Handbook of Driver Assistance Systems
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Brake-Based Assistance Functions

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Abstract

Many driver assistance systems (DAS) use the electronic stability control (ESC) system for their control tasks, while ESC itself uses the antilock brake system (ABS) and traction control system (ASR, TCS) for the control of the lateral dynamics of the vehicle. This chapter starts with the description of the systems ABS and ASR as they are used by ESC. Then the description of the system ESC as is used by DAS follows. At the end of the chapter, the brake-based assistance functions as used by DAS are described.

1 Introduction

In daily traffic, the longitudinal and lateral accelerations of a vehicle are seldom larger than 0.3 g. Therefore, the absolute value of the tire slip is seldom larger than 2 %, while the absolute values of the slip angles of the tires and the vehicle are seldom larger than 2°. Within those values, the tires and the vehicle behave more or less in a linear manner. Experience of most drivers in handling is thus limited to the linear behavior of the vehicle. If a vehicle approaches the physical limit between the tires and the road, its behavior becomes highly nonlinear. In those situations, most drivers are not able to handle the vehicle in a safe manner. Moreover, if the wheels lock during braking or spin during traction, then the driver is not able to influence the vehicle motion anymore and control over the vehicle is lost. If, e.g., the rear wheels lock before the front wheels, then the vehicle may skid (Fig. 1). Control systems which control the wheel rotation help the driver to keep the vehicle under control.

These control systems are the antilock brake system (ABS) which keeps the wheels from locking, the traction slip control system (TCS or ASR) which keeps the wheels from spinning, and the electronic stability control (ESC) which keeps the vehicle from skidding and leaving the turn. Since these control systems help the tires and the vehicle to behave in a predictable manner, they may be seen as vehicle assistance systems rather than driver assistance systems. Driver assistance systems may help the driver with his tasks to steer, accelerate, and decelerate the vehicle and to coordinate these tasks.

2 Fundamentals of Vehicle Dynamics

2.1 Stationary and Transient Behavior of Tires and Vehicles

This section deals with the handling of the vehicle in the linear and nonlinear region however not in full detail but only as is required to understand the control systems. The vehicle motion is mainly determined by the forces between the tires and the road. Therefore, understanding the tire behavior is a prerequisite for understanding the vehicle behavior. Since transients of the vehicle motion have a main influence on the control system, the transient behavior of the vehicle will also be discussed.

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Fig. 1 Skidding car on a dry asphalt road

If a wheel is not steered and neither braked nor driven, then its rotational velocity, called the free rolling rotational velocity ω_{WhlFre}, can be computed from the vehicle velocity v_v, $\omega_{\mathrm{WhlFre}} = v_v/r$ where r is the radius of the wheel. If the wheel is braked by a brake torque M_{BR}, then the rotational velocity ω_{Whl} will be smaller than the free rolling rotational velocity ω_{WhlFre}. In the description of the control systems, the wheel velocity is used instead of the rotational velocity. The wheel velocity is defined as the product of the rotational wheel velocity and the wheel radius r. The free rolling wheel velocity is then $v_{\mathrm{WhlFre}} = \omega_{\mathrm{WhlFre}} \cdot r$, while the wheel velocity is $v_{\mathrm{Whl}} = \omega_{\mathrm{Whl}} \cdot r$.

The difference between the free rolling and the braked wheel velocity is called the slip velocity. The slip velocity divided by the free rolling wheel velocity is called the wheel or tire slip λ. It is a dimensionless quantity whose value is one if the wheel is locked. Often, the slip is also expressed in percent, and the slip is 100 % if the wheel locks.

$$\lambda = \frac{v_{\mathrm{WhlFre}} - v_{\mathrm{Whl}}}{v_{\mathrm{WhlFre}}} \tag{1}$$

If a brake torque M_{B} acts on the wheel, then a brake force F_{B} between the tire and the road results. If the normal force on the tire is F_{N}, then the brake coefficient of friction between the tire and the road is $\mu_{\mathrm{B}} = F_{\mathrm{B}}/F_{\mathrm{N}}$. The road torque on the wheel M_{R} is defined as the product of the brake force F_{B} and the wheel radius r, $M_{\mathrm{R}} = \mu_{\mathrm{B}} \cdot F_{\mathrm{N}} \cdot r$. Between the brake slip and the brake coefficient of friction between the tire and the road, a nonlinear relationship exists, $\mu_{\mathrm{B}}(\lambda)$, called the μ-slip curve. Figure 2 shows typical μ-slip curves. The curves usually exhibit a maximum, but for loose road surfaces like snow and gravel, the curve may not exhibit a maximum. The slip value at the maximum of the μ-slip curve is often called the target slip λ_{T}. For values of the tire slip larger than λ_{T}, the μ-slip curve is called unstable, since there is no stable equilibrium between the brake torque and the road torque: the tire slip will not be stable for a constant brake torque but usually increase until the wheel locks.

If a free rolling wheel whose longitudinal velocity is v_x is pushed sideways, then the wheel will also move in the lateral direction and the resultant velocity of the wheel center is v_{V}. The angle between the resultant wheel velocity and the wheel plane is called the slip angle α. Because the wheel is pushed sideways, the road pushes with a side force F_{S} in the opposite lateral direction on the wheel. The side force

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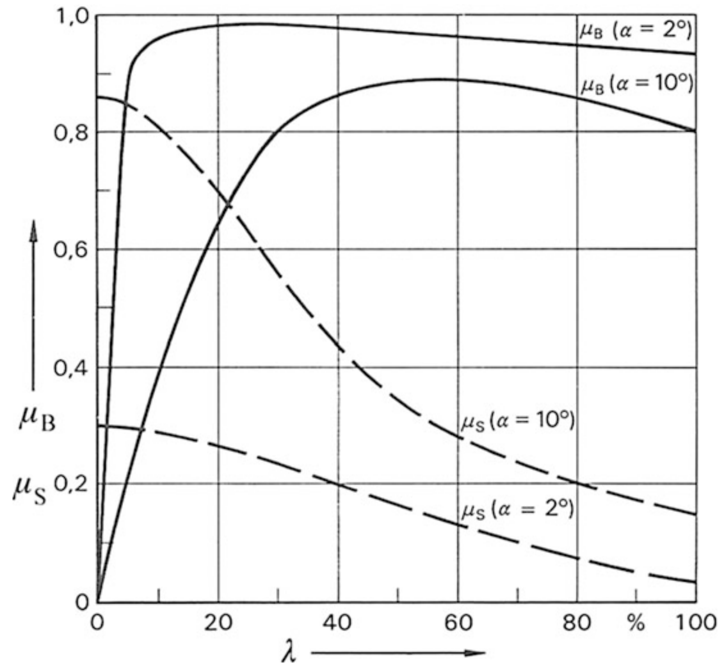

Fig. 2 μ-slip curves at some slip angle values and dependence of the lateral coefficient of friction on the slip

depends on the slip angle and on the normal force on the tire. Note that the maximum side force is not proportional to the normal force. The lateral coefficient of friction is the side force divided by the normal force on the wheel $\mu_S = F_S/F_N$. The nonlinear relation between the slip angle and the lateral coefficient of friction, $\mu_S(\alpha)$, is called the μ-slip angle curve which looks similar to that between the slip and the longitudinal coefficient of friction caused by braking.

The side force and the lateral coefficient of friction are also influenced by the tire slip and are reduced if the slip is increased. This is shown in Fig. 2. Similarly, if the slip angle is increased, then the brake force is reduced.

As mentioned above, for small steering wheel angles, vehicle handling is almost linear on a dry road. Handling is then described by the relationship between the steering angle of the front wheels and the vehicle yaw velocity using simple relations. First, the vehicle is simplified to a bicycle model that runs with constant velocity and where the transients have died out (Fig. 3). The lateral tire forces are supposed to increase linearly with the slip angles of the tires, where the slip angle is also influenced by compliance in the suspension and the steering system. This model is the basis for the vehicle dynamics control system ESC.

The steady-state yaw velocity can then be expressed by the following equation:

$$\dot{\psi} = \frac{v_X \cdot \delta}{(l_F + l_R) \cdot \left(1 + \frac{v_X^2}{v_{ch}^2}\right)} \tag{2}$$

The characteristic velocity v_{ch} determines the handling behavior of the vehicle. Its value depends on the effective lateral stiffness at the front axle, $c^{\text{'}}_{\alpha F}$, and at the rear axle, $c^{\text{'}}_{\alpha H}$, on the wheelbase $l = l_F + l_R$, on the vehicle mass m, and on the position of the front and rear axle w.r.t. the center of mass of the vehicle l_F and l_R, respectively.

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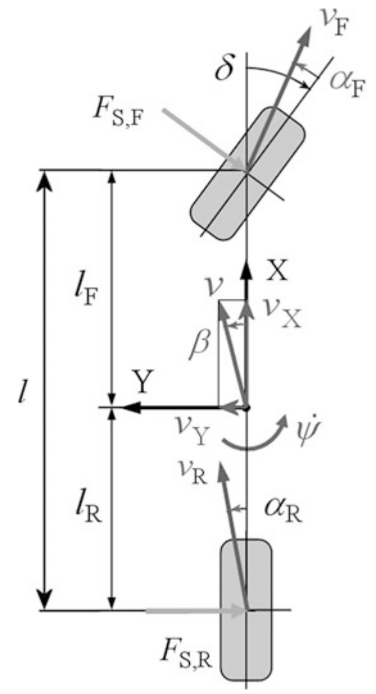

Fig. 3 Bicycle model of the vehicle

$$v_{ch} = l \cdot \sqrt{\frac{1}{m} \cdot \left(\frac{c'_{\alpha F} \cdot c'_{\alpha R}}{l_R \cdot c'_{\alpha R} - l_F \cdot c'_{\alpha F}} \right)} \tag{3}$$

However, since $c'_{\alpha F}$ and $c'_{\alpha H}$ depend almost linearly with the vehicle mass and the position of the axles, v_{ch} is in a first approximation almost independent of changes in the vehicle mass and changes in the location of the axles. If v_{ch} is positive, then the vehicle behavior is called understeer. If v_{ch} is infinite, then the vehicle behavior is called neutral steer, and if v_{ch} is imaginary, then the vehicle behavior is called oversteer.

Since the lateral acceleration of the vehicle is limited by the maximum lateral coefficient of friction between the tires and the road, $\mu_{S,max}$, the steady-state yaw velocity is also limited by $\mu_{S,max}$.

$$|a_Y| = \left| \frac{v_X^2}{R} \right| = \left| \dot{\psi} \cdot v_X \right| \leq \mu_{S,max}, \Rightarrow \left| \dot{\psi} \right| \leq \left| \frac{\mu_{S,max}}{v_X} \right| \tag{4}$$

in which R is the radius of the turn.

The yaw velocity as a function of the vehicle velocity according to Eqs. 2 and 4 is shown in Fig. 4a for several steering wheel angle values. In this figure, curves of constant lateral acceleration of the vehicle, a_Y, are also shown for several values of a_Y. If the vehicle velocity increases and the value of a_Y reaches the value of $\mu_{S,max}$ (in Fig. 4a: $a_Y = 0.775$ g), then the yaw velocity is limited according to Eq. 4. For larger values of the vehicle velocity, the yaw velocity is determined by Eq. 4.

In Fig. 4b, the yaw gain is shown, which is defined as

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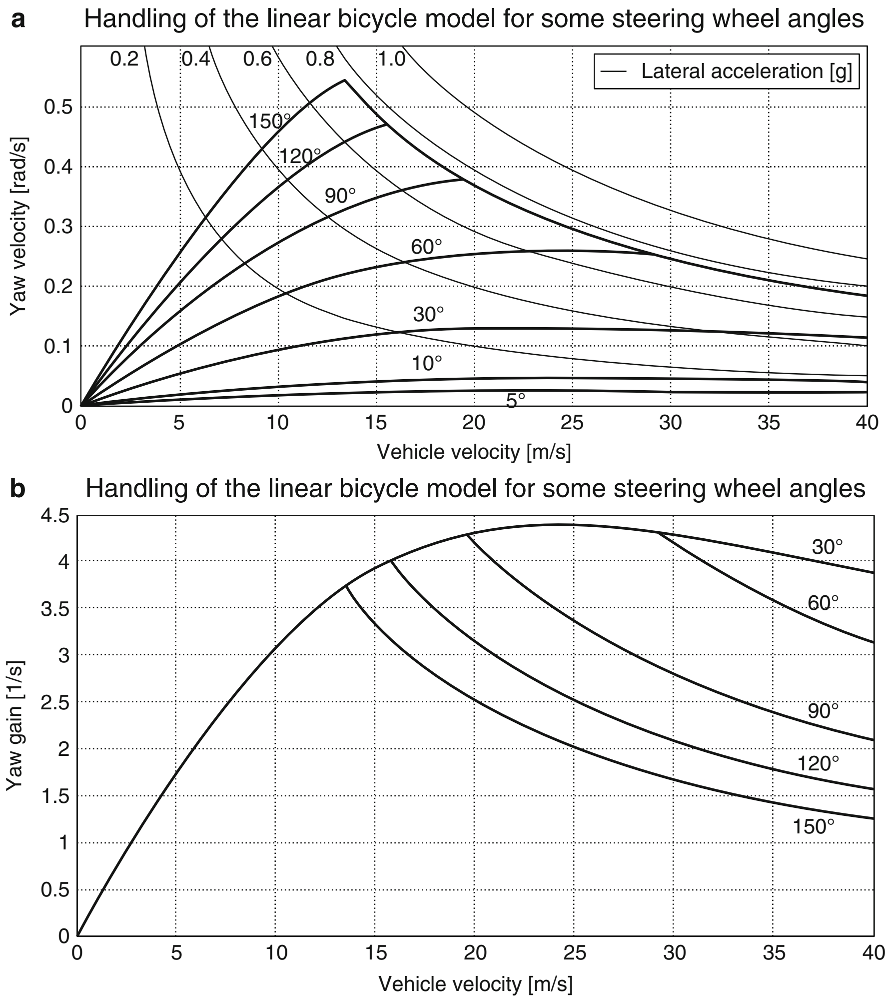

Fig. 4 Yaw velocity and yaw gain as a function of the vehicle velocity and the coefficient of friction of the road for various steering wheel angle values

$$\frac{\dot{\psi}}{\delta} = \frac{v_X}{(l_F + l_R) \cdot \left(1 + \frac{v_X^2}{v_{ch}^2}\right)} \tag{5}$$

Sudden rotation of the steering wheel can induce transients in the yaw velocity if the vehicle velocity is high enough (e.g., if $v_X > 60$ km/h). Figure 5 shows the yaw velocity after a sudden rotation of the steering wheel as measured in a front-wheel-drive middle-class car for two different car velocities. The measurement shows an oscillation of approximately 0.6 Hz in the yaw velocity. By comparison of the yaw velocities at the two car velocities, it follows that the decay of the oscillation is slower if the car velocity is higher. This can be explained by using the linear bicycle model for the evaluation of the yaw velocity and the lateral velocity of the car (Isermann 2006).

2.2 Rating Vehicle Dynamics

Handling properties are judged using vehicle maneuvers and include objective as well as subjective comparisons (Isermann 2006).

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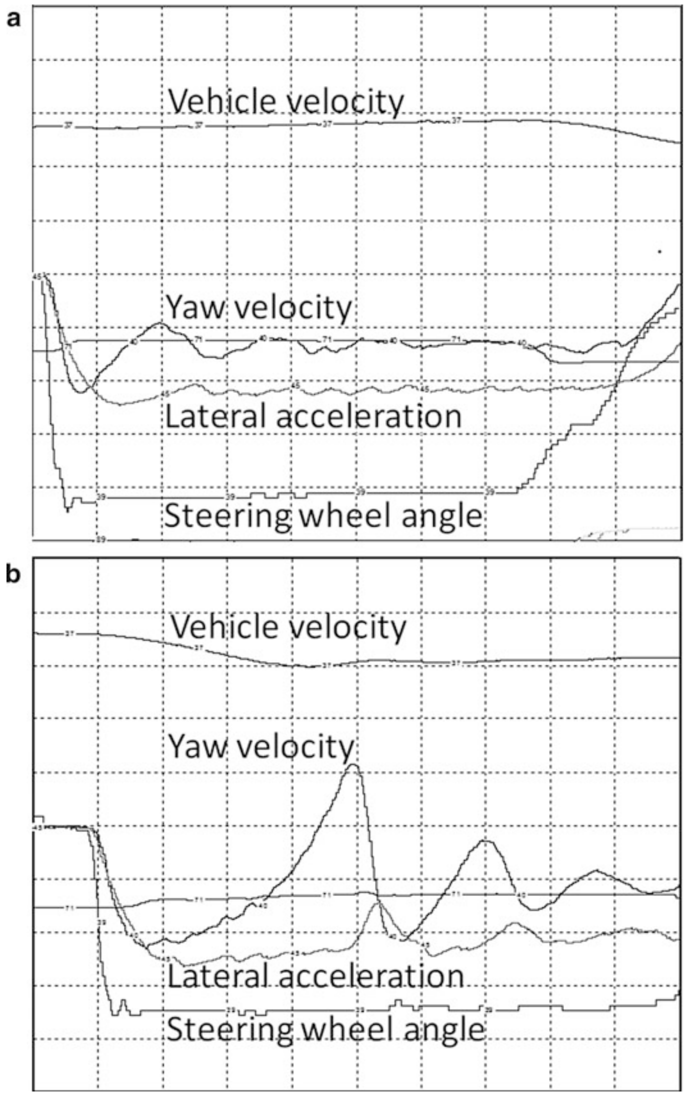

Fig. 5 Yaw velocity (**a**) after a step steering wheel angle input of 121° at a car velocity of 28 m/s and (**b**) after a step steering wheel angle input of 100° at a car velocity of 37 m/s. Plotting limits: time, 0–8 s; car velocity, −50 to +50 m/s; steering wheel angle, −145° to +145°; yaw velocity, −1 to +1 rad/s; lateral acceleration, −20 to +20 m/s^2

For the objective rating of ABS, several ISO requirements exist, like "ISO 7975 (1996): Passenger cars – Braking in a turn – Open-loop test procedure." In Germany, the braking distance is usually measured with straight-line braking on a dry asphalt surface with initial velocity just above 100 km/h. The test procedure is described in DIN 70028: "Passenger cars – Measuring the stopping distance with ABS in straight-ahead stops." The test is carried out a couple of times and an average is computed for the braking distance. Usually the test is done for cold and hot brakes. For a straight-line braking maneuver on a μ-split road surface, the German magazine "Auto Motor und Sport" has defined a rating system in which both the braking distance and the vehicle stability are considered.

In the USA, where ESC is compulsory on all cars and light trucks (FMVSS 126) since 2011, some standard maneuvers have been defined for the rating of vehicles with ESC. In particular with the test "sine with dwell steering," the vehicle has to fulfill well-described performance limits (Fig. 6).

The test maneuver is defined for a horizontal, smooth, dry, and solid road surface, with an initial velocity of 80 km/h, standard vehicle weight, tires with which the vehicle is sold, "sine with dwell

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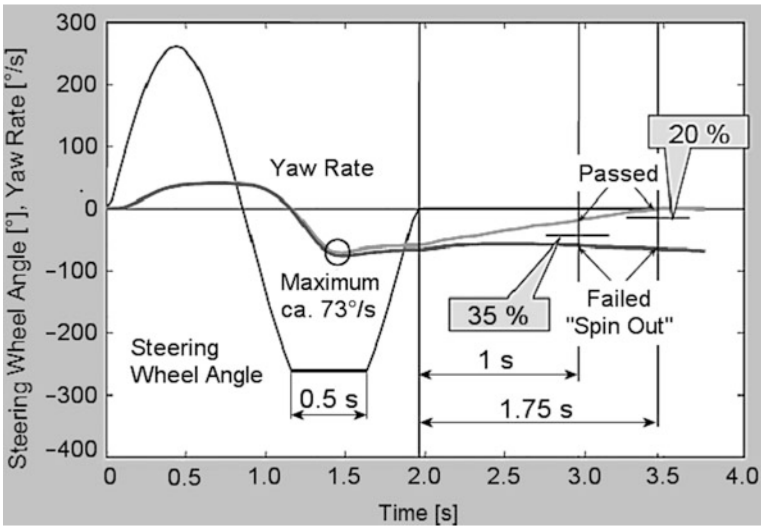

Fig. 6 Minimum requirements on ESC as defined by the NHTSA

steering" frequency 0.7 Hz, 500 ms dwell time, increasing the steering amplitude step by step by a factor of 0.5 of the initial steering angle up to the factor of 6.5 or up to the maximum of 270° steering wheel angle. The initial steering wheel angle is that angle at which the lateral acceleration of the vehicle is 0.3 g. The vehicle passes the test if 1 s after the sine with dwell steering the yaw velocity has dropped below 0.35 of its maximum value during the steering maneuver and below 0.2 of its maximum value during the steering maneuver after 1.75 s ("spinout" criteria of the NHTSA). For vehicles with gross vehicle weight up to 3500 kg, the lateral displacement must be at least 1.83 m after 1.07 s from the beginning of the steering maneuver. For heavier vehicles, this value is 1.52 m.

3 ABS, ASR, and MSR

3.1 Control Concept

In order to maintain some level of stability and steerability on all solid road surfaces, ABS must at least avoid the locking of the wheels during braking (Burkhardt 1993). For economical reasons, the vehicle velocity v_X is not measured, so that the free rolling wheel velocities and the slip levels during ABS control are unknown. For this reason, the control concept of ABS cannot be based on slip control. Instead, the control concept of ABS uses the wheel acceleration and compares it with limit values. Therefore, the ABS control concept is called control logic. The limit values are chosen such that the slip remains close to the slip at the maximum of the μ-slip curve, λ_T. The control concept of ABS is sometimes also called the optimizer principle.

In Fig. 7, the first part in time of a typical ABS control history of one wheel is shown (Robert Bosch GmbH 2004). In phase 1, the brake pressure is shown as applied by the driver. Because of the increasing brake torque at the wheels, the vehicle and the wheels decelerate. If the wheel deceleration has reached the limit value of $-a$, further increase of brake pressure at the wheel is stopped, and the brake pressure is kept constant. The brake pressure is not reduced yet, since the increasing brake force from the road on the wheel makes the wheel axle move in a longitudinal direction of the vehicle relative to the vehicle chassis because of the compliance of the wheel suspension. This motion results in a fast increase of the wheel deceleration although the wheel slip has not yet reached its value at the maximum of the μ-slip curve λ_T. The brake pressure is reduced only after the wheel velocity v_{Whl} has dropped substantially below the

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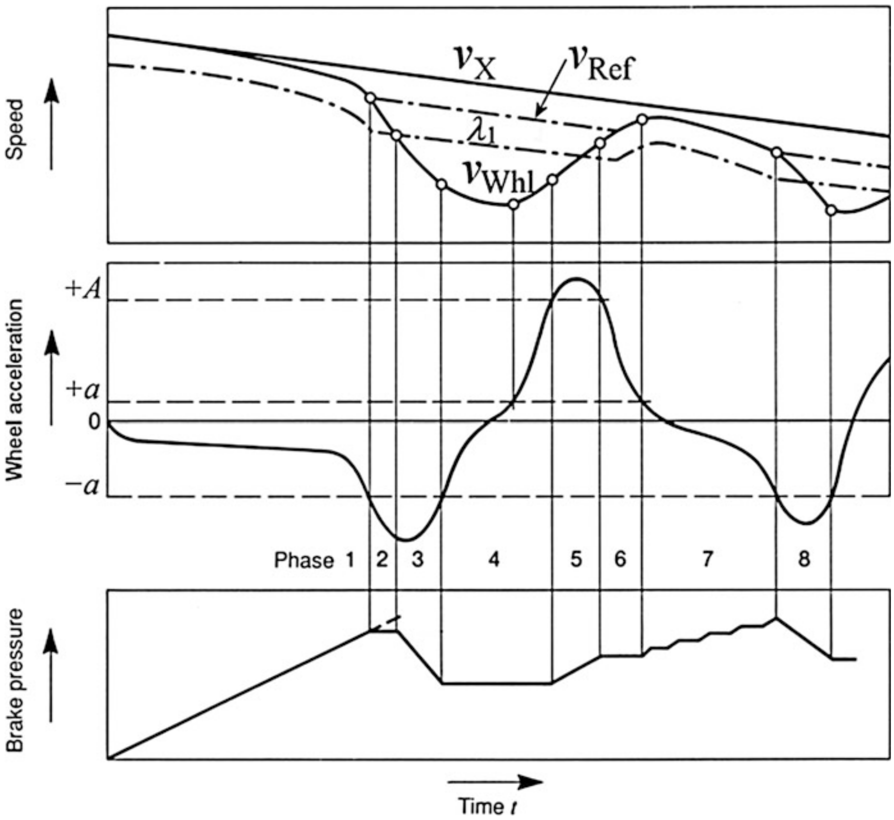

Fig. 7 Control concept at the beginning of ABS control

so-called reference velocity v_{Ref}. The reference velocity is an artificial signal which is supposed to be approximately equal to the wheel velocity if the wheel slip were λ_T. From the beginning of the brake application, the reference velocity follows the wheel velocity until the wheel deceleration has reached the limit value $-a$. Then the reference velocity is extrapolated with a certain rate, which is -0.3 g at the beginning of the ABS control. As soon as the wheel velocity has also dropped below the limit value λ_1, the brake pressure is reduced by the ABS control.

In phase 3, the brake pressure at the wheel is continuously reduced as long as the wheel deceleration is below the limit value $-a$. In phase 4, the brake pressure at the wheel is kept constant which results in an acceleration of the wheel. In phase 5, the acceleration increases above a limit value $+A$. The brake pressure is increased as fast as possible, and the increase is continued as long as the acceleration is larger than the limit value $+A$. Otherwise the brake pressure is kept constant (phase 6). If the acceleration has dropped below the limit value $+a$, it is assumed, that the brake force is nearly maximal. The slip has now almost reached a stable value near λ_T. Since the brake force is almost maximal, it suffices to increase the brake pressure slowly which is realized by a stepwise increase of the brake pressure. Thus, the brake force is almost maximal and stays maximal for a longer time. If the wheel deceleration drops again below the limit value $-a$, the brake pressure is reduced immediately in phase 8 since it is supposed that the brake force from the road on the wheel is almost constant because of the ABS control, and therefore the axle motion relative to the chassis is almost zero. The first step of the stepwise pressure increase is variable and chosen such that after the first step increase of the brake pressure, the tire slip is already close to λ_T.

If during propulsion the driven wheels spin, the lateral force on the vehicle cannot be influenced by changing the slip angles of the driven wheels. ASR, the traction slip control system, should prevent the wheels from spinning and controls the traction slip of the driven wheels by modulation of the engine

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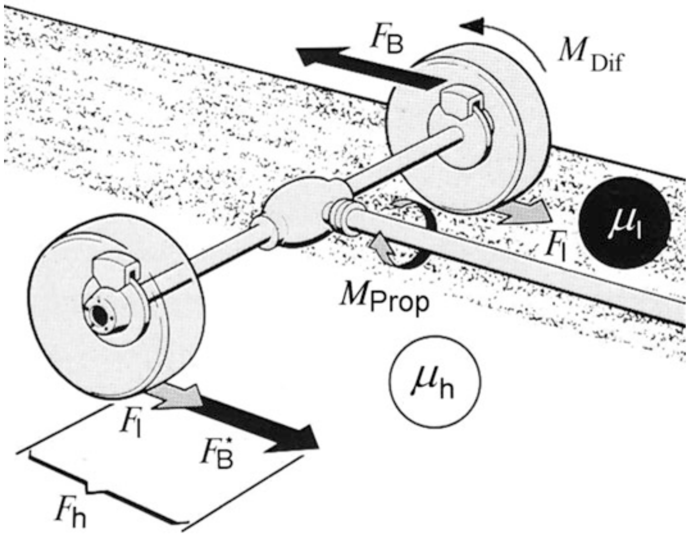

Fig. 8 ASR traction control on an asymmetric road

torque and, if necessary, also by active braking of the driven wheels. The μ-slip curve for traction slip looks very similar to the μ-slip curve for brake slip. However, ASR cannot adopt a control logic similar to that of ABS, which is based on the wheel acceleration. The reason is that the clutch is engaged so that the effective moment of inertia at the driven wheels is very large, particularly in the first gear. It is then not possible to clearly differentiate between stable and unstable slip values on the basis of the wheel acceleration. Moreover, the engine torque also depends on the engine speed. Therefore, the ASR control concept is different from the control concept of ABS.

ASR controls the wheel traction slip. For the evaluation of the wheel traction slip, the velocity of the nondriven wheels is used as the vehicle velocity. For all wheel-drive vehicles, this strategy cannot be used. For these vehicles only a traction control, which synchronizes the velocities of the driven wheels, was offered without engine torque modulation. However, with the introduction of ESC, the vehicle velocity is also estimated during ASR, so that the ASR control also became possible for four-wheel-drive vehicles.

During straight-line acceleration on a homogeneous road surface, the traction forces and the traction slips at the driven wheels on the same axle are approximately equal. Both driven wheels may spin if the traction slip reaches λ_T. ASR then reduces the engine torque to reduce the wheel traction slips. With gasoline engines, the throttle valve is closed, while with diesel engines, the control lever of the injection pump is retracted.

During straight-line acceleration on a μ-split road surface, the wheel on the low-μ road surface (μ_l) may spin first (Fig. 8). Without traction slip control, the traction forces at both wheels are equal to F_l. Traction slip control actively brakes the spinning wheel on the low-μ road surface with a force F_B on the brake disk to keep the wheel slip close to λ_T. This results in an increase of the traction force at the wheel on the high-μ road surface by F_B*. If the torque at the propulsion shaft M_{Prop} is so large, that the wheel at the high-μ road surface also starts to spin, i.e., $F_h = F_l + F_B$* has reached its maximum value, the engine torque is also reduced by ASR control. Ideally, ASR keeps the wheel velocities of both wheels approximately equal, since a difference between the velocities together with a high torque at the axle will stress the differential considerably.

If the accelerator pedal is released, the resulting braking torques at the driven wheels may be so high that in particular on low-μ road surfaces, the engine runs more or less in idle velocity. The brake slips at the driven wheels may then be much larger than λ_T. The driver may lose control of the vehicle. ASR can also be used in such situations to increase the engine torque in order to reduce the brake slips of the driven

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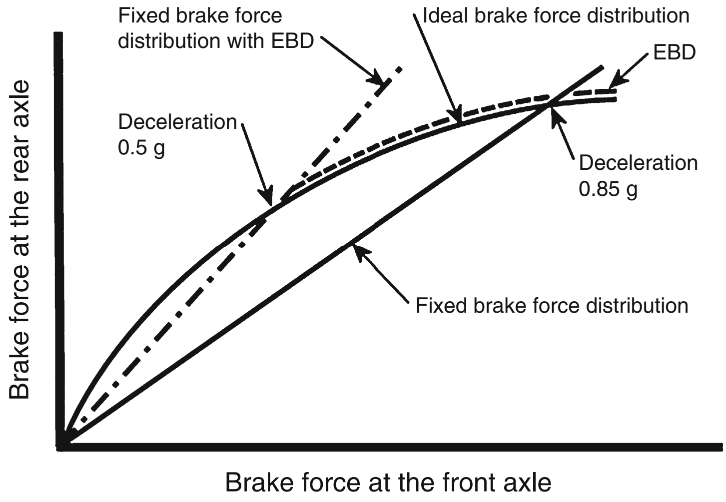

Fig. 9 Principle of the electronic brake force distribution, EBD

wheels. This control is called engine drag torque control (MSR). The ASR and MSR control will be described in more detail in the section on ESC.

Brake regulations (ECE13-H) require that during straight-line braking with vehicle decelerations below 0.85 g, the rear wheels must not lock before the wheels at the front wheel lock. This requirement can be achieved by appropriate dimensioning of the brakes at the front wheels and at the rear wheels. The brake force distribution front to rear is then fixed (Fig. 9), with the result that for most brake applications where the vehicle deceleration is below 0.3 g, the rear wheels hardly contribute to the total brake force on the vehicle. The brake force distribution is called ideal, if the coefficient of friction between the tires and the road surface at the front and the rear wheels are equal. The fixed brake force distribution can be modified to approach the ideal brake force distribution for small vehicle decelerations. However, the regulations may then be violated for vehicle decelerations below 0.85 g, e.g., for vehicle decelerations above 0.5 g (Fig. 9). Therefore, in the past, mechanical components have been installed to limit or reduce the brake pressure at the rear wheels for larger vehicle decelerations. Wear, corrosion, and other influences on the function of these mechanical components may again lead to violation of the regulations after some time.

The function of the mechanical components can also be implemented by an extension of ABS which is called the electronic brake force distribution (EBD). EBD monitors the wheel velocities and controls the brake pressure at the rear wheels such that the velocity at the rear wheels is not by much lower than the velocity at the front wheels. If the front and the rear wheel velocities are equal, then the slips of all wheels are equal, and the coefficients of friction between the tires and the road are equal at all wheels. Thus, EBD makes the brake force distribution to follow the ideal brake force distribution closely. For most brake applications, the vehicle deceleration is small (in Fig. 9 below 0.5 g), and the brake force at the rear wheels of the fixed distribution is smaller than the brake force of the ideal distribution. Thus, for most brake applications, the velocity of rear wheels is larger than the velocity of the front wheels, and EBD will not intervene. EBD interventions may produce some noise and some brake pedal feedback which may irritate the driver. However, for most brake applications with vehicle decelerations below 0.5 g where EBD does not intervene, there will be no irritation of the driver.

The EBD function is the following: If the velocity of the rear wheels is below the velocity at the front wheels by a certain first limit value, then the brake pressure at the rear wheels is kept constant. This first limit value depends on the vehicle velocity. If the velocity of the rear wheels is below the velocity at the front wheels by a certain second limit value which is larger than the first limit value, then the brake pressure at the rear wheels is reduced. The brake pressure reduction at the rear wheels is continued as long

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as the velocity of the rear wheels is below the velocity at the front wheels by the second limit value. Thus, the difference in the velocities between the front and rear wheels is kept small. If the difference becomes smaller than the first limit value, then the brake pressure at the rear wheels is increased in a stepwise manner. In case of a failure of ABS and if the EBD function fails, the driver is informed by a red light since ECE13-H regulation is violated.

4 ESC

4.1 Requirements

The requirements on ESC relate to the handling of the vehicle at the physical limit of lateral dynamics. Handling performance is judged by experts objectively as well as subjectively. However, the tuning of ESC on the vehicle largely depends on the expert and on the philosophy of the company. There is hardly any correlation between the subjective and objective rating of the vehicle handling performance.

At the physical limit, the tire forces on the road cannot be increased in magnitude. During full braking, a compromise must be found between the desire for maximum longitudinal forces on the tires for the shortest stopping distance on the one hand and for maximum lateral forces on the tires for smallest deviations of the vehicle from the desired path on the other hand. During free rolling of the vehicle, a compromise must be chosen between best steerability and stability of the vehicle on the one hand and unintended deceleration of the vehicle on the other hand.

The requirements of ABS and ASR also apply to ESC. Additional requirements on ESC describe the compromises mentioned above (Breuer and Bill 2013).

4.2 Sensors

For the evaluation of the vehicle state, cheap sensors for automotive applications are used. These are an inertial angular rate sensor for the yaw velocity and an accelerometer for the lateral acceleration of the vehicle. For the evaluation of the nominal vehicle motion, an angle sensor for the steering wheel angle and a pressure sensor for the pressure in the brake master cylinder (MC) are used. Furthermore, wheel velocity sensors as used for ABS and ASR are also used for the evaluation of the wheel velocities. These sensors are described in chapter of this book "▶ Vehicle dynamics sensors for Driver Assistance Systems".

4.3 Control Concept of ESC

ESC was developed on the basis of ABS and ASR with which the pressure in the individual brake wheel cylinders (WC) can be modulated, the wheels can be actively braked, and the engine output torque can be controlled. The control concept of ESC relies on the effect that the lateral force on a tire can be modified by tire slip (Fig. 2). Thus, the lateral dynamics of the vehicle can be influenced by modification of the slip value of each individual tire. For this reason, ESC uses the wheel slip as the vehicle dynamics control variable (van Zanten et al. 1994). Basically, the yaw moment from the road on the vehicle can be influenced by the slip value of each of the four wheels. However, a change of the slip value at a wheel also implies a change of the longitudinal force from the road on the wheel and with that an unintentional change in the vehicle acceleration.

The effect of a slip change of the wheel is a rotation of the resulting horizontal force from the road on the wheel. This is shown in Fig. 10. In the figure, a car is shown in a right turn with no braking forces nor traction forces on the wheels, and it is supposed that the lateral wheel forces have reached their maximum values, i.e., that the slip angles of the wheels are those where the slip angle curves have their maximum values. On the front left wheel, the resulting force from the road is initially $F_R(\lambda = 0)$ which is equal to the side force $F_S(\lambda = 0)$. If ESC applies a brake slip of λ_0 to the front left wheel, then the lateral force will be

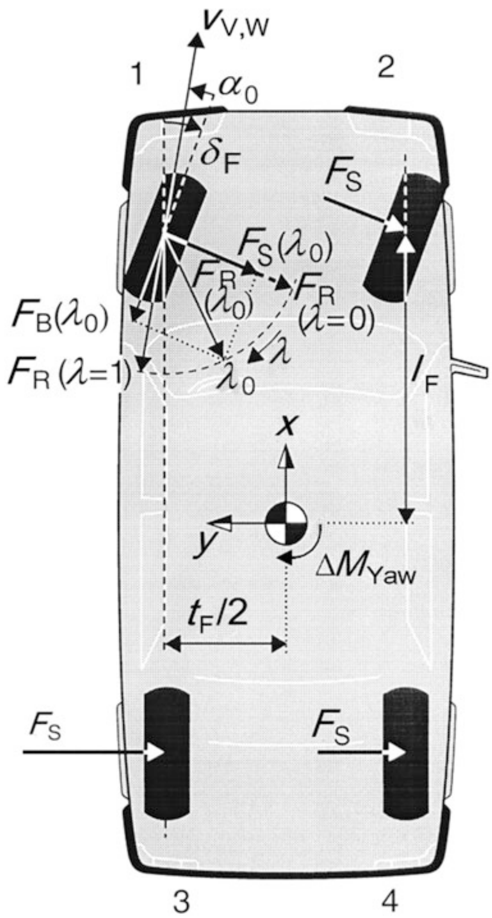

Fig. 10 Rotation of the resultant tire force F_R by a slip change from 0 to λ_0

reduced to $F_S(\lambda_0)$, and a brake force from the road on the wheel results which depends on the wheel slip $F_B(\lambda_0)$. The geometrical sum of these forces is then the resultant force $F_R(\lambda_0)$. Usually, the friction circle between the tire and the road is assumed (Schindler 2007). This means, that the absolute values of $F_R(\lambda = 0)$ and $F_B(\lambda_0)$ are equal. As a result of the brake slip application, the resultant wheel force from the road on the tire is rotated, and the yaw moment of the wheel about the center of mass of the car is modified by ΔM_{Yaw}.

$$\Delta M_{Yaw} = \frac{\partial F_S}{\partial \lambda} \cdot \Delta\lambda \cdot (l_F \cdot \cos\delta_F - 0.5 \cdot t_F \cdot \sin\delta_F) - \frac{\partial F_B}{\partial \lambda} \cdot \Delta\lambda \cdot (0.5 \cdot t_F \cdot \cos\delta_F + l_F \cdot \sin\delta_F) \quad (6)$$

The rotational angle of the resultant force increases with increasing brake slip but is limited by the maximum brake slip $\lambda = 1$. If the brake slip is 1, then the direction of the resultant brake force of the road on the wheel is opposite to the direction of the horizontal velocity of the car at the wheel center $v_{V,W}$.

In Fig. 11, the influence of changes in the values of brake slip on the yaw moment on the car at each wheel is shown for all brake slip values, for a dry asphalt road (μ ≈ 1.0), as well as for a packed snow road surface (μ ≈ 0.3) as computed by simulation of a simple car/tire model. It shows that by changing the brake slip of the front left wheel, the yaw moment on the center of mass of the car may be changed considerably, from approximately +5000 Nm at $\lambda = 0$ to −4000 Nm at $\lambda = 1$ on a dry asphalt road. On the packed snow road surface, the result is qualitatively the same; however, the values are smaller because of the lower coefficient of friction of packed snow. Variation of the brake slip of the front right wheel

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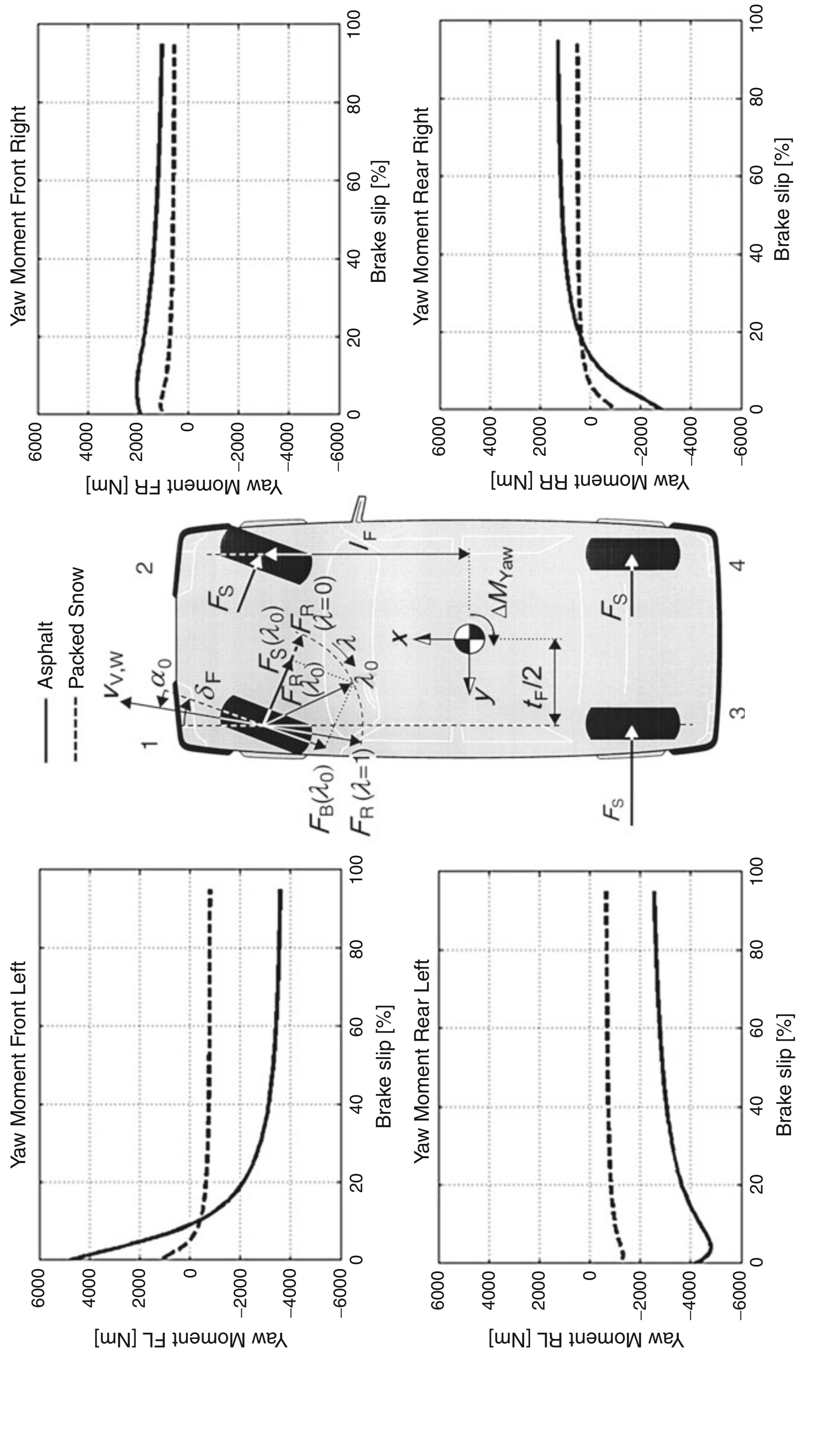

Fig. 11 Influence of a wheel individual brake slip change of the yaw moment on a dry asphalt road and on a packed snow road during stationary cornering with limit velocity and with radius of the turn 100 m

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shows little influence on the yaw moment. The reason is that the lever arm of the resulting wheel force to the center of mass of the car increases with increasing brake slip values first, but decreases again for larger brake slip values. As a result, the yaw moment curve shows a small maximum. Changing the brake slip at the front axle of the wheel on the outside of the turn has a much larger influence on the yaw moment of the car than changing the brake slip at the front axle of the wheel on the inside of the turn in this vehicle situation. This has to be considered with the control of the yaw moment on the car. During heavy cornering, the rear wheel on the inside of the turn may lift off, and brake slip interventions at this wheel will have no effect on the yaw moment on the car. This has to also be considered with the control of the yaw moment on the car. If in the free rolling situation of the car at the physical limit its behavior is oversteer during cornering, increasing the brake slip of the wheel at the front axle on the outside of the turn reduces the yaw moment and thus reduces the oversteer behavior of the car.

In Fig. 12, the influence of changes in the value of traction slip on the yaw moment on the car at each wheel is shown for all traction slip values, for a dry asphalt road ($\mu \approx 1.0$), as well as for a packed snow road surface ($\mu \approx 0.3$) as computed by simulation of a simple car/tire model. It follows that by changing the traction slip value of the front left wheel of a front-wheel-drive car, the yaw moment on the center of mass of the car does not change by much and shows a small maximum like explained for the front right wheel with brake slip above. On the packed snow road surface, the result is qualitatively the same; however, the values are smaller because of the lower coefficient of friction of packed snow. For a rear-wheel-drive car, the influence of changes in the value of traction slip on the yaw moment on the car is also shown. In particular, traction slip changes at the rear axle on the outside of the turn change the yaw moment on the car considerably from −4000 Nm at $\lambda = 0$ to +2000 Nm at $\lambda = -1$ (for historical reasons, traction slip is defined to be negative).

Because of the clear relationship between wheel slip and yaw moment, a hierarchical structure of the control is very useful, in which the control of the vehicle motion determines the required yaw moment change and in which the control of the wheel slips determines the rotation of the forces on the wheels. Control of the wheel slips includes the basic functions of ABS and ASR. Since ASR controls the wheel slips of the driven wheels, the ASR control can be used as wheel slip control for traction slip. However, since standard ABS control is based on wheel acceleration and not on wheel slip, ABS control cannot be used for wheel slip control during ABS braking. Instead, a novel ABS control is used which controls the wheel slip during ABS braking. Both the traction slip control and the brake slip control will be explained below. However, ESC control structures exist in which both an acceleration controller for the ABS function and a brake slip wheel controller for the modification of the yaw moment are used. This control structure is called modular.

The hierarchical control structure is shown in Fig. 13 with the vehicle as the process to be controlled and with the central elements the state variable control for the control of the vehicle motion, the brake slip control, and the traction slip control. An important component of the ESC control is an observer, with which the vehicle motion is analyzed and estimated. Another important component is the determination of the nominal motion of the vehicle expressed by its yaw velocity and its slip angle. These values are evaluated by using the driver input (the steering wheel angle δ, the brake master cylinder pressure p_{Circ}, and the driver-requested engine torque $M_{\mathrm{M,Pre}}$ as obtained from the engine management system), the lateral acceleration of the vehicle a_{y}, the longitudinal acceleration a_{X} of the vehicle, and the longitudinal velocity of the vehicle v_{X}. The output of the state variable controller is the required change in the yaw moment on the vehicle $\Delta M_{\mathrm{Yaw,Nom}}$. From the required change in the yaw moment on the vehicle $\Delta M_{\mathrm{Yaw,Nom}}$, the required changes in each wheel slip $\Delta\lambda$ can be computed, e.g., from Eq. 6 for the left front wheel. Since changes in the slip value at one wheel may have a much larger effect on the yaw moment on the vehicle than changes in the slip value at another wheel as explained by Fig. 10, the wheels that are chosen to change the yaw moment must be carefully selected. The brake and traction slip controllers compute the

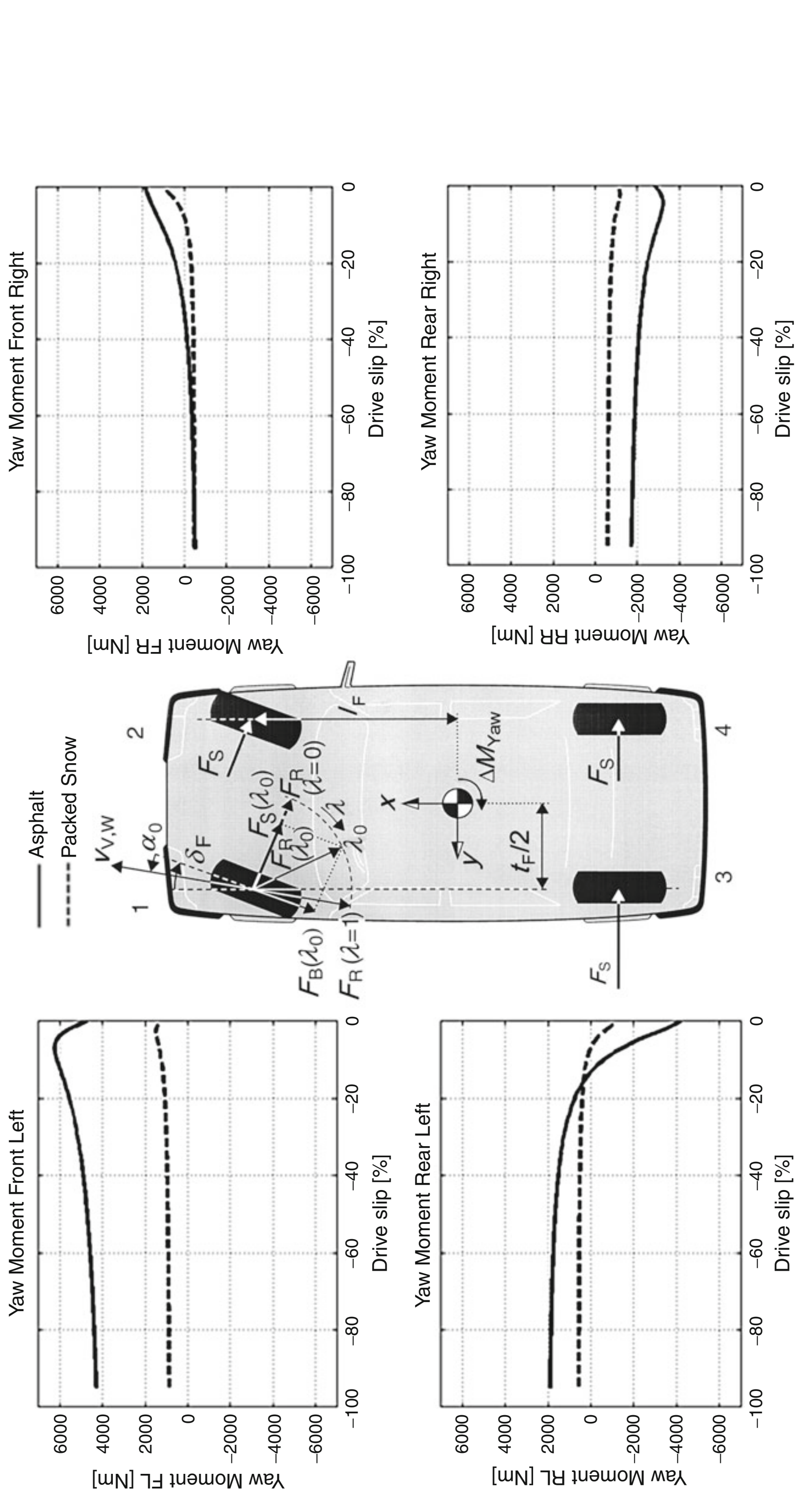

Fig. 12 Influence of wheel individual traction slip on the yaw moment on a dry asphalt road and on a packed snow road during stationary cornering with limit velocity and radius of the turn 100 m

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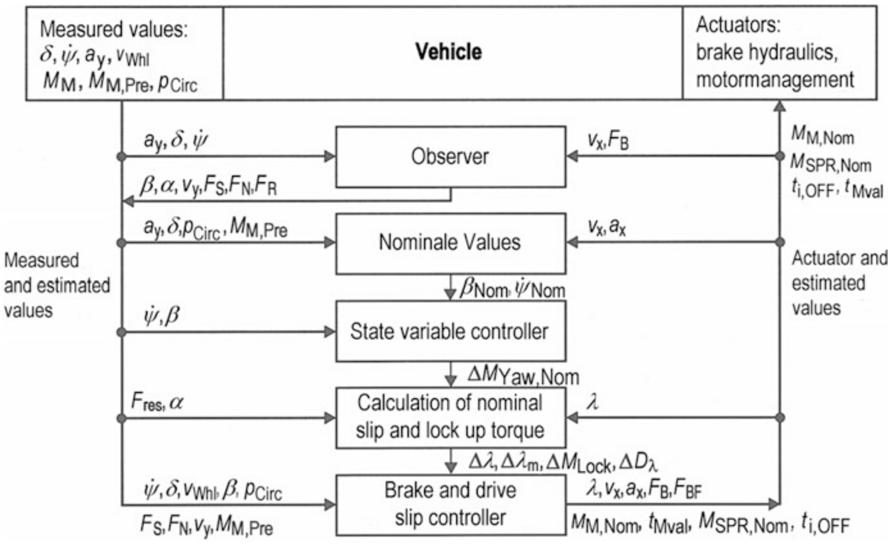

Fig. 13 Simplified block diagram of the ESC control with input and output quantities

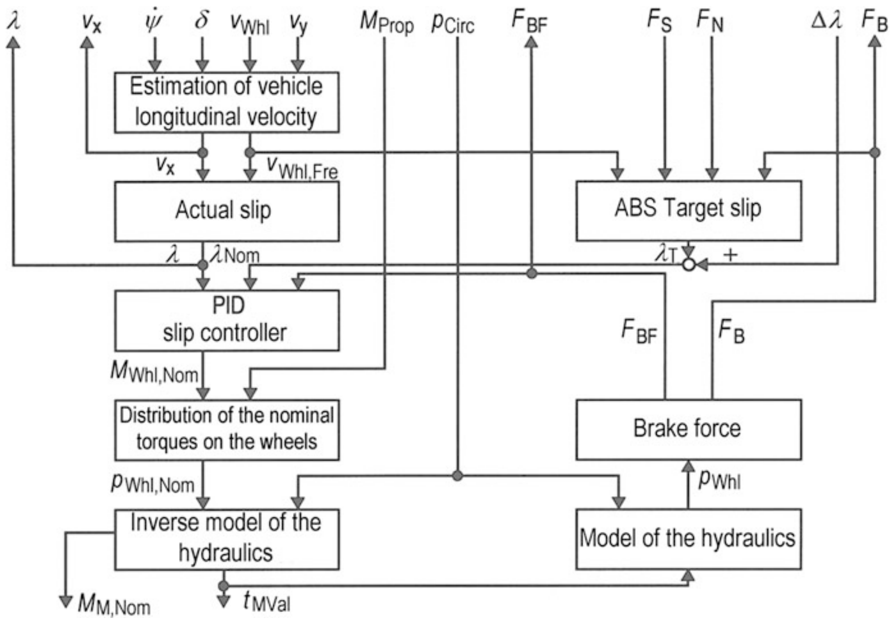

Fig. 14 Block diagram of the brake slip control with the most important modules and their input and output quantities

target slip values λ_T for maximum brake forces and maximum traction forces, respectively, dependent on the coefficient of friction of the road and of the vehicle velocity at each wheel. Each slip value λ_T is merely modified by a slip value change $\Delta\lambda$ required for the change in the yaw moment on the vehicle and results in the nominal slip value $\lambda_{Nom} = \lambda_T + \Delta\lambda$ for each wheel. Therefore, the structure of the slip controllers is independent of the yaw control, which demonstrates the hierarchical structure of the controller shown in Fig. 13. A detailed description of the control structure can be found in Isermann (2006).

The brake slip controller structure is shown in Fig. 14. The task of the brake slip controller is on the one hand the ABS function for shortest stopping distance and on the other hand the control of the wheel slip changes $\Delta\lambda$ which are demanded by the vehicle motion control. In order to combine both tasks in a unified

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manner, a novel brake slip controller was designed by which the ABS function is realized by brake slip control instead of wheel acceleration control as is used with standard ABS. For the ABS function, the brake slip controller determines the target slip, λ_T, which is the slip at the maximum of the μ-slip curve.

In order to compute the actual wheel slip, the longitudinal vehicle velocity must be known. Since this velocity is not measured, it must be estimated. The estimation procedure is described in Sect. 4.4. The nominal wheel slip is computed as described above.

$$\lambda_{\mathrm{Nom}} = \lambda_{\mathrm{T}} + \Delta\lambda \tag{7}$$

The traction slip controller is used to control the traction slip of the driven wheels during propulsion. Active slip interventions at the nondriven wheels which are required for changes in the yaw moment are controlled by the brake slip controller. The traction slip controller structure (Fig. 15) is described for a rear-wheel-drive vehicle.

Because of the differential, the transmission and the engine with the clutch the two driven wheels are mechanically coupled. This means that the equations of motion of the driven wheels are also coupled. By change of variables, the equations of motion become independent. The new variables are the mean velocity of the two driven wheels $V_{\mathrm{Prop}} = (v_{\mathrm{Whl,RL}} + v_{\mathrm{Whl,RR}})/2$ and the difference velocity of the two driven wheels $V_{\mathrm{Dif}} = v_{\mathrm{Whl,RL}} - v_{\mathrm{Whl,RR}}$ in which $v_{\mathrm{Whl,RL}}$ is the velocity of the left rear wheel and $v_{\mathrm{Whl,RR}}$ is the velocity of the right rear wheel (Isermann 2006). For this reason, the slip of each wheel is not controlled by itself, but instead the mean velocity V_{Prop} and the wheel velocity difference V_{Dif} are controlled. For the determination of the nominal mean velocity $V_{\mathrm{Prop,Nom}}$, a symmetrical nominal slip value λ_{m} is computed (where both driven wheels have the same nominal slip value λ_{m}). For the determination of the nominal value of the wheel velocity difference $V_{\mathrm{Dif,Nom}}$, an asymmetric slip value D_λ is computed, where the difference in the nominal slip values between the two driven wheels is D_λ. Using the longitudinal vehicle velocity v_{X}, the nominal values of the mean wheel velocity $V_{\mathrm{Prop,Nom}}$ and the wheel velocity difference $V_{\mathrm{Dif,Nom}}$ can be computed from the nominal symmetric slip value λ_{m} and the nominal asymmetric slip value D_λ.

The dynamics of the two equations of motion depend on the state of the controlled process. Therefore, the operating point is evaluated (which considers, e.g., the influence of the gear ratio i_{G} of the transmission) and used to adjust the controller gains. Control variables for the mean velocity are the engine torque and a symmetric brake torque of the driven wheels. The control variable for the difference velocity is the asymmetric brake torque control of the driven wheels.

As mentioned above, besides the hierarchical control structure of ESC, a modular control structure also exists (Rieth et al. 2001) where a brake slip controller is used for changes in the yaw moment, and a standard ABS controller is used for the ABS function. If ABS and the yaw moment controller work simultaneously, then the access of the controllers to the magnetic valve stimulations must be coordinated by a so-called arbitrator. Instead of the vehicle velocity v_{X}, the reference velocity v_{Ref} is used by the yaw moment controller (Fig. 7). Because of the arbitration, ABS control must be stopped once in a while. The same applies to the yaw moment control. Therefore, the control logic of ABS and of the yaw moment controller must be modified (e.g., for the determination of the integrals of ESC and for the "learning" of ABS). Also heuristic elements like the recognition of positive μ-jumps and adaptive modifications of gains must be modified. This is also to be considered during traction control.

4.4 Determination of Nominal Values and Estimation of Vehicle Dynamic Quantities

The nominal value of the vehicle slip angle β_{Nom} is defined by the requirements on ESC. The nominal value of the yaw velocity is defined by a model of the vehicle. This model is based on the linear steady-state bicycle model as described in Sect. 2.1. However, the yaw velocity of the linear steady-state bicycle

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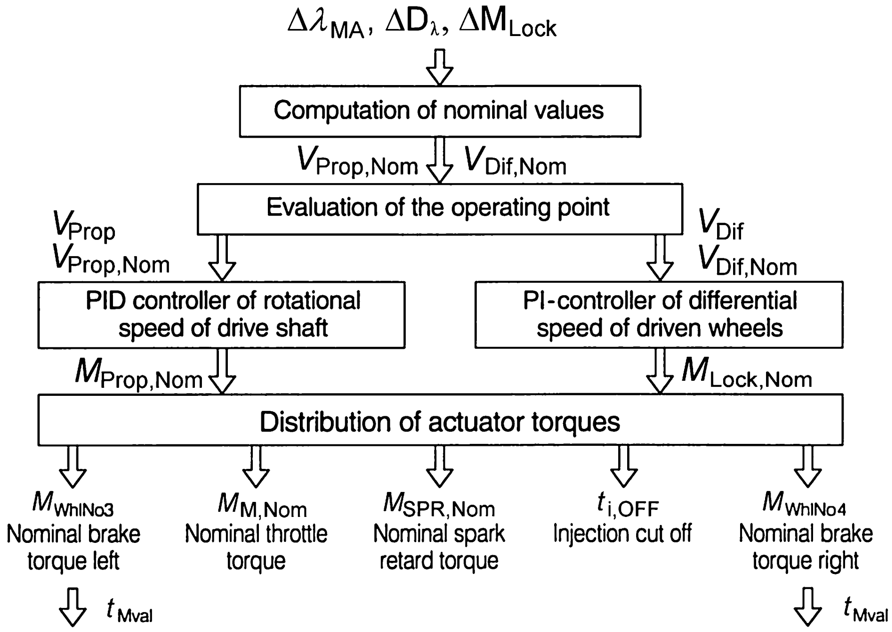

Fig. 15 Block diagram of the traction slip control with the most important modules and their input and output quantities

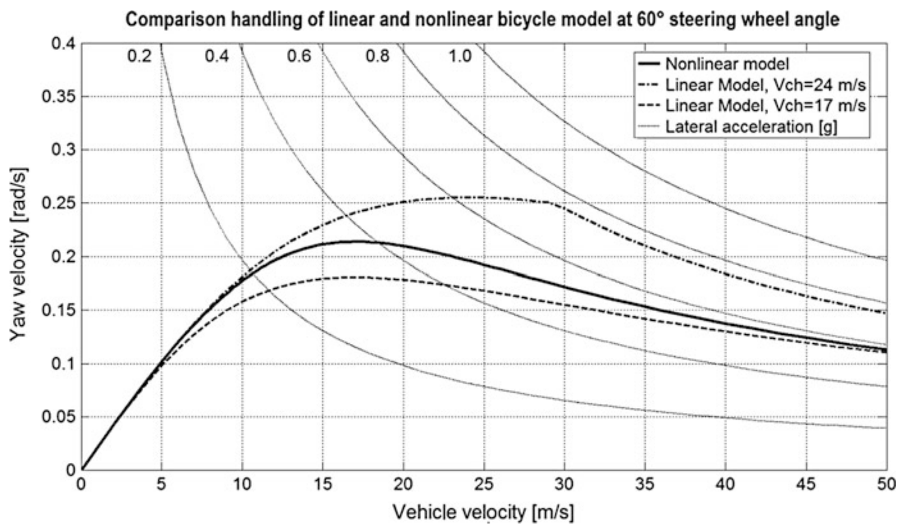

Fig. 16 Approximation of the nonlinear vehicle model by weighted interpolation between two linear bicycle models

model does not represent the yaw velocity of the vehicle with sufficient accuracy. In particular, the model must be extended to improve the accuracy of the yaw velocity if the vehicle dynamics approaches the physical limit. Otherwise, the yaw moment changes may occur late, when the vehicle is already skidding, or early, before the physical limit is reached. Expert drivers complain most if the latter situation occurs. Instead of one, linear steady-state bicycle model ESC uses two of them (Fig. 16). The difference between the two models is a different value of the characteristic velocity v_{ch}. Each of the linear steady-state bicycle models produces a yaw velocity for a certain vehicle velocity and steering wheel angle. The characteristic

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velocities are chosen such that the yaw velocity of the vehicle lays in between the yaw velocities of the two models. A weighted average is computed from these two yaw velocities to result in the nominal yaw velocity of the vehicle. The weights are chosen dependent on the lateral acceleration of the vehicle and of the driving situation.

Using a four-wheel model, the observer estimates from the measured yaw velocity, the steering wheel angle, and the lateral acceleration and from the estimated values of the braking and driving forces on the wheels, the slip angle of each wheel, the slip angle of the vehicle, and the vehicle velocity (Fig. 13). Also, the side and vertical forces on the wheels are estimated, and the resultant forces are computed. The four-wheel model also computes the transients and considers special situations like μ-split and banked roads.

On a horizontal, homogeneous road, the differential equation for the slip angle of the vehicle β is as follows:

$$\dot{\beta} = -\dot{\psi} + \frac{1}{v_{\mathrm{X}}} \cdot (a_{\mathrm{Y}} \cdot \cos\beta - a_{\mathrm{X}} \cdot \sin\beta) \tag{8}$$

in which a_{X} and a_{Y} are the longitudinal and the lateral acceleration of the vehicle, respectively. For small values of the longitudinal acceleration and of the slip angle of the vehicle, the equation can be reduced to

$$\dot{\beta} = \frac{a_{\mathrm{Y}}}{v_{\mathrm{X}}} - \dot{\psi}, \quad \beta(t) = \beta_0 + \int_{t=0}^{t} \left(\frac{a_{\mathrm{Y}}}{v_{\mathrm{X}}} - \dot{\psi} \right) \mathrm{d}t \tag{9}$$

Since the measured values of the lateral acceleration and of the yaw velocity as well as the estimated vehicle velocity include some errors, the error in the integral grows rapidly in time so that the computed slip angle of the vehicle cannot be trusted after a while.

For large values of the longitudinal acceleration of the vehicle, a Kalman filter can be used as an observer for the lateral velocity of the vehicle. The Kalman filter uses the equations of motion of the lateral velocity and the yaw velocity of a four-wheel model of the vehicle. Included in the observer is an estimate of the road inclination (Isermann 2006). Included in the observer are also simple estimates like those of the wheel normal forces which are based on the longitudinal and lateral acceleration of the vehicle.

For the control of the wheel slip, the longitudinal vehicle velocity must be known. Since this velocity is not measured for cost and other reasons like weather conditions, the velocity of the vehicle is derived from the velocities of the wheels. This is particularly difficult if all wheels show some slip like with ABS. Therefore, the derivation of the longitudinal vehicle velocity will be described for the case in which all wheels are controlled by ABS.

For the derivation of the longitudinal vehicle velocity during ABS, a single wheel is selected for which the ABS slip control to the nominal value λ_{Nom} is interrupted for some time period, called the adaptation time period or adaptation phase (Fig. 17). At the beginning of the adaptation time period, the wheel brake pressure is lowered by some amount and kept constant such that a stable wheel slip λ_{A} results. The free rolling wheel velocity is then

$$\mu_{\mathrm{A}} = \frac{F_{\mathrm{B,A}}}{F_{\mathrm{N,A}}} = c_\lambda \cdot \lambda_{\mathrm{A}} = c_\lambda \cdot \frac{v_{\mathrm{WhlFre,A}} - v_{\mathrm{Whl,A}}}{v_{\mathrm{WhlFre,A}}} \quad \rightarrow v_{\mathrm{WhlFre,A}} = v_{\mathrm{Whl,A}} \cdot \frac{c_\lambda}{c_\lambda - \dfrac{F_{\mathrm{B,A}}}{F_{\mathrm{N,A}}}} \tag{10}$$

where μ_{A} is the coefficient of friction between the tire and the road, $F_{\mathrm{B,A}}$ is the brake force, $F_{\mathrm{N,A}}$ is the normal force from the wheel on the road, c_λ is the slope of the μ-slip curve at its origin, $v_{\mathrm{WhlFre,A}}$ is the free

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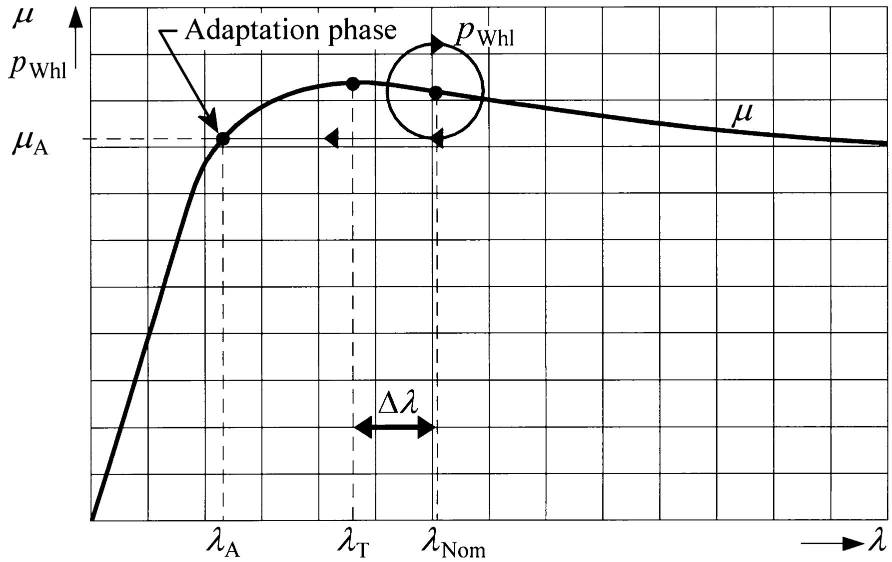

Fig. 17 Adaption phase during ABS brake slip control for the computation of the free rolling wheel velocity (the *circle* pR symbolically indicates the pressure modulation of ABS control)

rolling wheel velocity, and $v_{\mathrm{Whl,A}}$ is the measured wheel velocity, all during the adaptation time period. The free rolling wheel velocity is equal to the vehicle velocity in the direction of the wheel plane. Using the values of the yaw velocity, the steering wheel angle, the lateral velocity of the vehicle, and the geometrical data of the vehicle, this free rolling wheel velocity can be transformed to the center of mass. Using the transformed velocity in the longitudinal direction of the vehicle as the input to a Kalman filter, the longitudinal velocity v_{X} of the vehicle can be estimated. The estimated longitudinal vehicle velocity can then be transformed back to the locations of the four wheels to result in the free rolling wheel velocities of all four wheels. Thus, the actual slip values can be computed for all four wheels during ABS control. Usually, the left and right rear wheels of the vehicle are selected in an alternate manner. Then the stopping distance of the vehicle is not so much increased by the lower brake pressure during the adaptation time period, and the vehicle stability is improved.

For the Kalman filter, the differential equation of the longitudinal velocity of the vehicle is used in which the very small Coriolis acceleration due to the lateral and yaw velocity of the vehicle is neglected.

$$\dot{v}_{\mathrm{X}} = \frac{1}{m} \cdot \left\{ \left(F_{\mathrm{S,FL}} + F_{\mathrm{S,FR}}\right) \cdot \sin\delta - \left(F_{\mathrm{B,FL}} + F_{\mathrm{B,FR}}\right) \cdot \cos\delta - \left(F_{\mathrm{B,RL}} + F_{\mathrm{B,RR}}\right) \right\} - \frac{c_{\mathrm{W}} \cdot A \cdot v_{\mathrm{X}}^2 \cdot \rho}{2 \cdot m} - v_{\mathrm{X,\,offset}}. \tag{11}$$

$$\ddot{v}_{\mathrm{X,\,offset}} = 0 \tag{12}$$

in which the subscripts FL, FR, RL, and RR denote the front left wheel, the front right wheel, the rear left wheel, and the rear right wheel, respectively, c_{W} is the drag coefficient, A is the projected area of the vehicle, ρ is the air density, and the time derivative of $v_{\mathrm{X,offset}}$ is an offset in the vehicle acceleration induced by the inclination of the road. Equation 12 indicates that it is assumed that the inclination of the road changes slowly in time. The inclination of the road can then also be estimated by the Kalman filter (Isermann 2006).

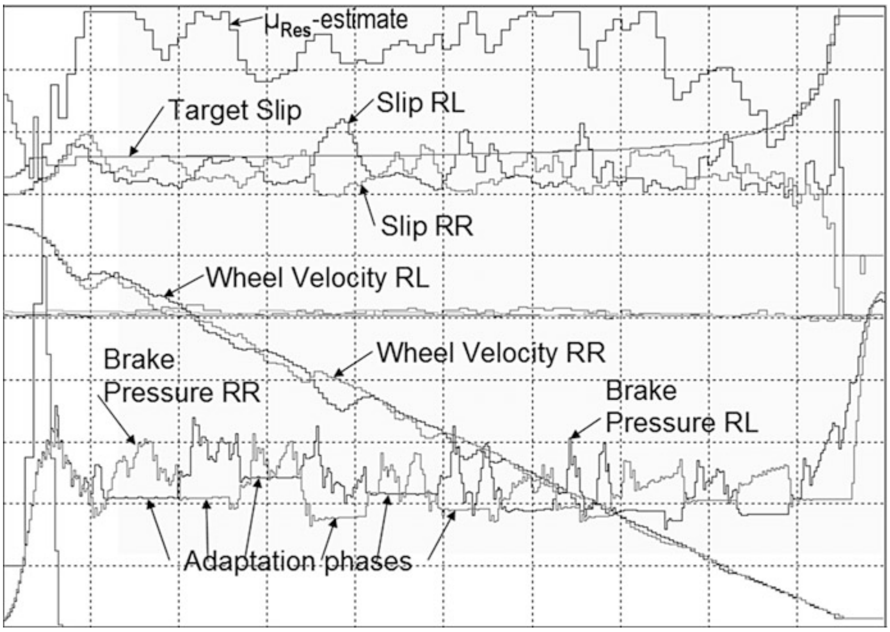

Fig. 18 Straight-line ABS braking with ESC from 120 km/h initial vehicle velocity on a smooth dry asphalt road. Plotting limits: time, 0–4.2 s; slip, −0.7 to +0.3; wheel velocity, 0 to 50 m/s; brake pressure, 0 to 250 bar

Figure 18 shows measured time histories of ESC during braking with ABS control in which the alternating adaptation time periods at the rear wheel pressures are clearly seen.

4.5 Safety Concept

ESC is a complex mechatronic system which influences the brake system which is a safety system of the vehicle. Therefore, the requirements on ESC with regard to failure rate and reliability are very high. Redundancy of components to increase the system safety and reliability must be avoided for cost reasons. Instead, components with inherent safety and component monitoring are used. Safety in this context does not mean the decrease of accidents in traffic situations by ESC but the safety of the vehicle in traffic situations in case of an ESC-component failure. Figure 19 shows the components considered in the total system driver-vehicle-ESC (Isermann 2006).

In order to increase the safety of the system, the actions of the driver are included in plausibility considerations. For instance, sudden changes in the measured value of the steering wheel angle can be compared with those a driver can maximally physically apply. The engine and the transmission are included in the system together with their controllers, since both of them are influenced by ESC. As shown in Fig. 19, the connections between the components are part of the total system and must also be monitored.

For the development of the safety concept, several methods are used. Such methods are the failure mode and effect analysis (FMEA) and fault tree analysis (FTA). A new method for monitoring and calibration of the sensors was included with the introduction of ESC. This method uses models for the monitoring of the steering wheel angle sensor, yaw rate sensor, and the lateral accelerometer and is called model-based sensor monitoring (Isermann 2006). This method is also called analytic redundancy.

Figure 20 shows the concept of the model-based sensor monitoring. The measured and during driving continuously calibrated sensor signal values of the steering wheel angle swa_{corr}, the lateral acceleration ay_{corr}, and the wheel velocities v_i are used as inputs for models whose outputs are the yaw velocity of the vehicle. Model 1 is the bicycle model of the vehicle, model 2 is a_y/v_X, and model 3 is $(v_{WhlFre,FL}\text{-}v_{WhlFre,FR})/t_F$ for the front wheels and $(v_{WhlFre,RL}\text{-}v_{WhlFre,RR})/t_R$ for the rear wheels, where t_F is

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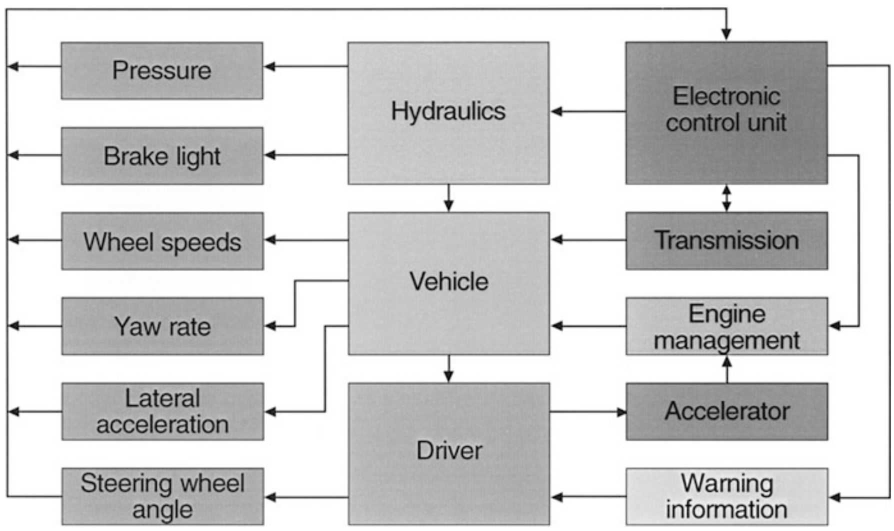

Fig. 19 Total system driver-vehicle-ESC considered for system safety

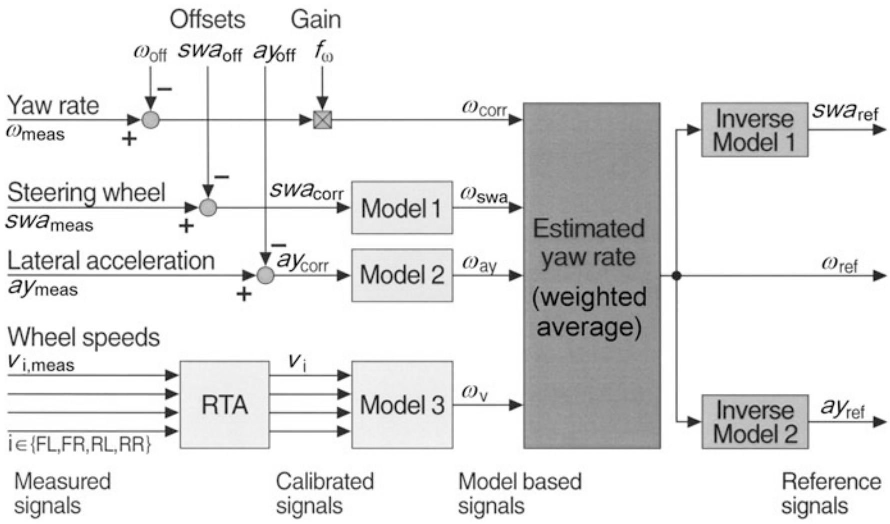

Fig. 20 Model-based monitoring of the yaw rate sensor, the steering wheel angle sensor, and the lateral accelerometer

the tread of the front axle and t_R the tread of the rear axle. Model 3 for the front axle with consideration of the steering angle is used for a rear-wheel-drive vehicle, model 3 for the rear axle is used for a front-wheel-drive vehicle, and a weighted mean value of the model outputs of the front and rear axles is used for four-wheel-drive vehicles.

Before model 3 can be used, a tire tolerance compensation (RTA) must be finished, which assures that during straight-line free rolling, all wheels show the same velocity. For the calibration of the other sensor signal values, calibration values can be found in EEPROM. The measured values of the yaw velocity, the steering wheel angle, the lateral acceleration, and the wheel velocity are ω_{meas}, swa_{meas}, ay_{meas}, and $v_{i,meas}$, respectively, where the subscript "i" denotes the different wheels. The offset values of the yaw velocity, the lateral acceleration, and the steering wheel angle are ω_{off}, ay_{off}, and swa_{off}, respectively. The sensitivity error of the yaw rate sensor is f_ω, which is stored in EEPROM together with the offset values. The subscript "corr" refers to the calibrated signal values.

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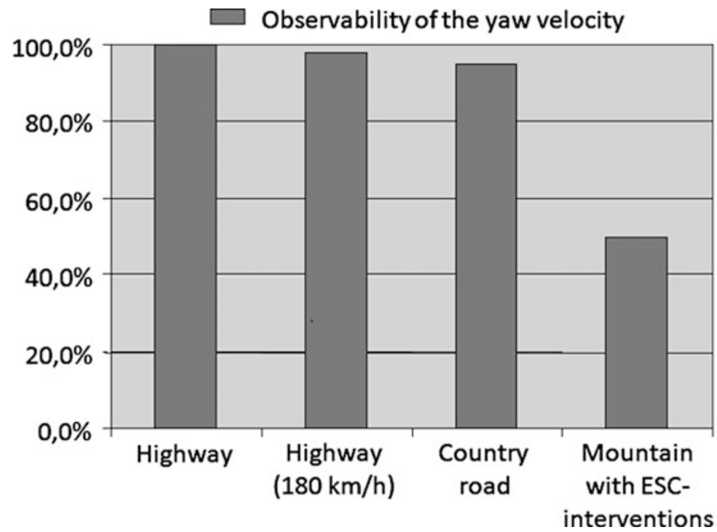

Fig. 21 Observability of the yaw rate sensor dependent on the route

The signal values ω_{swa}, ω_{ay}, and ω_{v} are estimates of the yaw velocity on the basis of the calibrated steering wheel angle sensor signal, the calibrated lateral accelerometer signal, and the compensated wheel velocity sensor signals, respectively. The computation of a weighted average of the four yaw velocity values in which the distance between the values and the distance between the time rate of change of the values are considered to compute the weight of each value results in the reference yaw velocity value ω_{ref}. This value of the yaw velocity is very reliable and delivers very good estimates for the true yaw velocity of the vehicle even during yaw moment control. From the reference yaw velocity, reference values of the steering wheel angle swa_{ref} and of the lateral acceleration ay_{ref} can be computed using an inverse model 1 and an inverse model 2 for the steering wheel angle and the lateral acceleration, respectively. These reference values can be used for monitoring the steering wheel angle sensor and the lateral accelerometer.

If the vehicle approaches the physical limit, the accuracy of the models loses precision. However, if the vehicle motion is still stable, the models compute yaw velocity values which can still be used for sensor failure monitoring. If the computed values of the yaw velocity ω_{swa}, ω_{ay}, and ω_{v} differ little, i.e., the weights of these values are large, then the signal of the yaw velocity sensor is called "observable," and sensor failure monitoring using the models can be continued. During driving, this is very often the case (Fig. 21).

The offset in the yaw velocity sensor signal is very important, because the outside of the turn is determined by the sign of the yaw velocity. As long as the offset value is not accurately determined, the dead zone of the vehicle motion controller is increased which makes the control less sensitive but which reduces the possibility of unintended ESC interventions.

Failures cannot always be detected early in time. The effect on the vehicle behavior because of some failures which are not timely detected can be limited by the following:

– **Banked turn logic** (Isermann 2006). As an example, if the lateral accelerometer signal is stuck at zero, then it might be that before the failure is detected, cornering of the vehicle is interpreted as "skidding on ice" and an ESC intervention should occur. However, this is a situation which can also occur while cornering in a banked turn. If the driver does not countersteer, then cornering of the vehicle is interpreted as cornering in a banked turn and no unintended ESC intervention occurs. After detection of the failure in the lateral accelerometer signal, ESC is switched off.

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- **Monitoring the time duration of the ESC interventions**. Since the ESC interventions are of short-time duration (usually less than 0.5 s), then a failure is suspected if the time duration of the ESC intervention occurs for a longer time period and then ESC is switched off.
- **Monitoring the time duration of ABS control**. Also, ABS control is physically limited in time duration. Therefore, a failure may be expected, and the system may be switched off if the ABS control lasts longer than a certain time period.
- **Plausibility between the signals of the brake light switch and the brake pressure sensor**. If the brake light switch is on for a longer time period although the pressure sensor signal remains zero, then the system is switched off.

5 Value-Added Functions

5.1 Special Stability Support

This group of value-added functions aims at discovering the tendency of the vehicle for instability and modifying the brake pressure, like with rollover mitigation.

5.1.1 Extended Understeer Control, EUC

If the handling of the vehicle becomes too much understeer, then ESC corrects the behavior with a slip intervention at the rear wheel on the outside of the turn. This makes the vehicle behave less understeer and the lateral forces at the rear axle increase because of the larger slip angle. Since this function is not always sufficient to keep the vehicle on track, EUC can reduce the engine torque and in addition also actively brake all wheels and thus reduce the vehicle velocity and thus also the centrifugal force on the vehicle considerably. The function starts if the driver wants to reduce the radius of the turn by more than that which is physically possible according to the coefficient of friction of the road and the momentary vehicle velocity.

5.1.2 Load Adaptive Control Mode for LCV/Vans, LAC

LAC includes an analysis of the vehicle handling (characteristic velocity v_{ch} of the bicycle model) and an estimation of the vehicle mass. The latter is based on Newton's second law of the vehicle motion in longitudinal direction during propulsion. Since the traction forces are known (estimated by the engine management system) and the vehicle acceleration is derived from the wheel velocities, the vehicle mass can be computed. With this information, some of the basic ESC functions as well as the function ROM can be adjusted and improved if the vehicle is loaded.

5.1.3 Roll Movement Intervention, RMI

Vehicles that do not tend to roll over during stationary cornering may use a reduced function ROM. For these vehicles, the stationary parts of the ROM function can be omitted, while those parts that deal with dynamic vehicle maneuvers are kept. The reduced function is called RMI and is simpler to apply than ROM.

5.1.4 Rollover Mitigation, ROM

Most vehicles do not roll over in daily traffic. Rollover is typical for vehicles that show a combination of a high center of mass and a soft suspension, as is the case with off-road and light commercial vehicles. There are two different situations in which rollover may occur:

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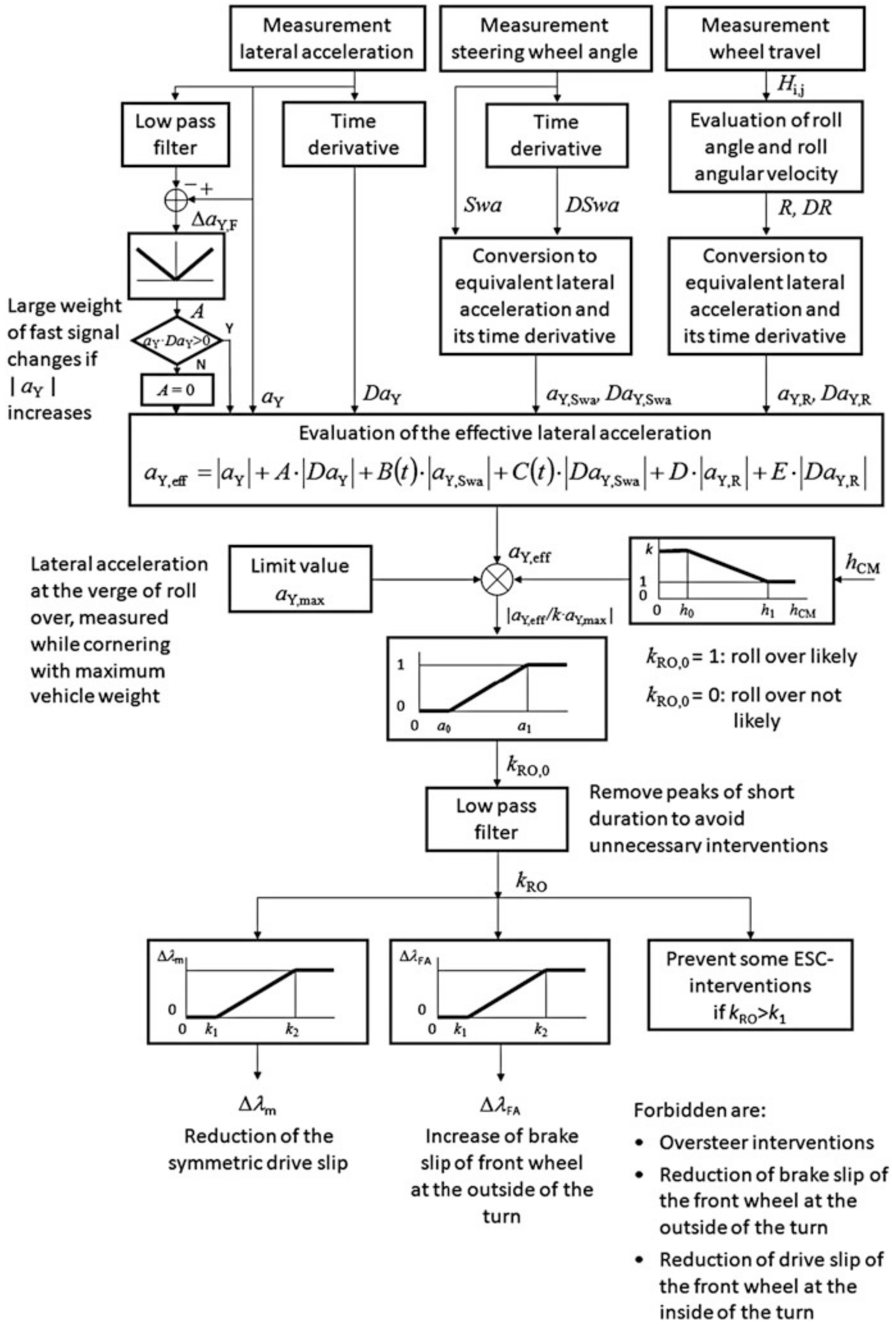

Fig. 22 Block diagram of the rollover function

– The rotation of the steering wheel is very dynamic (like with fast lane changes).
– The steering wheel angle continuously increases during limit cornering.

The key issue is the detection of an incipient rollover motion of the vehicle. Detecting rollover instability just on the basis of the lateral acceleration is not sufficient. However, by using additional signals, it is possible to reliably detect rollover critical vehicle situations. The rollover detection includes a prediction which uses the time rate of changes of the steering wheel angle and the lateral acceleration. For the detection, allowances are computed (also called “offsets”) which are added to the measured lateral acceleration to result in an “effective lateral acceleration” $a_{Y,eff}$ (Fig. 22).

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If the effective lateral acceleration exceeds some limit value, an incipient rollover is suspected, and, for instance, brake slip interventions at the front and rear wheels on the outside of the turn follow. Both the side force on the vehicle and the velocity of the vehicle are thus quickly reduced. Even if an accident still occurs, the damage will be reduced because of the reduced vehicle velocity. Since the rollover motion is fast, there is not much time left for rollover detection and intervention.

At highly dynamical situations like fast lane change maneuvers, the rollover risk is larger than for stationary maneuvers. Therefore, the time derivative of the lateral acceleration Da_{Y} and the time derivative of the steering wheel angle *DSwa* are included in the determination of the value of the rollover risk index k_{RO}. If the lateral acceleration a_{Y} is large and increasing, then the risk of rollover increases in time. In this situation, the product of a_{Y} and Da_{Y} is positive. Only if this product is positive, the time derivative of Da_{Y} is included in an offset. Fast changes of the lateral acceleration are indicated by the difference $\Delta a_{Y,F}$ between the lateral acceleration a_{Y} and its low-pass filtered value.

The steering wheel angle *Swa* and its time derivative *DSwa* are both used to generate offsets. A linear bicycle model is used in order to obtain equivalent values in the lateral acceleration $a_{Y,Swa}$ and $Da_{Y,Swa}$. These equivalent values are multiplied by parameters $B(t)$ and $C(t)$, respectively, which are time dependent such that the offsets decay in time.

Off-road vehicles sometimes have sensors with which the travel of each wheel suspension $H_{i,j}$ can be measured. Here, the subscript i denotes the front or rear axle, and the subscript j denotes the left or the right wheel on the axle. Using this information, the roll angle of the vehicle R and its time derivative DR can be evaluated. Using the roll stiffness of the vehicle, the equivalent lateral acceleration $a_{Y,R}$ and its time derivative $Da_{Y,R}$ can be evaluated. These values are multiplied by parameters D and E, respectively, and then used as offsets for the computation of the effective lateral acceleration $a_{Y,eff}$.

The effective lateral acceleration $a_{Y,eff}$ is compared with a limit value of the lateral acceleration $a_{Y,max}$ at which the vehicle is on the verge of rollover. This limit value $a_{Y,max}$ is obtained by experiment at stationary cornering with the fully laden vehicle. The vehicle is laden such that the height of its center of mass h_{CM} is maximal, h_1. If the vehicle is only partly laden, then h_{CM} will be less than h_1, and the risk of rollover will be less than that with the fully laden vehicle, and $a_{Y,max}$ can be increased by a factor k for the partly laden vehicle. The vehicle mass is estimated during driving and from this h_{CM} is estimated. Using a linear relationship, the factor k is evaluated dependent on the height of the center of mass h_{CM}.

The ratio between the effective lateral acceleration $a_{Y,eff}$ and the weighted maximum acceleration $k \cdot a_{Y,max}$ results in a first indication of the rollover risk. If this ratio is smaller than a_0, then rollover is unlikely. If this ratio is larger than a_1, then rollover is highly likely. In between those two values a_0 and a_1, the likelihood of rollover increases linearly with the ratio. Thus, a first value of the risk of rollover $k_{RO,0}$ is obtained. However, this value is truncated to result in a value between 0 and 1. Furthermore, since the signal from the accelerometer is corrupted by noise, the value $k_{RO,0}$ is filtered by a low-pass filter to finally result in the rollover risk factor k_{RO}.

The value of the risk factor k_{RO} is now used to control the slip values at the wheels. If the value of k_{RO} exceeds the value k_1, then the symmetric slip of the driven wheels λ_m is reduced proportional to the value $k_{RO}-k_1$ by the value $\Delta\lambda_m$, while the reduction is limited for values of k_{RO} larger than k_2. The reduction of the symmetric slip will reduce the engine torque. Similarly, if the value of k_{RO} exceeds the value k_1, then the brake slip of the front wheel at the outside of the turn is increased proportional to the value $k_{RO}-k_1$ by the value $\Delta\lambda_{FA}$, while the increase is limited for values of k_{RO} larger than k_2. This slip increase has an understeer effect on the vehicle. Destabilizing ESC interventions like oversteer interventions, reduction of brake slip of the front wheel at the outside of the turn, and reduction of traction slip of the front wheel at the inside of the turn are forbidden if $k_{RO} > k_1$.

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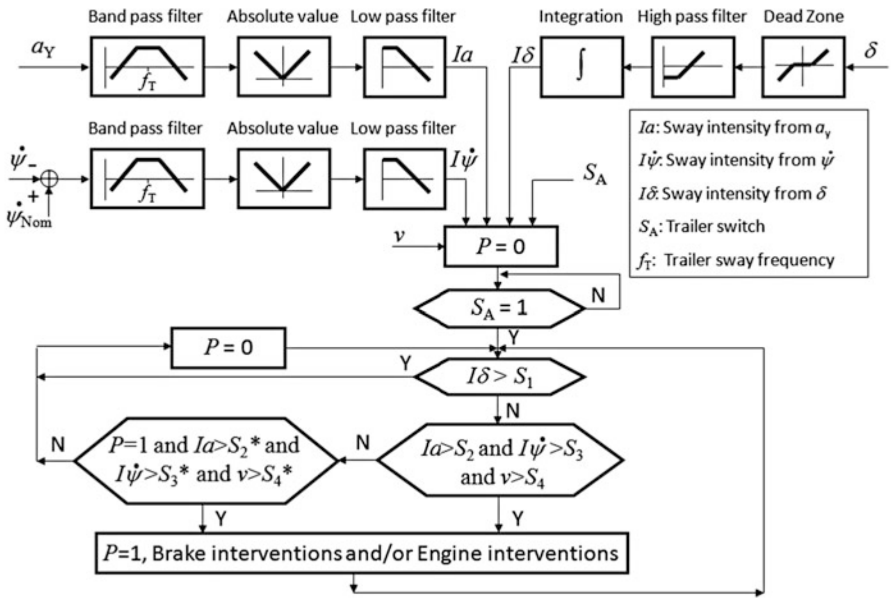

Fig. 23 Block diagram of trailer sway stabilization TSM

5.1.5 Trailer Sway Mitigation, TSM

Trailers that are towed by a vehicle may show sway oscillations if a certain critical velocity v_{crit} (e.g., 80 km/h) is exceeded. These trailer sway oscillations impose yaw moments on the towing vehicle which then shows oscillations in the yaw velocity of the same frequency as the trailer sway oscillation f_T. With increasing velocity of the train, the sway oscillation as well as the yaw oscillation becomes stronger. The yaw velocity of the vehicle may become so large that ESC interventions on the vehicle result. The detection of trailer sway oscillations is based on a frequency analysis of the yaw velocity of the vehicle. If trailer sway is detected, then the train is decelerated by active braking of the vehicle. The trailer sway oscillation decays by itself if the velocity of the train is lower than the critical velocity v_{crit}. The principle of TSM is shown by the block diagram in Fig. 23.

If trailer sway oscillations are detected, a sway indicator P is set to the value 1. At vehicle start, the sway indicator P is set to the value 0 (no sway oscillations). If the value of the trailer switch S_A is not 1, then it is presumed that there is no trailer connected to the vehicle, and further processing is stopped. Otherwise, the steering wheel angle is observed. If the driver steers violently, an oscillation in the yaw velocity of the vehicle may result which must not be interpreted as a trailer sway oscillation. This is the case if steering indicator $I\delta$ is larger than the limit value S_1. Otherwise, it is checked if the train velocity is larger than the activation limit value S_4, since at low velocities, trailer sway oscillations do not occur. If this is the case, the intensity values from the yaw velocity oscillation and of the lateral acceleration oscillation of the vehicle are checked to be larger than their activation limit values S_3 and S_2, respectively. If that applies, then the sway indicator is set to the value $P = 1$, and the engine torque is reduced in order to decelerate the train with a deceleration of -0.3 g. If the required deceleration cannot be achieved by an engine intervention alone, then all wheels are actively braked by the same brake pressure. The interventions in the engine torque and in the wheel brake pressures continue as long as the intensity values of the yaw velocity, the lateral acceleration, and the train velocity are larger than deactivation limit values S_3*, S_2*, and S_4*, respectively. If during the train deceleration of -0.3 g the intensity indicators of the yaw velocity and of the lateral acceleration increase further, then the pressure in the brake wheel cylinders is increased until the deceleration has reached the value of -0.5 g.

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The suppression of trailer sway can be improved further by a sidewise modulation of the wheel brake pressures at the axles. The resulting yaw moments on the vehicle then counteract the yaw moments from the trailer on the vehicle and thus reduce the oscillation of the yaw velocity directly. It has been shown that then the trailer sway oscillation is also reduced. It is important that the delay in the yaw moment modulation is small. Otherwise, the yaw velocity oscillation may increase instead of decrease. A delay in the yaw moment occurs if during the brake pressure modulation the pressure in the brake wheel cylinder is reduced to zero. In that case, dead times are introduced in the brake torque generation. These dead times occur, because if the pressure at a brake wheel cylinder is reduced to zero, the brake pad is retracted from the brake disk. If the pressure is increased again, the brake pad has to be moved back again to the brake disk which requires some time and which causes the delay.

5.1.6 Secondary Collision Mitigation, SCM

Vehicle crashes often result in vehicle motions which are very difficult to be handled by the driver and which may lead to a follow-up collision. On the one hand, the driver may be caught by surprise so that a definite reaction cannot be expected for a longer period of time. On the other hand, the driver may be injured and may not be able to react appropriately. A case study has shown, that in almost all crashes, except for frontal crashes, it is appropriate to fully apply the brakes. This can be realized by coordinating the airbag with ESC. ESC can then by active braking and active yaw moment generation reduce the severity of the secondary collision or even avoid it. The intervention is an automatic emergency braking maneuver with which the vehicle comes to a stop in a stable manner. The intervention occurs on the basis of the information “collision intensity” and “collision situation” from the airbag ECU. Using this information, the required vehicle deceleration level is evaluated in the airbag ECU. In the ESC ECU, the required brake pressures in the wheel brakes are computed in order to achieve this level of deceleration. If the driver applies the brake pedal, then the intervention is interrupted. However, it must then be assured that the driver does not apply the brakes unintentionally, e.g., by inertia forces of his body during the collision. The analysis of this situation is based on the time histories of the application of the brake pedal and accelerator pedal.

5.1.7 Side-Wind Assist, SWA

Sudden side-wind gusts may influence the lateral vehicle motion and lead to substantial deviations in the vehicle path. Such situations can be detected by using the sensors of ESC. All ESC sensor signals are used for the evaluation (yaw velocity, lateral acceleration, brake pressure, wheel velocities). In this evaluation, the yaw moment is computed which is required to reduce this unintentional lateral dynamics of the vehicle. The yaw moment is generated by ESC by brake slip interventions that are already described in the section on ESC. Through the reduction of the lateral dynamics of the vehicle, handling becomes easier, and the effort required from the driver to keep the vehicle in lane is reduced. Furthermore, the driver has more time for his own maneuvering tasks (Keppler et al. 2010).

5.2 Special Torque Control

This category of value-added functions supports the driver with stabilization, lane keeping, and propulsion of the vehicle and improves the handling of the vehicle.

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5.2.1 Dynamic Center Coupling Control, DCT

Four-wheel-drive vehicles with a controllable multi-disk clutch as a longitudinal lock between the driven axles offer an alternative to braking an axle. Instead of braking both spinning wheels at one axle to reduce the propulsion torque on that axle, the multi-disk clutch can be closed or opened, which is also of advantage from an energy economy point of view. Additional features of the controlled multi-disk clutch are:

– Improved performance at straight-line driving, on μ-split road surfaces, and at off-road.
– The clutch must be open during active ESC interventions (otherwise, slip interventions at one wheel may lead to slip changes at other wheels), during braking (otherwise, the electronic brake force distribution is corrupted), and during the adaptation phases for the computation of the vehicle velocity.
– If a visco-clutch is used, the unintended lock torque (which may occur in lifetime) must not be larger than 100 Nm. Otherwise, the computation of the vehicle velocity is corrupted.
– For rear-wheel-drive vehicles with optional four-wheel drive, the lock torque must be reduced if the vehicle understeers and increased if the vehicle oversteers.
– For front-wheel-drive vehicles with optional four-wheel drive, the lock torque must be increased if the vehicle understeers and reduced if the vehicle oversteers.
– For comfort reasons, the lock torque is modulated by a ramp function. The interventions are more comfortable than brake interventions. Therefore, the control of the vehicle dynamics using the multi-disk clutch can be tuned more sensitive than if the brakes were applied. Brake interventions are then less frequently required.

5.2.2 Off-Road Detection and Measures, ORD

It is well known that on loose road surfaces like gravel, the largest traction and braking forces are reached at large wheel slips, which is often the case off-road. For those roads, the stopping distance during ABS may be reduced if the target slip λ_T is increased as compared with that on normal roads (Fischer and Müller 2000). In the function, it is assumed that the road surface is loose if off-road is detected. Therefore, the off-road detection function was introduced. In the off-road detection function, the frequency spectra of the wheel velocities are analyzed. If the wheel velocities show spectral components of high frequency and if the amplitudes of these spectral components are also high, then off-road is presumed. Short-time periods with large high-frequency components in the wheel velocity must not immediately lead to off-road detection. For this reason, an off-road counter is used. The counter is increased in time if large high-frequency components exist and decreased in time if they vanish. If the counter reaches a limit activation value, then off-road is detected. For safety reasons, the target slip λ_T of the front wheels is increased from their normal values only for vehicle velocities below some limit value (e.g., 50 km/h). Sometimes, it is difficult to differentiate between ABS on scraped ice and ABS off-road. The differentiation is based on the relation between wheel slip and vehicle deceleration. If the vehicle deceleration is small but the wheel slip is large, then ice is detected, and the target slip λ_T is not increased. If the relation is not clear, the target slip λ_T is increased only at one front wheel. If the driver turns the steering wheel, the target slips are immediately reduced to the normal ones. The target wheel slips λ_T at the rear axle are not increased from their normal values.

Figure 24 shows an acceleration of a vehicle on a gravel road with a small slope with a successive full braking with ABS. During the acceleration, large oscillations occur in the wheel velocities which indicate an off-road situation. During the acceleration, the oscillations are analyzed and used in the off-road detection. After approximately one second, the off-road counter reaches the activation limit value, and off-road is detected. In this special maneuver, the relation between the wheel slip and the vehicle deceleration was not clear, so that it was not clear if the situation "scraped ice" or "off-road" applies.

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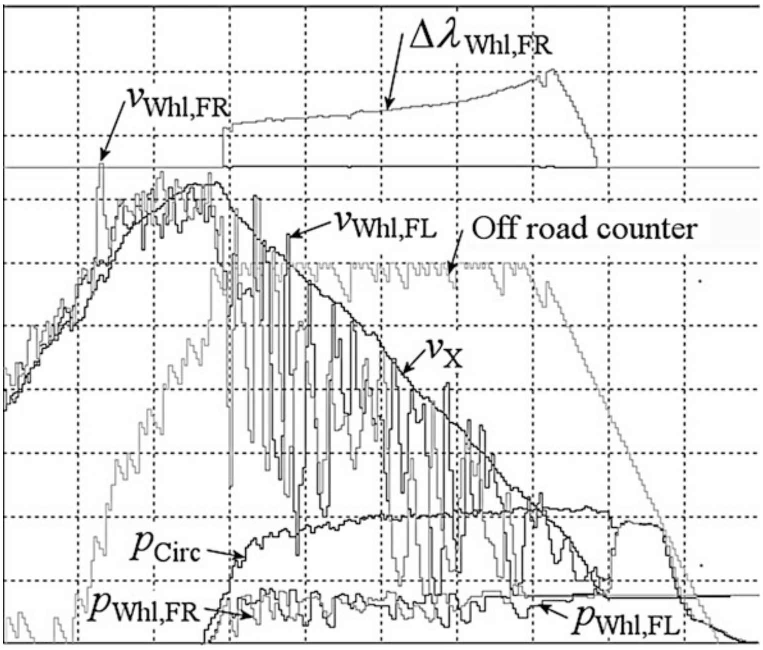

Fig. 24 Full braking off-road with off-road detection and increase of the target brake slip at the right front wheel. Plotting limits: time, 0–5 s; wheel/vehicle velocity, 0 to 10 m/s; brake pressure, 0 to 500 bar; off-road counter, 0 to 100; slip, −150 % to +50 %

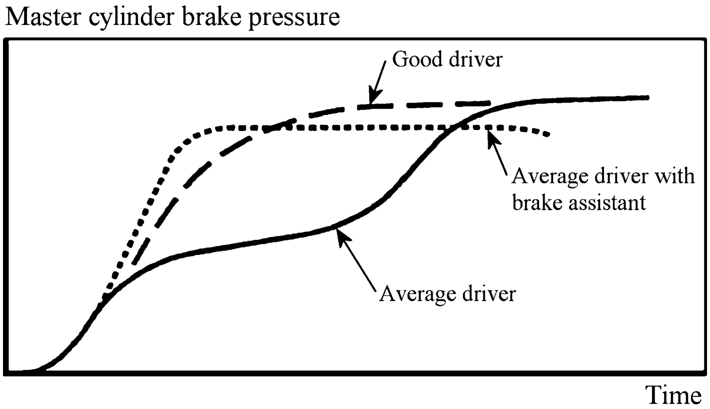

Fig. 25 Support of the driver at the start of braking by the brake assistant

Therefore, the target slip λ_T was increased only at the right front wheel and not at the left front wheel. The increase $\Delta\lambda_{\text{Whl,FR}}$ in the target slip λ_T of the right front wheel can be seen in the upper part of Fig. 24.

5.3 Brake and Boost Assist

In this category of assistance functions, the brake pressures and the brake boost functions are adjusted to the driving and system situations, for instance, with the brake assistant.

5.3.1 Hydraulic Brake Assist, HBA

Investigations with the driving simulator of Mercedes-Benz showed that normal drivers brake reluctantly in frightening situations (Fig. 25). The full application of the brake pedal by the driver after an initially fast but low brake pedal force application occurs after some delay. This delay has a large influence on the

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Table 1 HBA-Logic (See Fig. 26)

Situation	Detection logic
Phase 1 Emergency situation Panic braking	Brake pedal actuated and MC pressure gradient larger than activation limit value and MC pressure larger than activation limit value and vehicle velocity larger than activation limit
Phase 2 Reduction of brake pressure request	Pedal force (derived from MC pressure) below deactivation limit
Renewed activation	MC pressure gradient larger than activation limit
Standard braking	Brake pedal not actuated or MC pressure below deactivation limit or vehicle velocity below deactivation limit or pedal force large enough

stopping distance since at the beginning of the brake application the vehicle velocity is the highest during braking. The brake assistant (BA) overcomes the delay and thus reduces the stopping distance considerably. Key to the brake assistant is the detection of an emergency situation and the immediate increase of the brake pressure at all wheels beyond that induced by the driver up to ABS operation at all wheels.

The most important functional requirements on the brake assistant are the following (Robert Bosch GmbH 2004; Breuer and Bill 2013):

– Support of the driver in emergency braking situations and reduction of the stopping distance to those values that can usually be achieved by expert drivers only.
– Interruption of full braking as soon as the driver reduces the pedal force by a substantial amount.
– Conservation of the conventional brake boost function. Pedal feel and comfort should remain untouched during normal braking.
– Activation of the function only in real emergency situations, so that the driver does not get used to the function.
– No impairment of the conventional brake if the brake assistant function fails.

The hydraulic brake assistant uses the hydraulic unit of ESC to actively increase the brake pressure to values beyond that induced by the driver. Using the pressure sensor signal of ESC, the brake pedal application by the driver is analyzed. The detection of an emergency situation is based on the value of the pressure and on the value of its time derivative (Table 1). The HBA can be easily adapted to the properties of the vehicle and of the brake system by the choice of the activation limit values of the brake pressure and of its time derivative. The activation limit values are dynamically adjusted to the actual situation which includes the vehicle velocity, brake master cylinder pressure, state variables of the pressure control in the brake wheel cylinders, and an analysis of the time history of the braking maneuver. Also, the vehicle velocity must exceed a least velocity to activate the brake assistant function.

As soon as the activation conditions are fulfilled, the brake assistant is activated (number 1 in phase 1 in Fig. 26). Now, the brake assistant increases the brake pressure beyond the level induced by the driver at all four brake wheel cylinders up to ABS control at all wheels. The active brake pressure increases, and the brake pressure control occurs in a similar manner as with active interventions of ESC. As soon as the brake pressure passes the locking level, the brake slip controller ABS is started, and the brake slip is controlled for optimal brake forces.

If the driver releases the brake pedal and if the brake master cylinder pressure has dropped below a certain deactivation level (number 2), the system recognizes the driver's desire and can reduce the pressure in the brake wheel cylinders (Fig. 26, phase 2). From the beginning of phase 2 on the control strategy is changed. Now, the pressure in the brake wheel cylinders must follow the signal from the

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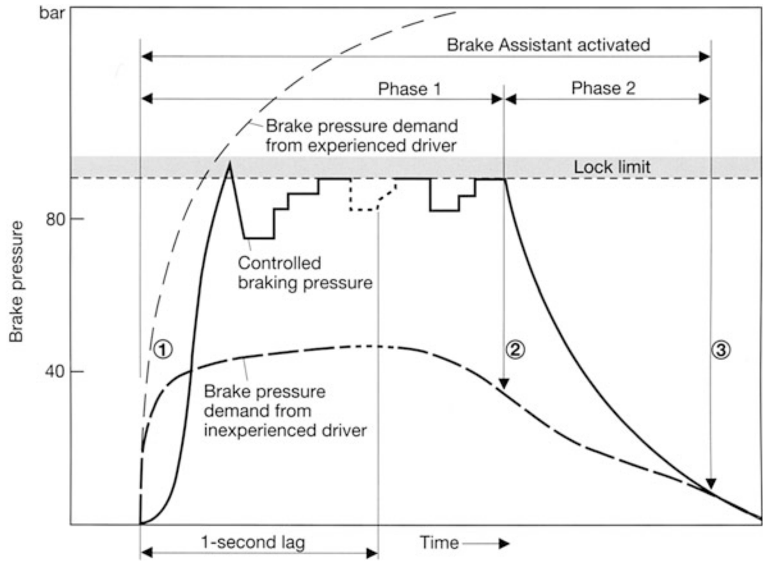

Fig. 26 Concept of the hydraulic brake assistant, HBA

pressure sensor in a smooth and comfortable transition back to the standard brake function. The brake assistant is switched off as soon as the enhanced pressure in the brake wheel cylinders reaches the value intended by the driver or falls below a deactivation limit value (number 3). The driver can now continue braking without the additional support of the brake assistant.

5.3.2 Brake Disk Wiping, BDW

Wet brake disks show a coefficient of friction between the brake lining and the brake disk which is lower than that of dry brake disks. If the brake disks are wet because it rains, then a small pressure in the brake wheel cylinders is actively generated and increased up to a very small level (approx. 1.5 bar) for a short period of time (approx. 3 s). The brake pad then removes the water film from the brake disk and thus improves the coefficient of friction between the brake lining and the brake disk. This procedure is repeated regularly (approx. every 3 min). During this procedure, the induced vehicle deceleration is so small that it is not noticed by the driver. Rain is signaled by the rain sensor. Another indication of rain is the operation of the wipers. The function is interrupted if the driver pushes the brake pedal. For this function, the hydraulic unit must be equipped accordingly, e.g., with precise control valves.

5.3.3 Electronic Brake Prefill, EBP

If the driver applies the brake pedal or if ESC actively generates pressure at the brake wheel cylinders, the brake pads move to the brake disks first before the pressure can be increased. During this time period, the brake torque at the wheels is zero. The brake force generation shows a delay from the beginning of braking (see also TSM), and the stopping distance increases with the delay. The ESC intervention is also delayed and stabilization of the vehicle may not be possible.

The situation can be improved if the brake pads are already in contact with the brake disks at the time the driver applies the brake or at the time the ESC intervention starts. Indicative for an expected fast brake pedal application is a fast release of the accelerator pedal. If this happens, a small pressure of approx. 3 bar at the brake wheel cylinders is generated actively by the ESC hydraulic unit. This pressure will already move the brake pads to the disks before the driver applies the brake.

An example of the use of this function with active ESC interventions shows the following situation. The driver steers the vehicle fast to the left, e.g., to avoid an obstacle. During this fast steering maneuver to the

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left, the pressure in the brake wheel cylinder at the front left of the vehicle is increased actively up to a small level of approx. 3 bar. This pressure value will push the brake pad at the front left wheel to the brake disk. If the driver steers the vehicle subsequently fast to the right, then it is highly likely that the vehicle behavior becomes oversteer to the right and unstable. In this situation, an ESC intervention at the left front wheel is required. Since the brake pad at the left front wheel is already pushed to the brake disk, there is no delay in the brake torque generation at that wheel brake. This improves the performance of ESC. However, if the driver does not rapidly steer to the right after the fast steering maneuver to the left, then the ESC intervention at the right front wheel is not required and may irritate the driver. Because of this ambiguity, the pressure in the front left brake wheel cylinder must be kept small and just sufficient to push the brake pad to the brake disk. For this function, the hydraulic unit must be equipped accordingly, e.g., with precise control valves.

5.3.4 Hydraulic Brake Boost, HBB

By far, the most brake actuations occur with brake pressures below 30 bar. For these situations, a small vacuum booster is sufficient to support the driver. A small vacuum booster is of advantage for packaging reasons. However, brake boosting must also insure that the driver is sufficiently supported during emergency braking with high brake pressures. In order to satisfy both requirements, the small vacuum booster is augmented with ESC boosting. ESC boosting starts if the small vacuum booster has reached saturation. Like with the hydraulic brake assistant, the ESC hydraulic unit can actively increase the brake pressure at the wheels. The harder the driver pushes the brake pedal (which is seen from the pressure sensor signal), the longer the recirculation pump is activated, the more brake fluid flows to the brake wheel cylinders, and the more the pressure in the brake wheel cylinders increases. For most brake actuations (brake pressure $<$30 bar), the function is not active, since the small booster supports the driver sufficiently. Only for very few brake actuations with large brake pressures, the function may become active, and the driver may notice some pedal feedback. Moreover, the function can compensate situations in which boosting is limited because of a low level of vacuum or even because of a complete failure of the vacuum booster.

5.3.5 Hydraulic Boost Failure Compensation, HFC

If the brake booster fails, the hydraulic unit of ESC can be used to support the driver with braking the vehicle with sufficient brake pressure. Similar to the HBB function, the pump of the ESC hydraulic unit is used to provide the brake wheel cylinders with sufficient brake fluid and to achieve the vehicle deceleration required by the driver.

5.3.6 Hydraulic Fading Compensation, HFC

During braking, the temperature at the wheel brakes increases and may reach very high values from which the brake efficiency and the vehicle deceleration suffer (brake fading). In order to keep a constant vehicle deceleration at a constant pressure in the brake master cylinder, the pressure in the brake wheel cylinders must be increased in such situations. For this pressure increase, the pump of the hydraulic unit of ESC is used. HFC supports the driver if at high brake pedal forces (evaluated from the pressure sensor signal) at which usually an ABS control results, the full vehicle deceleration is not achieved. The pump supplies brake fluid to the brake wheel cylinders continuously until the full vehicle deceleration is achieved, i.e., until ABS control at all wheels. If the pressure value in the brake master cylinder has fallen below a certain level, the function is stopped.

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5.3.7 Hydraulic Rear Wheel Boost, HRB

Normal drivers tend to keep the force on the brake pedal constant if they feel the start of ABS brake pressure modulation. Because of the stable brake force distribution, ABS modulation often starts at the front wheels first. This occurs, for instance, with straight-line braking on homogeneous roads at vehicle decelerations that are below a certain critical value (see Fig. 9). Thus, the brake force potential at the rear axle is not fully exploited although the situation requires this. Full exploitation of the brake forces at the rear axle can be obtained if the pressure at the brake rear wheel cylinders can be increased above that at the front wheels. This is possible by using the pump of the ESC hydraulic unit. HRB checks if the pressure at the brake front wheel cylinders is controlled by ABS. If that is the case but the pressure at the brake rear wheel cylinders is not controlled by ABS, then the pressure at the brake rear wheel cylinders is continuously actively increased by the pump delivery like with ESC interventions during partial braking. As a result, the rear wheel brakes will also start ABS control. The active pressure increase at the brake rear wheel cylinders is stopped if the front wheel brakes are no longer controlled by ABS or if the brake master cylinder pressure has dropped below a certain deactivation value.

5.3.8 Soft Stop, SST

At very low vehicle velocities, the coefficient of friction between the brake lining and the brake disk is larger than at higher vehicle velocities. A jerk can be felt by the driver just before the vehicle stops because of braking. This jerk can be avoided by the driver if the brake pedal is somewhat released just before the vehicle comes to a standstill. Using the control valves of ESC, this procedure can be realized without the action of the driver. Shortly before the vehicle stops, the pressure in the brake wheel cylinders is reduced from that value in the brake master cylinder induced by the driver.

5.4 Standstill and Speed Control

This category of value-added functions supports the driver at road inclination and with standing start, e.g., hill hold control and ACC stop&go. They allow the driver a comfortable ride.

5.4.1 Hill Descent Control, HDC

Off-road vehicles with engaged reduction gear can drive downhill with steep inclinations without braking, just with engine drag torque, and without a significant increase of the vehicle velocity. Off-road vehicles without this reduction gear use automatic braking to achieve the same effect (Fischer and Müller 2000). Part of the automatic braking is the function CDD-B.

HDC can be activated and deactivated by a tip switch at the dashboard. If HDC is activated and the vehicle velocity is below a certain level (e.g., 35 km/h) and the driver gives a little gas (accelerator position <20 %) and a road inclination is detected, then HDC is ready for operation. The time derivative of $v_{X,\mathrm{offset}}$ which is estimated by Eqs. 11 and 12 is used for the evaluation of the road inclination. Target for HDC is a constant vehicle velocity of 8 km/h. If the driver pushes the accelerator pedal, then HDC increases the target velocity to a higher value which, however, is limited to a maximum of 35 km/h. If the driver applies the brake, HDC reduces the target velocity to a lower value which is limited to a minimum of 6 km/h. Like with CDD-B, the brake lights are switched on during HDC operation.

HDC control is interrupted if the vehicle velocity increases above 35 km/h but continued again if the vehicle velocity drops below 35 km/h again. HDC is deactivated automatically if the vehicle velocity increases above 60 km/h.

During HDC control, the brake temperature may increase to high values which may damage the brake lining. If the temperature of the brakes at both wheels on one axle increases beyond 600 °C, the brake pressure is gradually reduced until the temperature has decreased below 500 °C. If the temperature is below 500 °C, the brakes may be activated again. The brake temperature is estimated using a thermal

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brake model. In the simulation, the heating time duration is considered as well as the cooling time period during which the brakes are not applied. Input to the simulation is the brake torque which is used to evaluate the generated thermal energy.

In off-road situations with uneven grounds, the normal forces on the wheels may vary significantly during driving. One or two wheels may even lift off from the ground. Because HDC actively applies the brakes, some wheel slips may increase very fast and start ABS control. This may induce large yaw moments on the vehicle which must be compensated by the driver through steering. In order to keep the velocity of the vehicle constant, HDC must then increase the brake pressure at the other wheels which are not yet under ABS control. However, the increase of the brake pressure at the other wheels will increase the yaw moment on the vehicle even further, and the steering task of the driver becomes increasingly difficult. The brake pressure increase may also initiate ABS control at the other wheels. However, the driver can fully concentrate on only his steering task since HDC takes over his task of keeping the vehicle velocity downhill constant.

5.4.2 Automatic Vehicle Hold with Acceleration Sensor, AVH-S

This driver assistance function is used to apply the brakes of the standing vehicle with a hold pressure so that it keeps standing and does not roll away. Using the ESC hydraulic unit, the pressure in the brake wheel cylinders is actively increased up to the hold pressure. Contrary to the function HHC-S, which can keep the hold pressure for only 2 s, the vehicle can be kept standing for several minutes without brake application by the driver. After some time, the hold function is taken over by the automatic parking brake.

For the pressure generation, the pump as well as the separation valve between the brake master cylinder and the brake circuit is stimulated (Breuer and Bill 2013). By reduction of the electrical current, the valve functions as an orifice. The brake fluid flow of the pump generates a pressure at the valve and in the brake wheel cylinders. Since the electrical current of the valve can be varied, it is also possible to vary the pressure in the brake wheel cylinders. Thus, a minimal pressure at the wheels can be set which can be variably adjusted to the longitudinal acceleration sensor signal and which stresses the ESC hydraulic unit minimally. If the pressure is sufficiently high to hold the vehicle, the electrical current of the valve is increased to its maximum value so that it closes and the pump motor can be switched off. The AVH-S function must be activated by the driver by pushing a switch or a tip switch. The brake must be released if after the stop the vehicle should accelerate. If the brake pressure is held by the ESC hydraulic unit, then the pressure is reduced by a reduced electrical current of the valve. As soon as the driver hits the accelerator pedal, the pressure in the brake wheel cylinders is reduced to a value which depends on the actual engine torque and the engaged gear.

5.4.3 Automatic Vehicle Release, AVR

This function allows the controlled reduction of the hold pressure at standstill. It is contained and described in the function AVH-S.

5.4.4 Cruise Control Basic, CCB

In the adaptive cruise control with environment sensor (ACC), the vehicle velocity is first reduced by a reduction of the engine torque. If the vehicle deceleration required by the ACC cannot be achieved by this single intervention, an active brake application is added using the hydraulic unit of ESC (see function CDD-B). For this basic function of the ACC brake pressures, up to 40 bar are required. Since the brake function must comply with high requirements on comfort, special, accurate and continuously controllable separation valves between the brake master cylinder and the brake circuits are required.

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5.4.5 Cruise Control Touch Activated, CCT

This function also uses the hydraulic unit of ESC to decelerate the vehicle smoothly. In contrast with CCB, the function CCT offers the driver the possibility to choose arbitrary acceleration and deceleration levels by using control elements at the steering wheel. In addition, it is possible to decelerate the vehicle up to a standstill and to keep it standing by, e.g., AVH-S. This function poses high requirements concerning stress and low-noise emission of the hydraulic unit of ESC.

5.4.6 Controlled Deceleration for DAS Basic, CDD-B

Many assistance functions require a definite vehicle deceleration, e.g., TSM, HDC, ACC, and the automatic partial braking in case of an expected crash. CDD-B is designed for cruise control systems and realizes vehicle decelerations of up to -3.5 m/s^2 at vehicle velocities larger than 35 km/h. Input to CDD-B is the required nominal vehicle deceleration, and output is the actual vehicle deceleration which is realized by active braking of all wheels. For active braking, the pump motor of the hydraulic unit of ESC is stimulated, and the separation valves are controlled by a variable current as described at AVH-S to achieve the required vehicle deceleration. Also, here the requirements concerning noise and comfort are very high so that high-precision separation valves must be used (Breuer and Bill 2013).

5.4.7 Controlled Deceleration for DAS Stop&Go, CDD-S

Vehicles are relatively often driven in the lower velocity range (0–30 km/h), i.e., in about 32 % of the total vehicle operation time. The traffic jam assistant helps the driver to avoid collisions in traffic jam situations in which the vehicle velocity is below 30 km/h. The system requires a short distance sensor (e.g., a radar sensor) for low velocities in order to recognize obstacles in front of the vehicle. In addition, a high-performance brake system is required in order to decelerate the vehicle at low velocities up to a standstill in a comfortable manner. If required, the vehicle will be decelerated by the traffic jam assistant up to a standstill. Like with CDD-B, the function CDD-S is used in cruise control systems to set the vehicle deceleration. CDD-S covers the complete velocity range, including stop&go. CDD-S can achieve high deceleration values up to -6 m/s^2. The vehicle can be kept standing by hydraulic means or by the mechanical parking brake. Because of the frequent operation of CDD-S, enhanced high-performance hydraulic units of ESC are required. If the vehicle in front stops, the driver can be warned by visual, by acoustical, as well as by haptical means, e.g., by AWB, in order to stimulate him to apply the brake. If the driver does not brake in time, then the system decelerates the vehicle up to a standstill.

5.4.8 Controlled Deceleration for Parking Brake, CDP

CDP can be used in vehicles that possess an electromechanical parking brake. This brake replaces the conventional hand brake lever: the cables of the parking brake are pulled by an electromotor. With the engine running, the ESC hydraulic unit takes over the task of the parking brake until the vehicle reaches a standstill and for some time period thereafter until the parking brake is activated and holds the vehicle. CDP is the interface to the ECU of the electromechanical parking brake and brakes the vehicle by active pressure generation at the wheels. All ESC functions remain available during braking.

5.4.9 Hill Hold Control with Acceleration Sensor, HHC-S

Accelerating a vehicle from a standstill on an inclined road requires a complicated coordination of releasing the brake pedal, engaging the clutch, releasing the handbrake, and actuating the accelerator pedal such that the vehicle does not roll back while the brake pedal is released. This process can be simplified to a normal acceleration on a horizontal road using the ESC hydraulic unit. The function holds the pressure in the brake wheel cylinders induced by the driver for up to 2 s after release of the brake pedal. An active increase of the brake pressure is not implied. The driver has now sufficient time to change from

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the brake pedal to the accelerator pedal. The brake pressure is reduced as soon as the acceleration procedure is detected. In order to determine the right starting time for the pressure release, the equilibrium situation of the vehicle is analyzed. The analysis uses the engine torque and the downhill force on the vehicle because of the vehicle weight. The downhill force is derived from the longitudinal acceleration sensor signal. The HHC-function is automatically activated. In order to avoid the driver leaving the vehicle while HHC is active, additional signals are monitored (e.g., clutch signal).

5.5 Advanced Driver Assistance System Support

In this category of value-added functions, the ESC interventions are modified based on sensor signals of other active and passive safety systems. For instance, the automatic warning brake helps to alert the driver in critical situations (see also chapter "▶ Development Process of Forward Collision Prevention Systems").

5.5.1 Adaptive Brake Assist, ABA

The stopping distance in an emergency situation may be shortened by early brake application. HBA helps the driver to continue fast brake application in emergency situations up to ABS control at all wheels. However, HBA starts operation only after some conditions are fulfilled, which takes some time. Moreover, because the brake pads must first be moved to the brake disks before the brake pressure can increase and the brake torque can be generated, there is some delay between brake application and vehicle deceleration. If sensors are used that scan the area around the vehicle, the emergency situation may be detected before the driver applies the brake pedal. If an emergency situation is detected this way, the conditions for activating the HBA can be reduced so that the function starts earlier. Moreover, the brake pads can be actively moved to the brake disks by the function ABP ahead of the brake application by the driver. If the driver then applies the brake, the HBA function starts earlier and the vehicle deceleration starts earlier and the stopping distance is substantially reduced. This function is also called "predictive brake assist" (PBA). If the vehicle has a brake by wire system, then the booster gain can also be increased. Even if a collision cannot be avoided, the function can reduce the severity of the accident. In an enhanced function, the signals of the sensors that scan the area around the vehicle can be used to estimate the required brake pressure in order to avoid a collision. If the driver then applies the brake, this required pressure is automatically generated immediately.

5.5.2 Automatic Brake Prefill, ABP

If an emergency situation is detected from the sensor signals that scan the area around the vehicle which can lead to a collision, then the brake pads are moved to the brake disks by a small active brake pressure. If the driver then applies the brake, the vehicle deceleration will follow without time delay. For the movement of the brake pads, the function electronic brake prefill (EBP) is used. The function is used, e.g., in the function ABA.

5.5.3 Automatic Emergency Brake, AEB

This function automatically generates full braking of the vehicle with ABS control at all wheels, even if the driver does not apply the brake on time. A prerequisite for this function is a reliable detection of the emergency situation. The function uses information from a sensor for long distance, as is used with ACC, and information from sensors that scan the near area around the vehicle (e.g., video camera). As with CDD-B, an active braking of the vehicle is initiated and continued up to vehicle standstill like with CDD-S. The pressure in the brake wheel cylinders is not increased up to a level at which a desired vehicle deceleration is reached but as fast as possible up to a level at which all wheels are ABS controlled like with HBA.

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5.5.4 Automatic Warning Brake, AWB

Various possibilities are available to attract the attention of the driver to emergency situations. The driver can be alerted by optical or acoustical signals if on the basis of sensor signals that scan the area around the vehicle potential danger is detected. Effective are haptic signals which can be felt by the driver as, for instance, a jerk on the vehicle which occurs with a fast change in the vehicle acceleration. With AWB, this jerk is obtained by a small active brake pressure impulse of approx. 10 bar. The active brake pressure impulse is generated by running the pump of the hydraulic unit of ESC. The separation valves between the brake main cylinder and the brake circuits are current controlled with an electrical current that corresponds to 10 bar. The pressure of 10 bar is kept for 250 ms at the wheels. Then the pressure is reduced by opening the separation valves, and the pump motor can be stopped.

5.6 Monitoring and Information

To this category belong functions that are based on ESC and that provide the driver with important information, e.g., the tire inflation pressure.

5.6.1 Tire Pressure Monitoring System, TPM

If the tire inflation pressure is lower than the nominal value, then the tire wear increases. At high vehicle velocities, the tires with low inflation pressure become hot and may burst because of the increased roll resistance and deformation work, in particular if the vehicle is loaded and at a high environment temperature. It is recommended that the driver checks the tire inflation pressure regularly. Often, the driver forgets to check the tire inflation pressure. An investigation in the USA showed that more than half of the vehicles in traffic drive with low tire inflation pressure. TPM monitors the tire inflation pressure during driving continuously and informs the driver if the pressure is too low. After many severe accidents in the USA which were caused by loss of tire inflation pressure, an automatic tire inflation monitoring is required on all new vehicles (cars and pickups) in the USA since 2008. The function warns the driver if the loss in inflation pressure exceeds 25 %.

In the function TPM, the tire inflation pressure is not measured directly (like with the so-called direct method, TPM-C) but deduced from the wheel velocity signals (this is called the indirect method). In TPM, the four-wheel velocities are compared during straight-line driving with constant velocity. The function performs well if only one tire loses pressure. However, it is also possible to warn the driver if all four tires or if two tires at one axle lose pressure. The detection is based on the fact that if a tire loses pressure, then the roll radius decreases and the wheel rotates faster. The velocity difference is very small, in particular with low-profile tires, and must be checked for values of 0.25 %. Therefore, a very slow filtering and mean value computation of the wheel velocities is necessary. After a change of a tire, the function has to be reset, e.g., by pushing a tip switch, and all tires must be inflated with their nominal pressure. Besides the evaluation of the wheel velocities, the frequency spectrum of the wheel velocities can be analyzed to detect a loss of inflation pressure of each tire by itself (TPM-F).

6 Outlook

Shortly after the introduction of ESC on the market in 1995, the important brake assistant was introduced. Since then, the number of assistance functions has exploded. In the beginning, the integration of ESC with other active vehicle dynamics control systems like active steering, active suspension, and active propulsion distribution was the center of attention (Isermann 2006). This development has been pushed since 2008, but the combination of the active safety system ESC with systems that are based on sensors that observe the area around the vehicle and with passive safety systems has also been pushed hard. Of key

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importance are the reliable detection of emergency situations and the safety of the integration of active systems. Safety determines the pace of progress in these fields. Therefore, it will take some years before integration and combination of the systems have been completely realized. Furthermore, it is difficult to integrate and couple systems which are produced by competitors. The exchange of highly confidential information like specifications and safety-relevant data between competitors (e.g., information on failure rates and risk numbers), which are a prerequisite for the safety of the total system, is a big challenge.

References

Breuer B, Bill KH (eds) (2013) Bremsenhandbuch. Vieweg Verlag, Wiesbaden
Burkhardt M (1993) Radschlupf-Regelsysteme. Vogel Buchverlag, Würzburg
Fischer G, Müller R (2000) Das elektronische Bremsenmanagement des BMW X5. ATZ 102(9):764–773
Isermann R (ed) (2006) Fahrdynamik-Regelung. Vieweg Verlag, Wiesbaden
Keppler D, Rau M, Ammon D, Kalkkuhl J, Suissa A, Walter M, Maack L, Hilf KD, Daesch C (2010) Realisierung einer Seitenwind-Assistenzfunktion fuer PKW. Tagungsband des 11. Braunschweiger Symposiums AAET 2010 (Automatisierungs-, Assistenzsysteme und eingebettete Systeme für Transportmittel), 10. And 11. February 2010, IST Niedersachsen e.V. Braunschweig 2010:178–199
Rieth P, Drumm S, Harnischfeger M (2001) Elektronisches Stabilitätsprogramm. Bibliothek der Technik, Bd. 223. Verlag Moderne Industrie, Landsberg/Lech
Robert Bosch GH (ed) (2004) Fahrsicherheitssysteme. Vieweg Verlag, Wiesbaden
Schindler E (2007) Fahrdynamik. Expert Verlag, Renningen
van Zanten A, Erhardt R, Pfaff G (1994) FDR – Die Fahrdynamikregelung von Bosch. ATZ96 (11):674–689