Adaptive Emergency Braking Control Using a Dynamic Tire/Road Friction Model*

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Abstract

A controller for emergency braking of vehicles in Automated Highway Systems (AHS) is designed. The scheme is based on the estimation of both the LuGre tire/road friction dynamic model and the braking system gain. The controller estimates the tire/road relative velocity which achieves maximum braking force based on a quasi-static solution of the LuGre friction model and sets the master cylinder pressure to track that relative velocity. This control system is designed to work in conjunction with antilock-braking-systems (ABS) providing two advantages: less chattering during braking and a source of a priori information regarding safe spacing.

1 Introduction

The concept of Automated Highway Systems (AHS) is expected to significantly increase current highways safety and capacity. A safe fault handling in AHS can involve emergency braking maneuvers [9]. Safety during this emergency braking is closely related to the braking capacity of vehicles that changes with the degradation in system performance due to adverse environmental conditions, gradual wear of AHS components and highway topology. From the perspective of emergency braking, the braking capacity is mainly determined by two factors: tire/road friction and available braking torque. These factors are not easy to determine precisely since their behavior is complex and the associated variables difficult to measure.

The goal of this paper is to design an on-line scheme for vehicles to estimate their own tire/road friction characteristics and the overall gain of the braking system. The wheel relative velocity that achieves maximum braking effort in a quasi-static LuGre friction model solution is estimated on line, assuming that the tire/road friction model dynamics is much faster than the vehicle braking dynamics. This information is used to design a controller that achieves near-maximum braking effort. Knowing the tire/road friction characteristics and the gain of the braking system allows vehicles to adjust their spacing for safety and to broadcast this information to the road-side infrastructure that can modify overall traffic conditions, if necessary.

Literature for tire/road friction estimation is abundant. Bakker *et al.* [2] and Burckhardt [3] describe two analytical models for tire/road behavior that are intensively used by researchers in the field. Kiencke [8] presents a procedure for real-time estimation of μ. The hypothesis in [7] postulates that by combining the slip and the initial slope of the μ versus λ curve it is possible to distinguish between different road surfaces. In [5] and [12], braking and traction control are carried out without knowledge of the optimal point of maximum friction force by use of sliding mode controllers. Alvarez *et al.* [1] gives an on-line estimation of the road/tire friction coefficient and an emergency braking controller based on that estimation with underestimation of the peak slip and peak friction values. Most of above work are based on the pseudo-static model for the road/tire friction estimations. A dynamic model for the road/tire characteristics and a traction controller are presented by [4]. In this paper we use such dynamical model to derive a pseudo-static formula with proper boundary conditions of contacting patches between tire and road and then use it for emergency braking control.

The paper is divided in five sections. In section 2, the LuGre tire/road friction model and a vehicle dynamic model are developed. In section 3 a stabilizing controller for emergency braking is designed, assuming that all state variables are measurable. Simulation work is illustrated in section 4 and section 5 contains concluding remarks and directions for future work.

2 Tire/road Friction and Vehicle Models

In this paper we utilize the so called LuGre dynamic friction model [4] given as

$$\begin{cases} \dot{z} = v_r - \theta \frac{\sigma_0 |v_r|}{h(v_r)} z \\ F = (\sigma_0 z + \sigma_1 \dot{z} + \sigma_2 v_r) F_n \end{cases} \tag{1}$$

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where θ is an unknown parameter of the tire/road conditions, $v_r = r\omega - v$ is the relative velocity, $h(v_r) = \mu_c + (\mu_s - \mu_c)e^{-|\frac{v_r}{v_s}|^{1/2}}$, σ_0 is the rubber longitudinal stiffness, σ_1 is the rubber longitudinal damping, σ_2 is the viscous relative damping, μ_s is the normalized static friction coefficient, μ_c is the normalized Coulomb friction, and v_s is the Stribeck relative velocity.

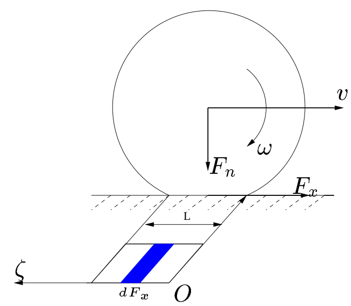

Figure 1: A schematic of one vehicle wheel with distributed force

The dynamic model in Eq. (1) is a lumped parameter model with only one internal state z. As discussed in [4] it is also possible to obtain a distributed friction dynamic model by assuming the existence of a contact patch between the tire and the road (see Fig. 1.) Following the same procedures as those given in [4] for the traction case, we can develop a quasi-static curve magic curve for the braking case. Construct a distributed model as

$$\begin{cases} \frac{d\delta z}{dt}(\zeta, t) = v_r - \frac{\sigma_0|v_r|}{h(v_r)}\delta z \\ F = \int_0^L (\sigma_0 \delta z + \sigma_1 \delta \dot{z} + \sigma_2 v_r)\delta F_n d\zeta \end{cases} \tag{2}$$

with boundary conditions as

$$\delta z(0, t) = \delta z(L, t) = 0, \qquad \forall t \geq 0, \tag{3}$$

where $\delta F_n = F_n/L$ and L is the length of the patch which is assumed to be constant. Assume that v and ω are constant and $\frac{\partial \delta z}{\partial t}(\zeta, t) = 0$ within an small enough interval of time, namely a quasi-static condition, then we have

$$\begin{cases} \frac{d\delta z}{d\zeta}(\zeta) = \frac{v_r}{r\omega} - \frac{\sigma_0|v_r/r\omega|}{h(v_r)}\delta z \qquad \zeta \in (0,\ L) \\ \delta(0) = \delta(L) = 0, \end{cases} \tag{4}$$

Defining $\eta = v_r/r\omega$ and solving the above equation for $\delta z(\zeta)$ with initial condition $\delta z(\zeta) = \zeta = 0$, we obtain

$$\delta z(\zeta) = \begin{cases} \frac{h(v_r)}{\sigma_0}(e^{-\frac{\sigma_0|\eta|}{h(v_r)}\zeta} - 1), & 0 \leq \zeta \leq \frac{L}{2} \\ \frac{h(v_r)}{\sigma_0}\left[e^{-\frac{\sigma_0|\eta|}{h(v_r)}(L-\zeta)} - 1\right], & \frac{L}{2} < \zeta \leq L \end{cases}$$

Calculating the friction force term by term using Eq. (2) we obtain

$$\begin{aligned} \int_0^L \delta z(\zeta) d\zeta &= -\frac{h(v_r)}{\sigma_0} L \left[1 + \frac{2h(v_r)}{\sigma_0 L|\eta|}(e^{-\frac{\sigma_0|\eta|L}{2h(v_r)}} - 1)\right] \\ \int_0^L \delta \dot{z}(\zeta) d\zeta &= -\frac{2vh(v_r)}{\sigma_0}\left(1 - e^{-\frac{\sigma_0|\eta|L}{2h(v_r)}}\right) \end{aligned}$$

then for constant velocity v we have

$$\begin{aligned} F(\eta, v) &= -F_n h(v_r)\left[1 + 2\gamma\frac{h(v_r)}{\sigma_0 L|\eta|}(e^{-\frac{\sigma_0 L|\eta|}{2h(v_r)}} - 1)\right] \\ &\quad -F_n \sigma_2 v_r \end{aligned} \tag{5}$$

where

$$\eta = \frac{v_r}{r\omega} = -\frac{\lambda}{1-\lambda}, \quad \gamma = 1 - \frac{\sigma_1|\eta|}{r\omega h(v_r)},$$

and $\lambda = 1 - r\omega/v$ is the longitudinal slip ratio. This formula is similar to the traction case derived in [4] that considers the angular velocity ω to be constant. $\lambda \in [0,\ 1]$ is used in [4] while $\eta \in (-\infty,\ 0]$ here.

Remark 1 In the pseudo-static curve in Eq. (5) derived here we assume constant velocity. If velocity changes, the curve changes as well. However, by taking a look at the dynamic equation for the internal state z we find that it changes much faster than the vehicle dynamics. Therefore, we can make the assumption that, for each time step, this formula can be used to calculate the approximated maximum peak value for the braking force produced by the tire/road friction.

Remark 2 The static curve in Eq. (5) is a function of longitudinal slip ratio λ. In the braking case it has been defined as $\lambda := \frac{v - r\omega}{v}$. When the vehicle's velocity becomes very small, the relationship does not have real physical sense. However, since we are interested in finding a controller strategy for braking the car at fairly high speeds, we can use this approach for large velocities and establish a lower bound v_{min} to obtain good braking; i.e. putting an upper bound $\bar{\lambda}$ for the maximum slip ratio $\lambda_{max} \in [0,\ \bar{\lambda}]$ in Eq. (5). In this paper we put $\bar{\lambda} = 0.4$, which corresponds to $v = 2.4m/s$ given the dynamic model parameters in Table 1.

Remark 3 The distributed model given by Eq. (2) is consistent with the lumped model given by Eq. (1) in the following sense: assuming that the patch region does not change with time and defining

$$\bar{z}(t) = \frac{1}{L}\int_0^L \delta z(\zeta, t) d\zeta \tag{6}$$

we have

$$\dot{\bar{z}}(t) = \frac{d}{dt}\left(\frac{1}{L}\int_0^L \delta z(\zeta, t) d\zeta\right) = \frac{1}{L}\int_0^L \frac{\partial}{\partial t}\left(\delta z(\zeta, t)\right) d\zeta \tag{7}$$

we know that

$$\begin{aligned} \delta \dot{z} &= \frac{d}{dt}(\delta z(\zeta, t)) = \frac{\partial}{\partial t}(\delta z(\zeta, t)) + \frac{\partial}{\partial \zeta}(\delta z(\zeta, t))\dot{\zeta} \\ &= \frac{\partial}{\partial t}(\delta z(\zeta, t)) + v\frac{\partial}{\partial \zeta}(\delta z(\zeta, t)) = v_r - \frac{\sigma_0|v_r|}{h(v_r)}\delta z(\zeta, t) \end{aligned}$$

then Eq. (7) becomes

$$\begin{aligned}\dot{\bar{z}}(t) &= \frac{1}{L}\int_0^L \left(\delta\dot{z} - v\frac{\partial}{\partial\zeta}(\delta z(\zeta,t))\right) d\zeta \\ &= \frac{1}{L}\int_0^L \left(v_r - \frac{\sigma_0|v_r|}{h(v_r)}\delta z(\zeta,t)\right) d\zeta \\ &\quad -\frac{1}{L}\int_0^L v\frac{\partial}{\partial\zeta}(\delta z(\zeta,t))d\zeta \\ &= \frac{1}{L}\int_0^L v_r d\zeta - \frac{\sigma_0|v_r|}{h(v_r)}\frac{1}{L}\int_0^L \delta z(\zeta,t)d\zeta \\ &\quad -\frac{v}{L}\left[\delta z(L,t) - \delta z(0,t)\right] \\ &= v_r - \frac{\sigma_0|v_r|}{h(v_r)}\bar{z} \qquad (8)\end{aligned}$$

in the last step we use Eq. (3).

Similarly, we can find the friction force given by Eq. (2) using the state $\bar{z}$ as

$$\begin{aligned}F &= \int_0^L (\sigma_0\delta z + \sigma_1\delta\dot{z} - \sigma_2 v_r)\delta F_n d\zeta \\ &= F_n\left(\sigma_0\bar{z} - \sigma_2 v_r + \frac{\sigma_1}{L}\int_0^L \delta\dot{z}d\zeta\right)\end{aligned}$$

note that

$$\begin{aligned}\frac{1}{L}\int_0^L \delta\dot{z}d\zeta &= \frac{1}{L}\int_0^L \left(\frac{\partial}{\partial t}(\delta z(\zeta,t)) + v\frac{\partial}{\partial\zeta}(\delta z(\zeta,t))\right) d\zeta \\ &= \frac{1}{L}\int_0^L \frac{\partial}{\partial t}(\delta z(\zeta,t))d\zeta \\ &\quad +\frac{v}{L}\left[\delta z(L,t) - \delta z(0,t)\right] \\ &= \frac{d}{dt}\left(\frac{1}{L}\int_0^L \delta z(\zeta,t)d\zeta\right) \\ &= \dot{\bar{z}}(\zeta,t)\end{aligned}$$

therefore

$$F = F_n(\sigma_0\bar{z} + \sigma_1\dot{\bar{z}} + \sigma_2 v_r) \qquad (9)$$

Remark 4 Comparing the lumped LuGre dynamic model in Eqs. (8) and (9) with the lumped model in Eq. (1), we found that the lumped model for the tire/road dynamics can be obtained by using the transformation in Eq. (6) and converting the PDE in Eq. (2) into the ODE in Eq. (8).

We compare the pseudo-static curve in Eq. (5) obtained from the dynamical model, against the magic formula for one of the tested tires in [11]. Fig. 2 shows the results using the formula in Eq. (5). From the figure we can see that the dynamical model can fit the tested data very well. The parameters for the dynamical model are listed in Table 1.

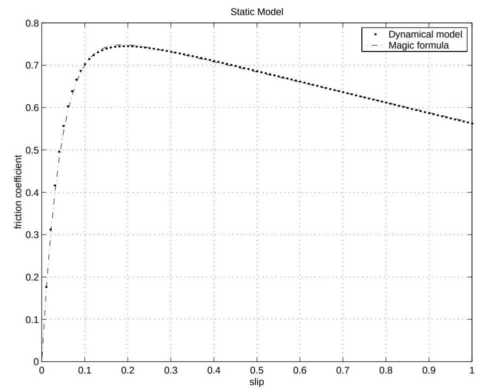

Figure 2: Comparison between the dynamical model and the magic formula for a tested tire in braking case with $v = 30mph$ and $\theta = 1$ in the model.

Table 1: Parameters used for the static curve by Eq.(5).

Parameters	Values	Units
σ_0	100	$1/m$
σ_1	0.7	s/m
σ_2	0.011	s/m
μ_s	0.5	–
μ_c	0.35	–
v_s	10.0	m/s
L	0.25	m

To model the longitudinal dynamics of the vehicle we consider the LuGre model [4] together with vehicle dynamics as:

$$\begin{cases}\dot{z} = v_r - \theta\frac{\sigma_0|v_r|}{h(v_r)}z \\ J\dot{\omega} = -rF_x - \sigma_\omega\omega - u_\tau \\ m\dot{v} = 4F_x - F_{av}\end{cases} \qquad (10)$$

where σ_ω is the coefficient of the viscous resistance moments, $F_{av} = C_a v^2$ is the aerodynamic force, u_τ is the traction/braking torque, and F_x is the traction/braking force given by the tire/road contacting. The braking force F_x is given by

$$F_x = F_n(\sigma_0 z + \sigma_1\dot{z} + \sigma_2 v_r) \qquad (11)$$

Define the state variables as

$$x_1 := z, \;\; x_2 := v, \;\; , x_3 := v_r = r\omega - v$$

and rewrite the system dynamics in Eq. (10) as

$$\dot{x}_1 = \dot{z} = x_3 - \theta\frac{\sigma_0|x_3|}{h(x_3)}x_1 = x_3 - \theta f(x_3)x_1 \qquad (12)$$

where $f(x_3) = \frac{\sigma_0|x_3|}{h(x_3)}$, $h(x_3) = \mu_c + (\mu_s - \mu_c)e^{-|\frac{x_3}{v_s}|^{1/2}}$,

$$\dot{x}_2 = g\left[\sigma_0 x_1 + \sigma_1(x_3 - \theta f(x_3)x_1) + \sigma_2 x_3\right] - C_{av}x_2^2 \qquad (13)$$

g is the gravity constant and $C_{av} = C_a/m$. We assume all four wheels of the vehicle applying the same braking force. For simplicity, we assume no road slope at this moment, and that weight of the vehicle is distributed evenly on the four wheels. We can relax these assumptions later. For the state variable x_3 we have

$$\begin{aligned} \dot{x}_3 &= -(g + \frac{F_n r^2}{J})\left[\sigma_0 x_1 + \sigma_1(x_3 - \theta f(x_3)x_1)\right] \\ &\quad -g\sigma_2 x_3 + C_{av}x_2^2 - \frac{r\sigma_\omega}{J}(x_2 + x_3) \\ &\quad -\frac{r}{J}K_b P_b \end{aligned} \tag{14}$$

In the above equation we use the formula $u_\tau = K_b P_b$, where K_b is the brake coefficient gain and P_b the brake pressure which is the controlled variable.

When emergency braking control is to be achieved, the estimation of the braking system gain K_b is important. This gain is uncertain and changes with temperature, vehicle velocity, physical wear and other parameters. If K_b is properly estimated, braking control can be realized using the pressure in the master cylinder as the control input. For this purpose consider the relationship proposed in [6, 10] between the brake pressure P_b and wheel braking torque u_τ

$$u_\tau = K_b P_b \tag{15}$$

Adaptation of K_b is important because its value can drop down to 60% of its normal value in some circumstances. In the following section the adaptation of K_b is realized in the context of the design of the emergency braking controller.

3 Controller Design

Our control goal is to achieve $x_2 \to 0$ and $x_3 \to x_{3d}$, where x_{3d} is the unknown maximum relative velocity. We now try to approximate x_{3d} by using the results of the previous section for the dynamic model. We define

$$x_{3d}(\hat{\theta}) := -\lambda_{max}(\hat{\theta})x_2 \tag{16}$$

$\lambda := -v_r/v$ is the longitudinal slip ratio for the braking case and

$$\lambda_{max}(\hat{\theta}) = \begin{cases} \arg\max_{\lambda\in[0,\bar{\lambda}]} F_x & \text{when } v \geq v_{min} \\ \bar{\lambda} & \text{when } 0 < v < v_{min} \end{cases} \tag{17}$$

Note that from section 2 we can produce a static curve for an infinitesimal time; i.e. constant velocity holds. Thus, we use the calculation from the static formula given by Eq. (5) as an approximation for λ_{max} when v is large enough, because the internal friction model dynamics is much faster than the vehicle dynamics. During each time step Δt, we assume that, for the vehicle braking control system, the internal state of the friction dynamic model converges to its steady-state. Thus, the maximum deceleration can be approximated well by Eq. (16). When v is small, we just bound v by v_{min}.

Arrange the system dynamics (12), (13) and (14) as

$$\begin{cases} \dot{x}_1 = x_3 - f_1(\mathbf{x})\theta \\ \dot{x}_2 = f_2(\mathbf{x}) - [g\sigma_1 f_1(\mathbf{x})]\,\theta \\ \dot{x}_3 = f_3(\mathbf{x}) - [\alpha\sigma_1 f_1(\mathbf{x})]\,\theta - \frac{r}{J}K_b P_b \end{cases} \tag{18}$$

where

$$\begin{aligned} \alpha &= -\left[g + \frac{F_n r^2}{J}\right], \qquad f_1(\mathbf{x}) = f(x_3)x_1 \\ f_2(\mathbf{x}) &= g\left[\sigma_0 x_1 + (\sigma_1 + \sigma_2)x_3\right] - C_{av}x_2^2 \\ f_3(\mathbf{x}) &= \alpha\sigma_0 x_1 + (\alpha\sigma_1 - g\sigma_2)x_3 + C_{av}x_2^2 \\ &\quad -\frac{\sigma_\omega}{J}(x_2 + x_3) \end{aligned}$$

Define

$$\tilde{s} := x_3 - x_{3d}(\hat{\theta}) = x_3 + \lambda_{max}(\hat{\theta})x_2$$

where $\lambda_{max}(\hat{\theta})$ is the peak value of the longitudinal slip λ under current conditions based on the estimated parameter $\hat{\theta}$ given by Eq. (17). Differentiate $\tilde{s}$ to obtain

$$\begin{aligned} \dot{\tilde{s}} &= \dot{x}_3 + \dot{x}_2\lambda_{max} + x_2\dot{\lambda}_{max} \\ &= -\frac{r}{J}K_b P_b - \left[\alpha\sigma_1 f_1(\mathbf{x}) + g\sigma_1 f_1(\mathbf{x})\lambda_{max}\right]\theta \\ &\quad + \left[f_3(\mathbf{x}) + x_2\dot{\lambda}_{max} + \lambda_{max}f_2(\mathbf{x})\right] \\ &= dK_b P_b + \beta_1(\mathbf{x})\theta + \beta_2(\mathbf{x}) \end{aligned} \tag{19}$$

where

$$\begin{aligned} d &= -\frac{r}{J},\ \beta_1(\mathbf{x}) = -\left[\alpha\sigma_1 f_1(\mathbf{x}) + g\sigma_1 f_1(\mathbf{x})\lambda_{max}\right] \\ \beta_2(\mathbf{x}) &= \left[f_3(\mathbf{x}) + x_2\dot{\lambda}_{max} + \lambda_{max}f_2(\mathbf{x})\right]. \end{aligned}$$

Let $M_b := \frac{1}{K_b}$ be the adaptation variable and define the error variables

$$\tilde{\theta} := \theta - \hat{\theta}, \qquad \tilde{M}_b := M_b - \hat{M}_b$$

Let the control input P_b as

$$P_b = \frac{\hat{M}_b}{d}\left[-\beta_1(\mathbf{x})\hat{\theta} - \beta_2(\mathbf{x}) - \eta\tilde{s}\right], \tag{20}$$

where $\eta > 0$ is a control gain. Then the dynamics for $\tilde{s}$ in Eq. (19) can be arranged as

$$\begin{aligned} \dot{\tilde{s}} &= K_b\hat{M}_b\left[-\beta_1(\mathbf{x})\hat{\theta} - \beta_2(\mathbf{x}) - \eta\tilde{s}\right] + \beta_1(\mathbf{x})\theta + \beta_2(\mathbf{x}) \\ &= \beta_1(\mathbf{x})\tilde{\theta} - \eta\tilde{s} + K_b\tilde{M}_b\left[\beta_1(\mathbf{x})\hat{\theta} + \beta_2(\mathbf{x}) + \eta\tilde{s}\right] \end{aligned} \tag{21}$$

Consider now the following Lyapunov candidate

$$V = \frac{1}{2}\tilde{s}^2 + \frac{1}{2\gamma}\tilde{\theta}^2 + \frac{1}{2\xi}K_b\tilde{M}_b^2$$

where $\gamma > 0$, $\xi > 0$ are gains. Then

$$\begin{aligned}
\dot{V} &= \tilde{s}\dot{\tilde{s}} + \frac{1}{\gamma}\tilde{\theta}\dot{\tilde{\theta}} + \frac{1}{\xi}K_b\tilde{M}_b\dot{\tilde{M}}_b \\
&= \tilde{s}\left[\beta_1\tilde{\theta} - \eta\tilde{s} + K_b\tilde{M}_b(\beta_1\hat{\theta} + \beta_2 + \eta\tilde{s})\right] \\
&\quad + \frac{1}{\gamma}\tilde{\theta}\dot{\tilde{\theta}} + \frac{1}{\xi}K_b\tilde{M}_b\dot{\tilde{M}}_b \\
&= \tilde{\theta}\left[\beta_1\tilde{s} + \frac{1}{\gamma}\dot{\tilde{\theta}}\right] + K_b\tilde{M}_b\left[\frac{\dot{\tilde{M}}_b}{\xi} + \tilde{s}(\beta_1\hat{\theta} + \beta_2 + \eta\tilde{s})\right] \\
&\quad -\eta\tilde{s}^2
\end{aligned}$$

letting

$$\begin{aligned}
\dot{\hat{\theta}} &= \gamma\beta_1(\mathbf{x})\tilde{s}, \\
\dot{\hat{M}}_b &= \xi\tilde{s}\left[\beta_1(\mathbf{x})\hat{\theta} + \beta_2(\mathbf{x}) + \eta\tilde{s}\right],
\end{aligned} \tag{22}$$

we obtain

$$\dot{V} = -\eta\tilde{s}^2 \le 0,$$

By Lyapunov's theorem, ($\tilde{s} = 0$, $\tilde{\theta} = 0$, $\tilde{M}_b = 0$) is a stable equilibrium point and $\tilde{s}$, $\tilde{\theta}$ and $\tilde{M}_b$ are bounded. Notice that

$$\ddot{V} = -2\eta\tilde{s}\dot{\tilde{s}}$$

is bounded. Thus, by Barbalat's Lemma, $\tilde{s} \to 0$ as $t \to \infty$, and

$$x_3 \to -\lambda_{max}(\hat{\theta})x_2.$$

Remark 5 In our controller design we assume that all state variables are measurable. However, in reality, we can only measure the angular velocity; i.e. ω. Therefore, we need to implement a state observer. The synthesis of combined controller and observer design can be found in [13].

Remark 6 Even if we can calculate the dynamic surface by Eq. (5), we still need to compute its time derivative. We approach this by numerical differentiation which can be easily implemented. A low-pass filter is used to smooth the control input due to numerical noise.

4 Simulation Results

In the simulation we take the parameters from the LeSabre cars used in the California PATH program. These parameters are: $M = 1701.0\ Kg$, $Ca = 0.3693\ N \cdot s^2/m^2$, $J = 2.603\ Kg \cdot m^2$, $R = 0.323m$ and take the road characteristic parameter to be $\theta = 1$, the brake coefficient gain $K_b = 0.9$, therefore $M = 1/K_b = 1.11$.

We simulate a vehicle starting an emergency braking maneuver at the initial velocity $v = 30m/s$, by applying the designed controller without an observer. Fig. 3 shows the vehicle's velocity change as well as the internal state z and the relative velocity $v_r = r\omega - v$.

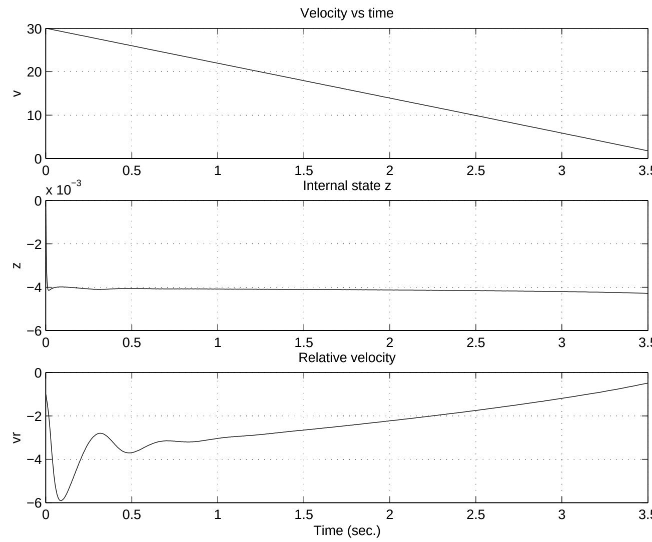

Figure 3: Vehicle velocity v (m/s), internal state z and relative velocity v_r (m/s).

The controlled brake pressure and the sliding surface errors are illustrated in Fig. 4. The adaptations of parameters θ and K_b are plotted in Fig. 5. From the simulation results we find that the vehicle stopped quickly with an almost constant deceleration (around $8m/s^2$) and both of the parameters K_b and θ converge to the true values respectively 0.9 and 1.0. Using this control, we can achieve braking around its peak value at each transient time, and this can be seen from the Fig. 6, which shows the friction coefficient and slip during the emergency braking. Note that the slip ratio converges to the value estimated by the dynamic model.

5 Conclusion

In this paper we discussed emergency braking control under unknown tire/road conditions and brake conditions, based on the dynamical friction model. We explored control design for brake pressure by assuming that vehicle velocity and the internal state are measurable. We used the static maximum slip as an approximation for the maximum deceleration when vehicle has fairly high longitudinal speed. The simulation results show that the vehicle can be stopped as quickly as possible by application of this controller. Compared with previous static approaches [1] and the experimental tire data, we found

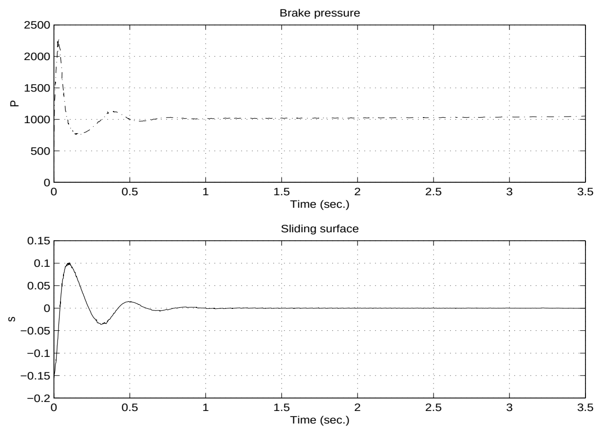

Figure 4: Brake pressure P (KPa) and sliding surface $\tilde{s}$.

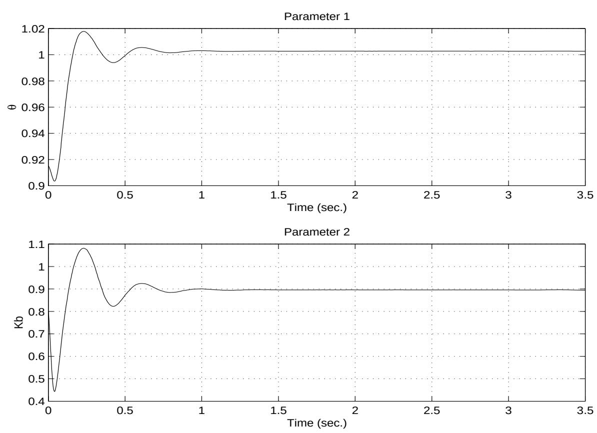

Figure 5: Friction characteristic parameter θ and brake coefficient K_b (Nm/KPa).

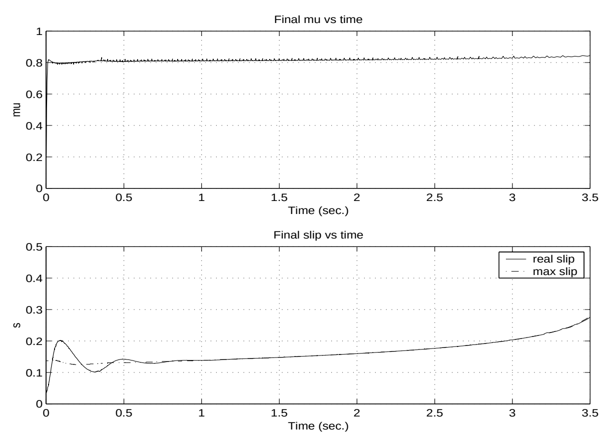

Figure 6: Friction coefficient μ and the slip λ during the emergency braking.

that the control design based on the dynamical model can achieve a better approximate maximum deceleration in a faster and more stable manner. Experimental works to test both static and dynamical model based controllers will be carried out at California PATH program and UC Berkeley.

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