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Brief paper

Robust nonlinear output feedback control for brake by wire control systems☆

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Abstract

This work proposes a nonlinear output feedback control law for active braking control systems. The control law guarantees bounded control action and can cope also with input constraints. Moreover, the closed-loop system properties are such that the control algorithm allows to detect—without the need of a friction estimator—if the closed-loop system is operating in the unstable region of the friction curve, thereby allowing to enhance both braking performance and safety. The design is performed via Lyapunov-based methods and its effectiveness is assessed via simulations on a multibody vehicle simulator.

Keywords: Vehicle dynamics; Automotive control; Nonlinear control design; Output feedback control; Lyapunov methods

1. Introduction

Electronic anti-lock braking systems (ABS) have recently become a standard for all modern cars. In fact, ABS can greatly improve the safety of a vehicle in extreme circumstances, as it maximizes the longitudinal tire–road friction while keeping large lateral forces which guarantee vehicle steerability. The current trend in braking control systems design is to move from threshold-based control logics, mainly based on the wheel deceleration measurement, to genuine slip control (see e.g., Drakunov, Ozguner, Dix, & Ashrafi, 1995; Johansen, Petersen, Kalkkuhl, & Lüdemann, 2003; Ünsal & Kachroo, 1999). The main motivation behind this major change in ABS design is due to the recent technological advances in actuators, both electro-hydraulic and electro-mechanical, which are replacing hydraulic brakes with discrete dynamics. These new actuators enable a continuous modulation of the braking torque, allowing to formulate slip control as a classical regulation problem. In the field of automatic braking control many approaches have been proposed, ranging from classical regulation loops based on linearized models to sliding mode, fuzzy-neural or hybrid control strategies, see e.g., Drakunov et al. (1995), Lin and Hsu (2002) and Somakumar and Chandrasekhar (1999). One of the main challenges in designing ABS systems is to devise control logics which are robust with respect to two significant sources of uncertainty affecting the braking dynamics, i.e., the highly nonlinear tire–road friction forces and the dynamic load transfer between front and rear axle. Many research efforts have been devoted to estimate the road characteristics on-line (see e.g., Canudas de Wit, Petersen, & Shiriaev, 2003; Gustafsson, 1997; Ono et al., 2003; Tanelli & Savaresi, 2006; Yi, Alvarez, & Horowitz, 2002). However, due to the high complexity of these techniques combined with the limited computing resources commonly available on commercial vehicles electronics control units (ECUs), a robust control logic which does not require on-line friction estimation is usually preferred. We propose a nonlinear output feedback control law, based on slip and wheel speed measurements, which does not require any knowledge either of the current road condition or of the instantaneous value of the normal force exerted on the tire. This control law guarantees bounded control action and can cope also with input constraints. The design is performed via Lyapunov-based methods and its validity is assessed via simulations carried out on a full multibody vehicle simulator. The overall controller

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performances are compared to those of a fixed structure wheel slip controller based on a linearized model so as to highlight the features of the proposed nonlinear controller. The distinctive feature of the proposed controller is that the closed-loop system properties are such that the employed control algorithm allows to detect if the closed-loop system is operating in the unstable region of the friction curve, thereby allowing to adapt the set point and enhance braking performance and safety. The paper is organized as follows. In Section 2 the quarter-car model is described and discussed. Section 3 states the control problem, while the nonlinear output feedback controller design is presented in Section 4. Section 5 assesses the effectiveness of the proposed nonlinear controller on a full-car simulator (MSC CarSim®) and compares its performance with that of a fixed structure wheel slip controller based on a linearized model of the braking dynamics.

2. System description

For the preliminary design and testing of braking control algorithms a simple but effective quarter-car model (Johansen et al., 2003) is typically used. The model is given by the following equations:

$$J\dot{\omega} = rF_x - T_b, \quad m\dot{v} = -F_x, \tag{1}$$

where ω (rad/s) is the angular speed of the wheel, v (m/s) is the longitudinal speed of the vehicle body, T_b (N m) is the braking torque, F_x (N) is the longitudinal tire–road contact force, J (kg m^2), m (kg) and r (m) are rotational inertia of the wheel, the quarter-car mass and the wheel radius, respectively. In the rest of the paper the following values will be employed: $J = 1\,\text{kg m}^2$, $r = 0.3\,\text{m}$, $m = 225\,\text{kg}$. The dynamic behavior of the system is hidden in the expression of F_x, which depends on the state variables v and ω. The most general expression of F_x is quite involved, since it depends on a large number of features of the road, tire and suspension. However, it can be well-approximated as follows: $F_x = F_z\mu(\lambda, \beta_t; \vartheta_r)$, where F_z is the vertical force at the tire–road contact point; λ is the longitudinal slip, which—during braking—is defined as $\lambda = (v - \omega r)/v$, hence $\lambda \in [0, 1]$; β_t is the wheel side-slip angle (Kiencke & Nielsen, 2000); ϑ_r is a set of parameters which characterize the shape of the static function $\mu(\lambda, \beta_t; \vartheta_r)$. For simplicity, in the rest of the paper we assume that the braking maneuver is performed along a straight line, i.e., $\beta_t = 0$. Accordingly, the dependence of F_x on β_t will be omitted. Note that the aforementioned assumption is not crucial to design the proposed controller. In fact, changes in β_t cause a shift in the peak position of the $\mu(\lambda; \vartheta_r)$ curve and act as a scaling factor (in this resembling the effect due to a change in the vertical load). Thus, as our controller does not require knowledge of the road condition or of the vertical load, in the same way it handles non-zero values of β_t. Many empirical analytical expressions for the function $\mu(\lambda; \vartheta_r)$ have been proposed; a simple and widely used model is (Kiencke & Nielsen, 2000)

$$\mu(\lambda; \vartheta_r) = \vartheta_{r1}(1 - \mathrm{e}^{-\lambda\vartheta_{r2}}) - \lambda\vartheta_{r3}. \tag{2}$$

Note that the vector ϑ_r has three elements only: by changing their values many different tire–road friction conditions can be modeled. In Fig. 1 the shapes of $\mu(\lambda; \vartheta_r)$ in four different road conditions are displayed. The parameters' values for the given

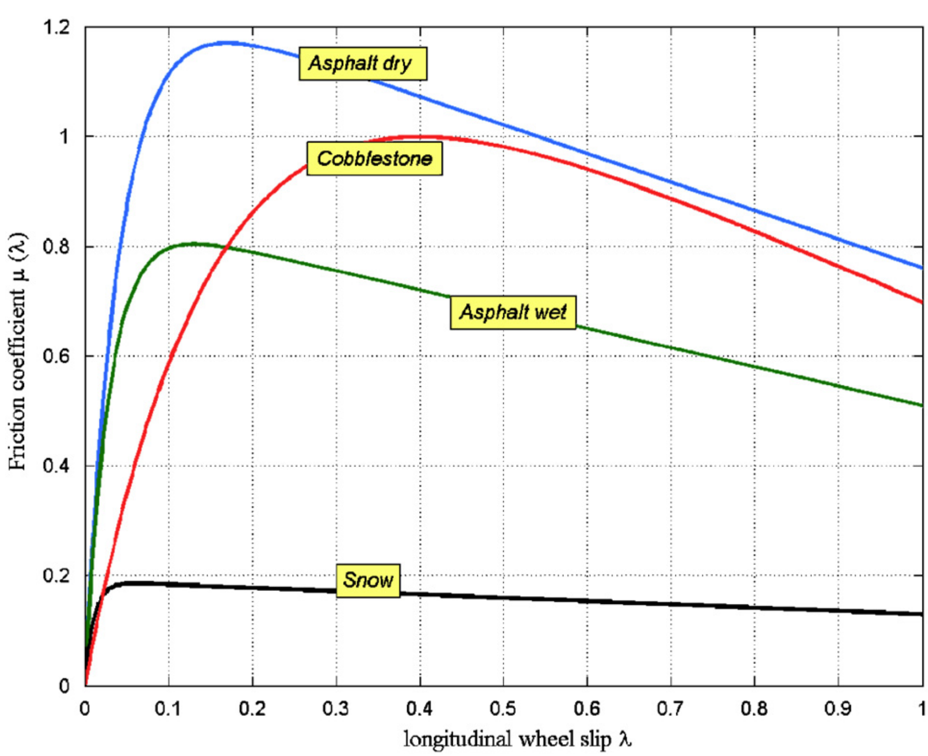

Fig. 1. Shapes of the function $\mu(\lambda; \vartheta_r)$ in different road conditions.