

A Tire Slip-Angle based Speed Control Driver Model for Analysis of Vehicle-Driver Systems at Limit Handling

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ABSTRACT

This paper presents a tire slip-angle based speed control race driver model. In developing a chassis control system for enhancement of high-speed driving performance, analysis of the vehicle-driver interaction at limit handling is one of the main research issues. Thus, a driver model which represents driving characteristics in a racing situation is required to develop a chassis control system. Since a race driver drives a vehicle as fast as possible on a given racing line without losing control, the proposed driver model is developed to ensure a lateral stability. In racing situation, one of the reasons which cause the lateral instabilities is an excessive corner-entry speed. The lateral instability in that moment is hard to handle with only a steering control. To guarantee the lateral stability of the vehicle while maximizing a cornering speed, a desired speed is determined to retain a tire slip-angle that maximizes lateral tire forces without front tire saturation. The proposed driver model has been validated with an expert driver's driving data, collected on the Korea International Circuit in Yeongam, Korea.

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I. INTRODUCTION

Numerous chassis control systems, such as Active Front Steer (AFS), Electronic Stability Control(ESC), Four Wheel Drive(4WD), Direct Yaw Moment Control(DYC) and Active Roll Control(ARC), etc., have been developed and commercialized to improve vehicle stability and driver's maneuverability. State-of-the-art research focuses on the integration of individual chassis control systems [1, 2, 3]. Recently developed integrated chassis control(ICC) system aims to enhance performance of high-speed driving as well as vehicle stability. In order to assess the performance of those vehicle control systems without cost and safety issues, a computer simulation has been widely used. A closed-loop simulation provides with means to analyze the interaction between a human driver and a vehicle with high-repeatability [4]. Thus, in the closed-loop simulation, modules which perform pre-defined path following and throttle/brake control function are essential parts. The modules are called lateral/longitudinal driver model respectively. This paper focuses on the driver model at limit handling.

The various types of lateral driver models for analysis of vehicle-driver systems at limit handling have been proposed in many researches. Johannes has developed the lateral driver model for improvements of handling under higher lateral accelerations regarding transient and the steady-state behavior of a vehicle [5]. Kritayakirana, Funke, et al. also have developed an autonomous steering controller, which neglects the effects of the rear tire saturation using center of

percussion(COP) [6, 7, 8]. From the view of lap-time minimization, the pre-defined optimal path trajectory, followed by the lateral driver model has been studied [8, 9, 10, 11]. Funke, et al. assumed the optimal path which is racing line as a clothoid-circular combined path because the COP base lateral driver model needs an ideal curvature profile [6, 7, 8] whereas, Braghin addressed that the optimal racing line is a compromise between the shortest path and the minimum curvature path [9]. Some research considered it as an optimization problem with vehicle dynamics [12, 13, 14]. These numerical approaches enable to obtain the optimal trajectory, velocity and acceleration profile of the vehicle simultaneously. However, this approach also has issues involved with computational load and existence of a solution. Thus, it can be hard to implementation in real-time simulation with full-course racing track.

An alternative approach to plan the optimal velocity profile for the longitudinal driver model at limit driving has been researched [11, 15-16]. Subbanna planned the optimal velocity profile with multi-preview point strategy in order to enhance a single-preview point strategy [11]. Velenis has developed the optimal velocity profile from a curvature profile by integration of ordinary differential equation(ODE) with simple constraints, adhesion limitation [15-16]. In this paper, in order to develop the driver model with real-time velocity planning, similar approach is utilized. However, with only these approaches it can be hard to secure the vehicle lateral stability at limit cornering or lap-time minimization because the velocity

profile from the approaches depend on assumed limitation of lateral/longitudinal acceleration. Thus, this paper focuses on the necessity of an additional feedback speed controller. In realistic driving, race driver who aims to minimize the lap-time decelerates or accelerates a vehicle to achieve maximum cornering speed without losing control. The speed control feedback part is one of the essential components in the proposed driver model for analysis of vehicle-driver systems at limit handling

In many case, tire saturation results in vehicle instability. Rear tire saturation may causes spinning motion of the vehicle (oversteer) and front tire saturation may causes plough motion of the vehicle (understeer) [17]. However most of commercialized vehicle are designed with understeer characteristic. Thus, the proposed driver model controls speed based on front tire slip-angle to prevent front tire lateral saturation.

Chassis control systems such as ABS, ESC, etc. to secure maneuverability and stability of the vehicle, control by individual brake actuators, whereas human driver can control only steering, throttle and brake with respect to the vehicle where an automatic transmission is applied. Thus, the final outputs of the proposed speed control driver model are throttle/brake.

The constitution of this paper are described as follows. The real-time velocity planning algorithm is derived first. For the realistic results, human driver's limit driving pattern is applied to the velocity planning algorithm. The driving test data is collected on the Korea International (KI) Circuit. Second, the slip-angle based speed control algorithm is introduced. The control law based on sliding mode control, is derived. And the un-measurable state, lateral velocity estimation algorithm is also derived. Finally, the proposed speed control driver model is validated via comparison with human driver's limit driving data.

II. THE PROPOSED SPEED CONTROL DRIVER MODEL

The racing drivers, who have acquired the optimal racing line for a given circuit by repeated training, determine throttle/brake action timing from a curvature of preview-path which comes into driver's view. When corner-entry speed exceeds a tire adhesion limit, the drivers lose maneuverability. On the contrary to this, if the corner-entry speed falls far short of a critical cornering speed, the drivers may get a poor lap-time. Thus, the drivers exert correction control in limit cornering. The proposed driver model has been developed to control a vehicle similarly to the human drivers as Fig.1.

The proposed driver model consists of an upper-level controller and a lower-level controller. The upper-level controller is composed of a feedforward part and a feedback part. The feedforward part performs real-time velocity profile planning. The velocity profile is generated every time-step from curvature profile of preview selected path. However, because of excessive corner-entry speed, it is hard to guarantee a stability of a vehicle in limit cornering using the planned velocity profile. The instability of the vehicle may results in loss of

maneuverability. Therefore, an additional acceleration/deceleration controller is needed to maximize driving speed without losing control. The feedback part performs that function.

The lower-level controller determines throttle/brake inputs to track the desired velocity, calculated from the upper-level output. The self-adaptive throttle/brake controller is utilized as the lower-level controller [18].

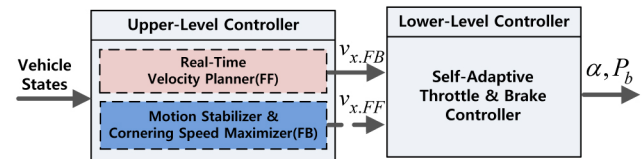


Figure1. The Proposed Speed Control Driver Model Diagram

A. Real-Time Velocity Profile Planning

There have been several researches related to optimal behavior of a vehicle at limit handling [12, 13, 14]. The results of the optimization problem with high order dynamical model are quite precise under certain assumptions. However, these numerical approaches have trouble in entire course simulation or a real-time implementation due to computational load or existence of solution. Thus, alternative solutions with low computational cost have been utilized.[11,15-16]

A process of the velocity profile planning consists of following 3 steps. All steps are repeated at each time-step by means of receding horizon implementation.

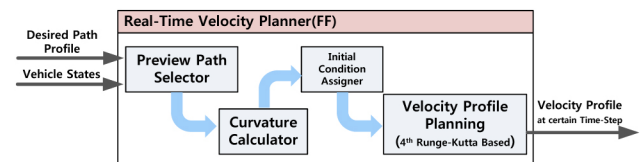


Figure2. Real-Time Velocity Profile Planner Diagram

1). Preview Desired Path Selection

A human driver recognizes a desired path by visionary information. The driver's visual angle and visual range depend on vehicle states, driving load, driving environments, etc. It is assumed that visual angle/range exponentially decrease with a velocity/yawrate of ego vehicle respectively. The functions between visual angle/range and a velocity/yawrate, are assumed as (1),(2). A desired path segment, in a fan-shaped preview area which is determined by visual angle/range, is defined as a preview-path.

$$\rho(\dot{\psi}) = (\rho_{\max} - \rho_{\min}) \cdot e^{-\gamma_{\rho} |\dot{\psi}|} + \rho_{\min} \quad (1)$$

$$\theta(v_x) = (\theta_{\max} - \theta_{\min}) \cdot e^{-\gamma_{\theta} |v_x|} + \theta_{\min} \quad (2)$$

where ρ , θ are assumed driver's visual range and angle respectively. γ_{ρ} , γ_{θ} are decay rate with respect to dependent states.

2). Curvature Calculation of the Selected Desired Path

A velocity profile is restricted to a curvature profile of the given preview path, because an excessive speed, beyond the critical speed on the certain point, causes lateral instabilities of the vehicle. The selected preview path is approximated as 1st/2nd/3rd polynomial curves (3). In order to calculate a curvature more accurately, the minimum residual curve is selected among the polynomial curves. The polynomial approximation is conducted with respect to the rotational transformed preview path, so as to avoid geometrical problems because, in some cases, an abstract shape of the given preview path may hard to be approximated by a polynomial curve, in ego vehicle coordinates. The angle of the rotational transformation is a slope of a line which joins the closest and the farthest points in the preview path with negative sign. The curvature of the certain point in the preview path is calculated as: (4).

$$\forall n \in \mathbf{P}$$

$$f(x) = p_n \cdot x^n + p_{n-1} \cdot x^{n-1} + \dots + p_0 \quad (3)$$

$$K(x) = \max \left[\frac{|f''(x)|}{\left[1 + \{f'(x)\}^2\right]^{3/2}}, 0.001 \right] \quad (4)$$

where $\mathbf{P} = \{n | n = 1, 2, 3\}$ is a set of polynomial order.

3). Velocity Profile Planning

An approach, which obtains velocity profile by integration of ODEs, is proposed by [15-16]. This approach is a competitive method in comparison with other optimization approach, in that computational cost is minimized and a solution will exist. Although a simple mass point model is considered in the definition of state variables, a reasonable velocity profile, compatible with a high-order vehicle model, can be obtained under proper several constraints.

States variable $\mathbf{Z} = [z_1 \ z_2 \ z_3]^T$ is defined as (5). State variables, z_1, z_2 are defined as a travelled distance and a velocity respectively. State variable, z_3 is added to apply delay effect to longitudinal acceleration at initial condition. Without 3rd state variable z_3 , there can be discontinuous rate of velocity at local maximum of a curvature profile which is initial point of integration. Thus, τ is time constant of delay (0.5s in this study)

$$\begin{aligned} z_1 &\triangleq s, & z_2 &\triangleq v_x = \dot{z}_1 \\ \dot{z}_1 &= z_2 \end{aligned} \quad (5)$$

$$\begin{aligned} \dot{z}_2 &= (1 - z_3) \cdot a_{x,plan} = (1 - z_3) \cdot \min[a_{x,max}, a_{x,limit}(v_x)] \\ \dot{z}_3 &= -\frac{1}{\tau} z_3 \end{aligned} \quad (6)$$

Where s represents a travelled distance, and then $a_{x,plan}$ denotes longitudinal acceleration used to plan velocity profile. $a_{x,limit}(v_x)$ is limitation of longitudinal acceleration which varies with longitudinal velocity. $a_{x,max}$ denotes maximum longitudinal acceleration on the friction circle which is assumed with certain frictional coefficients. Maximum longitudinal acceleration is determined by (10)

A travelled distance at local maximum point of curvature profile and critical velocity are initial conditions of integrations as (7).

$$\forall i \in \mathbf{S}$$

$$\mathbf{Z}_i = [z_{1,i} \ z_{2,i} \ z_{3,i}]^T = [s_{crit} \ v_{x,crit} \ 1]^T \quad (7)$$

where $\mathbf{S} = \{i | i = 2, 3, 4\}$. And i denotes number of points where ODEs integration starts. The subscript, *crit* describes states on local maximum curvature points.

Critical velocity which satisfies the lateral acceleration limitation, at each local maximum point of a curvature profile, can be determined as follow:

$$v_{x,crit} = \sqrt{a_{y,limit} \cdot R_{localmin}} \quad (8)$$

where $a_{y,limit}$ is limitation of lateral acceleration. And it is assumed to constant, 7.36 m/s²

Longitudinal/lateral acceleration of the ego vehicle is generated from resultant force of each tire. Longitudinal/lateral tire force for single tire are coupled each other at limit handling. The maximum force that the tire can generate under given driving conditions can be represented as a diagram called "Friction Circle". Friction circle for single tire is conceptually similar to the one for the whole vehicle [17]. When maximum resultant force, boundary of the friction circle is assumed to μmg , a relation between longitudinal/lateral acceleration at limit handling can be represented as (9)

$$\begin{aligned} (F_{resultant})^2 &= F_x^2 + F_y^2 \leq \mu mg \\ a_x^2 + a_y^2 &\leq (\mu g)^2 \end{aligned} \quad (9)$$

Where $F_{resultant}$ denotes resultant net tire force which exerts to ego vehicle. μ means frictional coefficient (1 in this study) and $a_{x,max}$ in (6) at limit handling can be represented by (10). The assumption that lateral acceleration on certain point cannot exceed maximum lateral acceleration is implied in (10)

$$a_{x,max} = \sqrt{(\mu g)^2 - \min \left[\frac{z_2^2}{R}, \dot{\psi}_{limit}^2 \cdot R \right]} \quad (10)$$

where $\dot{\psi}_{limit} \cdot R$ can be considered maximum lateral acceleration. The lateral acceleration at certain point of a given path is assumed as:

$$a_y = \dot{v}_y + \dot{\psi} \cdot v_x \approx \dot{\psi} \cdot v_x \approx \left(\frac{v_x}{R} \right) \cdot v_x \quad (11)$$

where $a_{x,limit}(v_x)$ in (6) and $\dot{\psi}_{limit}$ are limitation of longitudinal acceleration/yawrate respectively. Both quantities are function of longitudinal velocity as seen in Fig.3 and Fig.4. These constraints have great effect on a velocity profile. Thus, in order to obtain a velocity profile with realistic acceleration/deceleration pattern, the functions are obtained from human driver's limit driving data collected on the Korea International (KI) Circuit in Yeongam, Korea.

Two test driving results with same driver are illustrated in Fig.3 and Fig.4. It is found that the upper-boundary line in Fig.3 is quite similar to each test driving result because the upper-boundary line in Fig.3 depends on powertrain performance of the target vehicle. The equation of the upper-boundary line is represented as (12)

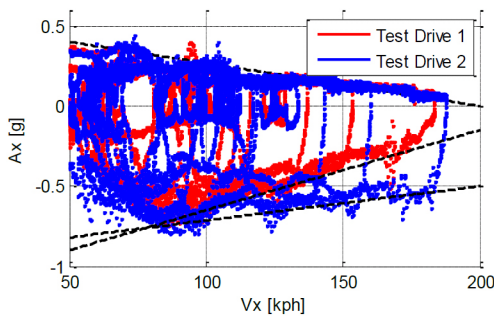


Figure3. Longitudinal Acceleration vs Longitudinal Velocity

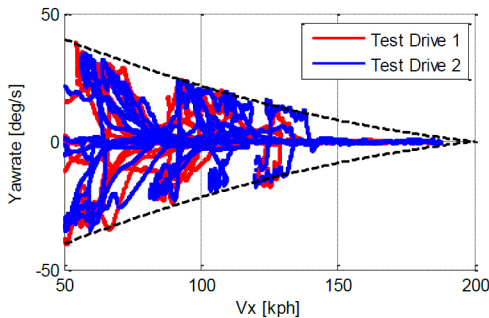


Figure4. Yawrate vs Longitudinal Velocity

On the contrary, the lower-boundary line in Fig.3 is different to each other. The lower-boundary represents driver's deceleration pattern. It can vary under driving conditions and vehicle states. In order to operate the proposed driver model more severely, the lower-boundary line with lower slope is used as a longitudinal deceleration limit function is defined (12)

$$\begin{cases} a_{x,limit} = -0.0943 \cdot v_x + 5.2287 & \text{Acceleration} \\ a_{x,limit} = +0.0752 \cdot v_x - 9.0906 & \text{Deceleration} \end{cases} \quad (12)$$

Limitation of yaw rate is considered as a boundary of envelop in Fig.4. Maximum yaw rate from human driver's limit driving decreases with longitudinal velocity because maximum lateral acceleration is bounded to 9.81 m/s² regardless of longitudinal velocity when the relation (11) is considered. A function, the boundary of envelop in Fig.4 is represented as (13):

$$\dot{\psi}_{limit} = 2.223 \times 10^{-4} \cdot v_x^2 - 0.0322 \cdot v_x + 1.1027 \quad (13)$$

Fig.5 illustrates process of ODEs integration. When the selected preview path has local maximum points in the curvature profile, forward/backward integrations are implemented from local maximum points and start/end points. The 4th order Runge-Kutta method is utilized as an integration method. As seen in Fig.5, velocity profiles intersect at midway in each section. Final velocity profile is obtained by trimming the larger one at intersected profile. In case of Fig.5, three of forward/backward integrations are implemented and blue solid line is final velocity profile at certain time. This series of process is repeated every time-step.

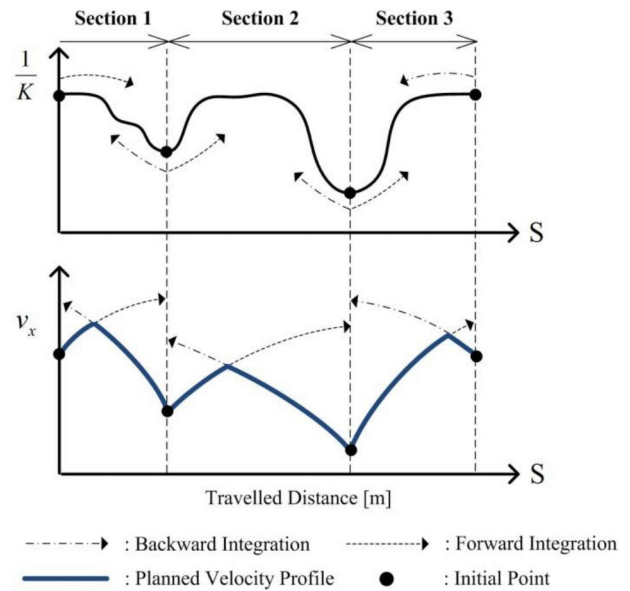


Figure5. Velocity Profile Planning Diagram

Fig.6 describes velocity profile planning of the proposed driver model. The golden solid line in trajectory figure is the pre-defined racing line. And the red solid line in that figure represents the selected preview path. The fan-shaped area denotes a driver's visual range. From the curvature profile of the red solid line, velocity profile is planned as seen in the bottom figure in Fig.6. In the bottom figure, the dashed blue line is the planned velocity profile at current time-step. The solid red line is the velocity profile, realized by the driver model so far.