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Model predictive control of vehicle stability using coordinated active steering and differential brakes☆

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ABSTRACT

This paper studies model predictive control of lateral stability of vehicles using coordinated active front steering and differential brakes. The controller is designed based on a bicycle model of the vehicle and the moment of the differential brakes is considered as an external torque. The prediction model calculates the prospective values of the vehicle's yaw rate, lateral velocity, and tire slip angles over the prediction window. The sideslip angle of the vehicle is enforced within a permissible range using soft constraints on the lateral velocity in order to guarantee persistent feasibility. Using computer simulations, the controller is shown to provide proactive control actions to control the vehicle's sideslip angle. The closed-loop response of the controller is also studied in experimental tests on an instrumented test vehicle. The results show satisfactory performance in various combinations of active front steering and differential brakes. In addition, the computational time of the controller is measured and shown to be safely below the sample time of the controller.

1. Introduction

Active safety features such as anti-lock braking system (ABS), traction control (TC) and electronic stability programs (ESP) have resulted in a significant reduction of single vehicle accidents [1–3]. In efforts towards zero accidents, advanced vehicle stability control systems continue to attract the attention of researchers.

Differential braking has been the center of attention in vehicle stability control over the past few decades. In this method, if vehicle oversteer is detected, the outer wheels of the vehicle are braked to reduce the yaw moment on the vehicle C.G. and control the sideslip angle. On the contrary, when the vehicle understeers, the inner wheels are braked to boost the steering yaw moment and improve maneuverability of the vehicle. Examples of the use of differential brakes in stability control include [4–7].

With the rise in market share of electric vehicles, torque vectoring is considered as an attractive alternative to differential brakes. Torque vectoring has the advantage of not affecting longitudinal vehicle dynamics. Therefore, it is a suitable choice for continuously adjusting the yaw rate response of the vehicle. Instances of stability control approaches using torque vectoring in the literature include [8–12].

With the advent of many intelligent features such as lane centering and lane keep assist, an increasing number of production vehicles are being equipped with active front steering (AFS) systems. This provides an opportunity to utilize the AFS systems in stability control programs in addition to the traditionally used actuation methods.

There have been several studies into the integration of active steering systems with differential brakes or torque vectoring. Some authors have studied a multi-level control architecture, where a high-level controller calculates the desired forces and moments at the vehicle C.G., and a low-level controller generates these forces and moments using the available actuators. For instance, Li et al. [13] designed an integrated two-level controller to achieve vehicle stability and yaw rate tracking. In the high level controller, the stabilizing forces and moments were calculated using a sliding mode scheme. At the distribution level, the required forces and moments were translated into wheel steering and wheel torque. Software simulations of step-steer and double-lane change maneuvers indicated improved handling response of the vehicle. Tjonnas and Johansen [14] studied a multi-level modular control structure for yaw rate tracking and indirect sideslip angle control. The control allocation problem is cast as a dynamic optimization problem that considers the actuator constraints. Differential brakes and active steering are the actuation mechanisms studied in this paper. Software simulations were conducted to assess the yaw rate tracking performance in typical understeer and oversteer situations. A similar multi-level control structure is used in [15,16].

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Nomenclature

α_i	slip angle of tires on axle i
δ^+	controller's wheel steering angle adjustment
δ_f	front wheel's steering angle requested by driver
$\hat{\alpha}_i$	slip angle of tires on axle i with zero steering angle
$\widehat{C}_{\alpha i}$	estimated value of cornering stiffness of axle i
ψ	vehicle's heading angle
a_y	vehicle's lateral acceleration
$C_{\alpha i}$	cornering stiffness of axle i
l_i	distance from C.G. to the center of axle i
M_x	yaw moment of the longitudinal tire forces
r	vehicle's yaw rate
R_w	effective radius of wheels
T_{ij}	total torque applied on wheel ij
T_{ij}^{drv}	driver's torque request on wheel ij
v_x	vehicle's longitudinal velocity
v_y	vehicle's lateral velocity
w_i	trackwidth of axle i
AFS	active front steering
AIT	acceleration-in-turn
CPU	central processing unit
DB	differential brakes
LPV	linear parameter varying
PWA	piecewise affine
RWD	rear-wheel drive

Despite advantages in modular control systems, separation of the actuation layer from vehicle dynamics means that the actuator dynamics and constraint are not explicitly considered in the closed-loop system and the optimality of the solution is compromised. For this reason, many authors adopt a holistic approach that results in globally optimal control actions. For instance, Zheng and Anwar [17] studied the yaw stability of vehicles using active steering. They designed a second order controller that decouples yaw and lateral motions of the vehicle using yaw rate and steering angle feedback. Experimental tests were conducted to verify the ability of the controller to maintain vehicle stability. The controller was then implemented in experimental tests. Yang et al. [18] proposed a coordinated use of active front steering and direct yaw control (differential brakes) using an optimal guaranteed cost controller (OGCC) approach. They used simulations to compare the performance of the proposed OGCC and an optimal coordination OC scheme based on an LQR approach. Slalom and single-lane change maneuvers on different road conditions were studied and the superiority of the OGCC approach was concluded. A similar holistic control structure is used in [19,20].

Model predictive control is an ideal candidate for the holistic vehicle stability control. MPC can explicitly address actuator and state constraints and provide proactive control actions. The main drawback of the MPC approach is its computational cost. However, vehicles are nowadays equipped with more powerful micro-processors, making MPC an attractive choice. Ren et al. [21] studied an integrated torque vectoring and active steering system for vehicle yaw rate tracking and traction control. They developed a nonlinear tire model and used a nonlinear MPC to solve the control problem. Software simulations were used to verify the performance of the controller in lane change maneuvers on dry and slippery surfaces. Cairano et al. [22] developed an MPC controller for yaw rate tracking using differential brakes and active front steering. They approximated the lateral tire forces with a piecewise affine (PWA) relationship in terms of the tire slip angles. A switching MPC controller was synthesized and tested in simulations and experimental tests on slippery surfaces. Falcone et al. [23] studied trajectory following in autonomous vehicles using model predictive control. They designed two controllers: one using a nonlinear prediction model, and one using a linear model obtained by successive linearization of the vehicle model at each time step. Under the assumption of prior knowledge of the desired trajectory, the controller calculated the front wheel steering angles so that the vehicle remains on path, even on slippery road conditions. Simulation and experimental tests were conducted to test the path tracking capability of the controlled vehicle in a double-lane change maneuver. Other notable examples of using MPC with active steering include [24,25].

The main contribution of the present paper is the development of a model predictive controller for vehicle stability control using both differential brakes and active front steering. The stability requirement is translated into a soft constraint on system states that guarantees persistent feasibility of the controller. The cornering stiffness of the front and rear axles are estimated using instantaneous measurements of the vehicle's lateral and yaw accelerations; therefore, the controller does not require explicit knowledge of road friction. Full spectrum of actuations from only differential brakes to only active front steering can be obtained simply by adjusting the controller gains. The designed MPC controller is tested in software simulations and experimental tests with an instrumented electric Chevrolet Equinox where high performance of the designed controller is demonstrated.

This paper is organized in 5 sections. In Section 2, the prediction model, objective function, and constraints of the MPC controller are developed. In Section 3, software simulations are carried out to investigate the closed-loop response of the vehicle in a critical driving situation. Experimental tests are presented in Section 4 where the performance of the controller is studied in an acceleration-in-turn maneuver on a slippery surface. In Section 5, the summary of the findings of this paper are provided.

2. Controller design

The predictive stability controller is designed in this section. First, a prediction model is introduced based on a bicycle model. This model is then represented in a discrete-time state-space format. Next, the objective function and input and output constraints are developed. These form a quadratic programming problem that is solved in real-time using a numerical toolbox. In this study, the driver's model is excluded from the control loop. The role of the driver in vehicle stability is considered in other studies, such as in [26,27].

2.1. Prediction model

The prediction model is based on a bicycle model of the vehicle (see Fig. 1), where the yaw moment of the longitudinal tire forces are added as an external moment at the C.G. of the vehicle. The states of the prediction model include:

$$\mathbf{x} = [r \quad v_y \quad \hat{\alpha}_f \quad \hat{\alpha}_r]^T \tag{1}$$

where r stands for the vehicle yaw rate, v_y is the vehicle's lateral velocity and $\hat{\alpha}_i$ is the slip angle of the front ($i = f$) and rear ($i = r$) tires at zero wheel steering angle. The inputs of the prediction model are:

$$\mathbf{u} = [T_{fl} \quad T_{fr} \quad T_{rl} \quad T_{rr} \quad \delta^+ \quad \epsilon]^T \tag{2}$$

where T_{ij} is the total torque on the wheel ij, δ^+ is the additive front wheel steering angle correction of the controller, and ϵ is the slack variable associated with the soft constraint on the lateral velocity, which is further explained in Section 2.2. The vehicle's yaw acceleration can be expressed in terms of the tire forces:

$$I_z\dot{r} = l_f C_{\alpha f}\alpha_f \cos\delta_f - l_r C_{\alpha r}\alpha_r + M_x \tag{3}$$

where δ_f is the steering angle of the front wheels as requested by driver,