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# Independent wheel steering control design based on variable-geometry suspension

**Balázs Németh** * **Dániel Fényes** * **Péter Gáspár** **

* *Systems and Control Laboratory, Institute for Computer Science and Control, Hungarian Academy of Sciences, Budapest, Hungary; E-mail: [balazs.nemeth,daniel.fenyes]@sztaki.mta.hu*

** *Institute for Computer Science and Control, Hungarian Academy of Sciences and MTA-BME Control Engineering Research Group, Budapest, Hungary; E-mail: peter.gaspar@sztaki.mta.hu*

**Abstract:** The paper presents the control design of an independent wheel steering system for in-wheel electric vehicles. The steering of the wheels is based on the variable-geometry suspension control, which influences the camber angle and the scrub radius of the wheel. A nonlinear formulation of the variable-geometry suspension is derived, which is validated through a high-fidelity suspension model. Furthermore, the impact of the independent steering in the lateral model of the vehicle is incorporated. In the paper the robust control design of the independent steering system in a hierarchical structure is presented. The trajectory tracking of the vehicle is guaranteed by the robust $\mathcal{H}_\infty$ control, while the intervention of the variable-geometry suspension is designed based on the Linear Quadratic theorem.



Keywords: Variable-geometry suspension, Independent steering, Robust control, In-wheel electric vehicle

## 1. INTRODUCTION AND MOTIVATION

The electrification of the vehicle drivelines provides a new possibility to enhance the stability and safety of road vehicles. A novel solution of the electric drive is the application of in-wheel electric motors. It makes possible the distribution of the traction forces between the wheels, by which additional functionalities can be achieved, e.g. the torque vectoring of the vehicle.

The in-wheel electric drive offers new challenges in the steering of the vehicle, such as independent steering. The goal of the independent steering concept is to improve the lateral dynamics of the vehicle using individually controlled wheels. The independent steering control for the rear wheels to modify the toe angle is presented by Lee et al. [1999]. The analysis of the independent wheel-steering system for heavy vehicles is found in Ronci et al. [2011]. Wang et al. [2011] proposes an indirect power steering measure called differential drive torque assisted steering, and validates its feasibility. Hu et al. [2015] presents a fault-tolerant control approach for a four-wheel independently actuated electric vehicle.

In the paper a new solution of independent steering is proposed, which is based on the variable-geometry suspension solution. The aim of the suspension control is the modification of the geometry, which results in a change in the camber or the toe angle. A rear-suspension active toe control for the enhancement of driving stability is proposed by Goodarzia et al. [2010]. 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 Piyabongkarn et al. [2004]. These vehicles require the design of innovative active wheel tilt and steer control strategies in order to perform steering similarly to a car on straight roads but in bends they tilt ilke motorcycles, see Suarez [2012]. 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 Evers et al. [2008], Lee et al. [2005].

In the paper the control design of an independent wheel steering system for the front wheels is proposed. The novelty of the paper is the application of the variable-geometry suspension in the steering solution. The contributions are the modeling and validation of the suspension system, the control design of the actuator and the robust trajectory tracking control of the vehicle. In the control solution the actuator and the tracking control are connected in a hierarchical structure.

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The paper is organized as follows. The formulation of the independent steering model incorporating the suspension and lateral vehicle models is presented in Section 2. The robust control design of the system in a hierarchical structure is found in Section 3. Section 4 demonstrates the efficiency of the control system. Finally, Section 5 summarizes the contribution of the paper.

## 2. MODELING OF INDEPENDENT STEERING

In this section the modeling of independent steering is proposed. The variable-geometry suspension performs the modification of the wheel position and orientation. Thus, the wheel camber angle and the scrub radius of the suspension vary. In the following, the formulation of the lateral vehicle model with the consideration of the camber angle and scrub radius is presented.

### *2.1 Modeling of the variable-geometry suspension*

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 tyre-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.

Since the variable-geometry suspension has one actuator in each wheel, it is necessary to find a suspension construction with which the lateral force generation of the camber and the scrub radius is in coordination. Thus, the forces from the wheel tilting and the steering from the scrub radius have the same effect on the vehicle dynamics. In the paper 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. Several forces influence the motion of the suspension and the wheel. The force of the suspension compression and damping $F_{susp}$ is formulated as

$$F_{susp} = s_{susp}\left(\frac{z_w + z_{w,0}}{\sin\epsilon_1}\right) + d_{susp}\frac{\dot{z}_w}{\sin\epsilon_1} \qquad (1)$$

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 lateral force on the tyre is $F_y$. It is derived from the Magic Formula, see Pacejka [2004]. $F_{tyre}$ is the force from the tyre compression, which has a direction to the wheel:

$$F_{tyre} = s_{tyre}\frac{(r_w\cos\gamma - l_{tyre}\sin\gamma - r_w - z_w) + z_{tyre,0}}{\cos\gamma} \qquad (2)$$

where $s_{tyre}$ is the tyre stiffness, $r_w$ is the wheel radius and $z_{tyre,0}$ is the static compression of the tyre.

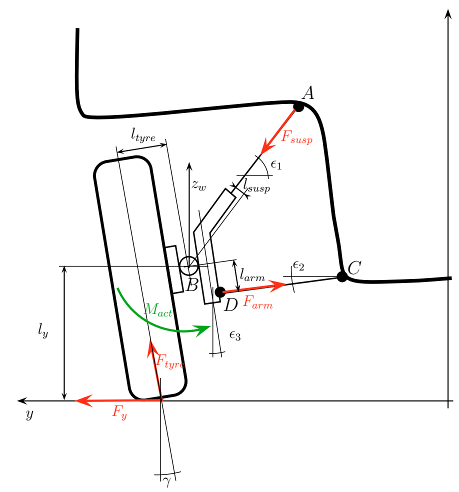


Fig. 1. Scheme of the suspension construction

The motion of the suspension is described by three dynamic equations. First, the vertical motion of the wheel hub $z_w$ is formulated (3). Second, the torque on the suspension is described to derive the rotation of the wheel hub $\epsilon_3$, see (4). Third, the tilting of the wheel $\gamma$ is formulated (5).

$$m_{susp}\ddot{z}_w = -F_{susp}\sin\epsilon_1 + F_{tyre}\cos\gamma + F_{arm}\sin\epsilon_2 \qquad (3)$$

$$J_{susp}\ddot{\epsilon}_3 = F_{susp}l_{susp} + F_{arm}l_{arm} - M_{act} \qquad (4)$$

$$J_w\ddot{\gamma} = M_{act} - F_{tyre}l_{tyre} - F_y l_y \qquad (5)$$

The arm of the lateral tyre force is

$$l_y = r_w\cos\gamma - l_{tyre}\sin\gamma. \qquad (6)$$

where $r_w$ and $l_{tyre}$ are construction parameters.

Since the inertia and the stiffness of the suspension through $F_{susp}$ and $F_{arm}$ are high, the effect of $M_{act}$ on the suspension motion is relatively small. Thus, in practice the angle $\epsilon_3$ is constant, thus $\ddot{\epsilon}_3 = 0$. Therefore, $F_{arm}$ is computed as:

$$F_{arm} = \frac{M_{act} - F_{susp}l_{susp}}{l_{arm}} \qquad (7)$$

Thus, it is assumed that $\epsilon_1, \epsilon_2$ and $l_{arm}, l_{susp}$ can be handled as constant suspension parameters.

The nonlinear variable-geometry suspension model is validated through the complex mechanical simulation system Matlab/SimMechanics. In SimMechanics the construction of the suspension has been built, and the modification of the wheel camber angle has been analyzed. Figure 2 shows a simulation example, in which the suspension model (3)...(5) and the SimMechanics model are compared. In the simulations the same chirp input signals $M_{act}$ are realized, see Figure 2(a). The camber angles of the model are presented in Figure 2(b). It shows that the resulting camber angles are very close to each other in a high operation range.

Figure 3 shows another example for the validation of the nonlinear suspension model. The step signal of the control input $M_{act}$ is shown in Figure 3(a). As an effect of the torque modification, the camber angle also varies, see Figure 3(b). The results show that the steady-state error of the camber angle is below 2%. It means that the proposed nonlinear suspension model fits the SimMechanics model well.