

Trajectory tracking based on independently controlled variable-geometry suspension for in-wheel electric vehicles

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Abstract—The paper presents a new variable-geometry suspension system which is applied in-wheel electric vehicles. It is able to realize the steering of the vehicle by independent wheel camber angles and wheel steering and create differential yaw moment by harmonizing the longitudinal forces. In order to perform the trajectory tracking of the vehicle the control signals are the virtual signals, such as differential yaw moment and steering by wheel camber angles. In the suspension system two physical active torques are realized on either side of the front axle and longitudinal forces are realized by in-wheel motors. The control design of the variable-geometry suspension system is based on a hierarchical structure. The purpose of the high-level controller is to calculate the virtual control inputs based on the performance specifications for the road trajectory. The purpose of the low-level controller is to realize the physical active torque of the wheels and track the required differential yaw moment.

I. INTRODUCTION AND MOTIVATION

Due to the growing demand for environmentally-friendly and economical transportation, in-wheel electric vehicles receive increasing attention by automotive companies and researchers as well. These vehicles have several advantages and very few drawbacks compared to conventional vehicles with internal combustion engines. The compact size of in-wheel motors and the absence of regular drive train components enable space-efficient passenger cabin design, also resulting in reduced overall weight.

Recent studies of in-wheel vehicles focus on exploiting the specific properties of the hub motors. Several papers focus on the lateral control of in-wheel vehicles, see [1], [2]. High performance wheel slip control is presented by several authors, see [3], [4]. Rollover avoidance methods are proposed by [5]. A fault-tolerant control system designed to accommodate hub motor faults by automatically reallocating the control effort among other healthy wheels is proposed by [6], [7]. 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 [8]. The analysis of the independent

wheel-steering system for heavy vehicles is found in [9]. An indirect power steering called differential drive torque assisted steering is proposed by [10]. A fault-tolerant control for a four-wheel independently actuated electric vehicle is proposed by [11].

In this paper a new solution to independent steering is proposed, which is based on the variable-geometry suspension system. The aim of the 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. Consequently, a longitudinal force on the wheel creates a moment on the wheel and, thus, realizes steering angle. The proposed mechanism of steering angle modification is a new solution, which is suitable for the in-wheel electric motors. There are further application possibilities of the variable-geometry suspensions. A rear-suspension active toe control for the enhancement of driving stability is proposed by [12]. The active tilt control system which assists the driver in balancing the vehicle 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 suspension system two controllers must be realized and integrated. An active torque for a wheel is designed in order to change the camber angle and perform wheel steering. A differential yaw moment is also designed in order to perform the trajectory tracking of the vehicle. In an earlier paper the control design of an independent wheel steering system for the front wheels was proposed, see [14].

The control design of the variable-geometry suspension system is based on a hierarchical structure. In the high level the virtual control inputs must be designed in order to meet the performance specifications concerning the road trajectory. In the low level the physical active torques of the wheels are realized in order to modify the camber angles of the wheels and the longitudinal forces are realized in order to track the required differential yaw moment.

The structure of the paper is the following. The independent steering model incorporating the vertical and lateral vehicle systems is presented in Section II. The robust control design of the integration and the hierarchical structure of the architecture are proposed in Section III. The operation of the control system is illustrated through the CarSim simulation environment in Section IV. Finally, Section V summarizes the contribution of the paper.

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II. MODELING OF THE VARIABLE-GEOMETRY SUSPENSION SYSTEM

In the variable-geometry suspension system two control tasks must be performed. In the first task the active torque is generated in order to modify the wheel position and the orientation of the wheel. Thus, the wheel camber angle and the scrub radius of the suspension vary. In the second task the differential yaw moment is designed by using individual in-wheel motors in order to perform the trajectory tracking of the vehicle.

A. Modeling of the relationship between the wheel camber angle and the active torque

The goal of the variable-geometry suspension is to perform the wheel camber angle and the scrub radius modification. The camber angle results in a lateral force on the tire-ground contact. Since the longitudinal force has a rotatory effect on the wheel, the scrub radius of the wheel influences the steering dynamics of the wheel. Therefore, a lateral force from the wheel steering through the scrub radius modification results. Two actuation torques on either side of the first axle realize the steering of the wheels at the front axle around the center of the vertical axis of the vehicle.

It is necessary to find a suspension construction by which the lateral force generation of the camber and the scrub radius is in coordination. The McPherson construction is used to perform the motion of the wheel. The actuator is incorporated in the suspension between the wheel hub and the wheel. It is able to generate an active torque M_{act} around B 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 A of the chassis. Moreover, the arm connects the hub D and the chassis C with joints, which are able to guarantee the rotation and the motion of the suspension. The scheme of the variable-geometry suspension is shown in Figure 1.

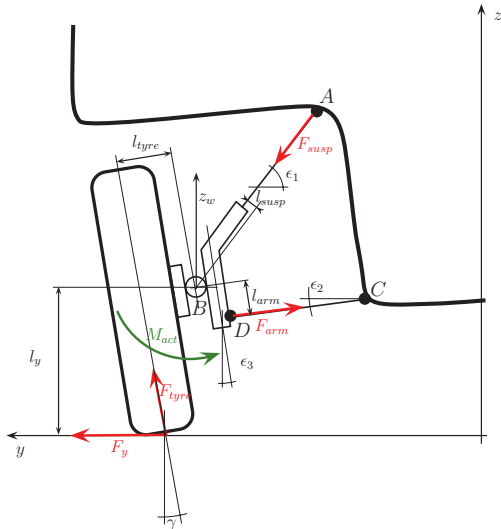


Fig. 1. Scheme of the suspension construction

The dynamic equations of the variable-geometry suspension system consist of the vertical motion of the wheel hub z_w and the tilting motion of the wheel γ .

$$m_{susp}\ddot{z}_w = -F_{susp} \sin \epsilon_1 + F_{tyre} \cos \gamma + F_{arm} \sin \epsilon_2 \quad (1a)$$

$$J_w\ddot{\gamma} = M_{act} - F_{tyre}l_{tyre} - F_y l_y \quad (1b)$$

where m_{susp} is the mass of the quarter-car, J_w is the wheel inertia and the arm of the lateral tire force is $l_y = r_w \cos \gamma - l_{tyre} \sin \gamma$, in which r_w and l_{tyre} are construction parameters.

The forces in equation (1) influence the motion of both the suspension and the wheel. The force of the suspension compression and damping F_{susp} is formulated as

$$F_{susp} = s_{susp} \left(\frac{z_w + z_{w,0}}{\sin \epsilon_1} \right) + d_{susp} \frac{\dot{z}_w}{\sin \epsilon_1} \quad (2)$$

where s_{susp} and d_{susp} are the stiffness and damping coefficients, $z_{w,0}$ is the joint position, resulting from the static suspension compression. The force from the tire compression F_{tyre} is formed, see [15]:

$$F_{tyre} = s_{tyre} \frac{(r_w \cos \gamma - l_{tyre} \sin \gamma - r_w - z_w) + z_{tyre,0}}{\cos \gamma} \quad (3)$$

where s_{tyre} is the tire stiffness, r_w is the wheel radius and $z_{tyre,0}$ is the static compression of the tire. The force of the active control F_{arm} is expressed by the following form:

$$F_{arm} = \frac{M_{act} - F_{susp}l_{susp}}{l_{arm}} \quad (4)$$

Thus, it is assumed that ϵ_1 , ϵ_2 , l_{arm} , l_{susp} are constant suspension parameters.

The nonlinear model (1) can be transformed into a linear control-oriented form. In order to perform the linearization the following assumptions are made. In the formulation small wheel tilting angles are considered, i.e., $\cos \gamma \approx 1$ and $\sin \gamma \approx \gamma$. Since γ values are small, the lateral tire force F_{tyre} is approximated in a linear form: $F_y = C\alpha$, where α is the side-slip angle of the tire and C is constant. During the wheel tilting motion $\alpha = \tan(r_w\dot{\gamma}/v) \approx r_w\dot{\gamma}/v$ results in the lateral side-slip angle, which is the angle between the longitudinal and the lateral component of the velocity vector.

$$F_y = C \frac{r_w \dot{\gamma}}{v} \quad (5)$$

The static compressions of the suspension and the tire are neglected, see (2) and (3). Finally, small wheel camber angles are considered, thus $\gamma \approx 0$ is also used.

Thus the linearized model of (1) is as follows:

$$m_{susp}\ddot{z}_w = \left(1 + \frac{l_{susp}}{l_{arm}} \sin \epsilon_2 \right) (-s_{susp}z_w - d_{susp}\dot{z}_w) + s_{tyre}(-l_{tyre}\gamma - z_w) + \frac{\sin \epsilon_2}{l_{arm}} M_{act} \quad (6a)$$

$$J_w\ddot{\gamma} = M_{act} - s_{tyre}l_{tyre}(-\gamma l_{tyre} - z_w) - C \frac{r_w^2 \dot{\gamma}}{v} \quad (6b)$$

The state-space representation of the variable-geometry suspension model is yielded

$$\dot{x}_{susp} = A_{susp}x_{susp} + B_{susp}u_{susp} \quad (7)$$

where the state vector is $x_{susp} = [\dot{z}_w \ z_w \ \dot{\gamma} \ \gamma]^T$ and the control input is the active torque $u_{susp} = M_{act}$. The measured signal is the camber angle of the wheel $y = \gamma$.

B. Lateral vehicle model

The lateral motion of the vehicle is influenced by the modification of the camber angle and generating longitudinal forces. Moreover, the camber angle influences the scrub radius of the wheel, which results in wheel steering. The lateral dynamics of the vehicle is influenced in two ways: the wheel camber angle and the steering angle have lateral force effect, while the longitudinal forces on either side create differential yaw moment. Therefore, in the lateral dynamics both effects are considered.

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

$$J\ddot{\psi} = C_1l_1\alpha_1 - C_2l_2\alpha_2 + M_d \quad (8a)$$

$$mv(\dot{\psi} + \dot{\beta}) = C_1\alpha_1 + C_2\alpha_2 \quad (8b)$$

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 (8) δ represents a common steering wheel, computed as the average of the left and right steering wheel angles: $\delta = (\delta_l + \delta_r)/2$. The steering rotation of a wheel is formulated using a dynamical equation $\ddot{\delta}_i = r_{\delta,i}F_{l,i}/J_{\delta,i}$, 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 and $i \in \{l, r\}$. The relationship between $r_{\delta,i}$ and γ_i is linear: $r_{\delta,i} = \varepsilon\gamma_i$, where ε is a construction parameter. The steering rotations of the wheels are extended to the lateral motion of the vehicle.

$$\ddot{\delta}_i = \varepsilon \frac{F_{l,i}\gamma_i}{J_{\delta,i}}, \quad i \in \{l, r\}. \quad (9)$$

These equations, which are also applied in the control design, show the relationship between the steering angles of the wheels and the longitudinal forces.

The lateral dynamics of the vehicle is formed using (8) and (11). The state-space representation of the lateral dynamics of the vehicle is formed using (8) and (11):

$$\dot{x}_{lat} = A_{lat}x_{lat} + B_{lat}u_{lat} \quad (10)$$

where the states of the model are defined as $x_{lat} = [\dot{\psi} \ \beta \ \dot{\delta}_l \ \delta_l \ \dot{\delta}_r \ \delta_r]^T$ and the control input is u_{lat} . The measured signal is the yaw rate of the vehicle.

The lateral dynamics of the vehicle is based on the control signal of the differential yaw moment. In this case the in-wheel electric motors are driven by different wheel moments

with which the differential yaw moment of the vehicle M_d is generated. The relationship between the longitudinal forces at the left and the right hand sides $F_{l,l}$ and $F_{l,r}$ and the differential yaw moment is the following:

$$M_d = (F_{l,l} - F_{l,r}) \frac{b_f}{2} \quad (11)$$

The purpose of the control design is to calculate the longitudinal forces from both the differential yaw moment M_d and the steering angles of the wheels $\ddot{\delta}_l$ and $\ddot{\delta}_r$, see equations (9) and (11). Three control signals are applied in the control-oriented model:

$$u_{lat} = [F_{l,l}\gamma_l \ F_{l,r}\gamma_r \ M_d]^T, \quad (12)$$

The vehicle performs a trajectory tracking task, in which both reference yaw rate and reference speed tracking must be guaranteed. In the paper only the yaw-tracking task is solved. The speed tracking is assumed to be handled by a cruise control, which generates longitudinal forces on either side of the front axle. These external forces are denoted by F_0 . In the control design proposed by the paper only the reference yaw-rate signal is followed, thus the longitudinal forces are modified in such a way that the required yaw moment M_d is created. The longitudinal forces on either side can be calculated from the following expressions:

$$F_{l,l} = F_0 + \frac{M_d}{b_f}, \quad F_{l,r} = F_0 - \frac{M_d}{b_r}. \quad (13)$$

where b_f is the track and the longitudinal forces. From the first two terms of the control input u_{lat} the wheel camber angle on either side can be calculated:

$$\gamma_l = \frac{F_{l,l}\gamma_l}{F_{l,l}}, \quad \gamma_r = \frac{F_{l,r}\gamma_r}{F_{l,r}}. \quad (14)$$

Concerning the longitudinal forces their values can be calculated directly from the lateral model if the longitudinal force equation of the vehicle is taken into consideration by forming a further equation. It is the basis of the future research.

III. ROBUST CONTROL DESIGN OF THE SYSTEM

The purpose of the independent control of the wheels is to solve two tasks. The control must generate an active torque for a wheel in order to change the camber angle and perform wheel steering. The control must also generate a differential yaw moment in order to perform the trajectory tracking of the vehicle.

The control design of the variable-geometry suspension system is based on a hierarchical structure. The goal of the high-level controller is to calculate the virtual control inputs u_{lat} based on the performance specifications for the road trajectory and the equation (12). The purpose of the low-level controller is to realize the physical active torques of the wheels in order to modify the camber angles of the wheels, see equation (7), and realize the longitudinal forces in order to track the required differential yaw moment.

A. Lateral control design

The goal of the lateral control design is to guarantee the trajectory tracking of the vehicle. It means that the vehicle must follow a reference yaw-rate signal $\dot{\psi}_{ref}$. The performance is defined as

$$z_{lat,1} = \dot{\psi}_{ref} - \dot{\psi}, \quad |z_{lat,1}| \rightarrow \min \quad (15)$$

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 $u_{lat} = [F_{l,l}\gamma_l \quad F_{l,r}\gamma_r \quad M_d]^T$:

$$z_{lat,2} = u_{lat}, \quad |z_{lat,2}| \rightarrow \min \quad (16)$$

The performances are compressed in a vector $z_{lat} = [z_{lat,1} \quad z_{lat,2}]^T$. The state-space representation of the system for the control design which incorporates the performances and the measurement is the following:

$$\dot{x}_{lat} = A_{lat}x_{lat} + B_{lat}u_{lat} \quad (17a)$$

$$z_{lat} = C_{lat,1}x_{lat} + D_{lat}u_{lat} \quad (17b)$$

and the measured signal is $y_{lat} = \dot{\psi}$. The augmented plant for the $\mathcal{H}_\infty$ design is illustrated in Figure 2. In the robust control

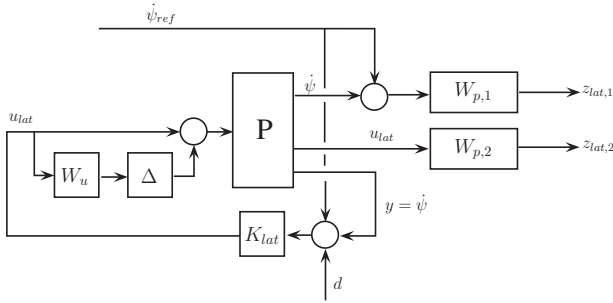


Fig. 2. Closed-loop interconnection structure

design weighting functions are considered, which are formed as transfer functions. The weighting functions of $z_{lat,1}$ and $z_{lat,2}$ are formulated as $W_{p,i} = (b_{1,i}s + b_{0,i})/(a_{1,i}s + a_{0,i})$ where $b_{1,i}, b_{0,i}, a_{1,i}, a_{0,i}$ are design parameters. The selection of these parameters has significant relevance. First, the parameters determine the accuracy of the yaw-rate tracking and the limitation of the signal of the actuator intervention. Second, the overshoot of the signal $\dot{\psi}$ is also determined by the parameters.

In the robust control design the uncertainties and the noises are also handled. It means that the performances of the lateral dynamics must be guaranteed under various circumstances. Since the steering of the wheels is realized through the camber angle actuation, it has an inaccuracy. Moreover, the wheel tilting intervention has its own dynamics, which is not considered in the lateral dynamics. These effects are considered in an input multiplicative uncertainty modeling approach. The uncertainty of the input u_{lat} is represented with Δ and its amplitude is scaled by the weighting function W_u , see Figure 2. Another robustness aspect in the control

design is the rejection of the external disturbances. In Figure 2 the sensor noise $w_l = d$ on the yaw-rate measurement is considered as a disturbance, which must be rejected by the robust controller.

The objective of $\mathcal{H}_\infty$ control is to minimize the $\mathcal{H}_\infty$ -norm of the cosensitivity function $T_{z_{lat}w}$. More precisely, the problem can be stated as follows [17], [18]. It means that it is necessary to find an admissible control K_{lat} , which satisfies the following design criteria: the closed-loop system must be asymptotically stable and the closed-loop transfer function from w_l to z_{lat} satisfies the following constraint

$$\|T_{z_{lat}w_l}\|_\infty < \Gamma, \quad (18)$$

for a given real positive value Γ . The robust control design guarantees the trajectory tracking of the vehicle through the computation of the appropriate $F_{l,l}\gamma_l, F_{l,r}\gamma_r, M_d$.

B. Variable-geometry suspension control

The objective of the variable-geometry suspension control is to realize the camber angle, which is yielded by the robust controller. The controlled system must guarantee the precise tracking of $\gamma_{ref,i}$ without high overshoots and time delays.

The performance of the suspension control is the minimization of the deviations of the current wheel camber angle from the reference angle both on either side of the first axle, such as

$$z_{susp} = [\gamma_{ref,l} - \gamma_l \quad \gamma_{ref,r} - \gamma_r]^T, \quad |z_{susp}| \rightarrow \min \quad (19)$$

where $\gamma_{ref,l}$ and $\gamma_{ref,r}$ are computed by the control inputs of the high-level controller, u_{lat} . The state-space representation of the suspension system is the following:

$$\dot{x}_{susp} = A_{susp}x_{susp} + B_{susp}u_{susp} \quad (20a)$$

$$z_{susp} = C_{susp}x_{susp} + D_{susp}u_{susp} \quad (20b)$$

and the measured signals are the camber angle of the wheels on either side of the front axle, i.e., $y_{susp} = [\gamma_l, \gamma_r]^T$. The sensor noises w_s are considered as a disturbance.

The control design is also based on the robust $\mathcal{H}_\infty$ method. The purpose is to design a suspension controller K_{susp} , which guarantees that the closed-loop system is asymptotically stable and the closed-loop transfer function from w to z_{susp} satisfies the following constraint

$$\|T_{z_{susp}w_s}\|_\infty < \Gamma, \quad (21)$$

for a given real positive value Γ . The realized control signals on either side of the first axle are the active torques around the center of the vertical axle of the vehicle, i.e., $M_{act,l}$ and $M_{act,r}$.

C. Architecture of the control systems

In Figure 3 the architecture of the lateral control and the variable-geometry suspension control are presented. It illustrates the cooperation of the different controllers.

The reference trajectory is designed through the velocity of the vehicle and the cornering intention of the driver,

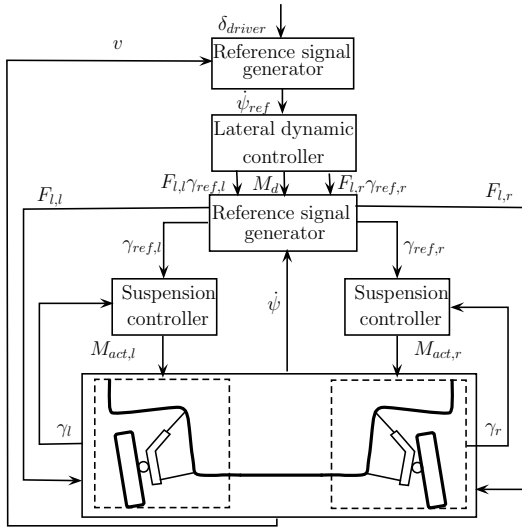


Fig. 3. Architecture of the control systems

estimated from the steering wheel angle δ_{driver} . The performance of the high-level control is the tracking of a reference yaw rate $\dot{\psi}_{ref}$. The reference signal is generated by the driver.

The lateral dynamic control, i.e., the high-level control in the hierarchical structure, computes the required control input $u_{lat} = [F_{l,l}\gamma_l \ F_{l,r}\gamma_r \ M_d]^T$, in which M_d is the differential yaw moment of the vehicle and $F_{l,i}\gamma_i$ is the steering by a wheel camber angle.

The suspension control, i.e., the low-level control in the hierarchical structure, generates the actuation torques $M_{act,l}$ and $M_{act,r}$, by which the steering of the wheels is realized. They require left and right wheel camber angles as reference signals $\gamma_{ref,l}$ and $\gamma_{ref,r}$. Since the high-level control generates u_{lat} , the longitudinal forces on the left and the right hand sides $F_{l,l}$ and $F_{l,r}$ and the wheel camber angles are calculated by applying a reference signal generator block. Since either side is controlled, in the low level two variable-geometry suspension controllers are applied, which operate independently of each other.

IV. SIMULATION EXAMPLE

The purpose of the simulation example is to show the efficiency of the variable-geometry suspension system in trajectory tracking. In the simulation a D-Class passenger car is used, whose mass is $m = 1690 \text{ kg}$. The camber angles and the active torques of the front wheels are generated by the hierarchical control structure. The simulations are performed through the high-fidelity CarSim software. Moreover, the complex multi-body SimMechanics model of the variable-geometry suspension is also built into the simulation system.

In the simulation example three control systems are applied and compared to each other. The first system applies only the steering with the control of the variable-geometry suspension system (K_{st} , denoted by Steering in the plots), the second system applies only the differential yaw moment (K_{ym} , denoted by Yawmoment in the plots) and the third

system integrates both the steering and the differential yaw moment (K_{int} , denoted by Steering + Yawmoment in the plots). It is assumed that a cruise control is applied in the vehicle, which generates the longitudinal forces in order to achieve the required speed.

The vehicle is driven along a section of the Michigan Waterford Hill course, see Figure 4(a). Since the track

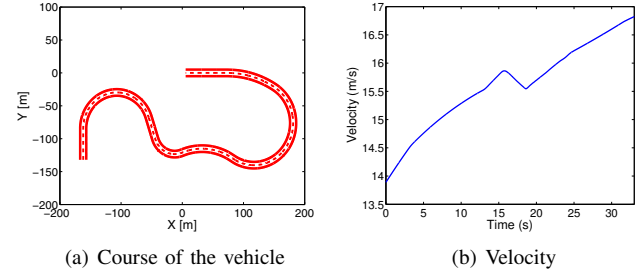


Fig. 4. Track of the vehicle

contains several bends, the velocity of the vehicle also varies, as shown in Figure 4(b).

The tracking of the reference yaw-rate signal is shown in all cases in Figures 5(a)-5(b), i.e., cases K_{st} and K_{ym} are in the figure on the left-hand side, while case K_{int} is on the right-hand side. Control systems which are based

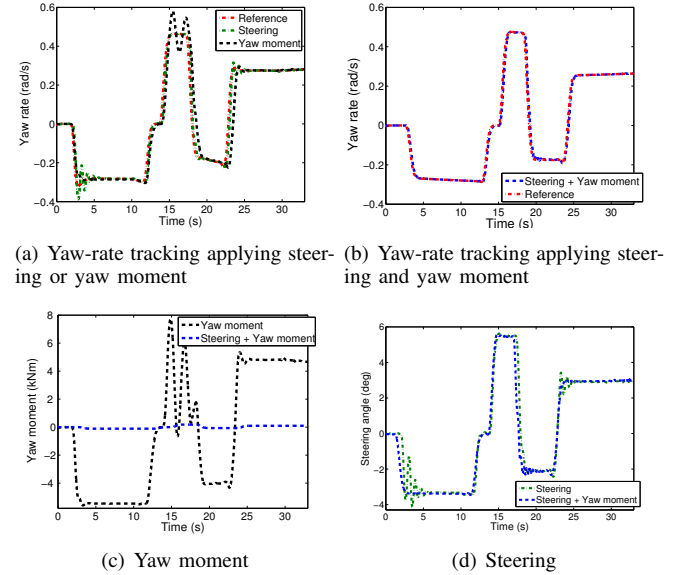


Fig. 5. Simulation results: yaw-rate tracking in different cases

on only one control signal show relatively long transients with overshoots and time delays in their time responses, while the control system which integrates both steering and differential yaw moment, K_{int} , shows smooth time responses. The yaw moment and the steering angle are shown in Figures 5(c)-5(d). In the integrated case the steering signal is smoother and faster compared to the steering in the K_{st} control. Moreover, in the integrated case the differential yaw moment significantly decreases compared to the differential yaw moment in the K_{ym} control.

Figure 6(a)...(d) show the active torques and the camber angles on either side and, moreover, the longitudinal forces. In the first maneuver the active torques in the control system which integrates both steering and differential yaw moment, K_{int} , are smaller and the transient is also shorter. However, in the second maneuver the active torques are larger and the transient is longer. Its reason is that the longitudinal forces are smaller in this interval, see Figure 6(e)(f). Note that the external force F_0 in equation (13) is constant in the simulation. According to (9) there is an inverse relation between the longitudinal force and the camber angle. In order to achieve the same steering and the longitudinal force decreases, the camber angle increases as it is shown in the example.

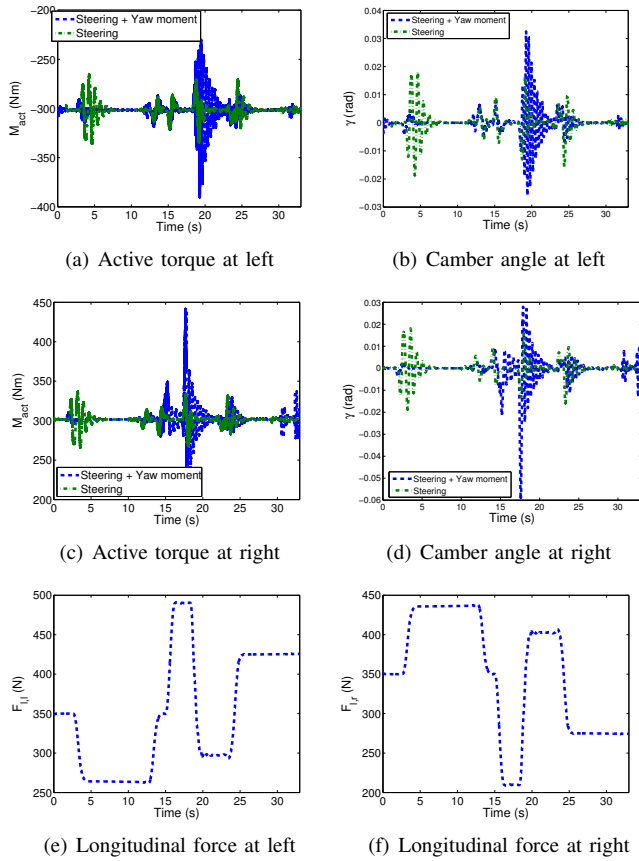


Fig. 6. Simulation results: torques, camber angles and longitudinal forces

V. CONCLUSION

A new variable-geometry suspension system which is integrated into in-wheel electric vehicles has been proposed in the paper. The variable-geometry suspension is an alternative way to the independent steering control. Since in-wheel electric vehicles are likely to have a significant impact in the future, further research on the variable-geometry suspension control is reasonable.

The variable-geometry suspension system generates active torque and modifies wheel camber angle in the wheels, moreover, by harmonizing the longitudinal forces both the

steering and the differential yaw moment can be created. The control design of the variable-geometry suspension system is based on a hierarchical structure. The goal of the high-level controller is to calculate the virtual control inputs based on the yaw-rate tracking for the road trajectory. In the reference signal generator block the longitudinal forces on either side and the wheel camber angles are calculated. The purpose of the low-level controller is to realize physical active torque of the wheels and track the required differential yaw moment. In the low-level control the actuation torques and longitudinal forces are realized.

The simulation examples show that although the reference yaw rate tracking task can be handled by all the three control solutions, the time responses of the performance signals show short and smooth transients in the control system which integrates both steering and differential yaw moment.

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