DRIVER ASSISTED CONTROL STRATEGIES: THEORY AND EXPERIMENT

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ABSTRACT

This work investigates the use of feedback controller augmentation of driver inputs to achieve a desired vehicle performance. The vehicle performance is specified as a Reference Model. The driver maintains nominal control of the vehicle by direct actuation of the front steering inputs. The controller then determines the appropriate rear steer inputs necessary for tracking the reference model. As a consequence the driver is able to specify, within limits, the type of handling behavior required of the vehicle. A strategy based on yaw rate control is presented. An appropriate vehicle model is developed and a polynomial pole placement technique is used to control the vehicle. To account for vehicle model changes due to variations in forward velocity, a continuous time Recursive Least Squares approach is examined for on-line identification and adaptive control. The strategy and control designs are implemented experimentally on the Illinois Roadway Simulator (IRS), a scale vehicle testbed for vehicle dynamics and controls. Results and limitations are discussed.

1. INTRODUCTION

An interesting avenue of vehicle control research is assisting the driver in his/her control of the vehicle, i.e. Driver Assisted Control (DAC). The control system would be used to provide stability and performance while still allowing the driver to dictate the path of the vehicle. Parameters such as yaw angle, yaw rate, lateral velocity, and lateral acceleration would be sensed or estimated to provide the control with the necessary information to achieve desired transient and steady state performance based on the driver's input to the vehicle.

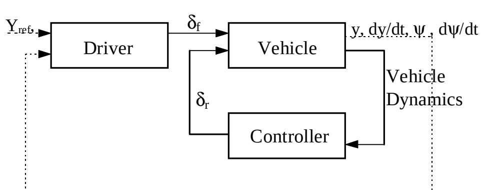

Figure 1: DAC Schematic

Implementing such a system on a vehicle would give the driver a "selectable" vehicle performance in some sense as well as providing additional stability. This type of controller can be equipped on a vehicle by using the rear wheels as the control input, still allowing the driver to steer the vehicle front wheels open loop as they are accustomed to; see Figure 1 above. A failure detection method can also be used to lock the rear wheels at zero position if the control becomes unstable for some reason. All of these features are advantages because they utilize the rear wheels as input to alter the dynamic response of the vehicle while still allowing the driver to dictate the vehicle path.

There is a wealth of literature on lateral vehicle dynamics modeling and automatic 2WS and 4WS control designs. The interested reader is referred to the survey article by Tomizuka & Hedrick (1995) and the extensive references therein. Additionally, there have been many studies done on 4WS approaches to change the vehicle's dynamics (Yamamoto, 1991, Sato, et al 1991, Lin, 1992, Inoue & Sugasawa, 1993, Furukuwa & Abe, 1997). Typically, the strategies associated with 4WS can be categorized into feedforward and feedback approaches (Inoue & Fugasawa, 1993). The feedforward approaches typically set the rear steer angles to be the product of a gain and the front steer angles:

$$\delta_r = K_{steer} \cdot \delta_f$$

In this case the goal of the controller is simply to minimize the steady state sideslip angle of the vehicle using a predefined gain as a function of the vehicle speed. Other feedforward approaches use a filtered or delayed value:

$$\delta_r = \frac{K_1 s + K_0}{\tau s + 1} \cdot \delta_f$$

where the values of the constants may be determined via Linear Optimal Control (Cho & Kim, 1995). The goal of the filtering is usually to alleviate the reverse vehicle sideslip during transient responses. There have been several feedback approaches to 4WS as well. The simplest ones have been straightforward extensions of the feedforward strategies where the vehicle yaw rate was fed back to the rear wheels.

$$\delta_r = K_{steer} \cdot \delta_f + K_{yaw} \cdot \text{yaw rate}$$

The goal of the yaw rate feedback is to utilize a measurable variable to provide additional side slip minimization at higher vehicle speeds. Some investigators considered full state feedback but lateral velocity states are inherently difficult to determine.

Many of 4WS control schemes consider the yaw rate regulation as well as robustness to disturbances such as wind gusts. However, relatively few of the previous investigations attempt to actually make the vehicle behave as if it had a different set of dynamics through feedback. Lee (1990) details work completed at General Motors Research Labs on several different strategies: rear wheel assist, front wheel and rear wheel assist, driver open loop (DOL), and driver in

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the loop (DIL). Several different combinations of feedback parameters were tried as well: yaw rate, sideslip, and lateral velocity. Ackermann (1995) gives a generic form of a robust yaw angle control method to assist the driver. The potential drawback of this implementation, as well as several of the controllers in Lee (1990), is the use of the front wheels as the control augmenting the driver's commands. A failure in the controller would not allow the driver to stabilize the vehicle because his/her input is the front wheels as well. The DAC strategies completed in the present work use the rear wheels as the control input for this very reason.

The goals of this work were to develop models of experimental vehicles as transfer functions from front and rear steer inputs to yaw angle/rate outputs. Then these models would be used to design DAC strategies with rear wheels as the control input. The motivations for this type of controller are (i) to give the driver a "tuneable" vehicle in terms of its handling performance and (ii) to give the vehicle a more stable and predicable response at higher speeds. The rest of the paper is presented as follows. Section 2 discusses the basic vehicle model and the hardware used for DAC experimentation. The Illinois Roadway Simulator (IRS) (Brennan et al, 1998) is a small scale vehicle dynamics testbed that is used to obtain experimental results. Section 2 gives modeling results for the experimental vehicle's yaw dynamics from front and rear steer inputs. Section 3 gives results for DAC strategies based on a desired yaw angle/rate performance for driver front steering input. The DAC uses the rear wheels as the control input while the driver steers the front wheels open loop. Section 4 gives results of on-line yaw model parameter estimation for DAC strategies. A conclusion summarizes the main points, including the benefits and limitations, as well as acknowledging future areas of research to be investigated.

2. VEHICLE DYNAMICS

Consider the standard planar "bicycle" vehicle model (Ellis, 1989). Assuming the longitudinal velocity remains constant, the two remaining degrees of freedom in the plane can be described by the following equations:

$$m(\dot{V} + rU) = -\frac{C_{\alpha f} + C_{\alpha r}}{U} V + \frac{bC_{\alpha r} - aC_{\alpha f}}{U} r + C_{\alpha f}\delta_f + C_{\alpha r}\delta_r \quad (1)$$

$$I_z \dot{r} = \frac{bC_{\alpha r} - aC_{\alpha f}}{U} V - \frac{a^2 C_{\alpha f} + b^2 C_{\alpha r}}{U} r + aC_{\alpha f}\delta_f - bC_{\alpha r}\delta_r \quad (2)$$

where:
m = mass of the vehicle
I_z = vehicle inertia about vertical axis at the center of gravity
U = vehicle forward velocity
C_{sf}, C_{sr} = front, rear cornering stiffnesses
a, b = distance from front, rear axle to the center of gravity
V = lateral velocity
r = yaw rate
δ_f = front steering angle
δ_r = rear steering angle

From Equations (1) and (2), the transfer function from input steer angle to vehicle yaw rate can be determined as:

$$\frac{r(s)}{\delta_f(s)} = \frac{aC_{sf}mU^2 s + C_{sf}C_{sr}(a+b)U^2}{I_z mU^2 s^2 + U(I_z(C_{sf}+C_{sr}) + m(C_{sf}a^2 + C_{sr}b^2))s + (mU^2(C_{sr}b - C_{sf}a) + C_{sf}C_{sr}(a+b)^2)} \quad (3)$$

$$\frac{r(s)}{\delta_r(s)} = -\frac{bC_{sr}mU^2 s + C_{sf}C_{sr}(a+b)U^2}{I_z mU^2 s^2 + U(I_z(C_{sf}+C_{sr}) + m(C_{sf}a^2 + C_{sr}b^2))s + (mU^2(C_{sr}b - C_{sf}a) + C_{sf}C_{sr}(a+b)^2)} \quad (4)$$

The principal difference between the two transfer functions is that the response from rear steer input to yaw rate has a negative high frequency gain. Positive rear steer input produces a negative yaw angle.

The dynamics presented in Equations (1) through (4) were used to identify the dynamics of the scaled IRS vehicle pictured below.

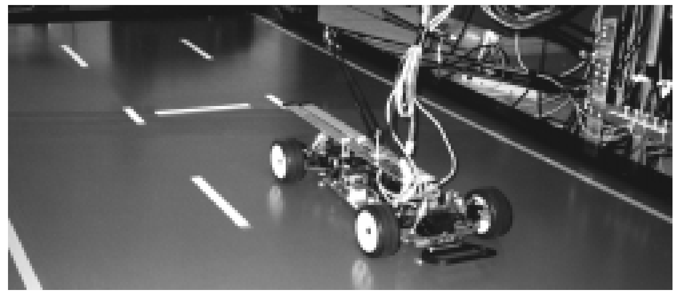
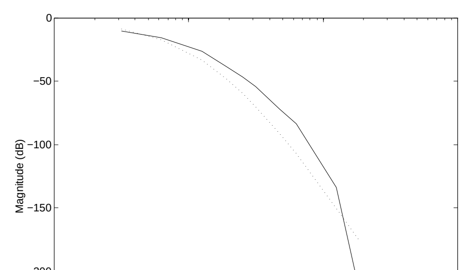


Figure 2: 4WS Experimental Vehicle

Both frequency domain and time domain model fits were attempted for the vehicle to develop as accurate a model as possible. Since the performance of the controller design presented in Section improves with system model accuracy, an accurate system identification is deemed beneficial. The bicycle model given in Equations (1)-(4) does not include many important aspects of Vehicle Dynamics including the effects that weight transfer and roll angle have on the steer dynamics. Roll steer, tire dynamics, suspension bump steer and a host of other factors will influence the vehicle beyond the simple 2 DOF model shown above. However, the simple model has been shown to be a very accurate representation of full size vehicles for non-extreme driving conditions (LeBlanc, et al 1996). Therefore, this model is used as a basis for the controllers to be developed. This is particularly justified since the control goal is to provide a driver-selectable variation in vehicle handling 'feel' and not an emergency stabilization. The data taken for system identification was done under a simple proportional closed loop controller acting to regulate the vehicle.



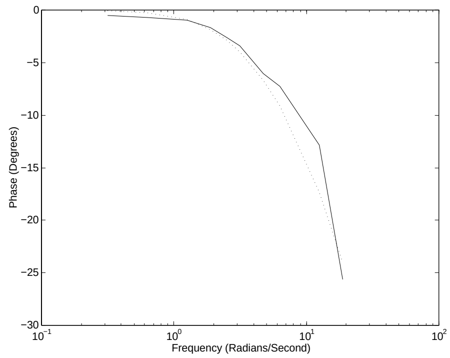


Figure 3: Closed Loop Frequency Response from Front Steer Angle to Yaw Angle
(solid line = experimental; dashed line = model)

The input in the closed loop system identification was a change in the reference. The reason for this is that the open loop vehicle contains free integrators and is therefore not stable. The IRS has a limited range of vehicle motion and so the identification was done under closed loop to avoid unintended vehicle departure from the roadway. Figure 3 shows the results of frequency domain identification for the closed loop system with a vehicle speed of 1.2 m/s. The transfer function in Figure 3 is from steer input to yaw angle, not yaw rate. The IRS sensor relied on encoder feedback that lacked the resolution to performs satisfactory differentiation to obtain yaw rate. This very important hardware limitation will be addressed again in future sections. Figure 4 shows the time domain response for an arbitrary reference signal. As is evidenced by the figure, the time domain fit is very good for the frequency domain model identified in Figure 3.

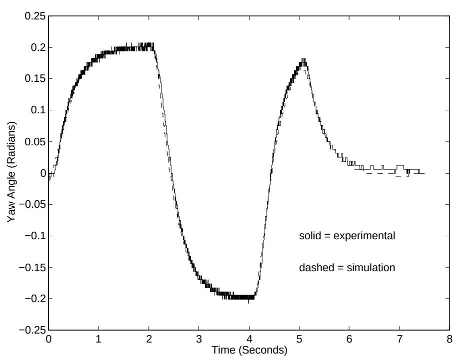


Figure 4: Closed Loop Time Response
(solid line = experimental; dashed line = simulated model)

The transfer function fit for the system in Figure 3 and 4 is given as

$$G_f(s) = \frac{40(s+10)}{s(s+13+j)(s+13-j)} \cdot G_{actuator}(s) \tag{5}$$

The free integrator in Equation (5) indicates that the transfer function is from steer input to yaw angle. Equations (1)-(4) ignore actuator dynamics and assume that the input to the system is the steer angles. The model fits of Figures (3) and (4) are unobtainable without including the dynamics of the actuator. The steer actuator is actually a rate limited 2nd order system as shown in the step responses of Figure 5

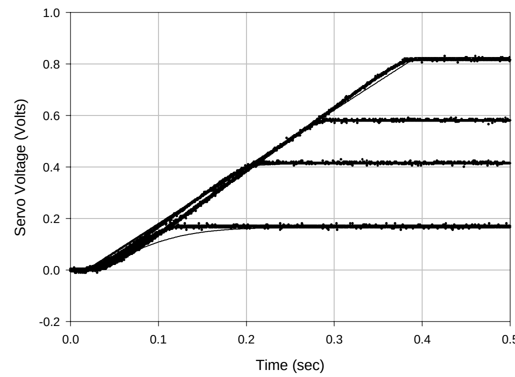


Figure 5 Actuator Dynamics: Model (solid) and Experimental (dots)

However, to simplify the controller design the actuator is modeled as a simple first order system. The rationale being that most steer angle commands will be small enough so that the rate limit is not encountered and a single real pole dominates.

$$G_{actuator}(s) = \frac{10.4712}{(s+24)} \tag{6}$$

Figure 6 and 7 show frequency and time domain identification for the rear steer angle input.

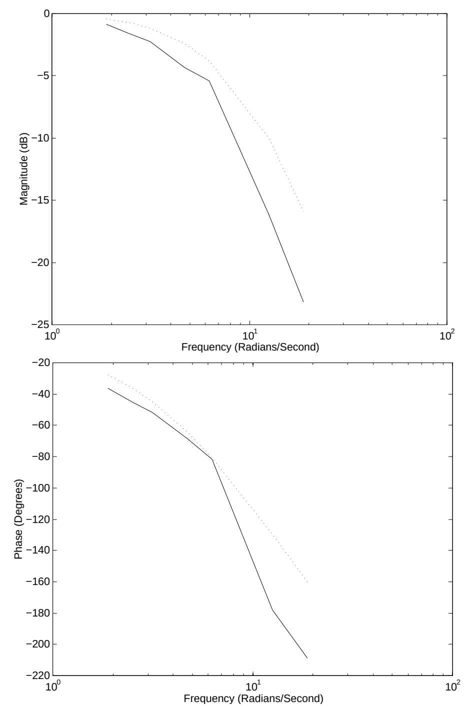


Figure 6: Closed Loop Frequency Response from Rear Steer Angle to Yaw Angle
(solid line = experimental; dashed line = model)

Similar to the front steer modeling, Figure 7 shows the time domain response comparing the identified system model with the actual experimental system for an arbitrary reference input. With the inclusion of the actuator dynamics, the performance of the identified model closely resembles that of the actual system.

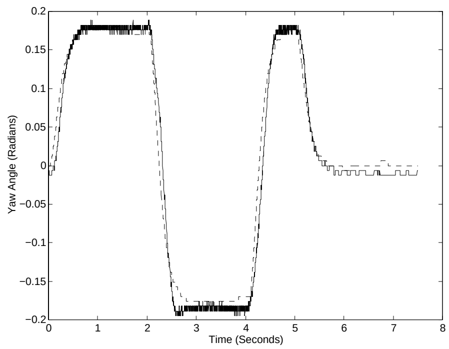


Figure 7: Closed Loop Time Response from Rear Steer Angle to Yaw Angle
(solid line = experimental; dashed line = simulated model)

The identified transfer function from rear steer command to yaw angle is given in Equation (7) where the actuator is assumed identical to Equation (6).

$$G_r(s) = \frac{120(s+15)}{s(s+20+3j)(s+20-3j)} \cdot G_{actuator}(s) \tag{7}$$

3. VEHICLE CONTROLLER DESIGN

The control design that will be used for the Driver Assisted Control (DAC) is Model Reference Control (MRC) which is a design method by which desired closed loop characteristics can be introduced into a system, i.e. a pole placement method (Astrom & Wittenmark, 1995). This is a natural formulation for DAC since the goal is to make the vehicle handle in some driver-prescribed fashion. MRC has a systematic method for the controller design. Assume the plant can be modeled as a ratio of two linear polynomials:

$$\frac{Y(s)}{U(s)} = \frac{B(s)}{A(s)}. \tag{8}$$

The polynomial A is assumed to be monic and of degree n. B can be non-monic and of degree less than or equal to n. It is also assumed that the polynomials are relatively prime, i.e. they have no common factors. The control law is given by

$$Ru(t) = Tu_c(t) - Sy(t) \tag{9}$$

where R, S, T are polynomials in the Laplace operator s. The controller consists of a feedforward term (T/R) and a feedback term (S/R). The idea behind the controller is to replace the unwanted plant dynamics with the designer's own desired dynamics.

To design the Laplace domain polynomials R, S, and T, it is necessary to take a closer look at the closed loop system in question.

$$Y(t) = \frac{BT}{AR+BS} U_c(t) \tag{10}$$

To have the closed loop system performance be identical to the desired reference model, the closed loop polynomials and the reference model must be identical.

$$\frac{BT}{AR+BS} = \frac{B_m}{A_m} \tag{11}$$

Equation (11) implies pole/zero cancellations occur between BT and AR+BS. If B is separated into stable and well damped zeros (B^+) and unstable or poorly damped zeros (B^-), B^- must be a factor of B_m. This fact also means that B^+ is canceled, so it must be a factor of AR+BS. A_m must also be a factor of AR+BS since it is the desired model characteristic polynomial. Therefore AR+BS must include A_m, B^+ and for causality conditions, the observer polynomial, A_o. Finally, since A and B are relatively prime and B^+ is a factor of B and AR+BS, B^+ must also be a factor of R. The polynomial R can then be factored into:

$$R = R'B^+ \tag{12}$$

To solve for R, S, and T, the logical steps just followed reduce the closed loop polynomial matching to:

$$AR' + B^-S = A_oA_m \tag{13}$$
$$T = A_o B_m' \tag{14}$$
$$\text{where} \quad B_m = B^- B_m'$$

As long as A and B are relatively prime, Equation (13) has a solution. For R, S, and T to be causal, the following conditions must be satisfied:

$$\begin{aligned} \text{degree}(S) &\le \text{degree}(R) \\ \text{degree}(T) &\le \text{degree}(R) \end{aligned} \tag{15}$$

Because equation (13), called the Diophantine Equation, has infinite solutions, it is simplest to implement a controller with the lowest degree possible, i.e. the minimum degree solution. This can be completed by restricting

$$\begin{gathered} \text{degree}(A_m) = \text{degree}(A) \\ \text{degree}(B_m) = \text{degree}(B) \\ \text{degree}(A_o) = \text{degree}(A) - \text{degree}(B^+) - 1 \\ B_m = B^- B_m' \end{gathered} \tag{16}$$

This design can be modified to cancel all, some, or none of the zeros by altering B_m, thus altering the MRC design. A more detailed discussion of this topic is given in Astrom and Wittenmark (1995).

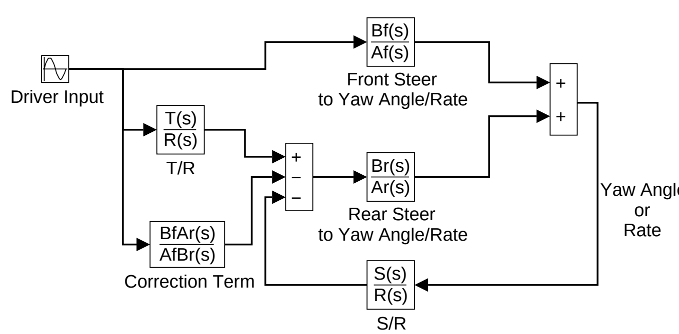


Figure 8: Rear Wheel DAC Yaw Angle/Rate MRC Implementation

The DAC implementation strategy is shown in Figure 8 for driver front steer input to desired yaw angle/rate output. The controller design has a model reference controller form, but the design has two distinct features which make it unique:

(1) the driver front steer input acts as a reference command to the MRC of the rear steer model while at the same time

(2) the driver input acts as an output disturbance (front steer model) to the MRC and is compensated for by an additional feedforward term called the Correction Term in Figure 8.

The controller design follows these steps:

(1) Choose a desired reference model and observer polynomial for front steer input to yaw angle/rate output.

(2) Use this reference model and the procedure described above to design the Laplace polynomials R, S, and T for the rear steer input model.

(3) augment the feedforward term T/R with the driver output disturbance Correction Term: $-B_f A_r / B_r A_f$.

The uniqueness of this model reference approach is that the effect of the driver's front steer input on the vehicle yaw angle/rate is formulated as an output disturbance which should be rejected by the controller governing the rear wheels. Obviously, the front steer will be aiding in the output yaw angle/rate tracking of the desired reference model for much of the time. Therefore, the controller's rear wheel steering will add only that incremental steer effort which is deemed necessary. Since the steady state gain of the yaw rate from front steer input is a function of the vehicle's understeer gradient(Ellis, 1989), the choice of both transient and steady state reference model performance completely dictates the vehicle's understeer gradient. The same can also be said for the yaw angle performance being a function of the understeer gradient.

Using the models for front and rear steer input with forward velocity U = 1.2 m/s along with a desired front steer input to yaw angle output reference model, the MRC was designed. Equations (17) to (21) give the resulting polynomials from the design.

$$R(s) = s^3 + 70s^2 + 1579s + 11310 \tag{17}$$
$$S(s) = 2.19s^3 + 147.22s^2 + 3260.92s + 23676 \tag{18}$$
$$T(s) = 3.788s^3 + 227.29s^2 + 4261.69s + 23676 \tag{19}$$
$$A_o(s) = s^2 + 50s + 625 \tag{20}$$

$$G_m(s) = \frac{4760(s+10)}{(s+8)(s+35)(s+13+j)(s+13-j)} \tag{21}$$

All polynomial transfer functions were digitally implemented as controllable canonical form state space filters on 75MHz Pentium PC's with Analog Devices RTI DAQ boards. Because of the quantization in the encoder signals, the sensed yaw angle was run through a 2nd order Butterworth lowpass filter with a 10 hertz cutoff frequency, as given in Equation (22), before being used as the input signal for the feedback transfer function [S(s)/R(s)] in Figure 8.

$$G_f = \frac{3947.841}{s^2 + 88.8576s + 3947.841} \tag{22}$$

For a square wave input, Figure 9 compares the output of the Reference Model in Equation (21) with the open loop response of the vehicle under front wheel steering; i.e. no DAC. The Reference Model was chosen to give fast, stable, transient response.

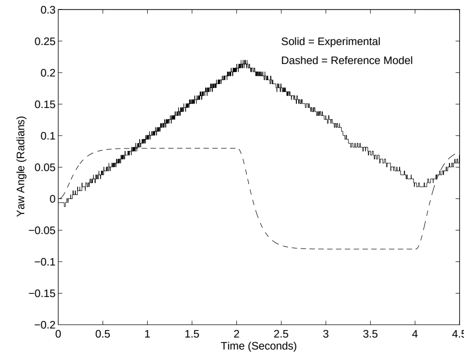


Figure 9: Reference Model vs. Driver Open Loop Yaw Angle

Figure 10 displays the results of front wheel steering with rear wheel DAC. The DAC strategy helps the driver to achieve the desired yaw angle profile. Error in the reference model following of Figure 10 can be accounted for by cumulative error from all three encoders when calculating the yaw angle and a compounding problem from steering actuator backlash. Despite these factors, the resultant yaw angle follows the desired reference model yaw angle for the given driver front steer input. The square wave driver front steer input was predetermined off-line and implemented via digital computer.

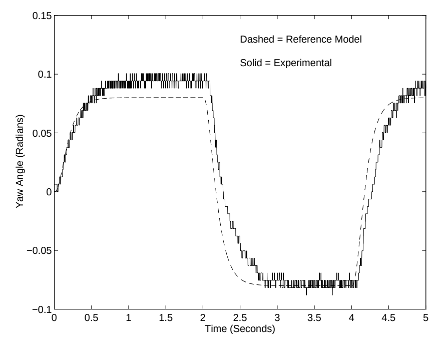


Figure 10: Reference Model vs. Experimental DAC Yaw Angle

The same controller can be used on the scaled experimental vehicle of Figure 2 with a human driver in the loop. A steering console, including a steering wheel and foot pedals, is interfaced to be part of the IRS (Brennan et al, 1998). By changing the steer angle on the driver console a subject can directly send voltages, via the intermediate computer interface, to the steer actuators. Testing completed by a human driver showed qualitatively that there was a significant difference in the feel or handling of the vehicle with DAC compared to driver open loop (DOL). Due to the reference model chosen, the vehicle 'felt' more stable and could make sudden changes in heading without feeling a loss of control.

A yaw rate DAC was attempted experimentally using the reference model given in equation (23).

$$G_m = \frac{525 \cdot (s+10)}{(s+5+5j)(s+5-5j)(s+35)} \tag{23}$$

However, before experimental implementation could be tested, the quantized yaw angle signal had to be differentiated to obtain a yaw rate feedback signal. The problem was alleviated by using a 4th order Butterworth lowpass filter with a cutoff frequency of 40 rad/sec. The low pass filter was convolved with a zero at the origin to produce a filtered yaw rate signal.

$$G_{filter} = \frac{2560000s}{s^4 + 104s^3 + 5462s^2 + 167240s + 2560000} \quad (24)$$

Figure 11 shows the experimental results in comparison to the desired reference trajectory. The plot shows that the vehicle does not track the desired yaw rate trajectory to an acceptable level. Several factors can account for this problem. The most important factor is in the filtering of the yaw angle signal to derive a filtered yaw rate. The signal is very noisy and not reliable as a method of acquiring yaw rate. The signal lags considerably due to the lowpass filter of Equation (24), but without the filter the data is too noisy to be of any use in a feedback scheme. Finding an alternative means of determining the yaw rate signal is an avenue of current research. Initial tests in which the encoder sensors are replaced with potentiometers have alleviated some of the problems associated with quantization.

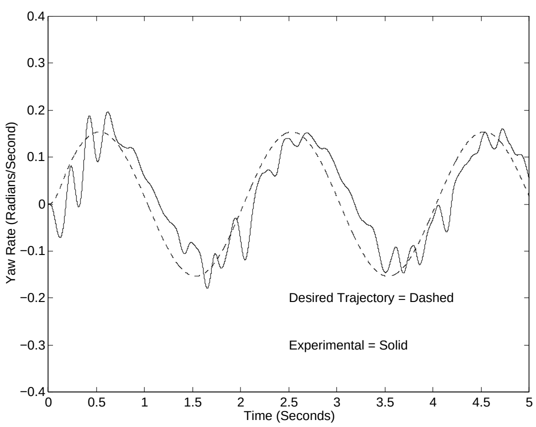


Figure 11: Reference Model vs. Experimental DAC for Yaw Rate

Implementation on an actual vehicle would necessitate the use of some type of gyroscopic device for yaw rate feedback. Additionally, the controller would have to account for the variation of the system models with forward speed. One method would be to develop several different controllers scheduled to the vehicle speed and interpolate between them. Another route would be to have the vehicle models estimated on-line. The second approach will be examined in the following.

4 ON-LINE PARAMETER ESTIMATION FOR DAC

The parameters of the bicycle model given in Section 2 can change significantly during normal vehicle operation. The most common source of the change in dynamics is the vehicle speed. In order to be able to adapt to changing vehicle parameters, particularly vehicle speed, on-line parameter estimation of the yaw dynamics was considered and experimentally tested. The method of estimation used was continuous time Recursive Least Squares (RLS) (Astrom and Wittenmark, 1995). The continuous time RLS estimator is formulated as follows. Assume the system to be estimated can be described by the model

$$A(s)y(t) = B(s)u(t) \quad (25)$$

where A(s) and B(s) are polynomials in the Laplace differential operator $s = \frac{d}{dt}$.

$$A(s) = s^n + a_1 s^{n-1} + ... + a_n \quad (26)$$

$$B(s) = b_1 s^{n-1} + ... + b_n \quad (27)$$

Because the RLS algorithm needs the derivatives of the input and output of the system, a stable filter H_f with a pole excess of n or more is used on both the input and output before the RLS uses those signals. This filtering is written as:

$$y_f(t) = H_f \cdot y(t) \quad (28)$$

$$u_f(t) = H_f \cdot u(t) \quad (29)$$

The role of H_f is to provide a causal representation of the input and output signal differentiations. The system can now be written as:

$$s^n y_f(t) = \phi^T(t)\theta \quad (30)$$

where

$$\phi^T = \left[-s^{n-1}y_f \quad \cdots \quad -y_f \quad s^{n-1}u_f \quad \cdots \quad u_f\right]^T \quad (31)$$

$$\theta = \left[a_1 \quad \cdots \quad a_n \quad b_1 \quad \cdots \quad b_n\right]^T \quad (32)$$

The vector θ is known as the *parameter* vector and ϕ is the *regressor* vector. The algorithm for continuous time RLS with exponential forgetting (Astrom & Wittenmark, 1995) is given by:

$$\frac{d\hat{\theta}(t)}{dt} = P(t)\phi(t)\left(s^n y_f(t) - \phi^T(t)\hat{\theta}(t)\right) \quad (32)$$

$$\frac{dP(t)}{dt} = \alpha P(t) - P(t)\phi(t)\phi^T(t)P(t) \quad (33)$$

where P(t) is referred to as the covariance matrix and α is called the forgetting factor. To drive the parameter estimates to their true values, a persistently exciting signal of order 2n is necessary. A summation of n sinusoidal inputs can be used for creating a sufficiently rich input signal for estimation provided the physical system actually satisfies the dynamics of Equations (26) & (27)

For the DAC approach given in Section 3, the transfer function models of Equations (5) and (7) could be used. These models, including actuator dynamics, consist of a relative degree = 3 system with one zero and four poles. However, for the purposes of implementing the RLS identification with the MRC DAC control, a 2^{nd} order model, as given below in equation (34) was used.

$$G(s) = \frac{K}{s(s+a)} \quad (34)$$

The motivation for a simpler approximation is the reduced complexity, and increased feasibility, associated with the implementation. However, the use of this reduced order model can be somewhat analytically justified from a pole/zero cancellation perspective. The previous modeling in Section 2 showed that the heavily damped pole pair was close to the zero while being between the pure integrator and the actuator pole; therefore some pole/zero cancellation occurs. Within the bandwidth relevant to this research, the cancellation results in the system approximating a pure integrator and a real pole which is slower than the actuator.

Experimental RLS was implemented on both the front and rear models to estimate the two parameters for each. The initial conditions for the RLS were: $P(0) = 1000*I$, $\alpha = 0$, $\hat{K}_f(0) = 45$, , $\hat{K}_r(0) = -45$, $\hat{a}_f(0) = 10$, $\hat{a}_r(0)=10$, respectively. The forgetting factor was set to zero so as to increase the speed of convergence. This estimation scheme was then used for a driver assisted control (DAC) strategy similar to Section 3. The procedure for the control is as follows. The vehicle was run to steady state at U = 1.2 m/s, the front wheel model was estimated, the rear wheel model was estimated, and then a controller was designed on-line and implemented for a specified driver front steer input. With the experiment still running, the speed was increased to U = 1.6 m/s and the same procedure was completed to show both parameter adaptation and acceptable DAC yaw angle tracking. The following reference model was used for the R, S, T polynomials to be formulated on-line using the estimated model parameters:

$$G_m(s) = \frac{5}{s^2 + 6s + 5}. \tag{35}$$

Figures 12 through 14 depict the results of the on-line RLS parameter estimation and DAC. The covariance matrix P(t), which is tantamount to a gain on the parameter update, decreases in an induced norm sense during the estimation. In order to counteract this, P(t) was reset to its initial value several times before the estimate was used for control in order to assure good estimates

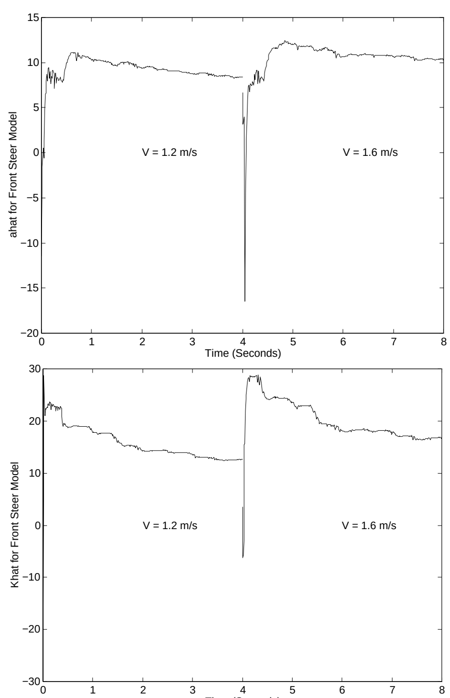


Figure 12: Front Steer Model Estimation for V = 1.2, 1.6 m/s

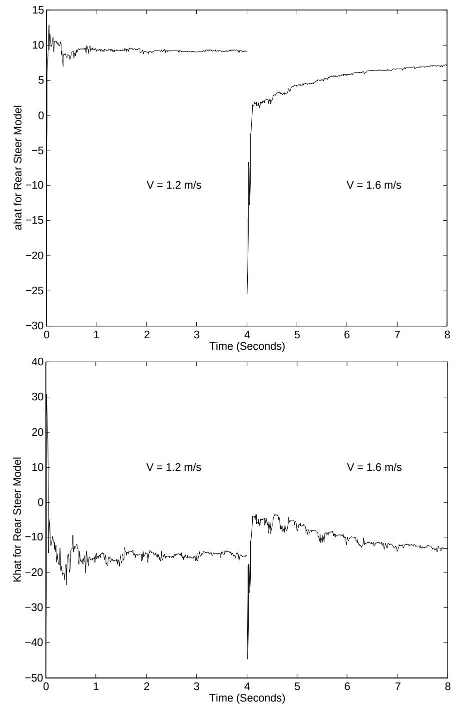


Figure 13: Rear Steer Model Estimation for V = 1.2, 1.6 m/s

More complex models (e.g. relative degree = 3) could be developed for estimation and control but would probably need to use digital control and estimation due to the need for additional derivatives in the continuous RLS algorithm. Increasing the number of poles and zeros increases the size of the filters needed to maintain causality, thereby increasing the complexity in implementation. However, the increased complexity and digital formulation may be necessary since higher vehicle speeds make the reduced order model less accurate since the pole pair would not cancel out the zero. Implementation on an actual vehicle would again use yaw rate as its output parameter, but physical limitations due to quantization of the yaw output caused this experimental work to be done with yaw angle.

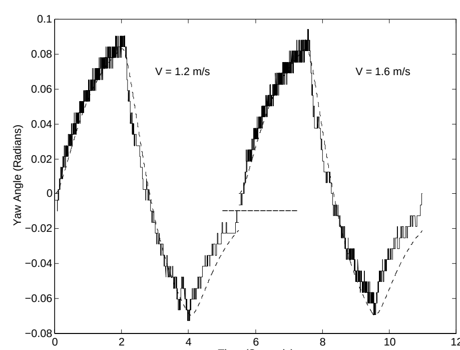


Figure 14: Model Reference Yaw Angle Tracking for DAC at U = 1.2, 1.6 m/s

CONCLUSIONS & DISCUSSION

This work has introduced a method for Driver Assisted Control based on Model Reference Control. The driver selects a desired vehicle handling model and the controller uses rear steer input to enable the vehicle to behave like the desired model. The driver maintains control of the front wheels, thus enabling a graceful degradation of performance in the event of a controller failure. The DAC control strategy was implemented on a scale testbed (IRS) and shown to be successful within sensor limitations. The unavailability of clean, accurate, yaw rate signals forced the implementation to be done on yaw angle feedback. The development of accurate yaw rate sensing is the subject of current research.

The MRC relies on an accurate model of the plant. In an effort to make the control approach feasible across a range of parameter variations, a continuous time RLS adaptive estimator was combined with the original DAC. The estimator operated on a reduced order model of the vehicle. The approach was shown to be successful for the experimental IRS system and was able to account for vehicle parameter changes due to increased speed

While some measure of success has been demonstrated in this work, there are several limitations to the present approaches that deserve mention. While the adaptive control procedure was successful in the laboratory, as shown in Section 4, it may not be feasible in practice. The proper parameter convergence for the front and rear steer transfer functions required more Persistence of Excitation than would probably be available in an actual vehicle. In a nominal driving condition which consists of relatively small driver input, the identification scheme may not obtain enough PE to get convergence to the actual parameters. Moreover, the reduced order model may not be accurate at higher speeds. Consequently, it is hereby suggested that a velocity scheduled control approach based on vehicle models identified off-line may be more realistic in practice. This avenue is also the subject of current research efforts.

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