Design of a Supervisory Integrated Control for Driver Assistance Systems

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Abstract—The paper proposes a control design method for a driver assistance system. In the operation of the system a predefined trajectory required by the driver with a steering command is followed. During maneuvers the control system generates differential brake moment and the auxiliary front wheel steering angle and changes the camber angles of the wheels in order to improve the tracking of the road trajectory. The performance specifications are guaranteed by the local controllers, i.e., the brake, the steering, and the suspension systems, while the coordination of these components is provided by the supervisor. The advantage of this architecture is that local controllers are designed independently, which is ensured by the fact that the monitoring signals are taken into consideration in the formalization of their performance specifications. The control system also uses a driver model, with which the reference signal can be generated. In the control design the parameter-dependent LPV method, which meets the performance specifications, is used.

I. INTRODUCTION AND MOTIVATION

The paper proposes the design of a driver assistance control. The purpose of trajectory tracking is to assist the driver in following road geometry at a predefined velocity and guarantee the road stability of the vehicle simultaneously. The control system includes several vehicle components such as the brake, the steering and the suspension systems. Primarily the driver generates the steering angle by using the steering wheel. This angle is corrected by the automatic control system by generating additional steering angle. A brake yaw moment is also created by generating a difference in brake forces, which affects the lateral tire forces directly. A variable-geometry suspension system changes the camber angles of the wheels and is able to modify the height of the roll center and the half track change.

The demand for vehicle control methodologies including several control components arises at several research centers and automotive suppliers, see e.g. [1]. A vehicle control with four-wheel-distributed steering and four-wheel-distributed traction/braking systems is proposed by [2]. A process to design the control strategy for a vehicle with throttle control and automatic transmission is proposed by [3]. A yaw stability control system in which an active torque distribution and differential braking systems are used is proposed by [4]. An integrated control that involves both four-wheel steering and yaw moment control is proposed by [5]. Active steering and suspension controllers are also integrated to improve yaw and roll stability [6]. A global chassis control involving an active suspension and brake is proposed by [7], [8]. In the integration of various control components the multiple model approach is also proposed, see [9]. A decentralized integration method by using the concepts of a multi-agent system is proposed by [10].

The solution to an integrated control proposed in the paper is a decentralized control structure in which the control components are designed independently, see [11]. In the structure the supervisor has an important role, since it must guarantee coordination between components and meet performance specifications. The supervisor has information about the current operational mode of the vehicle, i.e., the various vehicle maneuvers or the different fault operations gathered from monitoring components. A local controller focuses on their performance specifications and uses scheduling variables received from the supervisor. Thus the controller is able to modify or reconfigure its normal operations in order to improve safety. The structure also contains the fault-tolerant controllers, in which the detected fault signals are built in their performance specifications.

The paper is organized as follows. In Section II the control problem is set up and the control-oriented vehicle model is formalized. In Section III the components of the supervisory control are presented in detail. The coordination between controllers and priorities from actuators are provided. In Section IV the stability and performances of the entire system are presented. In Section V the integrated control is demonstrated through simulation examples. Section VI contains some concluding remarks.

II. CONTROL-ORIENTED MODEL FOR TRAJECTORY TRACKING

A. Control-oriented model for trajectory tracking

The control system of the lateral vehicle dynamics assists the driver in tracking road geometry. It has advantages in critical situations, where the driver is not able to ensure road stability. In trajectory tracking the vehicle is moving along the road as Figure 1 shows. The vehicle dynamics is determined by the driver while the lateral dynamics is assisted by the control systems.

The reference signal required by the driver is based on road geometry. In the specifications of the performance criteria the driver behavior can be taken into consideration.
through various driver models, see e.g. [12]. The input of
the driver is the position error from the defined line in lateral
direction $e_y$, while its output is the driver’s steering angle $\delta_d$.
During maneuvers the difference between the lateral position
of the vehicle and the reference lateral position is minimized
by the driver. Based on $\delta_d$ the reference yaw rate $\dot{\psi}_{\text{ref}}$ is
calculated using a first order proportional transfer function,
see [13]:

$$
G_{\dot{\psi}_{\text{ref}}, \delta_d} = \frac{v}{l_1 + l_2 + \frac{3}{2} v^2} \cdot \frac{1}{\tau s + 1} \quad (1)
$$

with an understeer gradient $\eta$ and the time constant $\tau$. In the
closed-loop interconnection structure $\dot{\psi}_{\text{ref}}$ is used instead of
$\delta_d$, which can be compared to the measured yaw rate $\dot{\psi}$.

In the trajectory tracking control the lateral dynamics must
be taken into consideration and the vehicle must follow
the reference yaw rate signal. The difference between the
measured yaw rate of the vehicle and the reference yaw rate
must be minimized: $|\dot{\psi}_{\text{ref}} - \dot{\psi}| \rightarrow \min$. Note that using this
approach the lateral error is minimized both by the driver
and the control systems.

B. Control inputs for trajectory tracking

Three actuators are used in the system: the front-wheel
steering angle $\delta$, the differential brake moment $M_{br}$, and
the wheel camber angle $\gamma$.

The formulism of the vehicle lateral dynamics is based
on the bicycle model. In the construction of the variable-
gameometry suspension system the change in the camber angle
of the wheels influences vehicle dynamics. Since the effect
of the front wheel camber angle on lateral dynamics is more
significant than that on the rear wheel camber angle, the
front wheel camber angle is applied in our method. The effect
of the wheel camber angle is formalized by a linearized
approach based on the Magic Formula, see [13].

The lateral tire forces in the direction of the wheel ground
contact are approximated linearly to the tire side slip angles:

$$
F_{yf} = C_f \alpha_f + C_{f\gamma} \gamma, \quad C_f = C_f(\alpha_f), \quad C_{f\gamma} = \text{coefficient}
$$

where $C_f$ is the cornering stiffness and $C_{f\gamma}$ is a coefficient, which represents the degree of offset and $\gamma$ is the wheel camber angle, see [14]. The tire side-slip angle $\alpha_f$ is expressed as $\alpha_f = -\beta + \delta - l_1 \cdot \dot{\psi} / v$, where $\psi$ is the yaw rate of the vehicle and $\beta$ is the side-slip angle. Moreover, the
tire slip angle at the rear is as follows: $\alpha_r = -\beta + l_2 \cdot \dot{\psi} / v$.

The difference in brake forces on the left and right hand side
at the front and the rear of the vehicle creates the differential
brake moment $M_{br}$. The lateral dynamics of the vehicle is formulated as:

$$
\dot{\psi} = C_f l_1 \alpha_f - C_r l_2 \alpha_r + M_{br} + C_f l_1 \gamma \quad (2)
$$

$$
\dot{\psi} + \beta = C_f \alpha_f + C_r \alpha_r + C_f \gamma \quad (3)
$$

where $m$ is the mass, $J$ is the yaw inertia of the vehicle
and $l_1$ and $l_2$ are geometric parameters. The state space
representation form of the motion equations of the vehicle:

$$
\dot{x} = A(x)u + B(x)w
$$

with the state vector of the system $x = [\psi \ 1 \ \beta]^{T}$ contains the yaw rate and the side-slip angle of the vehicle.
The system matrices depend on the velocity of the vehicle
nonlinearly. Using a scheduling variable $\rho = v$ the nonlinear
model is transformed into an LPV model. In the case of the
brake control the input of the system is $u = u_{br} = M_{br}$, in
the steering control the input is $u = u_{st} = \delta$, while in the
variable-geometry suspension the control input is the wheel
camber angle $u = u_{susp} = \gamma$.

C. Closed-loop interconnection structure

The control design is based on LPV methods, in which
both the the parameter varying model and the parameter-
dependent weighting functions are taken into consideration.
On the LPV control design level the controller must change
its behavior according to monitoring signals, i.e., the control
design method must enable the introduction of this type of
information in the performance specification.

The closed-loop system applied in the design of integrated
control includes the feedback structure of the model $G(\rho)$,
the compensator, and elements associated with the uncertainty
models and performance objectives:

$$
z = C(\rho)x + D_1(\rho)w + D_2(\rho)u
$$

where $w$ includes both the external disturbances and the
sensor noise and $u$ is the control input.

In the control design the purpose is to solve the tracking
problem. In trajectory tracking the reference signal is the
yaw rate defined by the steering angle of the driver $\psi_{\text{ref}}$.
The performance of the control system is the minimization of the
yaw rate tracking error

$$
z_1 = z_{\psi} = |\dot{\psi}_{\text{ref}} - \dot{\psi}| \rightarrow \min
$$

The design of the active steering control $K_{st}$, in which
the only performance is the yaw rate tracking, is based on a
robust control design method. The control input of active
steering is the front wheel steering angle $\delta$, which is added to
the steering angle of the driver $\delta_d$. In the design of the
active brake control $K_{br}$, similarly to the previous case, the
only performance is the yaw rate tracking.

In the control design of the variable-geometry suspension
system, however, other performances related to road stability
are also introduced. The role of the variable-geometry sus-
penion control $K_{susp}$ is to minimize the chassis roll angle
and the half-track change and, in an emergency, reduce the
yaw rate error.
The height of the roll center has an important role in the roll dynamics of the vehicle [15]. The roll dynamics is formulated in the following way:

\[(I + mΔh^2)\ddot{\phi} = mgΔh\dot{\phi} + mvΔh(\dot{\beta} + \dot{\psi}) - B_{i} \sum F_{s,i}\]

(7)

where \(Δh\) is the difference between the height of the center of gravity and the height of the roll center \((Δh = h_{CG} - h_{M})\), \(\phi\) is the chassis roll angle, \(I\) is the inertia of the chassis, \(B_{i}\) is the half track and \(F_{s,i}\) are the vertical forces of suspension. In order to minimize the chassis roll angle, the dynamic displacement of the height of the roll center \(|Δh|\) must be minimized. In practice, however, \(h_{M}\) has a physical limit, \(h_{\text{ref, max}}\). Therefore, a signal \(h_{\text{ref}}(< h_{CG})\) is introduced and applied as a reference signal for the tracking task: \(Δh_{M} = |h_{\text{ref}} - h_{M}|\), in which \(h_{M}\) is calculated from the measured γ according to suspension geometry. During traveling an additional important economy parameter is the half track change (ΔB). The lateral movement of the contact point between the tire and the road is relevant from the aspect of tire wear when the suspension moves up and down while the vehicle moves forward, see [14], [16].

Note that the performance requirements are in conflict, thus a balance must be achieved between them. Consequently, two additional performance signals for the variable-geometry suspension system are defined:

\[z_{2} = z_{Δh} = |Δh_{M}| \rightarrow \text{min}!\] (8)

\[z_{3} = z_{ΔB} = |ΔB| \rightarrow \text{min}!\] (9)

In the supervisory decentralized control proposed in the paper the role of LPV methods is fundamental. In the formalism of the control-oriented model, the selection of monitoring components and building them into signals, which are related to the performance requirements, are crucial points in the modeling. In the control design phase parameter-dependent LPV methods are applied. The designed controller meets robust stability and performance demands in the entire operational region, see [17], [18].

In the design of local controllers the quadratic LPV performance problem is to choose the parameter-varying controller in such a way that the resulting closed-loop system is quadratically stable and the induced \(L_{2}\) norm from \(u\) to \(z\) is less than \(\gamma\). The existence of a controller that solves the quadratic LPV γ-performance problem can be expressed as the feasibility of a set of LMI s, which can be solved numerically. The incorporation of a parameter dependent Lyapunov function implies a potentially less conservative approach by addressing limitations on the rate of change in the parameters, see [19].

III. DESIGN OF THE SUPERVISORY CONTROL

A. Design of the weighting strategy

In this section the dynamics of the actuators are analyzed in order to examine their effects on vehicle dynamics. The actuation of the steering, the brake and the wheel camber is related to their construction and adhesion limits. The construction limits must be taken into consideration all the time, e.g. the value of front-wheel steering must not exceed its upper bound. Therefore, in order to avoid reaching the steering limit, differential braking and the wheel camber angle must be increased. Brake control also has a physical (adhesion) reason. In order to avoid the skidding of tires, the value of differential braking must be reduced. The skidding is monitored by the estimation of the longitudinal slips \(\kappa\), see [20].

The different components have energy requirement. By using differential braking during traction the velocity of the vehicle is reduced, which must be compensated for by additional energy. Therefore the use of differential braking must be avoided during acceleration and front-wheel steering is preferred. During deceleration the brake is already being used, thus the lateral dynamics is handled by the braking for practical reasons. Thus differential braking is preferred, but close to the limit of skidding, front-wheel steering is more efficient.

The operations of the actuators are based on the following weighting strategy. The weighting functions for the front wheel steering and the brake yaw moment are the following:

\[W_{\text{act, } \delta} = \rho_{\text{st}}/\delta_{\text{max}},\] (10)

\[W_{\text{act, } \delta_{br}} = \rho_{\text{br}}/\delta_{br\text{max}},\] (11)

where \(\rho_{\text{st}}\) and \(\rho_{\text{br}}\) are scheduling variables applied to the steering and the brake control, \(\delta_{\text{max}}\) and \(\delta_{br\text{max}}\) are determined by the constructional maximum of the steering angle and that of the brake yaw moment. The weighting factors \(\rho_{\text{st}}\) and \(\rho_{\text{br}}\) are set by the supervisor according to the analyzed criteria as shown in Figure 2.

![Fig. 2. Selection of parameters \(\rho_{\text{st}}\) and \(\rho_{\text{br}}\)](image)

In the case of traction the front wheel steering is preferred, which is determined by factor \(\rho_{\text{st}}\), see Figure 2(a). The value is reduced between \(\delta_{1}\) and \(\delta_{2}\), which represents the constructional criterion of the steering system. In this interval differential braking is preferred for practical reasons. The values of \(\rho_{\text{st}}\) also depend on the velocity of the vehicle. The effect of velocity on the weighting factors is the consequence of the difference between the bandwidth values of the actuators. According to the inertia of steering, the bandwidth of steering is lower at each frequency than the bandwidth of differential braking. At higher velocities it is recommended to use differential braking, while at lower velocities the steering actuation is preferred for practical reasons. Consequently, two additional design parameters \((v_{1}\) and \(v_{2}\)) are also introduced. In the case of braking the tire longitudinal slip angle \(\kappa\) affects \(\rho_{\text{br}}\), see Figure 2(b). It
requires an interval to reduce tire skidding ($\kappa_1$, $\kappa_2$) and it also requires an interval to prevent chattering between steering and differential braking ($\kappa_3$, $\kappa_4$).

In the control design of the variable-geometry suspension system the performance requirements are in conflict, thus a balance must be achieved between them by using two signals, $\rho_{susp}$ and $h_{\text{ref}}$. In the control design parameter-dependent weighting functions are applied to the performances:

$$W_{e,\psi} = \rho_{susp}/e_{\psi_{\text{max}}}$$

$$W_{\Delta h_{\text{max}}} = (1 - \rho_{susp})/\phi_{\text{max}}$$

$$W_{\Delta B_{\text{max}}} = (1 - \rho_{susp})/\Delta B_{\text{max}}$$

where $\rho_{susp}$ is a scheduling variable, $e_{\psi_{\text{max}}}$ is the possible maximum of the yaw rate error, $\phi_{\text{max}}$ is the maximum of the chassis roll angle and $\Delta B_{\text{max}}$ is a maximum of the half-track change. In the following three vehicle scenarios are distinguished.

Case a./ In normal cruising the goal of the variable-geometry suspension control is to minimize half-track change $\Delta B$. This configuration is achieved by the selection $\rho_{susp} = 0$, $h_{\text{ref}} = h_{\text{ref,max}}$. This is the default configuration.

Case b./ When the roll angle $\phi$ increases significantly, the variable-geometry suspension control must minimize the roll angle. This configuration is achieved by the selection $\rho_{susp} = 0$, $h_{\text{ref}} = h_{\text{ref,max}}$.

Case c./ In an emergency maneuver, when the yaw rate error has increased extremely, the suspension system must focus on the tracking error $e_{\dot{\psi}}$ instead of the conventional two performances. This configuration is achieved by the selection $\rho_{susp} = 1$ and $h_{\text{ref}} = h_{\text{M}}$.

Note that it is possible to achieve vehicle maneuvers in which there is a balance between two performances, i.e., the reduction of the half-track change and that of the roll angle. In these configurations $\rho_{susp} = 0$ and $h_{\text{ref}}$ is selected in an interval $h_{\text{M}} < h_{\text{ref}} < h_{\text{ref,max}}$. When the suspension system must focus on the trajectory tracking, i.e., in emergency maneuvers, the scheduling variable $\rho_{susp} = 1$ is selected. The selections of the variables $\rho_{susp}$ and $h_{\text{ref}}$ are shown in Figure 3. In the selection mechanism $e_{\psi_{1}}, e_{\psi_{2}}$ and $\phi_{1}, \phi_{2}$ are design parameters.

In the control architecture the supervisor has information about the current operational mode of the vehicle, i.e., the various vehicle maneuvers or the different fault operations gathered from monitoring components and fault-detection and identification (FDI) filters. It uses the yaw rate error to improve trajectory tracking, and it also uses velocity, front wheel steering angle and longitudinal slips of tires to determine the scheduling variables $\rho_{\text{br}}, \rho_{\text{st}}, \rho_{susp}$.

The system architecture also contains the driver model, which generates the steering angle of the driver $\delta_d$ by using the difference between the lateral position of the vehicle and the reference lateral position is minimized by the driver. Several signal generator (SG) blocks are also built in the scheme. One of them calculates the reference yaw rate based on the driver steering angle, and another one calculates the error of the chassis roll height $e_{h,M}$. These signals are used in the steering, the brake and the suspension controllers, which generate the control signals, i.e., $\delta, M_{\text{br}}, a_y$.

C. Design of a fault-tolerant integrated control

Since fault information is also available the supervisor is able to guarantee the reconfigurable and fault-tolerant operation of the vehicle. In order to achieve that, the signals of various fault scenarios provided by FDI filters are built in the performance specifications of the controller. There may be various fault scenarios, e.g. the leakage of a hydraulic system, or the steering mechanism becomes jammed. The different changes in the operation of an actuator make it possible to detect a fault.

The steering and the brake are operated in cooperation in order to provide a reconfigurable fault-tolerant control system. In the case of a detected fault the brake yaw moment, the front wheel steering or the wheel-camber angle might be changed, i.e., their values differ from the required values. The supervisor handles the different fault scenarios by modifying the scheduling variables. In the following the responses of the supervisor to different fault scenarios are presented.

Case a./ When performance degradation occurs in the operation of the steering system, its role must be substituted for by using the braking system. In the fatal error of the
steering system the weight of steering must be masked: 
\( \rho_{st,new} = 0 \).

Case b./ When performance degradation occurs in the operation of a brake circuit the brake yaw moment must be substituted for by using the steering and suspension to provide trajectory tracking. Moreover, the reduction of the brake yaw moment is asymmetric. For example, in the case of a left-hand-side brake circuit fault in the rear the brake is not able to turn the vehicle anti-clockwise, therefore positive \( M_{br} \) is not allowed, i.e., \( \rho_{br} = 0 \). However, if \( M_{br} < 0 \) then \( \rho_{br} > 0 \). Consequently, if there is one fault in the brake system the weight of braking \( \rho_{br} \) depends on the sign of the desired brake yaw moment \( M_{br} \) and a gain \( \kappa_i \). In the realization of the gain \( \kappa_i \), either \( \kappa_{left} \) or \( \kappa_{right} \) must be set. The role of the variables \( \kappa_- \) and \( \kappa_+ \) is to prevent chattering. The actual modification of \( \rho_{br} \) is based on the sign of the desired brake yaw moment and the parameters \( \kappa_+ \) and \( \kappa_- \), which depend on the desired brake yaw moment:  
\( \rho_{br,new} = \kappa_i \rho_{br} \), where \( \kappa_i \) is selected according to Figure 5. For example \( \kappa_{right} = 0 \) if \( M_{br} \leq 0 \), \( \kappa_{right} = 1 - (\kappa_+ - M_{br})/\kappa_+ \) if \( 0 < M_{br} < \kappa_+ \), otherwise \( \kappa_{right} = 1 \).

IV. ANALYSIS OF STABILITY AND PERFORMANCES OF THE ENTIRE SYSTEM

In the entire system three controllers, such as the brake, the steering and the variable-geometry suspension, are used simultaneously. It is necessary to guarantee the stability and performances of the entire closed-loop system. Since each LPV controller contains one scheduling variable, the global system uses three additional scheduling variables \([\rho_{br}, \rho_{st}, \rho_{susp}]\). In order to provide a formal verification of the achieved control performance on a global level, the problem must be formulated globally.

The relationship between the supervisor and the local controllers guarantees that the system meets the specified performances. Applying parameter-dependent weighting functions balance, coordination and priorities between different controllers are achieved. In different critical cases related to extreme maneuvers or performance degradations/faults in sensors or actuators the controllers reconfigure their operations. The stability and performances for LPV systems are analyzed by using affinely parametrized Lyapunov functions in [19], [21], [22].

V. SIMULATION RESULTS

In the simulation example the vehicle is driven along a section of Waterford Michigan Race Track, which is shown in Figure 6(a). The road trajectory contains several difficult and sharp bends. Two scenarios with different velocities are shown. In each case the velocity also varies, see Figure 6(b). The driver without an assistance system is not able to follow the track, leaves the lane and the lateral error increases significantly as Figure 6(c) shows. At the same time, however, the supervisory integrated control significantly reduces the lateral error and improves the road stability of the vehicle. In the first bend the lateral error of the controlled vehicle is smaller at a higher velocity than that of the uncontrolled vehicle at a smaller velocity. It is the result of the minimization of the yaw rate error, see Figure 6(d).

The minimization of the yaw rate signal is achieved by integrating the control systems. Active front wheel steering angles \( \delta \) in the case of the two different velocities are shown in Figure 7(a). Without the driver assistance system the driver needs to turn the steering wheel in the entire steering range, which is uncomfortable and hazardous. In the controlled case the driver turns the steering wheel to a smaller degree and relatively smoothly. Consequently, the integrated control system is able to improve road stability and comfort simultaneously. The control moment of the brake system and the actuation of the suspension system are shown in Figure 7(b) and Figure 7(c). The suspension control modifies the wheel camber angle as Figure 7(d) shows. The direction of the camber angle is chosen according to the performance requirement and it results in the change of the wheel position. In normal conditions the minimization of the half-track change is preferred, see Figure 7(f). When the roll angle is in the focus, the height of the roll center is raised, which reduces the chassis roll angle, see Figure 7(e).

The actuator selection is based on monitoring signals which are used as scheduling variables. The selection variables of the steering and the brake controllers \( \rho_{st} \) and \( \rho_{br} \) are shown in Figure 8(a) and 8(b). In the variable-geometry suspension system the reference signal \( h_{ref} \) and the scheduling variable \( \rho_{susp} \) are used, see Figure 8(c) and 8(d). They create coordination between the minimization of the roll angle, half-track change and/or tracking error.
which meets the performance specifications is important. The simulation examples show that the integration of control components reduces the lateral error, improves road stability and reduces the roll angle.

**REFERENCES**


