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# A DATA COLLECTION SCHEME FOR IDENTIFICATION OF PARAMETERS IN A DRIVER MODEL

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## SUMMARY

When adapting a vehicle for use by a handicapped driver, it is often necessary to employ a high gain steering controller to compensate for limitations in the driver's range of motion. Because such a driver/vehicle system can become unstable as vehicle speed is increased, it is desirable to use a computer simulation of the driver/vehicle combination as a design tool to investigate the system response prior to construction of a controller and road testing. While there are a number of different driver models in existence, they all contain some unknown driver parameters which must be identified prior to use of the model for system analysis. This work addresses a means to collect the data necessary for identification of these driver model parameters without extensive instrumentation of a vehicle to measure and record vehicle states.

The procedure consists of three steps. First, a road test is conducted with the driver in a normal vehicle, during which only the steering wheel angle and the vehicle speed is recorded. Next, the data from the road test is input into a computer model of the vehicle which integrates the vehicle equations of motion with the given speed and steering inputs to yield the vehicle states, some of which the driver senses. Finally, with the sensed vehicle states as inputs and the recorded steering response as output, a least squares parameter identification procedure is used to compute the parameters in the proposed driver model.

Initial tests of the procedure identified all of the driver parameters with errors of 6% or less.



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## INTRODUCTION

The Rehabilitation Engineering Program at Texas A&M University is currently involved in the evaluation and design of low effort - high gain automotive control devices for handicapped drivers. Experience has shown that some vehicles with high gain steering are relatively easy to control at moderate speeds and others require maximum driver effort to maintain control at very low speeds. In order to better understand the causes of this wide variation in handling proterties and to quantify the effects of changes in various steering system parameters on vehicle response, development of a computer simulation of the driver-vehicle combination was begun.

At present, there are a number of vehicle models [3] and driver models [1,5,6] that are available. Use of a typical vehicle simulation requires knowledge of the geometry, inertial properties, and tire characteristics of the vehicle to be studied. Most of these can be obtained by direct measurement or are easily estimated.

As with the vehicle models, most of the driver models were found to have several coefficients whose magnitude is dependent on the characteristics of the driver or his environment. In order to use a driver model, a means to quickly and inexpensively identify these unknown parameters in a driver model is required. Classically, this identification is accomplished by running a road test and recording the driver inputs and all of the vehicle motion variables that the driver may respond to. As illustrated in Figure 1a, the vehicle motion data is used as an input for the driver model and the driver response data is compared to the results of the driver model to generate an error. There are a number of parameter identification techniques available [4] to determine the coefficients in the driver model that will minimize the error in predicted and measured driver response. The difficulty in employing this procedure lies in the instrumentation required to record the vehicle motion variables. Variables such as heading angle or yaw rate require a gyroscopic device which is expensive and bulky. The position of the vehicle on the road may be obtained with an optical tracking device or by integrating the output of accelerometers on an inertial platform. Either of these methods is expensive and the equipment is not easily moved from one vehicle to another.

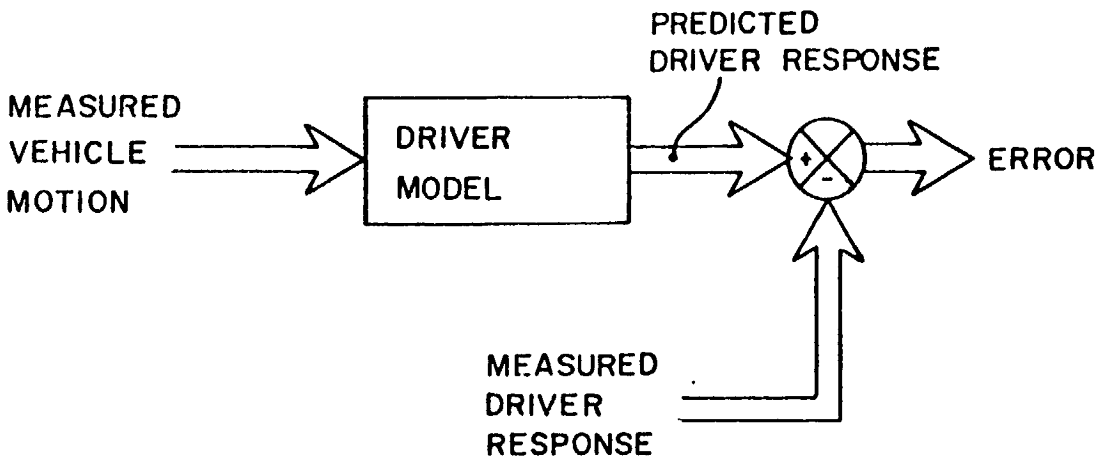


Figure 1a. Typical Parameter Identification Procedure

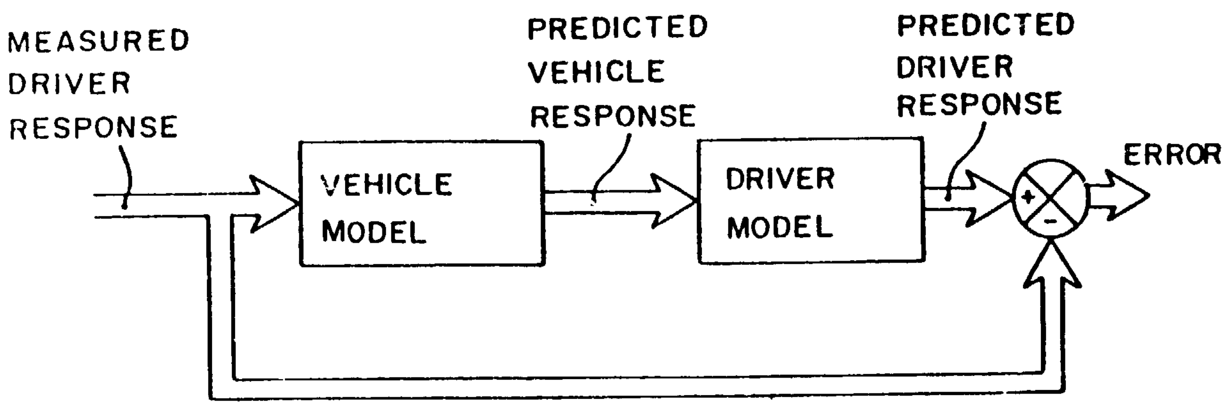


Figure 1b. Proposed Procedure

In order to minimize the instrumentation necessary to obtain the required data, an alternate procedure was examined. As illustrated in Figure 1b, the only measured variables are driver inputs (steering wheel angle and vehicle forward velocity). These inputs are then used with a vehicle simulation to determine what the motion must have been during the test. In this manner the required data for driver model identification is obtained with a minimum of instrumentation.

## VEHICLE MODEL

The vehicle model used in this procedure was chosen because it was considered to be the simplest model available that exhibited the handling properties under study. Figure 2a is an illustration of the vehicle model. This vehicle has three degrees of freedom including the forward and lateral position of the mass center P, ($r_{PX}$ and $r_{PY}$) and the heading angle ($\psi$). Two coordinate systems are employed. The X, Y, Z system is fixed to the roadway and has the associated unit vectors $\hat{I}$, $\hat{J}$, and $\hat{K}$. The x, y, z system is fixed to the car and has unit vectors $\hat{i}$, $\hat{j}$, and $\hat{k}$.

Using these definitions, the acceleration of the mass center, P, may be shown to be

$$\bar{a}_P = [\ddot{r}_{PX} \cos\psi + \ddot{r}_{PY} \sin\psi]\,\hat{i} + [\ddot{r}_{PY} \cos\psi - \ddot{r}_{PX} \sin\psi]\,\hat{j} \qquad (1)$$

The free body diagram for the vehicle is illustrated in Figure 2b. In order to simplify the analysis, secondary forces such as tire rolling resistance, self aligning torque, aerodynamic drag, and gyroscopic moment are considered negligible. Applying Newton's Laws to the free body diagram results in the following equations of motion.

$$F_{fL} \cos\delta_L + F_{fR} \cos\delta_R + F_{RL} + F_{RR} = m\ddot{r}_{PY} \cos\psi - m\ddot{r}_{PX} \sin\psi \qquad (2)$$

$$-a_5(F_{RL} + F_{RR}) + a_1(F_{fL} \cos\delta_L + F_{fR} \cos\delta_R) - (a_2 + a_4)\, F_{fL} \sin\delta_L$$

$$+ (a_2 - a_4)\, F_{fR} \sin\delta_R = I\ddot{\psi} \qquad (3)$$

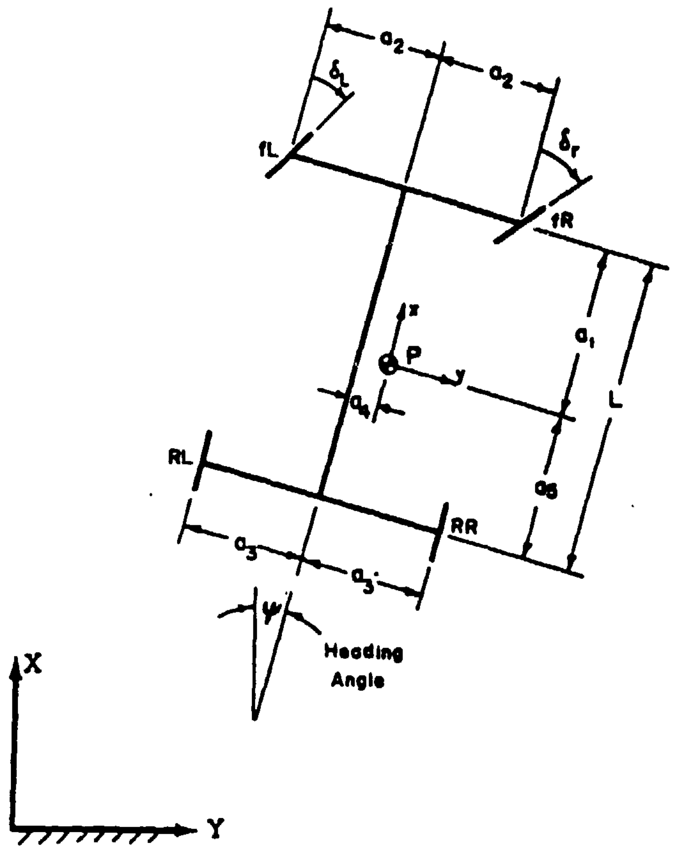


Figure 2a. Vehicle Model

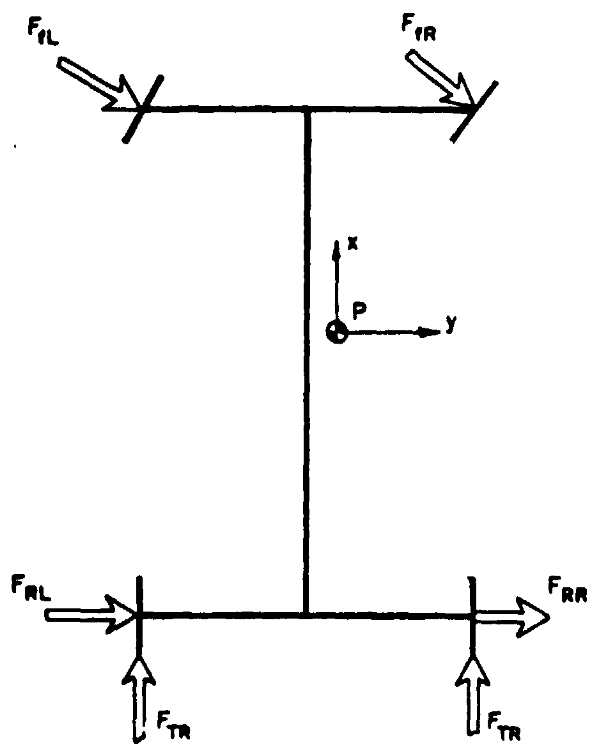


Figure 2b. Vehicle Free Body Diagram

A third equation may be obtained by recalling that the vehicle forward velocity is known. The x component of $\bar{a}_p$, therefore, may be expressed as

$$\ddot{r}_{PX} \cos \psi + \ddot{r}_{PY} \sin \psi = \dot{V} \tag{4}$$

Equations (2), (3), and (4) represent the three equations of motion for the vehicle. Unfortunately, the tire forces ($F_{fL}$, $F_{fR}$, $F_{RL}$, $F_{RR}$) are unknown and must be related to the vehicle motion parameters. It is well documented [2] that a rolling pneumatic tire under the influence of a lateral load exhibits a viscoelastic deformation of the tread surface. This deformation results in a deviation of the wheel center velocity from its expected direction. This deviation is termed the slip angle and may be expressed in terms of the vehicle motion variables. Using the defined coordinates, ($r_{PX}$, $r_{PY}$, and $\psi$) the slip angle at each wheel may be shown to be:

$$\alpha_{fL} = \delta_L - \tan^{-1}\left[\frac{\dot{r}_{PY} \cos \psi - \dot{r}_{PX} \sin \psi + a_1\dot{\psi}}{V + (a_2 + a_4)\dot{\psi}}\right] \tag{5}$$

$$\alpha_{fL} = \delta_R - \tan^{-1}\left[\frac{\dot{r}_{PY} \cos \psi - \dot{r}_{PX} \sin \psi + a_1\dot{\psi}}{V - (a_2 - a_4)\dot{\psi}}\right] \tag{6}$$

$$\alpha_{RL} = - \tan^{-1}\left[\frac{\dot{r}_{PY} \cos \psi - \dot{r}_{PX} \sin \psi - a_5\dot{\psi}}{V + (a_3 + a_4)\dot{\psi}}\right] \tag{7}$$

$$\alpha_{RR} = - \tan^{-1}\left[\frac{\dot{r}_{PY} \cos \psi - \dot{r}_{PX} \sin \psi - a_5\dot{\psi}}{V - (a_3 - a_4)\dot{\psi}}\right] \tag{8}$$

where $\delta_L$ and $\delta_R$ are left and right steering angles as defined in Figure 2a.

Once the slip angle at a given wheel is specified, the tire lateral force may be determined. For this work, the tire used is a Goodyear FR70-14. Figure 3 illustrates the relationship between slip angle and lateral force for various normal loads on the tread surface.

Given the equations of motion (2-4), the slip angle expressions (5-8) and the tire force function illustrated in Figure 3, the motion of the vehicle may be determined if the forward velocity V, and the front wheel angles, $\delta_L$ and $\delta_R$ are known. Assuming Ackermann Steering, $\delta_L$ and $\delta_R$ may be related to the steering wheel angle by the following relationships

$$x = \frac{L}{\tan (K_{ST}\delta)} \tag{9}$$

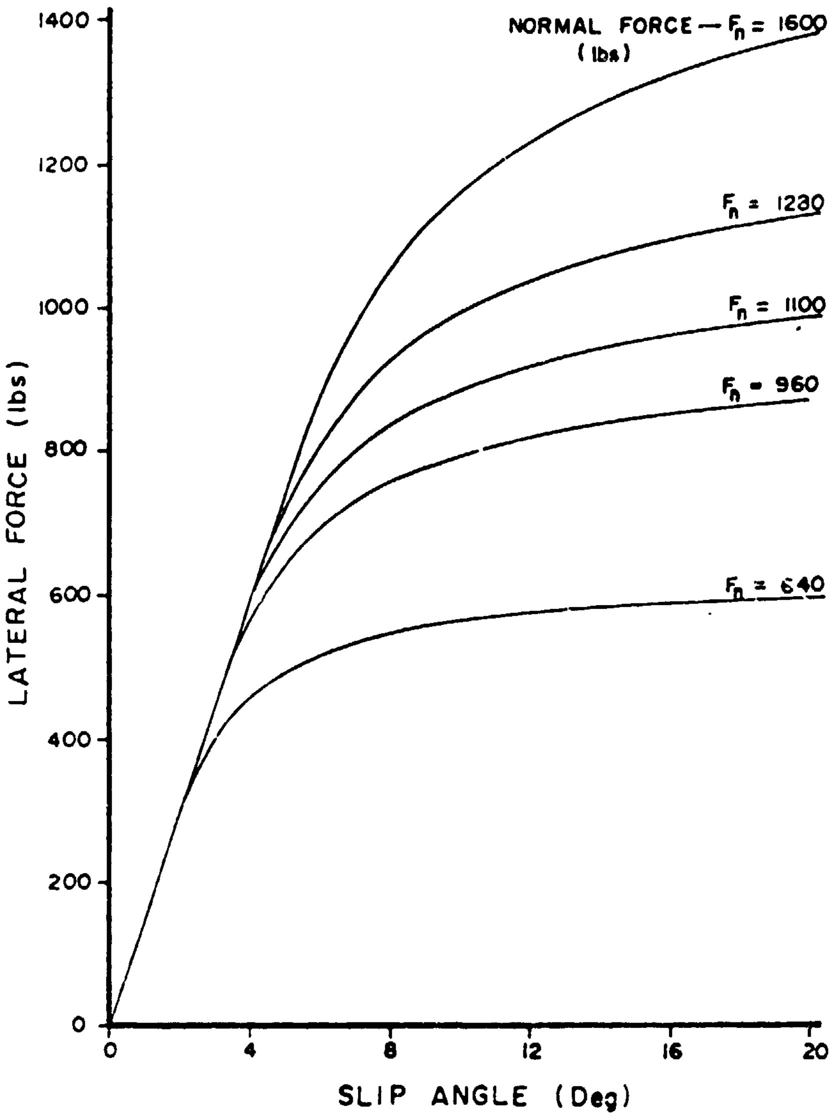


**Figure 3. Tire Lateral Force vs. Slip Angle**

$$\delta_R = \tan^{-1} \frac{L}{(x - a_2)} \tag{10}$$

$$\delta_L = \tan^{-1} \frac{L}{(x - a_2)} \tag{11}$$

where $K_{ST}$ represents the steering gain, $\delta$ is steering wheel angle, and L is wheelbase.

Now, given the driver inputs of steering wheel angle, $\delta$, and forward velocity, V, the motion of the vehicle may be determined. Because of the non-linearity of th .otion and constraint equations, the most expedient means of obtaining a solution is by utilizing a numerical integration procedure on a digital computer. The computer that was used in the following example was a PRIME 750 minicomputer. The program was written in FORTRAN and utilized a predictor-corrector numerical integration scheme.

## VALIDATION OF VEHICLE MODEL

In order to verify the accuracy of the vehicle simulation, a series of test runs was made with a vehicle that was instrumented to record several of the vehicle motion parameters as well as the driver responses. The driver responses were used as input data for the vehicle simulation The results of the vehicle simulation were then compared to the vehicle motions recorded during the test. Several test runs were made at different vehicle speeds and over different courses. Figure 4 illustrates the comparison between predicted and measured heading angle, yaw rate, and lateral acceleration as the vehicle passed through an offset alley. As shown in Figure 4, the vehicle simulation provides a good estimation of the vehicle motion. The large difference in measured and predicted heading angle is due to a difference in reference position. The simulation automatically sets the vehicle's initial heading to zero degrees while the measured heading depends on the vehicle orientation when the gyroscope unit is switched on. When this bias is removed, the results compare quite well.

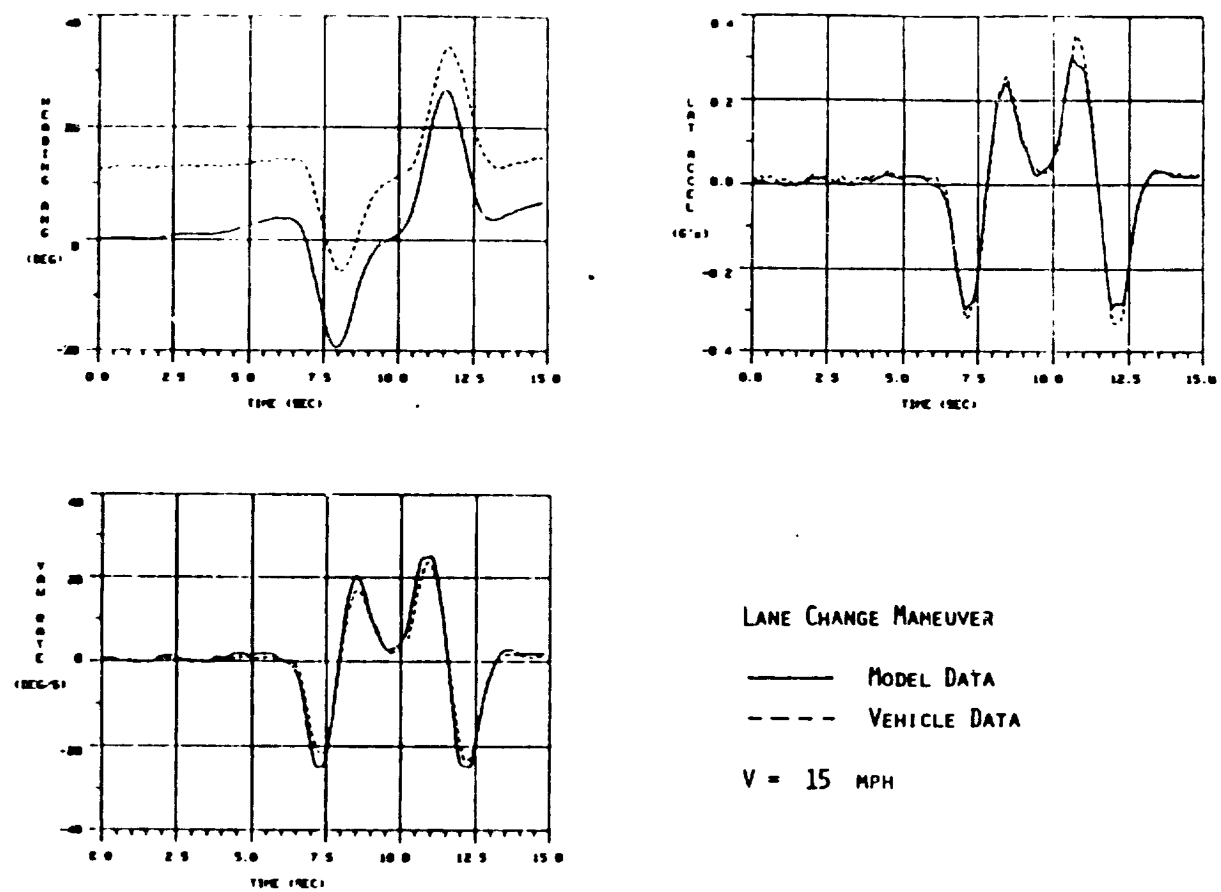


**Figure 4. Comparison of Measured and Simulated Vehicle Motion**

## VALIDATION OF IDENTIFICATION PROCEDURE

The driver model used to validate the parameter identification procedure assumes that the driver can be represented as a two part cascade of a brain response and a neuromuscular lag. Reaction time delay and precognition or preview are not included since they tend to cancel and, at any rate, the particular form of the driver model is not critical to the identification procedure at this stage. The inputs to the driver model are the lane position error, $\varepsilon_y$, and the handing angle error, $\varepsilon_\psi$. For a straight roadway $\varepsilon_y = r_{py}$ and $\varepsilon_\psi = \psi$. The output of the brain is the commanded steering wheel angle $\delta'$ and is modeled as

$$\delta' = K\,(r_{py} + \eta\psi) \tag{12}$$

where K is the brain gain and $\eta$ is a weighting factor. The brain output is the input to the neuromuscular system which is modeled as

$$\dot{\delta} = \frac{1}{\tau}\,(\delta' - \delta) \tag{13}$$

where $\tau$ is the neuromuscular time constant. Thus, the parameter identification procedure must compute values of K, $\eta$, and $\tau$ for which the computer model output best fits the measured data in a least squares sense.

Since the driver parameters K, $\eta$, and $\tau$ are not available from an actual driver/vehicle test a computer simulation was used to generate test data that allows direct comparison of computed parameter values to true parameter values. The test data was generated using the same driver model as the identification code, but a different vehicle model. The vehicle model used was more complete and had been thoroughly checked out.

The test maneuver was lane tracking on a straight road at a nominal speed of 55 mph with an initial lane error of five feet. As shown in Table 1 the largest error between the true and computed parameter values is 6%.

| Parameter | True Value | Identified Value | Percent Error |
|---|---|---|---|
| $\tau$ | 0.1 | 0.100 | 0% |
| K | -0.1 | -0.094 | -6% |
| $\eta$ | 100 | 94.40 | -5.6% |

Table 1 Errors in Identified Values of the Driver Model Parameters

## CONCLUSIONS

As shown in Figure 4, the computer model provides a very accurate simulation of the actual vehicle states. This is a critical result, since the data collection scheme proposed in this paper is based on the premise that vehicle states, which are inputs to the driver, can be accurately constructed from simulation, thus eliminating the need to instrument each vehicle to be tested with an inertial platform. Also, assuming that an appropriate driver model is used, the identification procedure accurately computes the driver model parameters as indicated by the data in Table 1. Thus, each part of the overall procedure has been independently validated.

The final test of the procedure will be identification of driver model parameters from an actual road test and comparison of computed vehicle states and driver outputs to actual measured values. This testing is currently in progress.

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