

ENHANCED BRAKING PERFORMANCE BY INTEGRATED ABS AND SEMI-ACTIVE DAMPING CONTROL

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ABSTRACT

This paper is focused on the optimization of the braking process integrating Antilock Braking System (ABS) and Continuous Damping Control (CDC). Strategies for reducing the braking distance derive from theoretical approaches. These strategies deal with sharing information between ABS and CDC in order to improve the slip-control quality and adjusting braking torque (ABS) and / or wheel load (CDC) coordinately. Quantities which influence the amount of the mean braking force and therefore the braking distance are identified methodically, regarding a standard control loop. Furthermore the influence of the time course of wheel load on the braking process is discussed.

In the second section of this paper, experimental results of straight-line ABS-braking tests for two methodically identified strategies are discussed. The results of the first experiments show the influence of passive damper settings (hard, soft) and the MiniMax damping control on the braking distance for various braking conditions (dry and wet roads, flat and unevenness roads,...). The MiniMax damping control aims for reduced body induced slip oscillations that usually disturb standard ABS-control. This damping control reduces the braking distance significantly in a statistical manner. The second experiment has been performed with a modified ABS which takes into account the information of the dynamic wheel load (due to pitching and lifting) additionally for the calculation of the braking force operation point. It is shown that the braking force operation point changes more, if dynamic wheel load information is implemented in ABS-control. Indeed the amount of modulated braking force operation point due to pitching or lifting is too small with respect to the demand, so further modifications are necessary.

Finally an outlook on the next steps for improving the braking process by integrated ABS and Continuous Damping Control is given.

INTRODUCTION

When designing a chassis of a passenger car, ride and handling are important criteria. In order to improve driving safety and comfort, several chassis control systems that control the vehicle's longitudinal, lateral and vertical dynamics were developed

and introduced in production cars in the last decades. Regarding vehicle safety, most important chassis control systems are ABS (Antilock Braking System) and ESP (Electronic Stability Program). These systems control longitudinal and lateral tire forces by adjusting tire slip, based on wheel speed sensor information. However, horizontal tire forces are limited by the amount of wheel load and friction. With information available in today's ABS the wheel load is estimated taking the vehicle mass and axle-load transfer into account only.

According to comfort, mainly the vehicle's vertical dynamic characteristics are important. With passive suspensions a known trade-off between comfort (usually measured as RMS on vertical body acceleration) and safety (usually measured as RMS on wheel load) exist. Thus, a compromise between different optimal suspension parameters for ride and handling has to be found. With adjustable damping and / or spring forces, vertical body accelerations and wheel loads can be influenced depending on the situation. Available for production cars are either semi-active, e.g. Continuous Damping Control (CDC), or active systems, e.g. Active Body Control (ABC) or Anti-Roll-System (ARS). In normal driving situations damping and / or spring forces are usually adapted according to a Skyhook algorithm in order to reduce vertical body movements (lifting, pitching and rolling). For this control strategy, vertical wheel accelerations and vertical body displacements or accelerations are detected by several sensors. Currently, semi-active suspension systems have greater market share compared to active suspension systems probably due to less energy consumption and component costs. This makes Continuous Damping Control interesting for the topic of this research: It is investigated if CDC in conjunction with ABS has potential to improve the braking process.

State of the Art

In critical driving situations, as ABS controlled braking or lane changing, the time course of wheel load should be optimized in order to realize maximum horizontal forces. In today's production cars, semi-active or active suspensions support slip-control systems (ABS or ESP) reducing body movements as pitching or lifting. These body movements cause wheel load oscillations which

disturb slip-control and lead to less mean horizontal tire forces. Therefore, for straight-line ABS-braking an aperiodic pitching behavior is intended in order to reduce disturbances. In production cars equipped with semi-active damping, this objective is aspired by switching the dampers to a rather hard setting, if ABS-braking is detected (refer to Becker et. al. [1]). However wheel load oscillations depend not only on body movements but on pavement excitations as well. This fact is not considered in this usually used control strategy. Apart from the Boolean signal “ABS-activity”, it is not of the author’s knowledge that additional information between ABS/ESP and semi-active or active suspension are shared in today’s production cars. So, information of “dynamic wheel loads” caused by pitching, rolling, lifting and pavement excitations are not taken into account in today’s ABS-control although the knowledge of the overall wheel load is necessary to adjust the braking force operation point correctly in today’s ABS systems.

In a research project, Niemz [2] developed a control strategy for semi-active damping, which is able to modify wheel load induced slip. For modifying slip in this control strategy, the wheel load is changed by means of switching the damper from hard to soft and vice versa. This control strategy has been reduced the mean ABS-controlled braking distance by 1.3% compared to those with series damper setting (constant hard setting), tested with an initial velocity of 70 km/h on a dry road in real braking tests. However, this damping control and the production ABS of the vehicle worked independently from each other, data exchange between both systems has not been taken place. Moreover, the performance and robustness of this control strategy for varying braking conditions is not investigated yet.

Motivation and Objective

For several years, coordination and information exchange of different control systems has been focused more and more by industry and research. This is based on the fact that in addition to ABS and ESP, other control systems as semi-active and active suspensions or Active Front Steering (AFS) take place in production cars. Sharing information between systems and coordination of those systems may provide greater overall performance compared to different stand alone working control-systems – often without additional production costs. As an example, the combination of ESP with AFS to so the called ESP II reduces the braking distances in μ -split situations [4]. Moreover, the combination of active antiroll-bars (ARS) with ESP and AFS reduced the braking distance even further (refer to [5]). Referring to previous research at TU Darmstadt which deals with ABS-controlled braking and semi-active damping (refer to [2] and [3]), an increased overall performance is expected if ABS and CDC

share information and work in conjunction. So, the results of this research project shall answer the question if and to what extend there is potential for reducing the braking distance by coordination of the chassis control systems ABS (as part of ESP) and CDC. This potential could be used both by sharing information and by modifying the horizontal and vertical tire forces in conjunction.

This paper is focused on the identification of possible strategies for improving the braking process with Continuous Damping Control in conjunction with ABS. Furthermore the results of braking tests with a modified damping control (MiniMax control) and a modified ABS are presented which show the potential of optimizing the braking process by means of integrated ABS and CDC. The modified ABS takes the dynamic wheel load in a very easy manner into account. Based on the theoretical approaches and experimental results, other strategies will be tested in the future which let expect potential for improving the braking process.

STRATEGIES FOR IMPROVING STRAIGHT-LINE BRAKING

Objective of this research project is the determination if and to what extend the braking process can be improved with combined operation of ABS and CDC systems. Improving the braking process to the authors view means:

- Reducing braking distance, which possibly avoids accidents or reduces their severity at last
- Faster decrease of the vehicle’s kinetic energy with respect to travelled distance, which reduces the severity of accidents that cannot be avoided
- Braking stability (lateral offset, yaw rate variation) must not be degraded

The braking distance is directly connected to the braking force, as can be seen in equation. (1).

$$d_B = \frac{1}{m_v} \cdot \int_{t_{BB}}^{t_{BE}} F_{B,total}(t) \cdot dt' \cdot dt \quad (1)$$

Due to the fact that total braking force is limited by friction coefficient μ_{max} and wheel load F_z it cannot be increased above a certain value. The average total braking force for braking to standstill is limited by friction coefficient and vehicle mass.

$$F_B = \mu \cdot F_z \quad (2)$$

$$F_{B,total,mean} = \mu_{mean} \cdot m_v \cdot g \quad (3)$$

The braking distance can be reduced either by increasing the mean friction coefficient μ_{mean} or by modifying the time course of wheel load. Neglecting aerodynamic effects the mean value of wheel load must be constant over the whole braking process – namely equal to the vehicle mass multiplied by gravity. This is due to the fact that the vehicle mass

cannot be changed within the braking procedure. The effect of wheel load distribution on the braking distance will be discussed latter.

Increasing the mean braking force

Regarding the first optimization aspect - increasing mean friction coefficient μ_{mean} to increase mean braking force - the friction-slip characteristics is important (refer to Figure 1). For stationary conditions, the maximum friction coefficient can be obtained with a characteristic slip value $\lambda_{B,opt}$, which depends on tire properties and road condition. If this slip value could be controlled exactly, the maximum total braking force for a given vehicle mass could be achieved. Braking slip λ_B is defined physically by the ratio of the wheel's rotational velocity v_W and the translational velocity of the wheel's center point $v_{W,x}$. For ABS control, $v_{W,x}$ is assumed to be equal to the vehicle velocity:

$$\lambda_B = 1 - \frac{v_W}{v_{W,x}} = 1 - \frac{r_{dyn} \cdot \dot{\varphi}_W}{v_{W,x}} \approx 1 - \frac{r_{dyn} \cdot \dot{\varphi}_W}{v_V} \quad (4)$$

Of course, braking distance can be reduced by increased friction as equation (2) and (3) show. For a given tire - road combination this can be achieved by modifying the tire properties. Although this is possible in certain applications as e.g. racing cars demonstrate this come along with other negative tire properties which usually are not acceptable for production cars. Optimizing the tire properties is not discussed within this research; hence a standard tire is assumed and used for the experiments of this research project.

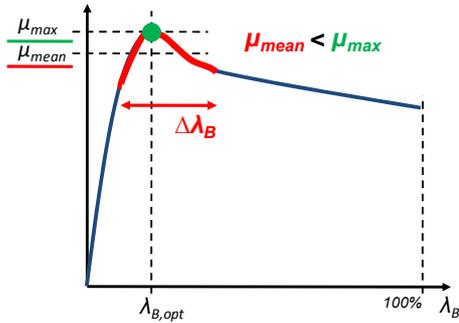


Figure 1. Example of friction-slip characteristics

For production ABS-controller, the so called target slip $\lambda_{B,Z}$ is less than $\lambda_{B,opt}$ and depends on the vehicle velocity among other things to keep slip-control smooth and effective. However in braking maneuvers slip oscillations exist. Assuming that $\lambda_{B,Z} = \lambda_{B,opt}$, slip oscillations in the nonlinear section of the μ - λ_B characteristics lead to a mean friction coefficient that must be less than the maximum friction coefficient. Therefore, as a first optimization parameter for this research, braking slip oscillations has to be as small as possible to obtain the mean friction coef-

ficient of a braking process μ_{mean} as close as possible to the maximum friction coefficient μ_{max} . This holds true for quasi-stationary conditions which are represented by the friction-slip characteristics in Figure 1. For dynamic situations, fast alternations of the braking torque or wheel load lead to even greater braking slip and friction oscillations, referring to Zegelaar [7]. The hypothesis “shorter braking distances are obtained by less slip oscillations” is not disproven yet and therefore hold true. This is shown in previous research [2] analyzing the correlation between braking distance and velocity difference ($v_V - v_W$), which is in fact the braking slip weighted with the vehicle velocity. According to [2] the correlation coefficient is between 41-63% depending on the damper settings. The physical mechanisms of the dynamic transfer behavior of braking torque and wheel load modifications on the braking slip will be examined in further research. So far it is assumed that the objective

$$\mu_{mean} \rightarrow \mu_{max} \quad (5)$$

can be achieved by minimizing slip oscillations:

$$(\lambda_{B,Z} - \lambda_B) \rightarrow 0 \quad (6)$$

Quantities which cause braking slip oscillations - In order to increase the mean braking force for a given friction-slip characteristics by means of increased mean friction due to decreased slip oscillations, the influences on braking slip have to be known. Taking a view on a rotating and braked wheel (Figure 2) with neglected vertical and longitudinal stiffness of the tire the principle of angular momentum delivers:

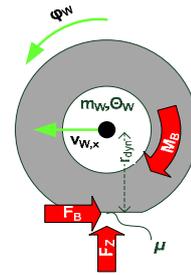


Figure 2. Quantities at a rotating wheel

$$\Theta_W \ddot{\varphi}_W(t) = F_B(t) \cdot r_{dyn} - M_B(t) \quad (7)$$

Integrating eq. (7) leads to

$$\Theta_W (\dot{\varphi}_W(t) - \dot{\varphi}_{W,0}) = \int F_B(t) \cdot r_{dyn} dt - \int M_B(t) dt \quad (8)$$

Substituting the angular velocity of the wheel with the definition of slip (eq.(4)), eq. (8) delivers:

$$\lambda_B(t) = 1 - \frac{r_{dyn}}{v_{x,W}(t)} \cdot (\Delta\dot{\phi}_W + \dot{\phi}_{W,0}) \quad (9)$$

Defining differences of the angular velocity with respect to a free rolling wheel as:

$$\Delta\dot{\phi}_W = \frac{\int \mu(\lambda_B(t)) \cdot F_Z(t) \cdot r_{dyn} dt - \int M_B(t) dt}{\Theta_W} \quad (10)$$

Equation (10) shows that differences in the wheel's angular velocity result from the integral of wheel load and /or braking torque variations in time domain. This means, there is a time delay between torque or force variations and the reaction of braking slip. Comparing the integral with a low pass filter this shows that torque or force variations with lower frequencies take even more effect on braking slip oscillations compared to higher frequencies with equal amplitudes. This means, that both increased amplitudes of braking torque or wheel load variations and low variation frequencies lead to higher braking slip oscillations. If those braking torque and wheel load variations are based on disturbances, this could result in less mean total braking force and longer braking distances. For straight-line braking, especially pitching and lifting influence wheel loads with low frequencies and high amplitudes and hence the braking slip with respect to eq. (10). Braking torque disturbances result e.g. from friction oscillations of the brake disc / pad combination. Furthermore ABS-control is not perfect due to several assumptions and actuator properties which can cause braking slip oscillations as well.

Strategies for decreasing braking slip oscillations

The previous section deals with the identification of variables which possibly cause braking slip oscillations. Thus less mean total braking force would be obtained. Braking torque and wheel load oscillations have been identified as disturbance variables. What can be done to reduce braking slip oscillations? In today's standard ABS, slip is controlled adjusting the braking torque only. Amongst others, inaccurate slip control could cause differences between applied and optimal braking torque. The latter is defined as the braking torque which is required for a specific situation. So, an increased accuracy of slip control has the potential to reduce slip oscillations, which would increase the mean total braking force.

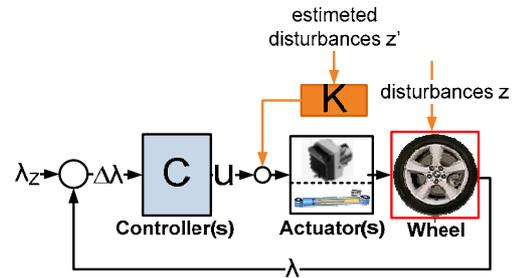


Figure 3. Simplified control loop with disturbance feed-forward (K)

Taking a view on a real ABS-control loop for straight-line braking we get to know that several quantities of the control loop (Figure 3) are estimated based on wheel angular velocities and the master brake cylinder pressure. These estimations are necessary for ABS-control because direct measurements are not available (referring to Figure 2 e.g., longitudinal wheel velocity, braking torque, braking force,...).

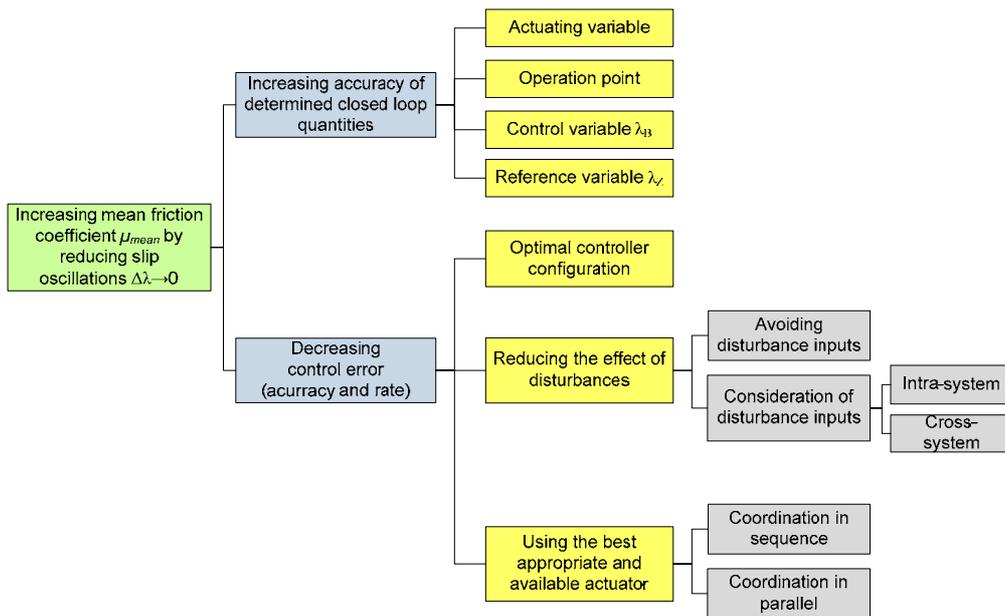


Figure 4. Strategies for increasing the mean total braking force $F_{B,total,mean}$

The control variable “braking slip” has to be determined by measuring the wheel’s angular velocity and the vehicle’s velocity. Assuming that vehicle velocity equals the wheel’s longitudinal velocity, which means that the longitudinal suspension flexibility is neglected, braking slip is calculated. However for ABS-controlled braking, the vehicle velocity is estimated as well because all wheels possess slip. So, the control variable “braking slip” is based on several estimations and might be inaccurate.

For slip control a reference variable “target slip” is needed. Desiring a high friction coefficient and braking stability (which means sufficient potential for lateral forces) the target slip derives from the friction slip characteristics for stationary conditions. This tire-characteristic varies by several inputs as wheel load, velocity, road conditions, temperature etc. Lots of these inputs are not well known and therefore not all of these influences are considered in the target-slip which results in deviations to the desired optimal target slip.

The actuating variable determines the desired braking torque of a wheel. Both operating point and the required braking torque differences to reduce control errors (controller output) are translated into caliper pressure that is influenced by the brake valves. So, the quality of slip control is also influenced by operating point and actuating variable.

In summary all of the control loop variable possibly influence slip control quality and thus braking slip oscillations which should be minimized. This is shown in the upper part of Figure 4.

As mentioned before, major task of the control loop is to minimize control errors. Regarding the control loop shown in Figure 3 and assuming target-slip and braking slip as determined precisely two components influence the control quality mainly: The controller configuration and the properties of the actuator(s), especially its dynamics and operation range. For the optimization of the controller settings the trade-off between fast error compensation and overshooting has to be dealt with. Although the controller compensates control errors the control quality can be improved by reducing the effects of disturbances on the control variable. Avoiding of disturbance inputs in general would be the best option in order to reduce slip oscillations. Instead of a closed-loop control this would allow an easier open-loop control. Although this is not possible for ABS braking the reduction of disturbance inputs should be aspired because of less control errors anyway.

Among other quantities, e.g. braking torque variations, dynamic wheel load variations due to low frequency pitching act as disturbance variable on slip control, as equation (10) shows. For ABS-controlled straight-line braking pitching results from the height difference between center of gravity and pitching center. The effect of pitching on wheel load oscillations can be reduced by increasing damping or spring forces – no or aperiodically pitching is

aspired ideally. If the vehicle’s suspension is passive, a trade-off with respect to comfort exists. For today’s production cars equipped with Continuous Damping Control (CDC) damping is switched to rather hard setting when ABS braking is detected so that wheel load oscillations and their effect on ABS control are reduced. As a consequence, hard damping results in shorter braking distances on flat roads compared to soft damping, referring to [2]. Taking into account that the integral of dynamic wheel load changes braking slip (eq. (10)) the ABS controller reacts on wheel load oscillations with a time delay in braking torque. This could be improved using the dynamic wheel load information additionally for slip control in order to decrease wheel load induced slip oscillations. In literature algorithm for active or semi-active suspension control principles are known which reduce wheel load oscillations (e.g. “Ground Hook control” [6], “Constant wheel load control” [3]) or that control wheel load induced slip oscillations directly [2]. All of these methods consider the disturbance variable “wheel load oscillations” intra-system with adjusting the vertical suspension parameters of the vehicle. However connecting semi-active or active suspension control with ABS wheel load could be taken into account for ABS control additionally which possibly reduces slip oscillation even further.

Apart from the controller, actuation of the control system “wheel” with its control variable “braking slip” is very important. In today’s production cars hydraulic valves moderate the braking pressure applied by the driver. So the braking torque is controlled by ABS. Referring to equation (10), apart from braking torque the wheel load modulates braking slip as well. With an active or semi-active suspension system that acts in vertical direction, wheel load and thus braking slip can be influenced temporarily. In order to modify braking slip, there are two different actuators available with brake torque (ABS) and wheel load (e.g. CDC) modulators. Depending on their properties ABS and CDC could act in sequence or in parallel but coordinated. The method of coordination depends on the specific properties of the actuator principle which could be different with respect to operation range, “minimal step size” and dynamics.

Wheel load influence on the braking process

At the beginning of this section it is shown that both mean friction coefficient and wheel load take effect on the braking distance in principle. In the previous section strategies for increasing the mean friction coefficient and so mean total braking force are deduced within a theoretical approach. The next section will discuss the influence of wheel load on the braking process.

Neglecting aerodynamical effects which may change wheel load with higher vehicle velocities the

overall wheel load has to be constant in steady state. Assuming wheel load and friction to be constant for the whole braking procedure the total braking force is constant, too:

$$F_{B,total}(t) = \mu \cdot m_v \cdot g \quad (11)$$

As a result of constant braking force, the braking distance is calculated by eq. (1):

$$d_B = \frac{1}{2} \frac{v_{x,0}^2}{\mu \cdot g} \quad (12)$$

Taking into account that the mean total braking force has to be constant over the whole braking process due to the constant vehicle mass (refer to eq.(11)) and assuming a linear decrease of braking force over time with eq. (14) this leads to a shorter braking distance for $\kappa > 1$, as eq. (15) shows. The factor κ describes the raise of the braking force at the beginning of the braking process (Figure 5):

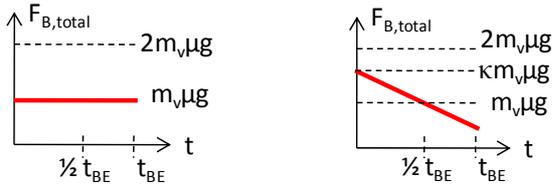


Figure 5. Constant (left) and time-variant (right) distributions of total braking force in time domain

$$\text{For } 0 \leq \kappa \leq 2: \kappa = \frac{F_{B,total}(t=0)}{F_{B,total,mean}} \quad (13)$$

The time course of total braking force is defined as

$$F_{B,total}(t) = \kappa \cdot \mu \cdot m_v \cdot g \cdot \left(1 + 2 \left(\frac{1}{\kappa} - 1 \right) \frac{t}{t_{BE}} \right) \quad (14)$$

$$d_B = \frac{v_{x,0}^2}{\mu \cdot g} \left(\frac{4 - \kappa}{6} \right) \quad (15)$$

Defining $\kappa = 2$ and assuming a linear decrease of braking force over time the braking distance is minimized to:

$$d_B = \frac{1}{3} \frac{v_{x,0}^2}{\mu \cdot g} \quad (16)$$

Compared to eq. (12) the braking distance is reduced by $1/3$ in this theoretical approach. The result shows that the time course of braking force influences the braking distance at a given mean total braking force, which depends on the vehicle mass

and mean friction coefficient μ_{mean} mainly. The higher the braking force at the beginning of the braking process the shorter the braking distance. So as a second optimization parameter, the braking force at the beginning of the braking process has to be maximized.

But how could the time course of the braking force be influenced? Of course the friction coefficient could be maximized especially at the beginning of the braking process. But this objective is aspired for the whole braking process and should not be considered at the beginning only.

Apart from the friction coefficient, the wheel load derives as a second quantity which influences the braking force (equation (2)). Controlling the time course of wheel load this could deliver higher but decreasing braking forces from the beginning of the braking process. It is possible to influence wheel load temporarily with active and semi-active suspensions. Wheel loads can be changed temporarily with these systems by adding additional spring and / or damping forces. In case of adjustable damping switching the damper in compression from soft to a hard setting increase damper force and thus wheel load (please see Niemz [2] for more details). The effect time is limited because additional suspension forces acts both on the wheel (increasing the wheel load) and the body. The latter accelerates the body upwards as long as the wheel load increases. Due to the raising displacement between body and wheel greater spring forces decelerate the body which decrease wheel load. The average wheel load equals zero but the time course of wheel load is changed temporarily.

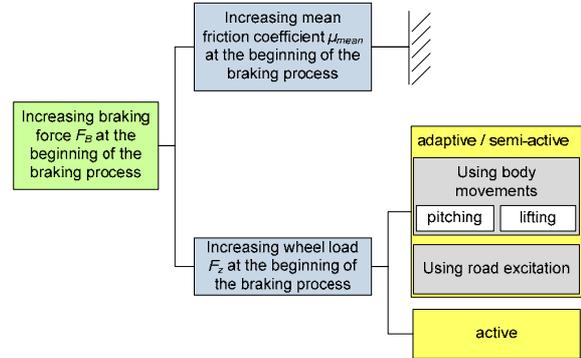


Figure 6. Strategies for increasing braking force F_B at the beginning of the braking process

Contrary to active systems with good controllability semi-active suspensions have to use system inherent energy for changing wheel loads which is less predictive. In case of semi-active damping, damper forces depend on damper velocity and damping characteristics which can be changed by a proportional valve. Without any damper velocity wheel load cannot be influenced. Energy that can be used for changing wheel load results from body movements due to pitching and lifting which appear in straight-line braking situations. Rolling can be used

in cornering situations to change wheel loads – but this is not taken into account for straight-line braking. In addition to body movements road excitations cause damper velocities as well and can therefore be used for changing wheel loads by semi-active systems too.

Summary of strategies for improving the braking process

Previous section deals with the identification of strategies in order to improve the braking process. Two quantities are identified which influence the braking process mainly:

The first is the mean friction coefficient. Limited by tire properties mainly that value should be as high as possible. This could be realized by minimized slip oscillations. A theoretical approach shows that slip oscillations result from both braking torque variations and wheel load oscillations. Strategies for reduced slip oscillations are deduced methodically considering a standard control loop. For minimized braking slip oscillations the quantities of the control loop have to be known as accurate as possible. In addition, disturbance inputs as wheel load oscillations can be reduced potentially by additional feed-forward controls in ABS and / or by control of vertical suspension systems. At last it is discussed that different actuator principles as ABS and CDC, which could be used coordinately, have the potential to improve the overall control performance “adjusting the braking slip to the target slip”. As a result it is expected that the braking distance is reduced due to higher mean braking forces.

As a second quantity, the wheel load should be maximized at the beginning of the braking process. Semi-active or active suspensions allow changing the total wheel load temporarily and can be used for this application in principle. Contrary to active suspension systems with a good controllability, semi-active suspensions systems as Continuous Damping Control have to use system inherent energy for changing wheel loads temporarily. This energy results either from body movements or road excitations.

EXPERIMENTS

In the previous section strategies for improving the braking procedure are deduced with theoretical approaches. The following section investigates the influence of selected strategies on the braking procedure in driving tests. Referring to (Figure 4) the following theoretical strategies are considered in the following sections:

1. The disturbance variable “wheel load induced slip oscillations” is minimized intra-system by using the semi-active damping control “MiniMax-control”. No ABS- modification or interaction takes place.

2. The “dynamic wheel load” is taken into account for a modified ABS-control in order to estimate the overall wheel load for each wheel. The braking force operation point is adjusted to dynamic wheel load oscillations due to pitching or lifting in order to decrease slip oscillations.

Test design - For statistical reasons straight-line braking tests with varying damping-control or ABS settings are usually repeated $N=35$ cyclically in order to compensate slow changing parameters as tire wearing or test track temperature. The start of the braking procedure, initiated automatically by a braking machine, is changed with respect to the position on the test track in order to compensate potentially particularities of the test track. The braking distance is determined using an optical Correvit-sensor, which measures the longitudinal velocity, and several light barriers reflectors with defined gaps. The braking distance is defined as the travelled distance during the time interval of $[t_{BB}, t_{BE}]$ – which represents the beginning and the end of the braking procedure in time domain. The beginning of the braking procedure is defined by a threshold of the left front calliper pressure which corresponds to the beginning of the maximal longitudinal deceleration of the vehicle. The end of the braking procedure is defined by the vehicle velocity $v_x \leq 3$ km/h. With this method, the braking distance is determined with an average accuracy of below 0.2 %. The determined braking distance cannot be compared to those which can be found in literature because the built-up time for braking pressure is not taken into account. However it is a proper method for the comparison of different damper or ABS settings with respect to braking distance because the built-up time is reproducible due to the braking machine.

1. Reducing wheel load induced slip-oscillations by semi-active damping control (MiniMax-control)

The MiniMax control strategy has been developed in previous research by Niemz [2]. Referring to equation (10) the integral of wheel load oscillations leads to slip oscillations. In order to decrease slip oscillations the MiniMax-controller switches the dampers to a hard or soft setting depending on the damper stage and the amount of the integral of dynamic wheel load which represents wheel load induced slip oscillations. For a detailed description of the controller refer to Niemz [2]. It has been proved that reducing the braking distance with this strategy is possible for an initial velocity of 70 km/h on a dry road with unevenness representative for a German Autobahn. The following results deal with the transferability to other conditions of the braking procedure. The braking tests were performed with the same test vehicle (referring to [2]), a GM Opel Astra

H equipped with CDC dampers (Continuous Damping Control). The dynamic wheel load information is estimated wheel individually by means of vertical accelerometers. The following parameters were varied in additional braking tests in order to prove the robustness of the MiniMax-controller:

- initial velocity: 70km/h and 100 km/h
- road condition: dry and wet road to vary the overall friction coefficient
- road roughness: flat, Germ. Autobahn like, very rough (nondeterministic)
- tire type: 205/55R16 summer (standard) and winter tyre of the same manufacturer

In addition to the MiniMax damping control, the braking tests have been performed with passive hard (standard setting for the production car) and soft damper setting. Any test design consists usually of N=35 braking tests per damper setting, 105 ABS-controlled braking tests overall. If less braking tests were performed for a scenario, it is noted in the figures. In order to compare the performance of the control strategy for different tests-conditions, the braking distance is normalized by the mean braking distance of all three damper settings hard (h), soft (s) and controlled (c):

$$d_{B,i, norm}^{h,s,c} = \frac{d_{B,i}^{h,s,c}}{\frac{1}{3N} \sum_{i=1}^N (d_{B,i}^h + d_{B,i}^s + d_{B,i}^c)} \quad (17)$$

It is defined that a parameter influences the performance of the control strategy (braking distance reduction) significantly if the range between minimum and maximum braking distance reduction of two test-scenarios does not overlap on a significance level of $\alpha=5\%$. A t-test is used to calculate the minimum difference of braking distances on a level of significance of $\alpha=5\%$, for the maximum difference of braking distance the 2σ limit of a Gaussian distribution is used.

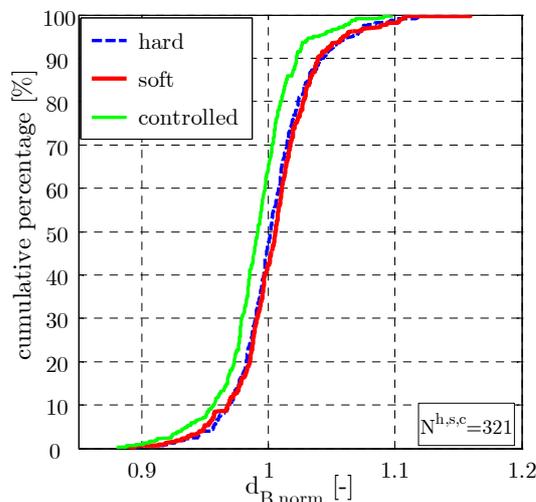


Figure 7. Overall results of N=963 braking tests

In summary, N=963 ABS-controlled braking tests with different damper settings were carried out. Figure 7 shows the normalized results of the braking distance. Assuming a Gaussian distribution, the control strategy reduces the mean braking distance compared to hard damping by approx. 1% and for soft damping by approx. 1.3% (level of significance $\alpha=5\%$).

The distributions of normalized braking distances for summer and winter tires and for the defined initial velocities do not show any significant differences with respect to the mean braking distance and deviation (refer to Figure 8 for some results of the influence of initial velocities). So, these results are not represented by a figure in this paper. For these test scenarios, the MiniMax control reduces the braking distance on dry roads significantly compared to the best passive damping (usually hard, except for very rough roads, where soft damping leads to better results).

The next sections describe the influence of different damper settings for different road roughnesses and friction conditions on the braking distance in more detail:

Influence of friction - With reduced friction μ on wet roads, less effect of the damper setting on the braking distance is expected due to less pitching and less body-induced wheel load oscillations. The mean acceleration (which is proportional to the mean total braking force and thus the braking distance, referring eq. (1)) on varying road conditions and initial velocities is presented in Figure 8. Contrary to dry roads different damper settings do not change the mean deceleration on wet “German Autobahn” like roads significantly ($\alpha=5\%$). Due to smaller damper velocities, the damping forces on wet roads distinguish less. Thus the influence of different damper settings on the wheel load oscillations is reduced. On a wet rough road, the control strategy increases the mean deceleration and reduces the braking distance significantly. A reason for this effect could be greater road induced wheel load oscillations compared to the body induced wheel load oscillations. With higher damper velocities on rough roads, the effect of switching the damper from one setting to another generates more effect on wheel load and braking slip. For proving this hypothesis, more tests are necessary.

Influence of road roughness - Driving tests on roads with varying roughness reveal the relevant excitation frequencies and the influence of damping on the braking distance. On flat roads (very flat test track), it is expected that the wheel load is influenced by pitching primary and only secondary by the road excitation. The expectation is based on the small power of road excitation compared to body induced wheel load. The influence of road excitation is expected to rise with increased road roughness.

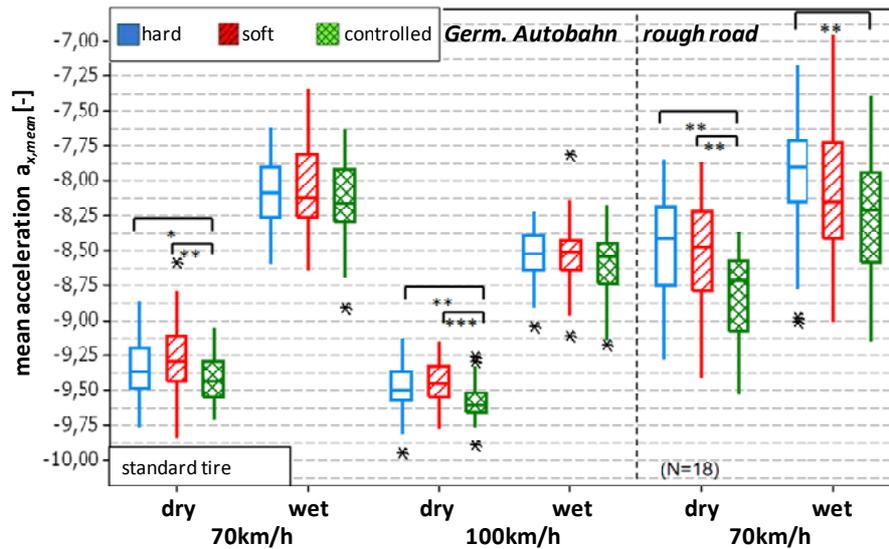


Figure 8. Variation of road conditions and initial velocities (significance levels: *(5%), **(1%), ***(0.1%))

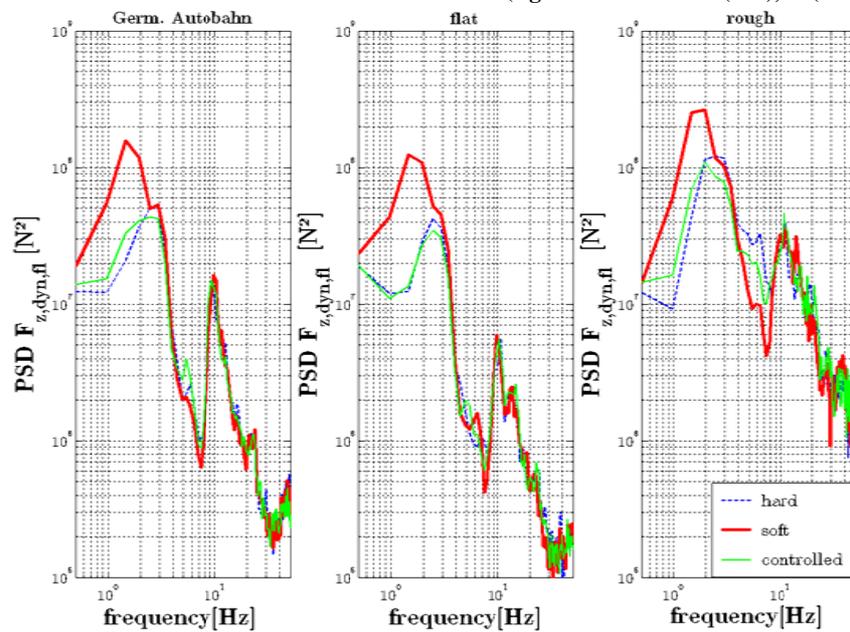


Figure 9. Mean Power Spectrum Density for the dynamic wheel load front left on varying road roughness and an initial velocity of $v_{x,0}=70\text{km/h}$

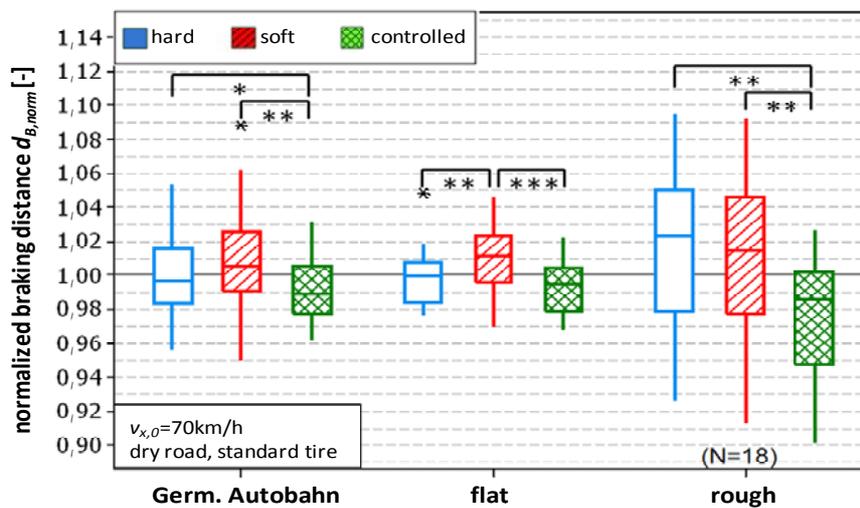


Figure 10. Variation of road unevenness (significance levels: *(5%), **(1%), ***(0.1%))

This effect is observed in the power spectral density of dynamic wheel load gained from the braking tests measured at the front left wheel, see Figure 9. As expected for the vibration characteristics the spectrum of 3-10 Hz between pitch eigenfrequency and vertical wheel eigenfrequency reveals greater wheel load oscillations with hard damping on rough roads than with soft damping. Next to the eigenfrequencies (especially pitch eigenfrequency at ~2-3 Hz), this behavior is quite contrary. The control strategy combines advantages of both damping characteristics: The wheel load oscillation in the spectrum of 3 – 10 Hz is reduced almost to the level of soft damping (see Figure 9 for rough roads). For flat roads the pitch eigenfrequency dominates the wheel load oscillation. For these roads the MiniMax control strategy produces the smallest wheel load oscillations. As a consequence, hard damping causes shorter braking distances on flat roads (“flat” and “German Autobahn”) compared to soft damping (Figure 10). According to the power spectral density (Figure 9), the body-induced wheel load oscillation dominates on these roads. On rough roads, soft damping leads to shorter braking distances compared to hard damping.

In none of the test scenarios the mean braking distance with activated MiniMax damping is longer than with passive damping (on a level of significance of $\alpha=5\%$). On German Autobahn like roads and rough roads, controlled damping reduces the braking distance on a level of significance of $\alpha=5\%$ (*) and $\alpha=1\%$ (**) respectively, compared to the best passive damper setting. The results on German Autobahn affirm Niemz’s results. MiniMax damping control reduces the mean braking distance by approx. 1%. On rough roads, the mean braking distance is reduced even by 3.5%. The control strategy solves the trade-off between hard and soft passive damper settings for different road roughnesses: hard damping for flat road surfaces, soft damping for rough roads. MiniMax control allows shortest braking distances and lowest standard deviations.

Interim Conclusion - Almost thousand braking tests analyzed with statistical methods show, that the semi-active damping control “MiniMax” can reduce the braking distance significantly, in a statistical manner. The performance depends on the conditions of the braking maneuver. With low friction and small body movements (due to pitching and lifting) the influence of damping on the longitudinal dynamics is small. On rough roads the trade-off between hard and soft damping with respect to the shortest braking distance is solved by MiniMax. Furthermore, the MiniMax controller reduces the mean braking distance by 3.5% compared to the best passive damping, which has been “soft” for these conditions. On a dry road with unevenness comparable to a German Autobahn, the MiniMax controller reduces the braking distance compared to best passive damping (hard), too. This holds true for an

initial velocity of 70 and 100 km/h. Changing the standard tire (summer) to a winter tire (same dimension) has not changed the positive effect of the MiniMax controller on the braking distance.

In summary it is proven for several braking conditions that it is possible to improve the braking process if disturbances are minimized intra-system (refer to Figure 4).

2. Extended ABS-control using dynamic wheel load information

In a previous section it has been mentioned that taking into account additional information, e.g. the dynamic wheel load, lets expect an improved slip control (refer to Figure 4). For preliminary studies with adjusted ABS-control the answer of a simple question is aimed: How does the implementation of dynamic wheel load into ABS affects the quality of slip-control and the braking distance if the ABS control parameters are not adapted to this modification? In order to answer this question ABS is modified by adding the dynamic wheel load information to the wheel load information, which is estimated by the ABS already. A standard ABS estimates the wheel load (e.g. for the wheel front left “FL”) by means of the longitudinal acceleration \ddot{x}_v shown in eq. (18) (h_{CG} : center of gravity height, l : wheelbase; m_v : vehicle mass):

$$F_{z,ABS,standard,FL} = F_{z,stat.,FL} - \frac{1}{2} \ddot{x}_v \cdot m_v \cdot \frac{h_{CG}}{l} \quad (18)$$

The modified ABS uses the overall wheel load, which takes wheel load oscillation due to road excitation and due to the body movements as pitching and lifting into account as well:

$$F_{z,ABS,modified,FL} = F_{z,ABS,standard,FL} + F_{z,dyn.,FL} \quad (19)$$

For ABS control wheel load and estimated friction define the optimal braking force operation point. The braking force operating point is an important quantity for ABS-control because it influences the amount of the caliper pressure and thus the amount of the braking torque strongly. If the operating point is chosen correctly in every braking situation, e.g. by a feed-forward-control, a slip controller would not be necessary. In previous industrial research with a BMW X5 (E70), the braking force operation point of the ABS has been adjusted continuously to the amount of weight transfer. As a result, the braking performance of this prototype ABS has been improved compared to the standard ABS.

Test vehicle and design - A BMW X5 (E70) with a programmable prototype ABS and CDC (Continuous Damping Control) is also used as test vehicle for this research. The ABS system, which only uses the weight transfer (eq.(18)) for adjusting

the braking force operation point, is used as reference for this research (“Reference-ABS”).

The vehicle is equipped with a braking machine due to reproducibility reasons. The dynamic wheel load is measured front left by a measurement rim. The dynamic wheel load information for the left front wheel is copied to the signal for the front right wheel assuming that the dynamic wheel load is dominated by pitching and lifting instead of road excitations. For the rear axle, no dynamic wheel load information is used in ABS-control. However the pitching centre is close to the rear axle, it is expected that pitching influences the wheel load oscillations of the rear axle less compared to the front axle. The determination of the dynamic wheel load by the use of a measurement rim reduces the transferability to further prospective applications, because the dynamic wheel load would be estimated by vertical sensors in production cars. For further research the dynamic wheel load will be estimated by information which are available due to the Continuous Damping Control. The dynamic wheel load information are transferred using the vehicle’s chassis high-speed CAN. The braking tests were performed according to the described test design (refer to the beginning of this section), which means N=34 cyclical repetitions of each ABS-setting with an initial velocity of 70km/h. The production car’s standard damping control has been used for these tests.

Results - The following section discusses the influence of adding dynamic wheel load information on the braking force operating point in addition to

weight transfer (eq.(19)), which has been already implemented in the test vehicle’s prototype ABS (“Reference ABS”). Apart from changing the wheel load calculation - in order to take oscillations due to pitching and lifting into account - the algorithms of the Reference-ABS have not been changed. However, for preliminary tests a very simple method has been chosen for orientation. Though, if this very simple method “adding the dynamic wheel load” already improves the braking process in terms of shorter braking distances it would be very easy to extend prospective standard ABS systems for production cars if they were equipped with semi-active or active suspensions. The Reference-ABS includes a feed-forward control in order to increase the slip control dynamics if wheel loads change (wheel load is estimated by horizontal accelerations only). If wheel load changes the braking force operating point for this wheel is adjusted directly to those changes. However, dynamic wheel load oscillations due to pitching or lifting are not taken into account for standard ABS- and Reference-ABS-control currently. As a result, the braking force operating point is not adjusted directly by the feed-forward control and so the braking slip changes slowly and with a delay due to the low-pass filter characteristics of the integral (refer to equation (10)). The braking torque is not modified till then a difference between target slip and braking slip is detected. It is assumed that taking the dynamic wheel load into account for feed-forward control of the braking force operating point this will improve ABS-control due to more dynamics.

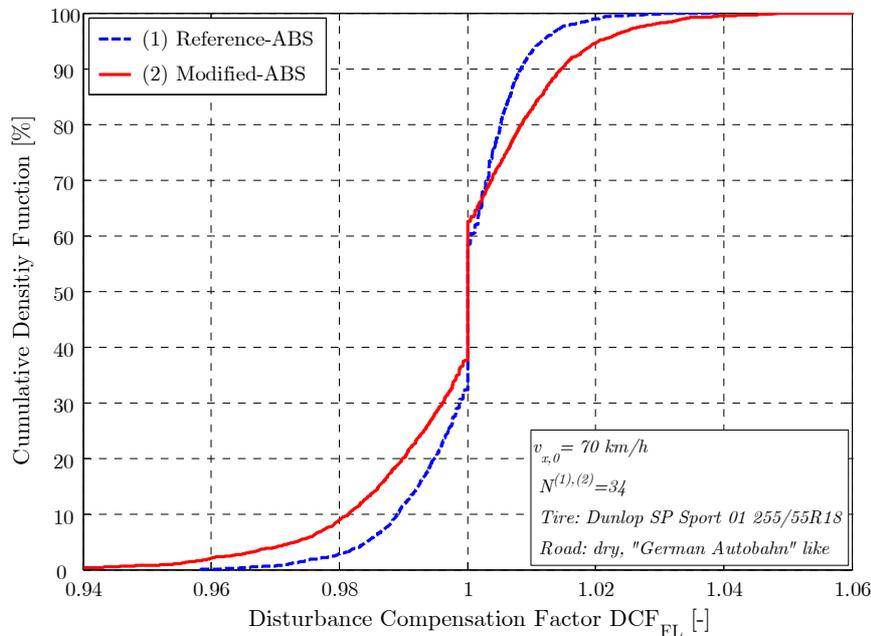


Figure 11. Deviations of the braking force operating point for Reference-ABS and Modified-ABS. The latter takes the dynamic wheel load into account

Taking a view on the results of the performed braking tests, the so called “Disturbance Compensation Factor” (DCF) is analyzed. This factor influence the braking force operating point: If the wheel load increase, the braking force operating point will be increased as well, i.e. $DCF > 1$. Figure 11 shows the effect of the dynamic wheel load on DCF . Providing the dynamic wheel load in ABS-control this modifies the braking force operating point more compared to the Reference-ABS algorithm. This is shown by the cumulative density function which is obtained from all of the carried out measurements. It can be interpreted as follow: In 50 % of the time, the DCF equals one, which means that the mean braking force operating point is not adapted in total. Regarding the range of the distributed values, with Reference-ABS (no dynamic wheel load is taken into account) the braking force operating point is adjusted by max. $\pm 4\%$. Taking the dynamic wheel load for the feed-forward control of the braking force operating point into account, the range of the DCF is increased by $\pm 2\%$ to $\pm 6\%$ in total. This is because of the fact that wheel load oscillations due to pitching and lifting are now considered by ABS-control additionally. In summary, Figure 11 shows that the dynamic wheel load information takes effect on adjusting the braking force operating point - the range is increased by max. $\pm 2\%$. Although the braking force operating point is more adjusted by the dynamic wheel load the demand is still unknown. The measurements in Figure 12 show that the ampli-

tude of the dynamic wheel load oscillation is approx. 2000 N in its maximum. Taking into account that the weight transfer is approx. 3000 N this is a rather high amount of wheel load which is not considered in standard ABS- and Reference-ABS-control. Assuming that it is optimal to adjust the braking force operating point to the time course of wheel load, the demand on a Disturbance Compensation Factor can be calculated by

$$DCF_{opt} = \mu(\lambda_z) \cdot \frac{F_{z,total,FL}}{F_{z,loadtransfer,FL} + F_{z,static,FL}} \quad (20)$$

Figure 12 shows the plot for a braking test with modified ABS. The disturbance compensation factor takes the dynamic wheel load into account. However it adjusts the braking force operating point in a range of $\pm 6\%$, as described before. Regarding the assumed DCF_{opt} for $\mu = 1$ on dry roads (for this pavement and tire $\mu_{max} = 1.15$), Figure 12 shows a demand of up to -20% to $+10\%$. So, the effect of the dynamic wheel load on adjusting the braking force operating point seems to be too small. The disturbance of wheel load oscillations due to pitching is much higher than expected from the ABS’s feed-forward control. As a consequence, the feed-forward control of the Reference-ABS should be adapted for further research.

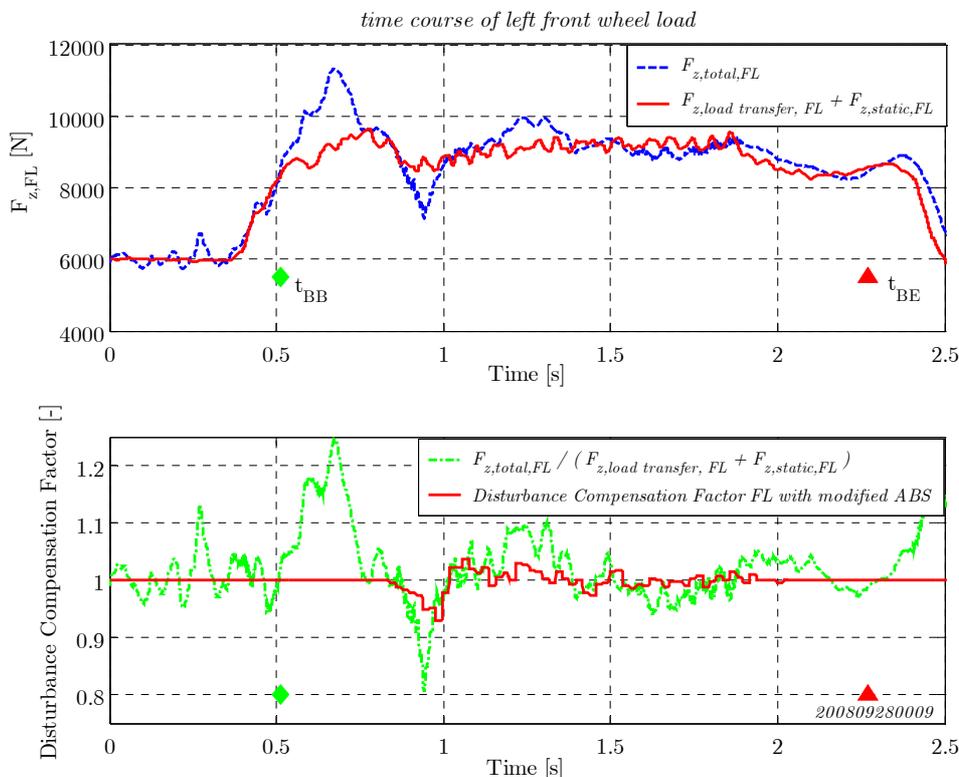


Figure 12. top: wheel load provided for standard and Reference-ABS (based on weight transfer only) and for Modified-ABS (total wheel load); bottom: available DCF vs. estimated demand

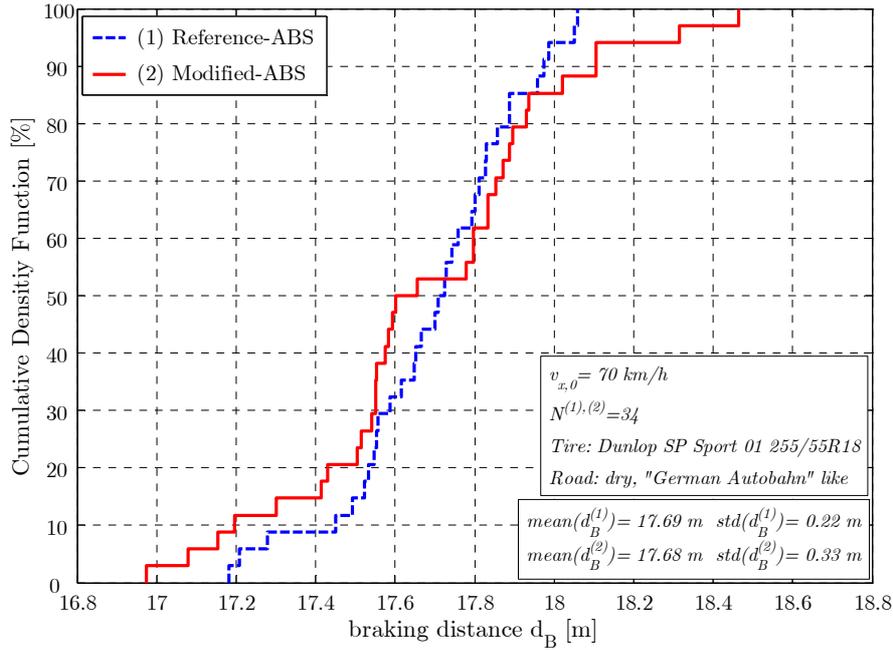


Figure 13. Braking distances, comparing Reference-ABS and Modified-ABS with dynamic wheel load

Looking at the measured braking distances no differences of the mean braking distance can be verified on a significance level of $\alpha=5\%$. It is checked with a t-test due to the normal distribution which is proven by a Lilliefors test. Figure 13 shows that the mean braking distances are similar. In addition the deviation of the braking distances with modified ABS increase which reduce the reproducibility of the braking procedure. Neither a positive nor a negative effect of the adjusted braking force operating point is proven statistically using both the feed-forward control of the Reference-ABS and the dynamic wheel load information. However it has to be considered that the dynamic wheel load has not influenced the braking force operating point only but other quantities as well during the tests. For one of these quantities, a strong negative influence of the dynamic wheel load is identified. With the available measurements, it cannot be excluded that an adjustment of the braking force operating point by adding the dynamic wheel load takes effect on the braking distance. Further braking tests are necessary to analyze the influence of dynamic wheel load on the braking force operating point and other quantities separately. It is estimated that adjusting the braking force operation point on the time course of dynamic wheel load allows shorter braking distances due to higher mean friction coefficients.

CONCLUSION AND OUTLOOK

Goal of this research project is the improvement of straight-line ABS-braking. The braking process is influenced by the vehicle's longitudinal and vertical behavior, or in more detail the braking torque and wheel load, mainly. Adjusting the braking torque

and the wheel load by ABS and Continuous Damping Control (CDC), two actuating quantities which influence the braking process mainly can be modified.

Based on theoretical approaches, the paper presents several strategies which seem to have potential to improve the braking process. They can be separated in those which increase the mean braking force by a greater mean friction coefficient and those which modify the time course of the braking force. For both optimization objects, possible strategies derive from a methodical analysis. With semi-active or active suspensions the time course of wheel load and thus the braking force can be influenced temporarily. This topic will be investigated in further studies by use of Continuous Damping Control.

Regarding the objective of less slip oscillations, the paper presents results of the MiniMax damping control for various braking conditions and the results of a preliminary study which deals with the modulation of the ABS internal braking force operation point depending on dynamic wheel load information. The braking distances with different damper settings show that in combination with a standard ABS the MiniMax controller reduces the braking distance significantly adjusting the wheel in a special matter. Furthermore the results let expect less potential for improving the braking process by means of integrated ABS and CDC control on wet roads than on dry roads due to less wheel load oscillations on wet roads.

In a preliminary study the dynamic wheel load is taken into account for ABS-control of the front wheels. A measurement rim has been used for these tests. Further tests will be performed with estimated dynamic wheel load information, based on sensors

similar to those used with CDC. In summary, taking the dynamic wheel load into account for ABS-control does not yet reduce the braking distance significantly in these preliminary tests. However it has to be considered that the dynamic wheel load has adjusted not only the braking force operating point but influences also other ABS-modules. In one of these modules, the additional dynamic wheel load information has lead to a negative effect, which possibly affects the braking distance as well. In further braking tests, the dynamic wheel load will be taken into account in ABS-modules separately in order to identify the influence of each modified module on the braking distance.

Furthermore the transfer mechanisms of both the braking torque to braking force and the wheel load to braking force will be investigated with theoretical approaches and experiments. The knowledge of both transfer mechanisms is aspired in depth in order to allow an optimization of the braking process which is transferable to other applications and vehicles. Research is ongoing and results will be published in the future.

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