ADVANCED NONLINEAR BRAKE SYSTEM CONTROL FOR VEHICLE PLATOONING

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Abstract

Successful longitudinal control of a vehicle in an Intelligent Vehicle and Highway System (IVHS) environment is highly dependent on the adequate control of the vehicle's subsystems. Once it had been demonstrated that it is feasible to control an automotive powertrain to maintain accurate longitudinal tracking, the next obvious step was to develop a brake system controller. This paper presents a second generation brake controller that evolved from the successes and drawbacks of the original one. Due to the nonlinearities present in the system, a nonlinear control method is proposed. The method suggested in this study is a modification of the technique known as sliding mode control. It was chosen due to its robustness to modeling errors and disturbance rejection capabilities. Simulation results are presented to illustrate the capability of a vehicle using this controller to follow a desired speed trajectory while maintaining constant spacing between vehicles.

1 Introduction

The concept of Advanced Vehicle Systems (AHS) as part of an Intelligent Vehicle and Highway Systems (IVHS) environment envisions platoons (convoys) of vehicles traveling on the highway at short spacing from each other. The actual number of vehicles in a platoon, the platoon speed and the spacing between vehicles will be dictated by a supervisory layer control. Maintaining the required speed and spacing will be controlled at the individual vehicle level. Members of the Vehicle Dynamics Laboratory (VDL) at the University of California at Berkeley have demonstrated the feasibility of controlling the powertrain for this purpose.

The next obvious step was the development of a brake controller. Since early attempts to control the brake system for the purpose of obstacle avoidance did not produce models adequate for closed-loop control, the results of such research proved to be of little value in developing a brake controller for platooning. More recent efforts in the area of automatic brake control have concentrated on consumer oriented products such as anti-lock brake system (ABS) and traction control system (TCS). Since ABS is concerned with preventing wheel lock-up by releasing brake pressure, the control algorithms developed in this area are not compatible with the task of brake actuation for longitudinal control. While TCS is able to actuate the brakes, the main requirement for such a system is to maintain traction in adverse conditions and therefore passenger comfort and accurate speed tracking are not important concerns in these cases. As such, these control algorithms cannot be easily adapted to an AHS environment. (Bowman and Law, 1993)

Finally, recent attempts to control the brake system in vehicle following situations have concentrated on applications to Autonomous Intelligent Cruise Control (AICC). However, the spacing requirements for AICC are far less stringent than the ones for platooning, making the hardware and control algorithms incompatible. (Martin, 1993)

This particular study emphasizes the development of a second generation automatic brake controller designed for vehicle platooning. Valuable lessons were learned from the previous system's successes and drawbacks and used to design the current hardware and control algorithm.

The brake system can be represented as a series of nonlinear elements. Due to these nonlinearities, linear control methods have failed to meet the demands placed on the system. The control algorithm must also ensure robust performance over a wide operating range in the presence of disturbances. Therefore a sliding mode control algorithm was implemented for this application.

Section 2 describes the longitudinal vehicle model used to design and simulate the controller. The dynamics modeled include a simplified powertrain and a brake system complex enough to capture the important characteristics but simple enough to facilitate the controller design and reduce real-time computation load. This section also includes a discussion on the development of the new hardware.

The control algorithm is developed in Section 3. In order to facilitate implementation, a multiple surface sliding controller was used. Three variations of this controller based on sensor availability are analyzed.

The control algorithm provided good tracking even under the presence of modeling errors and disturbances. The simulation results demonstrating the performance of these controllers are presented in Section 4.

2 Modeling

For simulation and control algorithm design a simplified powertrain model and a brake model developed specifically for the task of brake control in vehicle following were used.

Powertrain

A three state model was developed by Hedrick, McMahon and Swaroop (1993). For this model the following assumptions are made:

1. time delays associated with power generation in the engine are negligible
2. the torque converter is locked
3. no torsion of the drive axle
4. no slip at the wheels

Figure 1 shows a free body diagram of this model.

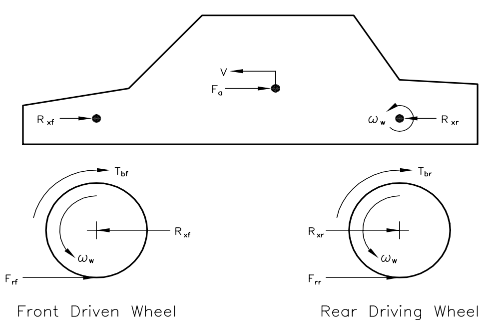


Figure 1. Vehicle Free Body Diagram

The two state equations for the engine are:

$$\dot{m}_a = c_1 TC(\alpha) - c_2 \omega_e m_a \qquad (1)$$

$$\dot{\omega}_e = \frac{1}{J_e}\left(T_i - T_f - T_r - T_d - T_b\right) \qquad (2)$$

where

m_a - mass of air in the intake manifold
ω_e - engine speed
$TC(\alpha)$ - throttle characteristic
J_e - effective vehicle inertia
$T_i = c_3 m_a$ - indicated torque
$T_f = c_4 \omega_e$ - friction torque
$T_r = h \cdot F_r$ - rolling resistance
$T_d = c_5 \omega_e^2$ - aerodynamic drag
T_b - total brake torque
h - effective tire radius

The third and final state is the total brake torque. There are several articles in the literature suggesting modeling the brake system as a pure delay followed by a first order lag. However such a model does not capture all the necessary dynamics and is therefore inadequate for control. The new model is discussed briefly in the following sections.

Brake system hardware

The first attempt to control an automatic brake for platooning is presented in detail in Maciuca, Gerdes and Hedrick (1994). Although we achieved very good speed tracking in the experimental phase, the passenger comfort, although a subjective characteristic, was deemed suboptimal. The vacuum booster was identified as being a source of problems due to its low actuation bandwidth caused by the air flow dynamics. One solution would have been to eliminate the hydraulic actuator and control the pressure in the vacuum booster through two air valves as it was suggested by Mitsubishi (Kishi, et.al., 1993). The shortcoming of this solution is a major design change of the vacuum booster. Furthermore, it does not solve the problem of low actuation bandwidth.

The solution chosen was to bypass the vacuum booster in automatic mode while still allowing a human driver to take advantage of the vacuum booster in manual driving mode. Therefore the actuator was placed between the vacuum booster and the master cylinder as shown in figure 2. There are several advantages to this configuration. The bandwidth of the system increased due to the elimination of the air flow dynamics in the vacuum booster. The model and control were greatly simplified since the force balances related to the booster operation were eliminated. The vehicle can still be easily operated by a human driver since the vacuum booster is functional with a driver input. There is however a need for increased hydraulic supply pressure since the amplification due to the vacuum booster is lost. And last, the simplicity of the design makes any vehicle easy to retrofit for AHS use.

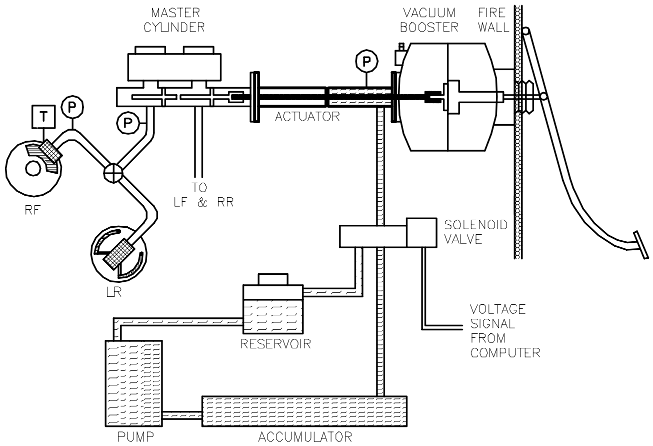


Figure 2. Brake System Diagram

Brake system modeling

An accurate yet simple model is needed to develop a control algorithm capable of meeting the stringent requirements of platooning. Based on recent analysis and experimental data a model of the brake system was developed specifically for the purpose of automatic control (Gerdes, Brown and Hedrick, 1995). The following is a brief description of the brake components model.

The response of the servo-valve has been modeled as a first order system. The flow from the solenoid valve to the actuator or from the actuator to the reservoir is proportional to the square root of the respective pressure differences:

$$Q_{act} = C_{act}\sqrt{P_s - P_{act}} \tag{3}$$

The relationship between the actuator pressure and the master cylinder pressure is inversely proportional to the ratio of the areas of the two cylinders so that

$$P_{mc} = (P_{act} \cdot A_{act} - F_{sp} - F_{sf}) / A_{mc} \tag{4}$$

where F_{sp} is the master cylinder spring preload and F_{sf} is the seal friction.

The relationship between the pressure at the master cylinder and the one at the slave cylinders has seen less attention in previous attempts to control the brakes. In reality, the behavior is an incompressible flow with nonlinear capacitance that follows Bernoulli's Law. The flow is therefore proportional to the square root of the pressure difference between the master cylinder and the slave cylinders at the wheels:

$$Q_w = C_w \cdot \sqrt{P_{mc} - P_w} \tag{5}$$

In a dynamic sense, this translates into a lag. Such lag is undesirable since it can adversely affect tracking and/or ride quality. However, through proper control, this lag can be greatly reduced.

Finally, there is the relationship between the pressure at the slave cylinder and the brake torque generated. Under steady state conditions a quasi-linear relationship can be assumed. However, that relationship changes with temperature, vehicle speed, friction material and several other parameters. Unfortunately many of these states are unmeasurable in real time and therefore inadequate for control. It is therefore suggested to use a sliding mode controller in order to compensate for the modeling errors. Furthermore, the concept of using a brake torque sensor in the feedback loop is analyzed. The simulation section shows the outcome of each control strategy.

3 Controller Development

Section 2 introduced a simplified powertrain model and the brake system model. A detailed discussion of the control methodology is presented in this section. The powertrain control algorithm has been developed and improved over the last several years. The focus of this study will be the development of a brake control algorithm. The goal is to track the velocity of the preceding vehicle while maintaining constant longitudinal spacing and passenger comfort.

Application to AHS

Due to the operation of the brake system, a natural approach is to use a multiple surface sliding controller (Green and Hedrick, 1990). A sliding controller forces a system to a surface and then tracks along that surface. The first surface of this system is based on the spacing error which translates into an engine speed error. The second surface is based on the brake torque error and it dictates the desired pressure at the master cylinder. Finally, the third surface is based on the actuator pressure error and it leads to the desired solenoid valve input voltage.

Vehicle speed control

Assuming that the automatic transmission is locked in overdrive, there is a linear relationship between engine speed and vehicle speed:

$$v = h \cdot R_g^* \cdot \omega_e \tag{6}$$

where

R_g^* = transmission gear-dependent variable

Therefore the desired engine speed can be expressed as:

$$\omega_{e,des} = \frac{v_{des}}{h \cdot R_g^*} \tag{7}$$

Due to this relationship it can be assumed that a change in the vehicle speed will be directly reflected in a change in engine speed. Therefore, a change in the throttle angle will change the engine speed which in turn will change the vehicle speed, while a change in the brake torque T_b will change the vehicle speed which in turn will affect the engine speed.

Brake torque control

From equation (2), by observation, the brake torque appears in the first derivative of the engine speed. Therefore, the first sliding surface is defined as:

$$S_1 \equiv \omega_e - \omega_{e,des} \tag{8}$$

Its first time derivative is then

$$\dot{S}_1 = \dot{\omega}_e - \dot{\omega}_{e,des} = \frac{1}{J_e}\left[T_i - T_f - T_d - T_r - T_{br}\right] - \dot{\omega}_{e,des} \tag{9}$$

Therefore

$$T_{b,des} = \hat{J}_e\left[\frac{1}{J_e}\left(T_i - T_f - T_d - T_r\right) - \dot{\omega}_{e,des} + K_1 S_1\right] \tag{10}$$

where $\hat{J}_e = \left(\sqrt{J_{e,min}^{-1} \cdot J_{e,max}^{-1}}\right)^{-1}$

In order for the controller to operate under parameter uncertainties, K_1 was designed to tolerate a 20% error in J_e and 10% error in T_i, T_f, T_d, and T_r. Therefore K_1, needing to account both for multiplicative errors and additive errors, is of the form (Slotine and Li, 1991):

$$K_1 = \frac{(1-\beta_{min})\left|\hat{\hat{f}}\right| + \alpha + \eta}{\beta_{min}} \tag{11}$$

where

$$\alpha = \left|.1c_3 m_a + .1c_4\omega_e + .1h \cdot F_r + .1c_5\omega_e^2\right|$$

$$\hat{\hat{f}} = \frac{1}{J_e}\left(T_i - T_f - T_d - T_r\right) - \dot{\omega}_{e,des}$$

$$\eta = 1.0$$

$$\beta_{min} = \frac{\hat{J}_e}{J_{e,min}}$$

Wheel brake pressure control

As it was mentioned in section 2 there is a quasi-linear relationship between wheel brake pressure and brake torque. Therefore

$$P_{w,des} = f\left(T_{b,des}\right) \tag{12}$$

The function *f*, however varies with friction material, vehicle speed and other environmental conditions. However, assuming the function is known, the desired wheel pressure is determined and the second surface is defined as the error between the actual and desired caliper pressures:

$$S_2 = P_w - P_{w,des} \tag{13}$$

and

$$\dot{S}_2 = \dot{P}_w - \dot{P}_{w,des} \tag{14}$$

However, $\dot{P}_w$ is proportional to the flow from the master cylinder to the wheel brake caliper. Therefore

$$\dot{P}_w = k_w \cdot C_w \cdot \sqrt{P_{mc} - P_w} \tag{15}$$

Substituting in equation (14)

$$\dot{S}_2 = k_w \cdot C_w \cdot \sqrt{P_{mc} - P_w} - \dot{P}_{w,des} \tag{16}$$

From the above equation the desired master cylinder pressure can be obtained:

$$P_{mc,des} = P_w + \left(\frac{\dot{P}_{w,des}}{k_w \cdot C_w}\right)^2 - K_2 \cdot S_2 \tag{17}$$

where $K_2 = \alpha + \eta$.

However, in this situation the system will run open-loop from the wheel pressure to the brake torque. Because there is an uncertain relationship between P_w and T_b, this presents a problem in tracking and passenger comfort. The suggested solution is to estimate the relationship between wheel pressure and torque and use sliding mode control with multiplicative error to compensate for the difference. Furthermore, this solution greatly reduces the transport lag between the master cylinder and the slave cylinder. Therefore:

$$P_{mc,des} = \hat{k}_b \cdot \left(T_{b,est} + \left(\frac{\dot{T}_{b,des}}{k_w \cdot C_w}\right)^2 - K'_2 \cdot S_2\right) \tag{18}$$

where

$$\hat{k}_b = \left(\sqrt{k_{b,min}^{-1} \cdot k_{b,max}^{-1}}\right)^{-1}$$

$$K'_2 = \frac{\left(1-\beta_{min}\right)\left|\hat{\hat{f}}\right| + \alpha + \eta}{\beta_{min}}$$

$$\beta_{min} = \frac{\hat{k}_b}{k_{b,min}}$$

However, if a brake torque sensor is installed, the actual brake torque measurement can be fed back and the second surface becomes

$$S'_2 = T_b - T_{b,des} \tag{19}$$

and

$$\dot{S}'_2 = \dot{T}_b - \dot{T}_{b,des} \tag{20}$$

Then, substituting in equation (18)

$$P_{mc,des} = \hat{k}_b \cdot \left(T_b + \left(\frac{\dot{T}_{b,des}}{k_w \cdot C_w}\right)^2 - K'_2 \cdot S'_2\right) \tag{21}$$

This solution eliminates the estimation error between the wheel pressure and the brake torque. More importantly, a brake torque measurement feedback closes the loop around the master cylinder/slave cylinder subsystem. Without one, the loop is closed around the vehicle speed. Since the dynamics of that system are much slower, poor tracking and diminished passenger comfort are the net result.

The simulation section shows the results using each of the above control methods.

Actuator pressure control

From equation (4), the actuator pressure can be determined within the model knowledge of the master cylinder spring preload and the seal friction. The difference between the actual and modeled values is treated as model error and compensated for in the sliding controller.

The final surface is based on the error between the actual and desired actuator pressures:

$$S_3 = P_{act} - P_{act,des} \tag{22}$$

and

$$\dot{S}_3 = \dot{P}_{act} - \dot{P}_{act,des} \tag{23}$$

But $\dot{P}_{act}$ is proportional to the flow from the valve to the actuator. Therefore:

$$\dot{P}_{act} = k_{act} \cdot C_{act} \sqrt{P_s - P_{act}} \tag{24}$$

where P_s is the commanded pressure at the solenid valve and thus proportional to the valve input voltage.

Substituting in equation (24)

$$\dot{S}_3 = k_{act} \cdot C_{act} \sqrt{P_s - P_{act}} - \dot{P}_{act,des} \tag{25}$$

From the above equation we can obtain the commanded pressure

$$P_s = \hat{r}\left(P_{act} + \left(\frac{\dot{P}_{act,des}}{k_{act} \cdot C_{act}} \right)^2 - K_3 \cdot S_3 \right) \tag{26}$$

and K_3 is designed to compensate for the additive modeling error (including master cylinder spring preload error and seal friction error) and multiplicative error (ratio of piston areas). Therefore:

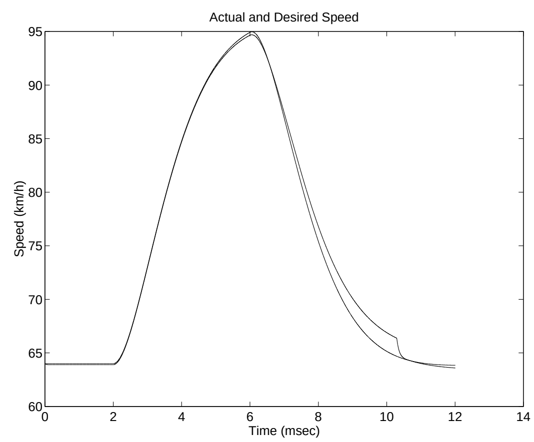
$$K_3 = \frac{(1-\beta_{min})\left|\hat{\bar{f}}\right| + \alpha + \eta}{\beta_{min}}$$

$$\hat{r} = \left(\frac{\sqrt{a_{min} \cdot a_{max}}}{a} \right)^{-1}$$

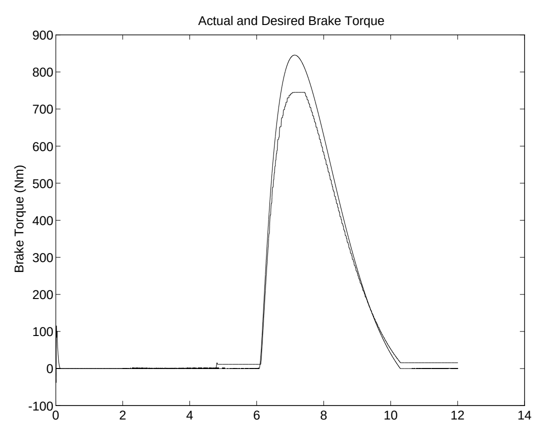
$$a = \frac{A_{mc}}{A_{act}}$$

4 Simulation Results

A smooth speed trajectory was designed for simulation purposes. It represents a typical acceleration/deceleration maneuver performed at highway speed under "normal" conditions. Each controller's ability to track the speed profile while maintaining passenger comfort is thus analyzed. Figure 3 shows the desired and actual speed trajectories using the open-loop relation between the wheel pressure and brake torque. Figure 4 shows the performance of the controller using an estimate of the pressure to torque relation and a sliding mode controller to compensate for the modeling errors. Finally, figure 5 shows the same results using a brake torque sensor in the feedback loop. As it can be seen from the "smoothness" of the actual brake torque, the goal of achieving passenger comfort while maintaining good speed tracking is achieved. However, using the brake torque measurement in the feedback loop provides the best tracking performance without compromising passenger comfort.

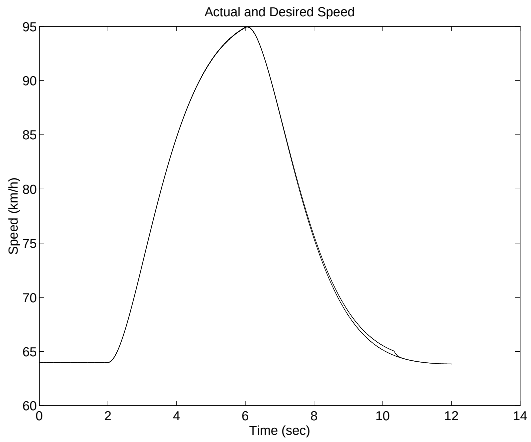


a. Actual and Desired Vehicle Speed

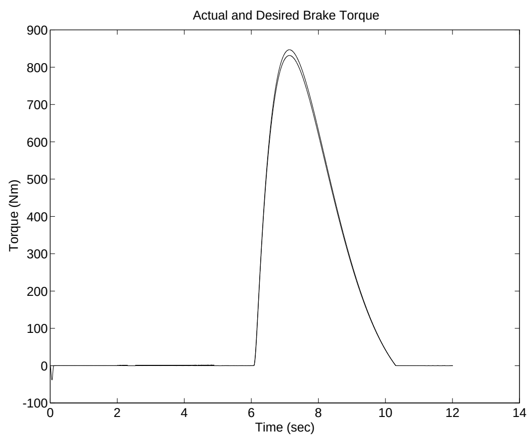


b. Actual and Desired Brake Torque

Figure 3. Open Loop Performance

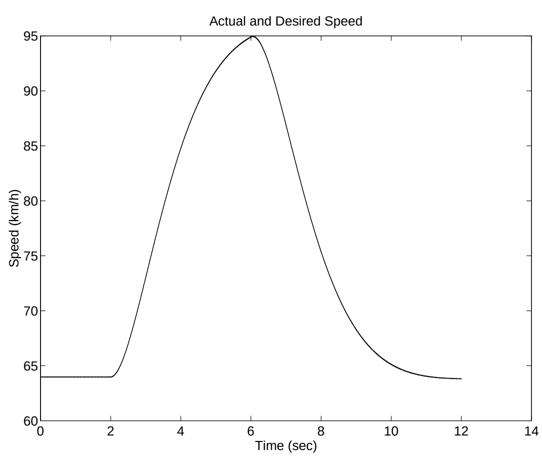


a. Actual and Desired Vehicle Speed

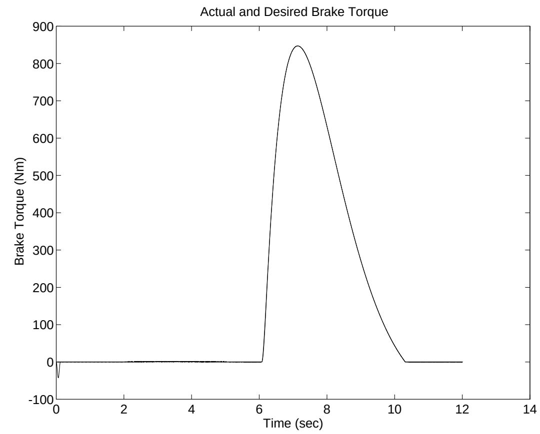


b. Actual and Desired Brake Torque

Figure 4. Torque Estimation and Sliding Mode Control



a. Actual and Desired Vehicle Speed



b. Actual and Desired Brake Torque

Figure 5. Brake Torque Measurement Feedback

5 Conclusions

A second generation nonlinear brake control algorithm for automated vehicle platooning was developed. Three versions of this controller were developed based on the sensors available. The simulation results show good speed tracking in all cases even in the presence of disturbances and modeling errors. Of the three versions however, using brake torque feedback shows the best performance. For experimental purposes, the control algorithm derived here will be implemented on test vehicles. It is expected that experimental results will corroborate the simulation results.

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References

[1] Bowman, J.E., and Law, E.H., "A Feasibility Study of an Automotive Slip Control Braking System," SAE Paper 930762, 1993.

[2] Cho, D., and Hedrick, J.K., "Automotive Powertrain Modeling for Control," *Transactions ASME Journal of Dynamic Systems, Measurements and Control*, Vol.111, No.4, December, 1989.

[3] Gerdes, J.C., Brown, A.S., and Hedrick, J.K., "Brake

System Modeling for Vehicle Control," *Proceedings International Mechanical Engineering Congress and Exposition*, 1995.

[4] Green, J.H., and Hedrick, J.K, "Nonlinear Speed Control of Automotive Engines," *Proceedings American Control Conference*, San Diego, 1990

[5] Hedrick, J.K., McMahon, D.H. and Swaroop, D., "Vehicle Modeling and Control for Automated Highway Systems," *California PATH Report*, UCB-ITS-PRR-93-24, November, 1993.

[6] Kishi, M., Watanabe, T., Hayafune, K., Yamada, K., and Hayakawa, H., "A Study on Safety Distance Control," *Proceedings of the 26th ISATA*, Aachen, Germany, 1993.

[7] Maciuca, D.B., Gerdes, J.C., and Hedrick, J.K., "Automatic Braking Control for IVHS," *Proceedings International Symposium on Advanced Vehicle Control (AVEC '94)*, Tsukuba, Japan, 1994.

[8] Martin, P., "Autonomous Intelligent Cruise Control Incorporating Automatic Braking," *ABS/TCS and Brake Technology*, SP-953, SAE Paper 930803, 1993.

[9] Slotine, J.-J.E., and Li, W., *Applied Nonlinear Control*, Prentice Hall, New Jersey, 1991.