Available online at www.sciencedirect.com

ScienceDirect

IFAC PapersOnLine 50-2 (2017) 265–270

Model Predictive Lateral Vehicle Guidance Using a Position Controlled EPS System

Jochen Gallep * **Vivan Govender** ** **Steffen Müller** ***

* *Department of Automotive Engineering, Technical University of Berlin, Berlin, Germany, (e-mail: jochen.gallep@tu-berlin.de)*
** *Daimler AG, Böblingen, Germany, (e-mail: vivan.govender@daimler.com)*
*** *Department of Automotive Engineering, Technical University of Berlin, Berlin, Germany, (e-mail: steffen.mueller@tu-berlin.de)*

Abstract: On the way to automated driving it is necessary to develop suitable control structures which ensure an accurate and robust path tracking during different maneuvers and speeds. This paper presents a Model Predictive Controller (MPC) for lateral vehicle guidance which calculates an optimal manipulated variable sequence for an inner low level steering angle controller of an Electric Power Steering (EPS). On the basis of prior work, a linear controller design is used to control the front steer angle and the resulting steering control loop behaviour is explicitly considered in the prediction model of the outer MPC. The MPC uses a speed-dependent adaptation of the prediction model and the cost function weights to ensure a stable and precise path tracking performance. The MPC real-time capability will be assessed and the performance of the proposed controllers evaluated.

Keywords: Autonomous vehicles, linear time-varying MPC, path tracking, steering angle control, electric power steering, steering analysis

1. INTRODUCTION

The lateral guidance of autonomous vehicles using Electric Power Steering (EPS) systems places high demands on the control accuracy of the lateral controller. This includes, in particular, a good reference tracking during different driving maneuvers to avoid potential dangerous situations. In order to reduce the lateral displacement while taking into consideration the future heading angle as well as EPS limitations, optimal reference steering angles values need to be calculated. The first step of the optimal control sequence is then used as an input for the subordinated steering angle controller. Due to the absence of a human driver, the steering wheel is free moving, which changes the steering system dynamics compared to a non-autonomous vehicle. These new steering dynamics is controlled using an angle controller, that ensures a predictable and accurate steering position control that can be considered in the outer lateral controller.

A possible control strategy for the outer lateral controller in the context of autonomous driving is Model Predictive Control (MPC). MPC honours constraints of the steering angle δ and steering angle velocity $d\delta/dt$ during the calculation of the optimal manipulated variable sequence. Because of the receding horizon principle, future course information is taken into account. Keviczky et al. (2006) introduces a MPC with nonlinear prediction model for lateral vehicle guidance. The calculation of the nonlinear optimization problem causes a high computational burden, which is a drawback for practical implementations. Katriniok and Abel (2011) and Turri et al. (2013) thus use linearizations of the nonlinear prediction model to calculate the manipulated variable. Additionally the calculation of the lateral displacement references to the vehicles center of gravity. Within their work, the controller performance is tested with lane change and evasive maneuvers at a constant speed of $15\,\mathrm{m/s}$.

In experimental test vehicles the position of the center of gravity is load-dependent and therefore not precisely known without prior measurements. Thus a clearly defined geometric reference point on the vehicle is used such as the intersection between the central axis and the front axle axis.

Within this work, the influence of the reference point on the controller design and controller performance will be examined. Furthermore, the operating area of the controller will be extended to different maneuvers and velocities. The focus will be on low speeds ($1-2\,\mathrm{m/s}$) with small curve radius ($<10\,\mathrm{m}$) as they occur during parking and manoeuvring through narrow spaces. An addition focus is on high dynamic manoeuvres at increased speed.

2. SYSTEM OVERVIEW

Figure 1 shows the path control system consisting of Vehicle, EPS and the MPC lateral controller. The reference heading angle $\Theta_{ref} = arctan(dy/dx)$ is calculated on the basis of the reference path information in xy-coordinates, which the car needs to follow.

The MPC receives the actual velocity v, yaw rate $\dot{\psi}$, yaw angle ψ, lateral displacement dy and the steering angle at the front axle δ. The MPC manipulated variable

10.1016/j.ifacol.2017.12.056

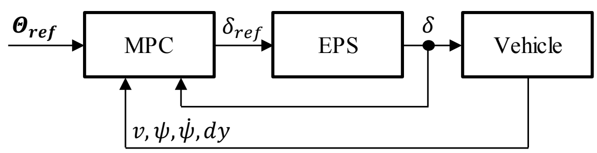

Fig. 1. Path control system

commands δ_{ref} are propagated to a subordinated steering angle controller inside the EPS.

3. PREDICTION MODEL

An important element of the MPC is the prediction model, which is used to predict the future vehicle behaviour. The key objective is to predict the lateral displacement dy for future manipulated variable commands δ_{ref}. Figure 2 shows all relevant variables for the prediction of dy. The variable v_v represents the velocity vector of the front wheel which encloses the angle γ relative to the inertial system.

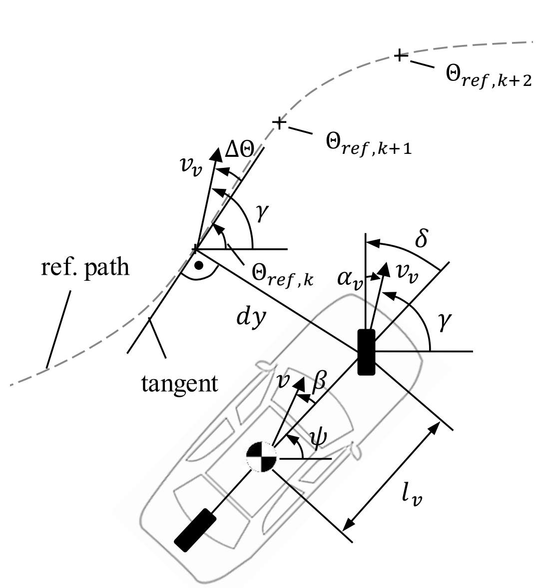

Fig. 2. Single track model and reference path

The discrete reference heading angle sequence $\Theta_{ref,k+i}$ represents the discrete reference heading angles for discrete time steps ahead of the vehicle. On the Basis of trigonometrical relationships the angle γ is calculated according to the following expressions:

$$tan(\delta - \alpha_v) = \frac{vsin\beta + l_v\dot{\psi}}{vcos\beta} \approx \beta + l_v\frac{\dot{\psi}}{v} \tag{1}$$

$$\Rightarrow \gamma = \psi + \delta - \alpha_v = \psi + \beta + l_v\frac{\dot{\psi}}{v}. \tag{2}$$

The lateral displacement rate results from the velocity v_v and small heading angle errors $\Delta\Theta = \gamma - \Theta_{ref}$ to:

$$\begin{aligned} d\dot{y} &= v_v sin(\gamma - \Theta_{ref}) \approx v(\gamma - \Theta_{ref}) \\ &= v(\psi + \beta + l_v\frac{\dot{\psi}}{v} - \Theta_{ref}). \end{aligned} \tag{3}$$

The calculation of the states $\dot{\psi}$ and β will be performed through a dynamic and a kinematic single-track model. The kinematic model describes the vehicle dynamics less accurately, but can be simply parametrized because it uses the wheel base as the only vehicle parameter and the model can also be used in case of zero velocity.

3.1 Kinematic Single Track Model

Ignoring vehicle mass and inertia, $\dot{\psi}$ and β are geometrically defined using Ackermann steering angle and can be written as

$$\dot{\psi} = \frac{v}{R} = v\frac{\delta}{L} \tag{4}$$

$$\beta \approx \frac{L}{2R} = \frac{\delta}{2}. \tag{5}$$

R corresponds to the curve radius and $L = l_v + l_h$ to the wheelbase.

3.2 Dynamic Single Track Model

The linear single-track model according Mitschke and Wallentowitz (2014) describes the lateral vehicle dynamics with a state-space representation and the parameters cornering stiffness (c_α), mass moment of inertia around the vertical axis (J), vehicle mass (m) and the distances between CoG and front/rear axle (l_v, l_h) as follows:

$$\frac{d}{dt}\begin{bmatrix}\dot{\psi} \\ \beta\end{bmatrix} = \begin{bmatrix} a_{1,1}(v) & a_{1,2} \\ a_{2,1}(v) & a_{2,2}(v)\end{bmatrix}\begin{bmatrix}\dot{\psi} \\ \beta\end{bmatrix} + \begin{bmatrix}\frac{c_{\alpha,v}l_v}{J} \\ \frac{c_{\alpha,v}}{vm}\end{bmatrix}\delta \tag{6}$$

$$a_{1,1}(v) = -\frac{c_{\alpha,v}l_v^2 + c_{\alpha,h}l_h^2}{vJ} \tag{7}$$

$$a_{1,2} = -\frac{c_{\alpha,v}l_v - c_{\alpha,h}l_h}{J} \tag{8}$$

$$a_{2,1}(v) = -1 - \frac{c_{\alpha,v}l_v - c_{\alpha,h}l_h}{v^2m} \tag{9}$$

$$a_{2,2}(v) = -\frac{c_{\alpha,v} + c_{\alpha,h}}{vm}. \tag{10}$$

3.3 Steering Dynamics

The reference value δ_{ref} calculated by the MPC needs to be controlled by the subordinated steering angle controller inside the EPS. The purpose of the steering angle controller is to reproduce the steering inputs achieved today by a human driver. Work done by Beck (2017) using a conventional EPS system examines a wide range of measurement data to reproduce the full operational bandwidth of the steering system. Using the power spectral density to analyse the data in the frequency domain and plotting the integral of the power spectrum over frequency, it can be seen in figure 3, that the bandwidth of 3 Hz represents 99 % of the energy excitation within the steering system. This