Any Intelligent Vehicle and Highway System (IVHS) longitudinal control scheme must incorporate an appropriate brake system. The focus of this paper is on the development of a controller for a retrofitted brake actuation system. Since the current actuation system involves maintaining the existing brake system, several difficulties arise due to the highly nonlinear dynamics of the vacuum booster. Due to these dynamics a nonlinear control method was proposed. The method suggested in this study is a modification of the technique known as sliding control. It was chosen due to its robustness under the presence of modeling errors and disturbance inputs. Also, due to the physical constraints of the booster operation, this technique had to be adapted to a case of discrete control input. Simulation and experimental results are presented to illustrate the capability of this controller to follow a desired speed trajectory while maintaining a constant spacing between vehicles.

Keywords/ Automatic Braking, Brake, Brake Dynamics, Advanced Vehicle Control System, Modeling

1. INTRODUCTION

The concept of IVHS and automatic brake control are not new. These topics have seen a lot of research in the 60's and 70's. Unfortunately, most of the early attempts have fallen short of the desired goal to automatically control the brakes smoothly and safely in an AHS environment. However, due to a need to increase capacity, reduce pollution and improve safety on the major highways, the topics have seen renewed interest in the recent years.

Partners for Advanced Transit and Highways (PATH) and CalTrans have supported IVHS research since 1988. This particular study emphasizes the development of an automatic brake controller for IVHS. The controller was developed for a Lincoln Town Car retrofitted for IVHS use, but it can be easily modified for other cars. Although retrofitting provides an easy, reliable and liability-free way to automate the brakes, it also makes the job of controlling them more difficult.

Recent analysis of the brake system shows that it contains a series of nonlinearities. First, the vacuum booster, which is an integral part of the brake system, contains deadzones, spring preloads, and nonlinear fluid flow dynamics. Second, the dynamics of vacuum booster are dependent on the engine manifold dynamics which are nonlinear. The third source of nonlinearities is the actuation system. Composed of a solenoid valve and a hydraulic piston, this actuation system has a high order dynamic response.

Due to these nonlinearities, the linear control methods have failed to meet the necessary requirements. The controller must also ensure good performance and reliability over a large range of operating points. Therefore a nonlinear controller was used for this study.

Section 2 describes the mathematical model used to design and simulate the controller. The model includes a simplified powertrain. The brake system relies on a model complex enough to capture the important dynamics, and yet simple enough to facilitate the controller design and reduce computation time.

The actual brake controller is developed in Section 3. The methodology used was the Lyapunov-based technique of Sliding Control. In order to accommodate implementation issues, a multiple surface sliding controller was used. Also, modifications were made to accommodate the discrete type input present in the vacuum booster.

The controller provided good tracking even under the presence of modeling errors and noise. This kind of performance was obtained both in simulation and experiments. These results are presented in Section 4.

2. MATHEMATICAL MODEL

The powertrain model was adapted from Cho and Hedrick (1989) and McMahon, Hedrick and Shladover (1990). Since the emphasis of this study is brake control, a simplified model of the powertrain is used. The model is complete enough though to capture the dynamics of the vehicle. The brake model was developed by Gerdes, et. al. (1993) specifically for the task of brake control in an IVHS environment.

2.1 Powertrain
A simplified three state model of an automotive powertrain was used. For a more complex model see [5]. 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 simplified model.

The two state equations for the engine are:

\[ m_a \dot{\omega}_e = T_c - T_f - T_r - T_d - T_h \]

\[ \omega_e = \frac{1}{J_v}(T_i - \omega_e - m_a) \]

where

- \( m_a \) - mass of air in the intake manifold
- \( \omega_e \) - engine speed
- \( T_c \) - throttle characteristic
- \( J_v \) - effective vehicle inertia
- \( T_i \) - indicated torque
- \( T_f \) - friction torque
- \( T_r \) - rolling resistance
- \( T_d \) - aerodynamic drag
- \( T_h \) - total brake torque
- \( h \) - effective tire radius

The third and final state is the combined brake torque. In the past it has been modeled as a first order linear system. The new model is discussed in detail in the following sections.

2.2 Retrofitted Brake System

This study revolves around a Lincoln Town Car retrofitted for IVHS studies. There are several reasons why a retrofitted brake system was chosen over a system specifically designed for IVHS. Most importantly, such a system is easy to design and implement. The time frame and financial cost are considerably reduced. Furthermore, such a system is more reliable since it relies heavily on the existing brake system.

Figure 2 is a schematic of the brake system present in the test vehicle.

As it can be seen from the above diagram, the voltage signal from a computer controls the flow of hydraulic fluid to the actuator through the use of a solenoid valve. Once the actuator starts pulling on the brake pedal through the firewall, the system operates in the same manner as if a human driver would apply the brakes. The only significant difference over a stock brake system is that in this system the brake pressure, the pressure in the two chambers of the vacuum booster and the pressure in the actuator can be measured.

One disadvantage that a retrofitted brake system has is that it is a lot more difficult to control. First, the current hydraulic actuation system has a high order dynamic response which is difficult to model. Further downstream, the vacuum booster is an integral part of any modern brake system. Its purpose is to amplify the operator's input by a factor of approximately 10 and thus reduce the operator fatigue and provide the higher braking pressure required for the operation of disk brakes.

The vacuum booster though adds to the complexity of controlling the brakes by introducing a series of nonlinearities. One key difficulty is the fact that the booster has a low bandwidth internal feedback loop. Another problem is that the available input is virtually a three stage discrete input: apply, hold and release. This makes it difficult, accurate operation difficult.
The operation of the vacuum booster is documented in literature (see Puhn, 1985 and Gerdes, et. al., 1993). However, what is important to notice in Figure 2.3 is the source of nonlinearities. Both the valve springs and the return spring have preloads that must be overcome before any motion can take place. This gives rise to several deadzones. The air flow dynamics from atmosphere to the apply chamber (in the apply mode) and from the apply to vacuum chamber (in the release mode) is also a phenomenon that presents nonlinear dynamics. Important to notice is also the fact that the power piston has three discrete modes of operation. In the apply mode air is allowed into the apply chamber from the atmosphere while the two chambers are sealed from each other. In the hold mode the chambers remain sealed but no air is admitted. In the release mode air is still not admitted in the apply chamber and the two chambers are connected thus equalizing the pressure across the diaphragm. As long as the pressure in the vacuum chamber is greater than the manifold pressure, the air drains to the manifold through a check valve.

2.3 Brake Model

Based on recent experimental data and analysis of the brake system a mathematical model was developed (Gerdes, et. al., 1993). As mentioned earlier, the automatic brake system is composed of the actuation system, the brake booster and the master cylinder and the brake lines.

Due to the solenoid valve behavior, the actuator presents a high order dynamic response to a step input. Furthermore, this response is very difficult to model. It is however possible, for control purposes, to bound it. This will be the topic of Section 3.5.

The brake booster dynamics have been modeled in two sections: force balance and air dynamics. The force balance equations are presented below:

\[ F_{in} - F_{at} - F_{vs} \cdot F_{mc} = 0 \]  
\[ F_{at} + F_{d} - F_{rs} \cdot (1 - r) \cdot F_{mc} = 0 \]

where

\[ F_{in} = \text{pushrod input force} \]
\[ F_{vs} = \text{valve springs force} \]
\[ F_{mc} = \text{master cylinder force} \]
\[ F_{d} = \text{booster diaphragm force} \]
\[ F_{rs} = \text{return spring force} \]
\[ r = \text{pushrod/master cylinder reaction washer ratio} \]

The booster diaphragm force can be approximated as the pressure difference over the area of the diaphragm

\[ F_{d} = (P_a - P_v) \cdot A_d = \Delta P \cdot A_d \]  

For the air dynamics, the ideal gas law dictates that

\[ P = \frac{m}{V} RT \]  

And therefore, the pressure change at constant temperature is

\[ \Delta P = \frac{m}{V} RT \cdot \dot{V} \text{m}RT \]

The temperature is assumed constant since the engine compartment temperature is constant under steady state operating conditions and the vacuum booster is located in the engine compartment.

From the booster operation it is also known that the check valve from the vacuum chamber to the manifold is always open when the vacuum chamber pressure is higher than the manifold pressure. Also, the booster valve air ports open and close in about 1ms and therefore it can be assumed that they have only discrete orifice areas. The booster valve operation can be summarized in the following table:

<table>
<thead>
<tr>
<th>Mode</th>
<th>Atmosphere to Apply</th>
<th>Apply to Vacuum</th>
</tr>
</thead>
<tbody>
<tr>
<td>APPLY</td>
<td>OPEN</td>
<td>CLOSED</td>
</tr>
<tr>
<td>HOLD</td>
<td>CLOSED</td>
<td>CLOSED</td>
</tr>
<tr>
<td>RELEASE</td>
<td>CLOSED</td>
<td>OPEN</td>
</tr>
</tbody>
</table>

Based on the air flow equations and the booster valve operation, the change in pressure difference over the booster diaphragm can be represented as:

\[ \Delta P = P_{at} \cdot u_1 - P_{AV} \cdot u_2 + P_{mv} \]  

where the subscripts

\[ a = \text{atmosphere} \]
\[ A = \text{apply chamber} \]
\[ V = \text{vacuum chamber} \]
\[ m = \text{manifold} \]

and \( u_1 \) and \( u_2 \) are described by the following table:

<table>
<thead>
<tr>
<th>Mode</th>
<th>( u_1 )</th>
<th>( u_2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>APPLY</td>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>HOLD</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>RELEASE</td>
<td>0</td>
<td>1</td>
</tr>
</tbody>
</table>

Lastly, from experimental results, it was determined that there is a linear relationship between brake torque (Nm) and the force applied to the master cylinder (N). Its empirical solution is:

\[ F_{mv} = 2.94 P_{mc} - 5351 \]  

3. CONTROLLER DEVELOPMENT

3.1 Introduction

The development of a simplified powertrain and a retrofitted brake models were discussed in Section 2. A detailed discussion of a control methodology is presented in this section. Since the longitudinal control using throttle angle has been investigated in the past, the focus of this study will be concentrated on longitudinal control using brakes. For this specific purpose, the control task is to track the velocity of the preceding vehicle while maintaining a constant spacing.

3.2 Application to Advanced Highway Systems

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 engine speed error. The second surface is based on the brake torque error and it dictates the mode in which the power piston should be in (apply, hold or release). Finally, the third surface is based on the
actuator force error and the resulting input is the voltage signal to the solenoid valve.

3.3 Brake Torque Controller

By observation, from Equation 33, the brake torque appears in the first derivative of the engine speed. Therefore the first sliding surface is defined as:

\[ S_1 = \dot{\omega}_e - \omega_{edes} \]  \hspace{1cm} (10)

Its first time derivative is then

\[ \dot{S}_1 = \ddot{\omega}_e - \omega_{edes} = \frac{1}{J_e} \left[ T_e - T_f - T_d - T_{br} \right] - \dot{\omega}_{edes} \]  \hspace{1cm} (11)

Therefore

\[ T_{br,des} = J_e \left[ \frac{1}{J_e} \left( T_e - T_f - T_d - T_{br} \right) - \dot{\omega}_{edes} + K_{br} \dot{S}_1 \right] \]  \hspace{1cm} (12)

In order for the controller to operate under parameter uncertainties, \( K_{br} \) was designed to tolerate a 20% error in \( J_e \) and 10% error in \( T_1, T_f, T_d, \) and \( T_{br} \). Therefore \( K_{br} \) is of the form (Slotine and Li, 1991):

\[ K_{br} = \frac{(1 - \beta_{min}) \dot{\hat{f}} + \alpha + \eta}{\beta_{min}} \]  \hspace{1cm} (13)

where

\[ \alpha = \left[ 0.377 + 0.001 \tau_0 + 0.000027 \omega_\text{rpm}^2 \right] \]

\[ \dot{\hat{f}} = \frac{1}{J_e} \left( T_e - T_f - T_d - T_{br} \right) - \dot{\omega}_{edes} \]

\[ \eta = 1.0 \]

\[ \beta_{min} = 0.82 \]

3.4 Booster Controller

In this section it will be assumed that the available pressure difference in the vacuum booster is high enough so that, in the apply mode, the booster has the potential to provide the required force without additional input force from the actuator. In this case the actuator force needs only be large enough to open the booster valve to atmosphere. Since the actuator can only provide a "pull" action, in release mode the actuator cannot help reduce the brake torque faster than the booster can purge the air to vacuum.

Under these conditions, a linear relationship between the desired pressure difference over the booster diaphragm and brake torque under specific operating conditions can be derived:

\[ \Delta p_{des} = 8031 + 5.6 T_{br,des} \]  \hspace{1cm} (14)

By inspection it can be seen that the controller appears after the first derivative, therefore

\[ S_2 = \Delta p - \Delta p_{des} \]  \hspace{1cm} (15)

\[ \dot{S}_2 = \dot{\Delta p} - \dot{\Delta p}_{des} = \dot{p}_{a_{in}} - \dot{p}_{a_{out}} - u_2 + \dot{p}_{a_{br}} - \Delta p_{des} \]  \hspace{1cm} (16)

As it can be seen, it would be very difficult to obtain the values for \( u_1 \) and \( u_2 \) from the above equations. Furthermore, a large number of parameter uncertainties would be introduced since the flows are highly dependent on the effective orifice areas which are difficult to calculate accurately. However, since \( u_1 \) and \( u_2 \) can only take the values described previously, the problem simplifies significantly. It’s also important to notice that in the apply mode, for example, \( V_A \) is positive or zero, \( V_y \) is negative or zero, \( m_{av} \) is positive or zero, \( m_{cv} \) is zero and \( m_{cv} \) is positive or zero. The objective is to have \( S_2 < 0 \). Therefore when \( S_2 < 0 \), \( S_2 > 0 \) which can only be achieved when \( u_1 = 1 \) and \( u_2 = 0 \) or apply mode. The opposite is true when \( S_2 > 0 \). This statement is also physically intuitive. It says that if the pressure difference is too low, more pressure is needed in the apply chamber, and so the brakes should be applied. The significance of this statement is even more far reaching. It implies that this control scheme can be used with any vacuum booster that operates in this manner as long as the pressure difference measurement is available.

The booster valve position, in turn, correlates with the valve spring force \( F_{VS} \). The above statements can be summarized in the following table:

<table>
<thead>
<tr>
<th>( S_2 )</th>
<th>Mode</th>
<th>( u_1 )</th>
<th>( u_2 )</th>
<th>( F_{VS} ) [N]</th>
</tr>
</thead>
<tbody>
<tr>
<td>( S_2 &lt; 0 )</td>
<td>APPLY</td>
<td>1</td>
<td>0</td>
<td>89.0</td>
</tr>
<tr>
<td>( S_2 = 0 )</td>
<td>HOLD</td>
<td>0</td>
<td>0</td>
<td>75.0</td>
</tr>
<tr>
<td>( S_2 &gt; 0 )</td>
<td>RELEASE</td>
<td>0</td>
<td>1</td>
<td>62.0</td>
</tr>
</tbody>
</table>

3.5 Actuator Control

The assumption made in the previous section will be revisited here. There will be cases when the air in the vacuum chamber would not have had enough time to drain to the manifold and the desired pressure difference in the booster will not be available fast enough. In such cases, the booster will not be able to provide the entire braking force. However, the hydraulic system present in the Lincoln Town Car is able to provide forces large enough to provide braking even in the absence of the vacuum booster. Therefore the actuator control should take advantage of this availability in order to provide faster and safer braking. Therefore the desired actuator force is:

\[ F_{in,des} = \left( F_{in,des} + r \cdot F_{mo} - r \cdot \frac{S_2}{1 - r} \right) / L_1 \]  \hspace{1cm} (17)

where

\[ p = \begin{cases} 0 \text{ if } S_2 > 0 \text{ (can’t push on the pedal)} \\ 1 \text{ if } S_2 \leq 0 \text{ (can pull on the pedal)} \end{cases} \]

\[ L_1 = \text{brake pedal linkage ratio} \]

In the above equation the first two terms represent the amount of force required to obtain the desired \( F_{VS} \) and the third term represents the compensation for not having enough pressure differential of the booster diaphragm.

At this point, the solenoid valve currently in operation cannot provide the actuator with a "clean" input. A step input on the valve reveals high order dynamics. The solution suggested to solve this problem is to bound the dynamics between two first order responses and to treat the difference in the two gains and time constants as parameter uncertainties which can be bounded. A typical step response for the valve along with the first order dynamics bounding it are shown in Figure 4.
Figure 4. Brake Actuator Step Response

Considering the above concepts the brake actuator dynamics can be estimated as:

$$F_m = \frac{(v - v_{off}) \cdot k \cdot A_{act} - F_{in}}{\tau}$$ (18)

where

- $v$ = input voltage
- $v_{off}$ = voltage offset
- $A_{act}$ = actuator area
- $k$ = proportionality constant between voltage and actuator pressure

Since the voltage is the input, by inspection, the input appears after the first differentiation. Therefore:

$$S_3 = F_m - F_{in,des}$$ (19)

$$S_3 = \dot{F}_m - \dot{F}_{in,des} = \frac{(v - v_{off}) \cdot k \cdot A_{act} - F_{in} - \dot{F}_{in,des}}{\tau}$$ (20)

and so

$$v = \frac{\tau}{k \cdot A_{act}} \left( \frac{v_{off} \cdot k \cdot A_{act} + F_{in} \cdot \tau + \dot{F}_{in,des} - K_{act} S_3}{\dot{f}} \right)$$ (21)

In implementation $F_{in,des}$ is approximated as

$$\left( F_{in,des}(t + \Delta t) - F_{in,des}(t) \right) / \Delta t$$ since it is difficult and computationally intensive to calculate it analytically. $K_{act}$ must be designed so that the higher dynamics fit in the bounds determined by the first order responses. Therefore (Slotine and Li, 1991):

$$K_{act} = \frac{(1 - \beta_{min})}{\beta_{min}} \left| \frac{\dot{f}}{\dot{f}} + \alpha + \eta \right|$$ (22)

where

$$\dot{f} = \frac{v_{off} \cdot k \cdot A_{act} + F_{in} \cdot \tau + \dot{F}_{in,des}}{\tau}$$

Allowing for 20% error in $k$ and $\tau$ and 10% error in $A_{act}$ and $v_{off}$ the following expressions were obtained for $\alpha$ and $\beta_{min}$:

$$\alpha = 2.24 + 26.41 F_{vq} + 0.0033 P_{ac}$$ (23)

$$\beta_{min} = 0.73$$

$$\eta = 10.0$$

However, since $F_{vq}$ is not a measurable state, its largest value can be used to calculate $\alpha$. Therefore

$$\alpha = 52651 + 0.0033 P_{ac}$$ (24)

4. BRAKE CONTROLLER PERFORMANCE

4.1 Simulation Results

A speed trajectory was designed for simulation purposes. It represents a typical acceleration/deceleration maneuver performed at highway speeds under "normal" conditions. Figures 5 and 6 show simulation results of the desired and actual speed trajectories and the speed error associated with them.
4.2 Experimental Results

The experiments conducted concentrated on the ability of the controller to maintain a constant spacing from the lead vehicle. For this purpose, a ramp speed profile was used simulating an idealized lead vehicle. Figure 7 shows the desired and actual speed used in these tests.

![Figure 7. Desired and Actual Speed (Experimental)](image)

More importantly however, is the ability of the controller to maintain a constant space from the lead vehicle or any other desired trajectory. As Figure 8 shows, the largest spacing error is less than 0.2m.

![Figure 8. Spacing Error (Experimental)](image)

5. CONCLUSIONS

An automotive brake system model was developed and used in designing an automatic brake controller. A powertrain model was also adapted for this purpose. Due to the nonlinear nature of both models a nonlinear control methodology was chosen. Due to its robustness in the presence of model errors and input disturbances, the method of Sliding Control was used. A multiple sliding surface was used since the relative degree of the system was greater than one.

This study showed that it is possible to control a retrofitted brake system in the frame of IVHS longitudinal control. Simulation results show good speed tracking for a platoon of two vehicles. The experimental results corroborated the simulation predictions. The three surface controller shows good tracking even in the presence of modeling errors.

The results derived here must be extended to cases when more severe commands are required such as emergency maneuvers. Experiments with larger platoons are planned in the near future.

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