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Drift control for cornering maneuver of autonomous vehicles

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ABSTRACT

Expert drivers have the ability to perform high side-slip angle maneuvers, like drifting, during racing to minimize lap time or avoid obstacles. Designing planning and control algorithms for autonomous drift maneuvers, however, is challenging because of the high lateral motion and nearly full saturation of rear tires. In this paper, the authors propose a complete path planning and motion control framework to plan and track a reference drift trajectory along a sharp bend in a track. The path planner divides the path horizon into three regions, finds a path using different planning sub-modules, and then concatenates the solutions to generate the reference trajectory. The controller then applies a mixed open-loop and closed-loop scheme to track the reference trajectory. We validate the planning and control algorithms in simulation using a high-fidelity model in Simulink/Carsim, and through experimentation using 1/10 scale Radio-Control (RC) vehicle.

1. Introduction

Drifting occurs when an expert driver intentionally maneuvers a vehicle to cause loss of traction in the wheels, characterized by large side-slip angles and near full saturation of the wheels. It is commonly seen in rally racing when a driver quickly turns a corner.

Drifting represents a particularly interesting control maneuver because of the tire saturation and limited control authority in a highly unstable region. Current chassis control systems, like anti-lock braking system (ABS) and traction control system (TCS), try to prevent drifting conditions from ever arising [1,2], but experimental evidence shows the high-drift maneuvers may be more efficient from the minimum time point of view or obstacle avoidance. In rally racing, expert drivers often bring the vehicle into a drift state in order to reduce lap time or avoid collisions, while still maintaining control of the vehicle. To better understand these dynamics, Velenis et al. analyzed the behavior of expert drivers during drift and provided an empirical description of the sequence of steps the driver implements to initiate and control drift [3,4].

Most research on drift maneuvers fall into one of two categories: sustained drift and transient drift. Sustained drift focuses on stabilizing the vehicle about an unstable equilibrium state, resulting in a steady state circular drift. Transient drift focuses on entering a drift state temporarily to perform a maneuver, like drift parking.

For sustained drift, researchers have used vehicle models of varying fidelity to study drift dynamics, ranging from a two-state bicycle model [5,6] to a seven-state vehicle model [7,8]. Regardless of model fidelity, the system model must accurately capture the tire forces that emerge throughout drift, especially when the tires saturate. The Fiala tire model [9] and the Pacejka tire model [7,10] have been used to capture these forces. For the control design of sustained drift, the central feature lies the coordination between steering and rear drive torque [7,9,11,12]. Gonzales et al. designed a controller by linearizing the vehicle model around one of its drift equilibria and then used a linear quadratic regulator (LQR) feedback policy to compute the steering angle and rear drive [12]. Hindiyeh et al. applied dynamic surface control to balance the inputs and enabled 'steering' of rear tire through novel usage of rear drive for lateral control [9]. Additionally, Hindiyeh provided stability guarantees in the control design using Lyapunov-based techniques. Outside of model-based optimal control, Cutler applied reinforcement learning with a motion capture system to achieve sustained drift [13–15].

Work on transient drift has also emerged as a research topic for vehicle applications [4,16–21]. Chakraborty et al. investigated methods for mitigating unavoidable collisions using nonlinear optimization. They found handbrake cornering drift to be optimal maneuver in some situations [16,17]. A probabilistic control strategy called multi-model LQR was applied by Kolter to slide a vehicle into a parking spot [18,19]. Velenis et al. reproduced in simulation a trail braking maneuver using nonlinear optimization. The nonlinear program was formulated to achieve maximum corner exit speed or minimal cornering time, using vehicle model with suspension dynamics as constraints [4,20].

The common strategies to control transient drift maneuvers are

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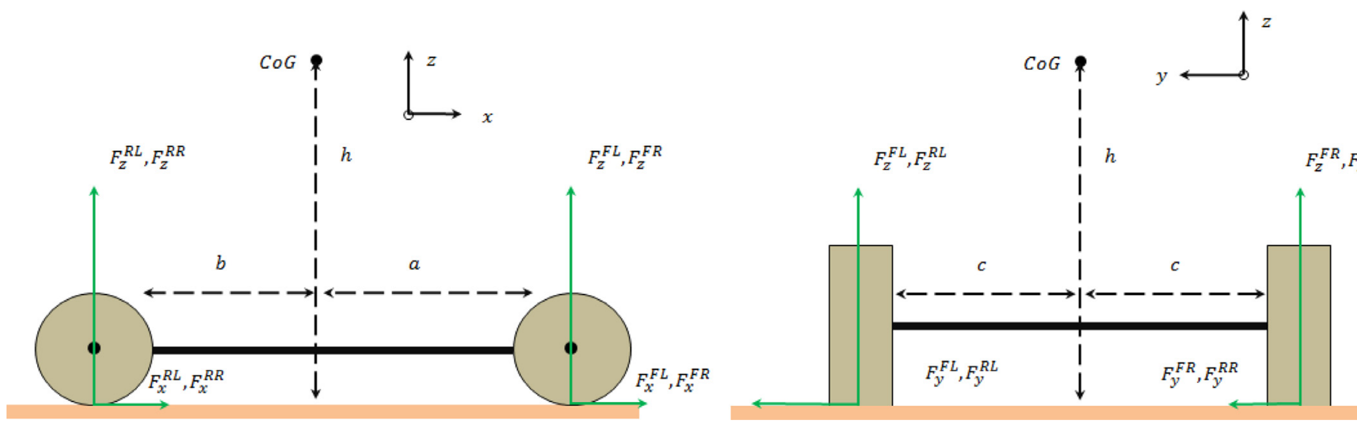

Fig. 1. x-z plane and y-z plane of the vehicle.

based on nonlinear control methods that use a high dimensional vehicle model. These models require numerous parameters, many of which are difficult to estimate, as well as significant computational resources, which makes them unsuitable for fast real-time implementation. In this paper, the authors extend previous work [22] and present a framework to plan and track a drift trajectory. Specially, the main contributions of this paper are:

- A hybrid path planning algorithm to generate a reference trajectory for drift. The planner divides the path horizon into three different types of regions, finds a path for each region using different planning algorithms (Rapidly-exploring Random Trees, rule-based sampling, Proportional Integral control) with different vehicle models, and then concatenates the solutions of each to construct the reference trajectory.
- A mixed open-loop and closed-loop control technique based on the standard bicycle model with linear tire model to track the drift trajectory, which is experimentally validated through a RC platform.

The remainder of this paper is organized as follows. Two kinds of vehicle models and tire models are discussed and compared in Section 2. Then, a hybrid rapidly-exploring random trees (RRT) and rule-based path planning algorithm is presented in Section 3. Section 4 summarizes mixed open-loop and closed-loop control strategy. Simulation and experimental results are presented in Section 5. Concluding remarks are given in Section 6.

2. System model

This section discusses vehicle models for the path planner and controller. Ideally, a single model would be used through the entire architecture, but conventional models, like the four-wheel model, begin to break down as the vehicle enters a drift state. The following subsections describe the low and high fidelity vehicle models that are used in the path planner and controller.

2.1. Four-wheel vehicle model

We use a planar four-wheel vehicle model from Falcone [23] to capture the vehicle dynamics, which models the vehicle as a single rigid body with forces acting at each of the wheels. The tire forces are modeled using the Pacejka tire model, which is a semi-empirical model similar in mathematical structure to physics-based models, which is a semi-empirical model based on fitting a curve to experimental data. The four-wheel vehicle model and Pacejka tire model describe the planar motion of the vehicle at the center of mass. The vehicle state and input are $z = [U_x, U_y, r, X, Y, \psi, \omega_{FL}, \omega_{FR}, \omega_{RL}, \omega_{RR},]$ and $u = [\delta, F_x^{RL}, F_x^{RR}]$, respectively. U_x, U_y and r are the vehicle's longitudinal speed, lateral speed and yaw rate in the body-fixed frame, and X, Y, ψ describe the position and yaw angle of the vehicle in earth-fixed frame. The four variables ω_{ij} are the wheel angular speed in tire-fixed frame, where the first subscript $i \in \{F, R\}$ indicates either the front or rear axle, and the second subscript $j \in \{L, R\}$ indicates either the right or left side. δ and F_x are the steering angle and rear drive force, respectively. For sake of brevity, we do not present the full set of equations from the models, but instead discuss only modifications to the model that take weight transfer into account.

Weight transfer means that the normal force of each tire (i.e. force in the vertical direction $F_z^{i,j}$) can change over time, especially when the yaw rate and acceleration are large. By assuming the vertical acceleration is zero and all rotations occur about the center of mass, we apply the following force constraints

$$F_z^{FL} + F_z^{FR} + F_z^{RL} + F_z^{RR} = mg \tag{1}$$

$$F_z^{FL} + F_z^{RR} = F_z^{FR} + F_z^{RL}. \tag{2}$$

For each force acting on the wheel, we use the notation $(\cdot)_k^{i,j}$, where i, j indicates the wheel, and $k \in \{x, y, z\}$ indicates the directional component of the force in the body frame of the vehicle. The $x - z$ and $y - z$ planes of vehicle are shown in the Fig. 1. We also apply balance of angular momentum equations about the center of mass

$$(F_z^{FL} + F_z^{FR})a + (F_x^{FL} + F_x^{FR} + F_x^{RL} + F_x^{RR})h = (F_z^{RL} + F_z^{RR})b \tag{3}$$

$$(F_z^{FL} + F_z^{RL})c + (F_y^{FL} + F_y^{FR} + F_y^{RL} + F_y^{RR})h = (F_z^{FR} + F_z^{RR})c \tag{4}$$

where m is the mass of the vehicle, g is the acceleration due to gravity, a, b are the distances from the center of gravity (CoG) to the front and rear axles, respectively. c is the distance from vehicle longitudinal axis to the wheels and h is the distance from the CoG to the ground.

The dynamics of the system are compactly expressed as

$$\dot{z}(t) = f^{4w}(z(t), u(t)). \tag{5}$$

where the superscript $4w$ indicates the four-wheel model.

2.2. Model accuracy

The four-wheel model from the previous section accurately describes the motion of the vehicle under typical driving conditions (i.e. non-extreme maneuvers), when the slip angle is small. This model begins to break down, however, once the slip angle grows to large values. To illustrate this, we compare the output of the four-wheel model and a high-fidelity vehicle model from CarSim. Both models use the same system parameters and the same input sequence that cause a large slip angle to emerge. The resulting trajectories are shown in Fig. 2a.

From the resulting trajectories, we observe that the four-wheel model and the CarSim model are nearly identical as the vehicle drives