Mechatronics 30 (2015) 338–347

Contents lists available at ScienceDirect

Mechatronics

journal homepage: www.elsevier.com/locate/mechatronics

Robust energy-to-peak sideslip angle estimation with applications to ground vehicles

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ARTICLE INFO

Article history:
Received 15 May 2014
Revised 15 July 2014
Accepted 22 August 2014
Available online 16 September 2014

Keywords:
Robust estimation
Four-wheel-independent-drive electric vehicle
Sideslip angle
Energy-to-peak

ABSTRACT

In this paper, the observer design problem for the sideslip angle of ground vehicles is investigated. The sideslip angle is an important signal for the vehicle lateral stability, which is not measurable by using an affordable physical sensor. Therefore, we aim to estimate the sideslip angle with the yaw rate measurements by employing the vehicle dynamics. The nonlinear lateral dynamics is modeled firstly. As the tyre model is nonlinear and the road adhesive coefficient is subject to a large variation, the nonlinear lateral dynamics is transformed into an uncertain model. Considering the variation of longitudinal velocity, an uncertain linear-parameter-varying (LPV) system is obtained. Based on the LPV model, a gain-scheduling observer is proposed and the observer gain can be determined with off-line computation and on-line computation. The off-line computation includes the calculation of a set of linear matrix inequalities and the on-line computation contains several algebraic operations. The proposed design methodology is applied to a four-wheel-independent-drive electric vehicle in simulation. It infers from different maneuvers that the designed observer has a good performance on estimating the sideslip angle.

1. Introduction

The sideslip angle of vehicle is of importance for vehicle lateral stability; see [1–4] and the references therein. It is a critical index for the vehicle safety. An ideal case is that the sideslip angle is kept zero during driving. However the ideal case is impossible to be achieved during the steering maneuvers with a high longitudinal velocity. In order to enhance the lateral stability, there are numerous techniques to stabilize the lateral dynamics and reduce the angle. Among the existing techniques, the sideslip angle is generally employed to construct the feedback loop [5]. However, the commercial physical sensor for the sideslip angle is quite expensive and it is impossible to apply the physical sensor in the practice. An alternative method is to estimate the sideslip angle with measurements from low-cost sensors.

During the past decade, the sideslip angle estimation has attracted considerate efforts and there exist a lot of methods in the literature [6]. The authors in [7] employed the GPS measurements to calculate the sideslip angle and the tyre slip. The proposed method is based on the kinematics model, which has the drift phenomena induced by the bias errors. The steering torque was applied in [8] to estimate the tyre slip angle which was further utilized to compute the sideslip angle. The Kalman filter can be seen in the sideslip angle estimation work of [9]. Since the measurement facility or the physical sensor is costly in many applications, the authors in [10] proposed the sideslip angle fusion estimation using low-cost sensors. Though there are a lot of existing approaches on the sideslip angle estimation, the approaches have some drawbacks such as the cost is high or the method depends on an accurate dynamics model or accurate priori information. Motivated by these points, we aim to develop robust estimator which has the capacity against the inaccuracy in the system model.

The robust control and estimation have been two popular research areas for a long time due to the fact that the uncertainty is unavoidable in the system modeling [11–16]. The robust scenario is not only against the system uncertainty, but also attenuate the effect of the external disturbance [17,18]. Different from the Kalman filtering, the priori information on the noise is not required. Typically, there are energy-to-energy and energy-to-peak strategies in the robust filtering. The first one is to minimize or constrain the energy-to-energy ratio and the second one is used for the energy-to-peak ratio. Both strategies have been successfully

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http://dx.doi.org/10.1016/j.mechatronics.2014.08.003
0957-4158/

applied to different kinds of systems; see the work in [19–23]. However, there is few result on the robust estimation of vehicle systems.

In this work, motivated by the gap between the robust estimation and the vehicle dynamics, we aim to develop the robust estimator design method for the vehicle sideslip angle estimation. Both the variations of the road adhesion coefficient and the tyre cornering stiffness are considered, an uncertain vehicle lateral dynamics model is obtained. Due to the fact that the longitudinal velocity is not a constant, we consider a varying longitudinal velocity which is within a range and the observer gain is scheduling with respect to the velocity. The robust energy-to-peak estimator gain can be obtained by solving a set of linear matrix inequalities.

2. Uncertain vehicle dynamics model

We consider a simplified bicycle model of vehicle lateral dynamics, which is a two-degree-of-freedom (2-DOF) model and has been widely adopted to describe the lateral dynamics, as shown in Fig. 1. In the model, the total mass of the vehicle is represented by m and the moment of inertia about the yaw axis through its center of gravity (CG) is denoted as I_z. The front axis is located at the distance l_f from the CG and the rear axis is located at the distance l_r from the CG. The steering angle δ is controlled by the driver and has the capacity to change the heading of the front tyres. The slip angles α_f and α_r of the front and rear tyres depend on the front and rear lateral tyre forces F_{yf} and F_{yr}, respectively.

The relationship between the lateral force and the side angle is complex and decided by the tyre characteristic, the road adhesion coefficient and the lateral acceleration. Since the load adhesion coefficient is not always a constant and difficult to be estimated, a practical tyre lateral model is approximated as [5]

$$\begin{aligned} F_{yf} &\approx (c_f + \Delta c_f N(t))\alpha_f = \bar{c}_f \alpha_f, \\ F_{yr} &\approx (c_r + \Delta c_r N(t))\alpha_r = \bar{c}_r \alpha_r, \end{aligned} \tag{1}$$

where the approximated coefficients $\bar{c}_f$ and $\bar{c}_r$ consist of nominal terms and uncertain terms, and $N(t) \leq 1$. The equivalent nominal coefficients c_f and c_r are calculated by

$$\begin{aligned} c_f &= \mu_n C_{fn}, \\ c_r &= \mu_n C_{rn}, \end{aligned}$$

and the bounds of the uncertainties are estimated by

$$\begin{aligned} \Delta c_f &= \mu_n \Delta C_f + \Delta\mu C_{fn} + \Delta\mu \Delta C_f, \\ \Delta c_r &= \mu_n \Delta C_r + \Delta\mu C_{rn} + \Delta\mu \Delta C_r, \end{aligned}$$

with the nominal value of road adhesion coefficients μ_n, the uncertainty bound $\Delta\mu$, the nominal values of cornering stiffness C_{fn} and C_{rn}, and the uncertainty bounds ΔC_f and ΔC_r. Note that the tyre lateral model in (1) can cover different road conditions and tyre characteristic. The tyre model is quite useful in the controller and estimator design. More details on the derivative of the tyre model can be seen in [5].

To study the vehicle lateral dynamics, a variable named sideslip angle at CG is defined as the tangential angle between the vehicle lateral velocity and the longitudinal velocity:

$$\tan\beta = \frac{v_y}{v}. \tag{2}$$

Here, v denotes the longitudinal velocity and v_y is the lateral speed. When the vehicle sideslip angle β is small, with the geometric relationship, the tyre slip angles are

$$\begin{aligned} \alpha_f &= \delta - \frac{l_f\omega}{v} - \beta, \\ \alpha_r &= \frac{l_r\omega}{v} - \beta. \end{aligned} \tag{3}$$

Here, ω denotes the yaw rate at CG. According to the Newton law, a state-space model of the lateral dynamics can be obtained as:

$$\dot{x}(t) = Ax(t) + B\delta + CM_z, \tag{4}$$

where

$$x(t) = \begin{bmatrix} \beta \\ \omega \end{bmatrix},$$

$$A = \begin{bmatrix} \frac{-\bar{c}_f - \bar{c}_r}{mv} & \frac{l_r\bar{c}_r - l_f\bar{c}_f}{mv^2} - 1 \\ \frac{l_r\bar{c}_r - l_f\bar{c}_f}{I_z} & \frac{-l_f^2\bar{c}_f - l_r^2\bar{c}_r}{I_z v} \end{bmatrix},$$

$$B = \begin{bmatrix} \frac{\bar{c}_f}{mv} \\ \frac{l_f\bar{c}_f}{I_z} \end{bmatrix}, \quad C = \begin{bmatrix} 0 \\ \frac{1}{I_z} \end{bmatrix},$$

M_z is the yaw moment exerted on the vehicle. In order to enhance the lateral stability, the direct yaw moment control is applied in many vehicles such as the four-wheel-independent-drive electric vehicle in [24]. When the direct yaw moment control is available for a vehicle, the value of yaw moment is known for the observer design. Moreover, the front wheel steering angle δ can be also measurable [25]. Therefore, both inputs of the lateral dynamics model can be assumed to be available for the estimator design.

It is necessary to mention that the lateral dynamics in (4) is an approximated model with several assumptions. To better describe the behaviors of the vehicle, we use the following uncertain model subject to disturbances:

$$\dot{x}(t) = Ax(t) + B\delta + CM_z + d, \tag{5}$$

where $d = \begin{bmatrix} d_1 \\ d_2 \end{bmatrix}$ is unknown but bounded. The disturbance d may arise from (1) the modeling error; (2) the mapping error by the control allocation; or (3) the hysteresis of the steering system. Note that the forward velocity v is not a constant such that the system matrix in the model (5) is time-varying. As the analysis of the linear systems has been well developed, we are going to convert the system in (5) into a linear parameter varying (LPV) form. The challenge is how to balance the conservativeness and the computational load. Suppose that the forward velocity v is bounded as:

$$v \in [\underline{v}, \ \bar{v}] \tag{6}$$

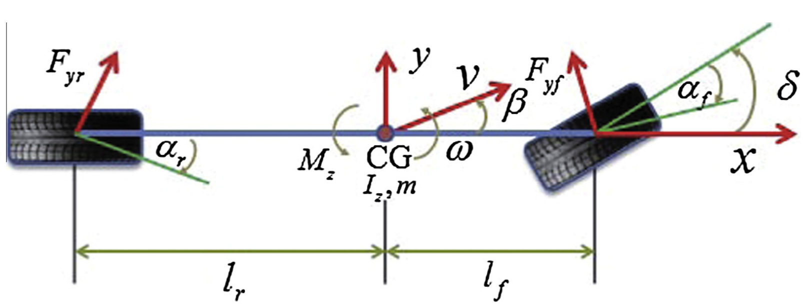

Fig. 1. Simplified two-degree-of-freedom (2-DOF) vehicle lateral dynamics model.