Control Engineering Practice 20 (2012) 69–81

Contents lists available at SciVerse ScienceDirect

Control Engineering Practice

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

Vehicle longitudinal motion modeling for nonlinear control

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ARTICLE INFO

Article history:
Received 26 November 2010
Accepted 23 September 2011
Available online 15 October 2011

Keywords:
Vehicle longitudinal motion
Longitudinal slip
Tire Kiencke's model
Speed control

ABSTRACT

The problem of modeling vehicle longitudinal motion is addressed for front wheel propelled vehicles. The chassis dynamics are modeled using relevant fundamental laws taking into account aerodynamic effects and road slop variation. The longitudinal slip, resulting from tire deformation, is captured through Kiencke's model. A highly nonlinear model is thus obtained and based upon in vehicle longitudinal motion simulation. A simpler, but nevertheless accurate, version of that model proves to be useful in vehicle longitudinal control. For security and comfort purpose, the vehicle speed must be tightly regulated, both in acceleration and deceleration modes, despite unpredictable changes in aerodynamics efforts and road slop. To this end, a nonlinear controller is developed using the Lyapunov design technique and formally shown to meet its objectives i.e. perfect chassis and wheel speed regulation.

1. Introduction

Vehicle longitudinal motion control aims at ensuring passenger safety and comfort. It is an important aspect in dynamic collaborative driving i.e. when multiple vehicles should coordinate to share road efficiently while maintaining safety. In this respect, several works have been devoted to what is commonly referred to adaptive cruise control of the main objective of which is maintaining a specified headway between vehicles (Ioannou and Chien, 1993; Moon, Moon, &Yi, 2009). Different control techniques have been used in these works including linear and adaptive control (You, Hahn, & Lee, 2009), genetic fuzzy control (Poursamad and Montazeri, 2008), sliding mode control (Liang, Chong, No, & Yi, 2003; Nouvelière and Mammar, 2007), anti-sliding control (Fang et al., 2011), scheduling gain control involving PIDs (Ren, Chen, & Chen, 2008) and estimating some state variables such as sideslip and tire force (Baffet, Charara, & Lechner, 2009). However, most previous works on longitudinal control were based on simple models neglecting important nonlinear aspects of the vehicle such as rolling resistance, aerodynamics effects and road load. In some studies, the controller performances were not formally analyzed (Ren et al., 2008). In Yamakawa, Kojima, and Watanabe (2007), longitudinal vehicle control has been studied focusing on torque management for independent wheel drive. It is worth noticing that in all previous studies on longitudinal vehicle control, the control design has been based on simple models not accounting for the tire–road interaction.

In the present study, the problem of longitudinal vehicle control is revisited, for front wheel propelled vehicles, focusing on speed regulation. The aim is to design a controller that is able to tightly regulate the chassis and wheel velocities, in both acceleration and deceleration driving modes, despite changing and uncertain driving circumstances. This problem has not been dealt with previously. A further originality of the present paper is that the control design relies upon a more complete model that accounts for most vehicle nonlinear dynamics including the tire–road interaction. That is, the study includes two major contributions. First, a suitable control model is developed for the vehicle longitudinal behavior. In this respect, recall that a convenient model is one that is sufficiently accurate but remains simple enough to be utilizable in control design. To meet the accuracy requirement, the model must account not only for aerodynamic phenomena but also, and especially, for tire–road friction. Modeling the tire/road contact is a quite complex issue involving multiple aspects relevant to tire characteristics (e.g. structure, pressure) and to environmental factors (e.g. road load, temperature). Several tire models have been proposed in the literature e.g. Guo's model (Guo and Ren, 2000), Pacejka's model (Pacejka and Besselink, 1997), Dugoff's model (Dugoff and Segel, 1970), Gim's model (Gim and Nikravesh, 1990), Kiencke's model (Kiencke and Nielsen, 2005). In the present work, Kiencke's model is retained because it proves to be a good compromise between accuracy and simplicity. Vehicle modeling is completed with chassis dynamics equations. It is carried out according to the bicycle model principle, applying the fundamental dynamics and aerodynamics laws. The full vehicle model turns out to be a combination of two nonlinear state-space representations describing, respectively, the acceleration and deceleration longitudinal driving modes.

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doi:10.1016/j.conengprac.2011.09.005

Nevertheless, it proves to be utilizable in vehicle control design. Such model development is in fact a major achievement of the present study. The other contribution is the design of a nonlinear controller that ensures global stabilization and longitudinal speed regulation during acceleration/deceleration driving modes. This is carried out based on the Lyapunov design technique (Khalil, 2002). It is formally proved that the developed controller actually achieves the stability and regulation objectives it was designed to. Furthermore, it is observed through numerical simulations that the controller is quite robust with respect to uncertainties on environmental characteristics.

The paper is organized as follows: Section 2 is devoted to modeling the acceleration/deceleration vehicle longitudinal behavior; the obtained model is used in Section 3 to design a controller and to analyze the resulting closed-loop system; the controller performances are illustrated in Section 4 by numerical simulations. Conclusion is in Section 5.

2. Modeling of chassis longitudinal motion

Except for aerodynamic forces, all external efforts acting on a vehicle are generated at the wheel–road contact. The understanding and modeling of the forces and torques developed at wheel–road contact is essential for studying properly the vehicle dynamics. These are discussed in the forthcoming sections. In this respect, recall that the vehicle motion is composed of two types of displacements: translations along the x, y, z axes and rotations around these same axes (Fig. 1).

2.1. Kiencke's tire modeling

The tire is a main component of the wheel–road contact as it ensures three important functions (Kiencke and Nielsen, 2005): (i) bearing the vertical load and absorbing road deformations; (ii) producing longitudinal acceleration efforts and contributing to vehicle braking; (iii) producing the required transversal efforts that help the vehicle turning.

The efforts generated at the wheel–road contact include longitudinal (acceleration/deceleration) forces, lateral guiding forces and self-alignment torque. The effect of these efforts on the vehicle behavior is determined by the tire–road adhesion. For small load variations, the longitudinal coefficient of friction is characterized by the following ratio:

$$\mu = \frac{F_{tx}}{F_v} \tag{1}$$

where F_{tx} denotes the longitudinal effort and F_v the vertical load. The ratio μ is called longitudinal adhesion or friction coefficient. The value of this coefficient depends on the tire slip resulting from the deformation of the tire in contact with the road (Kiencke and Nielsen, 2005). The longitudinal slip is characterized by the coefficient λ defined as follows.

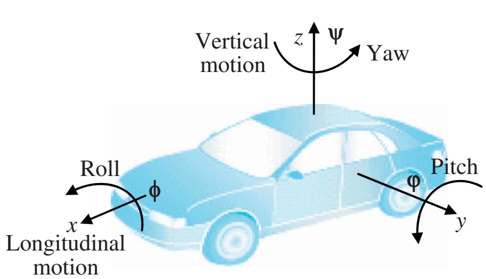

Fig. 1. Degrees of freedom of a vehicle.

In acceleration mode, i.e. $V_v < V_w$, one has

$$\lambda = 1 - \frac{V_v}{V_w} = 1 - \frac{V_v}{r_{eff}\Omega_w} \tag{2}$$

In deceleration mode, i.e. $V_v \geq V_w$, one has

$$\lambda = \frac{V_w}{V_v} - 1 = \frac{r_{eff}\Omega_w}{V_v} - 1 \tag{3}$$

where r_{eff} denotes the effective wheel radius, Ω_w designates the wheel angular velocity, $\vec{V}_w$ is the speed of the tire–road contact, $\vec{V}_v$ is the linear velocity of the wheel center (Fig. 2). A similar deformation occurs when the wheel presents a slip angle α i.e. the resulting lateral slip produces a lateral force $\vec{F}_{ty}$.

Modeling the efforts at the wheel–road contact has been given a great deal of interest over the last years. In this respect, several tire models have been developed with quite different properties, e.g. Guo and Ren (2000), Pacejka and Besselink (1997), Dugoff and Segel (1970), Gim and Nikravesh (1990), and Kiencke and Nielsen (2005). For control design use, the most suitable tire model is one that presents the best accuracy/simplicity compromise. From this viewpoint, Kiencke's model turns out to be a quite satisfactory choice (Kiencke and Nielsen, 2005). Indeed, this model is sufficiently accurate as it accounts for the main features such as the vertical load F_v, slip angle α, slip coefficient λ. On the other hand, it has already proved to be useful in designing simple estimators for state variables like slip angle and lateral efforts (You et al., 2009). In the present paper, this model will prove to be useful in control design.

2.2. Kiencke's model

This was developed in Kiencke and Nielsen (2005) using Burckhardt's extended model to compute the friction coefficient μ. Accordingly, the latter is a function of the combined longitudinal/lateral slip coefficient κ and the forces acting on the tire. Fig. 3 gives a schematic representation of Kiencke's wheel model where:

$$\mu = C_1(1-\exp(-C_2\kappa)) - C_3\kappa\exp(-\kappa C_4 V_G)(1-C_5F_v^2) \tag{4}$$

$$\kappa = \sqrt{\lambda^2 + \kappa_y^2} \tag{5}$$

$$\lambda = 1 - \frac{V_v}{V_w} \text{ and } \kappa_y = (1-\lambda)\tan(\alpha) \text{ (acceleration)} \tag{6}$$

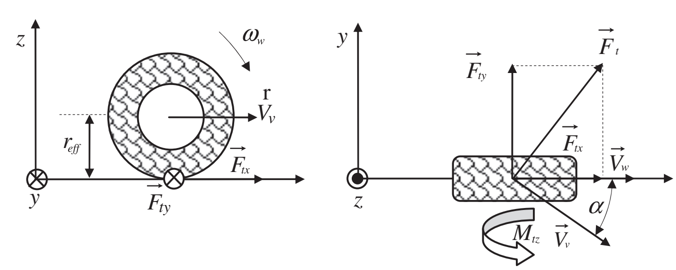

Fig. 2. Forces applied on the wheel.

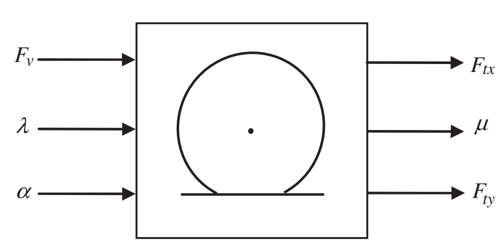

Fig. 3. Kiencke's wheel model.