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Application of Local Linear Steering Models with Model Predictive Control for Collision Avoidance Maneuvers

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Abstract:
Collision avoidance systems demand sophisticated control algorithms to ensure driving on a safe preplanned path. Model predictive control (MPC) is suitable for this task as different influences can be considered by the algorithm. However MPC requires a model of the plant to guarantee good control characteristics. Especially the task to generate a simple steering model covering the considered range of vehicle dynamics is challenging. In this work a concept for an adaptive steering model is presented. The adaptive steering model is integrated in a model predictive control concept for collision avoidance maneuvers. Simulation results show improvement in prediction quality, while achieving modest improvement in control performance.

Keywords: Automotive, collision avoidance, model predictive control, local linear model tree, steering system

1. INTRODUCTION

Vehicle dynamics control for evasive maneuvers have been the focus of different work for collision avoidance e.g. Schorn (2007), as it allows lower distance to the object at higher velocities compared to braking maneuvers when starting the maneuver, see Moshchuk (2015).

In order to assist the driver in emergency situations only, the principle to react at the last possible moment is followed, so that the system intervenes only in critical situations.

Model predictive control (MPC) has been one of the most popular control methods in the last years and is presented and discussed in Borrelli (2003). The idea of the control method is to use a time discrete model of the plant and predict the future behaviour of the system. According to a cost function and an optimization problem the control output with the lowest cost value will be chosen.

Several research works have been conducted for vehicle dynamics control with MPC. Falcone et al. (2007) presented linear time variant MPC approaches using a single track model and steering wheel angle as command signal combined with a solver for quadratic programming problems. Katriniok (2013) additionally incorporate the longitudinal deceleration into the MPC problem for combined braking and steering. All of the above works use the steering wheel angle as command for the system.

Most vehicles use steering torque as intervention interface. Steering torque ensures good interaction of lane keeping and lane centering concepts with the driver. Mature steering systems provide different functionality like friction compensation, torque amplification curves (boost curve) for different speeds and other arbitration modules. These features help the driver to handle complex and fast changing vehicle dynamics states and increase safety as well as user experience in the vehicle. Influenced by these features, the analytical steering model will change its character according to different vehicle dynamics conditions. Modelling of a steering system is thus a crucial task, to provide predictability for the overall vehicle dynamics control especially in maneuvers with short duration. Keller et al. (2015) proposed a cascaded control concept to control the demanded steering wheel angle in an inner loop by a linear controller thus using steering wheel torque as an interface for the vehicle. However tuning of the controller for different vehicle dynamics conditions is a tough task. Moshchuk (2015) provided a control concept using analytical steering system with constant parameters, which will further be used in this work.

The main contributions of this paper are as follows:

- The problem of steering system characteristics at different conditions is described. Thus the need for appropriate modelling, considering different vehicle dynamics conditions of the steering system, is deduced.
- Further, an adaptive model predictive controller for collision avoidance maneuvers is proposed. To improve prediction quality at different vehicle dynamics states, state dependant steering parameters are learned using local linear model trees, described in Nelles (2001).
- Results in a simulation environment with an embedded algorithm of the electric steering control module for different trajectories show the effectiveness of the proposed methods compared to constant steering parameters with respect to prediction quality.

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2. VEHICLE DYNAMICS DEPENDANT STEERING SYSTEM CHARACTERISTICS

Electrical steering support is provided in many vehicles to ensure comfortable and safe driving on the street. One of the functions of this support is to guarantee low actuation force for the driver in different speed ranges, which is done by using boost curves, describing the speed dependant torque amplification character. Also friction compensation, active steering wheel centering and other complex functionalities are integrated in the steering system. Parallel to these fundamental functions a steering torque interface needs to be provided for active safety driver assistance system functions. These functions need to work neatly with existing safety concepts.

In this work an Opel Insignia model year 2014 is used. It is equipped with an EPS (Electrical Power Steering) steering actuator, providing a 6Nm steering wheel torque interface. In the following we will show the causal relation with the steering system and the vehicle dynamics in Fig.1, which will be considered in our control concept. The EPS

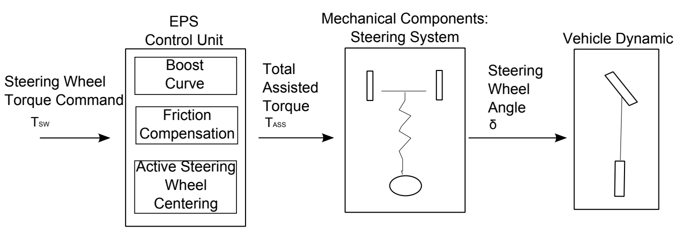

Fig. 1. Plant structure with Steering- and Vehicle Dynamics System

control unit receives a steering wheel command torque. Here different function modules have an influence on the total assisted torque provided by the electrical machine. The boost curve has the main influence on the steering system behaviour. It translates the commanded torque into a resulting total assisted torque generated by the electrical device. The amplification rate is speed dependant and is illustrated in Fig. 2 schematically. Here v_0 is a lower vehicle speed, therefore higher assistance torque is demanded at the same torque command than for a higher speed v_1. The applicable total torque is limited due to actuator limits of the steering system. In a next step the applied total assisted torque leads to a change in the steering wheel angle according to the mechanical dynamic of the steering system. In this work we use a simple equation

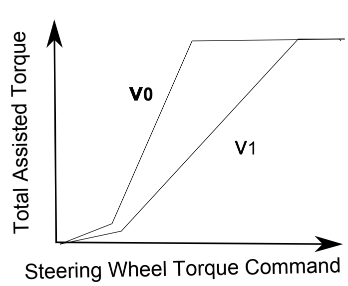

Fig. 2. Boost Curve for different velocities with $v_0 < v_1$

from Moshchuk (2015) to represent the steering system behaviour.

$$J_{Str}\ddot{\delta} + d_{Str}\dot{\delta} + D_{Str}\alpha_f = T_{ASS} \tag{1}$$

In the equation above, δ is the steering wheel angle, J_{Str} is the inertia of the steering wheel and steering column, T_{ASS} is the total assisted torque, α_f is the slip angle at the tires of the front axis, d_{Str} is the damping parameter, D_{Str} is the self alignment constant consisting of the cornering stiffness and pneumatic trail length.

Apart from boost curves different functionalities are implemented in the steering system like friction compensation and active steering wheel centering. A steering model with a simple structured steering system like (1) cannot consider all these features. Collision avoidance maneuvers induce transitions through a high range of vehicle dynamics accelerations, in which the features mentioned above have different effects on the vehicle dynamics. The collision avoidance algorithm needs to consider changing behaviour due to these features as well as changing environmental parameters like self-alignment torque at different speeds. Thus the steering parameters will be adapted using local linear model tree, which will be presented in the next sections. The main focus of this work lies in an easy and adaptive concept to consider parameter changes during control of a vehicle in different vehicle dynamics operation conditions.

3. VEHICLE DYNAMICS MODEL

3.1 Vehicle Model and Tire Model

Vehicle dynamics modeling has been discussed in various studies e.g. by Schorn (2007) and Mitschke and Wallentowitz (2004). In this section the single-track model is derived. The dynamical equations representing the lateral vehicle dynamics are given by

$$mv(\dot{\beta} + \dot{\psi}) = F_{yf} + F_{yr}, \tag{2}$$

$$J_z\ddot{\psi} = l_f F_{yf} - l_r F_{yr} \quad . \tag{3}$$

Here m is the mass of the vehicle, v is the vehicle speed, J_z is the inertia for the z-axis, l_f and l_r are the distances of the center of gravity to the front and rear axis respectively. β is the side slip angle, ψ is the yaw angle, $\dot{\psi}$ is the yaw rate, F_{yf} and F_{yr} are the lateral forces generated by the tires at the front and rear axis.

The lateral force is calculated using Pacejka's tire formula, given by

$$F_y(s_\alpha) = D\sin\{C\arctan[Bs_\alpha - E(Bs_\alpha - \arctan(Bs_\alpha))]\} \tag{4}$$

where s_α is the lateral tire slip given by $s_\alpha = \sin(\alpha)$ and B, C, D, E are tire specific parameters. In order to simplify the equation, Pacejka's tire formula is linearized around the operation point according to Choi and Choi (2014), such that the linear equation is given by

$$F_{yf} = c_{\alpha_f}\alpha_f + F_{yf0}, \tag{5}$$

$$F_{yr} = c_{\alpha_r}\alpha_r + F_{yr0} \quad . \tag{6}$$

with the front and rear slip angle

$$\alpha_f = \frac{\delta}{i_L} - \beta - \frac{l_f}{v}\dot{\psi}, \tag{7}$$

$$\alpha_r = -\beta + \frac{l_r}{v}\dot{\psi} \tag{8}$$