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Design of actuator interventions in the trajectory tracking for road vehicles

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Abstract— **The paper proposes a control design of vehicle systems including several vehicle components such as the brake, steering and driveline. The purpose is to design a cruise control system to track the predefined velocity and the road geometry. The effects of the actuators are analyzed and an actuator selection method in the control design is developed. The cruise control system based on robust LPV (Linear Parameter Varying) methods is designed with the emphasis on predefined velocity and path considerations.**

I. Introduction and motivation

The purpose of the trajectory tracking is to follow a road geometry with a predefined velocity and guarantee the road stability of the vehicle simultaneously. The control system includes several vehicle components such as the brake, steering and driveline systems. The difficulty in the design is that the actuators affect the same dynamics of the vehicle. In the control design the interaction between the actuators must be taken into consideration and a balance between them must be achieved.

The demand for vehicle control methodologies including several control components arises at several research centers and automotive suppliers. Here are a few examples for illustration. A vehicle control with four-wheel-distributed steering and four-wheel-distributed traction/braking systems is proposed by [1]. A process to design the control strategy for a vehicle with throttle control and automatic transmission is proposed by [2]. A yaw stability control system in which an active torque distribution and differential braking systems are used is proposed by [3]. An integrated control that involves both four-wheel steering and yaw moment control is proposed by [4], [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 ABS is proposed by [7], [8]. The driveline system and the brake are integrated in [9]. A possible integration of the brake, steering and suspension system is presented by [10]. Recently, important survey papers have also been presented in this topic, see e.g. [11].

An important task of the control design is to generate control forces into different directions, such as steering angle, difference in brake forces and longitudinal forces. The paper extends the control design with a new actuator selection procedure. Although this selection is usually performed by using practical considerations, in the paper a theory-based approach is used. The procedure is built in the control design by exploiting the advantages of the LPV methods. These methods allow us to take into consideration the highly nonlinear effects in the state space description in such a way that the model structure is nonlinear in the parameters, but linear in the states. Moreover, in the LPV method both performance specifications and model uncertainties are taken into consideration. The designed controller meets robust stability and performance demands in the entire operational region, see [12], [13].

The paper is organized as follows: in Section II the nonlinear control-oriented vehicle model is formalized. In Section III the effects of the actuators are analyzed and an actuator selection method is developed. In Section IV the design of trajectory tracking control based on LPV methods is proposed. In Section **??** the architecture of the tracking control is presented. In Section V simulation results are presented and in the last section the contribution is summarized.

II. Control-oriented modeling of the trajectory tracking

In the trajectory tracking task the vehicle is moving in the entire plane of the road, thus both the longitudinal and the lateral dynamics must be taken into consideration as Figure 1 shows. The lateral dynamics of the vehicle can

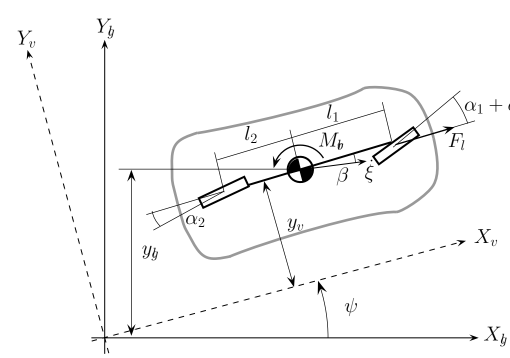

Fig. 1. Nonlinear model of vehicle

be approximated by the bicycle model of the vehicle, while the longitudinal dynamics can be formalized by using the longitudinal forces and with the notations $\alpha_f = \delta - \beta - \dot{\psi} l_1/\dot{\xi}$

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and $\alpha_r = -\beta + \dot{\psi} l_2/\dot{\xi}$:

$$m\ddot{\xi} = F_l - F_d \tag{1a}$$
$$J\ddot{\psi} = C_1 l_1 \alpha_f - C_2 l_2 \alpha_r + M_{br} \tag{1b}$$
$$m\dot{\xi}(\dot{\psi} + \dot{\beta}) = C_1 \alpha_f + C_2 \alpha_r \tag{1c}$$
$$\ddot{y}_v = \dot{\xi}(\dot{\psi} + \dot{\beta}) \tag{1d}$$

where m is the mass, J is the yaw-inertia of the vehicle, l_1 and l_2 are geometric parameters, C_1 and C_2 are cornering stiffnesses, ψ is the yaw angle of the vehicle, β is the side-slip angle, $\ddot{y}_v$ is the lateral and $\ddot{\xi}$ is the longitudinal acceleration. The actuators of the system are the front steering angle (δ), the brake yaw moment (M_{br}) and the longitudinal force (F_l). The disturbance force F_d affects the longitudinal dynamics of the vehicle. The sources of the disturbance are the rolling resistance, aerodynamic forces, road slopes, etc. The nonlinearity of the system is caused by the longitudinal velocity $\dot{\xi}$.

In the design of trajectory tracking it is necessary to guarantee that the longitudinal and lateral positions of the vehicle track the geometry of the road. The required longitudinal motion is ensured by velocity tracking, while the required lateral motion is controlled by the error between the lateral position and the road curve. The reference of the road geometry is defined by using a world coordinate system (X_{gl} and Y_{gl}), in which the coordinate system of the vehicle rotates together with the vehicle. The lateral position of the vehicle is calculated in both coordinate systems, see y_v and y_{gl} in Figure 1. Therefore, it is necessary to consider the rotation of vehicle at the calculation of the reference road geometry: $y_{v,ref} = -sin(\psi)\, x_{gl,ref} + cos(\psi)\, y_{gl,ref}$, where $y_{v,ref}$ is the lateral position of the reference road geometry in the coordinate system of the vehicle, $x_{gl,ref}$ and $y_{gl,ref}$ are the longitudinal and lateral coordinates of the reference road geometry in the world coordinate system.

In trajectory tracking both the longitudinal and lateral dynamics must be taken into consideration, i.e., the vehicle must track two reference signals. First of all it is necessary to ensure appropriate velocity tracking in the longitudinal direction: $\dot{\xi} \rightarrow \dot{\xi}_{ref}$. This requirement is formalized as an optimization criterion:

$$|\dot{\xi}_{ref} - \dot{\xi}| \rightarrow 0. \tag{2}$$

Second, the difference between the lateral position of the vehicle and the reference lateral position must be minimized:

$$|y_{v,ref} - y_v| \rightarrow 0. \tag{3}$$

These performances are built in a performance vector: $z_1 = [\dot{\xi}_{ref} - \dot{\xi};\ y_{v,ref} - y_v]^T$. Simultaneously, actuator saturations must be avoided. The maximal forces of both the driveline and braking systems are determined by their physical construction limits and so are the tyre-road adhesion conditions. These limits must be taken into consideration in the control design. Thus, they are formalized as performance criteria: $z_2 = \begin{bmatrix}\delta & F_l & M_{br}\end{bmatrix}^T$.

Note that the longitudinal and lateral dynamics are not independent; there are significant effects between the longitudinal velocity and the lateral displacement. This relationship is considered by applying the concept of safe cornering speed. Consequently, instead of designing independent longitudinal and lateral controllers it is necessary to design an autonomous tracking system. The motion equation of the vehicle is transformed into a state-space representation form:

$$\dot{x} = A(\rho)x + B_1(\rho)w + B_2(\rho)u \tag{4}$$

where the state vector of the system $x = \begin{bmatrix}\dot{\xi} & \xi & \dot{\psi} & \beta & \dot{y}_v & y_v\end{bmatrix}^T$ contains the longitudinal velocity and the displacement of the vehicle, the yaw-rate, the side-slip angle, the lateral velocity and position. The control input and the disturbance are $u = \begin{bmatrix}F_l & \delta & M_{br}\end{bmatrix}^T$ and $w = F_d$, respectively.

The system matrices depend on the velocity of the vehicle $\dot{\xi}$ nonlinearly. The velocity is assumed to be available, i.e., it is measured or estimated, see e.g [14]. Using a scheduling vector ρ with the scheduling variable $\rho = \dot{\xi}$ the nonlinear model is transformed into an LPV model. With the selection of the scheduling variable ρ the state space representation of the LPV model is valid in the entire operating region of interest. The measured output of the system is the velocity and the lateral position $y = \begin{bmatrix}\dot{\xi} & y_v\end{bmatrix}^T$. In practice the lateral position is measured by using a signal from an on-board video camera.

The controller calculates the required global longitudinal force and the brake yaw-moment. These signals must be distributed between the wheels in the realization of the control system. The purpose of the next section is to analyze the dynamics of the actuators and examine their effects on vehicle dynamics.

III. Analysis and synthesis of actuator intervention

The first constraint in the actuation of steering and brake derives from construction and adhesion limits. In both cases they have construction limits, e.g. the value of front-wheel steering can not exceed an upper bound. Although braking also has a construction limit, it cannot be exceeded, because of physical (adhesion) limits. It is necessary to avoid the skidding of tyres, thus in this case the generation of differential braking must be reduced, while the yaw-motion of vehicle must be controlled by front-wheel steering. Similarly, to avoid a steering limit differential braking must be increased.

Second it is necessary to consider that the actuations of the different components have energy requirement. By using differential braking the velocity of the vehicle is decreased, which must be compensated for by the driveline with 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 must also be generated. The actuation of differential braking causes increased strain on the tyres. When the vehicle moves in the lateral direction the positions of each tyre is longitudinal and they are not turned. It also shows that using front-wheel steering is more efficient.

The wheel force distribution strategy in the braking process depends on the construction of the brake system. During deceleration the load of wheels is modified due to the pitch dynamics of the vehicle. In order to improve pitch dynamics the braking of the front wheels must be stronger while the braking of the rear wheels must be reduced. The optimal distribution of brake forces can be determined according to the following form, see [15]:

$$F_r = -F_{fr} - \frac{mg\, l_2}{2h} + \sqrt{\frac{F_{fr}(l_1+l_2)\, mg}{h} + \left(\frac{mg\, l_2}{2h}\right)^2} \tag{5}$$

where F_r and F_{fr} are the wheel forces at the rear and at the front, respectively. The yaw moment of the brake also increases the brake forces of the vehicle, thus the yaw-moment should be distributed on one side of the vehicle between the front and rear wheels using (5). The distributed brake forces must be added to the brake forces coming from the distribution of decelerating longitudinal forces F_l. The forces must be monitored in the view of the momentary friction margin of the tire. It requires an estimation of the friction coefficient μ, see [16]. The measurement of the friction value requires the knowledge of the vertical load of the wheels, therefore the maximal longitudinal force of the wheels ($F_{i,max}$) is calculated and it is compared to the momentary longitudinal wheel forces (F_i). Note that the maximal longitudinal force depends on the maximal adhesion coefficient and the static and dynamic components of the vertical force at the wheel, i.e., the lateral and pitch dynamics:

$$F_{i,max} = \mu_{max}\{F_{z,stat} \pm ma_y h/2/L \pm m\dot{\psi} vh/b\}.$$

This calculation must be performed at all wheels and the highest rate of $\nu = F_i/F_{i,max}$ is selected. If a skidding incident is imminent the actuation of the brake moment must be reduced and it is replaced by the actuation of front wheel steering.

The third viewpoint in the analysis is the bandwidth of the actuation of front-wheel steering and differential braking. In case of front-wheel steering the inertia of the steering system effects a slower operation of the actuator. The dynamics of steering is described in the following form:

$$M_{st} = 2\vartheta\ddot{\delta} + M_{res} \tag{6}$$

where ϑ is the inertia of the wheel on the axle of steering, δ is steering angle, M_{st} is the moment, which rotates the wheels.

- M_{st} is controlled by a servo-power system according to the rack. $M_{res} = M_{gy} + M_{ch}$ is the sum of the steering resistances, which consists of two main components.
- In order to the steer the rotated wheels it is necessary to generate energy against the gyroscopic effect. It is formulated by using the following assumption:

$$M_{gy} = \frac{2J_w\dot{\xi}}{r_w}\dot{\delta}, \tag{7}$$

where J_w is the inertia of the wheel on the axle of rotation and r_w is the wheel radius.
- During steering the positions of wheels are modified. Since the axle of a steered wheel is skew the vertical position of the entire chassis also moves. Thus, it is necessary to generate energy to improve the lateral dynamics of the chassis:

$$M_{ch} = B\pi s_{susp}(\sin n)\; \delta, \tag{8}$$

where B is the wheel track, s_{susp} is the suspension stiffness and n is the angle between the road and the axle of wheel steering. Note, that n may be differ from the caster, if the axle of wheel steering is virtual.

The differential equation of the steering system, which is formulated in (6).. (8), can be transformed into a second-order proportional transfer function form (from M_{st} to δ):

$$G_{st}(s) = \frac{1}{\vartheta s^2 + \frac{2J_w\dot{\xi}}{r_w}s + B\pi s_{susp}(\sin n)} \tag{9}$$

Now, the bandwidth of steering and that of differential braking actuation can be compared. In case of differential braking the input of the model is the brake yaw moment, the output is the yaw-rate. In case of steering it is necessary to combine the bicycle model with the second-order steering model (9) in order to transform the steering angle into a steering moment. Thus the two models can be compared.

Both models depend on the velocity of the vehicle. The bandwidth of the actuation rate is analyzed as a function of velocity. Figure 2 shows that the bandwidth values change significantly with the change of velocity. According to the inertia of steering, the bandwidth of steering is lower at each frequency than the bandwidth of differential braking. The

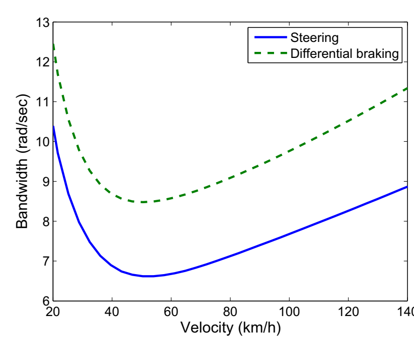


Fig. 2. Bandwidth values of actuators

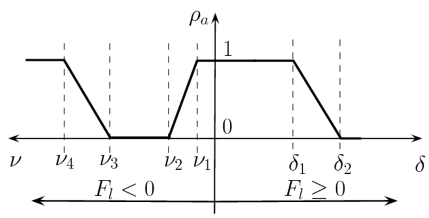
fast operation of actuators is an important feature mainly at high velocities. Therefore a design criterion is expressed by the dynamics of actuators. At higher velocities it is recommended to use differential braking, while at lower velocities the steering actuation is preferred for practical reasons.

The generation of the different actuators is based on the following weighting strategy. The weighting for the front wheel steering and for the brake yaw-moment are

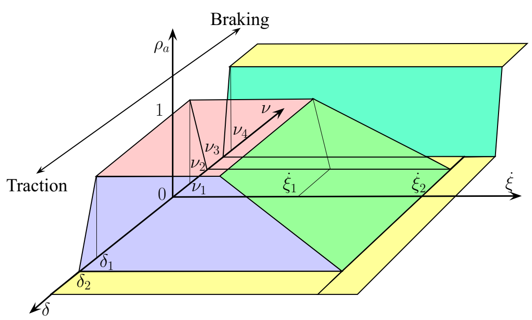
$$W_{act,\delta} = (1-\rho_a)/\delta_{max}, \tag{10}$$

$$W_{act,Mbr} = \rho_a / M_{brmax}, \tag{11}$$

where δ_{max} is determined by the constructional maximum steering angle and M_{brmax} is the maximum of brake yaw-moment. Weighting factor ρ_a is chosen according to the relationship between the brake yaw-moment and the front steering angle. This weighting factor must be chosen according to the the examined criteria. Figure 3(a) shows the characteristics of the weighting factor ρ_a as functions of δ and ν.



(a) Case $\dot{\xi} = 0$



(b) General case

Fig. 3. Selection of parameter ρ_a

The choice of ρ_a depends on several variables. In case of traction the front wheel steering angle δ determines the factor ρ_a. The value is reduced between δ_1 and δ_2, which represents the constructional criterion of the steering system. In case of braking the variable ν affects the factor ρ_a. In this interval differential braking is preferred for practical reasons. It requires an interval to reduce tire skidding and it also requires an interval to prevent chattering between steering and differential braking. Therefore four parameters are designed: ν_1 and ν_2 are used to prevent chattering and ν_3 and ν_4 are applied to prevent the skidding of tires. The value of ρ_a depends on the velocity of the vehicle as well. Figure 3(b) extends the results as a function of velocity. The effect of the velocity on the weighting factor ρ_a is the consequence of the interaction between the bandwidth values of steering and differential braking actuators. Two additional design parameters $\dot{\xi}_1$ and $\dot{\xi}_2$ are also introduced.

IV. LPV-BASED DESIGN OF LONGITUDINAL-LATERAL CONTROL

The control design is based on a weighting strategy, which is formalized through a closed-loop interconnection structure, see Figure 4. The selection of input and output

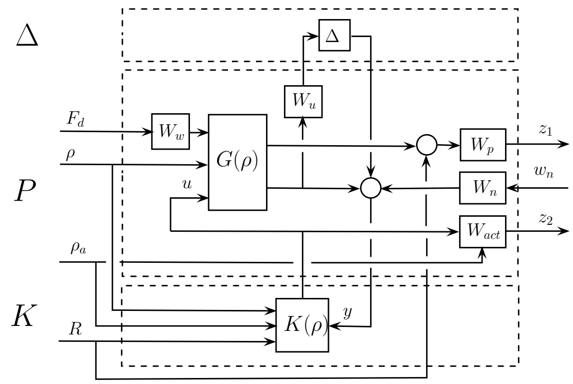


Fig. 4. Closed-loop interconnection structure

weighting functions is typically based on the specifications of disturbances and the performances. Usually the purpose of weighting function W_p is to define the performance specifications in such a way that a trade-off is guaranteed between them. They can be considered as penalty functions, i.e. weights should be large where small signals are desired and small where large performance outputs can be tolerated. The purpose of the weighting functions W_w and W_n is to reflect the disturbance and sensor noises. Δ block contains the uncertainties of the system, such as unmodelled dynamics and parameter uncertainty. The magnitude of the neglected dynamics is handled by a weighting W_u.

In the trajectory tracking problem two reference signals are introduced in order to guarantee the tracking of the road geometry. They are the reference velocity and reference lateral displacement $R = [\dot{\xi}_{ref} \; y_{v,ref}]^T$. The weighting function for performance specification is selected as a second-order proportional form: $W_p = \gamma_p(\alpha_2 s^2 + \alpha_1 s + 1)/(T_1 s^2 + T_2 s + 1)$, where $\alpha_1, \alpha_2, T_1, T_2$ are designed parameters. Similarly weighting functions W_w and W_n are also selected in a linear and proportional form. Note that although weighting functions are formalized in the frequency domain, their state-space representation forms are applied in the weighting strategy and in the control design.

Since the model to be used in the control design contains uncertain parameters (mass, inertia, cornering stiffness) and dynamic effects that are neglected, a robust control must be designed. The uncertainties of the model are represented by W_u and Δ. Design models exhibit high fidelity at lower frequencies, but they degrade rapidly at higher frequencies due to poorly-modelled or neglected effects. Thus, the weighting is selected as $W_u = \gamma_u(\alpha_u s + 1)/(T_u s + 1)$. The input scaling weight W_w normalizes the disturbances to the maximum expected values. W_n is selected as a diagonal matrix, which accounts for sensor noise models in the control design. The noise weights are chosen constant both for velocity and the lateral position.

The role of W_{act} is to guarantee wheel force distribution and limit actuator forces. Two weighting functions are applied, one to the brake yaw-moment and the other to the steering angle see (10) and (11). Since these weighting functions must be built into the performance of the actuators ρ_a is selected as a scheduling variable in the LPV design. In the following it is assumed that ρ_a is available. Its value depends on the adhesion coefficient and the longitudinal forces on the wheels. It is sufficient to estimate the actual adhesion coefficient within a relatively wide range, i.e., an estimation of an interval which fits the actual value is acceptable.

The control design is based on the LPV method that uses parameter-dependent Lyapunov functions, see [12], [17]. The quadratic LPV performance problem is to choose the parameter-varying controller $K(\rho)$ in such a way that the resulting closed-loop system is quadratically stable and the induced $\mathcal{L}_2$ norm from the disturbance and the performances is less than the value γ. The existence of a controller that solves the quadratic LPV γ-performance problem can be expressed as the feasibility of a set of Linear Matrix Inequalities (LMIs), which can be solved numerically. Finally, the state space representation of the LPV control $K(\rho)$ is constructed, see [13], [17]. When the controller has been synthesized, the relation between the state and the variable is used such that a nonlinear controller is obtained.

The purpose of the control design is to calculate the necessary longitudinal control force, front steering angle and brake yaw moment. The design of this upper level controller is based on the LPV method. Then the designed longitudinal force and brake yaw moment are distributed between the four wheels of the vehicle. Moreover, a third layer is also

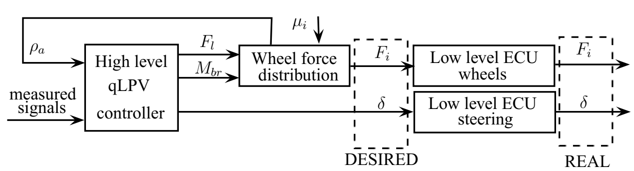
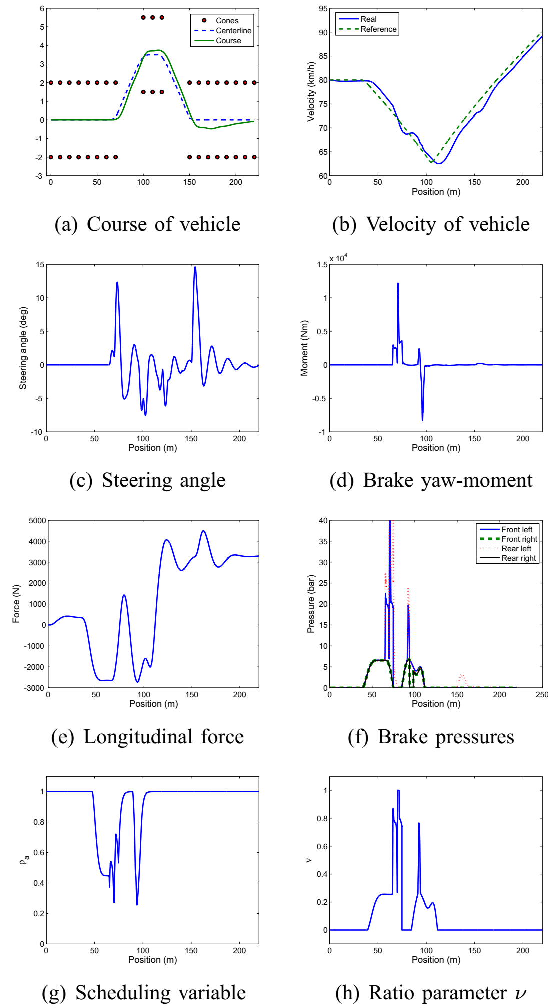


Fig. 5. Architecture of control system

necessary since the required control forces must be tracked by using a low-level controller. This controller transforms the wheel forces and the values of the steering angle into a real physical parameter of the actuator. These components are implemented by Electronic Control Units (ECUs). The structure of the multi-layer control system is illustrated in Figure 5.

V. Simulation Results

In the first example a double-lane-changing maneuver is analyzed. In this maneuver the vehicle must drive along a predefined corridor without knocking down the buoys, which are located along the edge of the corridor, see Figure 6(a). The initial velocity of the vehicle is 80 km/h, but



(a) Course of vehicle (b) Velocity of vehicle

(c) Steering angle (d) Brake yaw-moment

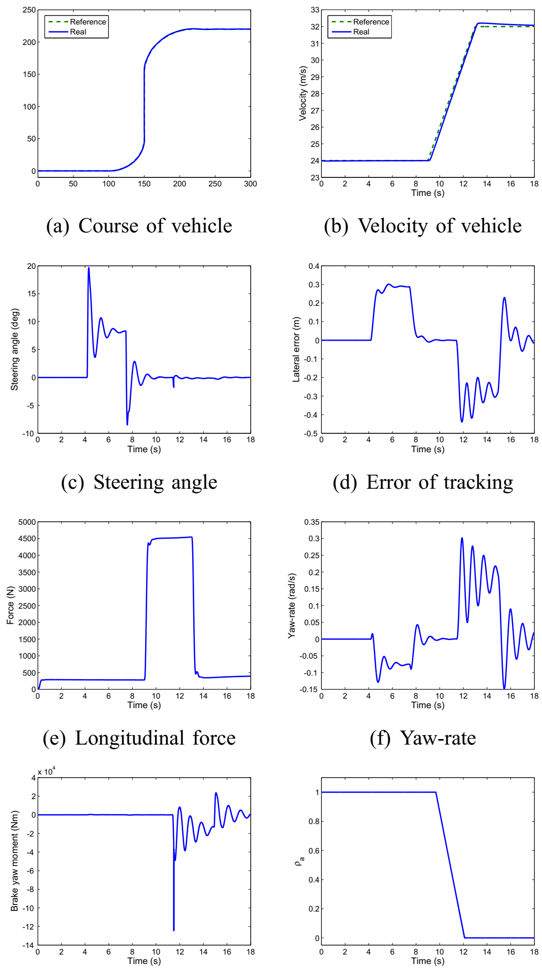
(e) Longitudinal force (f) Brake pressures

(g) Scheduling variable (h) Ratio parameter ν

Fig. 6. Double lane change maneuver

this velocity is modified along the course. The proposed controller must guarantee the tracking of both the longitudinal velocity and the lateral displacement. Figure 6(b) shows that the controller tracks the velocity accurately. In this simulation example there is a powerful braking, therefore the ratio parameter ν increases during braking, see Figure 6(h). According to this effect, the scheduling variable ρ_a decreases, see Figure 6(g). The reason for this change is the shape of the characteristics of ρ_a weighting factor. The change of ρ_a influences both the brake yaw moment and front wheel steering, see Figure 6(d) and Figure 6(c). It is shown that when ρ_a decreases the actuation of brake yaw moment increases. Therefore the braking pressures on the wheel cylinders also increase and they differ on the left and right hand sides. Above the critical value of skidding parameter (ν_3) the ρ_a scheduling variable increases to prevent the skidding of wheels. In this way the actuation of the brake yaw moment is decreased while steering is increased. The trade-off between the steering and the brake yaw moment guarantees that the wheels will not skid during the strong braking.

In the second example cornering maneuvers with different cornering radiuses are analyzed, see Figure 7(a). The initial



(a) Course of vehicle (b) Velocity of vehicle

(c) Steering angle (d) Error of tracking

(e) Longitudinal force (f) Yaw-rate

(g) Brake yaw-moment (h) Scheduling variable

Fig. 7. Cornering maneuver

velocity of the vehicle is 24 m/s (86 km/h), which is increased by a significant acceleration to 32 m/s (115 km/h), see Figure 7(b). Control inputs are generated in order to reduce the tracking error. The tracking of the velocity and the minimization of the error in the lateral displacement require strong brake forces on the wheels, see Figure 7(d) and 7(g). It is shown that the value ρ_a in Figure 7 (h) decreases because of the increased velocity. It indicates that during acceleration differential braking is preferred because of its faster dynamics. The actuations of steering and differential braking are shown in Figure 7(c) and Figure 7(g).

VI. Conclusion

The paper has proposed the design of a trajectory tracking system which is able to track road geometry with a predefined reference velocity. The actuators of the control system are the front steering, the brake yaw moment and the longitudinal force, thus the lateral and longitudinal dynamics are combined. The paper extends the control design with a new actuator selection procedure. This method is built into a weighting strategy applied to the control design. Although the selection of the actuator is usually performed by using practical considerations, in the paper a theory-based method is used in the weighting strategy. The control design is based on the robust optimal LPV method, in which both performance specifications and model uncertainties are taken into consideration.

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