Evolution of Electronic Control Systems for Improving the Vehicle Dynamic Behavior

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Starting with ABS (Antilock Brake System) the steps towards integrated active safety systems dealing with vehicle dynamics is shown. While ABS and TCS were initially designed as open loop controllers for the lateral vehicle motion a first approach towards closed loop control of the lateral vehicle motion for active safety systems was realized by ESP (Electronic Stability Program. Estimation algorithms and model following control are used with ESP to compensate for the lack of sensors. Since 2001 ESP is available for cars with an electro hydraulic brake system. The extension of ESP in combination with active front steering is expected to enter the market in 2003.

Keywords: Safety, Handling, Observers, Yaw moment, Brakes, Steering, Suspension

1. INTRODUCTION

The loss of yaw response of the vehicle to steering inputs during full braking while the wheels are locked has lead to very early investigations (in the beginning of the 20th century) to prevent wheel lock. ABS, which can prevent this can preserve a high level of handling performance during full braking [1]. Similarly, if the driven wheels spin due to excess engine torque handling becomes difficult, particularly if the driven and steered wheels are identical. This has lead then to the introduction of traction control systems which preserve also a high level of handling performance during driving with excess engine torque. Since neither the steering angle nor the yaw moment on the car were available, even a feed forward handling control by control of the brake and engine was not possible. With the introduction of ESP these important quantities became available. Together with the measurement of the yaw velocity, a feed back control of the vehicle handling could be introduced [2], [3], [4].

The difficulty of handling of a car at the physical limit can be explained by the research contribution of Shibahata [5]. In this paper the β-method was developed to analyze the influence of the slip angle of the vehicle on its maneuverability. One important result is, that the sensitivity of the yaw moment on the vehicle w.r.t. changes in the steering angle decreases rapidly as the slip angle of the vehicle increases (Fig. 1). Large slip angles mean here values at which the μ-slip angle curve of the tire has its maximum. Therefore, on dry surfaces vehicle maneuverability is lost at vehicle slip angle values larger than approximately 15°, whereas on packed snow this value is approximately 4°. Inagaki [6] has shown the inherent instability of the vehicle motion for certain combinations of the slip angle and its velocity (Fig. 2, dark areas). For other combinations, the slip angle velocity will return to zero and lead to a stable value of the slip angle (white area). However, for increasing steering angles the stability area shrinks to become eventually zero: the vehicle motion is unstable for all combinations of slip angle and slip angle velocity.

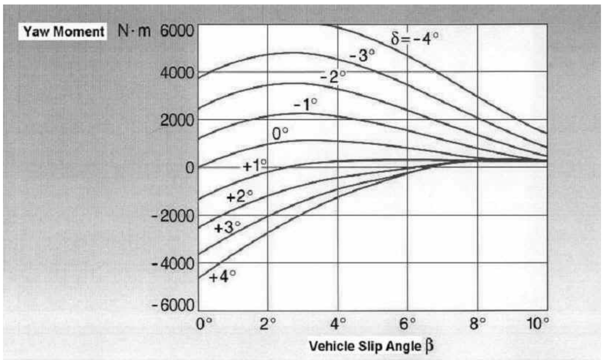

Fig. 1: Yaw moment in dependence of the slip angle for different steering angles (source: [5])

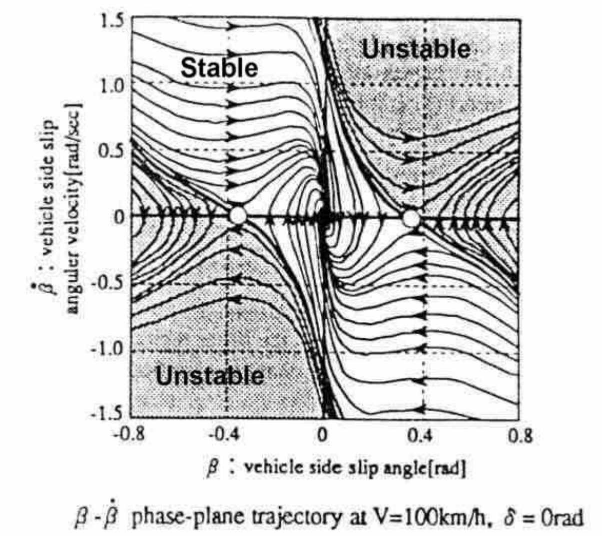

Fig. 2: Phase plane of the vehicle slip angle and its angular velocity at zero steering angle (source: [6])

Another reason for the problems normal drivers have in these situations is, that their driving experience is limited largely to driving well within the physical limit of adhesion. Förster [7] has analyzed this situation and set up some important rules. First, the driver can never recognize the coefficient of friction between the tires and the road and he has no idea of the vehicle's lateral stability margin. Second, if the limit of adhesion is reached the driver is caught by surprise and very often reacts in a wrong way and usually steers too much. This, he notes, is the real weak point in the system vehicle-driver-environment. Third, in traffic situations the need for the driver to act thoughtfully has to be minimized. Förster therefore comes to the conclusion that the concept of the vehicle including the tires and the suspension should very strongly account for the normal human behavior. Deviations from normal vehicle behavior that are inherent to the vehicle design must be controlled and reduced to negligible differences. Unexpected vehicle motions may lead to panic reactions of normal drivers. The design of vehicles should therefore center around the normal driver. Professional testers, test engineers and endurance testers are not at all typical for the real population of normal drivers, and therefore it is not unrealistic that they judge vehicle behavior according to criteria that are not relevant to the large number of average drivers.

Accidents are often said to be in 90% of all cases the result of driver errors. Käppler [8] however, notes that these statements must be taken very carefully since they originate from the police jargon. According to Brown [9] drivers are only in 19% of all cases responsible for the accidents. Vehicles are in 31% and the environment in 50% of all cases responsible for the accidents. Rompe et al [10] investigated the activities of drivers in critical driving situations just before the accidents happened. He found that steering was most often (50%) involved. Similarly Edwards et al [11] found that evasive maneuvers took place just ahead of 48% of all accidents, 50% just ahead of all collisions and 64% just ahead of all accidents in which the vehicle left the road.

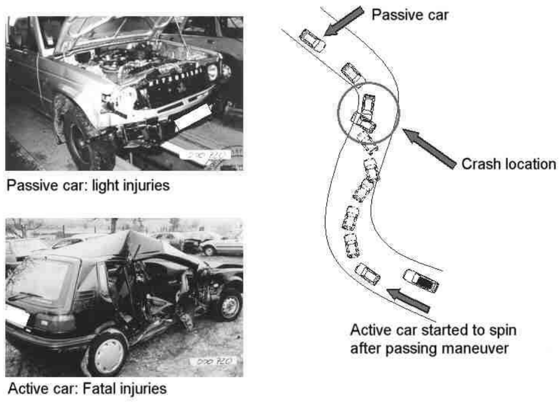

Fig. 3: Typical severe accident resulting from car spin (source: [12])

Statistics from the German Association of Insurance Companies (GdV) [12] show that severe accidents typically involve spinning cars (Fig. 3). Furthermore, while the number of people killed decreases continuously, the total number of people injured and killed remained approximately constant during an 8-year period (Fig. 4).

Contrary to the common belief that spinning of cars mainly occurs on slippery roads and at high speeds the statistics show, that by far most severe accidents occur on dry roads and at speeds between 60 km/h and 100 km/h.

A first approach to solve the problem of handling at the physical limit is given by van Zanten in [13]. Here the brake slip distribution is investigated during full braking while cornering which results in a reduced deviation of the vehicle motion from a desired nominal motion and simultaneously in a minimum stopping distance. Optimal control theory was used to get an idea of the best tuning of the brake control. It is shown, that given the momentary slip angle of a tire the brake slip of that tire must not necessarily be optimized to get the largest possible brake force.

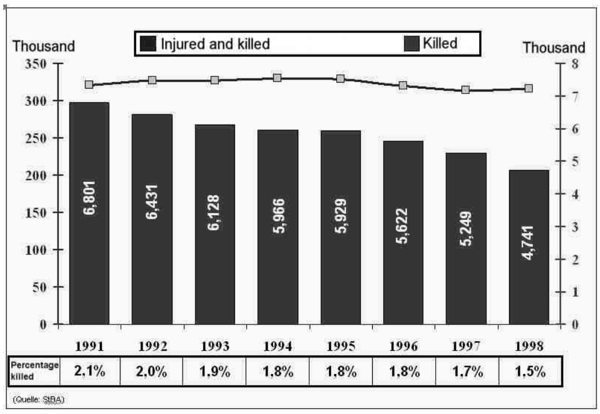

Fig. 4: Number of people injured and killed in Germany (source: [12])

An industrial approach is given by Heess [14]. He describes ways in which available control systems like Antilock Brake Systems (ABS) and traction control systems (ASR), suspension control systems and steering control systems can be used as subsystems for a superimposed vehicle dynamics control system. His suggestion applied to a full four wheel ABS/ASR led to the development of the Vehicle Dynamics Control system ESP of Bosch which controls the motion of the vehicle not only during full braking but in all situations like partial braking, coasting, acceleration and engine drag on the driven wheels. Its area of operation is therefore extended well beyond that of ABS (full braking control) and TCS (traction control).

2. ANTILOCK BRAKE SYSTEM

Locked wheels generate forces on the car which are in a direction opposite to the lineal wheel motion. Changing the steering angle has virtually no effect on the force vectors on the wheels. If the brake pressure induced by the driver is such that the wheels lock, then the brake pressure must be reduced to regain steerability.

For this task ABS uses a hydraulic unit which has electromagnetic valves to keep the pressure in the wheel brakes below the level induced by the driver

The main task of the control algorithm is to keep a high level of braking force while at the same time keeping a sufficient level of lateral force generation by steering to preserve a high level of handling performance. This information is not readily available and thus ABS relies on assumptions about the shape of the μ-slip curve and the wheel behavior during braking and cornering (Fig. 5).

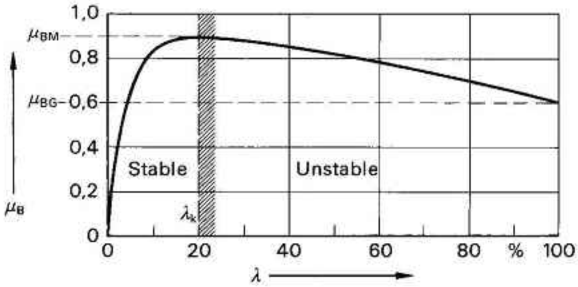

Fig. 5: Typical μ-slip curve

The question how much the wheel brake pressures should be reduced is solved by the control algorithm which monitors the wheel speeds but not the brake pressures (Fig. 6). If a wheel decelerates too fast (Phase 2), then its brake pressure is reduced and if because of the pressure reduction the wheel accelerates again then the pressure is increased again. The increase will be done in a stepwise manner in order to reduce the influence of the transients in the wheel behavior (phase 7) and the pressure may be reduced immediately if the deceleration becomes large (phase 8). Thus the average slip value is kept close to λ_k (Fig 5).

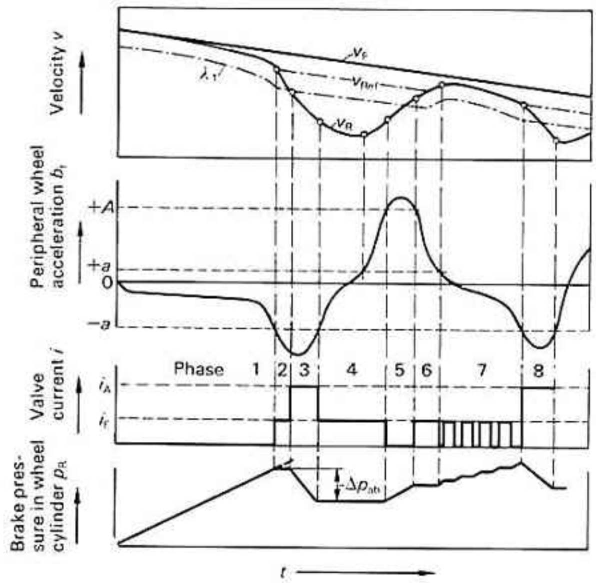

Fig. 6: ABS control concept

3. TRACTION CONTROL SYSTEM

As a first approach to control the driving forces on the driven wheels one may try to control the tire slip to the value λ_k of the μ-slip curve (Fig. 5) by the same concept chosen for ABS. This however fails, since not only the rotating inertia of the driven wheels is too large for a significant change in the wheel acceleration (because of the engaged engine and transmission) but also the engine torque is nonlinearly dependent on the engine speed, and thus also on the wheel speed. Therefore during traction control it is not possible to clearly differentiate between the stable and the unstable region of the μ-slip curve.

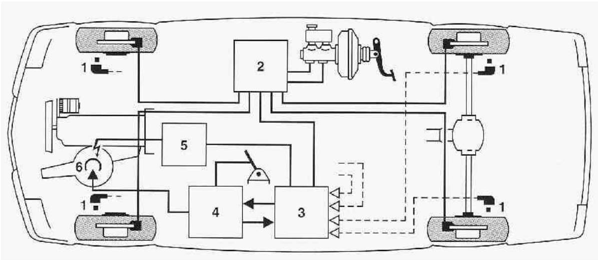

Fig. 7: Fundamental blocks of the traction control system. 1: Wheel Speed Sensor, 2: Hydraulic Unit, 3: TCS-ECU, 4: Throttle-ECU, 5 Engine-ECU

Fortunately, for single axle driven cars, the speeds of the non driven wheels may be used to compute the free rolling speed of the driven wheels. Therefore slip control of the driven wheels becomes now possible. The only unknown variable then is the value of λ_k which may vary with the road surface and the tire type and state. Traction control uses a mean value of λ_k and increases it for low car speeds. For four wheel driven cars, the determination of the free rolling speed is not easily possible and only estimates can be made which depend on the distribution of the engine torque between the front and the rear axle.

Drive slip can be reduced by reducing the engine torque or additionally by applying the brakes at the driven wheels (Fig. 7). By closing the throttle valve, reducing the spark advance or inhibiting fuel injection the engine torque may be reduced. Because of engine and exhaust emission regulations, all three interventions are not always and at the same time available. Furthermore, the time constants of these three different interventions are mutually different and vary with the state of the engine (cold start, low ambient temperature, state of the catalyst etc.). Similarly the time constant of the brake pressure varies, in particular with the temperature of the hydraulic unit.

4. ELECTRONIC STABILITY PROGRAM

Feedback control of the vehicle motion is possible by extending the traction control system with four additional sensors: steering wheel angle, brake pressure, yaw rate and lateral acceleration.

Since the nominal trajectory desired by the driver is unknown, the driver's inputs are taken to obtain nominal state variables that describe the intended vehicle motion instead. These inputs are the steering wheel angle, the engine drive torque as derived from the accelerator pedal position and the brake pressure.

The handling performance of the car can be improved if in dependence of the steering wheel angle the yaw moment on the car can be controlled. The main task of ESP as an active safety system is, however, to limit the slip angle of the vehicle β in order to prevent vehicle spin.

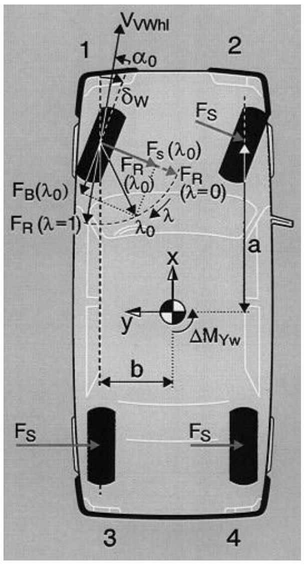

Fig. 8: Yaw moment change by slip control

ESP can control the yaw moment on the car by controlling the value of the slip at each wheel. This can be shown by the influence of some brake slip value λ_0 at the left front tire of a free rolling car in a right turn (Fig. 8). $F_R(\lambda=0)$ is the lateral force on the free rolling tire. Because of the brake slip λ_0 the lateral force will be reduced to $F_S(\lambda_0)$ where it is assumed, that neither the normal force F_N nor the tire slip angle α_0 are changed. As a result of the brake slip the brake force $F_B(\lambda_0)$ is generated. $F_R(\lambda_0)$ is the resultant force on the tire, which is the vectorial sum of $F_S(\lambda_0)$ and $F_B(\lambda_0)$. If the tire friction limit is reached, the magnitudes of $F_R(\lambda=0)$ and $F_R(\lambda_0)$ are approximately equal.

The influence of brake slip λ is now obvious: a change in the brake slip value results in a rotation of the resultant force on the tire. As a result of the rotation the yaw moment on the car is changed. However, simultaneously the lateral force and the longitudinal force on the car are influenced. The control concept determines by what amount the slip at each tire shall be changed to generate the required change in the yaw moment. Usually it is required that the driver must not have the impression that with ESP the car is slower than without ESP.

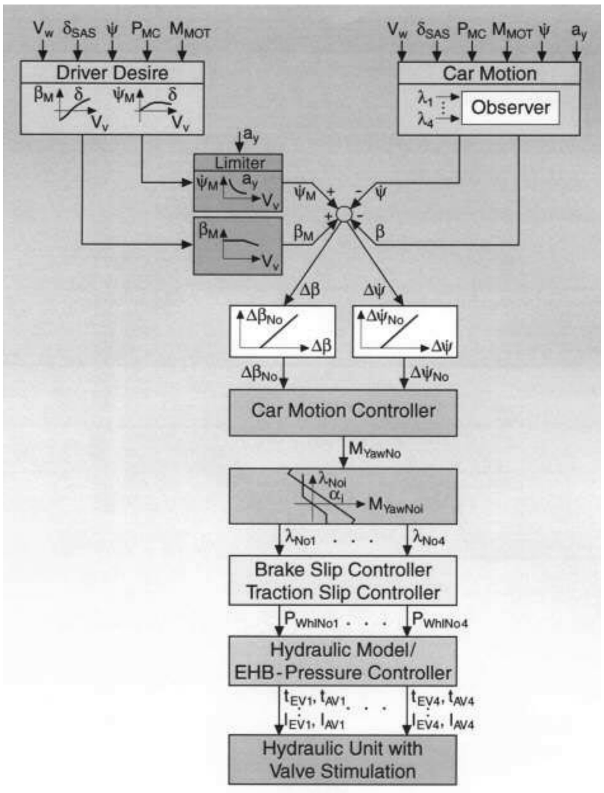

Fig. 9: Simplified block diagram of the ESP control

The vehicle dynamics controller part of ESP (Fig. 9) constitutes the upper part of a hierarchical control. Output are the nominal tire slips λ_{Noi}. In the lower part the slip values of the tires are controlled. The vehicle dynamics controller part consists of several processing blocks. On the top left the motion desired by the driver is derived from his inputs by a linear bicycle model (which uses a linear relationship between the slip angle and the lateral force of the tire). On the top right the motion of the car is measured and missing state variables are estimated.

As a first approach to estimate the slip angle of the car the derivative of the slip angle is used:

$$\dot{\beta} = -\dot{\psi} + \frac{1}{v_v}\left(a_y \cdot \cos\beta - a_x \cdot \sin\beta\right)$$

This estimate is valid if the pitch and roll angles of the car are neglected and furthermore, if the car moves on a horizontal plane. In this equation a_y is the lateral acceleration of the car and a_x is its longitudinal acceleration, v_v is its velocity and $\dot{\psi}$ is its yaw velocity. If the car velocity is constant and its slip angle is small then the estimate can be readily obtained by a simple time integration

$$\beta(t) = \beta_0 + \int_0^t \dot{\beta} dt = \beta_0 + \int_0^t \left\{ \frac{a_y}{v_v} - \dot{\psi} \right\} dt$$

Offset and other errors in the sensor and estimated signals may quickly lead to large errors in the estimate. Furthermore, during full braking the car deceleration can not be neglected. Therefore, during full braking an alternative estimate of the slip angle based on an observer is used.

The observer is based on a full four wheel model of the car and uses two dynamic equations, one for the yaw velocity and the other for the lateral velocity of the car ([15]). These equations are rearranged and discretized to be used as the model for a Kalman filter. Since the yaw velocity is measured, the solution of the differential equation of the yaw velocity is used to derive the measurement equation.

For these equations the longitudinal force F_B at any wheel is required and can be estimated by the following generic equation

$$F_B = c_p \cdot \frac{p_{whl}}{R} - \frac{M_{CaHalf}}{R} + \frac{J_{whl}}{R^2} \cdot \frac{d}{dt} v_{whl}$$

Here c_p denotes a known brake constant, p_{whl} denotes the brake fluid pressure in the brake wheel cylinder, R denotes the known tire radius, M_{CaHalf} denotes half of the engine torque at the axle, J_{whl} denotes the known moment of inertia of the wheel about its axis of rotation and v_{whl} denotes the wheel speed which is the product of the wheel angular velocity and the tire radius. The engine torque value can be obtained from the engine management system, while the rotational wheel velocity is measured by the wheel speed sensor. Finally by modeling the hydraulic unit the wheel brake pressure is estimated at each wheel.

The side forces are not readily available. Therefore a tire model is used. Specifically, the HSRI tire model as described in [15] is used which allows for a simple relation between the lateral and the longitudinal force.

$$F_S = \frac{C_\alpha \cdot \tan\alpha}{C_\lambda \cdot \lambda} \cdot F_B$$

In these equations, C_λ and C_α are the slip and cornering stiffness of the tire and λ and α are the tire slip and slip angle respectively.

The estimate of the lateral velocity by the Kalman filter is robust to tire changes as only the ratio of the lateral and longitudinal tire stiffness is used. For winter tires the ratio is nearly the same as for summer tires. The same is true for new and worn tires, conventional and wide tires etc. Thus both evaluations of the slip angle are more or less insensitive to changes in the tire properties.

Unfortunately the vehicle slip angle estimation is not always sufficiently accurate and the confidence level of its value is sometimes low. Therefore, the vehicle dynamics controller uses additionally a model following control for the yaw velocity of the car, for which the already mentioned linear bicycle model is taken. Output of the linear bicycle model is the nominal value of the yaw rate $\dot{\psi}_{No}$. Thus a first value for the nominal yaw velocity $\dot{\psi}_{No}$ is obtained (Fig. 10).

$$\dot{\psi}_{No} = \frac{v_x \cdot \delta_w}{l \cdot \left[1 + \left(\frac{v_x}{v_{ch}} \right)^2 \right]}$$

The wheel base l is a simple geometric parameter while the vehicle forward velocity v_x is estimated by the brake slip controller.

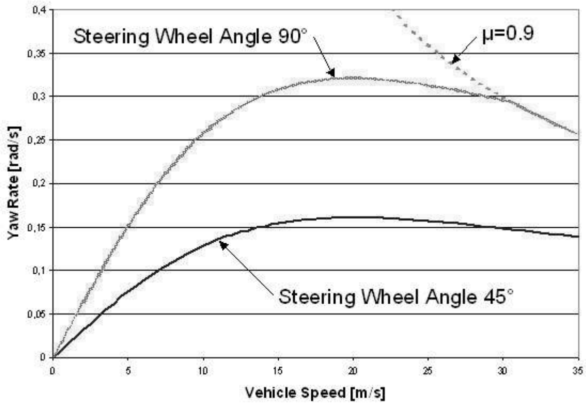

Fig. 10: Nominal yaw velocity from the linear bicycle model

The characteristic speed v_{ch} depends mainly on the lateral tire stiffness C_α of the tires. Therefore, the nominal yaw velocity changes with the tire type, make and state (new or worn). This change may occur suddenly if new tires are mounted. The model following control is thus sensitive to changes in the tire stiffness and ESP may suddenly change its behavior. This will be shown below. ESP must therefore be checked to correctly perform with all released tires.

Since the lateral acceleration of the car can not exceed the maximum coefficient of friction between the tire and the road μ, the nominal yaw velocity must be limited to a second value by the following relation (see the hyperbola in Fig. 10)

$$\left| \dot{\psi}_{No} \right| \leq \left| \mu \cdot g / v_v \right|$$

For summer tires the nominal yaw velocity is different from that of winter tires (Fig. 11). Similarly, for worn tires the yaw velocity is different from that of new tires. The vehicle becomes oversteer if on the front axle worn and on the rear axle new tires are mounted (Fig. 12). In such cases the vehicle

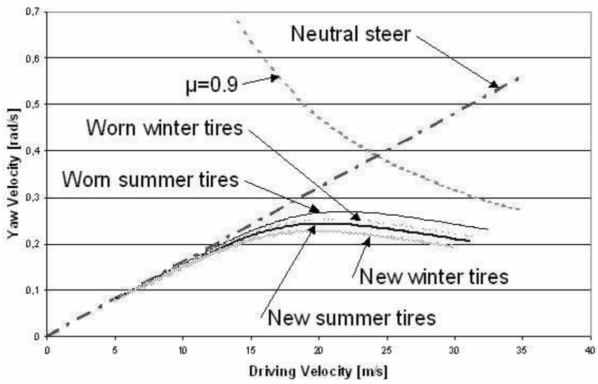

Fig. 11: Nominal yaw velocity from the full four wheel model with nonlinear new and worn summer and winter tires (steering wheel angle 60°)

behavior deviates significantly from the behavior of the linear bicycle model (Fig. 10) and ESP interventions can be expected for vehicle maneuvers which are well within the physical limit.

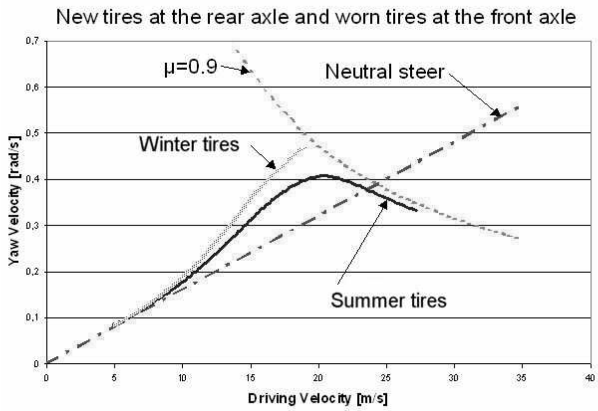

Fig. 12: Nominal yaw velocity from the full four wheel model with nonlinear summer and winter tires, with worn tires at the front axle and new tires at the rear axle (steering wheel angle 60°)

A first nominal limit value for the slip angle of the car (Fig. 9) is chosen as discussed using the Beta method in dependence of the coefficient of friction between the tires and the road. This value is reduced in dependence of the velocity of the car to a second value β_{No}, in order to improve the support for the driver at higher speeds.

If the state of the car as described by its yaw velocity and its slip angle differs from its nominal state, then the vehicle dynamics controller checks if this difference is within some tolerable dead zone. If not, a yaw moment is generated to reduce this difference to within this tolerable dead zone. Slip is controlled by the brake slip and traction slip controllers as described in [15].

5. ELECTROHYDRAULIC BRAKE SYSTEM

The Electro Hydraulic Brake System (EHB, also called SBC: Sensotronic Brake Control) is a brake by wire system, where the brake pressure in the wheel brakes are controlled in accordance with the brake master cylinder pressure using proportional valves.

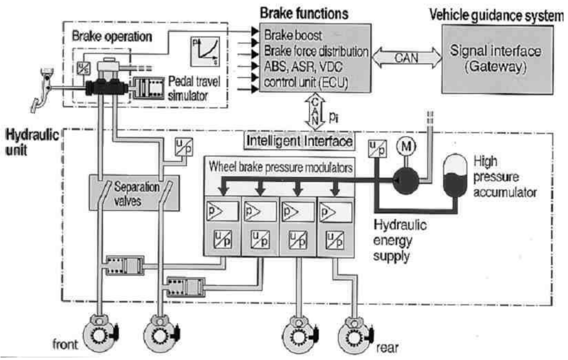

Fig. 13: Block diagram of the Electro Hydraulic Brake System

Fig. 13 shows the concept of EHB. ESP (here called VDC) is housed in the block named “Brake functions”. Essential for the control of the vehicle dynamics is, that the wheel brake cylinder pressures are measured. Furthermore, pressure can be very rapidly increased resulting in very fast active brake slip interventions.

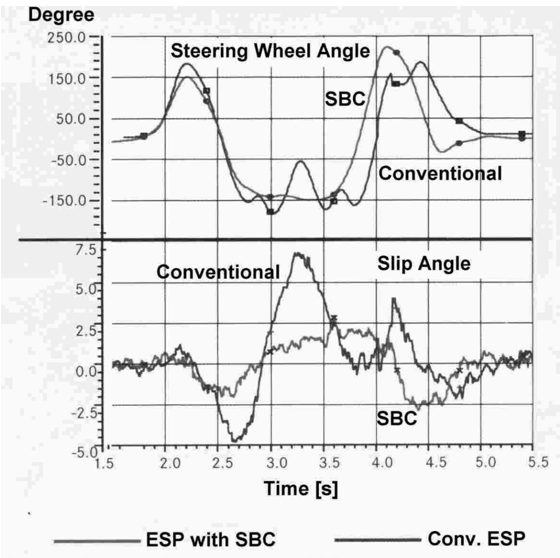

Fig. 14: Improvement of slip angle control of ESP during a double lane change maneuver by SBC (Source: [16])

Since the estimation of the brake pressure at the wheels is no longer required, the confidence level of the brake force estimation is higher than before. This improves the estimation and control of the vehicle slip angle as can be seen from the results of a double lane change maneuver (Fig. 14). Early interventions are possible so that they are less vicious and more comfortable.

Not only during maneuvers at the physical limit, but also during all brake maneuvers EHB improves the handling performance of the car. EHB allows for complete flexibility in choosing the brake force distribution in any situation, so that the brake force distribution can be chosen in dependence of the braking situation as a feed forward control. Thus almost neutral handling performance can be obtained during any combined braking and steering maneuver. Thus ESP is extended to control the vehicle handling well within the physical limit (Fig. 15).

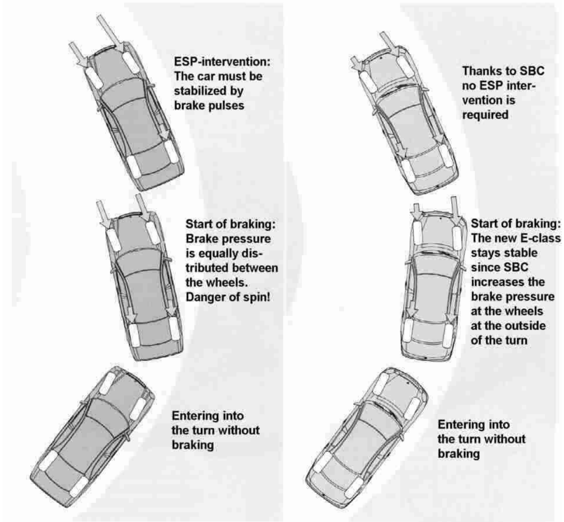

Fig. 15: Improvement of cornering stability by improved brake force distribution (Source: [16])

Although the ESP control performance is improved, limits on the improvement are set by the safety concept. It is not possible to immediately react on rapid changes in the sensor signals since the possibility of signal failures can not be disregarded. Internal tests like that of the yaw rate sensor triggered by the ECU introduce delays in the signal analysis like the estimation of its time derivative (Fig. 16). Thus the limits in the reliability of the signals limit the performance of the EHB.

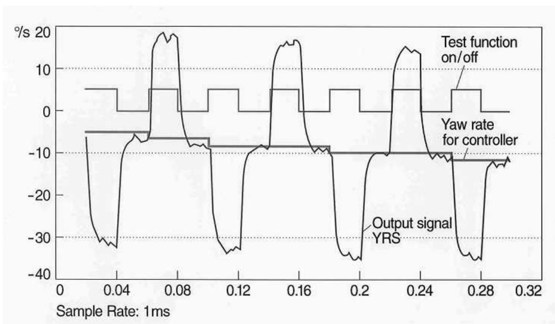

Fig. 16: Yaw rate signal with superimposed test signal

To partially compensate this drawback a new concept “suspicion of failure” was introduced. Only if a signal is suspicious of a possible failure the intervention of EHB is delayed by increasing the dead zone in the yaw rate (Fig. 9) and by reducing the pressure gradient. The general rule for the safety concept is to better not intervene where it would be beneficial for the driver then to surprise him with an intervention resulting from a corrupted sensor signal.

6. FUTURE SYSTEMS

If an active steering system is available, ESP can also use this system to control the handling behavior of the car, not only in limit situations but also during normal driving. Fig. 17 shows the concept of a first realization of such a system with which ESP can modify the steering angle [17]. Another system is shown in Fig. 18, in which ESP can modify the normal force distribution on the tires by control of special anti roll bars [17].

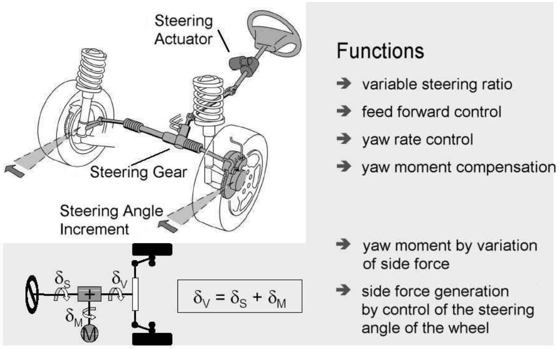

Fig. 17: Active steering system with incremental steering angle

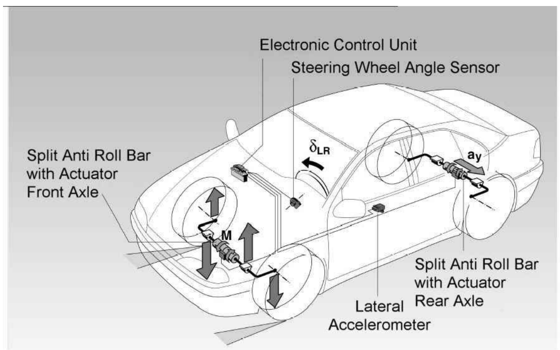

Fig. 18: Active suspension with split anti roll bar

Each of these systems can influence the yaw moment on the car. The question is now, how to implement the total yaw rate control. Early proposals were based on the observation of the different intervention bandwidths of the brake, engine, active steering and active suspension actuators. Each actuator had its own yaw rate controller (Fig. 19). The controller gains had to be tuned such that the interference between the interventions did not lead to undesirable or even unstable vehicle behavior. The charm of this approach is, that to some extend the development of the controllers can be done independent of each other. Obviously, if the controller gains must be drastically reduced to guarantee the peaceful coexistence of the controllers, important performance potential is lost. Identification, adaptation and learning control is not feasible.

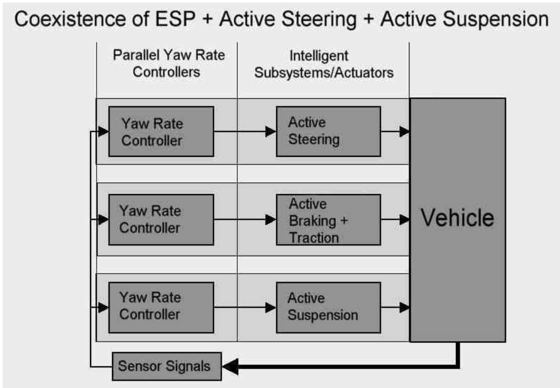

Fig. 19: Coexistence approach using the brakes, active steering and active suspension to control vehicle handling

Later proposals use a central yaw rate controller which takes the different properties of the actuators in consideration (Fig. 20). This is particularly important if the brake actuation is no more much slower than the steering actuation as is the case with EHB. The superimposed central yaw rate control uses then the subsystems as intelligent actuators.

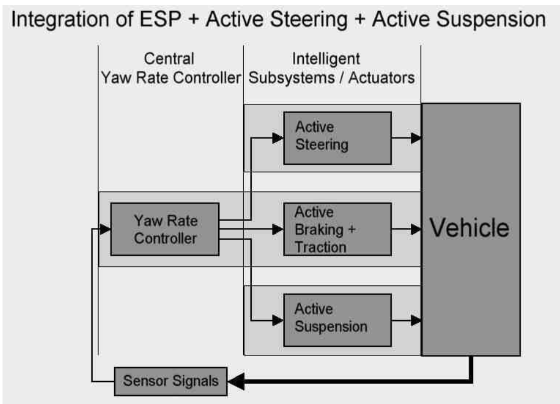

Fig. 20: Integration approach using the brakes active steering and active suspension to control vehicle handling

EHB and the future systems can influence the handling behavior of the car also in situations where the physical limit is not reached. If the passive car handling changes, e.g. because of changes in the tire properties, than these systems can restore normal vehicle behavior by interventions which are not noticed by the driver. These systems therefore have the potential of making handling robust against parameter changes. However, in order to fully exploit the high speed of the interventions, the test cycle of the yaw rate signal (Fig. 16) must be avoided which results in the requirement for redundant yaw rate signals.

7. CONCLUSION

ESP has extended active safety systems from a wheel behavior control (ABS, TCS) to a vehicle behavior control resulting in an increased active safety of the car. A practical system solution requires a model following yaw rate control which introduces robustness problems with changing vehicle parameters like the tire properties. Future systems may compensate these changes to a robust vehicle behavior but the safety concept must be changed to fully exploit the fast actuator interventions.

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