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Vehicle velocity estimation using nonlinear observers☆

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Abstract

Nonlinear observers for estimation of lateral and longitudinal velocity of automotive vehicles are proposed. The observers are based on a sensor suite that is standard in many new cars, consisting of acceleration and yaw rate measurements in addition to wheel speed and steering angle measurements. Two approaches are considered: first, a modular approach where the estimated longitudinal velocity is used as input to the observer for lateral velocity, and second, a combined approach where all states are estimated in the same observer. Both approaches use a tire-road friction model, which is assumed to be known. It is also assumed that the road is flat. Stability of the observers is proven in the form of input-to-state stability of the observer error dynamics, under a structural assumption on the friction model. The assumption on the friction model is discussed in detail, and the observers are validated on experimental data from cars.

Keywords: Automotive vehicles; Nonlinear observers; Side-slip estimation; Velocity estimation; Friction

1. Introduction

Feedback control systems for active safety in automotive applications have entered production cars. Many of these systems (for instance yaw stabilization systems such as ESP (van Zanten, 2000)) have in common that the control action depends on information about vehicle velocity. However, the velocity is rarely measured directly and must therefore be inferred from other measurements, such as wheel speed, yaw rate, and acceleration measurements.

The main goal of this work is to develop computationally efficient nonlinear observers for vehicle velocity with theoretical stability guarantees. We propose two nonlinear observer structures: first, a modular cascaded observer structure where the estimation of lateral and longitudinal velocity is separated, and second, a combined observer for both velocities. We establish input-to-state stability (ISS) of both observer structures under a realistic condition on the friction model.

Nonlinear observers are used in order to take nonlinear dynamics (due mainly to highly nonlinear friction and Coriolis forces) into account and to obtain simple designs with few tuning knobs (as opposed to extended kalman filter (EKF) designs). Another significant advantage of the proposed approach over the EKF is that real-time solution of the Riccati differential equations is avoided, such that the observer can be implemented more efficiently in a low-cost electronic control unit.

A nonlinear tire-road friction model is used in order to fully exploit the lateral acceleration measurement. An important parameter in many friction models is the maximum tire-road friction coefficient μ_H, which is known to vary significantly with road conditions. We will assume that this parameter is known, that is, measured or estimated. While simultaneous estimation of velocity and μ_H might be a feasible path, we claim that estimation of μ_H requires special attention depending on the application of the observer, since it will be only weakly observable for some driving maneuvers, requiring monitoring,

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resetting and other logic functions to be implemented (see e.g. Kalkkuhl, Johansen, & Ludemann, 2003). Estimation of friction parameters is therefore not considered in this paper. We also assume that the road bank and inclination angles are zero (or known), since these will introduce a gravity component in the horizontal acceleration measurements.

Earlier work on observers for estimation of lateral velocity is mainly based on linear or quasi-linear techniques, for example Fukada (1999), Venhovens and Naab (1999), Ungoren, Peng, and Tseng (2004), Farrelly and Wellstead (1996). A nonlinear observer linearizing the observer error dynamics is proposed in Kiencke and Daiss (1997), Kiencke and Nielsen (2000). The same type of observer, in addition to an observer based on forcing the dynamics of the nonlinear estimation error to the dynamics of a linear reference system, are investigated in Hiemer, vonVietinghoff, Kiencke, and Matsunaga (2005). The problem formulation there assumes that the longitudinal wheel forces are known, similarly to the observer implemented in ESP (van Zanten, 2000). In our work, we do not make this assumption, as such information is not always available.

An EKF is used for estimating vehicle velocity and tire forces in Ray (1995, 1997), thus without the explicit use of friction models. A similar, but simpler, approach is suggested in Farrelly and Wellstead (1996). An EKF based on a tire-road friction model which also includes estimation of the adhesion coefficient and road inclination angle is suggested in Suissa, Zomotor, and Böttiger (1996). In Best, Gordon, and Dixon (2000), the use of an EKF based on a nonlinear tire-road friction model is considered, which also includes estimation of cornering stiffness. The strategy proposed in Lu and Brown (2003) combines dynamic and kinematic models of the vehicle with numerical band-limited integration of the equations to provide a side-slip estimate. In Hac and Simpson (2000) the side-slip angle is estimated along with yaw rate in an approach which is similar to the one considered herein, but without yaw rate measurements. The approach is validated using experimental data, but no stability proofs are presented.

A unique feature of the work presented here is that explicit stability conditions are analyzed in a nonlinear setting, without resorting to linearizations.

2. Vehicle modeling

2.1. Rigid body dynamics

The geometry of the vehicle is illustrated in Fig. 1. The vehicle velocity is defined in a body-fixed coordinate system with the origin at the center of gravity (CG), the location of which is assumed constant. The x-axis points forward and the y-axis points to the left. There are also wheel-fixed coordinate systems aligned with each wheel and with origins at the wheel centers. The distance from the CG to each wheel center is denoted h_i, with i representing the wheel index. Together with the angles γ_i, this defines the vehicle geometry. Ignoring suspension dynamics, we assume that we can consider the vehicle a rigid body, for which the rigid body dynamics with respect to the CG

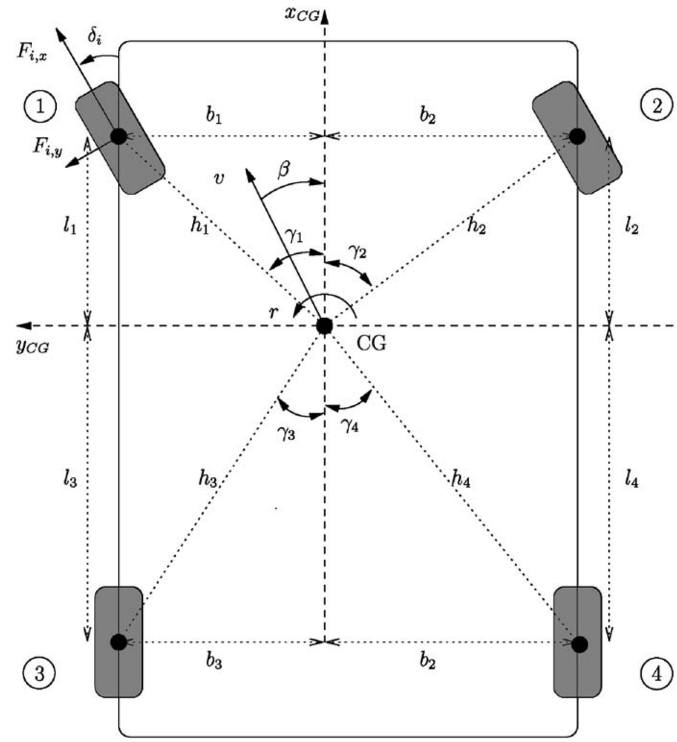

Fig. 1. Top view of vehicle: horizontal axis systems, geometric definitions, wheel forces, speed, slip angle and yaw rate.

coordinate system can be written (Kiencke & Nielsen, 2000)

$$\mathbf{M}\dot{\boldsymbol{v}} + \mathbf{C}(\boldsymbol{v})\boldsymbol{v} = \boldsymbol{\tau}, \tag{1}$$

where $\boldsymbol{v}$ is a vector containing the body generalized velocities. The matrices $\mathbf{M}$ and $\mathbf{C}$ are the inertia, and the Coriolis and centripetal matrices, respectively. The vector $\boldsymbol{\tau}$ consists of forces and torques acting on the vehicle; mainly friction forces acting via the wheels, but also gravitational and aerodynamic (wind and air resistance) forces.

By making the following assumptions:

- including only motion in the plane (ignore dynamics related to vertical motion, including roll and pitch),
- ignoring effect of caster and camber,
- including only forces caused by tire-road friction,

the vehicle dynamics are described by the longitudinal velocity v_x, lateral velocity v_y, and yaw rate r, for which

$$\mathbf{M} = \begin{pmatrix} m & 0 & 0 \\ 0 & m & 0 \\ 0 & 0 & J_z \end{pmatrix}, \quad \mathbf{C}(\boldsymbol{v}) = \begin{pmatrix} 0 & -mr & 0 \\ mr & 0 & 0 \\ 0 & 0 & 0 \end{pmatrix}.$$

The generalized forces $\boldsymbol{\tau} = (f_x, f_y, \tau_z)^\top$ are forces and torque acting on the vehicle generated by friction between the tires and the road. The friction forces $\mathbf{F}_i$ acting at each wheel (see Fig. 1) are functions of the velocity difference between the vehicle and the tires (cf. next section). These are transformed from the wheel coordinate systems to the body fixed coordinate system: in the body-fixed coordinate system, the forces generated by