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Abstract

Vehicle lateral dynamics are affected by vehicle mass, longitudinal velocity, vehicle inertia, and the cornering stiffness of the tires. All of these parameters are subject to variation, even over the course of a single trip. Therefore, a practical lateral control system must guarantee stability, and hopefully ride comfort, over a wide range of parameter changes. This paper describes a robust controller which theoretically guarantees stability over a wide range of parameter changes. The performance of the robust controller is then evaluated in simulation as well as on a test vehicle. Test results for experiments conducted on an instrumented track are presented, comparing the robust controller to a PID controller which was tuned on the vehicle.

1 Introduction

One of the fundamental goals of the Intelligent Vehicle-Highway Systems (IVHS) community is to develop automated highways where vehicles are capable of automatically driving down the road, either individually or in platoons of multiple vehicles. In order to implement such a system, a controller that can keep the vehicle centered in the lane is required. There are many factors which make automatic lateral control of vehicles difficult. These include changing vehicle parameters (tire pressure, tire wear, etc.), changing road conditions (rain, ice, bumps, crowns, etc.), as well as disturbances caused by wind and other factors. Another important consideration is driver comfort while performing lane changes and reacting to disturbances.

Initial research efforts on automated highway systems (AHS) were conducted by the Radio Corporation of America in cooperation with General Motors in the late 1950's [1, 2]. A significant amount of research, including the development of prototype experimental equipment, was conducted at Ohio State University between 1964-1980 [3]. This included research on both lateral and longitudinal control of highway vehicles. The largest current advanced vehicle control system (AVCS) research effort is being conducted at California PATH (Partners for Advanced Transit and Highways) [4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14].

The PATH program has been investigating a frequency shaped linear quadratic (FSLQ) optimal control approach for the lateral controller, with feedforward preview control to reduce feedback gains [4, 6, 7]. Although the FSLQ approach incorporates ride qualities into the performance index, other work which attempts to design a lateral controller taking into account ride comfort is described in [15]. Recent work on robust control applied to car steering is described in [16, 17, 18, 19, 20, 21]. While many of the previously mentioned efforts rely on buried magnets, electrified wires, or a microwave radar to determine the vehicle's lateral position, another promising approach involves using vision. Efforts at Carnegie Mellon University (CMU), at the National Institute of Standards and Technology (NIST), and in Germany have yielded promising experimental results using neural networks and classical vision algorithms [22, 23].

This paper describes a robust lateral controller which theoretically guarantees stability over a wide range of parameter changes. The controller is designed with the plant uncertainty modeled as unstructured additive perturbations in the frequency-domain. This approach, first described in [24], is reviewed in Section 2. Extensions to the current theory which are applied to the car problem are also described in Section 2. The modeling of the vehicle's lateral dynamics is discussed in Section 3. The controller design and simulation results are presented in Sections 4, and test results are presented in Section 5. A summary and discussion of planned future research is outlined in Section 6.

2 The Robust Stability Condition

Modeling system uncertainty as unstructured additive perturbations in the frequency-domain is described by equation (1), where the nominal plant transfer function is $p_0(s)$, and the uncertainty in the transfer function is $\delta p(s)$.

$$p(s) = p_0(s) + \delta p(s) \qquad (1)$$

The robust stability problem for additive unstructured perturbations then reduces to the problem of finding a strictly bounded real (SBR) function $u(s)$ which interpolates at the unstable poles of the nominal plant in the RHP. This interpolation problem is often referred to as the Nevanlinna-Pick interpolation problem. There are limitations to the approach presented by Kimura, which arise from limitations of the Nevanlinna-Pick interpolation theory. The current theory has difficulties with interpolation points with multiplicity, as well as with interpolation points on the jw axis. Techniques for handling these two cases are outlined in [25]. These techniques are applicable to the lateral control of automobiles because the lateral dynamics model contains a double integrator as described in the next section.

3 Model of Lateral Dynamics

The two degree-of-freedom linearized bicycle model for a vehicle's lateral dynamics will be used in this section to model the test vehicle, a GMC Jimmy. The estimated nominal parameters for the Jimmy and the

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Parameter	Nominal Value	Range
m, vehicle mass	$1590\ kg$	constant
V_x, longitudinal velocity	$8\ m/s$ ($28.8\ kmph$)	$5 - 10\ m/s$ ($18 - 36\ kmph$)
l_1, l_2, dist. from axles to $c.g.$	$1.17\ m, 1.42\ m$	constant
C_r, C_f, tire cornering stiffness	$42000\ Kn/rad$	0.85 to 1.15
I_z, inertia about z axis	$3200\ kg \cdot m^2$	constant
l^*, dist. from $c.g.$ to sensor	$2\ m$	constant

Table 1: Estimated Parameters for S-15 Blazer

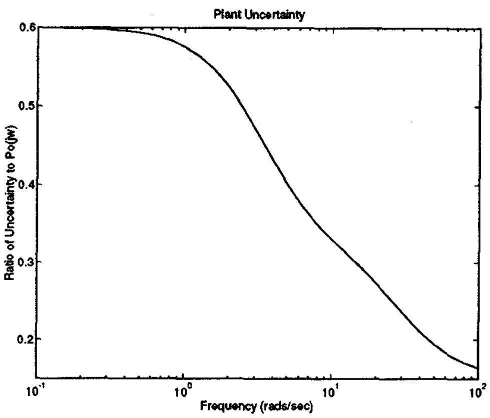

Figure 1: Simulation of Plant Uncertainty

xpected range of values are listed in Table 1. The ıominal speed of 8 m/s corresponds to 28.8 $kmph$, vhile the extreme speeds correspond to 18 $kmph$ and 6 $kmph$ respectively. These values were chosen beause the initial field testing will be conducted at relıtively low speeds. Simulation results for highway peeds appear in [26].

Using the values in Table 1, the nominal car model s given by equation (2).

$$\frac{E_f(s)}{\delta_f(s)} = \frac{114.2552(s^2 + 13.4391s + 31.4366)}{s^2(s^2 + 24.3156s + 151.9179)} \quad (2)$$

vhere E_f is the lateral error (m) at the sensor, and δ_f is the front steering angle (rad)

ı MATLAB® program was written to determine the ound on the frequency-domain uncertainty of the ıominal plant as the velocity and cornering stiffness ary over the ranges in Table 1. The program finds he magnitude of

$$|\delta p(jw)| = |p_0(jw) - p(jw)| \quad (3)$$

here $p_0(s)$ is the transfer function of the nominal lant and $p(s)$ is the transfer function of the actual lant as the parameters are varied. The results of he computer simulation are shown in Figure 1. The esults are plotted as the ratio $|\frac{\delta p(jw)}{p_0(jw)}|$. This format f data presentation was chosen because it facilitates e choice of $r(s)$ as a function of $p_0(s)$. This simplies the calculations required to arrive at the robust ontroller in the next section.

4 Controller Design

Using the data in Figure 1, a conservative bound on the plant uncertainty can be expressed as

$$r(s) = 0.6p_0(s) \quad (4)$$

With this bound, a robustly stabilizing controller can be designed if there exists an SBR solution to the interpolation problem. It should be noted that this choice of uncertainty is equivalent to the uncertain gain problem, whose solution is described in [27, 28]. However, a less conservative gain could be chosen which would require the use of the techniques described in [25].

A robustly stabilizing compensator for the lateral control problem was obtained in [25] and is shown in equation (5).

$$c(s) = \frac{(2s^2+1.5s+0.25)(s^2+24.3156s+151.9179)}{114.2552(0.64s^2+2.64s+1.16)(s^2+13.4391s+31.4366)} \quad (5)$$

The poles of the nominal-closed loop system are -2.5, -0.625, and -0.5. Test results for the robust lateral control algorithm are presented in the next section.

5 Test Results

The theoretical lateral control algorithm presented in the previous section was implemented on a test vehicle. For the purposes of this research, the 1989 GMC Jimmy was modified for drive-by-wire operation, where there are no mechanical linkages between the driver's steering commands and the motion of the wheels. A hydraulic control system is used to move the wheels under computer control. A one mile test track was instrumented with a wire reference system to sense the lateral position of the vehicle. The lateral control algorithm presented in the previous section was discretized using the bilinear transformation shown in equation (6).

$$s = \frac{2}{T}\frac{z-1}{z+1}, \ T = 100\ ms \quad (6)$$

The gain of the theoretical controller was increased by a factor of 2 to overcome approximately 1 degree of backlash in the steering actuator. A small integration term was also added to the robust controller to overcome center steering offsets and road superelevation. Testing was conducted on a straight section of the track, at speeds of 20 $kmph$, 30 $kmph$, and 40 $kmph$. The test results are shown in Figures 2 - 4.

A proportional-integral-derivative (PID) control algorithm was also implemented for comparison to the robust controller. The gains of the PID controller were hand tuned on the vehicle. The performance of the PID algorithm at the same three speeds is shown in Figures 5 - 7. A summary of the rms error for each test run appears in Table 2.

Speed ($kmph$)	Robust Controller rms error	PID Controller rms error
20	0.1085 m	0.0857 m
30	0.0751 m	0.0858 m
40	0.0953 m	0.0779 m

Table 2: RMS Error for PID and Robust Controllers

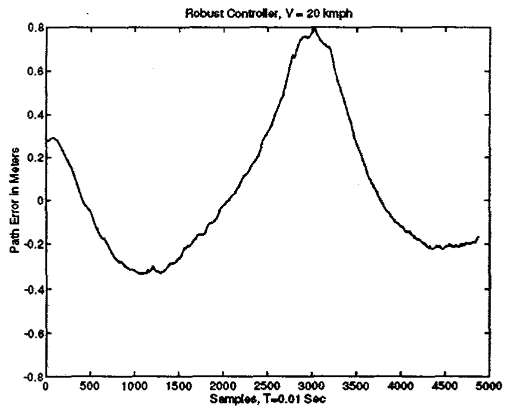

Figure 2: Robust Controller, $V_x = 20\ kmph$

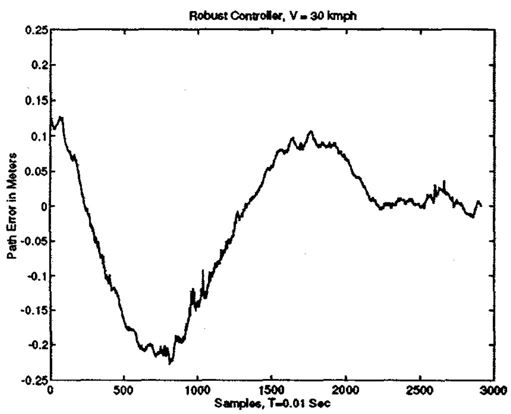

Figure 3: Robust Controller, $V_x = 30\ kmph$

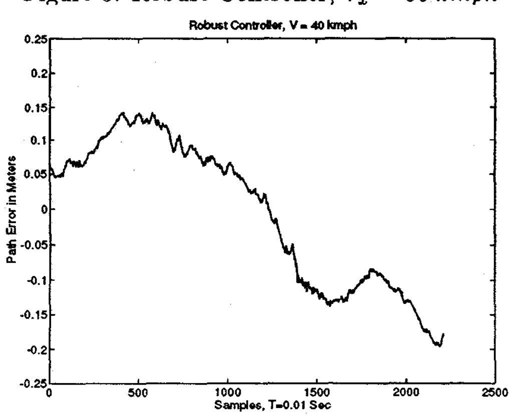

Figure 4: Robust Controller, $V_x = 40\ kmph$

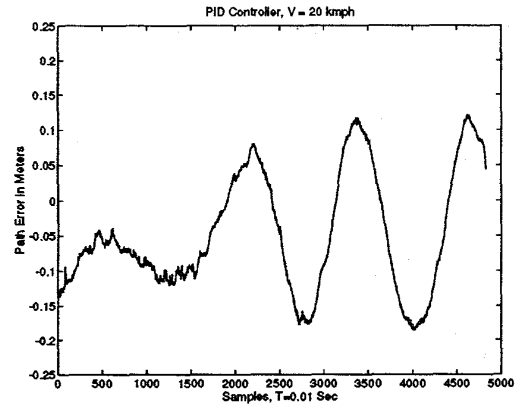

Figure 5: PID Controller, $V_x = 20\ kmph$

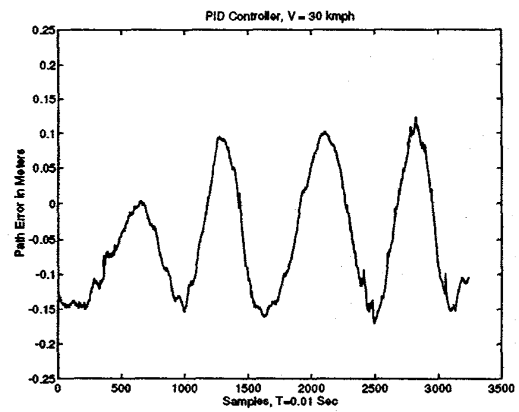

Figure 6: PID Controller, $V_x = 30\ kmph$

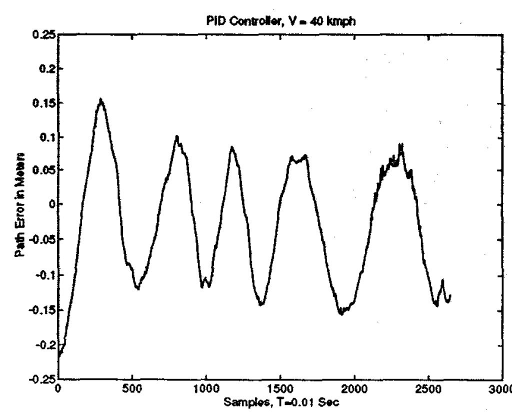

Figure 7: PID Controller, $V_x = 40\ kmph$

6 Summary and Conclusions

This paper presents experimental results for a robust lateral control system designed for an automated vehicle. The benefits of a robust control algorithm include guaranteed stability over a wide range of operating conditions and a fixed controller as opposed to gain scheduling. For this application, the uncertainty in vehicle velocity and cornering stiffness were modeled as unstructured additive perturbations in the frequency-domain. Based on the expected uncertainty modeling, interpolation techniques described in [25] were used to design a robust lateral control algorithm. The robust controller was then tested on a GMC Jimmy test vehicle that was modified for drive-by-wire operation. Testing was conducted on straight sections of an instrumented 1 mile test track. A PID controller, that was tuned on the vehicle, was also implemented for comparison purposes.

Although the robust controller performed satisfactorily, the vehicle control experiments highlighted several implementation difficulties. The uncertainty modeling was fairly conservative, which resulted in a performance tradeoff. Unmodeled uncertainties like steering actuator backlash and center steering offsets were also problematic. In addition, the robust controller was sensitive to initial vehicle orientation at the start of the test runs. Future research will focus on incorporating these types of uncertainty into the controller design as more structured uncertainty. By incorporating more knowledge of the system uncertainty into the robust design, performance can probably be improved while still maintaining the desired robustness to parameter changes.

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