

Robust Control Design for the Integration of Steering and Torque Vectoring Using a Variable-Geometry Suspension System

Balázs Németh, Péter Gáspár, Dániel Fényes, József Bokor

Abstract—The paper proposes a robust control design for the integration of steering and torque vectoring using a variable-geometry suspension system. Torque vectoring control is based on independent steering/driving wheel systems, which is an actual challenge in the field of electric vehicles. Simultaneously, the steering angle is generated by the variable-geometry suspension system by changing the camber angle and the position of the wheel-road contact independently at each wheel at the front. The integration of torque vectoring and wheel tilting control is in the focus of the paper. The modeling of the lateral and the steering dynamics of the vehicle leads to interconnection models. A hierarchical structure of both the modeling and the control design is formed. The control design itself is based on the Linear Parameter-Varying (LPV) method. The robustness of the control system can be achieved by estimating and considering the uncertainties in the different layers of the hierarchical structure. In the LPV-based control design the reconfiguration strategy is also applied.

I. INTRODUCTION AND MOTIVATION

One of the main constructional benefits of in-wheel vehicles is the space-efficient passenger cabin design, which is essential for small city cars. The in-wheel electric drive poses new challenges in generating the differential yaw moment of the vehicle. The independent steering control for the rear wheels to modify the toe angle is presented by [1], [2]. The independent, fast and precise torque generation of the hub motors lends torque vectoring capability to the vehicle with which manoeuvrability can be enhanced significantly, see [3], [4], [5]. By knowing the characteristics of the in-wheel motors and the hydraulic brake system, energy optimal torque distribution and high efficiency regenerative braking can be implemented, as proposed by [6], [7]. A fault-tolerant control system designed to accommodate hub motor faults by automatically reallocating the control effort among other healthy wheels is proposed by [8], [9].

In the paper the manoeuvres are solved by the variable-geometry suspension system, see [10], [11]. There are several application possibilities for the variable-geometry suspensions. A rear-suspension active toe control for the enhance-

ment of driving stability is proposed by [12]. The active tilt control system which assists the driver in balancing the vehicle and performs tilting in the bend is an essential part of a narrow vehicle system, see [13]. The novelty of the paper is the application of the variable-geometry suspension in both steering and trajectory tracking. It is able to generate the steering of the vehicle by independent wheel camber angles and wheel steering and create differential yaw moment by harmonizing the longitudinal forces.

In the paper a new conception of the integrated steering and driving is proposed, which is based on the coordination of steering and torque vectoring using the variable-geometry suspension system. The purpose of the variable-geometry suspension control is the modification of the geometry, which results in a change in the camber angle and the position of the wheel-road contact. Thus, the scrub radius is also modified. Note that scrub radius is a virtual distance, which can be considered the radius of wheel steering rotation. Consequently, a longitudinal force on the wheel creates a moment on the wheel and, thus, realizes steering angle. Therefore, the appropriate coordination of the wheel tilting and the driving can result in improved manoeuvring of the vehicle. The proposed mechanism of steering angle modification is a new solution, which is suitable for the in-wheel electric motors. In an earlier paper the modeling of a variable-geometry suspension based wheel steering system was proposed, see [14]. Furthermore, in [15] the integration possibility of the variable-geometry suspension and driving is examined. However, in these preliminary results several factors have not been taken into consideration, e.g. the robustness of the global system and the reconfiguration of the actuator interventions.

The contribution of the work is a robust Linear Parameter-Varying (LPV) control of the system, which is able to coordinate wheel tilting, steering and the longitudinal driving/braking forces on the wheels through a reconfiguration strategy. Thus, the manoeuvring and the trajectory tracking performance of the vehicle are improved.

The structure of the paper is the following. The modeling of the lateral dynamics, incorporating the steering dynamics is presented in Section II. The motivation and the architecture of the hierarchical modeling and control are proposed in Section III. The robust LPV control design of the controllers and the controller integration are presented in Section IV. The operation of the LPV control and the reconfiguration strategy of the integrated control system are illustrated through the CarSim simulation environment in Section V.

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II. MODELING OF THE LATERAL VEHICLE DYNAMICS

The scheme of the variable-geometry suspension is shown in Figure 1. The actuator is incorporated in the suspension between the wheel hub and the wheel. It is able to generate an active torque M_{act} to tilt the wheel. However, it also has a counter effect $-M_{act}$ on the hub. In the McPherson construction the suspension is able to rotate around the connection point of the chassis. Moreover, the arm connects the hub and the chassis with joints, which are able to guarantee the rotation and the motion of the suspension.

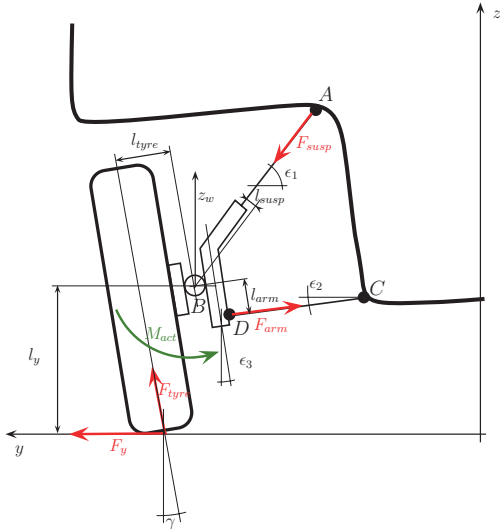


Fig. 1. Scheme of the suspension construction

The goal of the modeling of the lateral vehicle dynamics is to formulate the relationships between the wheel camber angle γ , the longitudinal wheel force $F_{l,i}$ and the vehicle motion. Since $F_{l,i}$, $i \in \{l, r\}$ has an effect on manoeuvring through torque vectoring and γ through the steering angle, both of the interventions in the dynamical equations are incorporated.

The lateral motion of the vehicle is described by the following equations, see [16]

$$J\ddot{\psi} = C_1 l_1 \alpha_1 - C_2 l_2 \alpha_2 + M_d \quad (1a)$$

$$mv(\dot{\psi} + \dot{\beta}) = C_1\alpha_1 + C_2\alpha_2 \quad (1b)$$

where m is the vehicle mass, J is the yaw-inertia of the vehicle, l_1 and l_2 are geometric parameters, C_1 and C_2 are cornering stiffness values, $\dot{\psi}$ is the yaw rate of the vehicle, β is the side-slip angle and M_d is the differential yaw moment. The side-slip angles of the front and rear axles α_1 and α_2 are expressed: $\alpha_1 = \delta - \beta - \frac{\dot{\psi} l_1}{v}$ and $\alpha_2 = -\beta + \frac{\dot{\psi} l_2}{v}$. In the single-track model (1) δ represents a common steering wheel, computed as the average of the left and right steering wheel angles: $\delta = (\delta_l + \delta_r)/2$.

First, the relation between the longitudinal force on the wheels and M_d is presented. $F_{l,i}$ influence not only the lateral motion of the vehicle, but also the longitudinal dynamics, e.g. the velocity profile. Moreover, the cruise control

of vehicle also generates longitudinal force requirement F_0 , which is an external signal for the lateral control system. In the distribution of M_d the signal F_0 must not be changed. Thus, the longitudinal forces on either side can be calculated from the following expressions:

$$F_{l,l} = F_0 + \frac{M_d}{b_f}, \quad F_{l,r} = F_0 - \frac{M_d}{b_r} \quad (2)$$

where b_f is the track of the vehicle on the front axle.

Second, the interconnection between γ and δ is formulated. The steering rotation of a wheel is described by the following dynamical equation

$$\ddot{\delta}_i = \frac{r_{\delta,i} F_{l,i}}{J_{\delta,i}}, \quad i \in \{l, r\}. \quad (3)$$

where $r_{\delta,i}$ is the scrub radius, $F_{l,i}$ is the longitudinal traction/braking force on the wheel, $J_{\delta,i}$ is the inertia of the steering system for one wheel. In practice, the relationship between $r_{\delta,i}$ and γ_i can be formulated linearly: $r_{\delta,i} = \varepsilon\gamma_i$, where ε is a construction parameter. The steering rotations of the wheels are extended to the lateral motion of the vehicle.

$$\ddot{\delta}_i = \frac{\varepsilon F_{l,i}}{J_{\delta,i}} \gamma_i, \quad i \in \{l, r\}. \quad (4)$$

These relations show the interconnections between the control interventions of the systems, such as the steering angle, the differential yaw moment and the longitudinal force. Through the mechanical connections the possibilities in the integration of steering and torque vectoring control are introduced.

III. HIERARCHICAL STRUCTURE OF THE VEHICLE SYSTEMS

In this paper the interconnection between δ_i and $F_{l,i}$ is solved by a hierarchical design of the integration of steering and torque vectoring by using the variable-geometry suspension system. The interconnected system is separated into three subsystems, such as the lateral dynamics of the vehicle and two steering wheel dynamics. The lateral dynamics of the system is described as

$$\dot{x}_{lat} = A_{lat}x_{lat} + B_{lat}u_{lat} \quad (5)$$

where $x_{lat} = [\dot{\psi} \quad \beta]^T$, $u_{lat} = [\delta \quad M_d]^T$ and $\delta = (\delta_l + \delta_r)/2$. The state-space representation of the steering wheel dynamics is formed as

$$\dot{x}_{st,i} = A_{st}x_{st,i} + B_{st,i}(F_{l,i})u_{st,i} \quad (6)$$

where $x_{st,i} = [\dot{\delta}_i \quad \delta_i]^T$ and $u_{st,i} = \gamma_i$.

Applying the mentioned separation one linear system (5) and two LPV systems (3) are yielded. The external varying parameter of (3) are the $F_{l,i}$ forces, computed from M_d using (2). The advantages of the separation are the following:

- The resulting systems are smaller, therefore the control design has reduced complexity.
- The steering wheel system (3) is an LPV system with an external parameter variation $F_{l,i}$. Thus, problems of the qLPV control design are eliminated.

- The interconnected system is observable if the measured signals are $y = [\dot{\psi} \ \dot{\delta}_l \ \dot{\delta}_r]^T$. However, in practice the measurement of $\dot{\delta}_i$ is difficult. Using the separation it is sufficient to measure $\dot{\psi}$ and δ_i signals.
- The LPV system contains one scheduling parameter $\rho_{st,i} = F_{l,i}$, which is an external signal and results from the distribution of M_d .

Control architecture of the systems

The trajectory tracking control design fits into the separation of the lateral and steering dynamics. Two different controllers must be designed, such as a high-level control for the lateral dynamics and low-level control for the steering wheel dynamics. The illustration of the hierarchy is found in Figure 2.

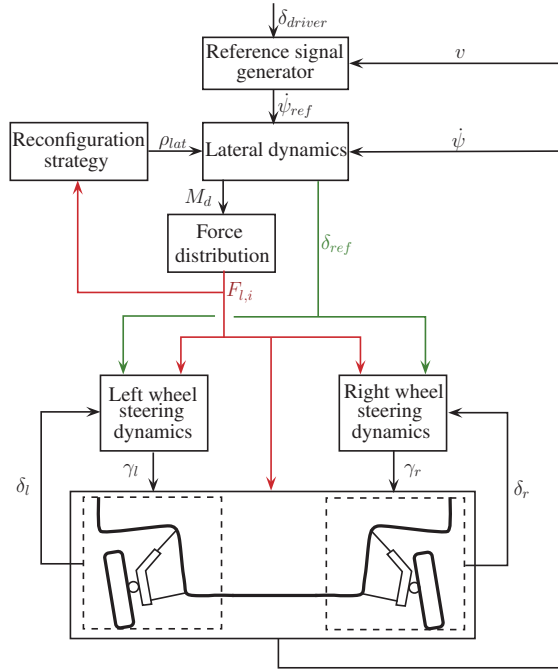


Fig. 2. Architecture of the system hierarchy

The reference trajectory is designed through the velocity of the vehicle and the cornering intention of the driver, estimated from the steering wheel angle δ_{driver} . It is generated by the driver himself in a driver assistance system. The reference yaw rate can be calculated by a first-order transfer function from the steering wheel angle of the driver δ_{driver} , see [16].

The lateral dynamic control, i.e., the high-level control in the hierarchical structure, computes the required control input $u_{lat} = [\delta_l \ \delta_r \ M_d]^T$. The differential yaw moment is distributed to the longitudinal wheel forces, thus the results of it are $F_{l,i}$. The control inputs $\delta_i, i \in \{l, r\}$ are the reference signals of the steering control, which results in the wheel camber angles γ_i . Moreover, these computations require longitudinal forces $F_{l,i}$. The longitudinal forces have a high impact on the low-level control. If $F_{l,i} = 0$ then the matrix of the control input γ_i is $B_{st,i} = [0 \ 0]^T$, which

means that the wheel camber angle is ineffective on the system. As a result only the M_d signal improves the motion of the vehicle, which can lead to the degradation of trajectory tracking. Therefore, a reconfiguration strategy is built in the hierarchical control design, by which the intervention of M_d is improved and the γ_i actuation is reduced to avoid the performance degradation. The reconfiguration strategy is also built into the control design.

IV. ROBUST CONTROL DESIGN OF THE SYSTEM

The purpose of this section is to propose the LPV-based control design of steering and torque vectoring. In the following, the control design on the steering dynamics together with the formulation of the uncertainty of the system are shown. Furthermore, the integrated design of the control interventions and the reconfiguration strategy are presented.

A. Design of suspension based steering control and uncertainty modeling

The control design of the steering dynamics is based on the state-space formulation (3). The purpose of the steering control is to guarantee the tracking of the reference steering signal $\delta_{ref} = \delta$, which is computed through the lateral controller:

$$z_{st,i,1} = \delta_{ref} - \delta_i, \quad |z_{st,i,1}| \rightarrow \min \quad (7)$$

Moreover, the performance $z_{st,1}$ must be reached using minimum control input:

$$z_{st,i,2} = \gamma_i, \quad |z_{st,i,2}| \rightarrow \min \quad (8)$$

The performances are formed in a vector, such as $z_{st,i} = [z_{st,i,1} \ z_{st,i,2}]^T$. The state-space representation of the steering dynamics, which incorporates the performances and the measurement, is the following:

$$\dot{x}_{st,i} = A_{st}x_{st,i} + B_{st,i}(\rho_{st,i})u_{st,i} \quad (9a)$$

$$z_{st,i} = C_{st,1}x_{st,i} + D_{st}u_{st,i} \quad (9b)$$

where the measured signal is $y_{st} = \delta_i$.

Thus, it is necessary to design controllers for both the left and the right wheels. Since matrix $B_{st,i}(\rho_{st,i})$ depends on a scheduling parameter, the LPV control design method is applied in the following. The LPV design is based on a weighting strategy, which is formulated through a closed-loop interconnection structure of (9), see Figure 3. The selection of input and output weighting functions is typically based on the specifications of disturbances and the performances. The purpose of weighting functions $W_{st,1}$ and $W_{st,2}$ are to define the performance specifications in such a way that a trade-off is guaranteed between $z_{st,i,1}$ and $z_{st,i,2}$. 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 weight of $z_{st,i,1}$ is chosen in the form $W_{st,1} = \frac{A_{st,1}}{T_{st,1}s+1}$, which scales the admissible tracking error. The actuation γ_i is scaled with the function in the form $W_{st,2} = A_{st,2}$, which determines the amplitude of the control signal. The aim of the

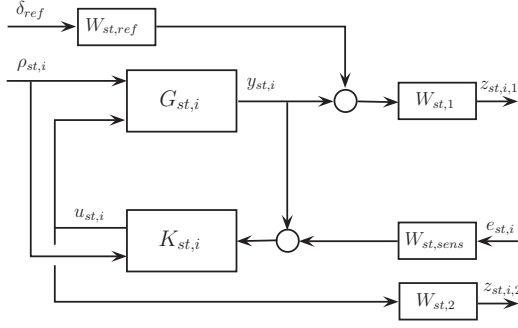


Fig. 3. Closed-loop interconnection structure of the steering system

function $W_{st,ref} = A_{st,3}$ is to scale the reference signal δ_{ref} . Furthermore, $W_{st,sens} = \frac{A_{st,4}}{T_{st,4}s+1}$ incorporates the scaling of the sensor noise, such as the maximum amplitude and the maximum frequency of the steering angle measurement error. The signals $e_{st,i}$ and δ_{ref} are handled as disturbances in the system, compressed in a vector $w_{st,i} = [e_{st,i} \ \delta_{ref}]^T$.

The quadratic LPV performance problem is to choose the parameter-varying controller $K_{st,i}$ in such a way that the resulting closed-loop system is quadratically stable and the induced $\mathcal{L}_2$ norm from the disturbance $w_{st,i}$ and the performance $z_{st,i}$ is less than a predefined small value Γ , see [17], [18], [19].

In the hierarchical structure the purpose of the steering controller is to guarantee the steering angle, which is required by the high-level lateral controller. Therefore, the accuracy of the low-level steering control has an important role in the stability and the performance of the entire system. The aim of the following analysis is to formulate the maximum tracking error of the low-level control. The result of the analysis is incorporated in the design of the high-level robust control. In this way the interconnection in the hierarchy is guaranteed.

The process of the analysis is the following. Several simulations are performed using different initial values $x_{st,i}(0)$ and δ_{ref} . The intervals of the initial values are $x_{st,i}(0) = \pm x_{st,max}$ and $\delta_{ref} = \pm \delta_{max}$. The intervals of $x_{st,i}$ and δ_{ref} are gridded with ϵ_{st} and ϵ_{ref} samplings, respectively. Altogether $\left(2 \frac{x_{st,max}}{\epsilon_{st}} + 1\right) \cdot \left(2 \frac{\delta_{max}}{\epsilon_{ref}} + 1\right)$ simulation scenarios are performed. In the scenarios the low-level control must guarantee achieving δ_{ref} from the $x_{st,i}(0)$. In each case the maximum tracking error during the scenario is calculated. The statistical results of the analysis are illustrated in Figure 4. It can be stated that the error of the wheel steering is quite small, the maximum of the error is below 0.04° , while its probability is approximately 3%.

The results of the simulation-based statistical analysis are used for the modeling of uncertainties in a multiplicative form. Since in the robust control design the worst case scenario is considered, the maximum tracking error of the previous analysis is used in the formulation. The consideration of the multiplicative uncertainty form in the control structure is found in Figure 5. Since the controller $K_{st,i}$ has inaccuracy, and $G_{st,i}$ has its own dynamics, there is a

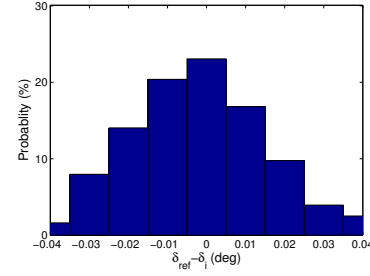


Fig. 4. Impreciseness of the steering control

difference between δ_{ref} and δ_i , see Figure 5(a). The relation between δ_{ref} and δ_i , using the upper bound of the previous analysis, is formulated in the transfer function $1 + W_\Delta$, see Figure 5(b). W_Δ represents the uncertainty on the control input signal, such as

$$W_\Delta = \frac{\alpha_{\Delta,2}s^2 + \alpha_{\Delta,1}s + \alpha_{\Delta,0}}{T_{\Delta,2}s^2 + T_{\Delta,1}s + T_{\Delta,0}}. \quad (10)$$

W_u scales the bound of input multiplicative uncertainty, where $\alpha_{\Delta,2}$, $\alpha_{\Delta,1}$, $\alpha_{\Delta,0}$ and $T_{\Delta,2}$, $T_{\Delta,1}$, $T_{\Delta,0}$ are design parameters. The ratio of $\alpha_{\Delta,2}/T_{\Delta,2}$ is selected according to Figure 4 and the ratio of $\alpha_{\Delta,0}/T_{\Delta,0}$ is chosen a very small value.

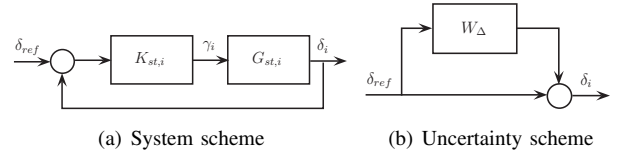


Fig. 5. Formulation of the steering uncertainty

B. Integrated design of suspension and torque vectoring control

The goal of the integrated control design is to guarantee trajectory tracking and the robust stability of the entire system through the computation of the control inputs δ_{ref} and M_d . The state-space representation of the high-level system is (5).

The vehicle must follow a reference yaw-rate signal $\dot{\psi}_{ref}$. The performance is defined as

$$z_{lat,1} = \dot{\psi}_{ref} - \dot{\psi}, \quad |z_{lat,1}| \rightarrow \min \quad (11)$$

where $\dot{\psi}_{ref}$ is computed using the velocity and the steering wheel angle of the driver, see [16]. Moreover, the performance $z_{lat,1}$ must be reached using minimum control input $u_{lat} = [\delta_l \ \delta_r \ M_d]^T$. Thus, the next two performances are defined:

$$z_{lat,2} = [\delta_l \ \delta_r]^T, \quad |z_{lat,2}| \rightarrow \min \quad (12a)$$

$$z_{lat,3} = [M_d]^T, \quad |z_{lat,3}| \rightarrow \min \quad (12b)$$

The performances are compressed in a vector $z_{lat} = [z_{lat,1} \ z_{lat,2} \ z_{lat,3}]^T$. The state-space representation of

the system for the control design which incorporates the performances and the measurement is the following:

$$\dot{x}_{lat} = A_{lat}x_{lat} + B_{lat}u_{lat} \quad (13a)$$

$$z_{lat} = C_{lat,1}x_{lat} + D_{lat}u_{lat} \quad (13b)$$

and the measured signal is $y_{lat} = \dot{\psi}$.

Although the system (13) is linear, a trade-off between the control inputs δ_i and M_d is reached by a scheduling variable ρ_{lat} . Therefore, in Figure 6 the augmented plant for the LPV design is illustrated. In the architecture three performance weighting functions are used. While $W_{lat,1}$ scales the admissible error on the yaw-rate tracking, the weights $W_{lat,2}(\rho_{lat})$ and $W_{lat,3}$ have impact on the actuation.

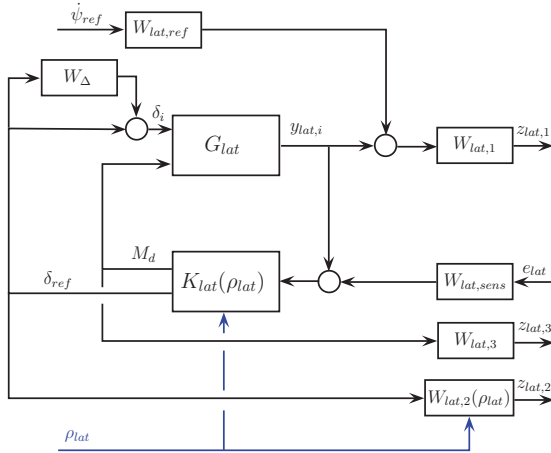


Fig. 6. Closed-loop interconnection structure of the lateral dynamics

The weight on the steering angle is selected as parameter-dependent, in the following form: $W_{lat,2}(\rho_{lat}) = \frac{\rho_{lat}}{A_{lat,2}}$, where $\rho_{lat} \in [\rho_{lat,min}, \rho_{lat,max}]$ is a scheduling variable of the system. The role of ρ_{lat} is to influence the actuation of the variable-geometry suspension through the steering intervention. In the case of small $|F_{l,i}|$ values the camber angle has a low impact on the steering angle and the vehicle dynamics. Therefore, the vehicle motion must be influenced through the differential yaw moment instead of δ_{ref} . The reconfiguration is guaranteed by the selection of ρ_{lat} . If $\rho_{lat} = \rho_{lat,min}$, then $W_{lat,2}(\rho_{lat,min})$ has a small value, which results in the increase of δ_{ref} . Similarly, if $\rho_{lat} = \rho_{lat,max}$, then $W_{lat,2}(\rho_{lat,max})$ has a high value, which reduces the steering actuation. The weight on M_d is selected a constant value, such as $W_{lat,3} = A_{lat,3}$. Thus, the trade-off between the actuation of δ_{ref} and M_d is determined by ρ_{lat} .

In the augmented plant the weight W_{Δ} is incorporated, which represents the multiplicative uncertainty of the low-level, see (10). Moreover, $W_{lat,ref}$ scales the reference yaw-rate and $W_{lat,sens}$ is the weight on the yaw-rate measurement. The signals e_{lat} and $\dot{\psi}_{ref}$ are handled as disturbances in the system, compressed in a vector $w_{lat} = [e_{lat} \ \dot{\psi}_{ref}]^T$.

The quadratic LPV performance problem is to choose the parameter-varying controller K_{lat} in such a way that

the resulting closed-loop system is quadratically stable and the induced $\mathcal{L}_2$ norm from the disturbance w_{lat} and the performance z_{lat} is less than a predefined small value Γ , see [17], [18], [19].

The reconfiguration strategy in the integrated LPV control is based on the selection of ρ_{lat} . The decision concerning the value of the scheduling variable is based on a function, illustrated in Figure 7. Since the low values of $|F_{l,i}|$ have a disadvantageous effect on the vehicle dynamics, $\rho_{lat,max}$ is selected if $|F_{l,i}| < F_{min}$, where F_{min} and F_{max} are design parameters. If $|F_{l,i}| > F_{max}$ then the actuation of the steering is preferred. The role of the section between F_{min} and F_{max} is to avoid the chattering in the steering signal.

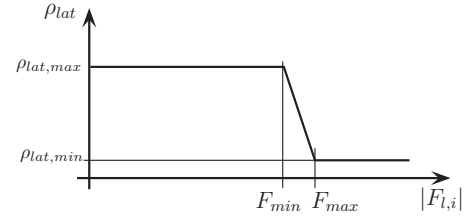


Fig. 7. Selection of ρ_{lat} value

V. SIMULATION EXAMPLES

The goal of the simulation scenario is to show the efficiency of the actuation integration and the operation of the reconfiguration strategy. In the simulation the previous vehicle is driven along a section of the Michigan Waterford Hill course, see Figure 8.

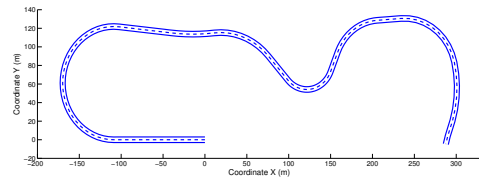


Fig. 8. Course of the vehicle

The track in the scenario contains several bends, therefore the velocity of the vehicle also varies, as shown in Figure 9(a). It results in the variation of the $F_{l,i}$ forces, see Figure 9(b). Although the longitudinal forces decrease below $F_{min} = 200N$ and $F_{max} = 1000N$, the tracking of the yaw-rate is accurate, as illustrated in Figure 9(c),(d).

The actuation of the steering and the differential yaw-moment are shown in Figure 10(a) and Figure 10(b), respectively. The resulting wheel camber angles are illustrated in Figure 10(c). Although γ_i of the wheels are small, these signals are sufficient to generate the required δ_{ref} value together with the high $F_{l,i}$ forces. Figure 10 shows the scheduling variable ρ_{lat} . This signal guarantees the reconfiguration strategy of the interventions. For example, in the section 420m...470m the longitudinal forces have a low value, which effect reduces the impact of γ_i on the steering.

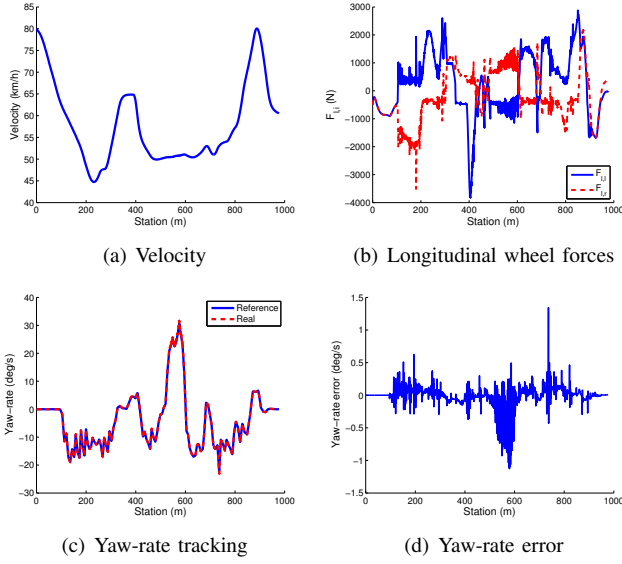


Fig. 9. Signals of the trajectory tracking

In this section ρ_{lat} is selected a high value, which results in the reconfiguration of the steering to torque vectoring control. Thus, δ_{ref} and γ_i have small values, while M_d significantly increases. The yaw-rate tracking of the vehicle is guaranteed by this reconfiguration.

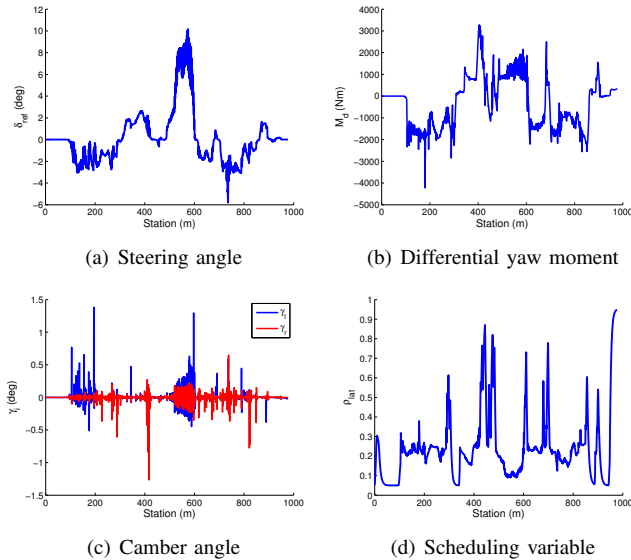


Fig. 10. Control signals and the scheduling variable

VI. CONCLUSIONS

In the paper a robust control design for the integration of steering and torque vectoring using the variable-geometry suspension system has been proposed. Torque vectoring control is based on the independent steering/driving wheel systems, while the steering angle is generated by the variable-geometry suspension system by modifying the camber angle independently at each wheel at the front. A hierarchical structure of both the modeling and the control design is

formed. The robustness of the control system can be achieved by estimating and considering the uncertainties in the different layers of the hierarchical structure. Moreover, the reconfiguration strategy is based on the LPV control design.

The operation of the variable-geometry suspension system is illustrated through simulation examples using a high-fidelity software for vehicle dynamics. The simulation examples show that the reference yaw rate tracking can be handled by integrating steering and the differential yaw moment. The operation of the reconfiguration strategy is also illustrated.

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