ADVANTAGES OF ACTIVE STEERING FOR VEHICLE DYNAMICS CONTROL

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Abstract

Yaw and roll dynamics of vehicles can be controlled efficiently by individual wheel braking or by active steering. Both approaches are compared on the basis of physical and application considerations. Two vehicle dynamics control concepts based on active steering are summarized. One of them focusses on the attenuation of yaw disturbances on the vehicle by robust unilateral decoupling of yaw and lateral mode. The other approach aims at rollover avoidance of road vehicles. There, in continuous operation, active steering improves the roll dynamics. In case of emergency an efficient strategy applies simultaneous steering and braking control.

1 Efficiency of braking and steering

Driver assistance systems for vehicle dynamics primarily produce a compensating torque for yaw disturbances. Such control systems can react faster and more accurately than the driver, when an unexpected deviation from the desired yaw rate occurs. The deviation is taken between the desired yaw rate (generated by a prefilter from the steering wheel input and the velocity) and the actual yaw rate (measured by a rate sensor). Also critical rollover conditions may be measured and fed back into a driver assistance system.

The actuation in such control systems is mostly by distribution of the brake force and in some cases also motor torque over the four wheels, ESP [1] is an example. The braking approach uses the existing ABS. Therefore only little additional hardware is required.

The question now arises: Is it worthwile to consider active steering as an alternative or in combination with an active braking system? We will first discuss the potential of both systems for yaw disturbance attenuation in terms of physical limits. Three simplifying assumptions are made for this discussion:

- The total force F_{max} which is transmittable by the tires does not depend on the direction in which the force acts (Kamm's circle).
- The center of gravity (CG) is assumed to be midway between the front and rear axles of the vehicle.

- The wheelbase ℓ is twice the trackwidth t.

Fig. 1 compares the physical limits of braking and steering. Obviously, the limits for rear wheel contributions to the corrective torque are the same in both cases. The available torque from front wheel braking is $F_{max} \cdot t/2$ and from front wheel steering it is $2F_{max} \cdot t$. In other words:

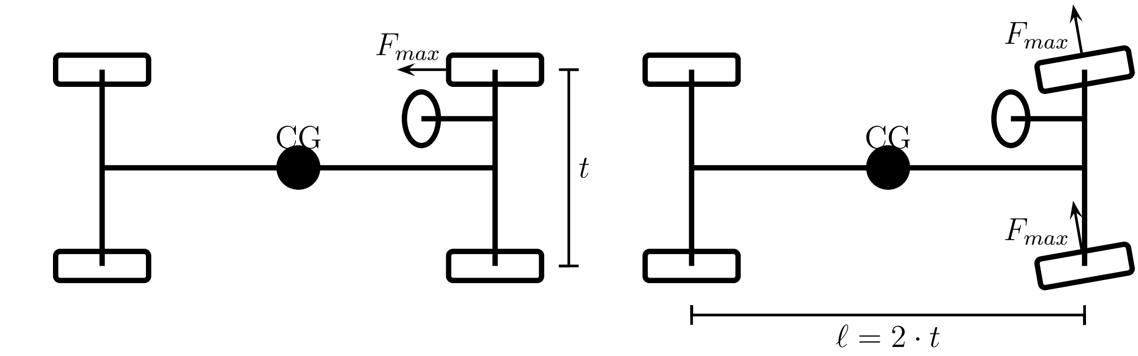

Fig. 1: Torques by front wheel braking (left) and front wheel steering (right).

Steering requires only one fourth of the front wheel tire force compared to asymmetric braking of the front wheels.

A further advantage of steering for generating a corrective torque is that it allows for a compensation of torques caused by asymmetric braking. An extreme μ-split braking situation with

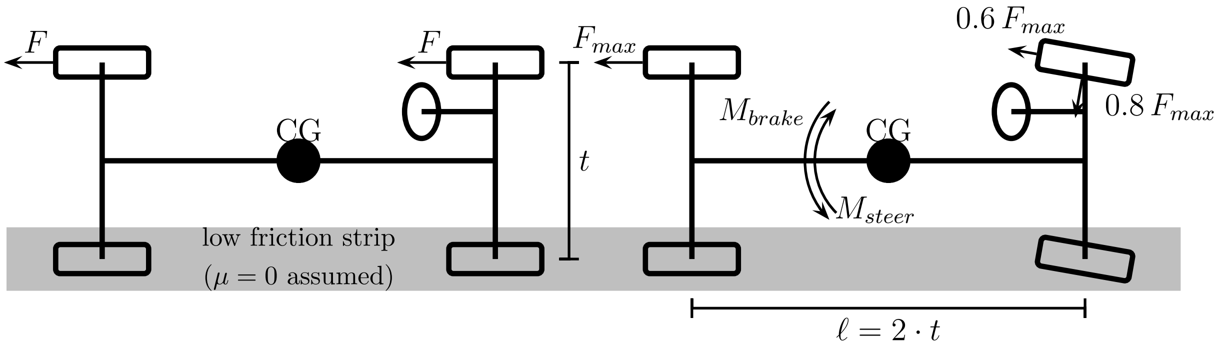

Fig. 2: For extreme μ-split braking the balance of torques yields zero brake force $F = 0$ for braking only (left) and $F = 1.6F_{max}$ brake force for combined braking and steering (right).

$\mu = 0$ under the right wheels is shown in Fig. 2. The combination of braking and steering (see right hand side of the figure) allows for a balance of torques $M_{brake} - M_{steer} = 1.6F_{max} \cdot t/2 - 0.8F_{max} \cdot t = 0$ and there still remains $1.6F_{max}$ for deceleration. (Note that $(0.6^2 + 0.8^2)F_{max}^2 = F_{max}^2$.) A mere ABS (see left hand side of the figure) would produce $F \le F_{max}$ and therefore a disturbance torque. In order to compensate for this yaw disturbance an individual wheel braking system can only make $F = 0$. Then, however, no deceleration is achievable any more.

The continuous operation of the active steering system yields additional advantages over an emergency braking system, in particular regarding comfort (e.g. under conditions of gusty wind, trailer pulling, and road irregularities) and safety (no discontinuity of vehicle dynamics in critical driving situations).

For a rollover protection system it may seem natural to harden the roll damping using active suspension and apply active counter roll inclination of the roll body towards the inner side of the curve. However, under a strict energy limitation a combined steering and decelerating action is much more efficient. Steering has an immediate effect on the roll dynamics, while deceleration involves more delay. A steering/braking control system allows larger obstacle avoidance maneuvers and supports the driver in case of emergency, i. e. when the vehicle is close to rollover.

From an application point of view there is no need to wait with active steering until steer-by-wire is state of the art in automobiles. Active steering systems for automobiles have been studied for a long time now. 30 years ago, Kasselmann and Keranen [2] designed an active control system that measures the yaw rate by a gyro and uses proportional feedback to generate an additive steering input for the front wheels, see Fig. 3. This early Bendix study never made it to a product. Some of its ideas are, however, still relevant for actual and future active front-wheel steering systems. Actuators for adding a feedback controlled steering angle to the driver commanded steering angle (see Fig. 3) may be placed in the rotational motion of the steering column [3] or in the lateral motion of the steering linkage [2]. A recent proposal by TRW is shown in Fig. 4.

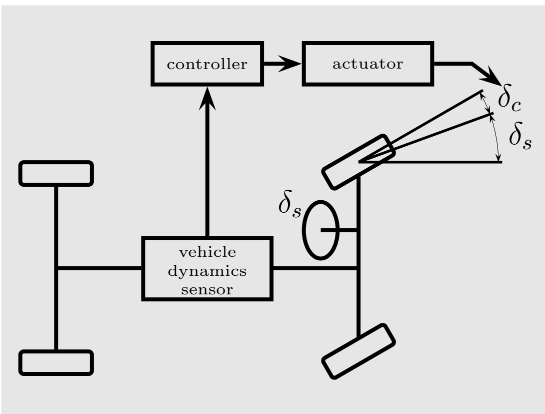

Fig. 3: Additional steering angle actuation principle.

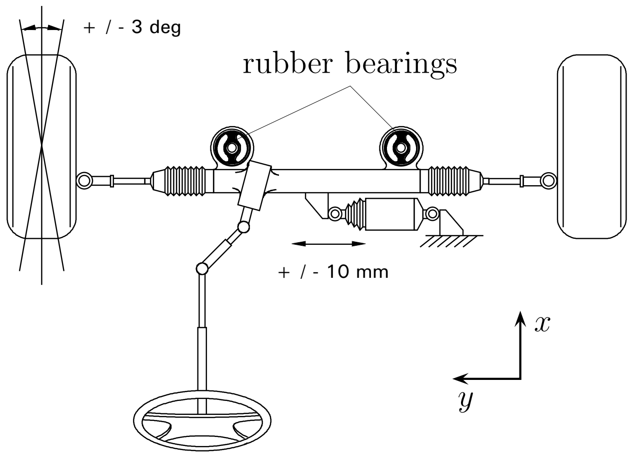

Fig. 4: Implementation example for additive steering actuator (courtesy of TRW).

The directional rubber elements connecting the steering gear housing with the car body are stiff in x-direction and elastic in y-direction. The y-direction is under control of an actuator that may be hydraulic or electric. In the latter case a low-friction spindle gear is suited to efficiently transform the rotational motion of the motor into a lateral shift [4].

We conclude that the high potential of active steering is available at reasonable cost. The following sections of this paper describe two robust control concepts for yaw disturbance rejection (section 2) and rollover avoidance (section 3), respectively.

2 Yaw motion control by active steering

In [5] a steering control method was presented which achieves robust unilateral decoupling of the vehicle's yaw and lateral motions, i.e. the yaw rate r is no longer observable from the lateral acceleration at the front axle a_{yF}, see Fig. 5. Remarkable about this method is, that the decoupling property holds robustly despite varying operating conditions such as speed, mass of the vehicle, and uncertain road conditions. It provides an advantagous separation of two basic tasks being under the responsibility of the driver so far: Path following and disturbance attenuation. With the robust steering control, the first task is left to the driver but the latter is managed by the active steering system [6]. This makes driving a car easier and provides convincing safety advantages. They were verified in driving experiments in cooperation with BMW [7]. In μ-split braking and sidewind maneuvers the decoupled car showed excellent disturbance rejection. In 1996 the IEEE Bode Prize [8] and the IFAC Nichols medal were awarded for the underlying scientific work. The robustly decoupling control law itself has a very simple structure; it basically consists of unity feedback of the integrated yaw rate to the front wheel steering angle.

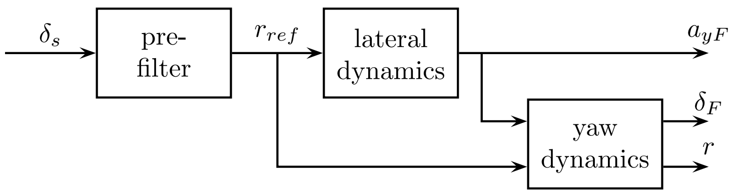

Fig. 5: Signal flow diagram of unilaterally decoupled steering dynamics.

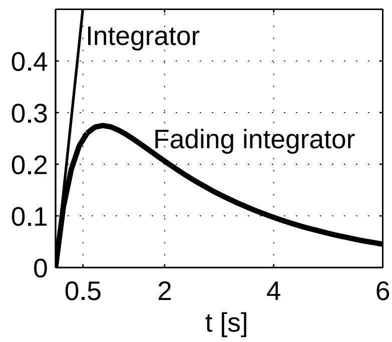

Fig. 6: Step responses of the integrator and the *fading integrator*.

The application of theoretic results usually calls for modifications in order to adapt them to practical requirements. In the case of the robustly decoupling steering control, three items were not satisfactory in the driving experiments: First, the damping of the separated yaw dynamics was not sufficient at high speed. Second, the integral feedback had been implemented only to achieve robust unilateral decoupling. However, it provides steady state accuracy according to a set point (r_{ref} in Fig. 5), which is considered neither necessary nor useful. It is rather wanted to preserve the steady state properties of the conventional car. A third effect was that limit cycles occurred due to actuator rate limitations.

Several remedies have been developed to cope with the drawbacks of unilateral decoupling [9]. Key element of the modifications to the robustly decoupling controller is the replacement of the integrator by the *fading integrator* [10], a second order linear filter. For illustration of its effect, the step response is compared to the step response of a perfect integrator in Fig. 6. The initial response of both integrators is the same. However, after about half of a second, the fading integrator's output returns to zero, yielding zero gain of this filter for low frequencies. This gives the same steady state behavior as for the conventional car, but it still provides the same advantageous disturbance rejection within 0.5 seconds, i.e. the driver gets support within the period of his reaction time. However, his responsibility for tasks which are not time critical is not unnecessarily cut. This makes the fading integrator an interesting element of the man-machine-interface of the steering control. The fading integration has some more benefits. By its application, the yaw damping gets improved and actuator saturation in steady state cornering is avoided. A simple nonlinear controller modification is described in [9, 11] which proves to be very efficient for the prevention of limit cycles. The prefilter for generation of the yaw rate's set point was refined in [12] such that good handling qualities are robustly achieved.

By the original unilateral decoupling, the characteristic polynomial of the controlled car was robustly factorized into a term representing the lateral dynamics and a second order term representing the yaw dynamics. This allowed the explicit calculation of the damping of the yaw mode. The robust factorization is preserved with an approach described in [13], where additional proportional feedback of the yaw rate is used to improve the yaw damping. Here, the damping of the yaw motion can be set by the choice of controller gains. Gain scheduling with velocity allows for a tradeoff between exact decoupling at medium speeds and good yaw damping at high speeds.

3 Vehicle rollover avoidance by active steering and braking

Driving situations, which can directly induce vehicle rollover, are excessive speed when entering a curve, severe lane change or obstacle avoidance maneuvers or disturbance impact like sidewind gusts. Vehicles with a high CG are especially prone to rollover. Moreover, many driver mistakes result from an overestimation of the vehicle's roll stability which varies due to large changes of the load dependent CG height. From common sense it is clear that the ratio of the track width and the height of the CG (the socalled track width ratio) is the most important vehicle parameter affecting the rollover risk, and accident analysis results [14] confirm this fact. A rollover coefficient R is defined in [15] that basically depends on the track width ratio and the lateral acceleration at the center of gravity of the vehicle's sprung mass. For values of $|R| < 1$ the vehicle is rollover stable, for $R = \pm 1$ the vehicle's left or right wheels lift off the road.

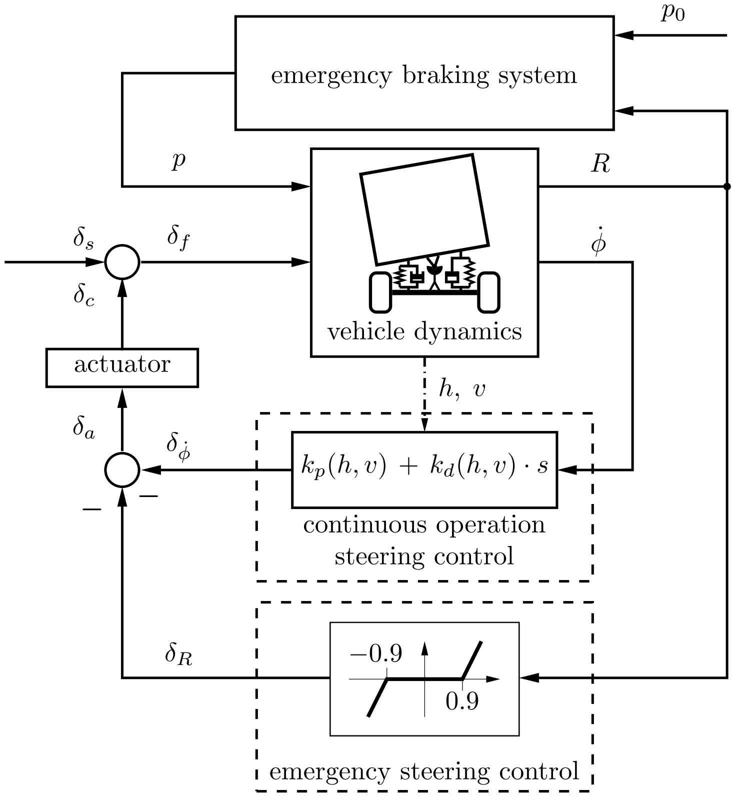

Fig. 7: Structure of the rollover avoidance system.

In [16] a control concept for rollover avoidance based on active steering and simultaneous braking is presented. The assumed controller structure shown in Fig. 7 is composed of three feedback loops: emergency steering control, emergency braking control and continuous operation steering control.

In case of emergency, indicated by $|R|$ approaching 1, the key idea is that rollover avoidance is given priority over ideal lanekeeping. To drive the sharpest curve which is physically possible, maximum lateral acceleration must be applied. The feasible lateral acceleration is limited by rollover. The relevant boundary is reached if the vehicle is steered such that the inner wheels are just about to lift off the road. This corresponds to $|R| = 1$ and is called the rollover limit. The optimal strategy to keep the sharpest curve possible while avoiding rollover would be to keep the vehicle at the rollover limit. Here, this idea is implemented in a nonlinear steering control law keeping a safety margin. Therefore, if the magnitude of R exceeds e.g. a value of 0.9, then the overstepping difference $\delta_R = k_R(R - 0.9)$ is fed back to the front wheel steering angle such that the curvature of the course is slightly reduced and rollover is avoided (k_R denotes the slope of the nonlinear deadzone characteristics in Fig. 7). At the same time deceleration of the vehicle is forced by operation of the brakes to improve the effect of rollover risk reduction. The magnitude of the braking pressure p may be controlled such that the course according to the driver's steering command is maintained.

Feedback of R requires the knowledge of the lateral acceleration at the CG and the height of the CG. Therefore, the height of CG is estimated once at the start of a ride. Since this estimation is not time critical offline-estimation methods, i.e. parameter identification can be applied. After the height of the CG is estimated the controller is switched on. Then, the lateral acceleration at the CG can be computed by interpolating measurement signals of two accelerometers fixed at two positions, e.g. at the bottom and the roof of the sprung mass, respectively.

In addition, continuous operation steering control is applied for improvement of the vehicle's roll-damping and roll-disturbance attenuation. The transfer function from the steering wheel angle δ_s to the roll rate $\dot{\phi}$ (see Fig. 7) is shaped by a control law such that the roll damping is optimized satisfying two constraints: 1.) steering amplification is reduced and disturbances are attenuated in the range of the roll resonance frequency, 2.) the controlled vehicle's dynamic is at least as fast as that of the conventional vehicle (see [17]). The roll rate, which can be measured by a rate sensor, is fed back by a PD-controller to the front wheel steering angle. The gains of the feedback law are scheduled with the vehicle velocity and the height of the CG, i.e. $k_p = k_p(v, h)$, $k_d = k_d(v, h)$, such that for any operating point optimal performance is ensured.

For this given controller structure the scheduling law for the continuous operation steering control, i.e. the gains of the roll rate and roll acceleration feedback $k_p(v, h)$, $k_d(v, h)$ and the slope k_R of the nonlinear dead zone characteristics in the emergency feedback loop in Fig. 7 were determined in [16, 17, 18]. In [17, 18] the continuous operation steering controller was designed using PARADISE (Parametric Robustness Analysis and Design Interactive Software Environment) [19], a Matlab-based toolbox for parametric robust control. In [16] the Popov criterion was applied to prove that for a given k_R absolute stability of the steering control feedback loop, i.e. continuous operation and emergency steering control, is ensured.

For illustrating the effect of the controller concept, in Fig. 8 some simulation results are presented. The simulations were performed using a simple nonlinear dynamic vehicle model [16]

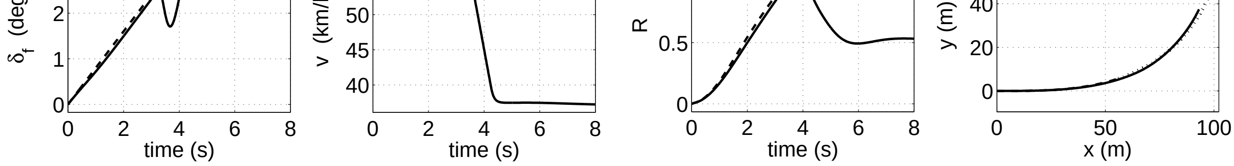

Fig. 8: Simulation results for a driver ramp like steering input.

which describes the vehicle's longitudinal, lateral, yaw and roll dynamics, assuming dry road, an initial velocity of 60 km/h and an unfavorably large height of the CG. The vehicle data were taken from [20].

Fig. 8 shows the responses of the conventional (dashed line) and the controlled vehicle (solid line) when a ramp-like input signal is applied to the steering wheel angle δ_s. This maneuver corresponds to driving through a highway exit with a clotoidal transition from straight road to a circle with a constant radius. After about 3 s the conventional vehicle reaches the rollover limit. Here, the dashed line ends, but for the sake of comparability the line is continued until the end of the maneuver with dotted linestyle. Note, that the simulation model is no longer valid after the occurrence of $|R| > 1$. The difference of both vehicles until 3 s indicates the effect of the continuous operation steering control, which is of minor importance in this maneuver.

Emergency steering and braking control is switched on after about 3 s when the rollover coefficient R implies that the vehicle is close to rollover, i.e. $|R| > 0.9$. After about 4.5 s the emergency control is switched off since here $|R| < 0.9$. Due to the fast and precise steering intervention rollover is avoided. Only little track error occurs in the vehicle's position plot in Fig. 8 because the vehicle is simultaneously decelerated by the emergency braking system. Comparably advantageous results were obtained when variations of v and h and other maneuvers were investigated in further simulations.

4 Conclusions and outlook

Active steering is an efficient means to influence a vehicle's yaw and roll dynamics. A comparison with vehicle dynamics control systems which make use of individual wheel braking reveals the fundamental advantages and drawbacks of both approaches. Individual wheel braking is implementable with less hardware effort since the actuators and wheel speed sensors are available by anti-lock brakes. However, active steering is more efficient with regard to fundamental mechanical considerations. Thus, the physical limits in terms of maximal force between tire and road can be further exploited to provide additional safety margins. In this paper two active steering concepts have been summarized, which are suitable to improve the yaw disturbance attenuation and to reduce rollover risk respectively. The potential being inherent in active steering will be easily usable once steer-by-wire is established. But one does not have to wait until the present hurdles in both the legal and technological sense are overcome. There are already technical solutions which provide the possibility to set an auxiliary steering angle additionally to the one directly transmitted by the steering wheel. At the present stage of technology, active steering suggests itself as stand-alone or as a powerful complement to present individual wheel braking systems.

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