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Design of a Model Reference Adaptive Controller for Vehicle Road Following

R. H. BYRNE
Department 9616, MS 1125, Sandia National Laboratories
Albuquerque, NM 87185-1125, U.S.A.
ray@eece.unm.edu

C. T. ABDALLAH
E.E.C.E. Department, University of New Mexico
Albuquerque, NM 87131, U.S.A.
chaouki@eece.unm.edu

Abstract—The design of a model reference adaptive controller (MRAC) for vehicle road following is discussed. There are many factors which make automatic lateral control of vehicles difficult. These include changing vehicle parameters, changing road conditions, as well as disturbances caused by wind and other factors. Traditional controllers have difficulty guaranteeing performance and stability over a wide range of parameter changes. Model reference adaptive controllers on the other hand, can guarantee both stability and performance over a wide range of slowly varying parameter changes as long as several conditions are met. Difficulties with model reference adaptive controllers (as well as other controllers) arise when saturation appears on the input or states. For vehicle applications, the steering angle is limited. Therefore, the adaptation gains for the MRAC designed in this paper were chosen so that the steering command remained reasonable, even during adaptation. We finally show that there is a tradeoff regarding input amplitude and input frequency during adaptation.

Keywords—Adaptive control, IVHS, Vehicle control.

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 autonomously 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 road 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 [3–12]. 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 as part of the Program on Advanced Vehicle Technology for the Highway (PATH, more recently Partners for Advanced Transit and Highways [13]) in California [13–16]. The PATH program has been investigating a frequency shaped

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Typeset by $\mathcal{A}_{\mathcal{M}}\mathcal{S}$-TEX linear quadratic (FSLQ) optimal control approach for the lateral controller, with feedforward preview control to reduce feedback gains [13,16]. 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 [17]. Recent work on robust control applied to car steering is described in [18–20]. 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) and in Germany have yielded promising experimental results using neural networks and classical vision algorithms [21,22].

Traditional controllers have difficulty guaranteeing performance and stability over a wide range of parameter changes. Robust controllers can guarantee stability over a wide range of parameter changes, but have difficulty in guaranteeing performance (i.e., percent overshoot, damping, etc.) as well as stability. Another approach, feedback linearization, is promising for dealing with the nonlinearities present in the car model, but has difficulties with plant variations over time unless they are known *a priori*. Optimal control can minimize a performance index related to control performance and passenger ride comfort with some inherent robustness properties (infinite gain margin and ± 60 degree phase margin), but these robust properties are only for a particular type of uncertainty, and are lost without full state feedback. (Some of the optimal control robustness properties can be regained with loop transfer recovery.) Model reference adaptive controllers, however, can guarantee both stability and performance over a wide range of slowly varying parameter changes as long as several conditions are met. These conditions are described in Section 3.

Difficulties with model reference adaptive controllers (as well as other controllers) arise when saturation appears on the input or states. For vehicle applications, the steering angle is limited mechanically, and also practically because the linearized car model is only valid for small steering angles, especially at higher velocities. For example, the steering on the GMC Blazer discussed in this paper is limited to approximately ± 28 degrees. However, depending on the velocity and vehicle trajectory, much smaller steering angles are needed to keep the lateral acceleration less than 0.2 g (on the order of ± 4 degrees). In this paper, the adaptation gains were chosen so that the steering command remained reasonable, even during adaptation. There is, however, a tradeoff regarding input amplitude and input frequency during adaptation. Some work has been done on investigating the stability of adaptive systems with constrained states/inputs, but this work is not applicable to the control of unstable plants with relative degree 2 (the car problem) [23–25]. Therefore, this remains an area of further research.

This paper is organized as follows. The vehicle model is discussed in Section 2. Section 3 outlines the adaptive controller design. Simulation results are presented in Section 4. Finally, a summary of the results and our conclusions appear in Section 5.

2. VEHICLE MODELING

Several different models have been used to simulate the dynamics of an automobile [26]. One common approach is to treat a four-wheeled vehicle as a two-wheeled system. This is often called the "bicycle model" and is shown in Figure 1. If the assumption is made that the steering angle is kept small, the equations describing the lateral motions of a vehicle with the steer angle as the only input variable are given in (2.1) and (2.2) (see [26]).

$$m\dot{V}_y + \left[mV_x + \frac{2l_1C_{\alpha f} - 2l_2C_{\alpha r}}{V_x}\right]\Omega_z + \left[\frac{2C_{\alpha f} + 2C_{\alpha r}}{V_x}\right]V_y = 2C_{\alpha f}\delta_f(t), \tag{2.1}$$

$$I_z\dot{\Omega}_z + \left[\frac{2l_1^2C_{\alpha f} + 2l_2^2C_{\alpha r}}{V_x}\right]\Omega_z + \left[\frac{2l_1C_{\alpha f} - 2l_2C_{\alpha r}}{V_x}\right]V_y = 2l_1C_{\alpha f}\delta_f(t). \tag{2.2}$$

Field tests at Ohio State University in the 1970's showed that the bicycle model is accurate for lateral accelerations of less than 0.2 g [3].

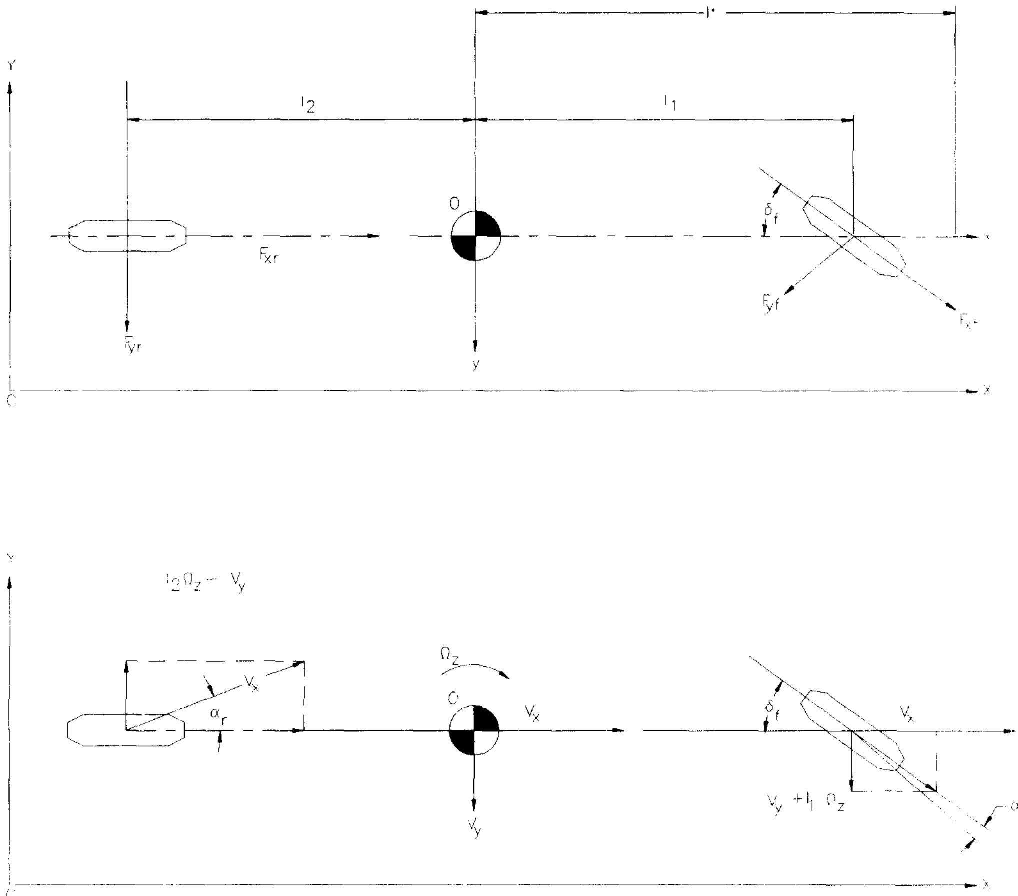

Figure 1. Simplified vehicle model for analysis of transient motions [26].

The parameters in equations (2.1) and (2.2) are described in Table 2.1. Estimated values for a GMC S-15 Blazer are also listed in the table. Rearranging equations (2.1) and (2.2) yields the state space system described by (2.3).

$$\dot{x} = Ax + Bu, \tag{2.3}$$

where

$$A = \begin{bmatrix} a_{11} & a_{12} \\ a_{21} & a_{22} \end{bmatrix}, \qquad B = \begin{bmatrix} b_1 \\ b_2 \end{bmatrix}, \quad u = \delta_f(t), \quad \text{and} \quad x = \begin{bmatrix} V_y \\ \Omega_z \end{bmatrix}.$$

$$\begin{aligned}
a_{11} &= -\left[\frac{2C_{\alpha f} + 2C_{\alpha r}}{V_x m}\right], \\
a_{12} &= -\left[V_x + \frac{2l_1 C_{\alpha f} - 2l_2 C_{\alpha r}}{V_x m}\right], \\
a_{21} &= -\left[\frac{2l_1 C_{\alpha f} - 2l_2 C_{\alpha r}}{V_x I_z}\right], \\
a_{22} &= -\left[\frac{2l_1^2 C_{\alpha f} - 2l_2^2 C_{\alpha r}}{V_x I_z}\right], \\
b_1 &= \frac{2C_{\alpha f}}{m} \qquad b_2 = \frac{2l_1 C_{\alpha f}}{I_z}.
\end{aligned}$$

Table 2.1. Description of parameters.

Parameter	Value	Description
m	1727 kg	vehicle mass
l_1	1.17 meter	distance from front axle to center of gravity
l_2	1.42 meter	distance from rear axle to center of gravity
$C_{\alpha f}$	47 KN/rad	front tire cornering stiffness
$C_{\alpha r}$	47 KN/rad	rear tire cornering stiffness
I_z	2867 kg • m^2	inertia about z-axis
Ω_z		yaw velocity about center of gravity
V_y		lateral velocity
V_x	20 m/s	longitudinal velocity (controlled by a cruise control), approximately 45 mph
δ_f		steering angle input, in radians

The two state variables are yaw rate and lateral velocity. For this application, the quantity to be minimized is the lateral path error E. The lateral path error E is a function of the lateral velocity V_y, the heading Θ, and the longitudinal velocity V_x.

This relation is shown in equations (2.4) and (2.5).

$$\dot{E} = V_y + V_x\Theta, \tag{2.4}$$

$$\dot{\Theta} = \Omega_z. \tag{2.5}$$

The augmented state space model is shown in equation (2.6).

$$\begin{bmatrix} \dot{V}_y \\ \Omega_z \\ \dot{\Theta} \\ \dot{E} \end{bmatrix} = \begin{bmatrix} a_{11} & a_{12} & 0 & 0 \\ a_{21} & a_{22} & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 1 & 0 & V_x & 0 \end{bmatrix} \begin{bmatrix} V_y \\ \Omega_z \\ \Theta \\ E \end{bmatrix} + \begin{bmatrix} b_1 \\ b_2 \\ 0 \\ 0 \end{bmatrix} \delta(t). \tag{2.6}$$

It is assumed that the lateral path error E is the measurable output of the system. Lateral path error may be sensed in a number of ways which include vision, sensing buried magnets, and sensing a buried wire. Therefore, the C matrix is described by equation (2.7).

$$C = [0 \quad 0 \quad 0 \quad 1]. \tag{2.7}$$

The lateral path error is also the quantity which must be controlled. The transfer function of interest is thus the transfer function from steering angle input to path error output. This may be determined using equation (2.8).

$$\frac{E(s)}{\delta_f(s)} = H(s) = C(sI - A)^{-1}B + D. \tag{2.8}$$

Because of the difficulty in computing $(sI - A)^{-1}$ symbolically, the transfer function will be computed using the parameters estimated for a GMC S-15 Blazer which were outlined in Table 2.1. It is also assumed that the vehicle speed will be held constant by a cruise control. The adaptive controller designed in the next section will allow cruising at different velocities, but the model becomes nonlinear if the speed is not held constant. For the nominal values in Table 2.1, the transfer function from steering to lateral error is:

$$\frac{E(s)}{\delta_f(s)} = \frac{54.43\left(s^2 + 4.02s + 84.91\right)}{s^2\left(s^2 + 7.33s + 21.50\right)}. \tag{2.9}$$

Note that the relative degree is 2, the numerator is Hurwitz, the denominator has a double root at the origin, and the sign of the high frequency gain is known (positive). Now that the structure of the plant is known, the next section describes the design of a model reference adaptive controller for lane following.

3. CONTROLLER DESIGN

The block diagram for a model reference adaptive controller [27] based on a Lyapunov approach is shown in Figure 2. The input to the system is $r(t)$, and the reference model is

$$Wm(s) + Km\frac{Zm(s)}{Rm(s)}. \tag{3.1}$$

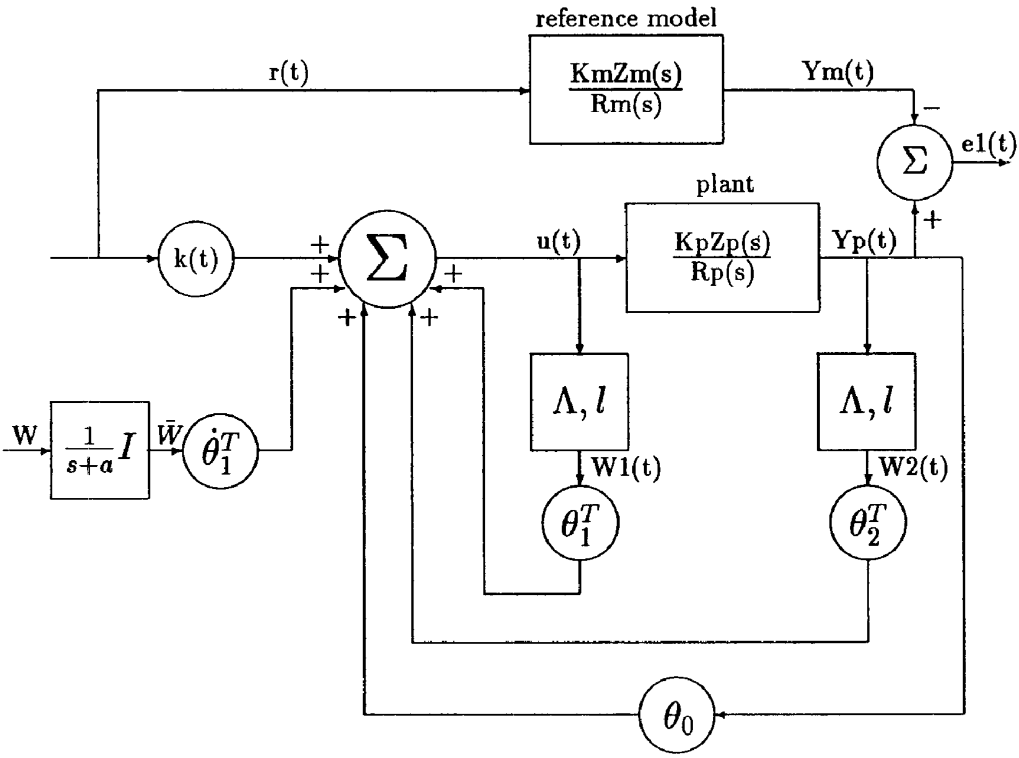

Figure 2. Block diagram of model reference adaptive controller.

The plant is represented by

$$Wp(s) = Kp\frac{Zp(s)}{Rp(s)}. \tag{3.2}$$

The following assumptions about the plant are made:

(i) The sign of the high frequency gain Kp is known.
(ii) An upper bound m on the deg $Rp(s)$ is known.
(iii) $m^* =$ deg Rp–deg Zp is known $= 2$.
(iv) $Zp(s)$ is a monic polynomial that is Hurwitz.

The input to the plant is $u(t)$, the output of the plant is $Yp(t)$, the output of the model is $Ym(t)$, and the error or difference between the reference model and plant output is $e1(t)$. The adaptive controller gains are $k(t)$, $\theta_0(t)\,\theta_1(t)$, and $\theta_2(t)$. In Figure 2, Λ is an $(n-1)\times(n-1)$ stable matrix (where $n = \deg Rp(s) = 4$), ℓ is an $(n-1)$ vector, and (Λ, ℓ) is controllable. (Λ, ℓ) filter the plant input $u(t)$ and plant output $Yp(t)$. $W(t)$ and $\theta(t)$ are defined as

$$W(t) = \begin{bmatrix} r(t) \\ W1(t) \\ Y_p(t) \\ W2(t) \end{bmatrix}, \qquad \theta(t) = \begin{bmatrix} k(t) \\ \theta_1(t) \\ \theta_0(t) \\ \theta_2(t) \end{bmatrix}. \tag{3.3}$$

$W1$, $W2$, θ_1, and θ_2 are $(n-1)$ vectors while r, k, Y_p , and θ_0 are scalars. $\overline{W(t)}$ is defined as $\overline{W(t)} = L^{-1}(s)W(t)$ where $L(s) = s + a$.

Normally, the reference model must be strictly positive real (SPR) for a stable adaptive controller [27]. SPR is defined below.

DEFINITION 2.1. [27] *A rational function $H(s)$is SPR if and only if*

(1) *$H(s)$ is analytic in* $\mathrm{Re}[s] \geq 0$,
(2) $\mathrm{Re}[H(jw)] > 0 \; \forall w \in (-\infty, \infty)$, and
(3) (a) $\lim_{w^2\to\infty} w^2 \mathrm{Re}[H(jw)] > 0$, when $n^* = 1$, and
 (b) $\lim_{|w|\to\infty} (H(jw))/jw > 0$ when $n^* = -1$,

where n^* is the relative degree of $H(s)$.

Since the model cannot be SPR (because its relative degree $= 2$), there is some difficulty in obtaining a stable adaptive controller. This is overcome in [27] by choosing $L(s) = s + a$ such that $L(s)Wm(s)$ is SPR. Then, the controller in Figure 2 yields a stable adaptive system if the assumptions about the plant are met [27].

Assumptions (i)–(iv) are met for the automobile system described in Section 2, so if an adaptive law of the form

$$\dot{\theta} = -\mathrm{sgn}(Kp)\, e_1(t)\, \overline{W}(t) \tag{3.4}$$

is used, the overall system is stable and $\lim_{t\to\infty} e_1(t) = 0$ [27]. Using this expression for $\dot{\theta}$, the control input becomes

$$u(t) = \theta^\top(t)\, W(t) - \mathrm{sgn}(Kp)\, e_1(t)\, \overline{W}^\top(t)\, \overline{W}(t). \tag{3.5}$$

Using equation (3.2), the adaptive controller update rules are given by:

$$\dot{k} = -k_1 e_1(t)\, \bar{r}(t), \tag{3.6}$$
$$\dot{\theta}_1 = -k_1 e_1(t)\, \overline{W1}(t), \tag{3.7}$$
$$\dot{\theta}_0 = -k_1 e_1(t)\, \overline{Y}_p(t), \tag{3.8}$$
$$\dot{\theta}_2 = -k_1 e_1(t)\, \overline{W2}(t). \tag{3.9}$$

The gain k_1, sometimes referred to as the adaptive gain, is a design parameter which affects how quickly the adaptive controller adapts. For this application, k_1 was chosen to be 0.001. The performance of the controller is fairly robust with respect to changes in k_1. If k_1 is increased by several orders of magnitude, oscillations can occur. On the other hand, if k_1 is decreased by several orders of magnitude, the control system will take much longer to adapt.

Λ was chosen as $-0.1 \times I_{3\times3}$ to filter the slowest modes in the plant, and l was chosen as $l = [\,10 \quad 10 \quad 10\,]^\top$. This yields the following equations for W1 and W2.

$$\begin{bmatrix} \dot{W1}_a \\ \dot{W1}_b \\ \dot{W1}_c \end{bmatrix} = \begin{bmatrix} -0.1 & 0 & 0 \\ 0 & -0.1 & 0 \\ 0 & 0 & -0.1 \end{bmatrix} \begin{bmatrix} W1_a \\ W1_b \\ W1_c \end{bmatrix} + \begin{bmatrix} 10 \\ 10 \\ 10 \end{bmatrix} u(t), \tag{3.10}$$
$$\begin{bmatrix} \dot{W2}_a \\ \dot{W2}_b \\ \dot{W2}_c \end{bmatrix} = \begin{bmatrix} -0.1 & 0 & 0 \\ 0 & -0.1 & 0 \\ 0 & 0 & -0.1 \end{bmatrix} \begin{bmatrix} W2_a \\ W2_b \\ W2_c \end{bmatrix} + \begin{bmatrix} 10 \\ 10 \\ 10 \end{bmatrix} Y_p(t). \tag{3.11}$$

Using equations (3.3), (3.5), and the notation used in (3.8), the input to the plant $u(t)$ can be written as

$$\begin{aligned} u(t) = kr &+ \theta_{1a} W1_a + \theta_{1b} W1_b + \theta_c W1_c + \theta_0 Y_p + \theta_{2a} W2_a + \theta_{2b} W2_b + \theta_{2c} W2_c \\ &- e_1 \left(\bar{r}^2 + \overline{W1}_a^2 + \overline{W1}_b^2 + \overline{W1}_c^2 + \overline{Y}_p^2 + \overline{W2}_a^2 + \overline{W2}_b^2 + \overline{W2}_c^2 \right). \end{aligned} \tag{3.12}$$

The desired model behavior for the system will be chosen as

$$Wm(s) = \frac{11.47(s+2.5)(s+1.5)}{(s+1)(s+2)\left(s^2+7.33s+21.50\right)}. \tag{3.13}$$

This model retains the stable poles of the car, while moving the two integrator poles in the plant into the left half plane. The poles at -2 and -1 were chosen to be stable, yet not faster than the existing poles of the car to reduce the required control effort. The gain of the reference model was chosen so that a commanded input of 1 meter would result in a 1 meter output in the steady state (dc gain = 1). The performance of the model was chosen so that the system is over damped, with no overshoot to increase ride comfort. The step response of the model is shown in Figure 3.

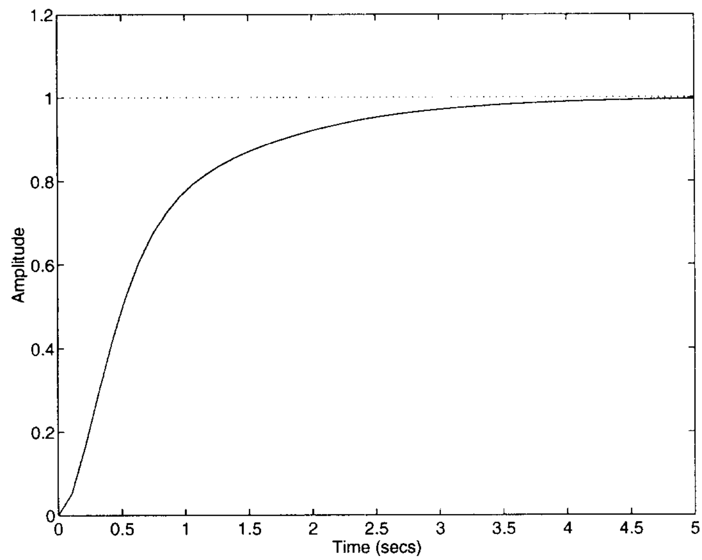

Figure 3. Model step response.

If $L(s)$ is chosen such that $L(s) = s + 4$, then $L(s)Wm(s)$ is SPR. The Nyquist plot of $L(s)Wm(s)$ is shown in Figure 4. Note that $\mathrm{Re}[H(jw)] > 0$ for all w, thus satisfying the definition of SPR.

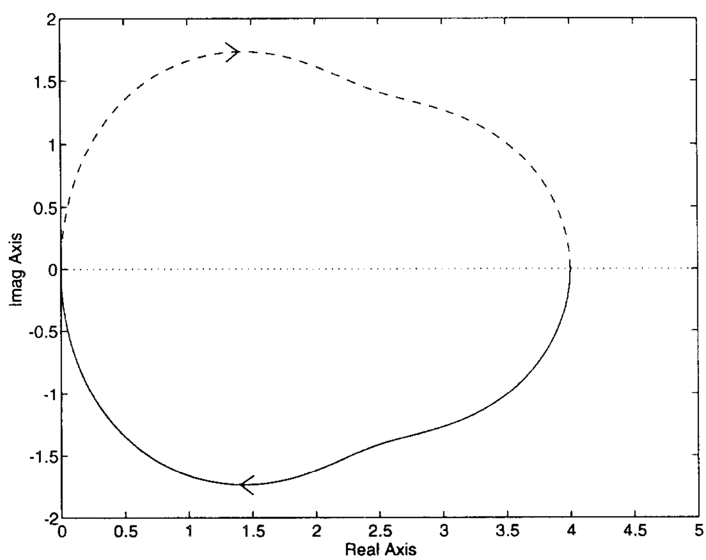

Figure 4. Nyquist plot of $L(s)Wm(s)$.

Equations (3.5)–(3.13) and (2.9) completely describe the model reference adaptive controller in Figure 2. The design parameters include the choice of k_1, Λ, l, and $Wm(s)$ which have been chosen in this section. The next section presents simulation results.

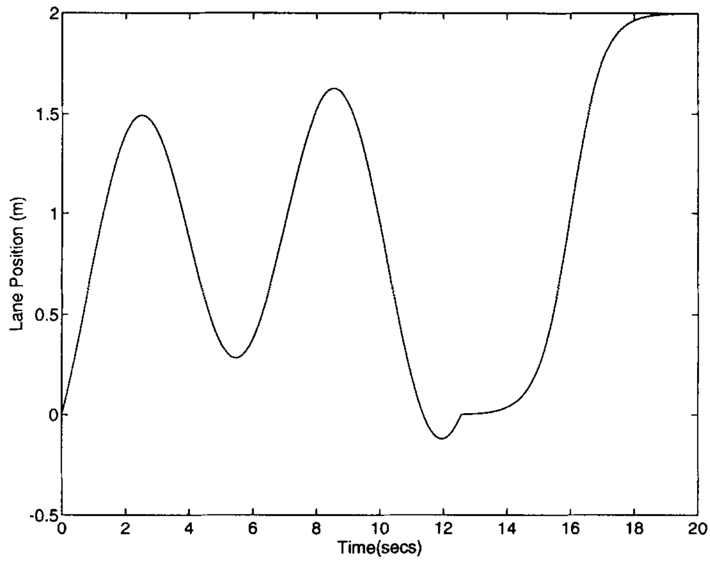

Figure 5. Figure desired trajectory.

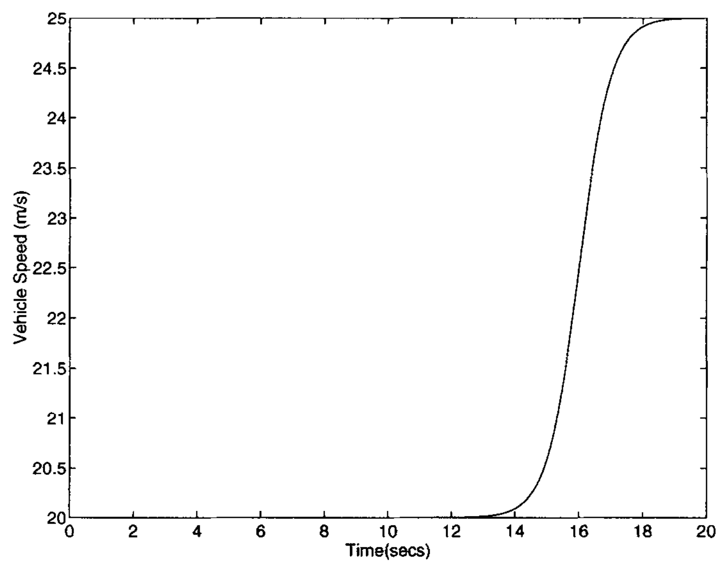

Figure 6. Vehicle speed.

4. SIMULATION RESULTS

The model reference adaptive controller for vehicle lateral control designed in the previous section was simulated using MATLAB. The commanded lateral path position was the sum of 3 sinusoids and a constant term for adaptation, and then a lane change of 2 meters. The tanh $(t - 16)$ function was used to generate a smooth trajectory for the lane change. A plot of the desired trajectory $r(t)$ is shown in Figure 5. The velocity of the vehicle was kept constant at 20 m/s for the initial adaptation, and then increased to 25 m/s while executing a lane change (45–55 mph). The velocity profile as a function of time is shown in Figure 6. The vehicle states are shown in

Figure 7. The transients that occur at about 2.5 seconds are a result of the adaptation. After the gains have reached their steady state values, the control system exhibits excellent performance. The various adaptive gains are shown in Figure 8.

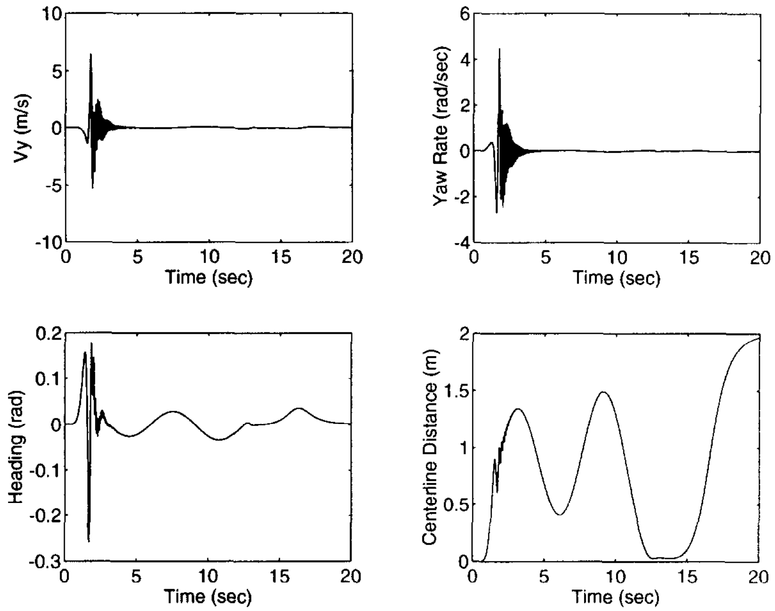

Figure 7. Vehicle states.

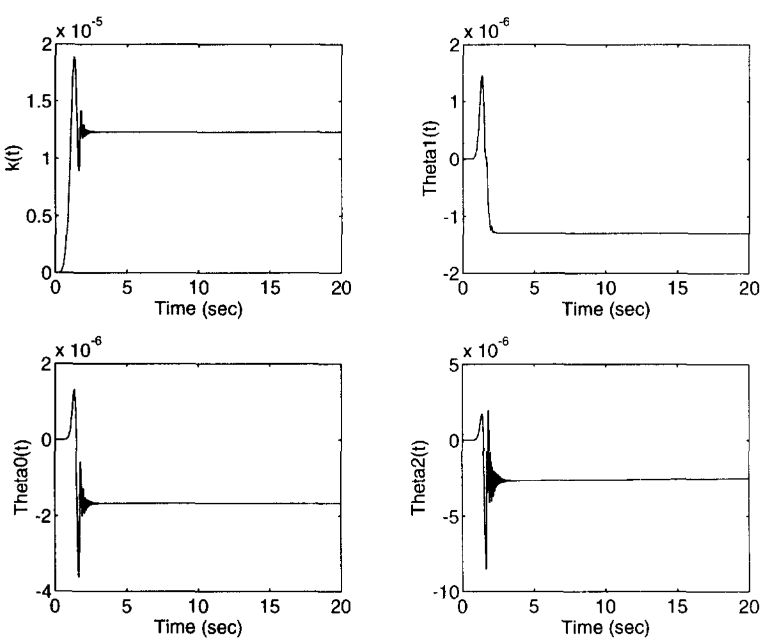

Figure 8. Adaptive gains.

The error between the plant and model is plotted in Figure 9. The maximum error is approximately 0.25 meters, which occurs during adaptation. Once the adaptive gains have reached steady state values, the plant-model error is negligible. Finally, the commanded steering angle is shown in Figure 10. The adaptive gains were adjusted to keep the magnitude of the steering command reasonable during adaptation. However, by doing so the frequency content of the steering was increased during adaptation.

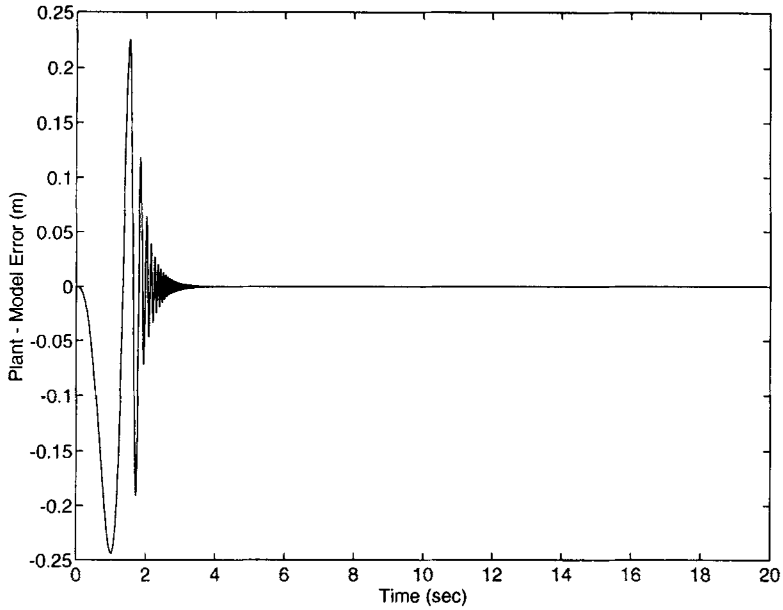

Figure 9. Plant-model error.

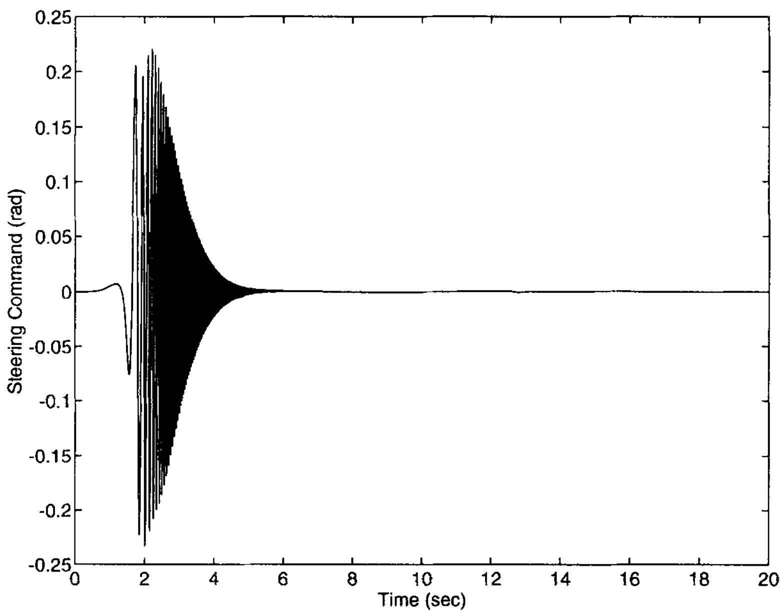

Figure 10. Steering command.

5. CONCLUSIONS

The design of a model reference adaptive controller for lateral vehicle control was outlined in this paper. The simplified bicycle model of an automobile was used to model the vehicle's lateral dynamics. Based on the vehicle parameters for a GMC S-15 Blazer, a suitable reference model was developed. The reference model retained the two stable poles of the car, and moved the unstable poles to -1 and -2. These two poles were chosen because they are stable and slower than the existing stable car poles to reduce the required control effort. A suitable $L(s)$ was chosen so that $L(s)Wm(s)$ is SPR. The MRAC was then simulated using MATLAB.

The steering angle of a car is limited, both physically and also practically, by the linear region of the simplified model. Therefore, the adaptive gains were adjusted to reduce the magnitude of the commanded steering. This resulted in an increased frequency content. It should also be noted that the MRAC controller did not adapt when saturation was introduced and the steering command saturated during adaptation. Saturation and steering bandwidth are two practical implementation issues. The steering will be limited by actuator motion, and the bandwidth of the steering commands will be limited by the actuator as well as driver comfort.

Once the controller adapted (after about 2.5 seconds), the tracking performance was excellent, even as the velocity of the vehicle was increased during a lane change. Therefore, it would appear that adaptive lateral controllers could overcome many of the problems associated with fixed controllers when various parameters are subject to change. The problem of saturation of inputs with adaptive controllers has been investigated for stable plants [23], but the stabilizability of unstable plants with input saturation has not been investigated. Therefore, based on these preliminary simulations, areas which require further research include the stability of model reference adaptive controllers for unstable plants with input saturation as well as the tradeoffs between input amplitude and frequency.

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