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Variable-Geometry Suspension Design in Driver Assistance Systems

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Abstract— **The paper proposes variable-geometry suspension design to enhance road stability during vehicle maneuvers. The orientation of wheels is modified by a suspension actuator, which results in an additional steering angle and a camber angle. A detailed analysis shows that the variable-geometry suspension system affects both the steering and the camber angle. Consequently, the integration of steering and wheel tilting can be handled by the variable-geometry suspension system. It is also shown that the suspension construction affects the control design. The control system must guarantee various vehicle performances such as trajectory tracking, roll stability and geometry limits. The operation of the designed control system is illustrated through simulation examples.**

I. Introduction and motivation

In the last decade several new research and development has been in the focus in the automotive industry [1]. The focus is on urban mobility and transport, alternative fuels, the electrification of the vehicle safety applications in co-operative systems, suitable materials, environment-friendly and efficient manufacturing. Within R&D activities the driver assistance systems play an important role, since the requirements of vehicle systems have become more stringent. Here are some examples: a need to enhance passenger comfort, improve road holding and the safety of travel, etc. Several important journal and conference papers have been presented in this topic, see e.g. [2], [3].

The variable-geometry suspension system provides a new possibility in the driver assistance systems to enhance road stability and safety. This system affects critical components such as the height of the roll center and the half track change. The advantages of the variable-geometry suspension are the simple structure, low energy consumption and low cost compared to other mechanical solutions such as an active front wheel steering, see [4], [5]. Since various safety and economy properties of the vehicle are determined by the suspension geometry it has an influence on the control design. The control input of variable-geometry systems is the camber angle of the front and rear wheels, with which the driver is supported to perform the various vehicle maneuvers, such as a sharp cornering, overtaking or double lane changing. During maneuvers the control system must guarantee various crucial vehicle performances such as trajectory tracking, roll stability and geometry limits.

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Several papers for various kinematic models of suspension systems have been published, see [6]. A nonlinear model of the McPherson strut suspension system was published by [7]. The kinematic design of a double-wishbone suspension system was examined by [8]. Seeking to meet the performance requirements often leads conflicts and requires a compromise considering the kinematic and dynamic properties, see [9]. The vehicle handling characteristics based on a variable roll center suspension were presented by [10]. A rear-suspension active toe control for the enhancement of driving stability was proposed by [11].

In our project, the number of possibilities of the variable-geometry suspension system are increased, see [12]. It has been shown that the control design is in interaction with the construction of the system [13]. A design criterion has been formed which results in optimal variable-geometry suspension systems [14]. This paper proposes the variable-geometry suspension system as part of the driver assistance system based on its combination with steering.

This paper is organized as follows. Section II proposes the dynamic interconnection between the steering angle and the camber angle. The construction of the suspension also has a significant effect on the actuation of the variable-geometry suspension, see Section III. There are several performances in a variable-geometry based driver assistance system, which are detailed in Section IV. In Section V the integration of the control design and the construction of variable-geometry suspension system is performed. Section VI illustrates the operation of the control system through different vehicle maneuvers using Simulink and CarSim softwares. Finally, the last section summarizes the contributions.

II. Dynamic effects of the variable-geometry suspension system

The actuation of the variable-geometry suspension system has effects on both the position and orientation of the front wheels. In the aspect of driver assistance system, steering angle δ_c and camber angle of the front wheels γ are relevant [13]. Through these signals the variable-geometry suspension has effects on the lateral tyre forces. In the following two sections the dynamic effects of the variable-geometry suspension system on steering and wheel tilting are presented.

A bicycle model of the vehicle is extended by the wheel camber effect, see Figure 1. The Magic form of the tire dynamics describes the effects on the steering angle, the camber angle and the lateral tyre forces (F_y), see [15]. Although it results in an accurate approximation of the lateral tire forces, in control design tasks a simplified form is used.

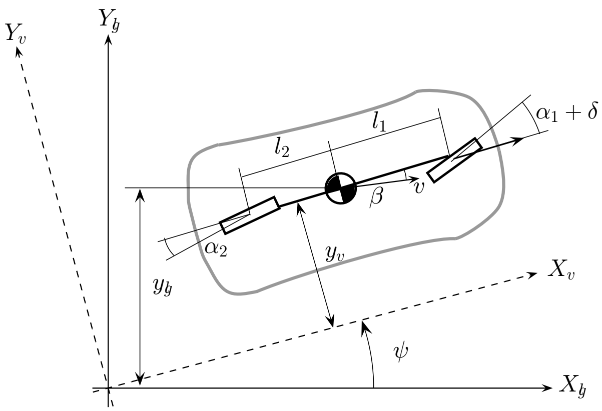


Fig. 1. Bicycle model of vehicle

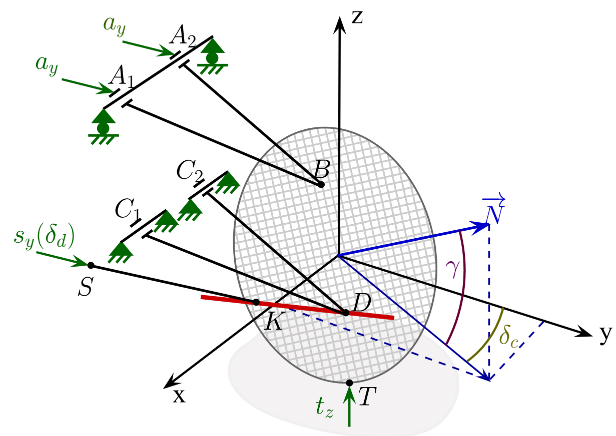


Fig. 2. Wheel position related to steering and camber angle

Based on the Magic form [14] proposes a linear relationship between δ_c, γ and the lateral tire forces at the front or rear $F_i = C_i\delta_c + C_{i,\gamma}\gamma$, where C_i is the cornering stiffness and $C_{i,\gamma}$ is the wheel camber stiffness. Then the following bicycle model is formed:

$$J\ddot{\psi} = C_1 l_1 \alpha_f - C_2 l_2 \alpha_r - C_{1,\gamma} l_1 \gamma(a_y) \tag{1a}$$

$$mv(\dot{\psi} + \dot{\beta}) = C_1 \alpha_f + C_2 \alpha_r + C_{1,\gamma}\gamma(a_y) \tag{1b}$$

with

$$\alpha_f = \delta_d + \delta_c(a_y) - \beta - l_1\dot{\psi}/v$$
$$\alpha_r = -\beta + l_2\dot{\psi}/v,$$

where m is the mass, J is the yaw inertia of the vehicle, l_1 and l_2 are geometric parameters, ψ is the yaw of the vehicle, β is the side-slip angle and v is velocity. In the equation the steering angle generated by the driver δ_d has an important role.

(1) shows that three signals have effects on lateral dynamics: δ_d, δ_c and γ. δ_d is performed by the driver, while the other two signals are control signals of the driver assistance system. However, δ_c and γ are not independent of each other, both of them depend on variable-geometry suspension actuation a_y: $\delta_c = \delta_c(a_y)$, $\gamma = \gamma(a_y)$.

The following facts can be stated:

- It is possible to realize the steering angle and the camber angle using one actuator. It leads to a simple and economic driver assistance system.
- The effects of the steering angle and the camber angle can be integrated to enhance road stability.
- The construction of the suspension system affects the balance between δ_c and γ. By using an appropriate construction the efficiency of driver assistance system is improved.

The interaction between these signals is determined by the construction, which will be presented in a detailed form in the following section.

III. Constructional interactions of the variable-geometry suspension

The scheme of the variable-geometry suspension system is illustrated in Figure 2. By modifying the camber angle the suspension geometry is modified and it affects the rotation of the front wheel. In the case of a double wishbone suspension camber angle γ modifies the wheel rotation around an axis, which is determined by the steering track-rod end K and the connection point of the lower arm D. Thus, the position of K has an important role in the rotation of the wheel. The angle between the axisBK and the road plane defines the relationship between camber angle γ and steering angle δ_c. When the variable-geometry suspension operates besides the changes in the camber angle an additional steering angle is generated. Consequently, a suitable suspension geometry is able to improve the lateral force on the tire not only by the camber angle, but also by the steering angle.

In the following the position of the wheel will be computed. A kinematic analysis of the variable-geometry suspension with double-wishbone construction has been proposed in [14]. In this paper the relationship between actuation a_y, disturbance t_z and suspension points B and D are formulated. Note that lateral and vertical movement of suspension points B and D are determined by control input a_y and disturbance t_z. They are denoted by b_y, b_z, d_y, d_z. Moreover, the rotation of the wheels is also determined by point K.

First, the orientation of the plane BDK is characterized by its normal vector $\overrightarrow{N}$. Steering angle δ_c and camber angle γ are computed by the movement of N in the following way:

$$\delta_c = \arctan\left(\frac{N_x}{N_y}\right) \tag{2a}$$

$$\gamma = \arctan\left(\frac{N_z}{\sqrt{N_x^2 + N_y^2}}\right) \tag{2b}$$

where N_x, N_y, N_z are the components of the normal vector

$\overrightarrow{N}$, which is computed as:

$$\overrightarrow{N} = \overrightarrow{DB} \times \overrightarrow{DK} \tag{3}$$

where $\overrightarrow{DB}$ and $\overrightarrow{DK}$ are vectors between the suspension points and steering track-rod end. Positions B and D points have been analyzed, see [14]. They require the measurement of suspension compression and actuation a_y, which provides definite information about positions B and D.

In the computation of point K in directions x, y, z the following statements must be considered:

- The axis of steering in a double-wishbone suspension is determined by axis BD.
- The relative positions of points B, D and K to each other are constant, because these points are the part of a solid wheel-hub. However, the position and orientation of the wheel-hub change.
- The length of the steering track-rod is also constant.

The statements guarantee that the position of point K can be determined accurately if the positions of B, D and steering rack movement (by the driver's steering wheel) are known. The steering track-rod interconnects the steering rack (S) and the wheel (K). Since the steering rack is able to move only in lateral directions S_x and S_z are constant, while S_y is determined by the driver's steering. Thus the steering rack movement is noted with S_y, which is directly measured or calculated by the measurement of the steering wheel angle.

There are several ways to calculate the position of K. An analytical way is to imagine K as a point in the intersection of 3 balls. The sections BK, DK, SK are constant, the positions of B, D and S are known, therefore the following coordinate geometry equations can be formulated:

$$BK^2 = (K_x - B_x)^2 + (K_y - B_y)^2 + (K_z - B_z)^2 \tag{4a}$$
$$DK^2 = (K_x - D_x)^2 + (K_y - D_y)^2 + (K_z - D_z)^2 \tag{4b}$$
$$SK^2 = (K_x - S_x)^2 + (K_y - S_y)^2 + (K_z - S_z)^2 \tag{4c}$$

The equations contain three unknown variables (K_x,K_y,K_z), which are the coordinates of K. They depend on the different points B, D, S Although (4) results in an analytic definite solution to the problem, there are some difficulties in its numerical solution.

The model of the variable-geometry suspension system is built in SimMechanics toolbox of Matlab, see Figure 3. The arms and bodies of the system are elements which are connected to vehicle chassis by joints. The joints A_1 and A_2 are actuated in lateral directions, which results in the change of wheel position and orientation. The coordinates of the points are measured, angles δ_c and γ are calculated by using (2).

Figures 4(a), 4(b) and 4(c) show angles δ_c, γ and ΔB at different K_z heights. The aim of the example is to present the relationship between signals. The variation of K_z has a great influence on angle δ_c and it modifies γ slightly. KB is the axis of wheel rotation during the actuation a_y, therefore its orientation influences the relationship between these angles. Since generally δ_c and γ are in conflict, it is necessary to find an appropriate solution to parameter K_z. In the analyzed

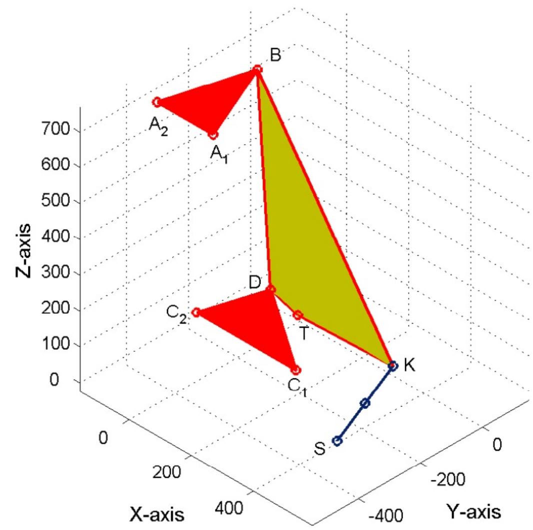
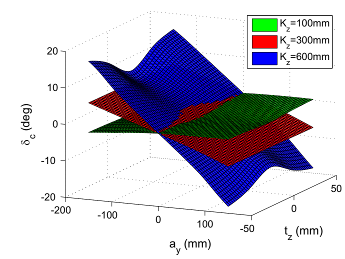
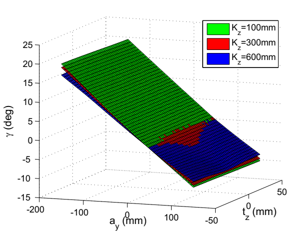


Fig. 3. Mechanism of the suspension system

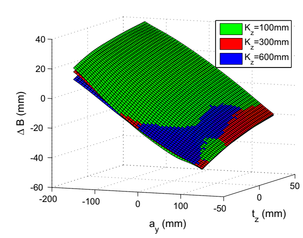
construction K_z has a significant influence on δ_c and with an increased K_z it is possible to achieve high lateral tire force, see Section II. Besides, K_z influences the lateral movement of T, i.e., the half-track change which is denoted by ΔB. It has an important role in tire wear, see [16]. Consequently, the steering angle, the camber angle and the half-track change are functions of the actuation, i.e., $\delta_c = \delta_c(a_y)$, $\gamma = \gamma(a_y)$, $\Delta B = \Delta B(a_y)$. Moreover, by applying a higher K_z it is possible to achieve an increased lateral tire force.



(a) δ_c change



(b) γ change



(c) ΔB change

Fig. 4. Influence of K_z on the relationship between δ_c, γ and ΔB

IV. Performances of the Variable-Geometry Suspension

The variable-geometry suspension system assists the driver during maneuvers, i.e., trajectory tracking can be performed by generating additional steering angle and modifying the camber angle. Besides, the variable-geometry suspension has an effect on other dynamic features. The selection of the roll center of the vehicle modifies the chassis roll angle. The control of the wheel position has an effect on the lateral movement of tire-road contact, which results in half-track change. Consequently, several performance requirements must be defined, such as yaw-rate tracking, the roll angle and the half track change. Note that the performance specifications are related to both the construction of the variable-geometry suspension system and the design of the control method.

A. Trajectory tracking

In the trajectory tracking control the vehicle must follow the reference yaw rate. The goal is to minimize the difference between the reference yaw rate and the measured yaw rate of the vehicle:

$$z_{e_{\dot{\psi}}} = |\dot{\psi}_{ref} - \dot{\psi}| \to min \quad (5)$$

The reference yaw rate represents the driver requirement, which depends on the steering input of the driver δ_d and geometry parameters. It is computed by using the following first order reference system, which is represented by a transfer function from steering angle δ_d to reference yaw-rate signal $\dot{\psi}_{ref}$, see [17], [18]:

$$G_{ref}(s) = \frac{v}{d} \cdot \frac{1}{\tau s + 1} \quad (6)$$

where d depends on velocity and geometry parameters $d = l_1 + l_2 + \frac{\eta}{g} v^2$, η is an understeer gradient, g is the gravitational constant and τ is the time constant, see [15].

B. Minimization of chassis roll angle

The height of the roll center has an important role in the vertical dynamics of the vehicle as it determines the roll motion. A possible way to minimize the chassis roll angle is the minimization of the height of the roll center h_M. In this case the difference between the roll center and the center of gravity must be minimized:

$$z_{\Delta h_M} = |h_{CG} - h_{M,st}| \to min \quad (7)$$

It can also be established that the height of roll center in steady state is determined by the suspension construction. Besides, the vertical movement of the roll center is determined by t_z and a_y, where a_y is the control signal. Thus, the minimization of the roll center is determined by both the construction and the control of the suspension.

C. Half-track change minimization

An additional important economy parameter is the half-track change $\Delta B = t_y = f(t_z, a_y)$. The lateral movement of the contact point is relevant from the aspect of tire wear [16], when the suspension moves up and down while the vehicle moves forward. By using an appropriate variable-geometry control the unnecessary movements can be eliminated:

$$z_{\Delta B} = |\Delta B| \to min \quad (8)$$

It has been shown that the lateral movement of the tire-road contact point ΔB depends on actuation a_y. Moreover, the relationship between a_y and ΔB is determined by the suspension construction, i.e., the positions of points K and D, see Figure 2. KD determines the axis of the wheel rotation. Thus, there is a direct relationship between the construction and ΔB.

D. Control input minimization

During the control tasks it is necessary to prevent a large control input, which is the lateral movement of the suspension arm a_y. It has construction limits, therefore the performance focuses on the minimization of the input displacement:

$$z_{act,susp} = |a_y| \to min \quad (9)$$

Note that the construction also influences performance $z_{act,susp}$ indirectly. Both steering δ_c and wheel tilting γ have lateral dynamic effects, see (1). However, the degree of cornering is different, i.e., $C_1 \neq C_{1,\gamma}$. Usually the cornering stiffness is greater than the wheel camber stiffness, therefore δ_c can be more efficient compared to γ in some cases. If a given control signal a_y induces greater δ_c and less γ, then the lateral force on the tire increases. In this case the actuation is more effective, which requires less actuation to generate lateral tire forces. The relationship between $\gamma(a_y)$ and $\delta_c(a_y)$ depends on the construction of the suspension system.

V. Control and construction design of the system

General form of the design method

Several performances which must be guaranteed by the driver assistance system have been formulated in the previous section:

$$Z = \begin{bmatrix} z_{e_{\dot{\psi}}} & z_{\Delta h_M} & z_{\Delta B} & z_{act,susp} \end{bmatrix}^T \quad (10)$$

The goal of the control design is to guarantee performances simultaneously. Since performances are in conflict, they require different control inputs. Thus, a balance between performances must be achieved. To emphasize the different importance of the performances weighting factors $W_i, i \in [1, 4]$ are introduced. The controller $\mathcal{K}$ significantly determines the properties of the controlled system. Since construction parameter K_z determines the balance between γ and δ_c and it also has an important role in tire-wear, its effect must be taken into consideration in the control design.

The aim of variable-geometry suspension design is to determine K_z and $\mathcal{K}$, which guarantee performances. The control design is based on the state-space representation of the system, which is formed by using equation (1):

$$\dot{x} = A(\rho)x + B_1(\rho)w + B_2(\rho)u \quad (11)$$

where the state vector contains the yaw-rate and the side-slip angle $x = \begin{bmatrix} \dot{\psi} & \beta \end{bmatrix}^T$, $w = t_z$ represents road disturbances and $u = a_y$ is the control signal of the variable-geometry suspension. The system matrices depend on the velocity of the vehicle nonlinearly, which is assumed to be a measured signal. Using a scheduling variable $\rho = v$ the nonlinear model is transformed into a Linear Parameter Varying (LPV) model. The performance signals are also formed in the state space representation form

$$z = C_1(\rho)x + D_{11}(\rho)w + D_{12}(\rho)u \qquad (12)$$

Generally, the following optimization task of the variable-geometry suspension system is formulated:

$$\min_{K_z,\mathcal{K}} \mathcal{J}(Z(K_z, \mathcal{K})) \qquad (13)$$

where $\mathcal{J}$ is a cost function of the performances. The optimization problem shows that the control design and the construction design are not independent. It can be solved in an iterative way.

Formulation of the suspension optimization problem

If the construction is fixed, the control design must be performed. The control design requires the formulation of the closed-loop interconnection structure of the system, see Figure 5.

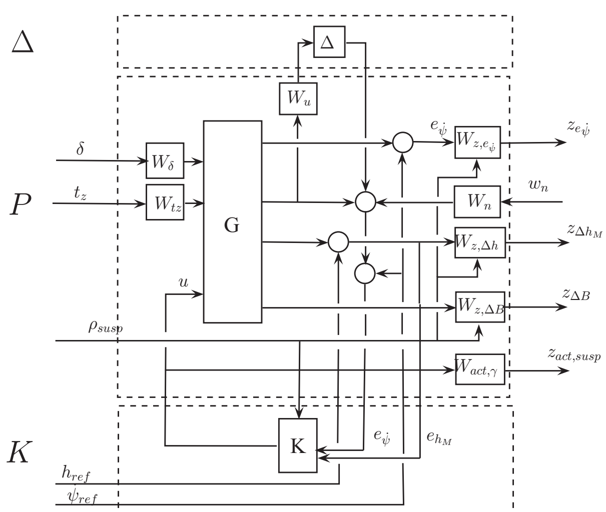


Fig. 5. Closed-loop interconnection structure

The control design of the variable-geometry suspension is based on the LPV method, which uses parameter-dependent Lyapunov functions, see [19], [20]. 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 $\mathcal{L}_2$ norm from the disturbance and the performances is less than a predefined value

$$\inf_K \sup_\Delta \sup_{\|w\|_2 \neq 0, w \in \mathcal{L}_2} \frac{\|\mathcal{Z}\|_2}{\|w\|_2}. \qquad (14)$$

where w is the disturbance and Δ represents the unmodelled dynamics. The $\mathcal{L}_2$ norm level for an LPV system represents the largest ratio of disturbance norm to performance norm over the set of the scheduling variables and the set of unmodelled dynamics.

Note that in an earlier paper of our project the simultaneous design of robust control and the construction of a relatively simpler structure of the variable-geometry suspension system has already been analyzed, see [14].

VI. SIMULATION EXAMPLE

In the simulation example the interaction between γ and δ_c is presented through the operation of a typical mid-size car. The control design of the suspension system is performed by the Matlab/Simulink software, while the verification of the controller is performed by the CarSim software and the SimMechanics toolbox of Matlab softwares. The vehicle dynamics in the CarSim is represented with high accuracy. The aim of the simulation example is to present the performances of the designed system.

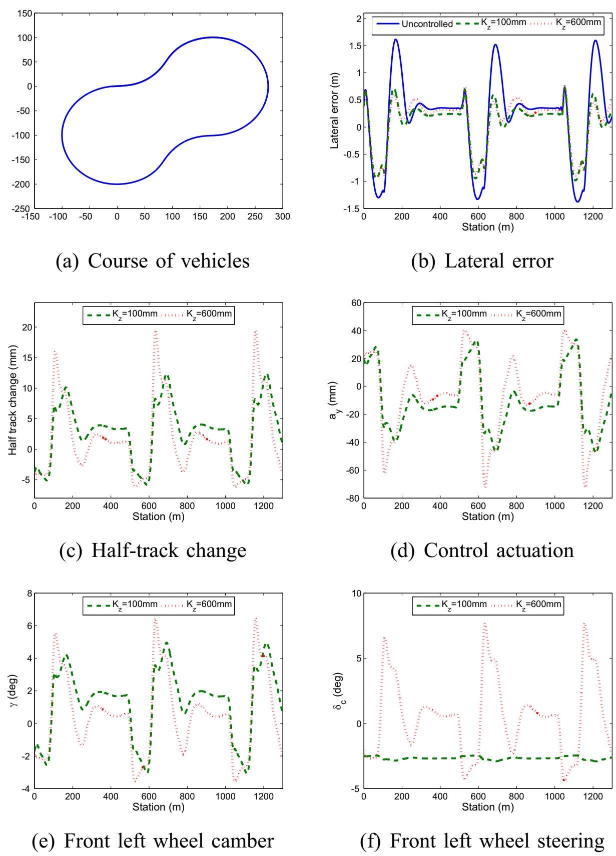
The analysis of the variable-geometry suspension system has shown that K_z affects wheel camber angle γ and steering angle δ_c significantly. Thus, two suspension constructions in which K_z is selected at different values are analyzed. They are $K_z = 100mm$ and $K_z = 600mm$. In the example the driver performs various maneuvers, in which the designed variable geometry suspension systems assists him. The results of the control systems are compared to the car without a driver assistance system.

Figure 6 shows the results of simulations. The operations of three systems are compared. The uncontrolled system is illustrated by solid blue line, the controlled system, in which $K_z = 100mm$ is illustrated by dashed green line, while the control system, in which $K_z = 600mm$ is illustrated by dash-dotted red line.

Figure 6(a) illustrates the course of vehicles. The vehicle is driven along the course at $95km/h$ velocity, which can be dangerous for the vehicle in the middle sections of the road because of sharp bends. Figure 6(b) shows that the lateral error of the uncontrolled vehicle is unacceptable. There are sections in which the deviation of the centerline exceeds $1.5m$, which may cause lane departures. Using the variable-geometry control system as a driver assistance system the error is reduced significantly, which is shown in Figure 6(b). Note that the reduction of the lateral error is independent of K_z, it is based on the designed controller.

The half-track change of the suspension system is shown in Figure 6(c). If $K_z = 100mm$ construction is set, in general, the half-track change is better than in the case of $K_z = 600mm$. However, the peak value of the half-track change is significantly worse in the $K_z = 100mm$ case. Besides, the actuation of control systems is greater in the $K_z = 100mm$ construction, see Figure 6(d). Generally, the tendency of control input signals are the same in both constructions. An interaction between ΔB and a_y is also found. When the $K_z = 600mm$ construction is set the peak values of the signal a_y increase compare related to the construction $K_z = 100mm$.

In terms of γ and δ_c the effects of the suspension constructions are different. In the case of $K_z = 100mm$ the control system is able to affect mainly the modification of wheel camber angle γ, see Figure 6(e). γ values are higher than in the other case because this system guarantees trajectory tracking by modifying γ. In case of $K_z = 600mm$ it is able to affect both wheel camber angle γ and steering angle δ_c, see also Figure 6(f). The steering angle actuation of the variable-geometry suspension system is shown in Figure 6(f). Since in this suspension system the steering wheel angle cooperates with wheel camber angle, a reduced a_y actuation is sufficient to perform trajectory tracking.



(a) Course of vehicles (b) Lateral error

(c) Half-track change (d) Control actuation

(e) Front left wheel camber (f) Front left wheel steering

Fig. 6. Simulation results in vehicle maneuvers

VII. Conclusion

The paper has proposed the design of the variable-geometry suspension system. The orientation of wheels is modified by a suspension actuator, which results in both an additional steering angle and a camber angle. The integration of steering and wheel tilting can be handled by the variable-geometry suspension system. The control system must guarantee various vehicle performances such as trajectory tracking, roll stability, half-track change and geometry limits. The system is able to create a cooperation between wheel camber angle and steering angle. The simulation example presents the efficiency of the variable-geometry suspension system and it shows that the system is suitable to be used as a driver assistance system.

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