested with a paired t-test. This shows that the average lateral deviation was lowered by 6.4 ± 4.4 cm, using a 95% confidence interval. This is calculated after 24 m of longitudinal displacement, where the maximum deviation occurs on average.

Fig. 4.5 contain corresponding time series. Colouring used is the same as in Fig. 4.4. The speed profiles are as expected similar for all runs apart for some where the driver has pressed the brake pedal gently. The following subfigure is showing change in StW angle, as defined in Eq. (4.1). Here, early overshoots indicate that some drivers are affected by the applied destabilising StW torque. Again a paired t-test was run, showing a significant difference between 0.3 s and 0.5 s. The StW torque curves show an apparent difference between the two settings used. From the last subfigure it can be seen that the yaw rate response roughly show a one period sine wave, c.f. Fig. 4.3. Corresponding frequency, 0.7 Hz, also happens to match the resonance frequency of several truck combination types.

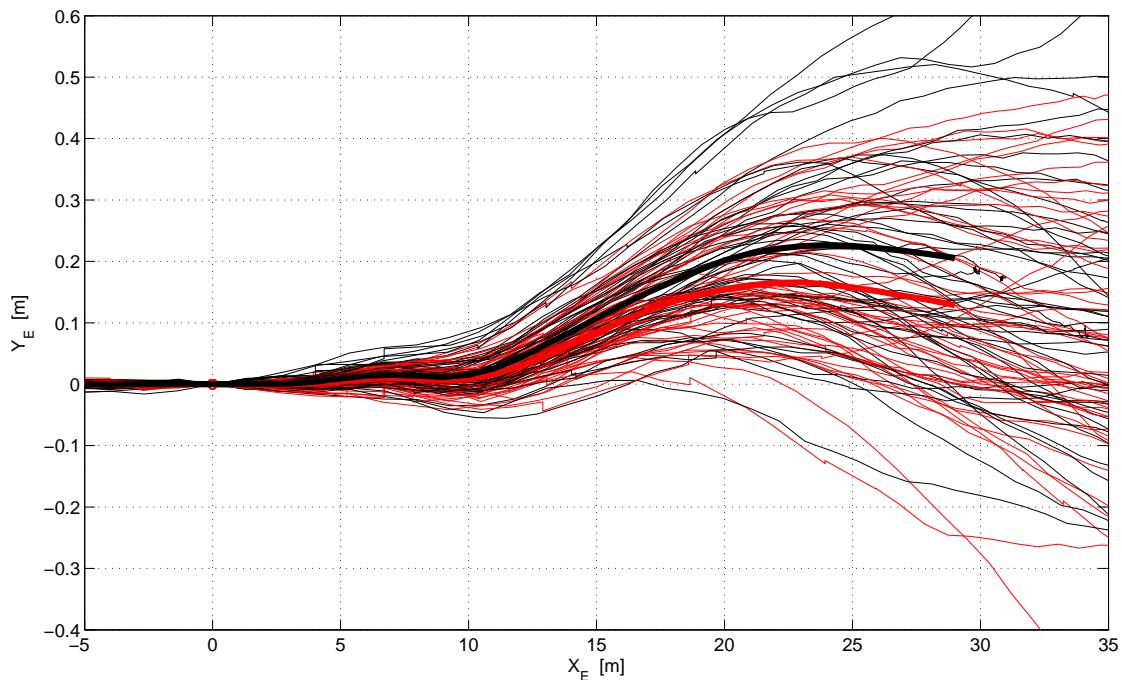


Figure 4.4: Position of drive axle for all emulated front left blow-out runs. The curves have been rotated and moved so that blow-out is initiated at position (0,0) m running at zero heading. Thin red lines correspond to steering-axis offset at ground 0 cm. Thin black lines correspond to steering-axis offset at ground 12 cm. Bold red line correspond to average of 0 cm runs. Bold black line correspond to average of 12 cm runs.

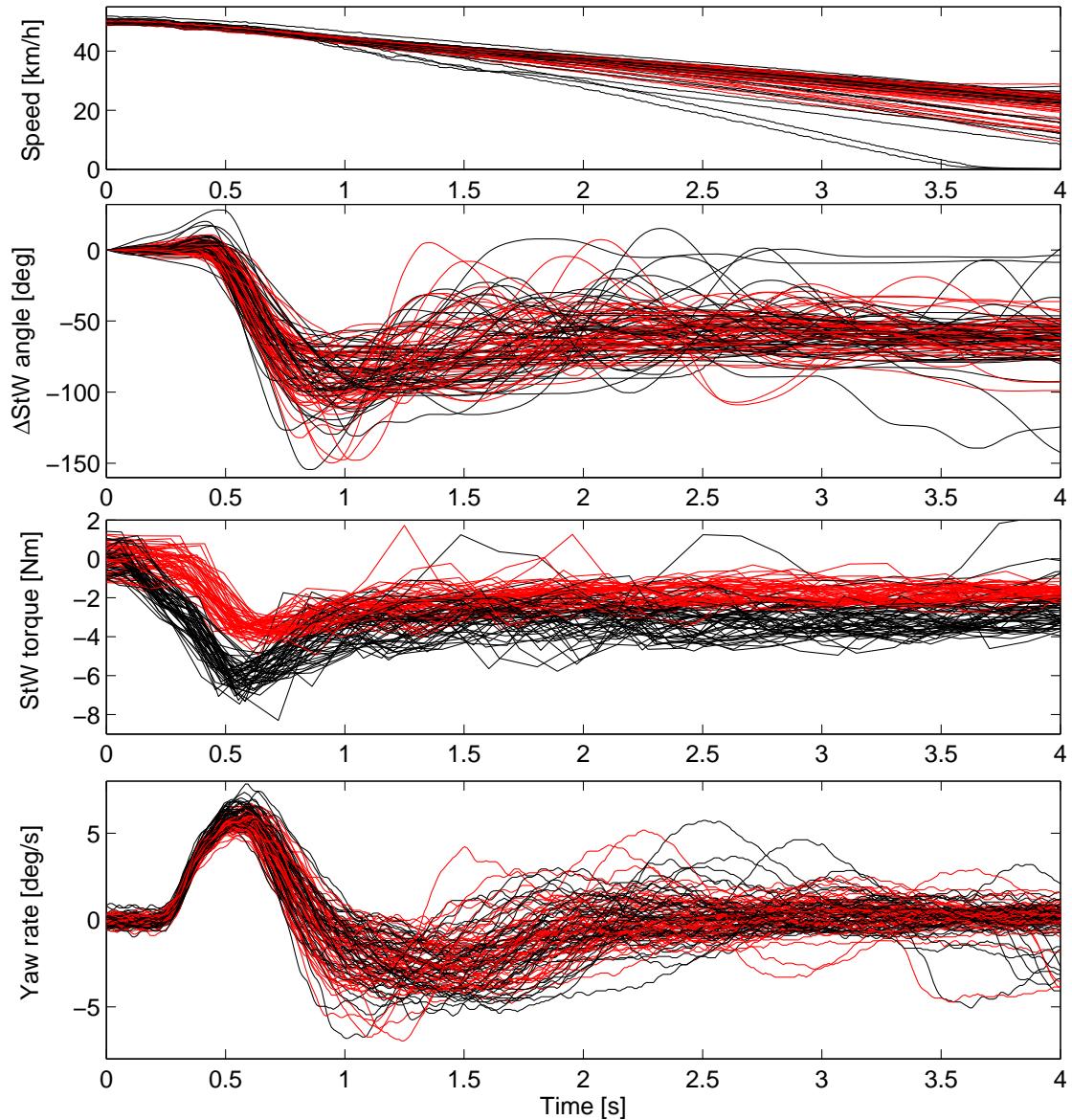


Figure 4.5: Time series for all emulated tyre blow-out runs. The blow-out is initiated at time 0 s. Red lines correspond to steering-axis offset at ground 0 cm. Black lines correspond to 12 cm.

4.4 Discussion

For the automatic braking case the lateral deviation observed was higher in the first runs, when drivers were unaware, compared to repeated runs. An identified reason for this was shorter reaction time. Measured levels suggest that the risk of collision, due to lateral deviation, is low for an alert driver. For a distracted driver more support might be required. As was obvious from the repeated runs an aware driver, knowing what will come, is more effective in reducing lateral deviation. This underlines the fact that the warning phase, which is already a part of AEBS, is important.

For the blow-out case a small yet significant difference was observed in lateral deviation when steering-axis offset at ground was reduced. The difference would increase for drivers who do not have a firm grip on the StW. In particular, the improvement for drivers not holding the StW at all would be several meters. Also the risk of roll-over would be very high.

In both cases the yaw response frequency matches the resonance frequency of several truck combinations. It is however not clear if driver response would be identical when trailers are included.

Chapter 5

Concluding Remarks & Future Challenges

In this chapter the overall conclusions from the thesis are stated. This is followed by ideas identified for future research.

5.1 Conclusions

The key objective of this work has been to strengthen the knowledge about driver and steering interaction. This knowledge is intended to be used when designing effective active safety functions. The main conclusions from this are here given in a list.

- First of all the theoretical overview on important components, forces, and torques in the steering system discloses that new technology will enable a paradigm shift. New technologies like electronic power steering, angle overlay and steer-by-wire open up for a plethora of functions. The forces acting on the wheels will however remain and should be taken into account when designing everything from a completely new steering system to just a new steering function.
- In order to better utilize steering research and development in other areas than the heavy truck side a lot of attention has been spent on trying to map functions and results between vehicle platforms. It can be concluded that some tuning will always be required when functions are e.g. inherited from cars to trucks. It is however possible to create a set of simple rough rules, describing how to scale functions and results when important dimensions change. From this the main findings are
 - Steering wheel size: Drivers perceive force rather than torque, it is therefore important to scale steering wheel torque. Here the entire steering characteristics should be considered.
 - Steering ratio, wheelbase and understeer gradient: For torque overlay a change in any of these dimensions will call for scaling of steering wheel torque. The degree of change is however depending on driver admittance when considering vehicle response. When the driver is considered out of the loop, it will be the steering characteristics that will provide the final link for scaling. For angle overlay the relation is somewhat simpler. Here, it is essentially vehicle

response that should be conserved. This can be derived from basic vehicle dynamics theory.

- In order to understand driver behaviour when brakes are automatically triggered and a yawing disturbance is present an experiment was performed emulating split friction. The conclusions are
 - Measured levels suggest that the risk of collision, due to lateral deviation, is low for an alert driver. For a distracted driver more support might be required.
 - A powerful way of reducing lateral deviation is to prepare the driver on what will come. This will reduce reaction time which has a substantial influence on the resulting lateral deviation.
 - The induced yawing disturbance should be limited based on driver capabilities.
 - A reduction in steering-axis offset at ground could potentially reduce the lateral deviation. This has not been investigated.
- Heavy trucks have positive steering-axis offset at ground, also known as kingpin offset at ground. To better understand what consequences this has in a front tyre blow-out scenario another experiment was carried out. The conclusions are
 - Elimination of steering-axis offset at ground has a small, yet statistically significant, effect on the lateral deviation induced at tyre blow-out.
 - The main factor for high lateral deviation in a real accident scenario is however the effect from surprise, as seen in previous research [17], and in the automatic braking scenario.
 - Proper countermeasures for blow-out accidents are identified as, brake based electronic stability control developed specifically for blow-out, steering guidance, and tyres that cannot blow-out other than slowly.

5.2 Future Steps

Based on the recommendations given, for the two scenarios in focus, it is possible to take actions. This can potentially save lives in the continuation. In this, the importance of the truck combination resonance should be analysed further.

Also recommended for future studies is the admittance of a driver to comply with steering guidance in various scenarios and how this guidance should be designed to be effective.

In general, when designing vehicle motion support systems, it is important to consider the capacity of the driver to handle possible vehicle disturbances and consequences thereof. If this is done in real time it would even be possible to design less conservative systems. E.g. for an automatic braking system the stopping distance, on split friction, could be reduced when either the driver is in an alert state or when the traffic situation makes this possible.

Trucks are driven with trailers. This is often forgotten when considering driver vehicle interaction. What understanding do truck drivers have about the motion of the trailer? And is it possible to add on to this understanding e.g. to reduce number of blind spot accidents? Two questions that exemplify a research field that is still open.

Human drivers can adapt and assess new situations. Computer controlled vehicles can do boring things millions of times in a reliable and quick way. Let's make the best out of this.

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Paper 1

Title: The influence of steering wheel size when tuning power assistance

Journal: Int. J. of Heavy Vehicle Systems, 2014 Vol.21, No.4, pp.295 - 309

DOI:10.1504/IJHVS.2014.068099

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Version: Author's post-peer review

To cite this article: K. Tagesson, B. Jacobson, and L. Laine, "The influence of steering wheel size when tuning power assistance," *International Journal of Heavy Vehicle Systems*, vol. 21, no. 4, 2014.

To link to this article: <http://dx.doi.org/10.1504/IJHVS.2014.068099>

Notation conversions: $T_d = M_H$; $T_{d,0} = M_{H,0}$; $\theta = \delta_H$; $v_x = \vec{v}_X$

The Influence of Steering Wheel Size when Tuning Power Assistance

Abstract: This paper describes how steering assistance should scale with steering wheel size. A method has been developed to scale complete torque felt by the driver, both for continuous and discontinuous feedback. This was used in an experiment with 17 subjects all driving a truck with three differently sized steering wheels. The test took place on a handling track at 45 km/h to 90 km/h. Continuous feedback was evaluated subjectively; discontinuous feedback by measuring angular response. Results show that torque feedback should decrease as steering wheel size decreases. A rule of thumb is to keep driver force level constant to maintain perceived handling and comfort. This also maintained the average steering wheel angle change response to discontinuous assistance. Furthermore large variance in angular response was observed. The direction, measured 0.25 s after start of a pulse, was the same as that of the pulse applied in 88% of the recordings.

Keywords: steering wheel size; tuning; power assistance; steering assistance; torque feedback; force feedback; trucks; angular response; handling; comfort; heavy vehicles; steering wheel diameter; steering wheel radius; steering haptics

1 Introduction

Power steering, also known as power assistance, has been used for many years to lower the required steering effort when turning a vehicle or a vessel (Howe, 1956). In recent years various more advanced steering systems have emerged. These include new refined methods for applying assistance torque and changing steering ratio (Heissing and Ersoy, 2010). Also, they are most often electronically controlled which allows for a greater flexibility in behaviour. It also opens up for precise tuning of steering support level at every speed and driving situation.

The steering wheel size, i.e. the diameter, in cars has been kept rather constant historically. Today it is 35-40 cm in most passenger cars (Wheelskins-Inc, 2009). In heavy trucks it is larger as a consequence of legislation. As stated by UNECE (2005) the driver should be able to manoeuvre with limited steering forces also in the case of an assistance failure. With common wheelbase and steering ratio this typically means a steering wheel, StW, diameter of 45-50 cm on modern heavy trucks. Until recently truck power steering systems have exclusively been hydraulic (Volvo-Trucks, 2013; ZF-Lenksysteme, 2012). When moving away from pure hydraulic power assistance systems and introducing new redundant electronic steering systems the legislation requirement becomes less relevant. Hence, the StW size could be chosen more freely also in a heavy truck.

The importance of force feedback for vehicle handling has been analysed in several studies (Ciarla et al., 2012; Kim and Cole, 2011; Pfeffer, 2006). Anand et al. (2011) carried out a simulator study where steering effort was tuneable when driving. The subjects reported their preferred level when satisfied. It was shown that the level varied highly between

individuals; this is also confirmed by Barthenheier and Winner (2003). Newberry et al. (2007) performed a thorough experiment to analyse what aspect of force feedback that a driver senses - whether it is torque or force. An indoor test apparatus with a strain gauge was used to measure StW torque. The StW angular degree of freedom was locked (isometric test). It was concluded that a driver perceives force rather than torque. When the angular degree of freedom was unlocked and torque set to zero it was further found that a driver perceives StW angle rather than hand translation. The StW centre position was not adjusted when changing StW.

It has not been shown how force feedback should vary with StW size in a real vehicle where isometric motion no longer holds and where the driver subjectively decides on optimal balance between handling and comfort. Also no such test has been performed where the subjects are free to choose seating and StW position which could be different when changing StW. This is the first identified gap analysed in this paper.

Now, continuing on next topic. In recent years new functions have been developed where more than vehicle forces can be sensed in the StW. Examples are functions for positioning in lane, crash avoidance and vehicle stabilisation (Rossetter, 2003; Volvo Cars Corporation, 2008; Yang, 2013). Some of these are not continuous, meaning that the torque which is added to the steering column is ramped up almost instantaneously. The intention is not only to provide tactile information to the driver. It is also to directly effect the steering wheel angle before the driver has had time to understand the seriousness of the situation. During that time the driver is however still in contact with the StW. This highly effects the angular response of the additional torque.

Cole (2012) developed a driver-vehicle model which was used to analyse a driver's ability to do path-following when exposed to a lateral force disturbance. Also driver response from both angle and torque disturbance inputs was analysed. The model was validated with a driving simulator experiment where angle overlay was performed. Groups were divided as tensed or not tensed. Abbink et al. (2011) measured driver frequency response to torque disturbance in a fixed based simulator. Pick and Cole (2007) used a similar set up to validate a linear mass-damper-spring model. The angular response was shown to be captured up to 6 Hz for random torque disturbances. The model contained arm inertia, damping and stiffness. It was observed that damping and stiffness increased as the driver co-contracted.

Several models have been developed and tests have been performed to study driver response to discontinuous StW torque feedback. No in vehicle measurements have been performed to show differences of driver response when StW size is changed and when the tension level of the driver is a consequence of the main mission of following a real road, or similar. This is the second identified gap analysed in this paper.

When changing StW size old knowledge about truck steering tuning would have to be transferred to apply for another StW size. I.e. how steering assistance should be tuned as a function of StW size. Considered are both continuous properties, e.g. normal driving characteristics, and discontinuous, e.g. guiding pulse. Hereafter referred to as part 1 and 2 respectively. The work also intends to serve as a mapping when comparing results received from vehicles with different StW sizes.

This paper describes an experiment conducted on a test track to study:

Part 1 - Tuning of Continuous Characteristics: How continuous steering properties should depend on StW size to maintain a subjective balance between handling and comfort.

Part 2 - Discontinuous Column Torque: How discontinuous steering properties should depend on StW size to maintain the same angular driver response.

Similar questions could be stated for the properties steering gear ratio and wheelbase. These are not considered and therefore kept fixed.

In the next section of the paper the developed method for scaling is presented and the performed experiment is described. In Section 3 received results are presented and discussed. A conclusion is given in Section 4.

2 Method

In this section basics about steering characteristics are described and also how a scaling method was applied. The chosen pulse for discontinuous testing is then described. Finally used vehicle, track and experimental set up are shown.

2.1 Part 1: Continuous Steering Characteristics

Steering assistance systems together with vehicle geometries make up for the basic steering characteristic, which is of high importance for the experienced stability of the vehicle. Figure 1 show the basic characteristic properties at $v_x = 80$ km/h, high friction and low steering wheel angular rate for the truck used in the experiment. A hysteresis, such as seen in Figure 1, appears due to friction and damping in the steering system. On how to measure the characteristics, see e.g. (Salaani et al., 2004). The slope and level of hysteresis of this curve is of high importance to get a comfortable and directionally stable vehicle (Rothhämel, 2010). The term steering characteristics hereafter refer to the relation between StW torque and StW angle (with time derivatives). When changing steering wheel diameter

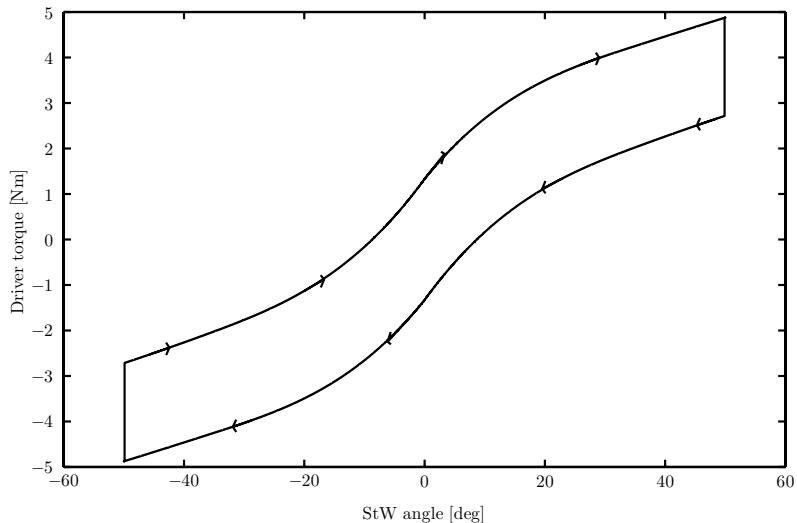


Figure 1 Baseline StW torque characteristics at speed $v_x = 80$ km/h for used truck.

the lever arm for the driver will change. One natural hypothesis would be that the drivers force level should be kept constant, when steering angular ratio is maintained. In other

words the required steering torque should be scaled linearly with steering wheel radius. In this experiment a truck was equipped with Volvo Dynamic Steering (Volvo-Trucks, 2013). This allows for great flexibility in shaping the required steering torque. The torque was changed with a gain factor according to

$$T_d(\theta, \dot{\theta}, v_x) = k_g \cdot T_{d,0}(\theta, \dot{\theta}, v_x) \quad (1)$$

where T_d denotes required driver torque, θ steering wheel angle, $\dot{\theta}$ steering wheel angular rate, v_x vehicle longitudinal speed, $T_{d,0}$ baseline steering torque characteristics (as in Figure 1) and k_g a scaling parameter. In Figure 2 characteristics for $k_g = 0.5$ and $k_g = 1$ are shown. Note that not only the slope but also the hysteresis (damping and friction) is scaled, i.e. the total torque is scaled with k_g at all times.

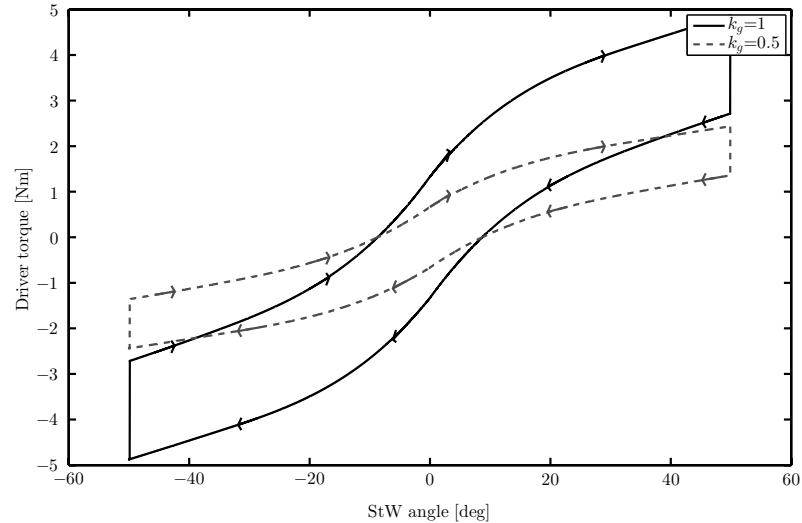


Figure 2 Steering wheel characteristic at speed $v_x = 80$ km/h with scaling parameter k_g .

If the hypothesis of maintained driver force would hold true the scaling value k_g should have a linear dependence on StW size, for each subject, i.e. each test driver. If on the other hand the StW torque was to be maintained k_g should not change when the StW size varies. On top of the here described torque characteristics it was possible to superimpose more torque. This was used in the second part, described in next section.

2.2 Part 2: Discontinuous Steering Characteristics

A discontinuous steering column torque input was produced by a sudden vertical offset to the characteristics in Figure 2. In the experiment a square pulse was used as discontinuous torque input. The pulse lasted for one second and was $-3 \cdot k_g$ Nm. This gave obvious impact on the driver but was still considered as safe on the used test-track. The continuous characteristics were also scaled for part 2 as k_g also effect relation (1). Again to scale complete torque applied on the StW. The rise time of the steering servo motor, from torque request to actual torque, was below 5 ms.

2.3 Experiment

To find out how force feedback should scale with StW size an experiment was set up where the previously described scaling method was used.

Truck Specification

A rigid Volvo FH16 truck was used. The axle arrangement was a single front axle, two rear axles whereof one driven. The rear most axle was lifted in all runs. Specifications are shown in Table 1. Three StWs were used, one Volvo FH 450 mm diameter (*large*), one Volvo V50 390 mm diameter (*medium*) and a Sparco R333 330 mm diameter (*small*). A quick release was installed to allow fast change of StW.

Table 1 Specification of Volvo FH16 truck used

Property	Value	Unit	Description
L	4.8	m	Wheelbase, distance between front and drive axle
$F_{z,f}$	63044	N	Front axle vertical load
$F_{z,d}$	72275	N	Drive axle vertical load
$F_{z,t}$	0	N	Tag axle vertical load (lifted)
i_s	23.2	-	Steering ratio, road wheel angle to StW angle
I_{StW}	[0.038, 0.019, 0.015]	kg m^2	StW inertia for <i>large</i> , <i>medium</i> and <i>small</i> StW including StW and column above power steering unit
r_{StW}	[0.225, 0.195, 0.165]	m	StW radius for <i>large</i> , <i>medium</i> and <i>small</i> StW measured from centre to rim edge

The truck was equipped with Volvo Dynamic Steering which is a torque overlay electric steering servo mounted on-top of a hydraulic steering gear. The servo was controlled with a dSpace MicroAutoBox to realize scaling as described in section 2.1. In this way the value k_g could be changed from a keyboard even at speed. More in detail, relation (1) was in fact implemented and verified fully. Including terms for aligning torque, damping, friction, steering wheel eccentricity and pulse. The only aspect not included in scaling was torque from StW inertia. The inertia of the StW together with steering column, I_{StW} , was assumed small enough to be neglected.

Track

A handling track having a lot of bends and one straight section was used. The top speed was limited to 90 km/h. The track was approximately 3100 m long and 6.5 m wide with two lanes. Traffic was unidirectional and other vehicles, driving in the same direction, occurred. The track was dry and some short sections had normal disturbances such as dips, small crests or were rutted. The maximum lateral acceleration reached by most drivers was around 3 m/s^2 . Drivers were advised not to drive slower than 45 km/h. One third of the track had an alternative route, with more bends. The subject were free to choose their preferred path, how they positioned and their speed as long as they stayed on road and within speed limits.

Subjects

Totally 17 drivers participated; all holding truck driving license. From these 10 were full time test drivers and 7 development engineers. Two were female. The average driver was 39 years old, got the truck driving license in 1996 and drove 60000 km yearly in a truck.

Set-up

Each person was initially informed about purpose of the activity. This was followed by a warm up lap to get to know the truck and the track. Then the actual test was run.

Each subject ran with all three StWs. The order of the StWs was randomized. For each StW subjective tuning of continuous characteristics was first run, then followed by objective recording of discontinuous response. Here described:

Part 1: Tuning of Continuous Characteristics. The scaling parameter k_g was used to adjust torque level when driving. The initial value of k_g was randomized between 0.5 and 1.5, the subjects were told that the initial value was a random number and the actual value of k_g was never shown to subjects. All of this was made to minimize cognitive bias (Fine, 2008), i.e. anticipation from subjects. The subjects were allowed to drive as many laps as needed to adjust the value of k_g to find their optimal trade-off between comfort and handling. The subjects were instructed to change the value of k_g by requesting either full, half or a quarter step up or down, where one full step corresponded to 0.1 in delta change of k_g . In this way the subjects were able to actively modify steering force level while driving and finally report their preferred level.

Part 2: Recording of Discontinuous Column Torque. The subjects were told to continue driving around the track, put both their hands on the StW and prepare for pulses to come (c.f. tensed mode in (Cole, 2012)). k_g was set in random order to value 1.0, 0.85 or 0.7. These levels roughly correspond to StW diameter in relation to the *large* StW diameter. Note that changing k_g effects both continuous characteristics and pulse size and that pulse was only given in one direction. An operator fired off pulses, unpredictable in time to subject, and all vehicle signals were recorded.

3 Results and Discussion

Here received results from the two parts are presented.

3.1 Part 1: Continuous Properties

The 17 subjects all decided on optimal scaling values of the driving torque characteristics. This was done for all the three StWs - resulting in 51 values of k_g , see Table 2. In Figure 3 the same data is shown but normalized with respect to the *large* StW k_g value per subject. The data is normalized since it is assumed that the preferred level is individual. Two lines are also included to visualise the hypothesis of maintained force and torque. This corresponds to slope 1 and 0 respectively.

By assuming normal distribution a paired t-test was used to show the likelihood of listed hypothesis, where three groups were used representing all StW sizes used. The hypothesis of maintained torque show $p < 0.1\%$ and is therefore rejected. The data is a lot more likely when assuming maintained force. Looking more in detail at the distribution of Figure 3 it can be seen that the number of data points are too few, as the variance is high, to guarantee

Table 2 Final value of k_g for all 17 subjects and StWs.

Subject ID	Large	Medium	Small
1	0.9	0.8	0.55
2	1	0.7	0.5
3	0.9	0.85	0.55
4	0.8	0.65	0.5
5	0.9	0.75	0.65
6	0.8	0.65	0.5
7	0.85	0.8	0.55
8	0.8	0.8	0.5
9	0.9	0.55	0.5
10	1	1	0.75
11	1.05	0.8	0.7
12	0.9	0.6	0.7
13	1.05	1	0.7
14	1.15	1.05	0.8
15	0.75	0.75	0.55
16	0.7	0.6	0.55
17	0.9	0.65	0.45

a linear relation. There are other non-linear models that would fit as good or better. The linear model of maintained force level is therefore only suggested as a rule of thumb for continuous steering assistance scaling. I.e. can be used as a first good guess.

During the trials some drivers complained about too slow StW free response return when k_g was low, which could suggest a hypothesis of maintained free response return rate. The steering wheel return acceleration is a consequence of aligning torque minus dissipative forces, i.e. what is actually scaled with k_g . The method for scaling total torque should therefore only be used as a rough rule of thumb. More precise tuning of e.g. damping would also be needed, in fact it can be shown that damping should scale as k_g^2 to maintain free response return rate. The method used for scaling, where complete torque was varied with only one parameter, could be extended to make more requirements fulfilled.

Figure 3 show a rather large variance amongst drivers. Some additional tests were carried out with some of the subjects after completion of the main section. By changing StW and again initializing k_g to a random number the question was if the driver would replicate his/her previous selection of k_g . Most of the subjects were not able to reproduce their previous value. In fact some drivers were obviously more sensitive than others. Some needed several laps to feel a difference when k_g changed and some felt it instantaneously. This showed that most of the variance in Figure 3 comes from resolution ability amongst subjects. The Just Noticeable Difference, JND, describe the minimum difference required between two stimuli before a human can notice the difference between them. For the arm joints the JND of sensing force is around 7% and 2 deg for angle (Tan et al., 1994). Assuming that the subjects compared steering stiffness, in terms of force per StW angle, at 40 deg the JND for steering stiffness becomes 8.6%, calculated with Taylor expansion of error propagation. This could explain a large part of the variance in Figure 3. Subjects were not able to discriminate between settings when k_g changed by less than the JND. Whether a driver perceive force or torque would here be irrelevant, since the relative JND would be the same. Furthermore whether a driver perceive e.g. lateral acceleration rather than StW angle

is left for future work. For this matter it could only add to the JND value, since sensing of force is here the dominant term. It should also be pointed out that the subjects had to keep the truck on the road, contributing with more uncertainty.

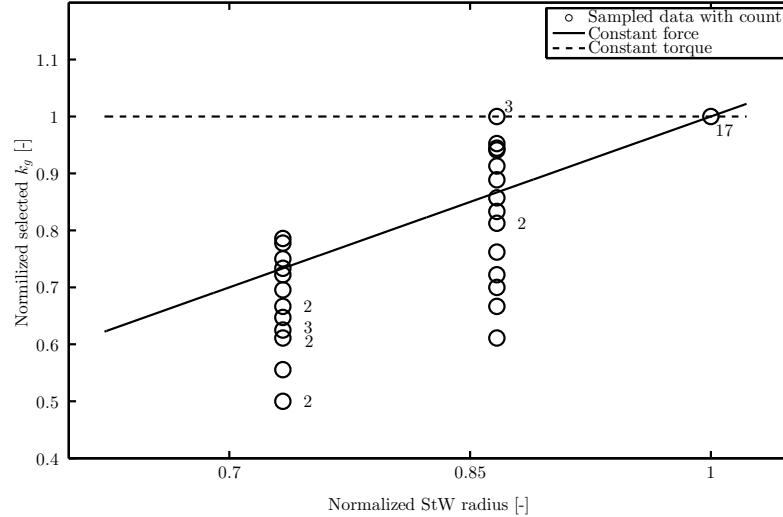


Figure 3 Result from test of steering assistance continuous properties. k_g values are divided with k_g from large StW per subject. r_{StW} is divided with r_{StW} from large StW. E.g. all data-points for the large StW will lie in point (1,1). Numbers are used to denote multiple occurrence of data-point. Also included is two lines, one with slope 1 and one with slope 0, these are constrained to run through point (1,1). The two lines correspond to constant force and constant torque respectively.

In the actual implementation of (1) it was assumed that torque from StW and column inertia could be neglected. A recording along the track is shown in Figure 4. It also includes an estimate of maximum error that is induced when neglecting scaling of inertia. The estimate is derived from StW angular acceleration times rotational inertia. As can be seen inertia is a lot smaller than other terms.

Used test procedure was set up to avoid bias in results from subject anticipation. Figure 5 show how initial value of k_g effected the final value selected by the subjects. As can be seen the correlation is very low, which suggests that anticipation from subjects is low. After the continuous tuning activity all drivers were exposed to several pulses, as described in Section 2.3, this is the topic of the next section.

3.2 Part 2: Discontinuous Properties

In total 1080 pulses were recorded. Given two conditions, that speed has to be above 50 km/h and that the magnitude of the StW angular rate has to be below 0.2 rad/s prior to pulse, 858 remain. Figure 6 show what effect that the pulses had on the StW angle 0.25 s after start of pulse. This was the point in time when the highest average deviation was seen. The shape of the distribution is not a normal distribution as seen. It is not symmetrical around the mean

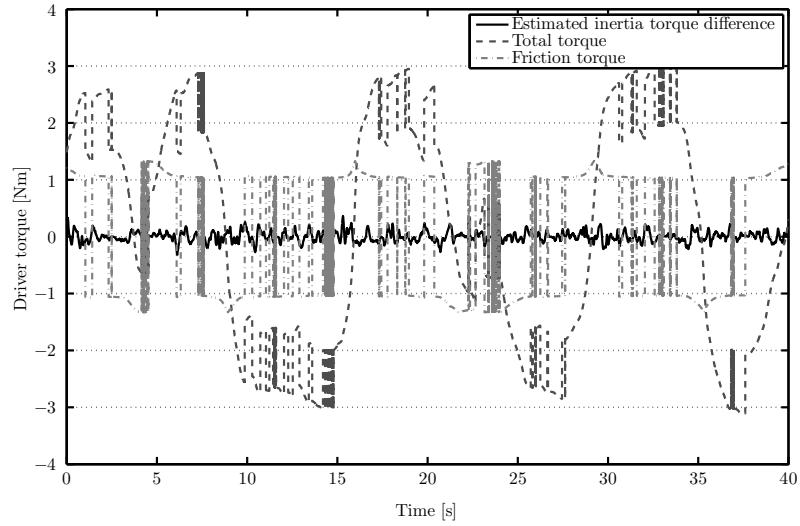


Figure 4 One measurement of total driver torque and friction along the track together with an estimate of error in torque arising when neglecting scaling of inertia.

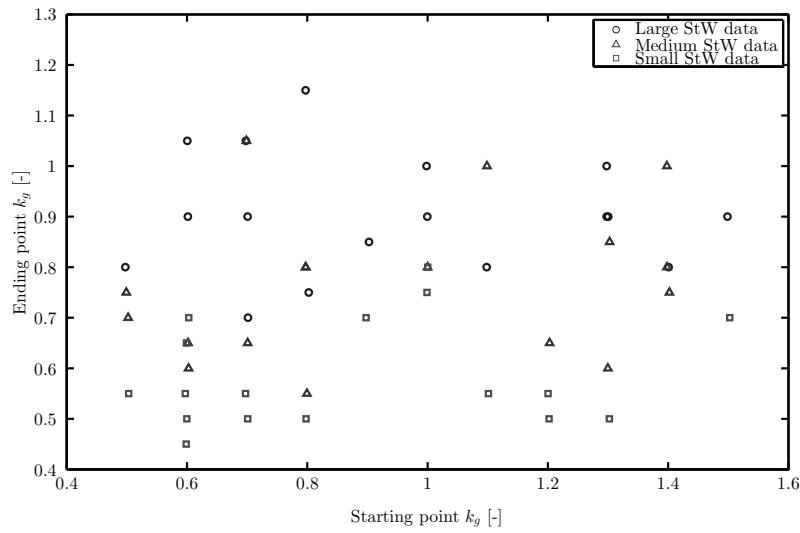


Figure 5 Final value of k_g shown together with the randomized initial value. The correlation is very low.

value and has a long tail on the left side. In total 88% responded in the same direction as the added torque. Possible reasons for the non-symmetrical shape are varying tension level of subjects, StW motion prior to the pulse and of course that there is different pulse sizes and StWs involved. However there are 12% of the pulses that even lead to a response opposite to the applied pulse. This indicate that the a priori uncertainty of a response is very high.

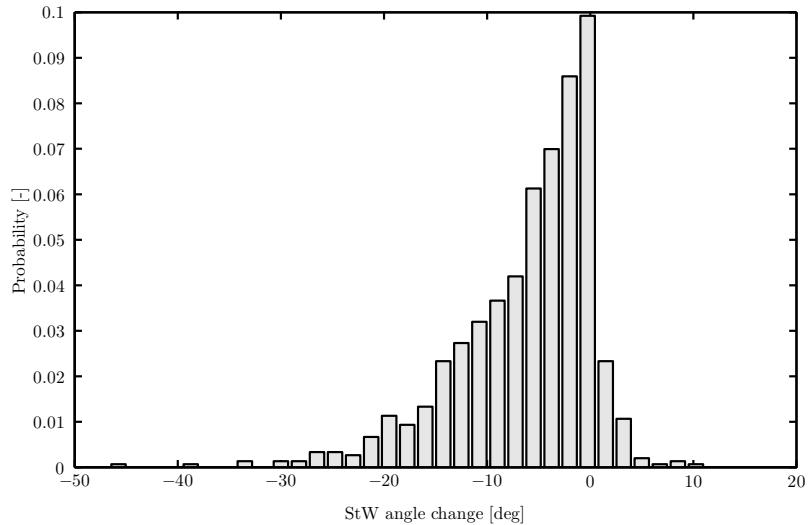


Figure 6 Sampled probability density function of all the 858 pulses. The variable shown is StW angle delta change 0.25 s from the start of the pulse.

The 858 pulses can be divided into nine classes, three StWs times three different values of k_g . Again mentioned, the pulse size was $-3 \cdot k_g$ Nm. For these histograms are shown in figure 7. Of special interest are the histograms on the diagonal, StW large $k_g = 1$, StW medium $k_g = 0.85$ and StW small $k_g = 0.7$. These are close to be tuned as suggested in the rough rule of thumb presented in Section 3.1 - namely scaling of driver torque linearly with StW size to maintain force. I.e. the continuous characteristics are tuned to maintain driver force along the diagonal.

Can it also be said that a pulse should be scaled linearly with StW size to preserve driver angular response? To answer this question two more figures should be considered, Figure 8 and 9. These show how the mean delta change move as time passes. In Figure 8 the pulse is equal for all the StWs. In Figure 9 the pulse is scaled with StW size. The mean delta change is almost the same for all StWs, in the latter case, which is not so in the former case. This supports the hypothesis to scale discontinuous steering torque with StW size. The uncertainty is however big as seen from pooled standard deviation which is also included in both figures.

The inertia of the three StWs was not the same. Figure 10 show that the difference in torque, for the driver, cannot be fully neglected since the pulse itself is of order 3 Nm and the error up to 0.5 Nm. However inertia conserves energy and the torque absorbed initially will be released when the motion is decelerated. This would result in a delay of the peak

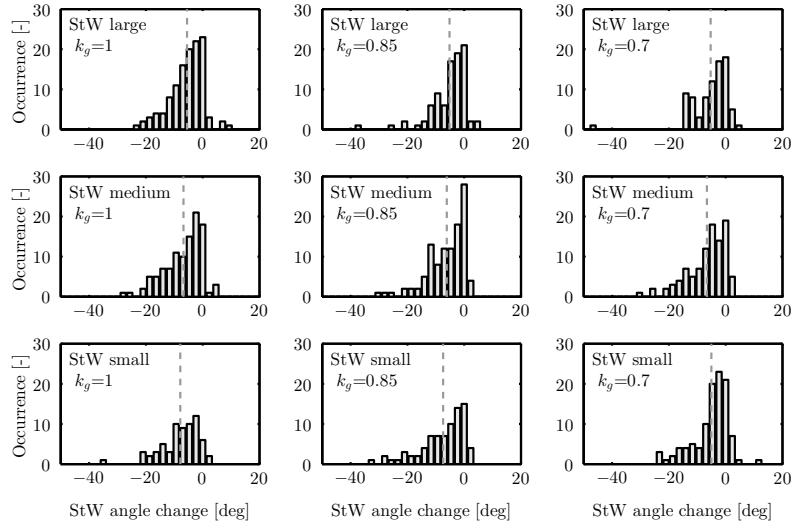


Figure 7 Histogram of StW angle delta change 0.25 s from the start of the pulse. Mean value of samples is shown with a vertical line. The y-axis show number of times that the delta change was observed.

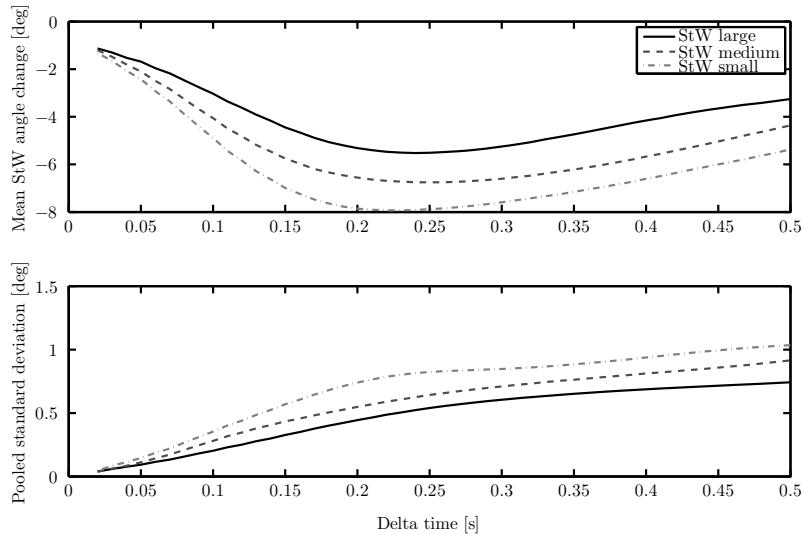


Figure 8 Mean StW angle delta change from start of pulse when $k_g = 1$. Also pooled standard deviation is seen to grow as time moves.

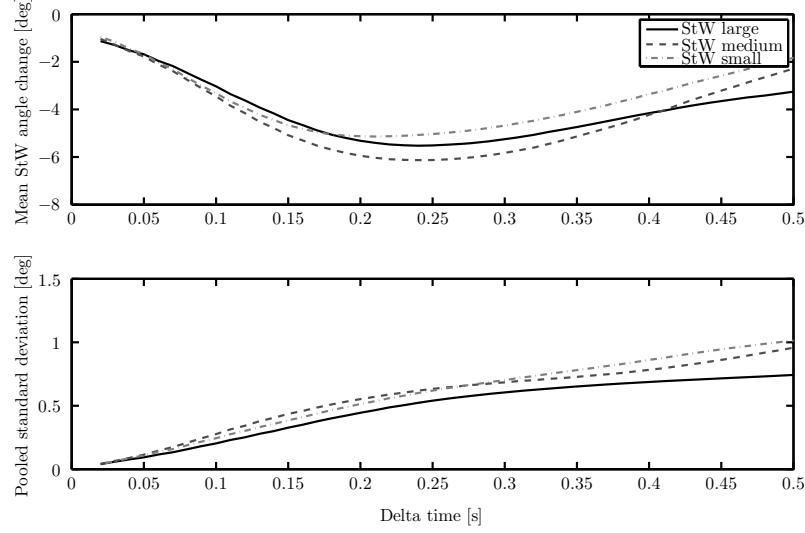


Figure 9 Mean StW angle delta change from start of pulse. For StW large $k_g = 1$, medium $k_g = 0.85$, small $k_g = 0.7$. Also pooled standard deviation is seen to grow as time moves.

value in Figure 8 when the StW has higher inertia. Only small such tendencies can be seen. The error induced from inertia is therefore assumed to be of secondary effect.

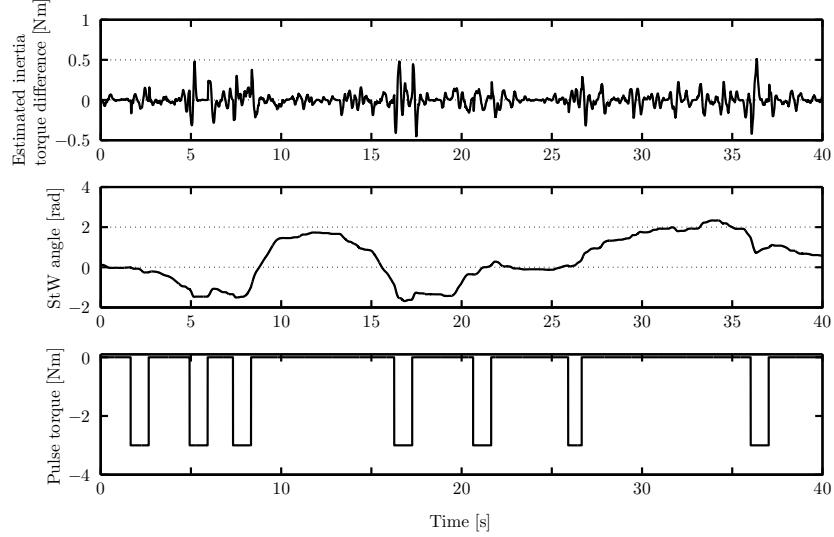


Figure 10 Maximum difference in torque from inertia, StW angle and pulse torque over time from a recording.

4 Conclusions

An experiment has been set up, on a test-track, to show how continuous steering torque should depend on StW size; this to maintain a subjective balance between handling and comfort. A method was developed and implemented to scale complete torque felt by the driver at speed in a truck. 17 subjects tuned torque level for three different StWs, of different size. Initial scaling was random and not shown to the subjects. Also the order of the StWs was varied to avoid bias from anticipation. Results indicate that driver torque feedback should be scaled when StW size changes. A rule of thumb is to use linear scaling of total torque to accomplish maintained driver force level, which is in accordance with (Newberry et al., 2007). Further adjustment of damping, friction etc. might be needed to realise conservation of steering wheel free response return rate. The subjects had some problems to be consistent in their subjective judgement. The human resolution was identified as the main reason for this. Further studies on tactile resolution of steering forces are therefore suggested.

Also analysed was how discontinuous steering properties should depend on StW size to maintain the same angular driver response. I.e. how a torque pulse should scale with StW size to have the same effect on steering wheel angle, thereby vehicle lateral motion. The same rule of thumb as for continuous properties was shown to work, namely maintain force feedback level. The StW angle delta change response was seen to vary a lot. The distribution was not symmetrical and did thus not follow a normal distribution. In (Cole, 2012; Pick and Cole, 2007) the cases tensed and relaxed arms differ widely in response. The large variance here observed could therefore be a consequence of tension level. Other possible factors are driver arm inertia, strength and if the driver is entering/leaving a turn. By monitoring driver state it could be possible to reduce the uncertainty in repose (Abbink et al., 2011). This would be useful when designing steering support functionality and therefore recommended for future studies.

The study was conducted with fixed wheelbase, fixed understeer properties and fixed steering gear ratio. These are other possible reasons for having different power assistance level when varied.

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Paper 2

Title: Driver response to automatic braking under split friction conditions

Publisher: JSAE

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Version: Accepted article

To cite this article: K. Tagesson, B. Jacobson, and L. Laine, “Driver response to automatic braking under split friction conditions,” in *The 12th International Symposium on Advanced Vehicle Control*, 2014, pp. 666-671.

Notation conversions: $X = X_E$; $Y = Y_E$

Driver Response to Automatic Braking under Split Friction Conditions

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At normal pedal braking on split- μ a driver can actively steer or adjust brake level to control lateral drift. The same driver response and thus lateral deviation cannot be assumed when brakes are automatically triggered by a collision mitigation system, since the driver can be expected as less attentive. To quantify lateral deviation in this scenario a test was run at 50 km/h with 12 unaware drivers in a heavy truck. Brakes were configured to emulate automatic braking on split- μ . Results show that the produced maximum lateral deviation from the original direction was 0.25 m on average. Two drivers deviated by 0.5 m. This can be compared to 2.2 m which was reached when steering was held fixed.

Topics/Active safety and driver assistance systems, Driver modelling

1. INTRODUCTION

A road section with significantly different level of friction between left and right vehicle sides is said to have split friction or split- μ . Common reasons for split- μ are: oil spillage, uneven ice coating, and one-sided aquaplaning. When cruising or accelerating slowly the driver may not even notice the effect, but when braking hard in an emergency situation the effect from unbalanced braking forces may cause serious rotation of the vehicle towards the side of high friction. For trucks towing one or more trailer this can also lead to jack-knife [1].

At the event of modest rotation the driver can steer and balance uneven braking forces. However if the driver is surprised by the situation and thus unprepared it is likely that substantial lateral deviation from ego lane can occur before the driver has responded. This can result in run off road or collision with oncoming traffic. Fig. 1 provides an example of this where a truck ends up in the opposite lane.

Furthermore a hypothesis is that when braking is activated by an advanced emergency braking system, AEBS, automatically the surprise would become even bigger. And thus also produce bigger lateral deviation and higher risk of jack-knifing. One can also note the similarity to front tyre blow-outs that yearly leads to some fatal accidents e.g. see [2]. In [3] a truck simulator study was performed where the left front tyre exploded. It was observed that driver behaviour very much

depended on if the blow out came as a surprise or not. Hence it is important to include the effect from surprise also when investigating automatic braking under split friction conditions.

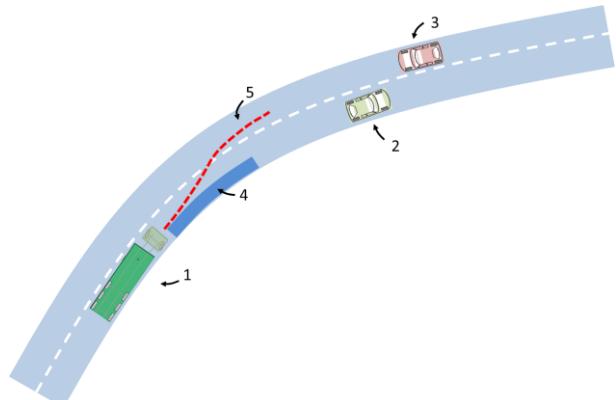


Fig. 1 An example with a truck (1) that brakes because of a stationary car (2). A patch of one-sided low friction (4) causes the truck to yaw and move sideways (5). When adding an oncoming car (3) into the scene an accident is imminent.

The split friction braking scenario has been a well-known hazard for decades and many innovations have been presented to reduce the effects [4]. Some have been proven more effective than others. E.g. brake pressure limiting approaches are already used on many vehicles, but have the effect of reducing brake

performance. There are also legal requirements for split- μ braking that limits the allowed lateral deviation, under certain conditions [5]. The aim of this paper is to study if these legal requirements together with commonly used functionality are enough, on split- μ , when AEBS is introduced and soon mandatory on heavy trucks in Europe [6]. Or if there is a need of further supporting the driver.

In order to understand the severity in automatic braking under split friction conditions it is important to know how a driver reacts. For this reason a test was set up exposing drivers to a rather sudden situation, where the truck pulls sideways during automatic braking.

In section 2 the arrangement of the experiment is described, results follow in section 3 and finally section 4 present some conclusions. Notations and properties used, especially sign conventions, are compliant with ISO 8855 [8].

2. METHOD

The test was run with a 9 ton solo tractor on a test track where 12 drivers were exposed to sudden automatic braking. Research results were obtained through informed consent. Brakes were controlled to emulate split- μ conditions on an even test-track. The drivers were not aware of the true purpose of the test in order to preserve the effect from surprise. The test was carefully designed to guarantee safety.

Only professional drivers, normally driving durability tests of trucks, took part. The average age was 42, the oldest driver was 60 and the youngest 27. Only one driver had experience from pure brake or handling tests. Drivers were told that the intention of the test was to record normal positioning in lane and that they should run back and forth inside a straight lane for 300 m. Cruise control was set to 50 km/h. After running back and forth for 5 minutes, without any intervention, an operator fired of automatic braking as described. After the first unexpected intervention two repeated runs were made at the same speed, followed by two more at 70 km/h.

2.1 Vehicle and Track

A 6x2 Volvo FH pusher tractor was used in the experiment having the pusher axle lifted. Since the test was set up for the first time it was run without any trailers to ensure safety. The same goes with the selection of speed. It was low initially to guarantee safety. For more details on the vehicle used see Table 1.

The test was run on a test track in Sweden during two days in December. Temperature was 3-8°C. The track was slightly wet, but it did not rain. No other vehicle was nearby; therefore cones were put in the adjacent lanes creating a sense of danger. Fig. 2 provides an illustration of the set-up.

Table 1 Vehicle System Parameters

Description	Value	Unit
Wheelbase, distance between front and drive axle	4.1	m
Vehicle width, to outer wheel side	2.25	m
Front axle vertical load	58470	N
Pusher axle vertical load (lifted)	0	N
Drive axle vertical load	29430	N
Overall steering ratio	23.2	-
Steering wheel radius, measured from centre to rim edge	0.225	m
Wheel effective radius	0.5	m

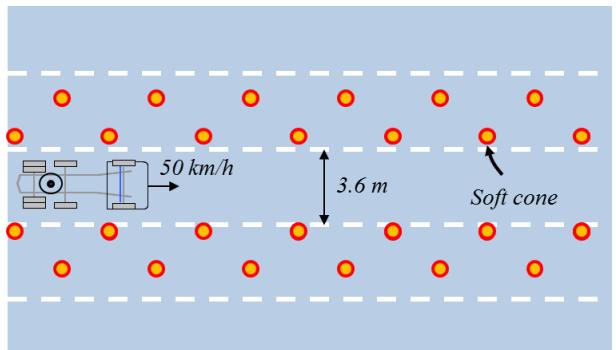


Fig. 2 The test was run at 50 km/h using cruise control. Soft cones were used to create a sense of danger.

2.2 Brake Controller

In [6] requirements implicitly say that an AEBS system shall be capable of performing deceleration by at least 2.2 m/s^2 during the emergency braking phase. Also stated is that "the AEBS shall be designed to minimise the generation of collision warning signals and to avoid autonomous braking in situations where the driver would not recognise an impending forward collision", i.e. nuisance should be avoided. In practise this means that the automatic emergency braking phase would have to be triggered closer to the imminent collision. In [9] normal braking behaviour of drivers in cars is analysed. At 80 km/h and time to collision, TTC, at 2.7 s it is a 75% chance that a driver would treat the required brake action as hard, in order not to collide with a moving target vehicle. In [10] a study on truck driver deceleration behaviour was made. At 80 km/h it was observed that normal braking does occur as late as TTC 3.9 s. This is based on 10000 normal brake interventions from euroFOT data. With the combined findings in [9] and [10] and the requirement regarding nuisance in [6] an AEBS should at least be capable of decelerating at 3.5 m/s^2 during the emergency braking phase. Here assuming that brake initiation is delayed until TTC 4.0 s and that the brake system has a delay of 0.2 s from brake request until full deceleration is reached (this delay was verified on used truck). Therefore as target deceleration 3.5 m/s^2 was used in a brake controller.

The controller consisted of a feedforward and a

feedback part. The feedforward part was constant and the feedback part was a PI controller with integrator saturation and delayed initiation.

The sum of the feedforward part and the PI controller was fed into a static allocation function. The proportion between left and right brake torque was fix and set to 4. This value was derived from real split- μ testing, using normal factory brake system settings. In this mode the tractor was compliant with [5] since it had a brake pressure limiting function setting allowed difference between left and right brake pressure. The relation between front and rear brake pressure was set according to static normal loads. A linear relation was assumed between brake pressure and brake force. In Fig. 3 a block diagram of the controller is shown. All properties included are listed in Table 2.

In case the driver pressed the brake pedal a select high pressure routine was used per wheel. If the driver pressed the accelerator pedal the test was aborted.

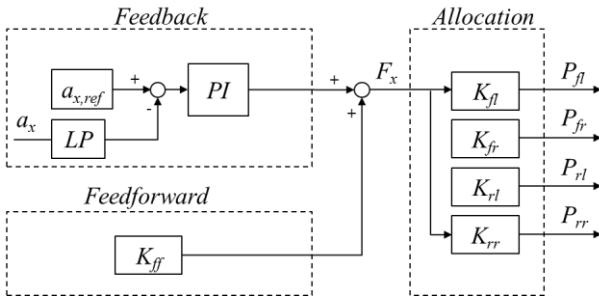


Fig. 3 Block diagram of brake controller

3. RESULTS

All 12 trajectories relating to the very first exposure of automatic braking interventions are shown in Fig. 4. All lines have been rotated and moved so that (0, 0) m correspond to where the operator activated the automatic braking, i.e. brake onset. The stopping distance, counting from brake onset, ranges from 30.2 m to 33.3 m. Two drivers instinctively deactivated the intervention by pressing the accelerator pedal. This corresponds to the two trajectories that continue to travel even after 35 m. The mean maximum lateral deviation was 0.25 ± 0.07 m, using 95% confidence level. Two drivers deviated by 0.5 m. The open loop response produced by locking the steering wheel, StW, is also shown. It deviates by 2.2 m at standstill.

Corresponding time series are shown in Fig. 5. Looking at the speed curves it can again be seen that two drivers instinctively deactivated the intervention by pressing the accelerator pedal. The second subfigure shows StW angle relative change, calculated as:

$$\Delta\delta_H(t) = \delta_H(t) - \delta_H(0) \quad (1)$$

where $\delta_H(t)$ is StW angle at time t . After 0.6 s, on average, drivers started steering. Some drivers

responded with a smooth and steady movement of the steering wheel, whereas others oscillated widely.

The positive scrub radius, which acts destabilizing, can be observed in the StW torque plot. Around -2.5 Nm of the disturbance reached the driver.

As seen in the last subfigure yaw rate starts building up after 0.3 s and also the response shows a one period sine wave. Corresponding frequency, 0.5 Hz, happens to match the resonance frequency of several truck combination types, see [7]. This highlights the importance of extending the study for multi-unit truck combinations.

Table 2 Brake Control Parameters

	Notation	Value	Unit
Feedback			
Measured longitudinal acceleration	a_x	-	m/s^2
First order low pass filter with time constant 0.2 s	LP	-	-
Target longitudinal acceleration	$a_{x,\text{ref}}$	-3.5	m/s^2
PI saturation		6000	N
P-gain		4000	N/m/s^2
I-gain		20000	N/m/s
Integrator saturation		2000	N
PI activation time		1.5	s
PI error linear ramp up duration		0.5	s
Feedforward			
Braking force	K_{ff}	-18000	N
Allocation			
Total longitudinal force	F_x	-	N
Allocation constant	K_{fL}	-2.04×10^{-4}	Bar/N
Allocation constant	K_{fR}	-5.10×10^{-5}	Bar/N
Allocation constant	K_{rL}	-6.79×10^{-5}	Bar/N
Allocation constant	K_{rR}	-1.70×10^{-5}	Bar/N
Brake pressure (front/rear left/right)	$P_{fL/fR/rL/rR}$	-	Bar

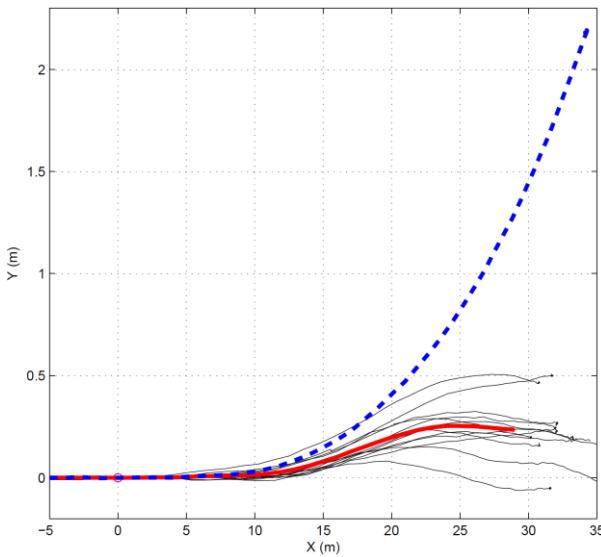


Fig. 4 Position of tractor rear axle during unexpected brake intervention, starting at $(0, 0)$ m. One black solid curve per driver, thick solid red is average of all drivers, dashed thick blue is reference run with fixed steering. Lines have been moved and rotated to get zero offset and heading at beginning.

Continuing with the repeated runs, Fig. 6 and 7 give all trajectories and time series from the repeated runs. Four runs have been filtered out since the drivers pressed the accelerator pedal early on during the exposure. The average maximum lateral deviation observed is 0.13 ± 0.03 m, again using 95% confidence level. There seems to be a reduction in lateral deviation, as drivers become aware of the true purpose of the test. To investigate this further a paired t-test was performed on the StW response. Fig. 8 show average change in StW angle from both the initial runs and the repeated runs. Also shown is the average difference between these runs per driver. Since each driver conducted only one initial run and two repeated runs the comparison is made between the initial run and the average of the repeated runs. In the four cases where runs are excluded only one repeated run is used. The difference between the lines indicate that there is a difference in reaction time between the runs. This is also clearly confirmed with the t-test which show a significant difference after 0.5 s, run with 95% confidence level and 11 degrees of freedom. Looking at the time where the on average -5 deg is passed the difference is about 0.1 s.

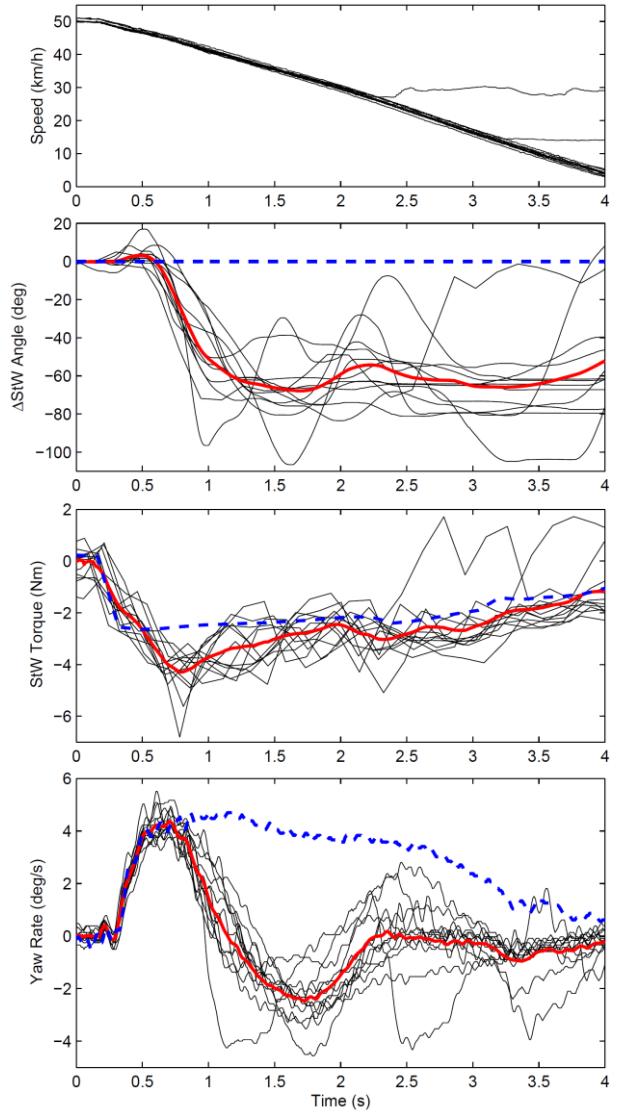


Fig. 5 Response to unexpected brake intervention, starting at time 0 s. Line styles same as in Fig. 4.

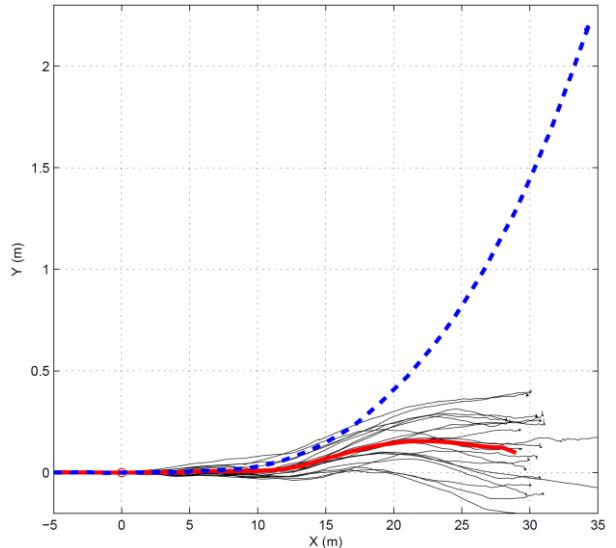


Fig. 6 Position of tractor rear axle during repeated brake intervention. For line styles see Fig. 4.

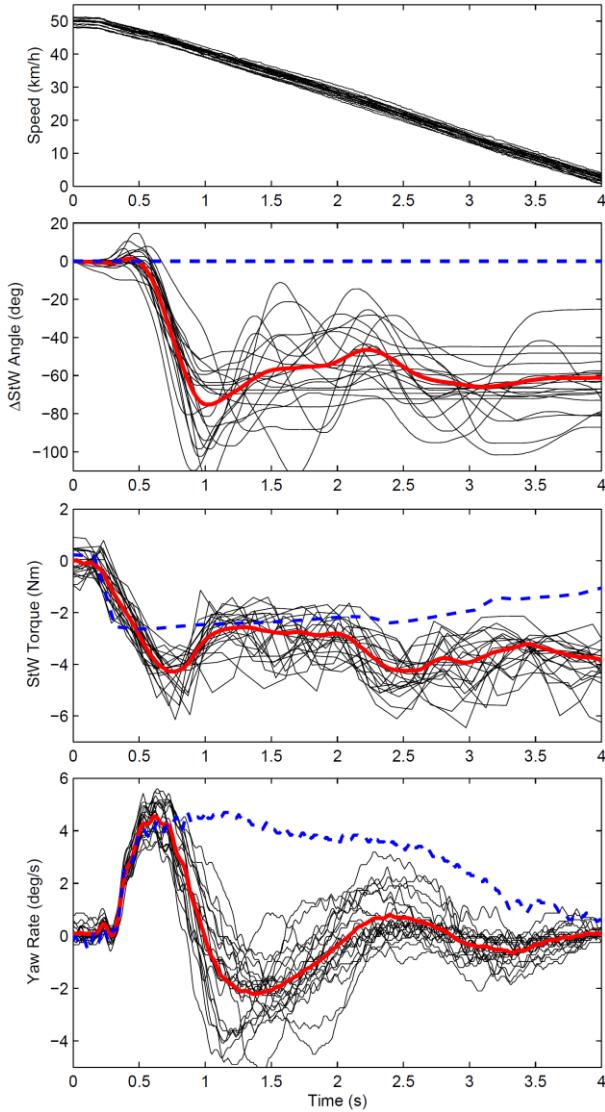


Fig. 7 Response to repeated brake intervention, starting at time 0 s. Line styles same as in Fig. 4.

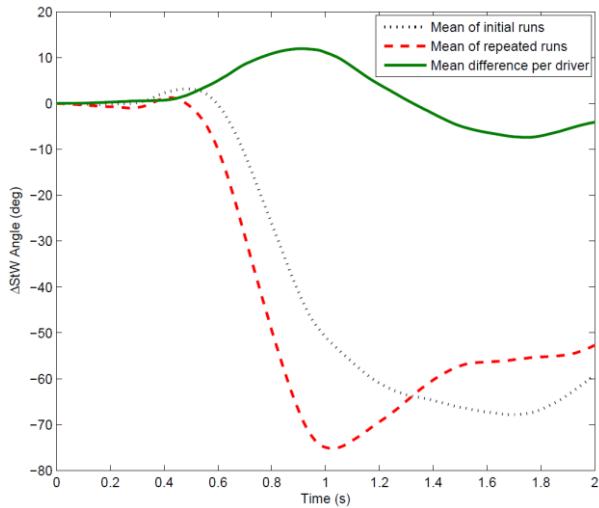


Fig. 8 Mean value of change in StW angle from the first initial runs and from the repeated runs. Also shown is the average difference between the initial run and the repeated runs.

For the repeated runs done at 70 km/h the lateral deviation observed was at a similar level as in the repeated runs done at 50 km/h. The average maximum lateral deviation observed was 0.10 ± 0.04 m, using 95% confidence level. The reaction time, before reaching a StW angle of -5 deg, was again lowered.

4. CONCLUSION

The combination of split- μ and automatic brake intervention has been tested in a truck with 12 unaware drivers. Even though drivers were unaware of brake intervention they were still all aware of being part of a study and consequently more observant than normal. The lateral deviation observed was higher in the first runs, when drivers were unaware, compared to repeated runs. An identified reason for this was shorter reaction time. Measured levels suggest that the risk of collision, due to lateral deviation, is low for an alert driver. For a distracted driver more support might be required. This was motivated by the runs where steering was held locked. As was obvious from the repeated runs a driver which knows what will come is more effective in reducing lateral deviation. This underlines the fact that the warning phase, which is already an important part of AEBS, should not be underestimated.

Also beneficial would be a low value, or even negative value, of scrub radius since this limits the destabilising StW torque which has to be taken care of by the driver. Using even more sophisticated approaches for StW torque, like overlay torque guidance, might even improve the results further. This is however not obvious.

Angle overlay, steer-by-wire or rear axle steer systems are other ways of reducing lateral deviation even further, e.g. see [4]. These also have the potential of reducing the stopping distance, when combined with brake controls. Stopping distance is obviously important when AEBS has activated due to risk of collision.

No major difference in lateral deviation was observed when increasing speed from 50 km/h to 70 km/h. Drivers did however show shorter reaction time in this case, most likely since it was the last part of the test.

Finally said, the yaw rate frequency matched, in most runs, the resonance frequency of several truck combinations. It is however not clear whether drivers respond in the same way when having trailers connected. Therefore additional tests, including trailers running at a higher speed, are of importance to completely be able to investigate the combination of automatic braking on split friction.

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Paper 3

Title: Driver response at tyre blow-out in heavy vehicles & the importance of scrub radius

Publisher: IEEE

DOI:10.1109/IVS.2014.6856544

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Version: Accepted article

To cite this article: K. Tagesson, B. Jacobson, and L. Laine, “Driver response at tyre blow-out in heavy vehicles & the importance of scrub radius,” in *The 2014 IEEE Intelligent Vehicle Symposium*, 2014, pp. 1157-1162.

To link to this article: <http://dx.doi.org/10.1109/IVS.2014.6856544>

Notation conversions: $X = X_E$; $Y = Y_E$

Driver Response at Tyre Blow-Out in Heavy Vehicles & The Importance of Scrub Radius*

Kristoffer Tagesson¹, Bengt Jacobson² and Leo Laine¹

Abstract—Front tyre blow-outs lead to several fatal accidents involving heavy vehicles. Common for most heavy vehicles is a positive scrub radius. This can result in a destabilising steering wheel torque at front tyre blow-out. In this study the safety improvement achieved when reducing scrub radius is quantified. By using a heavy truck equipped with a modified electric power steering system it was possible to change the scrub radius virtually. Brakes were configured to emulate front tyre blow-out which appeared as a sudden disturbance on one of the front tyres. In total 20 drivers took part in the study which was run on a test track at 50 km/h. Results show that the produced average lateral deviation from the original direction was 23 cm, when scrub radius was 12 cm, compared to 16 cm, when scrub radius was 0 cm. The main cause of the observed difference was a small, yet significant, initial overshoot in steering wheel angle which can be derived from the destabilising steering wheel torque.

I. INTRODUCTION

Tyre failures are involved in many fatal accidents every year. In [1] it was found that damaged tyres was the second most common vehicle defect reported at fatal accidents between 1995 and 1997 in USA. Blow-outs occurred in 0.35% of all fatal truck crashes. In particular front tyre blow-outs seemed more critical than blow-outs on other axles. In [2] another sample was taken from a French motorway network of 2000 km, during the period from 1996 to 2002. It showed that 3.5% of all trucks involved in accidents, with property damage or injury, were reported with blown out tyres. A higher criticality of front axle blow-outs was again confirmed.

Tyre blow-outs frequency can be reduced using correct tyre pressure and thereby avoid overheating. Overloading and excessive wear should also be avoided. Tyre pressure and loading monitoring systems have therefore been suggested and are already in use on many vehicles [1], [2]. A road hazard is another cause of tyre blow-outs. This problem is not removed by previously suggested countermeasures. In summary, it can be expected that the total number of tyre blow-outs on the roads will decline, yet a considerable number will remain. It is therefore of high importance to

support drivers by designing vehicles that are insensitive to tyre blow-outs.

There are three different reasons for vehicle instability at front tyre blow-outs. Firstly, vehicle yaw torque is induced. A damaged tyre produces a lot higher rolling resistance than a normal tyre. At worst it even stops rolling and instead develops full slip, similar to a locked up tyre. Since this force is offset from vehicle centre a resulting torque around centre of gravity will be acting. Secondly, common for most heavy vehicles is a positive scrub radius, which is a consequence of wheel and axle geometry. This can result in a destabilising steering wheel, StW, torque during front tyre blow-out. A deflated tyre has a smaller radius than a normal tyre. This creates even higher scrub radius and consequently also higher StW torque [3]. Thirdly, a vehicle towing one or more trailers will experience forces in the connection point. E.g. if the towing vehicle is slowed down because of a blown out tyre a heavy towed trailer will create high forces in the connection point. If an angle has developed, between the units, this force will act destabilising on the towing vehicle. Combined, these effects can result in run of road, collision with oncoming vehicles, roll-over or jack-knife, unless the driver is able to balance the effects by steering or braking. When designing vehicles it is therefore important to know how a driver reacts at a tyre blow-out. More precisely put, it is important to understand driver behaviour as a function of all the three above mentioned instability factors. In this work we focus on driver behaviour in a heavy vehicle and try to distinguish between vehicle yaw torque and StW torque.

Many have studied and modelled the motion of cars and trucks at tyre blow-out, e.g. [4] and [5]. Few have studied the variety in behaviour among different drivers at tyre blow-out. One exception is [6] where a truck simulator study was run. It was observed that driver behaviour was very much dependent on the effect of surprise. The lateral deviation on the first blow-out was a lot higher than on the following trials. The level of StW torque induced was however not varied.

To the best of the authors' knowledge no one has yet deeply analysed the influence of StW torque during tyre blow-out. In particular not for heavy vehicles where dynamics and steering geometries, e.g. scrub radius, are different than for cars. This will be the scope of this paper.

The outline of the paper is as follows: In section II the performed test track experiment is described and the corresponding results in section III. Finally some conclusions are given in section IV. Sign conventions used for vehicle quantities complies with ISO definitions, see [7].

*This work was supported by Volvo Trucks and Vinnova

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II. METHOD

A test was set up with a 9 ton solo semi-trailer truck, commonly known as tractor unit, on a test track where 20 drivers were exposed to several repetitions of emulated front tyre blow-out. Research results were obtained through informed consent. The test was part of a larger program, e.g. see [8]. Drivers were not aware of the intention of the test, but had been exposed to three similar interventions prior to the blow-out runs, all pulling the vehicle left.

A. Test Track

The test was run on a closed test track in Sweden during two days in December. Temperature was 3-8°C. The track was slightly wet, but it did not rain. For safety reason a 300 m long and 3.6 m wide straight marked lane on a large brake and handling area was used. This provided sufficient safety margins. To make drivers avoid crossing lane markings soft cones were put in the adjacent lanes. The set-up is illustrated in Fig. 1.

B. Test Vehicle

A solo 6×2 pusher tractor was used in the experiment having the pusher axle lifted. Brakes were controlled to emulate tyre blow-out. This was performed by applying 350 kPa of brake pressure on one of the front tyres. This level was selected just below tyre locking. The produced tyre force was thereby nearly maximised, but discontinuities relating to ABS control was eliminated. The relatively high level was selected to produce worst case blow-out forces, which is still not far above what has been measured, e.g. see [4]. In the case that the driver pressed the brake pedal a select high pressure routine was used. If the driver pressed the accelerator pedal the test was aborted. Tyre dimensions were selected on purpose to get high scrub radius. This resulted in 12 cm which in the default set up produced around 3 Nm of torque on the StW. For more details on the vehicle used see Table I.

The vehicle was also equipped with Volvo Dynamic Steering, which is an electric power steering unit. The system contains the ability to fully suppress steering torque disturbances coming from tyre road interaction, analogous

TABLE I
SPECIFICATION OF VOLVO FH TRACTOR USED

Property	Value	Unit	Description
L	4.1	m	Wheelbase, distance between front and drive axle
$F_{z,f}$	58470	N	Front axle vertical load
$F_{z,p}$	0	N	Pusher axle vertical load (lifted)
$F_{z,d}$	29430	N	Drive axle vertical load
i_s	23.2	-	Steering ratio, road wheel angle to StW angle
r_{StW}	0.225	m	StW radius, measured from centre to rim edge

to 0 cm of scrub radius in the blow-out case. The system was made configurable also to function as a conventional power steering system, however preserving the normal torque characteristics, which then is analogous to 12 cm of scrub radius. I.e. the two modes will behave the same during normal driving, but deviate when blow-out occurs. By changing mode in-between runs all drivers were exposed to tyre blow-outs both with 0 cm and 12 cm of scrub radius.

The on-board truck sensors were recorded during the whole test. That includes e.g. yaw rate, lateral acceleration, StW angle, StW torque, wheel speeds, brake pressure, accelerator pedal position and brake pedal position. A high precision GPS, placed above the drive axle, was also used and recorded.

C. Test Drivers

In total 20 professional drivers took part, normally driving durability tests of trucks. Only one driver had experience from brake or handling tests. The average age was 43.5, the oldest participant was 63 and the youngest 27. There were 17 male and 3 female.

D. Test Procedure

Drivers were told that the intention of the test was to record normal positioning in lane and that they should run back and forth inside the straight lane for 300 m. Cruise control was set to 50 km/h. An operator fired off emulated tyre blow-outs on the front left wheel, as described, at random locations. At the same time cruise control was deactivated.

Each driver was exposed to three blow-outs per scrub radius. The order of the exposures was reversed for every new driver to avoid bias from learning. For some drivers an additional blow-out on the front right wheel was fired off.

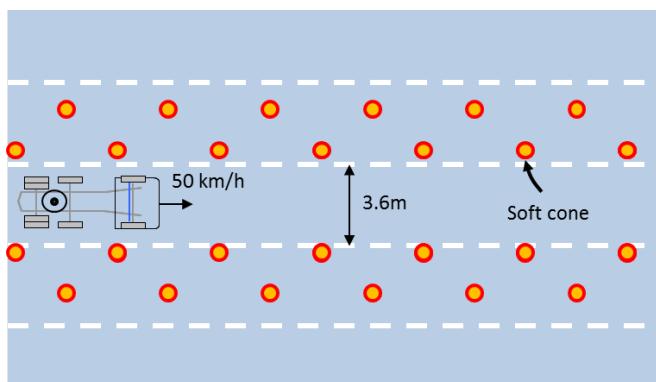


Fig. 1. Sketch of track set-up. Soft cones were used to create a sense of danger in the adjacent lanes.

III. RESULTS AND ANALYSIS

All trials have been checked with respect to; initial speed range 50 ± 2 km/h, correct brake pressure, that the driver did not press the accelerator pedal, and that the brake pedal was not pressed hard. After this 103 front left blow-outs remain, where 51 are run with scrub radius 12 cm and 53 are run with scrub radius 0 cm. In this series all drivers are represented in at least one run per scrub radius setting. Additionally, 15 front right blow-outs are also kept.

A. Left Blow-Out Path and Time Series

Fig. 2 show all trajectories produced for front left blow-out runs. Black colour is used for runs with 12 cm scrub radius. Red colour is used for runs with 0 cm scrub radius. Bold lines are used for average. The produced average lateral deviation from the original direction is 23 cm, when scrub radius is 12 cm, compared to 16 cm on average, when scrub radius is 0 cm. There is however large variance in data, so a direct comparison will not prove a significant difference. Some drivers deviated left by more than 50 cm.

Fig. 3 show time series of speed, StW angle, StW torque and yaw rate for all front left blow-out runs. Colouring used is the same as in Fig. 2. The speed profiles are as expected similar for all runs apart for some where the driver has pressed the brake pedal gently. The StW angle curves initially indicate that some drivers, exposed to a destabilising, StW torque turn left before they turn right. Furthermore during the first second the steering profile is rather consistent. After that, very different profiles appear. The StW torque curves show an apparent difference between the two settings used.

Continuing on analysing Fig. 3, it can be seen that the yaw rate response roughly show a one period sine wave. Corresponding frequency, 0.7 Hz, happens to match the resonance frequency of several truck combination types, see [9]. This highlights the importance of extending the study for multi-unit truck combinations.

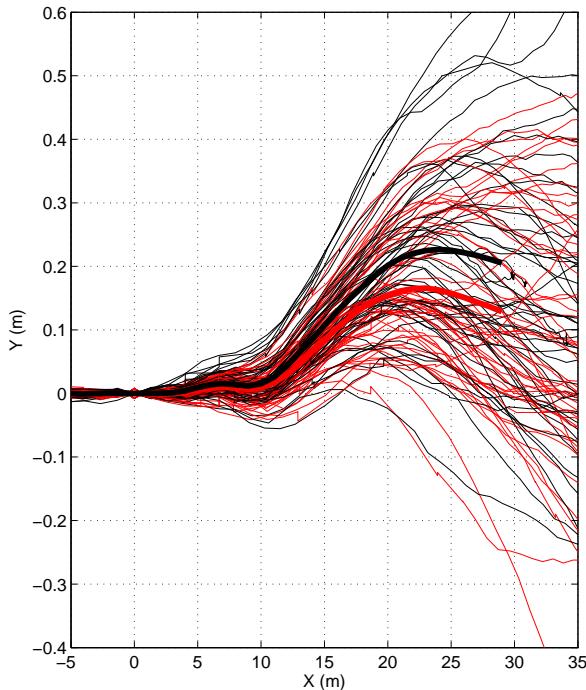


Fig. 2. Trajectories of centre of drive axle for all emulated front left blow-out runs. The curves have been rotated and moved so that blow-out is initiated at position (0,0) m running at zero heading. Thin red lines correspond to scrub radius 0 cm. Thin black lines correspond to scrub radius 12 cm. Bold red line correspond to average of scrub radius 0 cm runs. Bold black line correspond to average of scrub radius 12 cm runs.

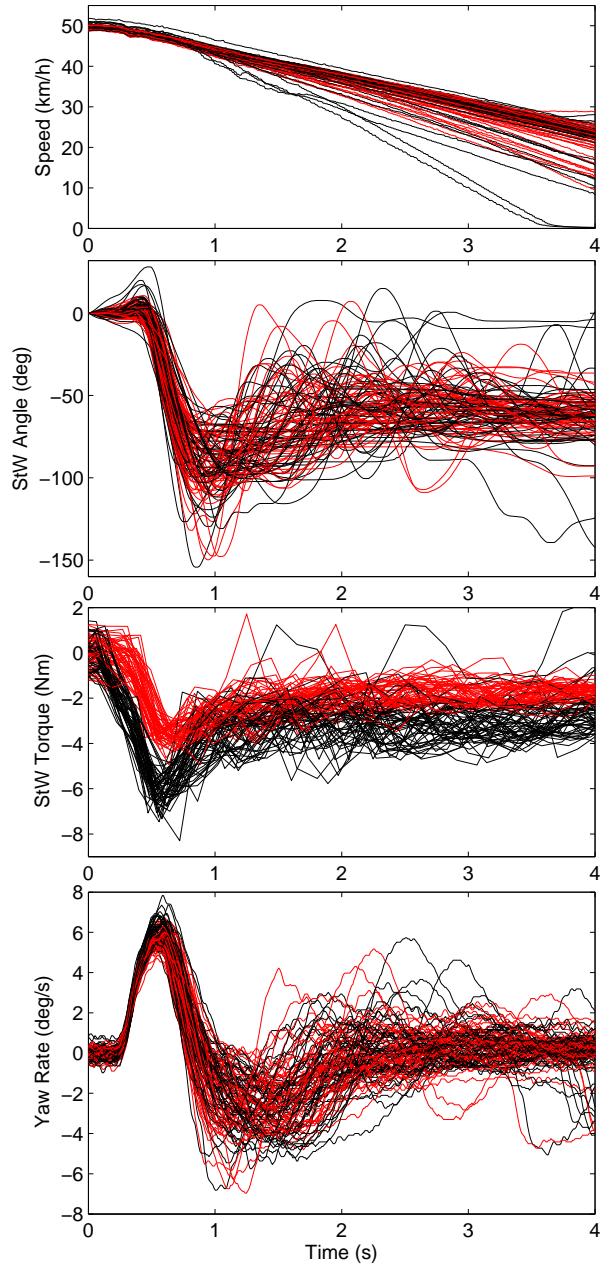


Fig. 3. Time series for all emulated tyre blow-out runs. The blow-out is initiated at time 0 s. Red lines correspond to scrub radius 0 cm. Black lines correspond to scrub radius 12 cm. In the first subfigure drive axle wheel speed is shown. The second subfigure show StW angle which is adjusted to 0 deg at time zero. The third subfigure show StW torque. The last subfigure show yaw rate.

In general, drivers that got low lateral deviation responded early and used high StW angle rate.

B. Statistical Analysis of Scrub Radius Settings

Trajectories, seen in Fig. 2, StW angle and yaw rate, seen in Fig. 3, indicate a difference when scrub radius was changed. The variance is however so high that this difference is not significant when the two groups are treated as independent, but drivers in the two groups are actually not independent. The same drivers have been used in both

groups. Therefore we can use a paired difference test to analyse the relative change for each driver. By doing so the variance used when comparing the groups will be scaled by $1/n$, where n is the number of drivers, in this case 20. The two groups will hereafter be denoted as the 12 cm and the 0 cm group respectively.

In Fig. 4 a paired t-test is performed on the travelled path data from left tyre blow-outs. First, the average path is calculated for each driver, with the two groups kept apart. Then, for each driver, the average path from the 12 cm runs is subtracted from the 0 cm runs. This is shown in black in the first subfigure. In other words it is the measured reduction in lateral deviation for each driver achieved when lowering the scrub radius. The average of these 20 curves is shown in bold red. After 24 m of longitudinal displacement the average improvement is 6.4 ± 4.4 cm, using a 95% confidence interval. 24 m is also where the maximum average displacement is observed in Fig. 2. Fig. 4 also include t-value with 19 degrees of freedom. To test if the average reduction is significant a two-tailed t-value with 98% confidence is used. This gives a t-value threshold of 2.54 which is also marked in the graph (for 99% confidence level the value is 2.86). The 98% confidence limit is surpassed after 15 m of longitudinal displacement. The highest t-value, 3.14, is reached after 21 m. It can therefore be concluded with confidence that drivers are affected by the StW torque they are subjected to. Also that the lateral deviation is lowered by having a lower scrub radius, or as in the case of the tested vehicle a power steering system that eliminates disturbances.

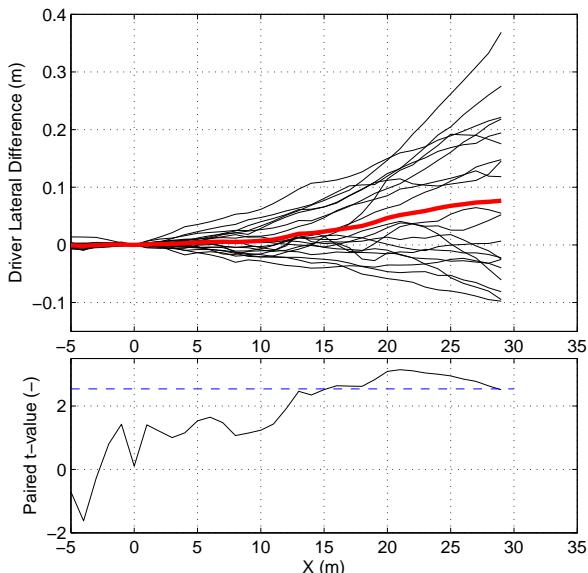


Fig. 4. A paired t-test of trajectories. In black the first subfigure show average lateral difference per driver between runs with 0 cm scrub radius and 12 cm scrub radius. The red bold curve is the average of all drivers. The second subfigure show the corresponding t-value, in solid black. Also included is a dashed blue line at 2.54 which is equal to $t_{19,0.99}$, i.e. the two-sided 98% cumulative probability value for 19 degrees of freedom.

To get a better understanding of the cause of the improvement identified we perform the same paired test also for

StW angle and yaw rate. The result is shown in Fig. 5 and Fig. 6 respectively. For StW angle we can now prove the significance for the groups between 0.3 s and 0.5 s. Drivers running with scrub radius 12 cm are here pulled by the disturbing StW torque in the wrong direction before they react and actively start to balance the blow-out by steering. However it should be noted that the significance is not strong. When using 99% confidence level the difference would not prove significant. For yaw rate a difference is also observed. Here the significance is stronger. A rough estimate show that the observed difference in StW angle is large enough to cause the observed difference in yaw rate. And the observed difference in yaw rate is large enough to cause the difference in lateral displacement.

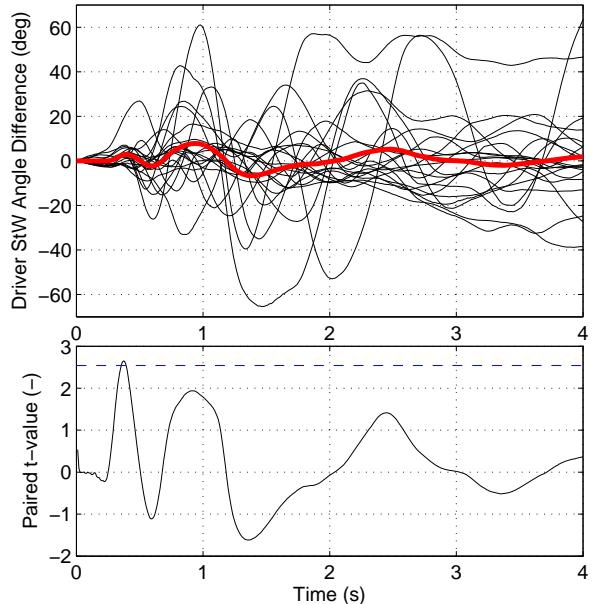


Fig. 5. A paired t-test of StW angle. In black the first subfigure show average StW angle difference per driver between runs with 0 cm scrub radius and 12 cm scrub radius. The red bold curve is the average of all drivers. The second subfigure show the corresponding t-value, in solid black. Also included is a dashed blue line at 2.54 which is equal to $t_{19,0.99}$, i.e. the two-sided 98% cumulative probability value for 19 degrees of freedom.

C. Open Loop Response

Fig. 7 show all trajectories, just like Fig. 4, but here a dashed green line is also included to show the open loop vehicle response. I.e. a run where StW angle was locked at 0 deg. For this run the lateral deviation quickly becomes more than a lane. The importance of having an alert driver is obvious.

D. Subjective Comparison of Scrub Radius

The virtual change made of scrub radius between the first and the last trials was kept secret to the drivers. Directly after the last run all drivers were asked if they had experienced any difference. Some reported that they had perceived the disturbance at blow-out as higher in some of the runs compared to others. Objectively, the level of the disturbance was the same

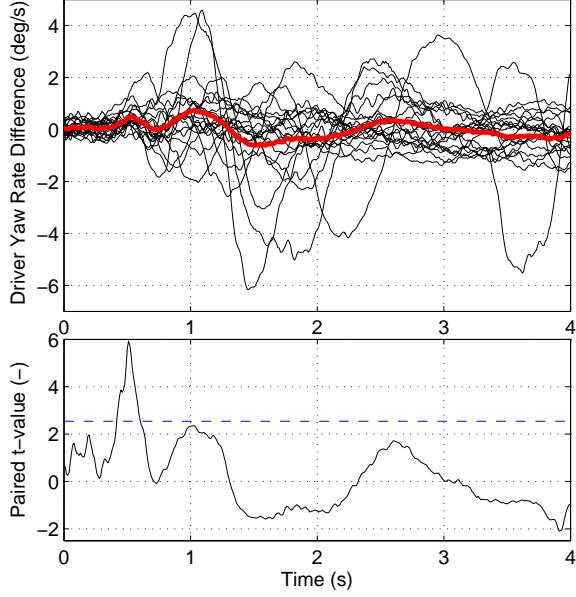


Fig. 6. A paired t-test of yaw rate. In black the first subfigure show average StW angle difference per driver between runs with 0 cm scrub radius and 12 cm scrub radius. The red bold curve is the average of all drivers. The second subfigure show the corresponding t-value, in solid black. Also included is a dashed blue line at 2.54 which is equal to $t_{19,0.99}$, i.e. the two-sided 98% cumulative probability value for 19 degrees of freedom.

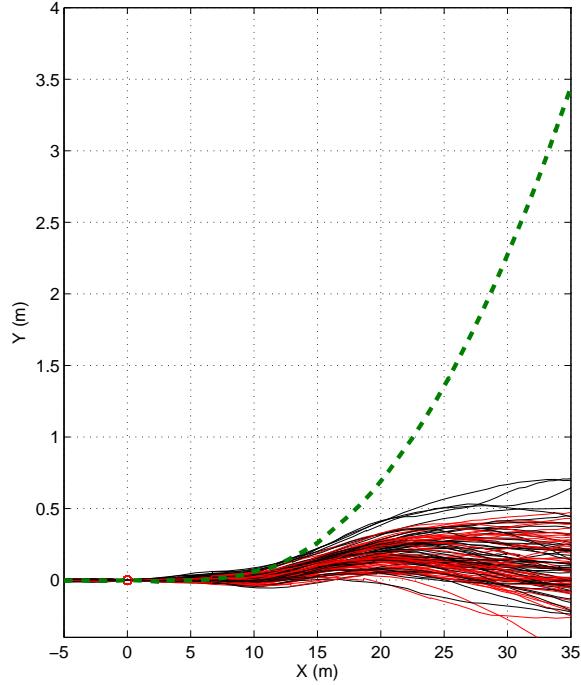


Fig. 7. Trajectories for all emulated left right blow-out runs in comparison to an open loop response. The curves have been rotated and moved so that blow-out is initiated at position (0,0) m running at zero heading. Thin red lines correspond to scrub radius 0 cm. Thin black lines correspond to scrub radius 12 cm. Bold dashed green line correspond to the open loop response, i.e. StW angle locked at 0 deg.

for all runs. No one reported that they had felt a difference connected to steering. As a follow-up question, all drivers were also asked if they had experienced any difference in the steering system. No one had. In Fig. 3 the difference in terms of steering wheel torque is apparent between the two settings. About 3 Nm of disturbance reaches the driver when scrub radius is 12 cm. As a separate experiment 3 Nm was applied to the StW during normal driving for a few drivers. All noticed that a disturbance had been applied. The difference between the normal and the critical situation is obviously an example of how the mental ability, to perform concurrent tasks, is dependent on the intensity of the main task.

E. Right Blow-Out

Some drivers were also exposed to a blow-out on the right front wheel directly after the main series of blow-outs on the front left wheel. Fig. 8 show trajectories for these runs. The average lateral deviation produced increased compared to left-blow out runs. Drivers had become used to a disturbance on the left front wheel. In line with [6] this show that repeated exposures will reduce the lateral deviation, as the driver focus harder and learn the manoeuvre.

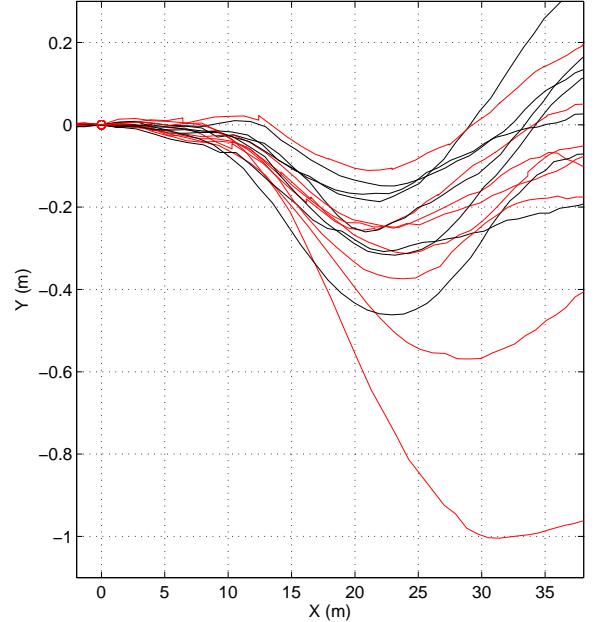


Fig. 8. Trajectories for all emulated front right blow-out runs. The curves have been rotated and moved so that blow-out is initiated at position (0,0) m running at zero heading. Thin red lines correspond to scrub radius 0 cm. Thin black lines correspond to scrub radius 12 cm.

IV. CONCLUSION

A test was set up with a 9 ton solo tractor on a test track where 20 drivers were exposed to repeated exposures of emulated worst case front tyre blow-outs. By using a configurable power steering system it was possible to alter

between two scrub radius settings, one corresponding to 12 cm and one to 0 cm. It was observed that the lateral deviation produced at a blow-out was lowered by 6.4 ± 4.4 cm when scrub radius was changed from 12 cm to 0 cm. The difference would increase for drivers holding the StW loose. In particular, the improvement for drivers not holding the StW at all would be several meters. Low scrub radius or a power steering system, that removes disturbances, could therefore ultimately slightly reduce the number of fatalities caused by tyre blow-outs.

Results also reveal that the response received from different drivers vary widely at a blow-out, irrespective of scrub radius setting. This is for instance reflected in reaction time and steering rate. These two measures have been identified as very important as to be able to maintain low lateral deviation. The test was set up with drivers that knew they would be exposed to some sort of challenge. As seen in [6] it is therefore most likely that e.g. reaction time would be higher under normal circumstances, as also partly was confirmed with results from right front tyre blow-outs. On average there is a statistically significant improvement of lowering scrub radius, but for an inattentive and less skilled driver yet more support would be needed to secure all scenarios.

To further reduce the number of accidents involving defective tyres there are several additional solutions that can be developed. First observation, reaction time is obviously critical. Designing tyres that always deflate slowly at the event of failure would therefore be beneficial. Developing stability support using brakes or additional steering, to reduce the initial heading error, would be another method. Using stabilising steering torque could be a third way. Next observation, when considering a vehicle combination with more than one unit the dynamics of the full vehicle must be considered. Here knowledge about driver behaviour is missing. When this information is available all previously suggested methods may need adjustments. And final observation, all drivers being part of this study had improved their deviation when exposed to a blow-out by the end of the session. Practical training should not be underestimated.

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