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Development of Driver/Vehicle Steering Interaction Models for Dynamic Analysis

Interim Technical Report

for

The U. S. Army Tank Automotive Command

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16. Abstract

This report summarizes the results and findings of the work conducted during the first period (July 1985 to June 1986) of the research project. Activities and accomplishments to date have included: 1) construction of a generalized set of directional vehicle dynamics to be used by the driver steering models developed under the project, 2) development and analysis of three basic methods for steering and controlling single-unit and powered-articulated vehicles, 3) usage of the DADS simulation program for installing the developed driver models within the DADS / TACOM software, and 4) completion of a preliminary test plan for conducting vehicle/driver tests during the Phase II project period. Results of the first year's work have shown basic agreement with previous simulation and experimental experience of driver/vehicle systems undergoing path tracking maneuvers. Further, the newly developed models and concepts, which are applicable to tracked and powered-articulated vehicles, appear to offer interesting alternatives for extending the present driver control methods to these systems.

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steering, driver, control, driver/vehicle, models, closed-loop, path-following, vehicle dynamics

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1.0 Overview of Phase I Work (July 1985 - June 1986)

The primary project activities to date have included: 1) construction of a generalized set of directional vehicle dynamics to be used by the driver steering models developed within this project, 2) development and analysis of three basic methods for steering and controlling single-unit and powered-articulated vehicles, 3) usage of the DADS program to gain familiarity for installing the developed steering models into the DADS software, and 4) completion of a preliminary test plan for conducting the Phase II vehicle/driver tests.

The generalized set of vehicle dynamics developed to date is intended to act as an *internal* representation of the vehicle dynamics within the driver model itself for the purpose of predicting / estimating future vehicle position. The specific internal (driver-based) vehicle dynamics is intended to permit representation of three basic vehicle types: a) single-unit steered wheel vehicles (e.g., the HMMWV), b) single-unit tracked vehicles, and c) articulated vehicles having both steered wheels and/or powered articulation joints (e.g., the LVS). At present, the set of internal vehicle dynamics is linear and time invariant. For the single-unit model, two degrees of freedom are represented: yaw and lateral displacement. For the articulated model, three degrees of freedom are included: yaw and lateral displacement of the lead unit, and yaw (or constrained translation) of the following unit. Future additions to the degrees of freedom for the internal model will include roll motion and influence of speed change (braking).

Three different methods for steering the subject vehicles were examined. Each method is a variation on the UMTRI driver model [1] used previously for representing closed-loop steering of conventional automobiles and trucks. The first method adapts this model to vehicles equipped with both *front and / or rear wheel steering*. This method can be applied to vehicles having such characteristics, or, to vehicles having front wheel steering and a light articulating rear frame which, in effect, approximates steering of the rear axle.

The second method utilizes a pure *yaw torque* applied to the vehicle as a steering control. This is primarily intended to represent, to first order, means by which tracked vehicles are steered. That is, the vehicle is first yawed by the control torque (accomplished by means of differential track speeds), and subsequently translates laterally in response to the resulting vehicle sideslip forces acting on the tracks. This method is also applied to the powered-articulated vehicle class which may also be accompanied by steered wheels.

The third and most general method utilizes a *lateral control force* applied at an arbitrary point ahead of the neutral steer point (fore/aft center of lateral ground forces) of the vehicle. Application of such a control force produces a yaw moment as well as a lateral force acting on the vehicle, similar to the effect produced through use of steered wheels. In fact, an equivalence between the two aforementioned methods and this method can be shown to exist for appropriate fore/aft points of application of the control force and its gain value. For example, application of a small lateral force at a very large forward distance approximates the pure moment method; whereas, application of a sizeable lateral force at the neutral steer point duplicates a purely lateral force method. Specification of the gain of the control force as well as its point of application between these two extremes permits any mixture of side force and yaw moment controls to be duplicated. As a result of this greater generality, the lateral force control method has been included to permit a wide range of flexibility for representing and visualizing possible internal vehicle systems utilized by drivers. Preliminary results to date suggest that for the different types of vehicle systems examined, and for which closed-loop directional control is implemented by steering control actions/strategies of 'typical' drivers (regardless of how the control forces/moments are actually applied to the vehicle), such systems can be modelled adequately by means of a single concept --- a lateral resultant control force acting at some fore/aft point (possibly variable) along the vehicle axis. The simplicity of this concept helps to facilitate not only the mathematical representation and analysis of the resulting closed-loop driver control system, but how it may be visualized and explained as well. This concept may become more important for modelling the internal vehicle dynamics used by the driver model on more complex vehicle systems, particularily those for which no simplified set of directional dynamics are readily available.

The other major activity to date has been familiarizing UMTRI personnel with the DADS program for the purpose of installing the developed driver models into the DADS code. So far, an example of the developed driver model has been installed into the DADS - 2D (planar motion version) and the results compare quite favorably with results obtained previously by UMTRI using its own models. Installation of the same models into the DADS - 3D (full motion version) has been delayed until the first portion of Phase II. The HMMWV vehicle will serve as the baseline single-unit vehicle for testing and evaluating the installed driver models in that DADS version.

Finally, a preliminary plan for conducting full-scale vehicle/driver tests was developed during the latter stages of Phase I. The primary vehicle tests consist of driver-controlled maneuvers involving curve negotiation and lane-changes using the HMMWV vehicle. Because of a re-scheduling of the Phase II testing from the late summer of this year until the spring of 1987, the test plan will remain subject to modification for the remainder of this year in the event alternate ideas regarding the testing activities are offered.

The following sections of the report provide further background and details related to the items discussed briefly in the overview. Section 2.0 covers the development and implementation of the three control schemes using the UMTRI driver model as a starting point. Results are shown for various parameter variations. Section 3.0 covers the DADS work to date, including the code installed into the DADS program for the 2-D version. Finally, Section 4.0 presents the preliminary test plan for next year's driver /vehicle test program.

2.0 Steering Model Implementation

The equations appearing in the UMTRI driver model [1] were modified to provide for vehicles controlled by: a) front and rear wheel steering, b) application of a pure yaw moment, and c) a mixture of an applied lateral force and a yaw moment. For the case of a single-unit vehicle having both front and rear wheel steering , the vehicle dynamics equations are shown as follows:

$$y' = v + U \psi \tag{1}$$

$$v' = [\,-2(C_{af} + C_{ar}) / m\,U\,]\,v + [\,2(b\,C_{ar} - a\,C_{af}) / m\,U - U\,]\,r + (2\,C_{af} / m)\,\partial_{fw} + (2\,C_{ar} / m)\,\partial_{rw} \tag{2}$$

$$r' = [\,2(b\,C_{ar} - a\,C_{af}) / I\,U\,]\,v + [\,-2(a^2 C_{af} + b^2 C_{ar}) / I\,U\,]\,r + (2\,a\,C_{af} / I)\,\partial_{fw} - (2\,b\,C_{ar} / I)\,\partial_{rw} \tag{3}$$

$$\psi' = r \tag{4}$$

where,

' denotes differentiation with respect to time

y is the inertial lateral displacement of the vehicle mass center

v is the lateral velocity in the vehicle body axis system

r is the yaw rate about the vertical body axis

ψ is the vehicle heading (yaw) angle

∂_{fw} is the front tire steer angle, control variable

∂_{rw} is the rear tire steer angle, control variable

and the paramters appearing in equations (1) -> (4) are:

U forward vehicle velocity

$C_{af, ar}$ front and rear tire cornering stiffnesses

a, b forward and rearward locations of tires from the vehicle mass center

m, I vehicle mass and yaw inertia

These equations can be extended to represent typical rear wheel steering system implementations in which the rear wheels are proportionately slaved to the front wheel angle by a gain constant, k :

$$\partial_{rw} = k\,\partial_{fw} \tag{5}$$

By also adding lateral force and yaw moment control terms, the equations (1) -> (4) can be written in a somewhat more general form as:

$$y' = v + U\psi \tag{6}$$

$$\begin{aligned} v' = {} & [-2(C_{af} + C_{ar}) / m\,U]\,v + [2(b\,C_{ar} - a\,C_{af}) / m\,U - U]\,r \\ & + \{A\,[2(C_{af} + k\,C_{ar}) / m] + B / m\}\,u_c \end{aligned} \tag{7}$$

$$r' = [\,2(b\,C_{ar} - a\,C_{af})\,/\,I\,U\,]\,v + [\,-2(a^2\,C_{af} + b^2\,C_{ar})\,/\,I\,U\,]\,r$$

$$+ \{\,C\,[\,2(a\,C_{af} - k\,b\,C_{ar})\,/\,I\,] + D\,/\,I\,\}\,u_c \qquad (8)$$

$$\psi' = r \qquad (9)$$

where now,

u_c takes on the role of the general purpose control variable (u_c may be interpreted as either front wheel steer angle, lateral control force, or yaw moment control, depending upon the values of the control parameters A, B, C, and D).

and,

A, B, C, and D are control coefficients specified to allow various types of control schemes to be represented. For example, by specifying A and C as 1.0 and k = B = D = 0, the conventional front wheel steered vehicle is represented with u_c interpreted as the front wheel steer angle control variable. Or, by specifying A = B = C = 0 and D as 1.0, a vehicle controlled by a pure yaw moment control is represented with u_c now interpreted as the yaw moment control variable. To represent a lateral force control scheme, A and C are selected as 0, B = 1.0, and D is the distance forward of the vehicle mass center at which the lateral control force, u_c, acts.

These equations, (6) -> (9), represent the internal set of vehicle dynamics utilized by the driver model for the category of single-unit vehicles. For the case of an articulated vehicle, a similar set of equations result and are shown in Appendix A. The equations in Appendix A are for a linear, constant velocity articulated vehicle having front steerable wheels as well as an articulation joint torque as control variables --- similar in concept to a simplified LVS.

Figures 1 - 3 depict the three control schemes for a single-unit vehicle. Fy_{FW} and Fy_{RW} are the front and rear lateral tire forces; M_z is the applied control torque in Figure 2. In Figure 3 the $F_{Control}$ force is applied at a distance "c'" ahead of the mass center. The neutral steer point "ns" seen in Figure 3 lies a distance, d', ahead of the mass center.

Figure 1. Vehicle with front- and rear-steered wheels.

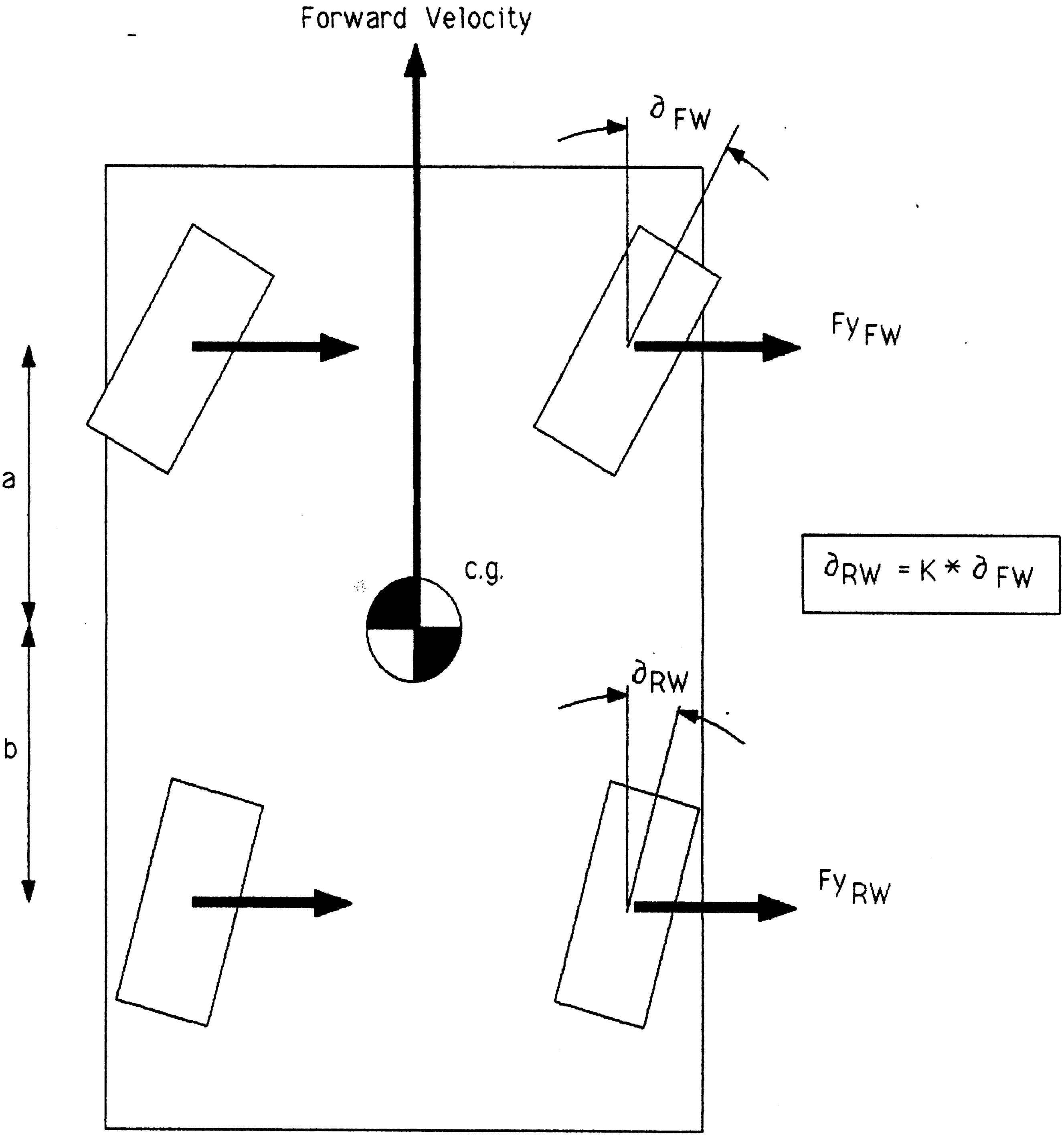


Fig 2. Moment Control Method

(Mz is calculated control moment)

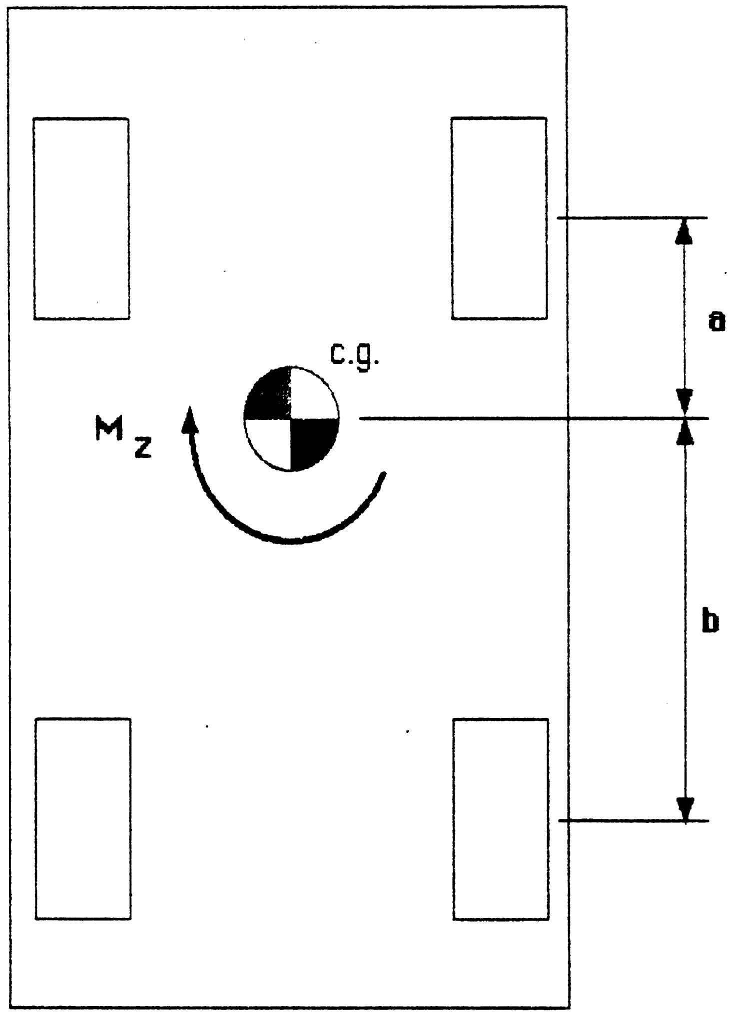
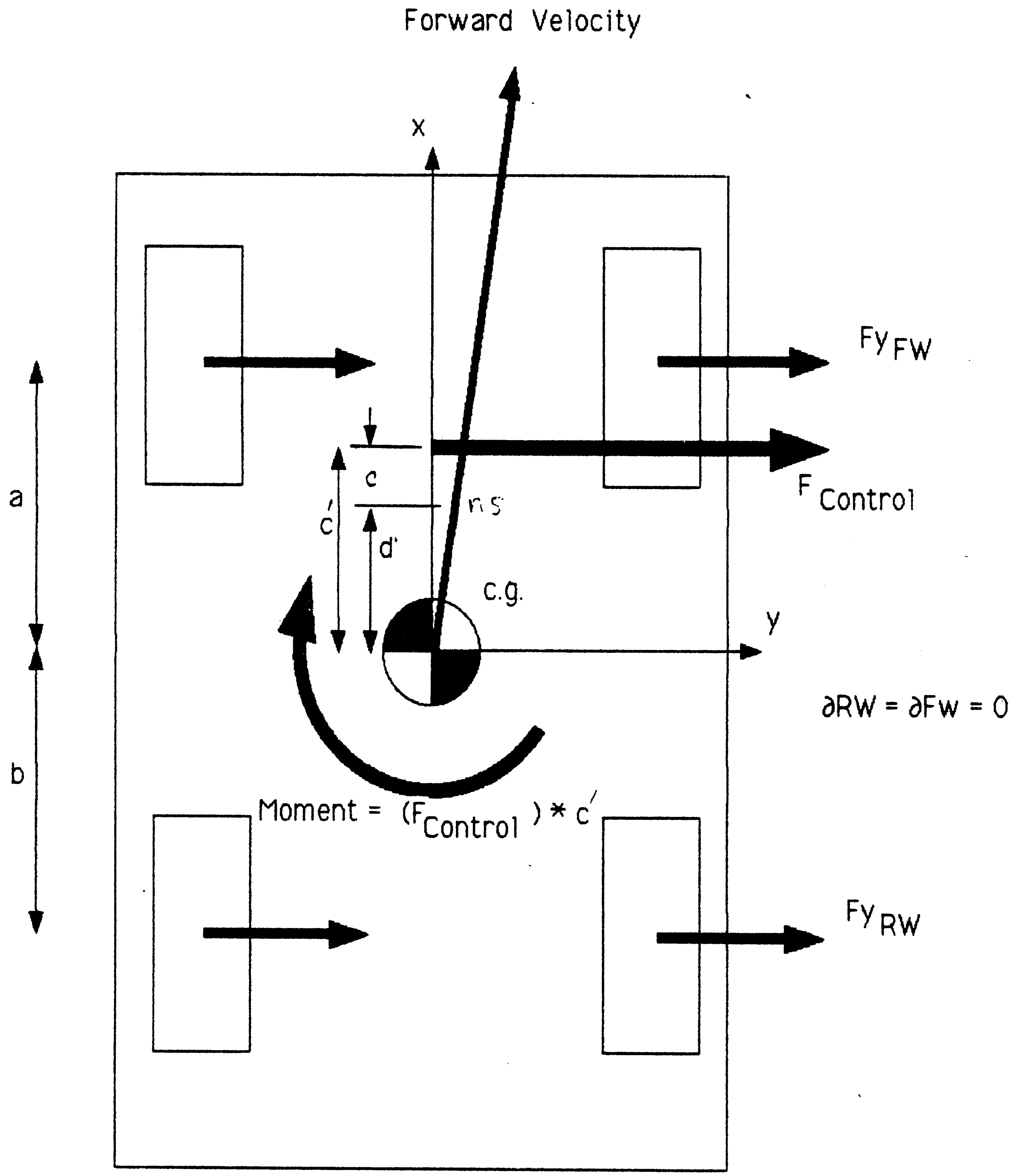


Figure 3. Vehicle Force/Moment Steer Control.



$F_{Control}$ is the calculated steering control applied

to the vehicle at a distance c' ahead of the c.g.

A number of analyses were performed using the developed control schemes to study the relative stability and controllability of each control concept using a baseline vehicle. The mass and geometric properties of the HMMWV vehicle served as the baseline configuration for these analyses since it is expected to be selected as the test vehicle in the first series of tests. Table 1 lists estimates of the mass and geometric parameters used in the HMMWV analysis. Figure 4 shows the cornering stiffness values used in the analysis, as well as some earlier UMTRI measurements on a HMMWV tire.

Three basic types of analyses were performed for each control concept: 1) root loci diagrams which were numerically calculated from the known dynamics of the closed-loop system, 2) frequency domain analyses obtained by using a random-like disturbance input to the front wheels while operating in a straight-line closed-loop driving mode (resulting in Bode plots and Nichols charts), and 3) time domain results for a 12-foot lane change maneuver. In each analysis, various key parameters related to the control schemes were varied in a systematic fashion. For example, in the front and rear wheel steering case, the rear wheel steering gain factor, k, was varied to examine its influence on the relative stability of the baseline driver/vehicle (closed-loop) system. Results for this variation are seen in Figure 5, where k is changing from -0.5 to +0.95. Values of k greater than 1.0 produced an unstable system response.

Similarly in Figure 6, a root locus diagram is shown for the force control method. The parameter "c" (distance ahead of the neutral steer point) is being varied from a distance of 4 feet to a distance of 0.1. As seen, if the point of application of the lateral control force approaches the neutral steer point, the stability of the system is diminished. At large values of "c" the low frequency mode is lightly damped and improves as the value is decreased.

Figures 7a and 7b show results from the disturbance input simulation test conducted for a 25 second time interval. A sum of low frequency sinusoids was applied in parallel with the driver steering model as a disturbance to the front wheel steering angle during straight-line driving. The task of the driver model was to keep the vehicle, as best it could, moving on a specified straight-line course. The maximum disturbance to the system by the random steer input is small and amounts to about plus and minus 0.1 g's of lateral acceleration. By knowing what the disturbance input is and then recording the simulated driver steering response, the closed-loop frequency response of the total driver/vehicle system can be subsequently calculated [2 , 3]. Figure 7a shows the resulting frequency response plot for the driver/vehicle system. By plotting these same results in the form of a Nichols chart, Figure 7b,

Table 1. Linear Model Parameter Estimates for the HMMWV Vehicle

Parameter	Symbol	Value	Units
Front tire cornering stiffness	Caf	360	lb/deg
Rear tire cornering stiffness	Car	226	lb/deg
Vehicle weight	W	5860	lb
Front axle to c.g.	a	5.15	ft
Rear axle to c.g.	b	8.18	ft
Yaw inertia	RI	5800	ft^2 -slug

All analyses shown to date were performed for a travel speed of 60 mph.

Driver model preview time: 1.5 sec

Driver model transport lag: 0.25 sec

Figure 4. HMMWV Tire Cornering Stiffness vs. Vertical Load

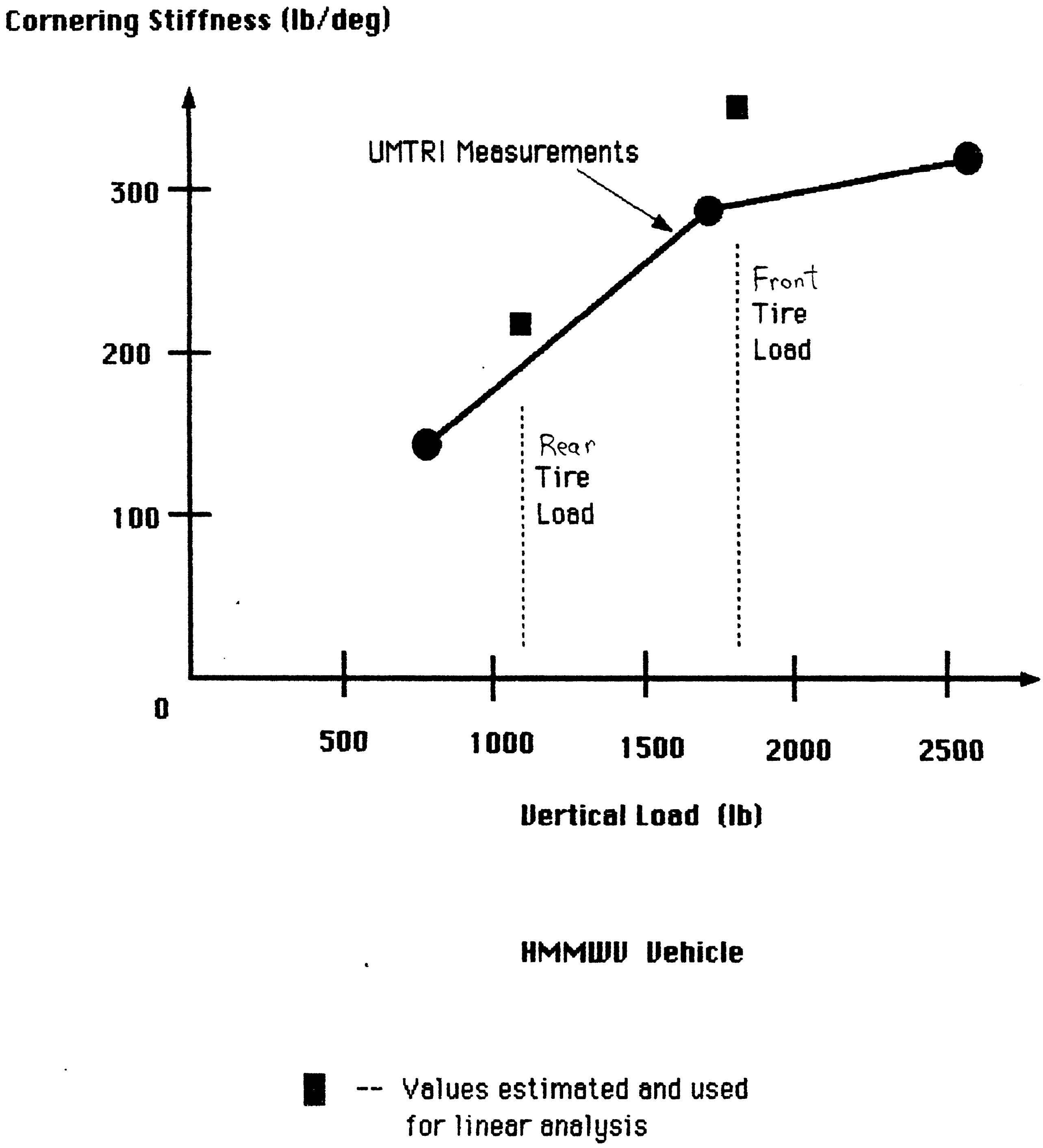


Figure 5. Root Locus

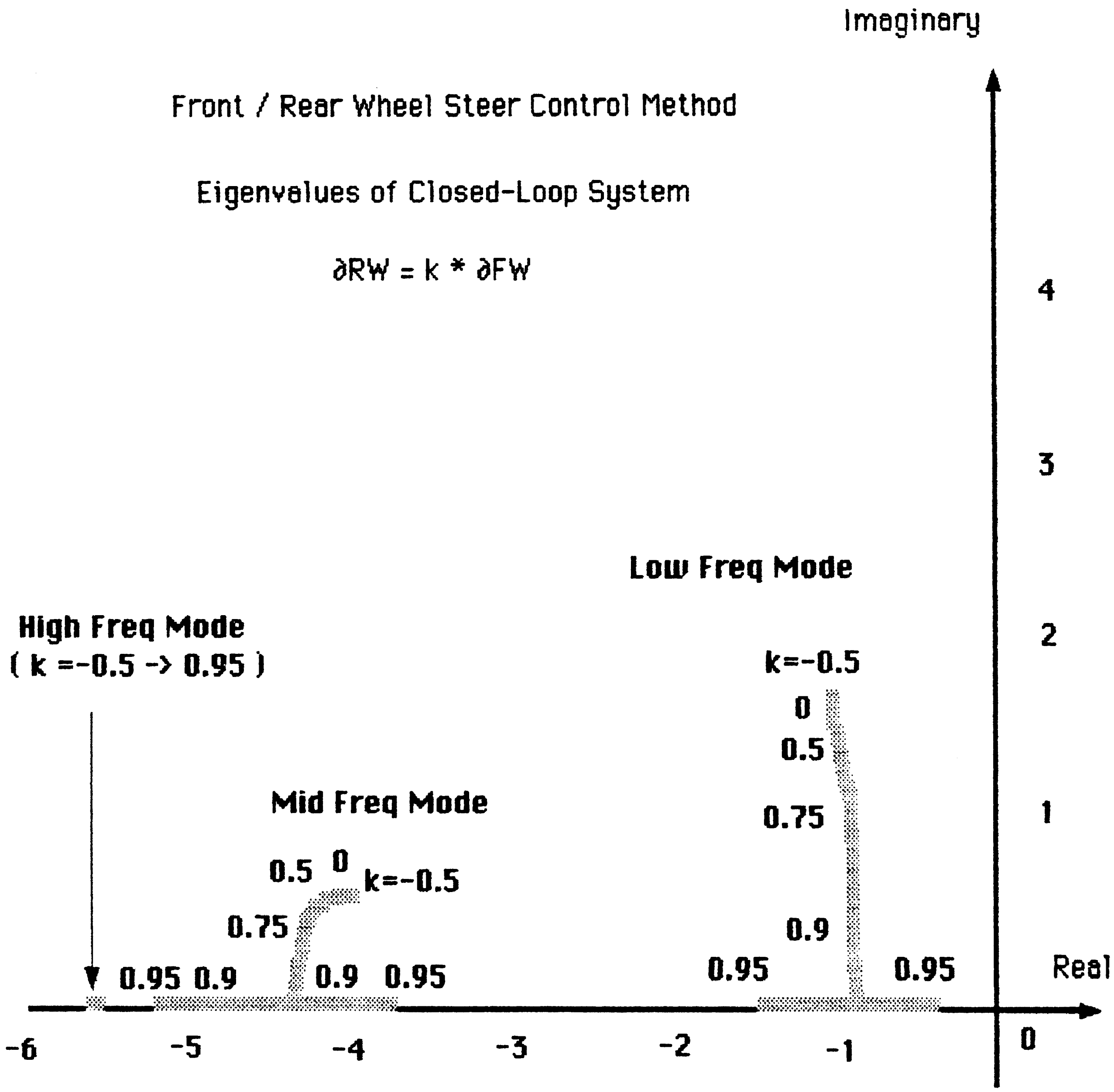
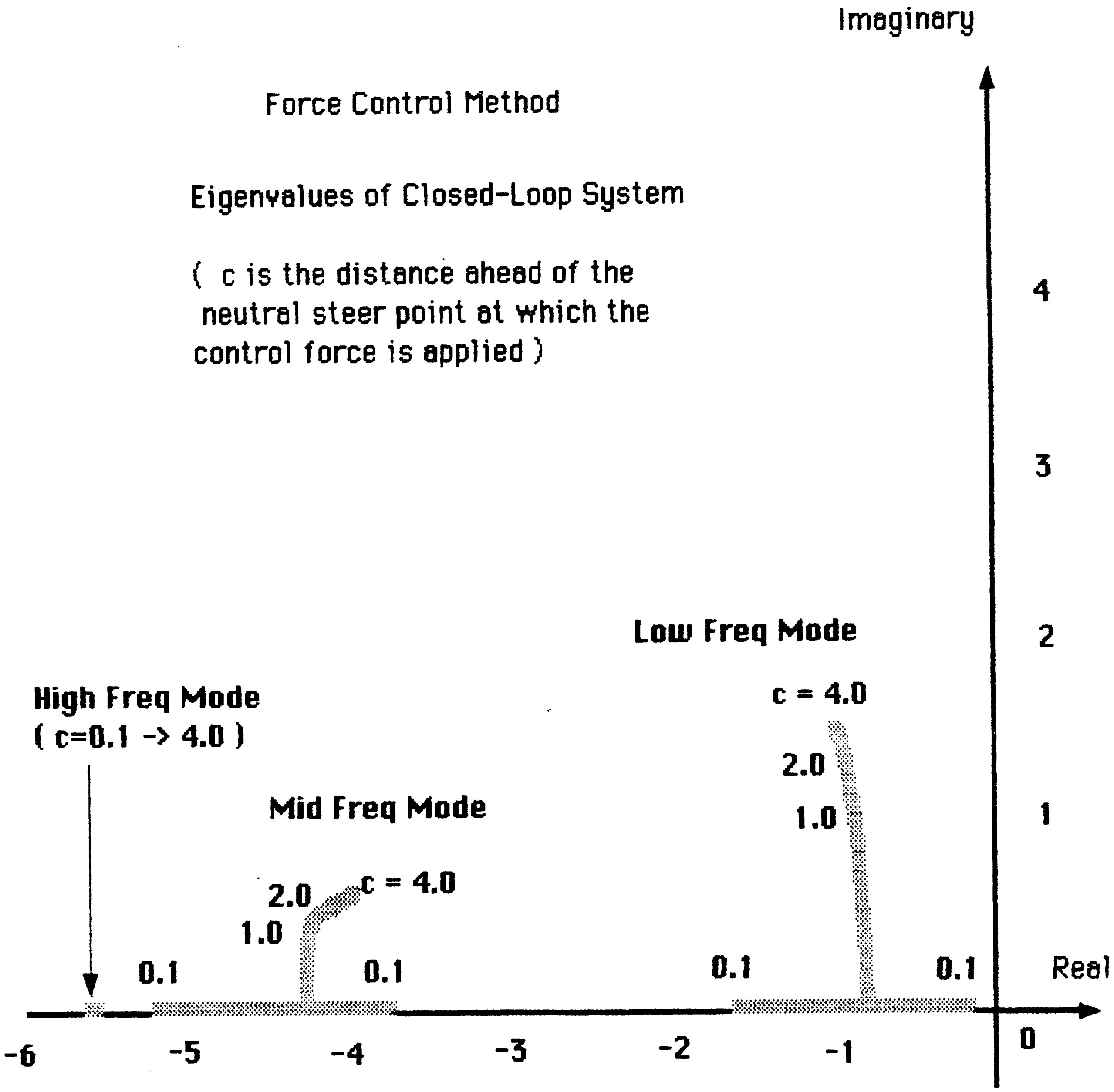


Figure 6. Root Locus



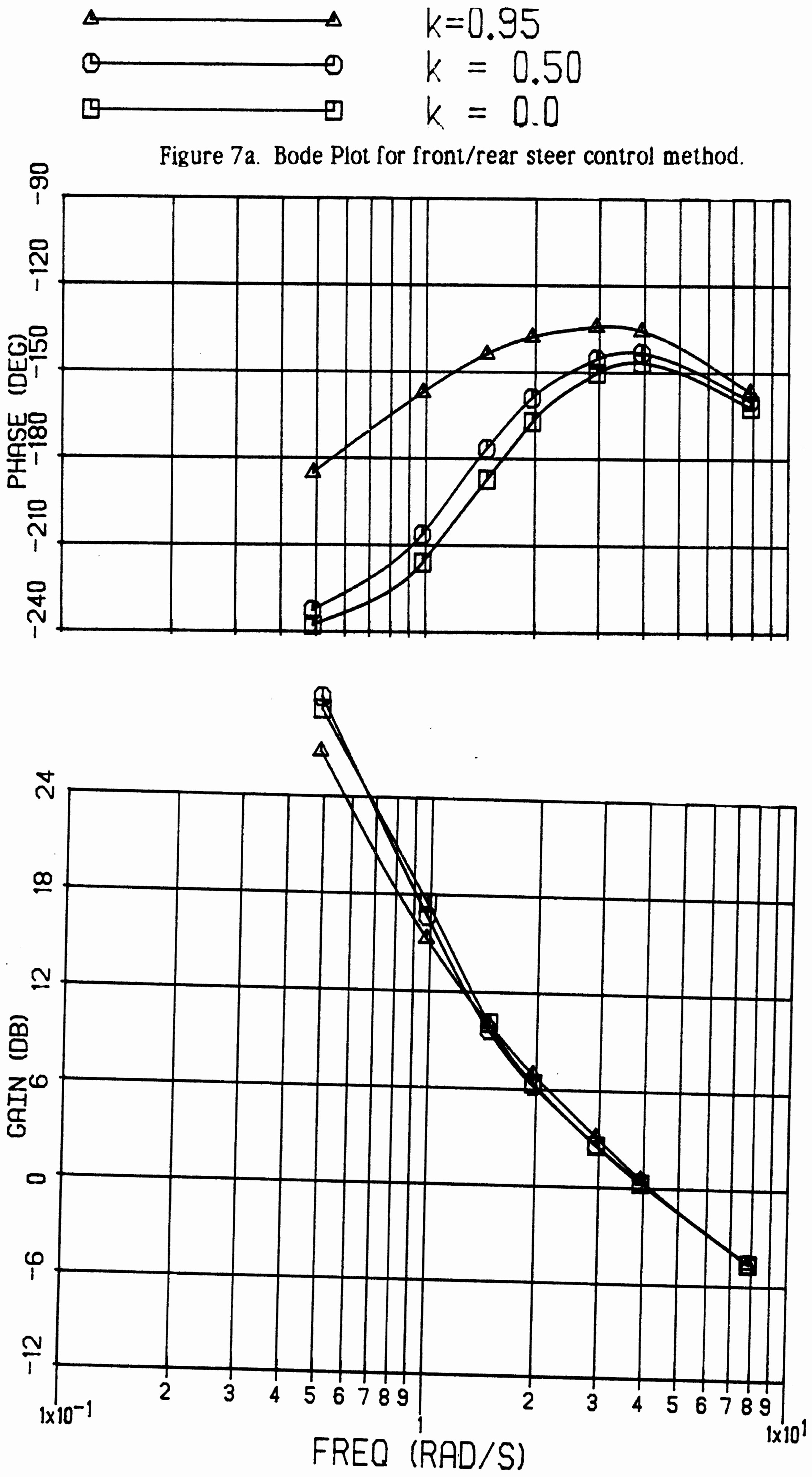
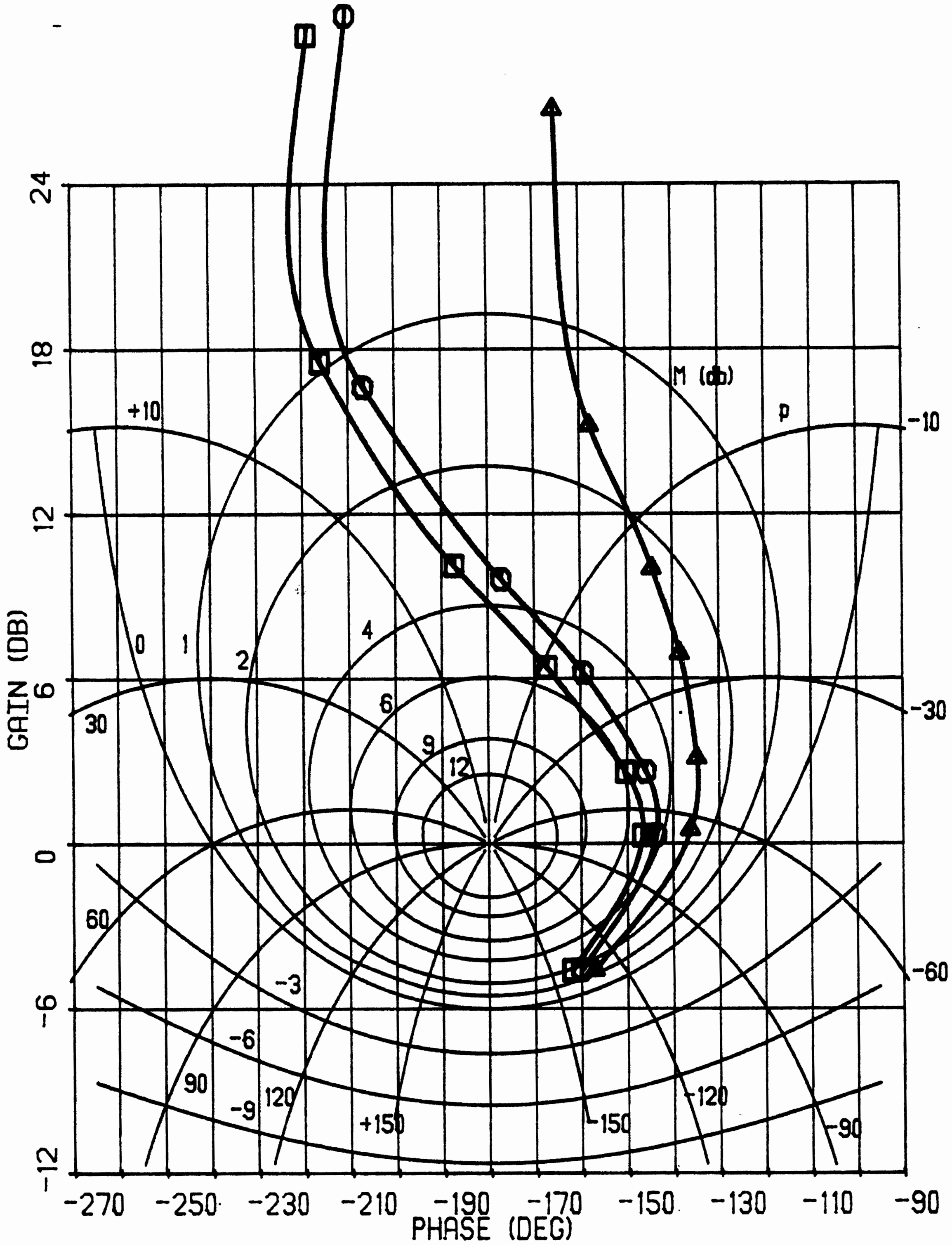


Figure 7a. Bode Plot for front/rear steer control method.

Figure 7b. Corresponding Nichols Chart.



the relative stability of the different vehicle or control system configurations being examined can be more easily evaluated [3]. As seen in Figure 7b, improved stability and damping of the closed-loop system is achieved by increasing the value of k to values less than 1.0. This result simply confirms the same result obtained using the root locus diagram. However, this type of test and analysis would normally be used in a vehicle test environment where the ideal dynamics of the driver/vehicle system are not ordinarily known (as is the case for the root locus calculations seen above).

Figure 8a and 8b show comparable frequency domain results for the force control method. In these figures, the distance "c" ahead of the neutral steer point is being varied from 4.0 feet to 0.1 feet. Again, the Nichols chart of Figure 8b supports the results seen earlier in the root locus diagram of Figure 6.

In Figures 9a and 9b this technique is used to compare the moment control method (M_z control) with that of a conventionally steered vehicle (D_{fw} control; k=0). Interestingly, the calculations show very little difference, perhaps a slight degradation in stability for the moment control method.

Lastly, a sequence of similar parameter variations was performed on the baseline single-unit driver/vehicle system using a standard 12-foot lane-change maneuver to excite the system. The lane-change maneuver is performed within a forward travel distance of 100 feet as defined by a specified path. Time histories of typical system responses were recorded for evaluating the degree of damping present in the time domain traces. Figure 10 shows the baseline configuration with front-wheel-only steering (k = 0 case). The first four time histories seen are: lateral displacement (positive to the right), heading angle (positive to the right), sideslip velocity relative to vehicle body axis system (positive to the right), and yaw rate (rate of change of heading angle). The fifth time history is the "control variable" which corresponds to u_c in the earlier discussion (front wheel steer, or yaw torque, or lateral force - depending upon the analysis). In Figure 10 the control variable is front wheel steer angle, consequently the units of measurement are degrees. The next plot is lateral acceleration of the vehicle mass center in g's. The final plot is of "MSE Preview Path Error" and is simply a measure of the mean squared previewed path error over the preview interval observed by the driver model at each instant of time. Larger values of this signal would imply greater difficulty in minimizing the detected future vehicle/path errors and presumably greater effort required in controlling the vehicle.

Figure 8a. Bode Plot for force control method.

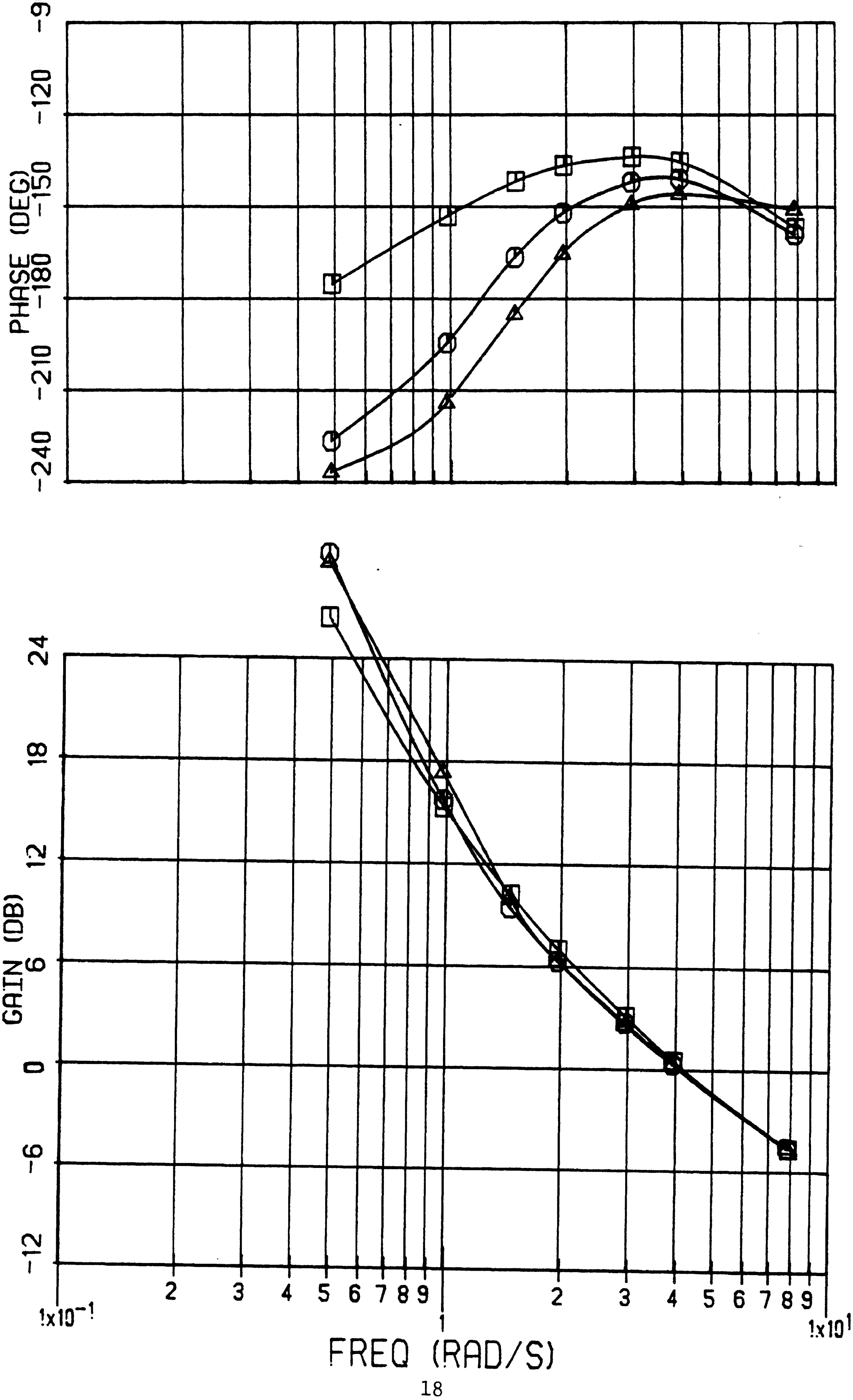
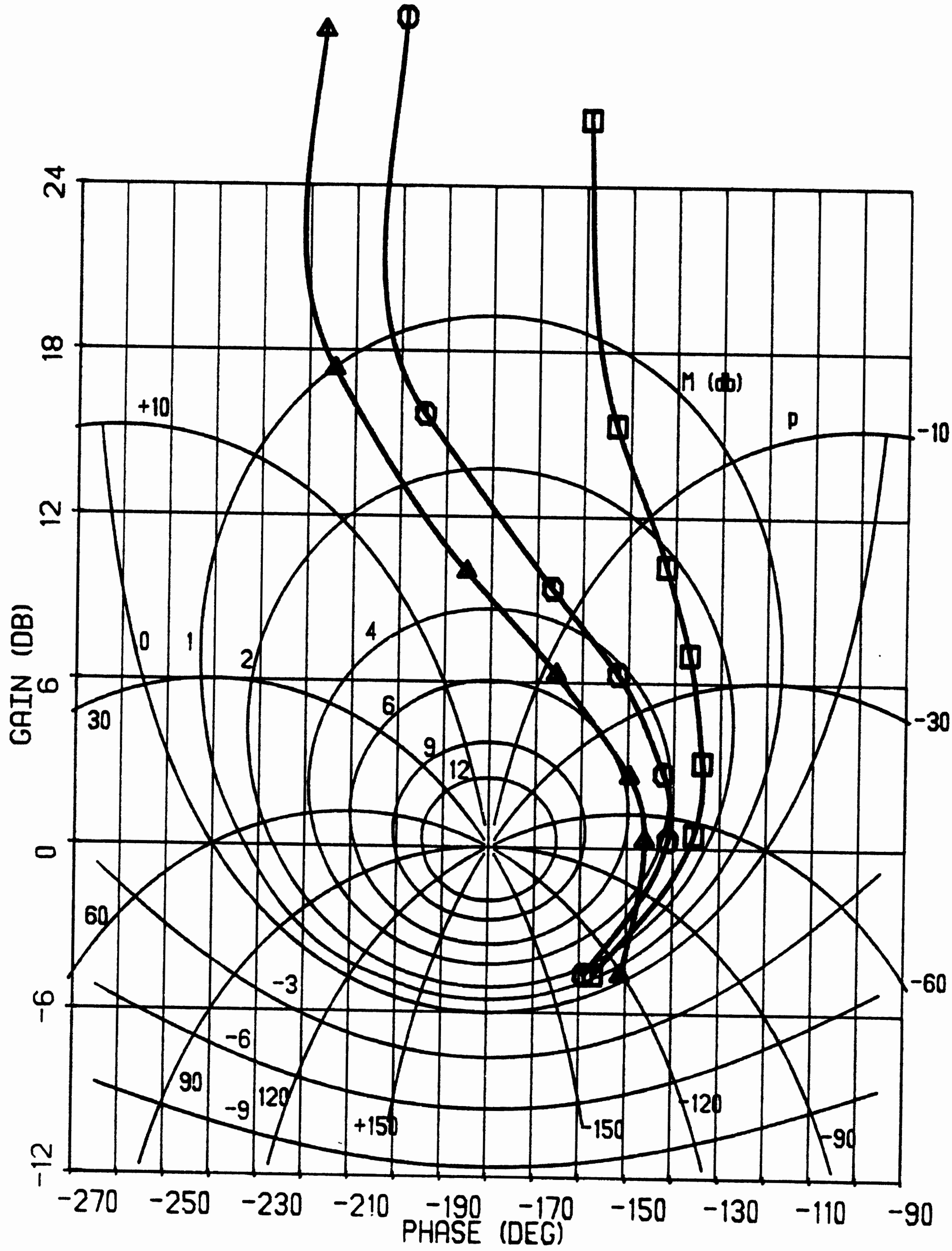


Figure 8b. Corresponding Nichols Chart.



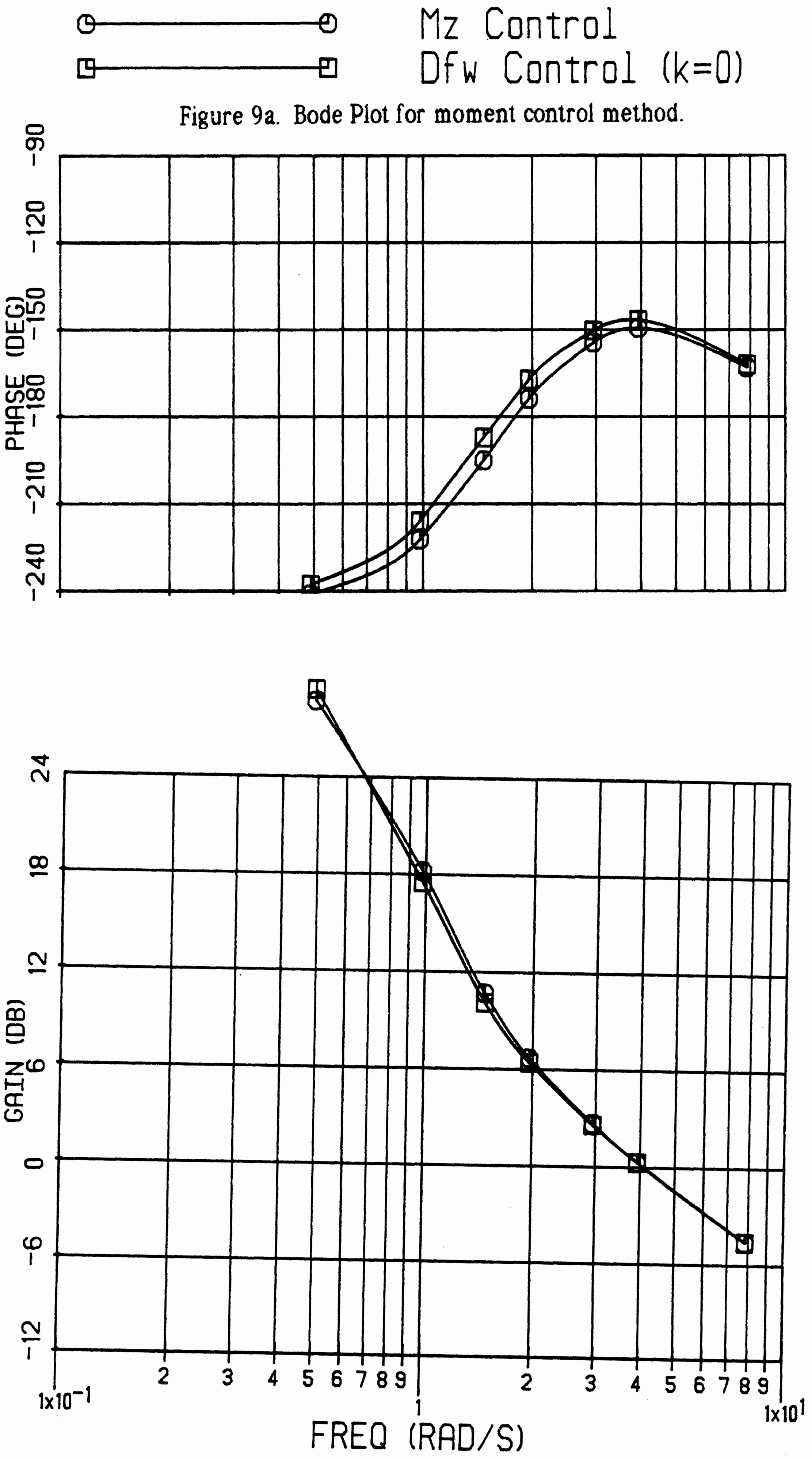
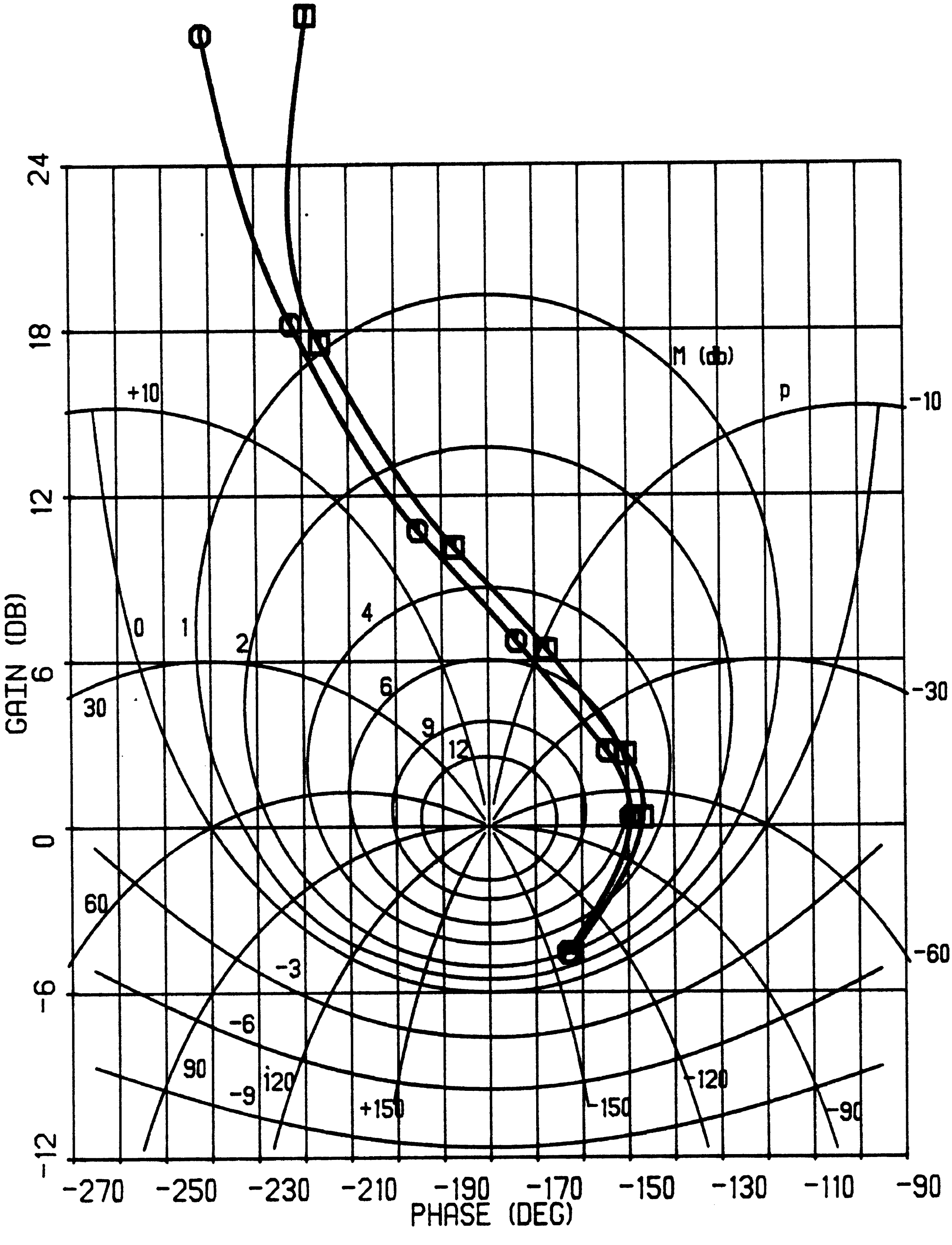
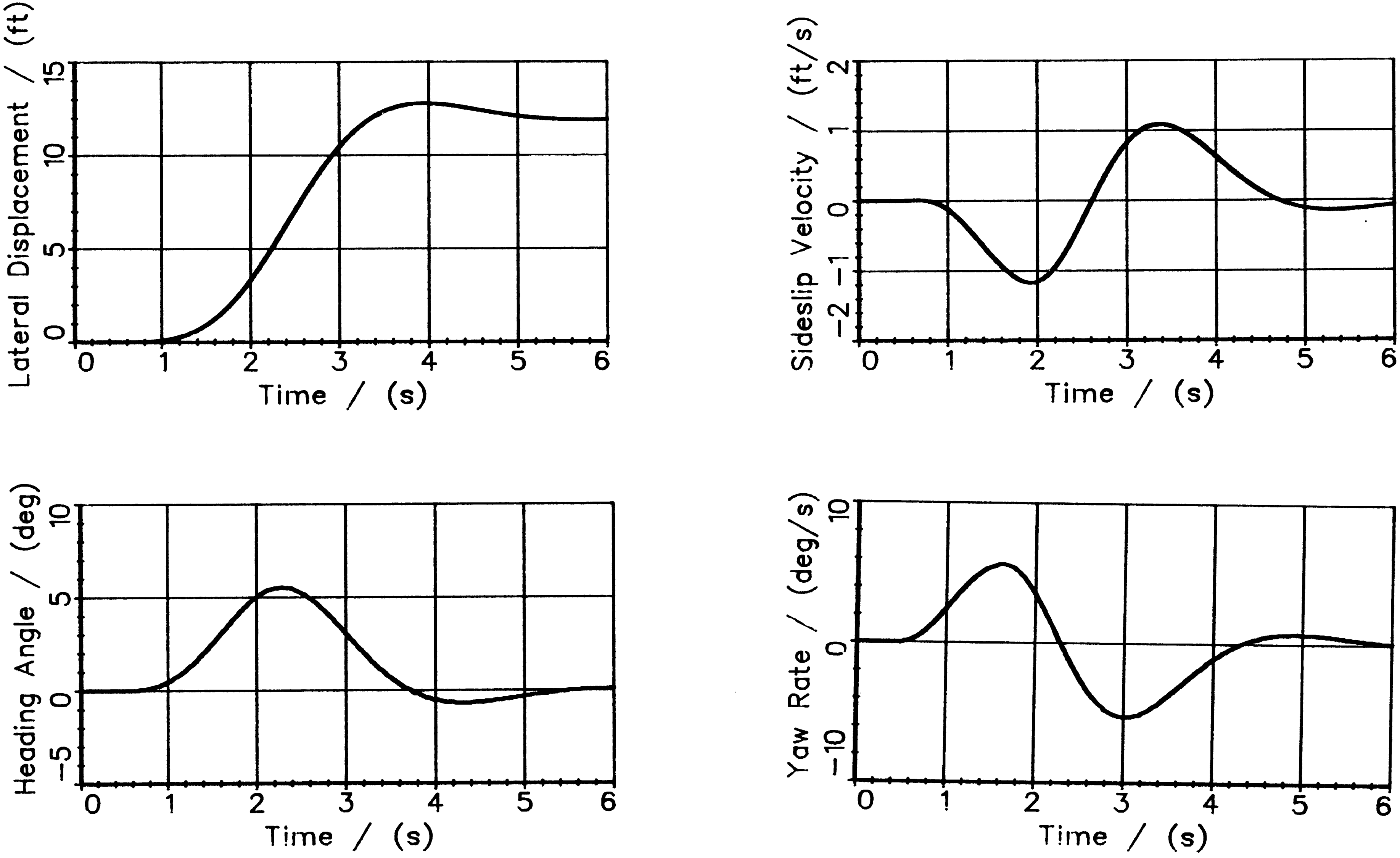


Figure 9a. Bode Plot for moment control method.

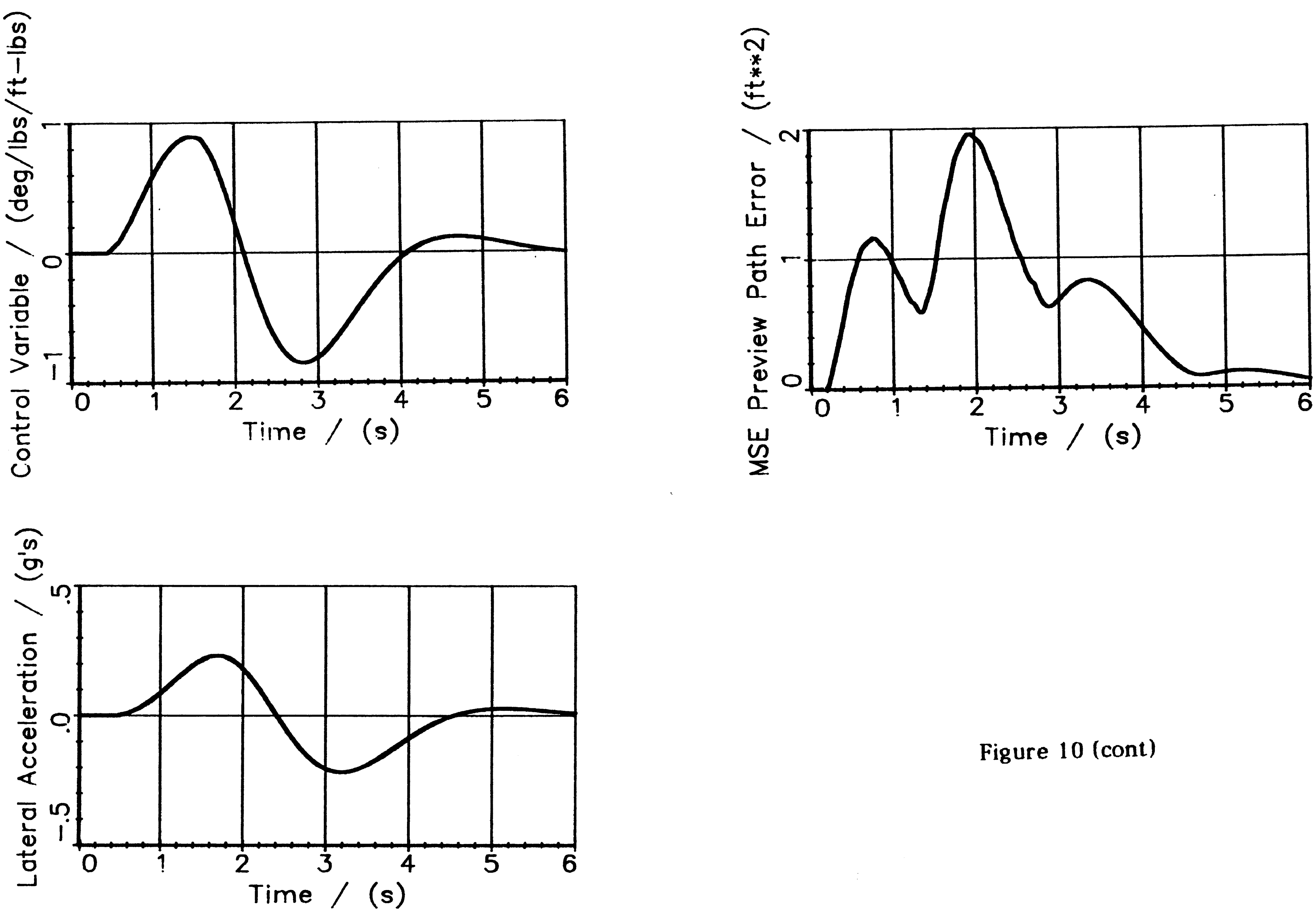
Figure 9b. Corresponding Nichols Chart.





Two Axle Vehicle --- 12 ft Lane Change

Figure 10. Time histories for conventional front wheel steer only (k = 0 case).



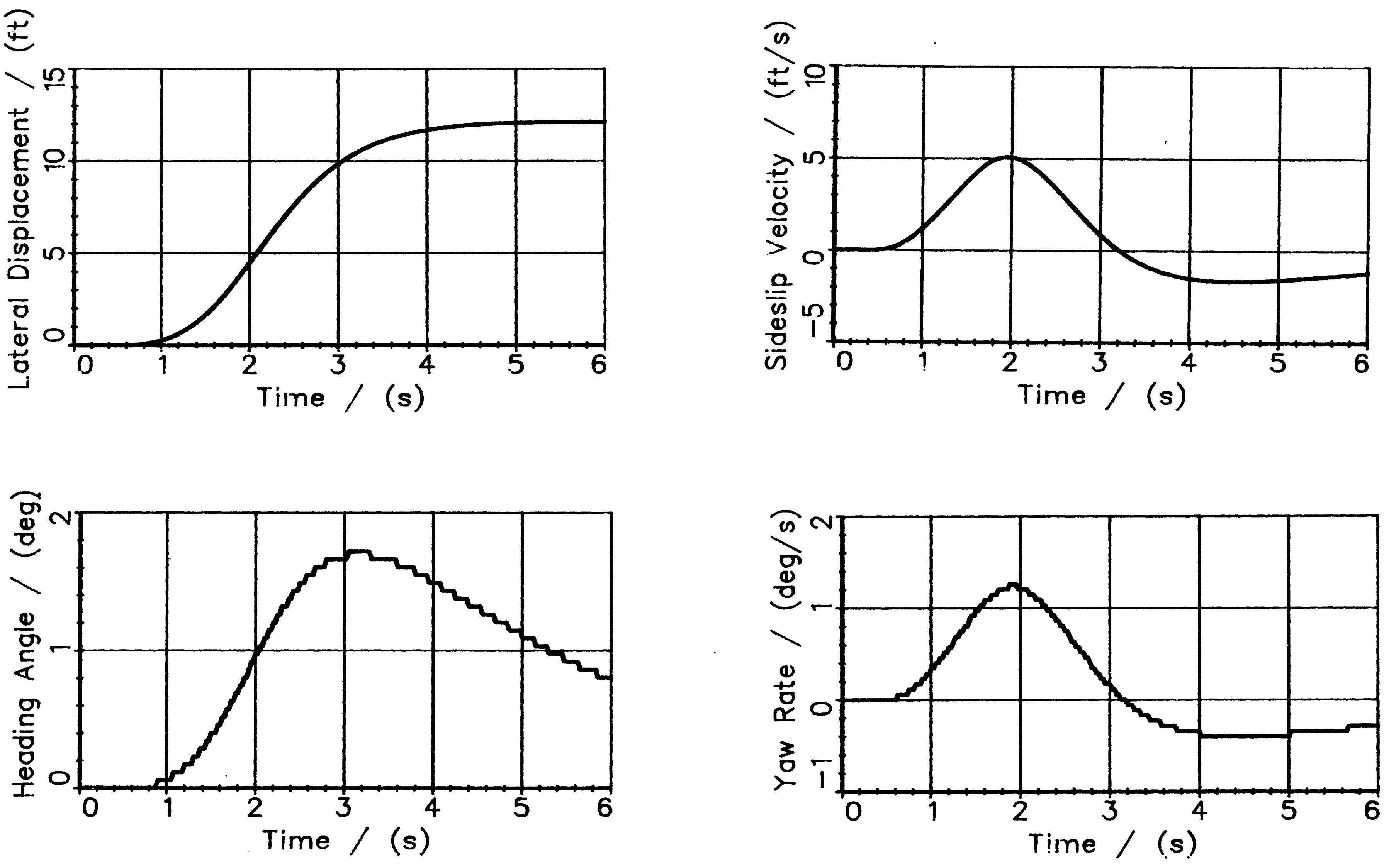
Two Axle Vehicle --- 12 ft Lane Change

Figure 10 (cont)

A set of comparable time histories is seen next in Figure 11 for the case of the rear wheel steering gain set to 0.95. The increased damping predicted before by the root locus and Nichols chart analyses is very evident in this plot. No path overshoot occurs in the lateral displacement time history, while the heading angle remains very small throughout the maneuver. The control variable is again in terms of the front steering wheel angle (degrees). The peak lateral acceleration achieved is somehat less than that of the front-only steered vehicle.

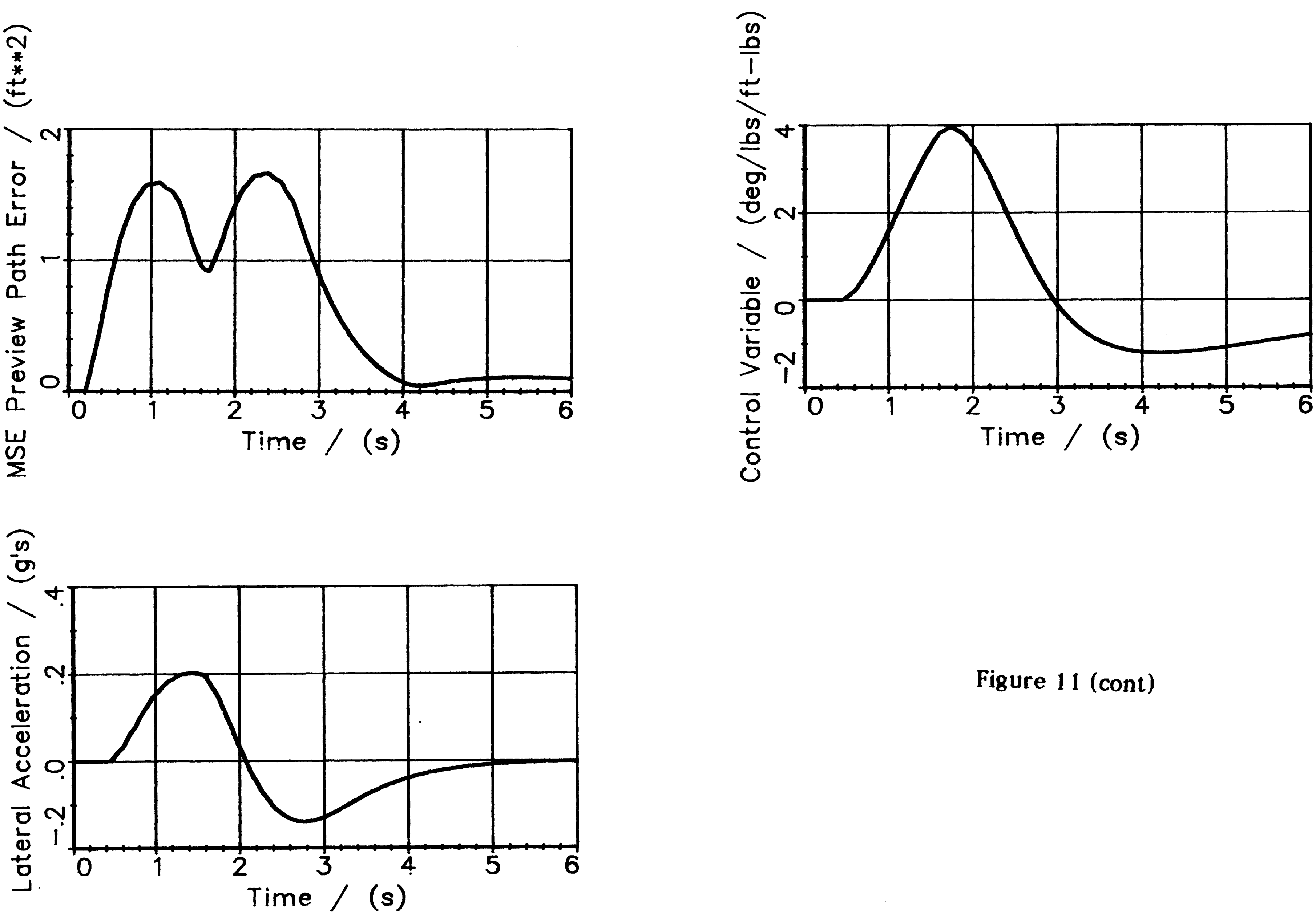
Figure 12 shows the lane change results using the yaw moment control method to steer the vehicle. As suggested by the frequency domain result above, the results are very similar to those obtained with the front-wheel-only steered vehicle.

Finally, Figures 13 and 14 apply to the lateral force control method. Figure 13 is for the force being applied 0.1 feet ahead of the neutral steer point and exhibits very heavy damping. Figure 14 illustrates the destabilizing effect of placing the control force aft of the vehicle neutral steer point. The driver/vehicle system in this case loses directional control and the vehicle spins counter-clockwise as it performs the approximate lane-change maneuver. Strong control over the closed-loop system response and damping can be achieved by the fore/aft position of the applied control force for this method.



Two Axle Vehicle --- 12 ft Lane Change

Figure 11. Time histories for front/rear wheel steer method (k = 0.95 case).



Two Axle Vehicle --- 12 ft Lane Change

Figure 11 (cont)

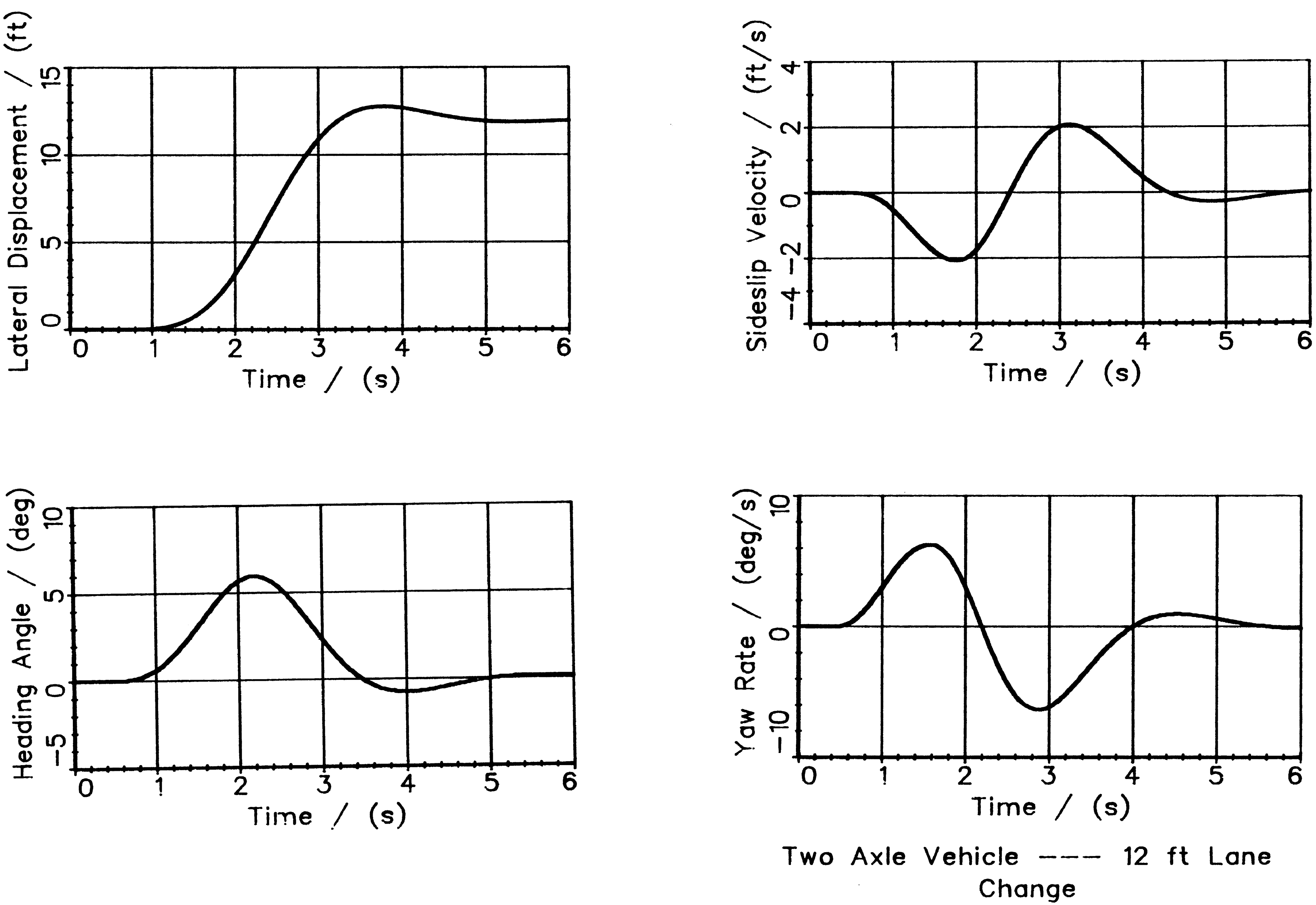
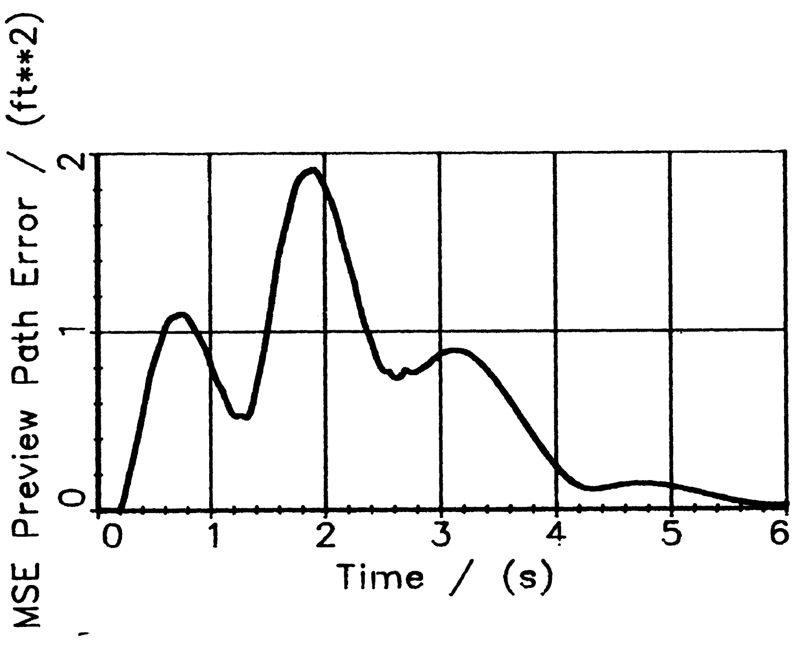
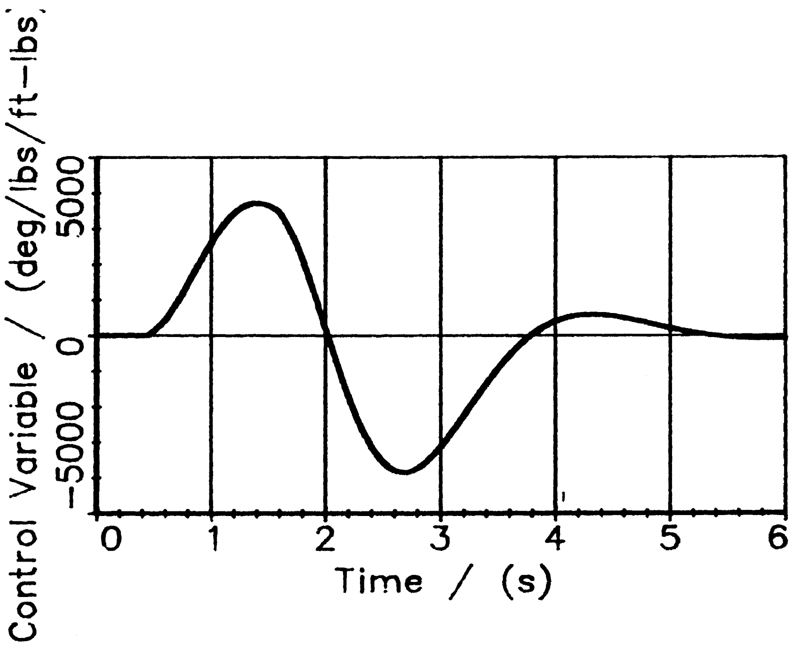
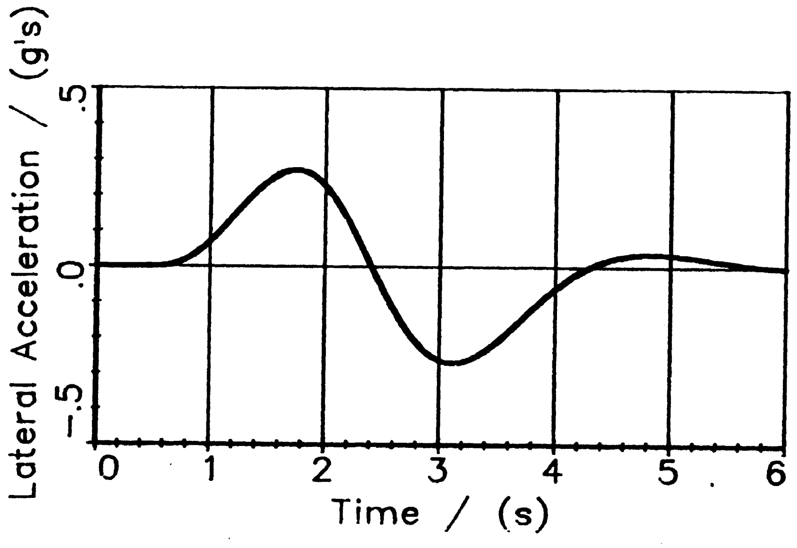


Figure 12. Time histories for moment control method.

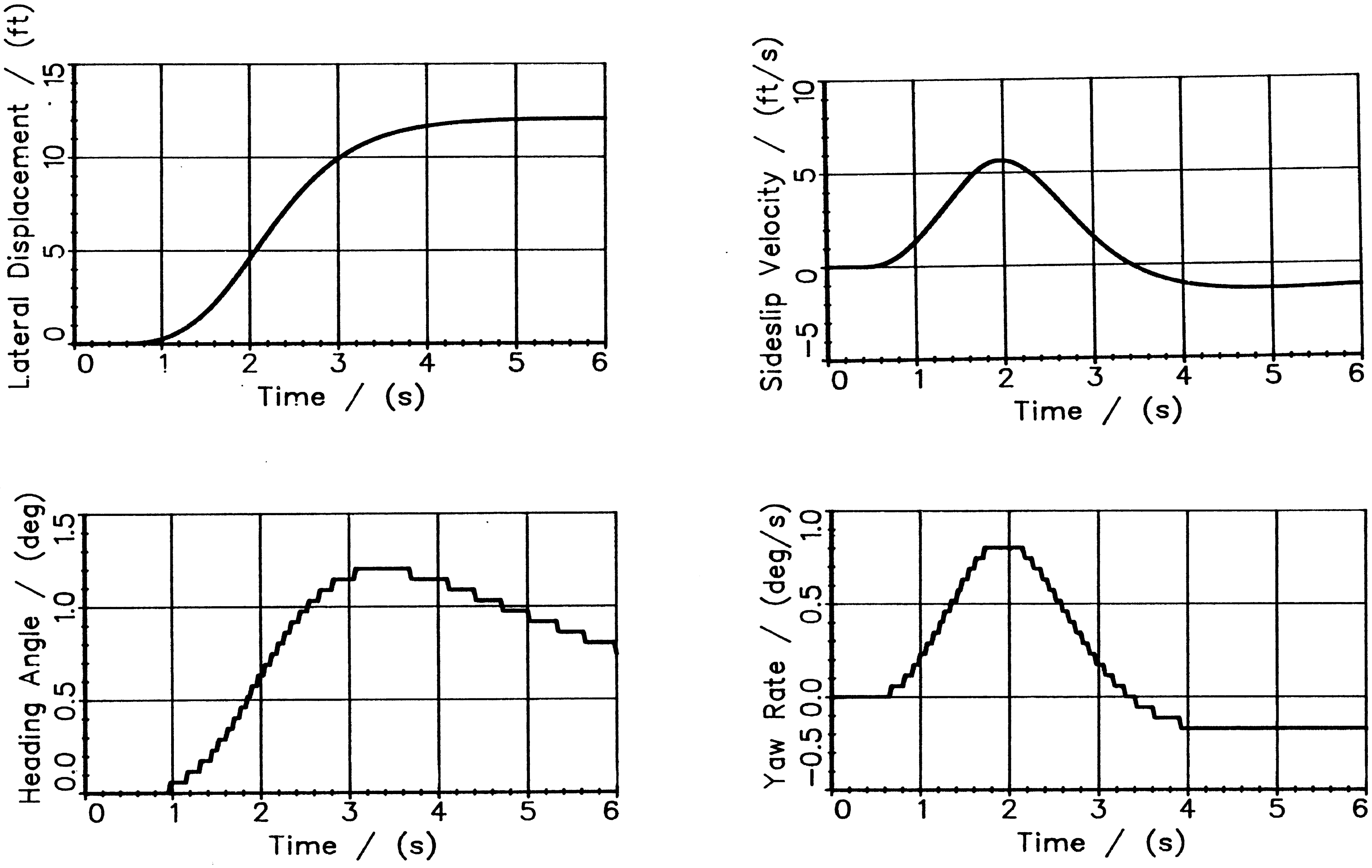






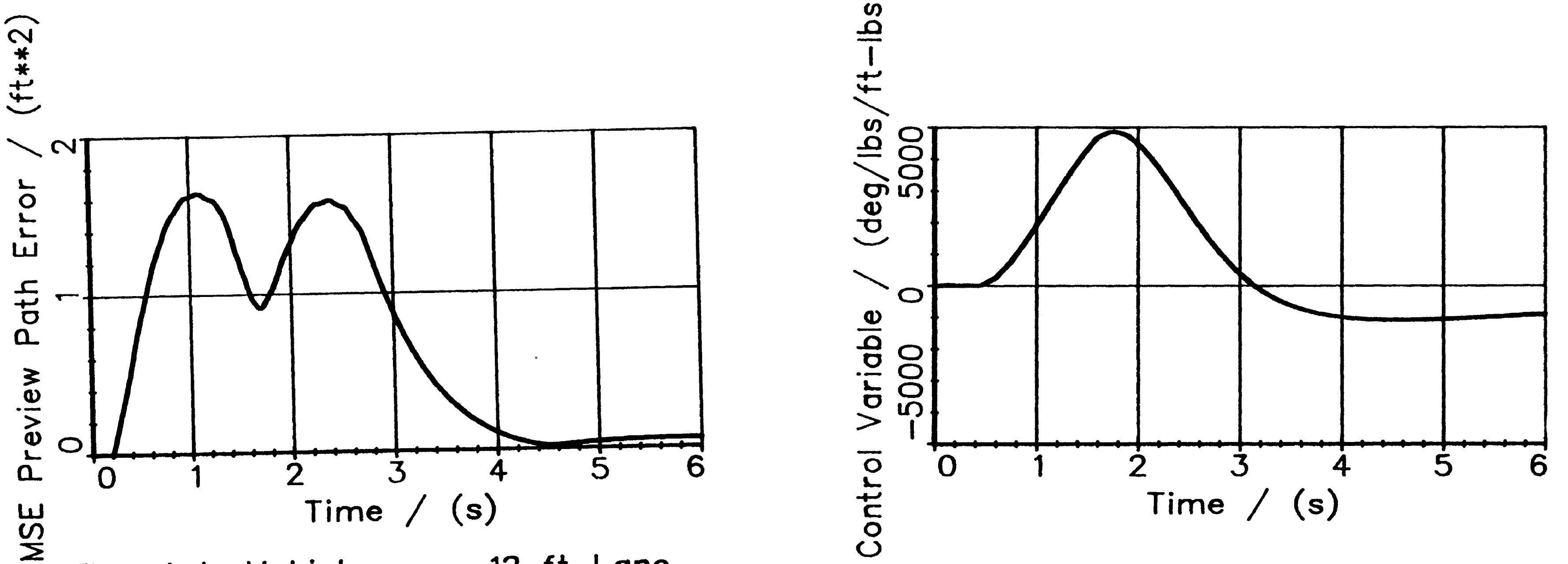
Two Axle Vehicle ---- 12 ft Lane Change

Figure 12 (cont)



Two Axle Vehicle --- 12 ft Lane Change

Figure 13. Time histories for force control method (c = 0.1 case).



Two Axle Vehicle --- 12 ft Lane Change

Figure 13 (cont)

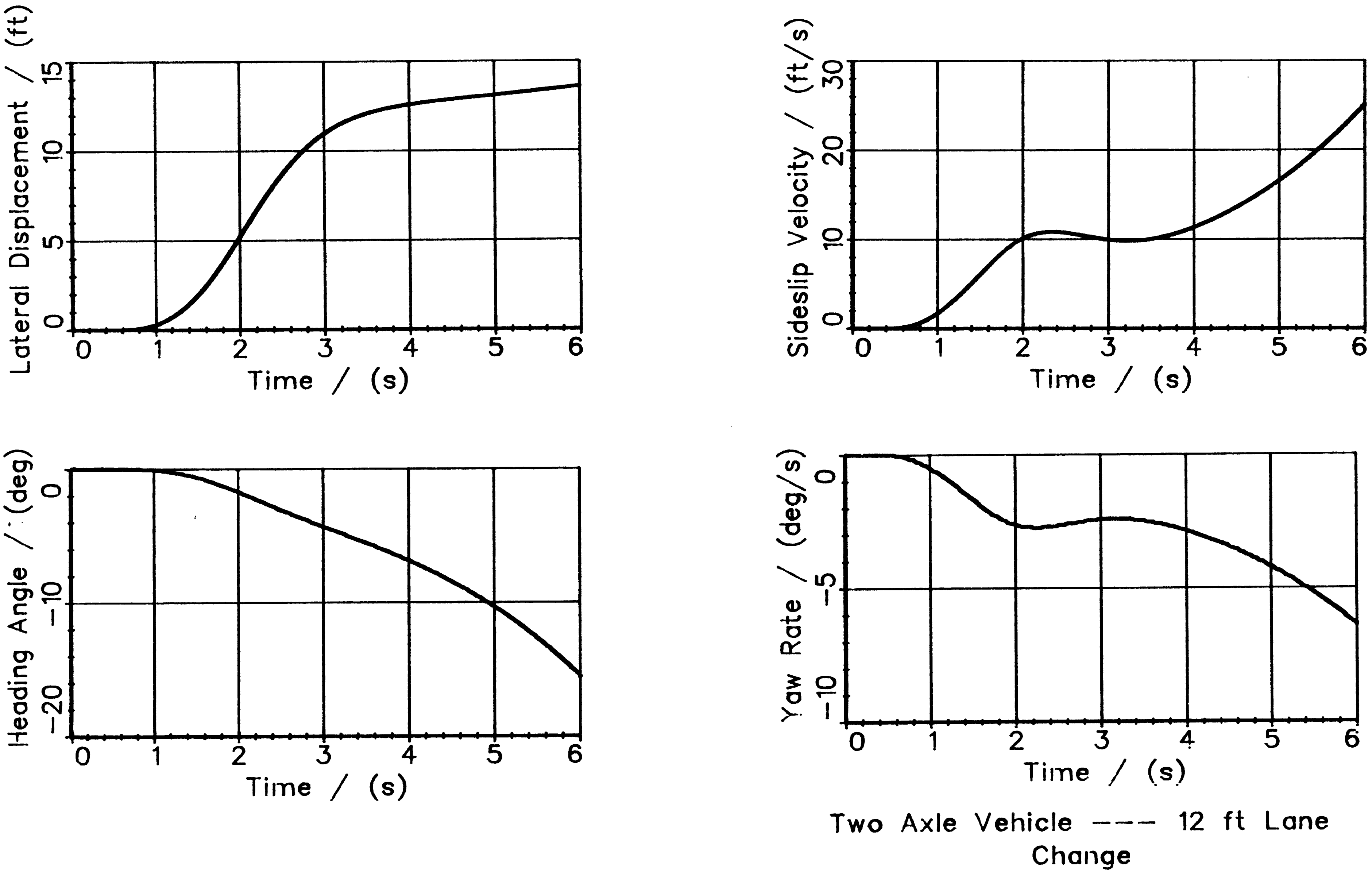


Figure 14. Time histories showing an unstable response due to placement of the applied control force at a point aft of the neutral steer point (c = -0.2 case).

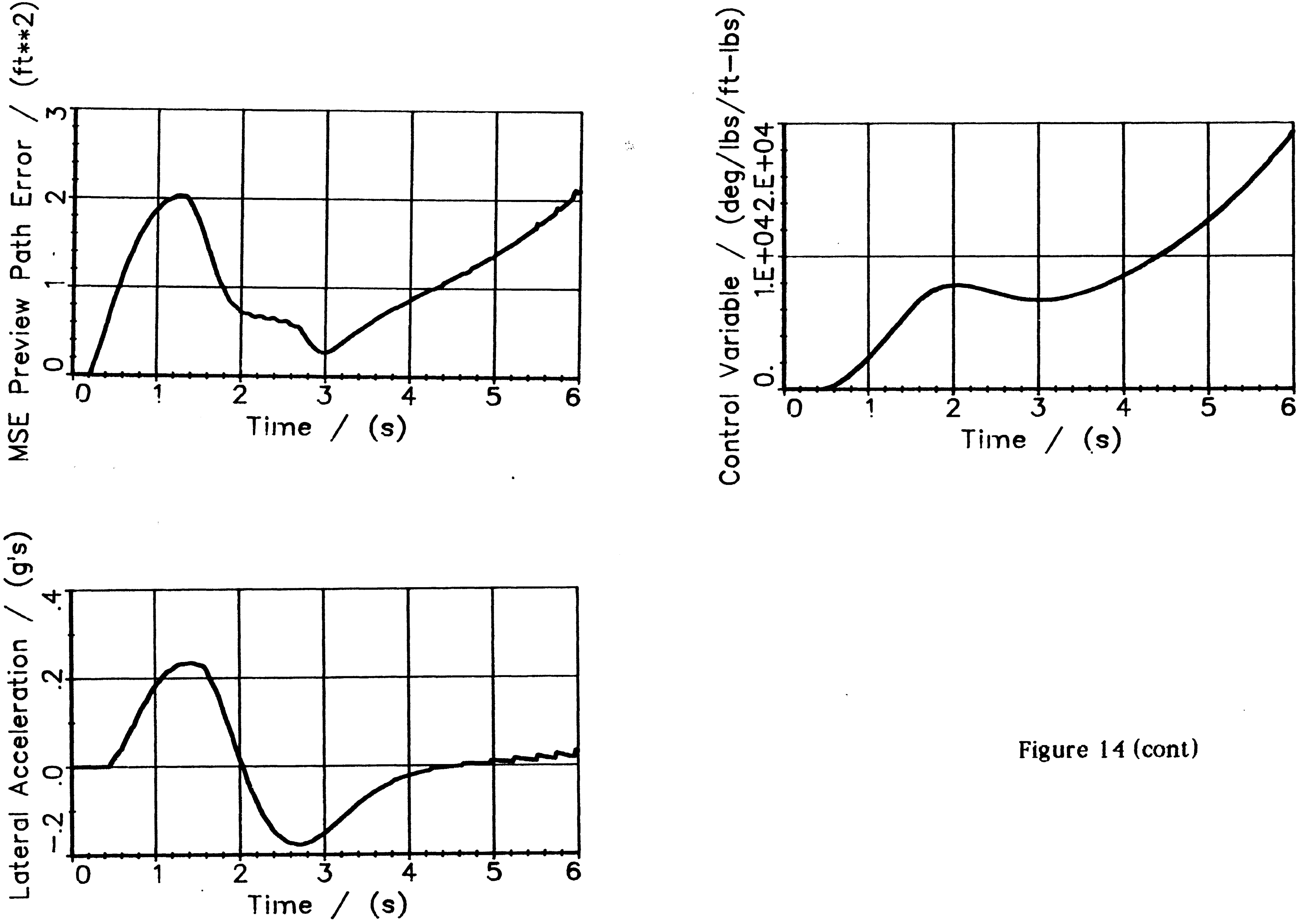


Figure 14 (cont)

Articulated Vehicle Model Results

Since the controlled system dynamics, now represented by an articulated vehicle with an additional degree of freedom in its simplest form, has been extended beyond that of a single-unit vehicle assumed by the original UMTRI driver model, the original UMTRI driver model was accordingly modified and extended to account for the additional degree of freedom. (This particular version of the driver model is to be used to steer and control articulated vehicles similar to that described below.) A diagram of the basic model of the articulated vehicle used by the driver model is seen in Figure 15. A list of corresponding variable and parameter definitions appears in Table 2. The model has two control inputs: the front wheel steer angle, δ, and the articulation torque, M_c. The complete equations defining the model are contained in Appendix A. (A later simplification slaves the articulation control torque to the front wheel steer angle by a proportional gain factor, thereby resulting in essentially one control input and a gain parameter for controlling the front-wheel-steer / articulation-torque "mixture.")

The LVS vehicle was used as a preliminary guide for defining the characteristics of the model. Estimates of LVS mass and geometric parameters were therefore used in the subsequent analyses for the articulated vehicle and are listed in Table 3.

Figure 16 contains time histories for the described driver/vehicle system performing a 12-foot lane-change. The input path was identical to that used for the single-unit vehicles. The first four plots correspond to the lead unit. The second four plots apply to the trailing unit. The last group of time histories show lateral acceleration traces for the lead and trailing units, the control variable (degrees of front wheel angle), and the aforementioned "MSE Preview Path Error." In this particular case, the articulation control torque was active and was slaved to the front wheel steer angle (10,000 ft-lb of articulation control torque / degree of front steer angle).

Figure 15- Articulated Vehicle Model

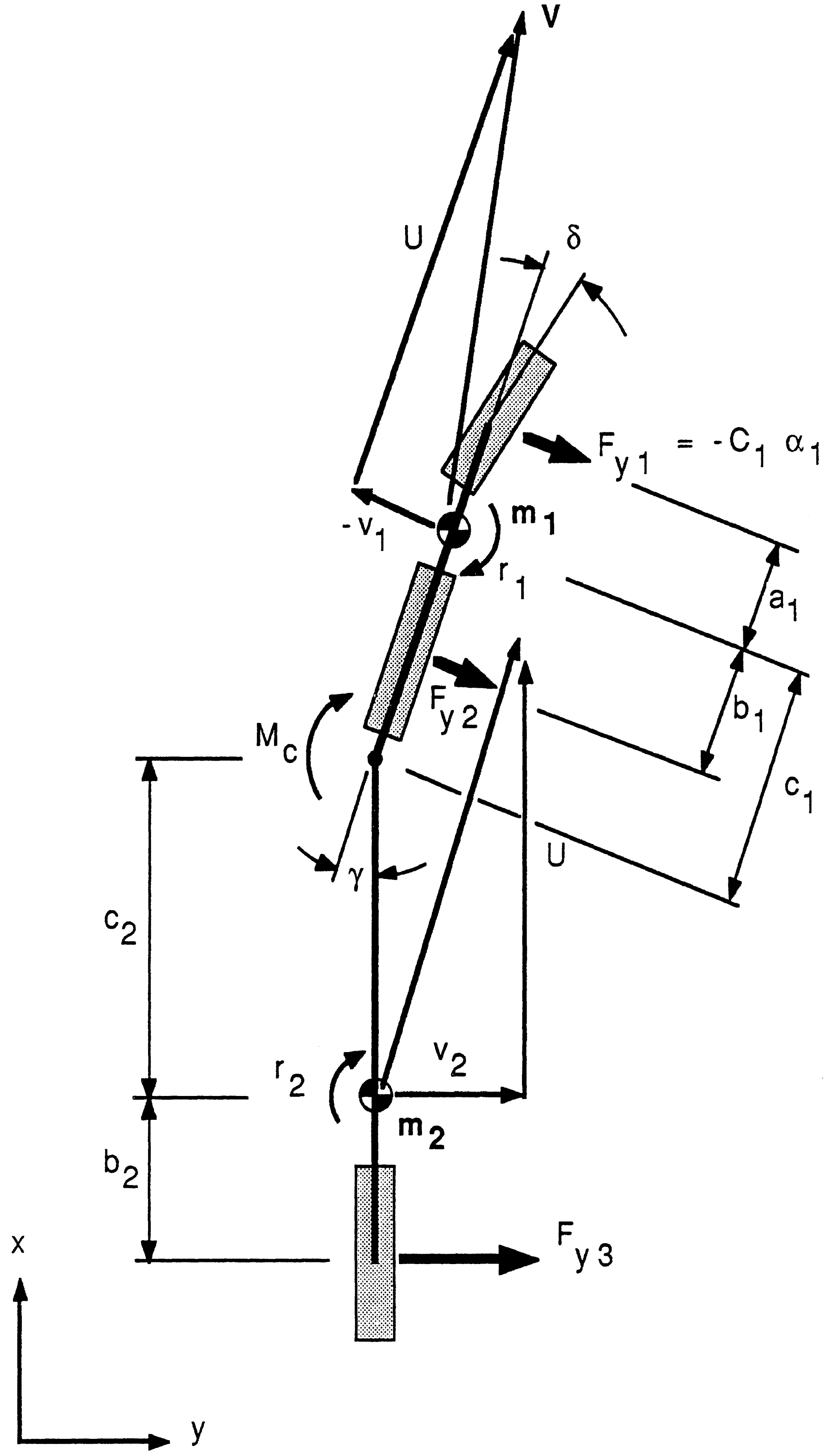


Table 2. Articulated Vehicle Model

(Internal Driver Model Representation)

Parameter	Description
m1	mass of front unit
m2	mass of rear unit
v1	lateral sideslip velocity of unit 1
v2	lateral sideslip velocity of unit 2
r1	yaw rate of unit 1
r2	yaw rate of unit 2
a1	distance form c.g. of unit 1 to axle 1
b1	distance form c.g. of unit 1 to axle 2
b2	distance form c.g. of unit 2 to axle 3
c1	distance form c.g. of unit 1 to hitch
c2	distance form c.g. of unit 2 to hitch
U	forward velocity component in x-body axis
δ	front axle steer angle - control variable
γ	articulation angle
α_i	tire sideslip angle (axle i)
C_i	tire cornering stiffness (axle i)
M_c	articulation torque - control variable
F_{yi}	lateral tire force (axle i)

Table 3. Preliminary LVS Parameter Values / Estimates

Parameter	Value
m1	750 slugs
m2	515 slugs
I1	9,000 ft^2-slug (lead unit yaw inertia)
I2	20,000 (rear unit " ")
a1	2.5 ft
b1	2.5 ft
c1	5.3 ft
b2	2.0 ft
c2	14.25 ft
U	88 ft/sec
C_1	42,000 lb/rad/suspension-side (tire cornering stiffness)
C_2	68,000 "
C_3	62,000 "
TF	1.5 sec (driver preview time)
Tau	0.25 sec (driver transport lag)
RKMOM	570,000 ft-lb/rad (articulation control moment-front steer gain)

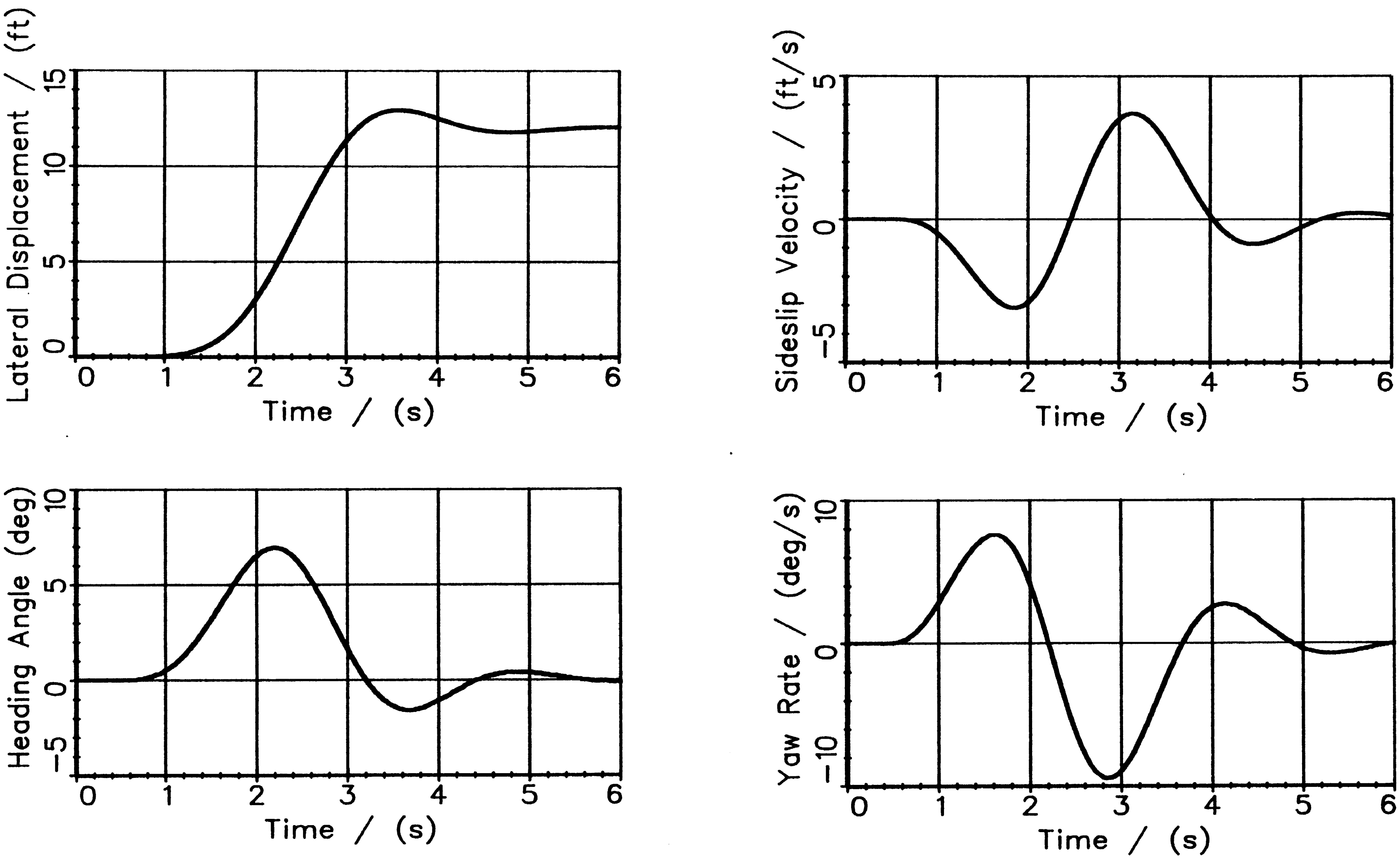


Figure 16. LVS Baseline Simulation - 12 ft Lane Change

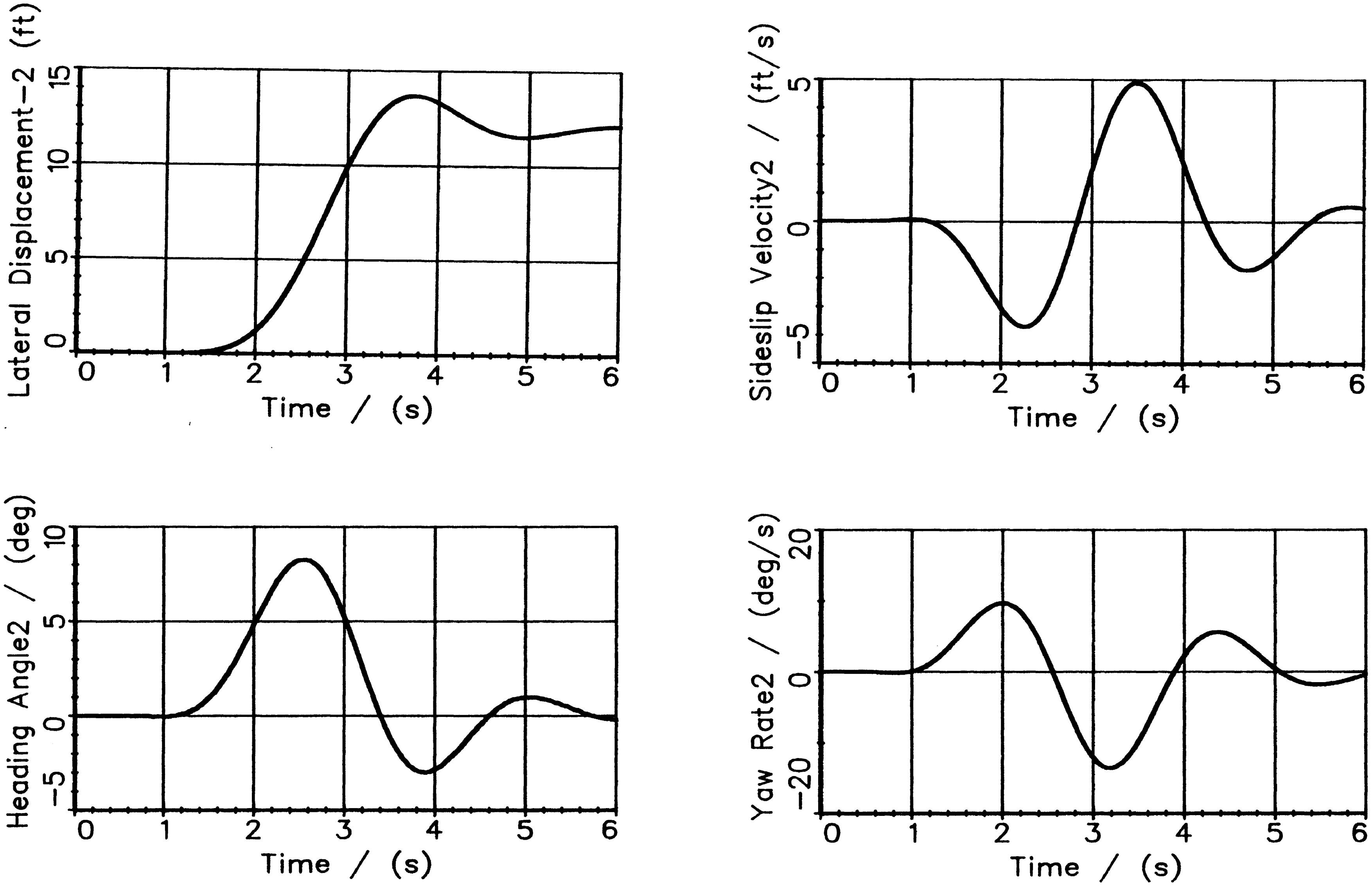
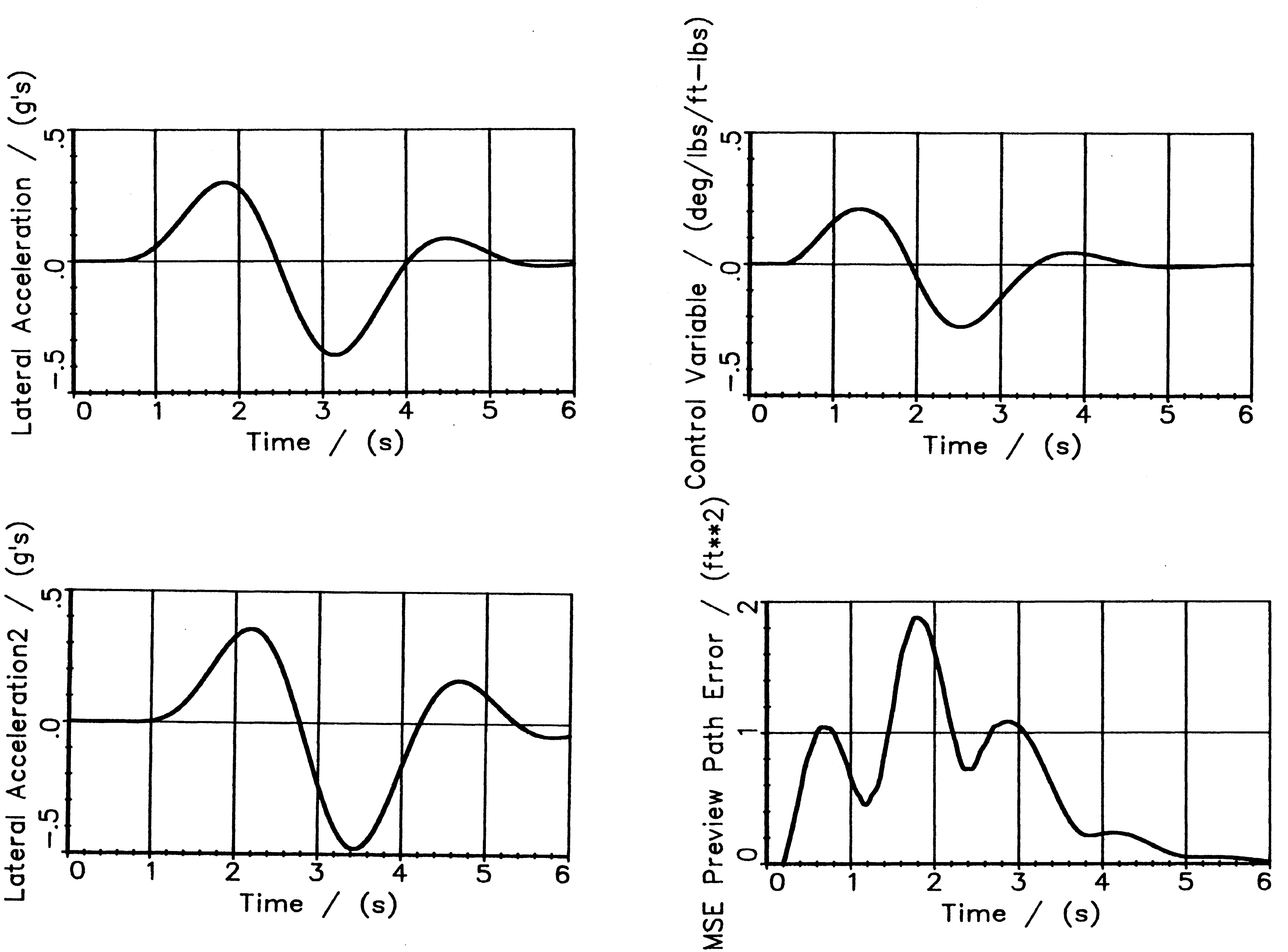


Figure 16 (cont)



Articulated Vehicle

Figure 16(cont)

3.0 Installation of the Driver Models into DADS Software

The driver models described in the previous section have been written in FORTRAN 77 code in order to transport and install them into the DADS vehicle model used by TACOM. A FORTRAN listing of the single-unit and articulated models is contained in Appendix C. The basic structure of these driver model programs is shown in Figure 17. Each program begins by reading parameters associated with the driver characteristics (preview time and transport delay) and the x-y trajectory defining the desired path to be followed (Subroutine DRIVE1). Based upon this information and certain vehicle related parameters (e.g., tire cornering stiffnesses, axle weights, etc.) the driver model initializes, within Subroutine DRIVE2, its own internal vehicle model to represent a simplified version of the actual vehicle being controlled (i.e., DADS vehicle). The state transition matrix is then calculated for the identified closed-loop system used by the driver model in Subroutine TRANS. This calculation allows the driver model to predict / estimate future vehicle positions based upon current conditions during a simulated maneuver. Finally, Subroutine STEER contains the code for calculating the closed-loop steering function returned to DADS during the numerical integration process.

In order to interface these routines to the DADS software, auxiliary subroutines are provided within DADS for permitting users to install and interface their own programs. As of this reporting, the driver model for the single-unit vehicle has been installed and successfully tested within the DADS-2D version. Example time histories from the DADS post-processor are seen in Figures 18 and 19. Figure 18 shows lateral displacement and lateral acceleration of the mass center recorded during a 12-foot lane-change maneuver. Figure 19 shows the conventional DADS heading angle, ϕ, and yaw rate, ϕ', variables recorded from the same run. The results agree quite accurately with prior results obtained by UMTRI using equivalent models, or, when compared with the simplified linear analysis calculations reported in Section 2.0

Figure 17. Basic Structure of the Driver Model

Subroutine Drive1

Read driver model parameters
and trajectory information

Echo parametric data

Subroutine Drive2

Initialization of internal vehicle
model parameters used by the
driver model

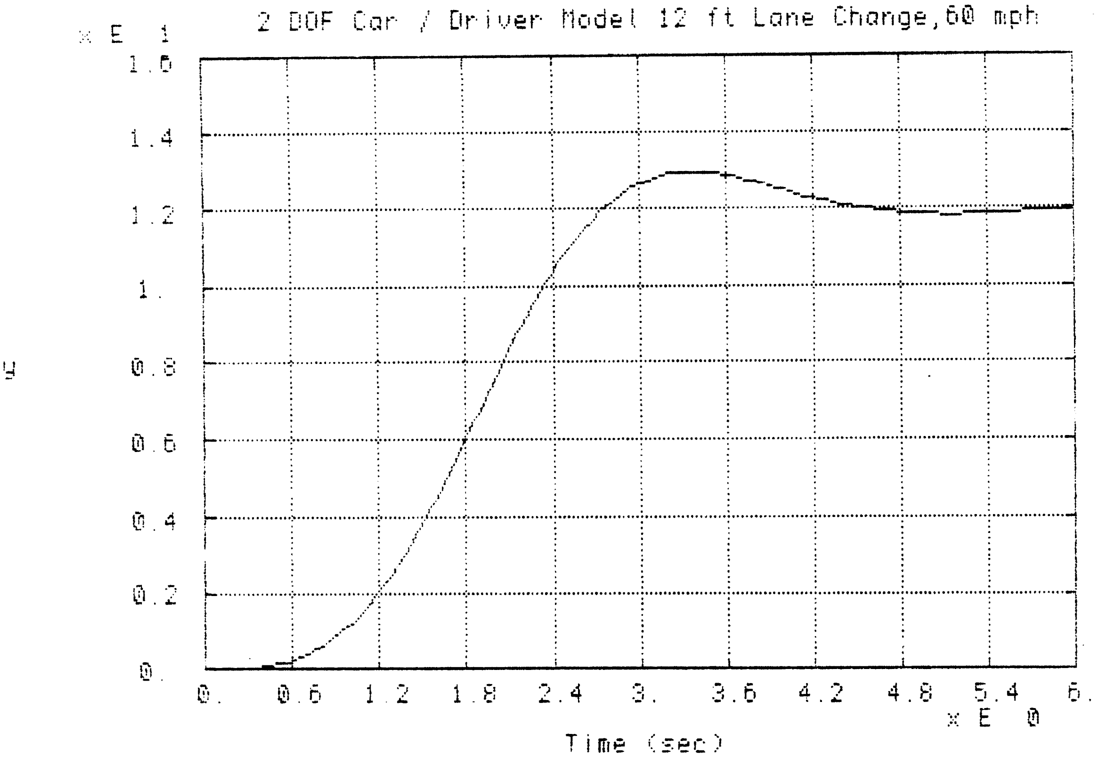
Subroutine Trans

Transition matrix calculation

Subroutine Steer

Closed-loop steer calculation

Figure 18. DADS - 2D



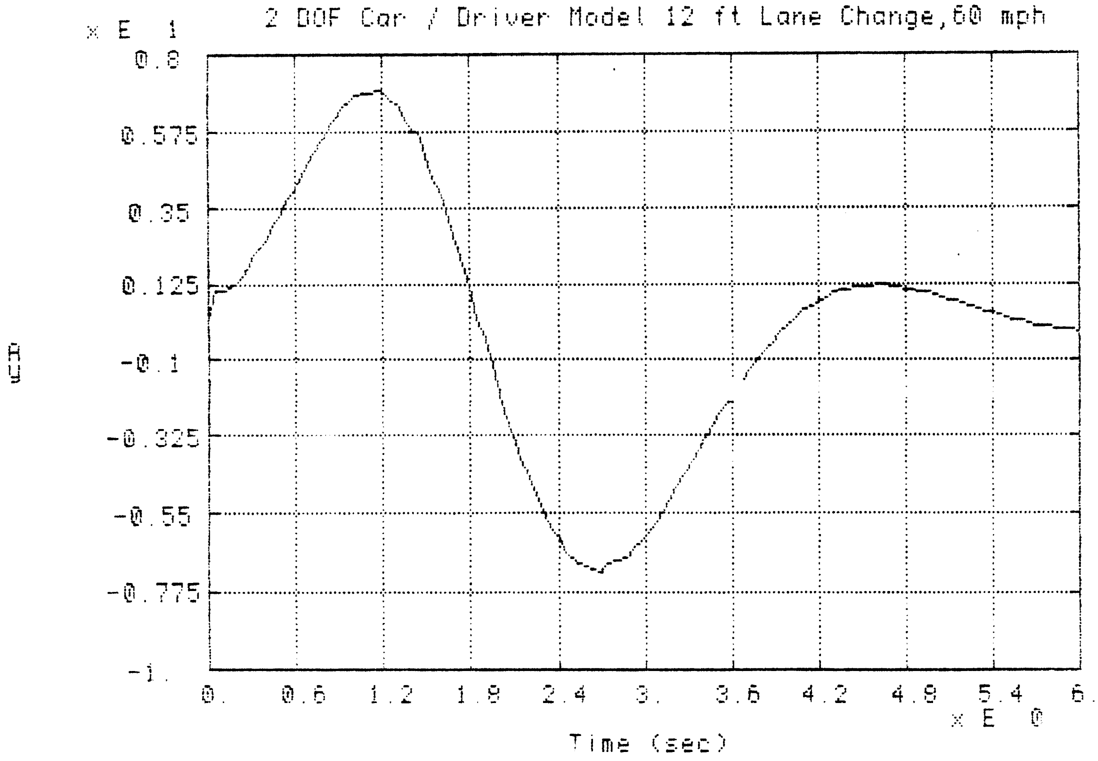
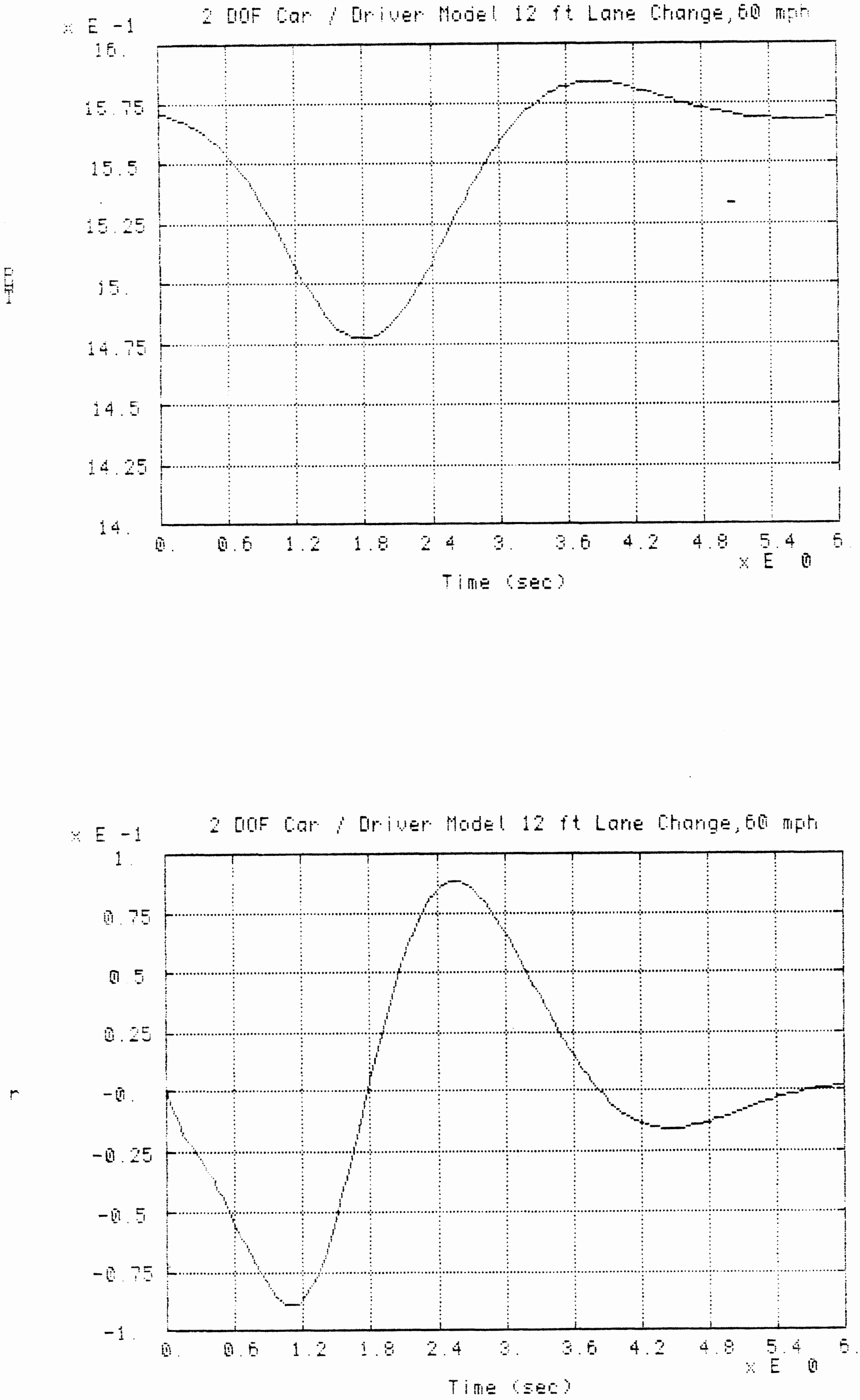


Figure 19. DADS - 2D



4.0 Plan for Driver / Vehicle Testing (1987)

The test plan (contained in Appendix B) applies to the first set of driver /vehicle tests currently scheduled for the spring of 1987. Because of the recent revision in the project schedule which moved the first set of tests from the fall of 1986 to the spring of 1987 and, because of the initial plans for Phase II tests during 1987, two test series will be conducted during the 1987 time frame. However, the plan described here only pertains to the first series of driver/vehicle tests to be conducted during the spring of 1987.

Because the planned tests are presently nine months away, it was advisable that completion of this planning task (task 5 within the Phase 1 project schedule) be deferred until January of 1987 in order to allow for likely changes and improvements to the test plan between now and the commencement of testing. Consequently, the plan is viewed as preliminary and subject to further refinement by TACOM and UMTRI over the remainder of this year.

The plan describes two basic types of tests to be conducted with the HMMWV vehicle. The first is a simple closed-loop turning maneuver along a circular path described to the driver by a sequence of cones. The driver approaches the circular path along a straight tangent and then transitions into the curve, finally reaching a steady-state turning condition. The curvature and speed are selected to produce approximately a 0.3 g steady turning lateral acceleration condition. Driver steering behavior and associated vehicle responses during the transition phase, as well as during the steady turning phase, will be used to refine and validate the developed steering models. Tests at different speeds or curvatures will be run to evaluate consistent variations in driver behavior (as likely reflected through use of different preview strategies for different conditions), as well as to estimate the likely level of randomness to be expected in driver steering behavior during such maneuvers.

The second vehicle/driver test is a transient lane-change maneuver used to excite the closed-loop system and observe the degree of controllability and damping present for different speeds and course geometries. The basic maneuver is designed to cause the driver to steer from one lane of travel to an adjacent lane within a prescribed forward distance. The nominal steering maneuver imparts a sinusoidal-like excitation to the vehicle/driver system. Control over the severity of the maneuver is accomplished by varying the chute-to-chute forward travel distance. The primary items of interest for this maneuver are the path tracking capabilities of the vehicle/driver system, peak levels of vehicle responses achieved during the maneuver, and the ability of the closed-loop system to become damped once excited.

5.0 Summary & Conclusions

The work to date aimed at developing improved methods for simulating closed-loop (driver) steering control with large scale vehicle models (such as the DADS / TACOM vehicle model) has shown good progress. This is particularly true for vehicle maneuvers which lie well within the low and moderate acceleration regimes where the vast majority of "normal" driving occurs. The results reported here indicate good agreement with previous simulation and experimental experience of vehicle/driver systems undergoing path tracking maneuvers. Furthermore, the newer models and concepts, which are applicable to the less conventional tracked and powered-articulated vehicle configurations, also appear to offer interesting alternatives for extending the present driver control methods to these systems.

During the course of the following year (Phase II), the current model will be fully implemented on the TACOM / DADS system and further extended to account for the effects of variable terrain and speed changes. Full-scale vehicle/driver tests will conclude the Phase II activities with the HMMWV tests being conducted at the Chrysler Proving Grounds.

References

1. MacAdam, C. C., "Application of an Optimal Preview Control for Simulation of Closed-Loop Automobile Driving," *IEEE Transactions on Systems, Man, and Cybernetics*, Vol. SMC-11, No. 6, June 1981.

2. McRuer, D. T. et al., "Measurement of Driver/Vehicle Multiloop Response Properties with a Single Disturbance Input," *IEEE Transactions on Systems, Man, and Cybernetics*, Vol. 5, No. 5, May 1975.

3. MacAdam, C.C., "Frequency Domain Methods for Analyzing the Closed-Loop Directional Stability and Maneuverability of Driver/Vehicle Systems," *Proceedings of the International Conference on Modern Vehicle Design Analysis*, London, UK, June 1983.

Appendix A

Articulated Vehicle Equations

The equations appearing in this appendix apply to the articulated vehicle model shown in Figure 15 and Table 2. This is the internal vehicle model employed by the driver control algorithm when steering an articulated vehicle. The sum of lateral forces and sum of yaw torques acting on each of the two articulated masses produces the following four dynamical equations (primes denoting differentiation with respect to time):

$$m_1 v_1' = Fy_1 + Fy_2 - m_1 U r_1 + f_y \qquad \text{(A-1)}$$

(f_y is the lateral hitch constraint force)

$$I_1 r_1' = a_1 Fy_1 - b_1 Fy_2 - c_1 f_y + M_c \qquad \text{(A-2)}$$

$$m_2 v_2' = Fy_2 - m_2 U r_2 - f_y \qquad \text{(A-3)}$$

$$I_2 r_2' = - b_2 Fy_3 - c_2 f_y - M_c \qquad \text{(A-4)}$$

The kinematic constraint for the articulation joint produces the following algebraic relationship between the lateral displacement, y_i, and heading angle variables, ψ_i :

$$y_1 - c_1 \sin \psi_1 = y_2 + c_2 \sin \psi_2 \ \psi \qquad \text{(A-5)}$$

Differentiating twice and assuming small angles for the heading angles results in the equivalent constraint equation expressed in terms of the associated accelerations:

$$v_1' + U r_1 - c_1 r_1' = v_2' + U r_2 + c_2 r_2' \qquad \text{(A-6)}$$

The tire forces Fy_i can be expressed in terms of the tire cornering stiffnesses and tire sideslip angles as:

$$Fy_i = - C_{\alpha i} \alpha_i \qquad \text{(A-7)}$$

where,

$$\alpha_1 = \tan^{-1}[(v_1 + a_1 r_1)/U] - \delta \qquad \text{(A–8)}$$

$$\alpha_2 = \tan^{-1}[(v_1 - b_1 r_1)/U] \qquad \text{(A–9)}$$

$$\alpha_3 = \tan^{-1}[(v_2 - b_2 r_2)/U] \qquad \text{(A–10)}$$

Assuming small tire sideslip angles for the α_i (replacing the arctan by the angle), Equations (A-1) -> (A-4) become after these substitutions:

$$m_1 v_1' = - C_{\alpha 1}(v_1+a_1r_1)/U - C_{\alpha 2}(v_1-b_1r_1)/U + C_{\alpha 1}\delta - m_1 U r_1 + f_y \qquad \text{(A-11)}$$

$$I_1 r_1' = -a_1 C_{\alpha 1}(v_1+a_1r_1)/U + b_1 C_{\alpha 2}(v_1-b_1r_1)/U - c_1 f_y + a_1 C_{\alpha 1}\delta + M_c \qquad \text{(A-12)}$$

$$m_2 v_2' = - C_{\alpha 3}(v_2-b_2r_2)/U - m_2 U r_2 - f_y \qquad \text{(A-13)}$$

$$I_2 r_2' = b_2 C_{\alpha 3}(v_2-b_2r_2)/U - c_2 f_y - M_c \qquad \text{(A-14)}$$

Expressed in matrix algebra terminology, the equations of motion (A-11) -> (A-14) become:

$$\mathbf{M}\,\mathbf{v'} = \mathbf{A}\,\mathbf{v} + \mathbf{G}\,\delta + \mathbf{N}\,f_y + \mathbf{E}\,M_c \qquad \text{(A-15)}$$

and the kinematic constraint equation (A-6) becomes:

$$\mathbf{C}\,\mathbf{v'} = \mathbf{D}\,\mathbf{v} \qquad \text{(A-16)}$$

where, $v = \{ v_1, r_1, v_2, r_2 \}^T$.

Solving (A-15) and (A-16) for the constraint force, f_y, and upon back substitution, results in the following set of dynamical equations free of the constraint force:

$$\mathbf{v'} = \mathbf{F^*}\,\mathbf{v} + \mathbf{g^*}\,\delta + \mathbf{h}\,M_c \qquad \text{(A-17)}$$

For a powered articulation scheme with $M_c = K[\delta - \eta(\psi_1 - \psi_2)]$, where the parameter K controls the degree to which the articulation torque is slaved to the front wheel steer angle, δ, and η provides an optional torsional spring effect about the articulation joint, equations (A-17) become:

$$\mathbf{v}' = \mathbf{F}^* \mathbf{v} + (\mathbf{g}^* + K\,\mathbf{h})\,\delta + \psi_i \text{ terms} \qquad \text{(A-18)}$$

Addition of the lead unit lateral displacement, y_1, and heading angle state variable, ψ_i, equations, results in the final set of seven linear dynamical equations:

$$\begin{bmatrix} y_1' \\ \psi_1' \\ v_1' \\ r_1' \\ v_2' \\ r_2' \\ \psi_2' \end{bmatrix} = \begin{bmatrix} 0 & U & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & & & & & & \\ 0 & & & & & & \\ 0 & \{-\eta K \mathbf{h}\} & & & \mathbf{F}^*_{4x4} & & \{\eta K \mathbf{h}\} \\ 0 & & & & & & \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} y_1 \\ \psi_1 \\ v_1 \\ r_1 \\ v_2 \\ r_2 \\ \psi_2 \end{bmatrix}$$

$$+ \begin{bmatrix} 0 \\ 0 \\ \mathbf{g}^* + K\,\mathbf{h} \\ 0 \end{bmatrix} \delta \qquad \text{(A-19)}$$

or in matrix algebra notation:

$$\{X'\}_{7x1} = [F]_{7x7}\,\{X\}_{7x1} + \{g\}_{7x1}\,\delta_{1x1} \qquad \text{(A-20)}$$

where,

$$\mathbf{M} = \begin{bmatrix} m_1 & & 0 & \\ & I_1 & & \\ & & m_2 & \\ & 0 & & I_2 \end{bmatrix}$$

$$\mathbf{A} = \begin{bmatrix} -(C_{a1}+C_{a2})/U & (-C_{a1}a_1+C_{a2}b_1)/U-m_1U & 0 & 0 \\ (-a_1C_{a1}+b_1C_{a2})/U & -(C_{a1}a_1^2+C_{a2}b_1^2)/U & 0 & 0 \\ 0 & 0 & -C_{a3}/U & -b_2C_{a3}/U-m_2U \\ 0 & 0 & b_2C_{a3}/U & -C_{a3}b_2^2/U \end{bmatrix}$$

$$\mathbf{G} = \begin{bmatrix} C_{a1} \\ a_1C_{a1} \\ 0 \\ 0 \end{bmatrix} \qquad \mathbf{N} = \begin{bmatrix} 1 \\ -C_1 \\ -1 \\ -C_2 \end{bmatrix} \qquad \mathbf{E} = \begin{bmatrix} 0 \\ 1 \\ 0 \\ -1 \end{bmatrix}$$

$$\mathbf{C} = \{ \quad 1, \quad -C_1, \quad -1, \quad -C_2 \quad \}$$

$$\mathbf{D} = \{ \quad 0, \quad -U, \quad 0, \quad U \quad \}$$

and,

$$F^* = M^{-1}(A + N[C M^{-1} N]^{-1}(D - C M^{-1} A)) \qquad \text{(A-21)}$$

$$g^* = M^{-1}(G - N[C M^{-1} N]^{-1} C M^{-1} G) \qquad \text{(A-22)}$$

$$h = M^{-1}(E - N[C M^{-1} N]^{-1} C M^{-1} E) \qquad \text{(A-23)}$$

Appendix B

Vehicle / Driver Test Plan

Task A -- Instrumenting and Preparation of the Test Vehicle

At the present time, it is expected that the initial test vehicle will be the four-wheeled HMWWV. The test vehicle will be instrumented with transducers and the UMTRI data acquisition package to measure the following signals and vehicle responses:

- Lateral Acceleration
- Vehicle Velocity
- Yaw Rate
- Roll Rate
- Front Wheel Steer Angles
- Steering Wheel Angle
- Longitudinal Acceleration

and possibly,

- Pitch Rate
- Vehicle Sideslip Angle

The UMTRI stable platform will be used to measure the lateral and longitudinal accelerations as well as the yaw and roll rates. A conventional fifth wheel will be used to measure forward velocity. Front wheel angles will be measured by linear potentiometers;

steering wheel angle by a rotary potentiometer. If available at the time of the tests, the MIRA trolley currently stored at UMTRI by FHWA will be used to measure vehicle sideslip response.

Additional transducers and measurement devices may be provided by TACOM.

Task B -- Vehicle Weight & Length Measurements

The test vehicle will be weighed in its test condition (with instrumentation and two passengers) to obtain front and rear tire loads and total weight. Estimates of yaw, pitch, and roll inertias will be estimated based upon past practice, if not available from previous measurements of the same vehicle. Likewise, center of gravity height will also be estimated, if not available from previous measurements. Measurements of wheelbase, wheel track, suspension locations, and overall geometry will also be performed.

Task C -- Tire Force Measurements

One tire from the test vehicle will be tested on the UMTRI flat-bed tire test machine to obtain lateral tire force measurements at three different loads (test load, 50% test load, 150% test load) and six slip angles (-1, 0, +1, 2, 4, 8 degrees). Tire cornering stiffness parameters needed by the driver model in subsequent model/test validation activities, as well as complete lateral tire force representation within the DADS model, will be based upon these measurements.

Task D -- Full Scale Vehicle Tests

Three basic maneuvers are planned for the test program. The first test maneuver is simple steady turning with a fixed steering angle (open-loop). The purpose of this test is to obtain open-loop estimates of the vehicle understeer and basic cornering properties. The second maneuver is similar to the first, but conducted in a closed-loop manner using the test driver to track a turn of fixed radius. Finally, the third maneuver will be a standard driver-controlled 12-foot lane change maneuver. The driver-controlled steady turning and lane-change maneuvers should provide adequate data for validating and comparing the basic driver model responses within the DADS model.

All of the tests will be conducted at the Chrysler Proving Grounds at Chelsea, Michigan.

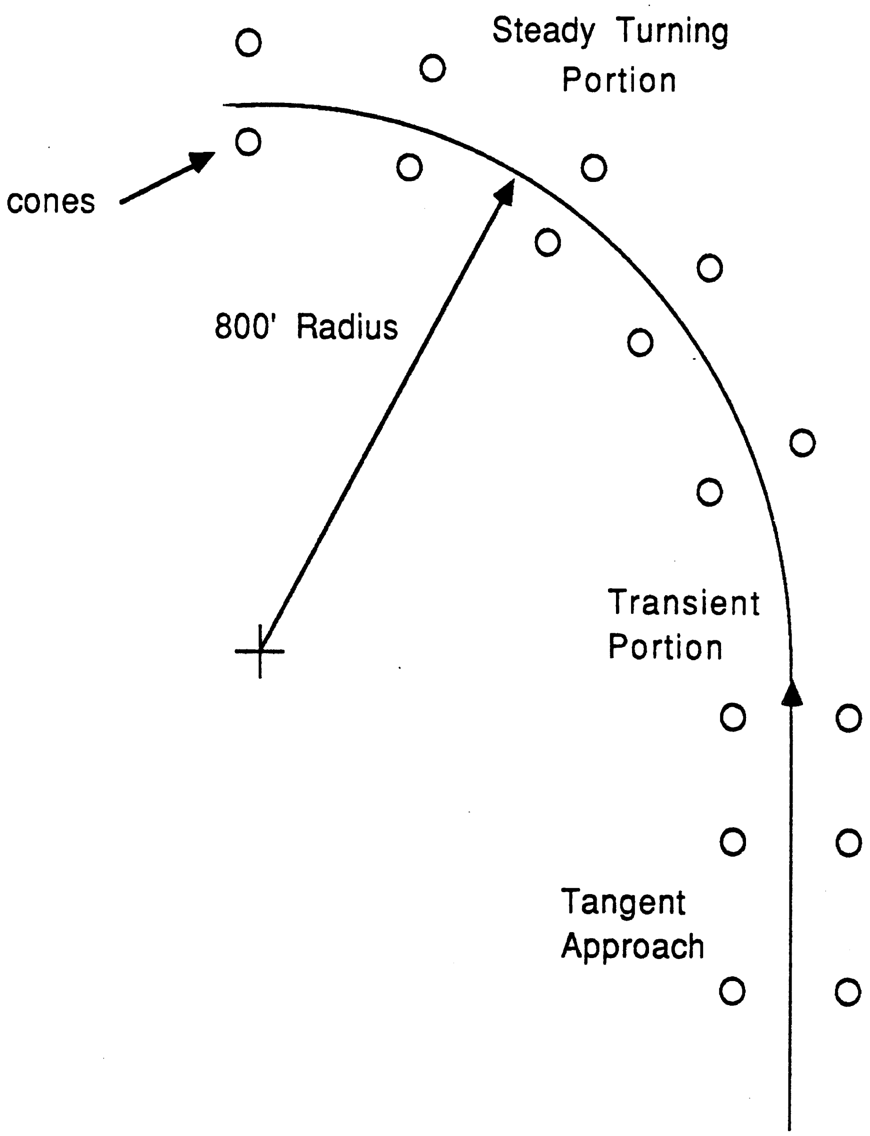
- Constant Velocity Fixed Steer Tests

Fixed steering wheel angle tests will be conducted at speeds of 30 and 60 mph for several different steer angles, concentrating primarily on lateral accelerations levels within the linear regime response of the vehicle. If possible, the vehicle will be equipped with a sideslip trolley device for measuring vehicle sideslip angle. (The trolley referred to is currently on loan to UMTRI from FHWA and may be available for the TACOM vehicle tests next year.) The sideslip measurements will augment the conventional steady-turning test results by providing a back-up means for estimating understeer characteristics of the vehicle. Since both front wheel angles and steering wheel angle measurements will be gathered, some idea of effective steering gear ratio and steering system properties will also be available, --- short of measuring the steering system itself.

- Driver-Controlled Constant Radius Turning Tests

These tests will be conducted also at 30 and 60 mph but with the test driver attempting to track a cone-marked turn of fixed radius. A radius of 800 feet will produce a lateral acceleration of 0.3 g's at 60 mph and can easily be accommodated on the Chrysler skid pad. The maneuver will begin by having the driver approach the circular turn along a straight tangent and then tracking the fixed radius curve at more or less constant speed. See Figure 1. Transient driver/vehicle response information due to entering the curve, as well as steady-state driver/vehicle response information due to tracking the curve, will be gathered from these tests. Influence of forward speed upon system damping will be obtained by conducting the same tests at the two different speeds.

Figure 1. Driver-Controlled Fixed Path Radius Turning Test



- Driver-Controlled Lane Change Maneuver

The purpose of this test is to gather transient driver/vehicle response data for a routine maneuver under relatively normal roadway operating conditions. The nominal maneuver consists of performing a 12-foot lane change (marked by a coned course) at forward speeds of 30 and 60 mph. See Figure 2. The 60 mph lane change geometry (shoot-to-shoot forward travel distance, L) will be adjusted to provide approximately 0.3 g's of peak lateral acceleration during the maneuver. The course geometry will remain fixed for the 30 mph tests in order to study the influence of forward speed upon driver preview and system damping.

Data Acquisition Equipment

Test data will be collected using the UMTRI portable data acquisition system. The system consists of a Texas Instruments TM 990 microprocessor, signal-conditioning units, programmable filters, and analog/digital converters. A CRT unit and keyboard are used to operate and control the system. Data are stored on high capacity digital tape cartridges for subsequent post-processing. Simple statistical calculations and background calibrations can be performed as well.

Test Schedule

Figure 3 shows the nominal schedule to be followed for completing the vehicle tests during the spring of 1987. Tasks A and B (vehicle instrumentation & preparation; weight & length measurements) will begin in March. Approximately 2-3 weeks of time should be allotted to these tasks to allow for possible UMTRI staff technician commitments to other projects. Task C (flat-bed tire tests) will be scheduled as well during this same time frame, however, an additional week is provided to account for possible staff conflicts. Finally, Task D (vehicle testing) is allocated one month of time during April, allowing for weather and other possible unanticipated problems.

Figure 2. Driver-Controlled Lane Change Maneuver

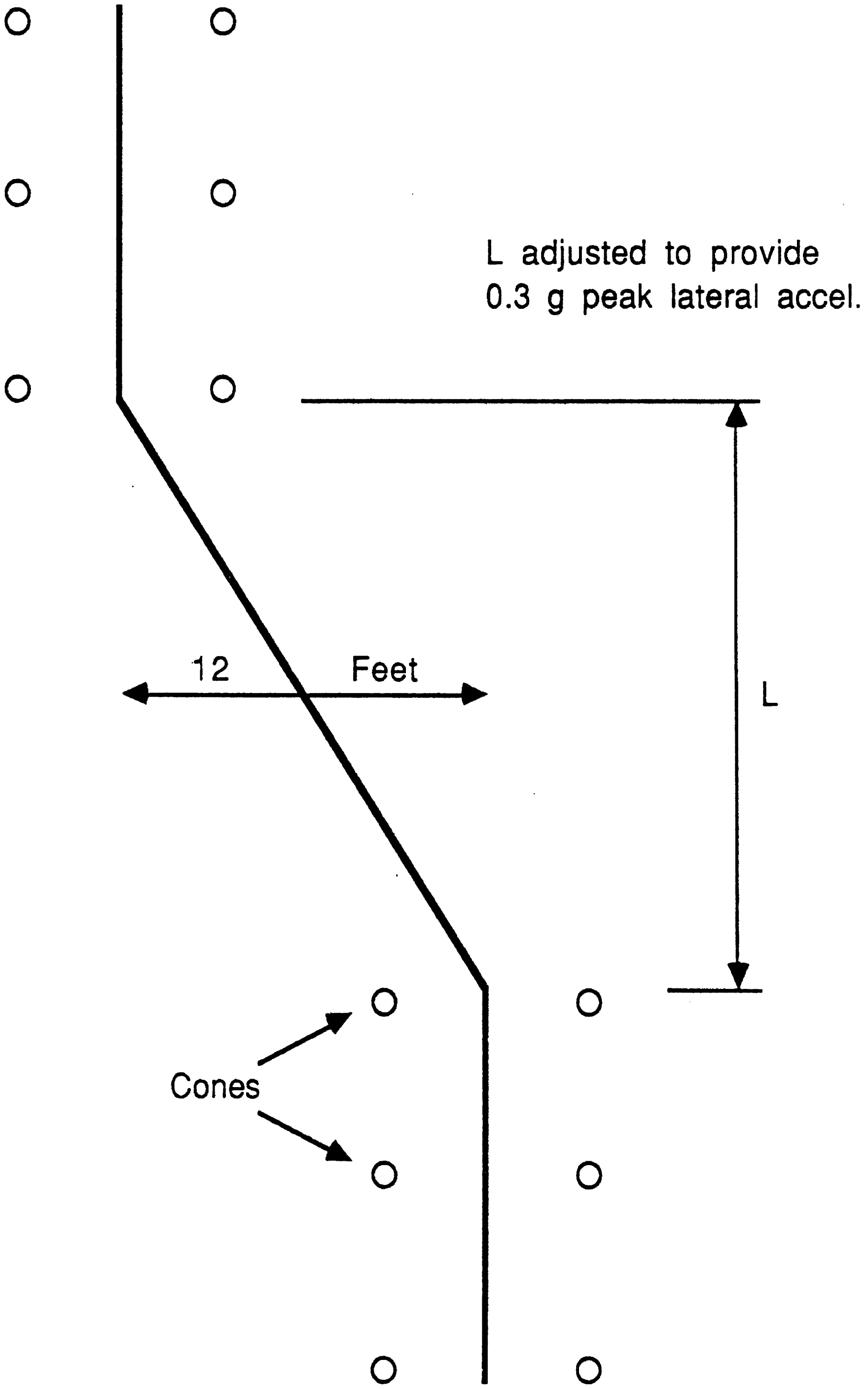
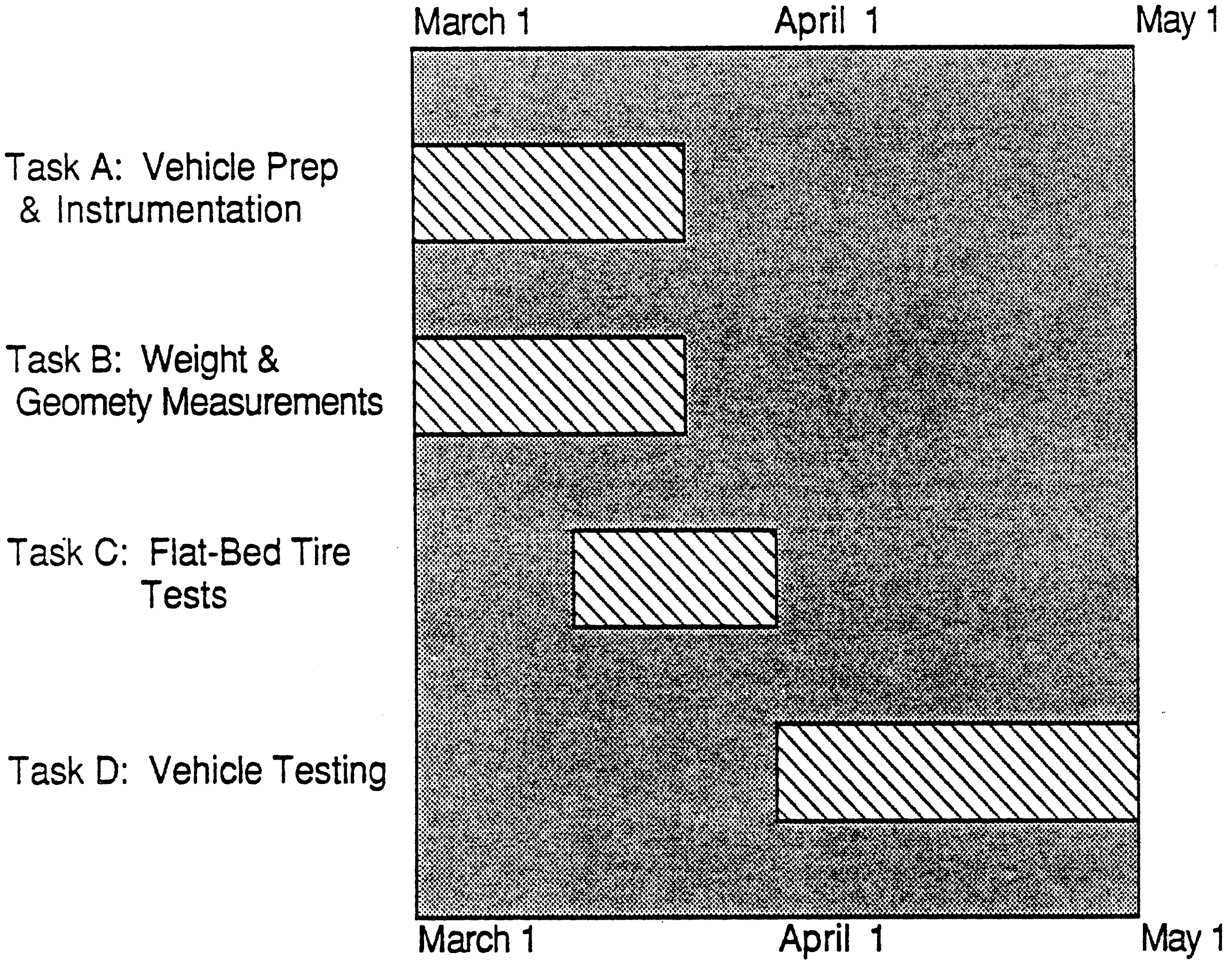


Figure 3. TACOM Test Schedule / Spring 1987



Appendix C

Single-Unit and Articulated Vehicle Driver Model Program Listings

This appendix contains FORTRAN 77 listings of the two basic driver models developed to date under this project. The principal difference between the two is only in the complexity of the internal vehicle model used by the driver model when steering different types of vehicles. The articulated vehicle version of the driver model (Listing 2) equips the simulated controller with a more enhanced understanding of the dynamics of the vehicle being controlled. (A less accurate, though frequently used, alternative is to allow it to ignore the direct dynamic effects of the rear unit.) The COMMON statements appearing in these listings are from the UMTRI Phase 4 vehicle model used to originally check and verify the developed code. These statements are replaced by equivalent statements in the DADS program code and are used to exchange information between the main vehicle model and the driver subroutines.

Listing 1

```
C
C   *** Single-Unit Driver Model Subroutine ***
C
C  Initialization
C
      SUBROUTINE DRIVE1(ISTEER)
      INTEGER R, W
      COMMON /PAGE/ NPG, HEAD(20), R, W
      COMMON /DRIV/ CAF, CAR,  WF, WR, U
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1        TFF, RM, A, B, RI, PSI0, NTF, NP, TLAST, DFWLST, TILAST,
     2        DMEM(100,2), XT(100), YT(100)
C
      NP = -ISTEER
      GRAV = 32.16666
      TICYCL = 0.0100
      TSS = 0.0
      DMAX = 2.0
C
      WRITE (W,10)
   10 FORMAT ('0', T20, 'CLOSED-LOOP PATH FOLLOWING MODE', /, '0', T20,
     1        'X-Y ', 'PATH', ' COORDINATES :', /, '0', T50, 'X', T60,
     2        'Y', /, /'0', T47, '(FEET)', T57, '(FEET)')
C
   20 FORMAT (I3)
      DO 40 J = 1, NP
        READ (R,30) XP(J), YP(J)
   30   FORMAT (2F10.2)
        WRITE (W,50) XP(J), YP(J)
   40 CONTINUE
   50 FORMAT (' ', T43, 2F10.2)
      READ (R,60) TAUMEM, TFF
   60 FORMAT (F10.4)
      WRITE (W,70) TAUMEM, TFF
   70 FORMAT (' ', /, ' ', T20, 'DRIVER TRANSPORT LAG (SEC) :', T60,
     1        F4.2, /, ' ', T20, 'END OF PREVIEW INTERVAL (SEC) :', T60,
     2        F4.2/)
      RETURN
      END
C
C Initialize Internal Vehicle Model Parameters
C
      SUBROUTINE DRIVE2(VEL)
      COMMON /DRIV/ CAF, CAR,  WF, WR, U
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1        TFF, RM, A, B, RI, PSI0, NTF, NP, TLAST, DFWLST, TILAST,
     2        DMEM(100,2), XT(100), YT(100)
      COMMON /SLOPES/ SLOPEY(4,2,2,2)
      COMMON /KEY/ MVEH, KEY(4,2), KDOLLY(4)
COMMON /SPMASS/ WHBS(4), BB1(4), A3(4), APHI(4), DELTA(4), VW(4),
     1        VJ(4,3), PW(4), PX(4), PZ(4), PJ(4,3), SNL(4,2), D(4),
     2        PH(4), TOL(4), MC5
      COMMON /FCTOUT/ XBAR(4,3), PHIBAR(4,3), UBAR(4,3), PBAR(4,3)
      COMMON /STATIC/ NS(4,2,2), FT(4), SF(4,2,2,2)
      DIMENSION  TALIGN(4,2,2,2)
      DIMENSION TOR(4,2,2,2), SRS(4,2,2,2), XXS(4,2,2), DT(4,2,2,2),
```

```
     1        DERY(112)
      DATA XXS /16*0./
      DATA SRS /32*0./
      DATA DERY /112*0./
      UBAR(1,1) = VEL
      U = VEL
      WF = NS(1,1,1)
      WR = NS(1,2,1)
      CALL TIRE(1, 1, 1, 1, 2, TOR, DT, SRS, XXS, DERY, TALIGN, 0.)
      CAF = SLOPEY(1,1,1,1) * NS(1,1,1) / 2.
      CALL TIRE(1, 2, 1, 1, 2, TOR, DT, SRS, XXS, DERY, TALIGN, 0.)
      CAR = SLOPEY(1,2,1,1) * NS(1,2,1) / 2.
      IF (KEY(1,2) .LE. 0) GO TO 70
      CALL TIRE(1, 2, 2, 1, 2, TOR, DT, SRS, XXS, DERY, TALIGN, 0.)
      CAR = CAR + SLOPEY(1,2,2,1) * NS(1,2,2) / 2.
      WR = WR + NS(1,2,2)
 70   RM = (WF + WR) / GRAV
      B = WHBS(1) * WF / (WF + WR)
      A = WHBS(1) - B
      RI = A * B * RM
      PSI0 = 0.0
      NTF = 10
      DO 80 J = 1, NP
        XT(J) = XP(J) * COS(PSI0) + YP(J) * SIN(PSI0)
        YT(J) = -XP(J) * SIN(PSI0) + YP(J) * COS(PSI0)
   80 CONTINUE
      TLAST = 0.
      DFWLST = 0.
      TILAST = 0.
      DFW = 0.
      DO 90 I = 1, 100
        DMEM(I,1) = 0.
   90 DMEM(I,2) = -1.
      RETURN
      END
C
C  Transition Matrix Calculation
C
      SUBROUTINE TRANS
      INTEGER R, W
      COMMON /PAGE/ NPG, HEAD(20), R, W
      COMMON /DRIV/ CAF, CAR,  WF, WR, U
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1       TFF, RM, A, B, RI, PSI0, NTF, NP, TLAST, DFWLST, TILAST,
     2       DMEM(100,2), XT(100), YT(100)
      COMMON /TRSSTR/ TTT(4,4,10), TTT1(4,4,10), G(4)
      DIMENSION SV(4), SD(4), SVI(4)
C
      DELT = 0.01
      A1 = -2. * (CAF + CAR) / RM / U
      B1 = 2. * (CAR*B - CAF*A) / RM / U - U
      A2 = 2. * (CAR*B - CAF*A) / RI / U
      B2 = -2. * (CAR*B*B + CAF*A*A) / RI / U
      C1 = 2. * CAF / RM
      C2 = 2. * CAF / RI * A
      ULAST = U
      G(1) = 0.
      G(2) = C1
```

```
      G(3) = C2
      G(4) = 0.
      DO 70 J = 1, 4
        NBEG = TSS / DELT + 1
        NEND1 = (TFF + .001 - TSS) / NTF / DELT
        NENDV = NEND1
        DO 10 L = 1, 4
          SV(L) = 0.0
          SVI(L) = 0.0
   10   CONTINUE
        TIME = 0.
        SV(J) = 1.0
        DO 60 I = 1, NTF
          DO 40 K = NBEG, NENDV
            SD(1) = SV(2) + U * SV(4)
            SD(2) = A1 * SV(2) + B1 * SV(3)
            SD(3) = A2 * SV(2) + B2 * SV(3)
            SD(4) = SV(3)
            DO 20 L = 1, 4
              SV(L) = SV(L) + SD(L) * DELT
   20       CONTINUE
            TIME = TIME + DELT
            DO 30 L = 1, 4
              SVI(L) = SVI(L) + SV(L) * DELT
   30       CONTINUE
   40     CONTINUE
          DO 50 L = 1, 4
            TTT(L,J,I) = SV(L)
            TTT1(L,J,I) = SVI(L)
   50     CONTINUE
          NBEG = NBEG + NEND1
          NENDV = NENDV + NEND1
   60   CONTINUE
   70 CONTINUE
      RETURN
      END
C
C  Closed-Loop Steer Calculation
C
      SUBROUTINE STEER(X, Y, DFW, DFWNOW)
      INTEGER R, W
      COMMON /PAGE/ NPG, HEAD(20), R, W
      COMMON /FCTOUT/ XBAR(4,3), PHIBAR(4,3), UBAR(4,3), PBAR(4,3)
      COMMON /DRIV/ CAF, CAR,  WF, WR, U
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1       TFF, RM, A, B, RI, PSI0, NTF, NP, TLAST, DFWLST, TILAST,
     2       DMEM(100,2), XT(100), YT(100)
      COMMON /TRSSTR/ TTT(4,4,10), TTT1(4,4,10), G(4)
      DIMENSION Y(5), YC(5)
      DIMENSION DUMV11(4)
      DIMENSION DUMV1(4), VECM(4)
      DIMENSION DUMM1(4,4), DUMM2(4,4)
      DATA VECM /1.0, 3*0.0/
C
      T = X
      IF(T .GT. TLAST) GO TO 5
      TLAST = T
      RETURN
```

```
  5   U = UBAR(1,1)
      EPSI = ABS(Y(4) - PSI0)
      DO 10 I = 1, 5
   10 YC(I) = Y(I)
      IF (EPSI .LE. .02) GO TO 30
C
C  Update Coordinate Transformation
C
      PSI0 = Y(4)
      DO 20 J = 1, NP
        XT(J) = XP(J) * COS(PSI0) + YP(J) * SIN(PSI0)
   20 YT(J) = -XP(J) * SIN(PSI0) + YP(J) * COS(PSI0)
C
   30 Y0 = -Y(5) * SIN(PSI0) + Y(1) * COS(PSI0)
      X0 = Y(5) * COS(PSI0) + Y(1) * SIN(PSI0)
      YC(1) = Y0
      YC(4) = Y(4) - PSI0
      EPSY2 = 0.
      TSUM = 0.
      SSUM = 0.
      IF (T - TILAST .LE. TICYCL) RETURN
      DO 50 I = 1, NTF
        TJI = (TFF - TSS) / NTF * I + TSS
        DO 40 J = 1, 4
          DO 40 K = 1, 4
            DUMM1(J,K) = TTT1(J,K,I)
   40   DUMM2(J,K) = TTT(J,K,I)
        CALL GMPRD(VECM, DUMM1, DUMV11, 1, 4, 4)
        CALL GMPRD(VECM, DUMM2, DUMV1, 1, 4, 4)
        CALL GMPRD(DUMV1, YC, T1, 1, 4, 1)
        XCAR = X0 + U * TJI
        CALL TRAJ(XCAR, XT, YT, YPATH, I)
        CALL GMPRD(DUMV11, G, S1, 1, 4, 1)
        EP = T1 + S1 * DFWNOW - YPATH
        TSUM = TSUM + EP * S1
        SSUM = SSUM + S1 * S1
        EPSY2 = EPSY2 + EP * EP * (TFF - TSS) / NTF
   50 CONTINUE
      EPSY = SQRT(EPSY2) / (TFF - TSS)
      DFW = -TSUM / SSUM + DFW
      IF (ABS(DFW) .GT. DMAX) DFW = DMAX * SIGN(1.,DFW)
      DO 60 J = 1, 2
        DO 60 I = 1, 99
   60 DMEM(101 - I,J) = DMEM(100 - I,J)
      DMEM(1,1) = DFW
      DMEM(1,2) = T
      TTAB = T - TAUMEM
      DO 70 I = 1, 99
        IF (DMEM(I + 1,2) .LE. TTAB .AND. DMEM(I,2) .GE. TTAB)
     1     GO TO 90
   70 CONTINUE
      WRITE (W,80)
   80 FORMAT ('0', '***** TAUMEM PROBABLY TOO LARGE *****')
      CALL EXIT
   90 DFW = DMEM(I,1)
      TLAST = X
      TILAST = X
      RETURN
      END
```

Listing 2

```
C  ***  Articulated Vehicle Driver Model  ***
C  Subroutine Drive1 -  Read & Echo Driver Model Parameters
C
      SUBROUTINE DRIVE1(DFW)
      INTEGER R, W
      COMMON /INOUT/ R, W
      COMMON /DRIV/ CF1, CF2, CF3, WHBS1, WHBS2, W1, W2, W3, W4, RM1,
     1       RM2, RI1, RI2
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1       TFF, PSI0, NTF, NP, TLAST, DFWLST, TILAST, A1, B1, A2, B2,
     2       C1, C2, DMEM(100,2), XT(100), YT(100)
C
      GRAV = 32.16666
      TICYCL = 0.0100
      TSS = 0.0
      DMAX = 2.0
C
      WRITE (W,10)
   10 FORMAT ('0', T20, 'CLOSED-LOOP PATH FOLLOWING MODE', /, '0', T20,
     1       'X-Y ', 'PATH', ' COORDINATES :', /, '0', T50, 'X', T60,
     2       'Y', /, /'0', T47, '(FEET)', T57, '(FEET)')
C
      READ (R,20) NP
   20 FORMAT (I3)
      DO 40 J = 1, NP
        READ (R,30) XP(J), YP(J)
   30   FORMAT (2F10.2)
        WRITE (W,50) XP(J), YP(J)
   40 CONTINUE
   50 FORMAT (' ', T43, 2F10.2)
      READ (R,60) TAUMEM, TFF
   60 FORMAT (F10.4)
      WRITE (W,70) TAUMEM, TFF
   70 FORMAT (' ', /, ' ', T20, 'DRIVER TRANSPORT LAG (SEC) :', T60,
     1       F4.2, /, ' ', T20, 'END OF PREVIEW INTERVAL (SEC) :', T60,
     2       F4.2/)
      PSI0 = 0.0
      NTF = 10
      DO 80 J = 1, NP
        XT(J) = XP(J) * COS(PSI0) + YP(J) * SIN(PSI0)
        YT(J) = -XP(J) * SIN(PSI0) + YP(J) * COS(PSI0)
   80 CONTINUE
      TLAST = 0.
      DFWLST = 0.
      TILAST = 0.
      DFW = 0.
      DO 90 I = 1, 100
        DMEM(I,1) = 0.
   90 DMEM(I,2) = -1.
      RETURN
      END
C
C   Subroutine Drive2 - Initialization of Internal Vehicle Model
C   (Articulated Vehicle)
C
      SUBROUTINE DRIVE2(VEL)
```

```
      DIMENSION DUMM1(4,4), DUMM2(4,4), DUMV1(4), DUMV2(4), DUMV9(4)
      DIMENSION LW(4), MW(4)
      DIMENSION FP(4,4), HV(4), GPV(4)
      COMMON /BLOCK1/ A(4,4), GV(4), U, OPTION, CCCC
      COMMON /FILT/ FILT1, FILT2, FILT3, FILT4
      COMMON /BLOCK8/ FF(7,7), GG(7)
      COMMON /DRIV/ CF1, CF2, CF3, WHBS1, WHBS2, W1, W2, W3, W4, RM1,
     1       RM2, RI1, RI2
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1       TFF, PSI0, NTF, NP, TLAST, DFWLST, TILAST, A1, B1, A2, B2,
     2       C1, C2, DMEM(100,2), XT(100), YT(100)
      DIMENSION RM(4,4), RN(4), EV(4), CV(4), DV(4)
C
C  Obtain Vehicle Parameters for Simplified Internal Vehicle Model
C
      DATA RNU, RKMOM /0.0, 0.0/
      DATA ETA /0.0/
      FILT1 = 1.0
      FILT2 = 0.0
      FILT3 = 1.0
      FILT4 = 0.0
C
      NDIM = 8
      DFW = 0.0
      A1 = W2 / (W1 + W2)
      B1 = WHBS1 - A1
      A2 = W4 / (W3 + W4)
      B2 = WHBS2 - A2
      RM1 = (W1 + W2) / GRAV
      RM2 = (W3 + W4) / GRAV
      RI1 = A1 * B1 * RM1
      RI2 = A2 * B2 * RM2
C
C  CALL TIRE ROUTINE TO GET CF1,...,3
C
      CF1 = 1.
      CF2 = 1.
      CF3 = 1.
      DO 10 I = 1, 7
        GG(I) = 0.
        DO 10 J = 1, 7
          FF(I,J) = 0.
   10 CONTINUE
      CALL SMPY(RM, 0., RM, 4, 4, 0)
      CALL SMPY(A, 0., A, 4, 4, 0)
      CALL SMPY(GV, 0., GV, 4, 1, 0)
      CALL SMPY(EV, 0., EV, 4, 1, 0)
      CALL SMPY(DV, 0., DV, 4, 1, 0)
      CALL SMPY(RN, 0., RN, 4, 1, 0)
      CALL SMPY(HV, 0., HV, 4, 1, 0)
      CALL SMPY(GPV, 0., GPV, 4, 1, 0)
      RM(1,1) = RM1
      RM(2,2) = RI1
      RM(3,3) = RM2
      RM(4,4) = RI2
      A(1,1) = -2 * (CF1 + CF2) / U
      A(1,2) = 2 * (CF2*B1 - CF1*A1) / U - RM1 * U
      A(2,1) = 2 * (CF2*B1 - CF1*A1) / U
```

```
      A(2,2) = -2 * (CF2*B1*B1 + CF1*A1*A1) / U
      A(3,3) = -2. * CF3 / U
      A(3,4) = 2. * CF3 * B2 / U - RM2 * U
      A(4,3) = 2. * B2 * CF3 / U
      A(4,4) = -2. * CF3 * B2 * B2 / U
      GV(1) = 2. * (CF1 + ETA*CF2) * FILT1 + FILT2
      GV(2) = 2. * (A1*CF1 - ETA*B1*CF2) * FILT3 + FILT4
      GV(3) = 0.0
      GV(4) = 0.0
      RN(1) = 1.
      RN(2) = -C1
      RN(3) = -1.
      RN(4) = -C2
      EV(2) = 1.
      EV(4) = -1.
      CV(1) = 1.
      CV(2) = -C1
      CV(3) = -1.
      CV(4) = -C2
      DV(2) = -U
      DV(4) = U
      CALL MINV(RM, 4, DET, LW, MW)
      CALL GMPRD(CV, RM, DUMV9, 1, 4, 4)
      CALL GMPRD(DUMV9, RN, SC, 1, 4, 1)
      SC1 = 1. / SC
      CALL GMPRD(DUMV9, A, DUMV1, 1, 4, 4)
      CALL GMSUB(DV, DUMV1, DUMV2, 1, 4)
      CALL GMPRD(RN, DUMV2, DUMM1, 4, 1, 4)
      CALL SMPY(DUMM1, SC1, DUMM1, 4, 4, 0)
      CALL GMADD(A, DUMM1, DUMM2, 4, 4)
      CALL GMPRD(RM, DUMM2, FP, 4, 4, 4)
C
   20 CONTINUE
      CALL GMPRD(DUMV9, GV, SC2, 1, 4, 1)
      SC3 = SC1 * SC2
      CALL SMPY(RN, SC3, DUMV1, 4, 1, 0)
      CALL GMSUB(GV, DUMV1, DUMV2, 4, 1)
      CALL GMPRD(RM, DUMV2, GPV, 4, 4, 1)
C
      CALL GMPRD(DUMV9, EV, SC4, 1, 4, 1)
      SC5 = SC1 * SC4
      CALL SMPY(RN, SC5, DUMV1, 4, 1, 0)
      CALL GMSUB(EV, DUMV1, DUMV2, 4, 1)
      CALL GMPRD(RM, DUMV2, HV, 4, 4, 1)
C
      FF(1,2) = U
      FF(1,3) = 1.
      FF(2,4) = 1.
      FF(7,6) = 1.
      DO 30 I = 1, 4
        DO 30 J = 1, 4
          FF(I + 2,J + 2) = FP(I,J)
   30 CONTINUE
      DO 40 I = 1, 4
        FF(I + 2,2) = -RNU * RKMOM * HV(I)
        FF(I + 2,7) = RNU * RKMOM * HV(I)
        GG(I + 2) = GPV(I) + RKMOM * HV(I)
   40 CONTINUE
```

```
C
      RETURN
      END
C
C  Transition Matrix Calculation
C
      SUBROUTINE TRANS
      INTEGER R, W
      COMMON /BLOCK8/ FF(7,7), GG(7)
      COMMON /INOUT/ R, W
      COMMON /DRIV/ CAF, CAR, WHBS, WF, WR, U
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1       TFF, RM, A, B, RI, PSI0, NTF, NP, TLAST, DFWLST, TILAST,
     2       DMEM(100,2), XT(100), YT(100)
      COMMON /TRSSTR/ TTT(7,7,10), TTT1(7,7,10), G(7)
      DIMENSION SV(7), SD(7), SVI(7)
C
      DO 10 I = 1, 7
        G(I) = GG(I)
   10 CONTINUE
      TLAST = 0.
      TILAST = 0.
      DFW = 0.
      DFWNOW = 0.0
      DELT = 0.01
      ULAST = U
      DO 80 J = 1, 7
        NBEG = TSS / DELT + 1
        NEND1 = (TFF + .001 - TSS) / NTF / DELT
        NENDV = NEND1
        DO 20 L = 1, 7
          SV(L) = 0.0
          SVI(L) = 0.0
   20   CONTINUE
        TIME = 0.
        SV(J) = 1.0
        DO 70 I = 1, NTF
          DO 50 K = NBEG, NENDV
            CALL GMPRD(FF, SV, SD, 7, 7, 1)
            DO 30 L = 1, 7
              SV(L) = SV(L) + SD(L) * DELT
   30       CONTINUE
            TIME = TIME + DELT
            DO 40 L = 1, 7
              SVI(L) = SVI(L) + SV(L) * DELT
   40       CONTINUE
   50     CONTINUE
          DO 60 L = 1, 7
            TTT(L,J,I) = SV(L)
            TTT1(L,J,I) = SVI(L)
   60     CONTINUE
          NBEG = NBEG + NEND1
          NENDV = NENDV + NEND1
   70   CONTINUE
   80 CONTINUE
      RETURN
      END
C
```

```
C  Subroutine Steer - Closed-Loop Steering Calculation
C
      SUBROUTINE STEER(X, Y, DFW, DFWNOW)
      INTEGER R, W
      COMMON /BLOCK8/ FF(7,7), GG(7)
      COMMON /INOUT/ R, W
      COMMON /DRIV/ CAF, CAR, WHBS, WF, WR, U
      COMMON /DRVST1/ GRAV, TICYCL, TSS, DMAX, XP(100), YP(100), TAUMEM,
     1       TFF, RM, A, B, RI, PSI0, NTF, NP, TLAST, DFWLST, TILAST,
     2       DMEM(100,2), XT(100), YT(100)
      COMMON /TRSSTR/ TTT(7,7,10), TTT1(7,7,10), G(7)
      DIMENSION DUMV11(7), DUMV1(7), VECM(7), DUMM1(7,7), DUMM2(7,7)
      DIMENSION YC(8), Y(8)
      DATA VECM /1.0, 6*0.0/
C
      IF (X - TLAST .GT. .001) GO TO 10
      TLAST = X
      RETURN
   10 CONTINUE
      T = X
      SUM = SUM + EPSY * EPSY * (X - TLAST)
      TLAST = X
      EPSI = ABS(Y(2) - PSI0)
      DO 20 I = 1, 8
   20 YC(I) = Y(I)
      IF (EPSI .LE. .02) GO TO 40
      PSI0 = Y(2)
      DO 30 J = 1, NP
        XT(J) = XP(J) * COS(PSI0) + YP(J) * SIN(PSI0)
   30 YT(J) = -XP(J) * SIN(PSI0) + YP(J) * COS(PSI0)
   40 Y0 = -Y(8) * SIN(PSI0) + Y(1) * COS(PSI0)
      X0 = Y(8) * COS(PSI0) + Y(1) * SIN(PSI0)
      YC(1) = Y0
      YC(2) = Y(2) - PSI0
      EPSY2 = 0.
      TSUM = 0.
      SSUM = 0.
      IF (T - TILAST .LT. TICYCL) RETURN
      DO 60 I = 1, NTF
        TJI = (TFF - TSS) / NTF * I + TSS
        DO 50 J = 1, 7
          DO 50 K = 1, 7
            DUMM1(J,K) = TTT1(J,K,I)
   50   DUMM2(J,K) = TTT(J,K,I)
        CALL GMPRD(VECM, DUMM1, DUMV11, 1, 7, 7)
        CALL GMPRD(VECM, DUMM2, DUMV1, 1, 7, 7)
        CALL GMPRD(DUMV1, YC, T1, 1, 7, 1)
        XCAR = X0 + U * TJI
        CALL TRAJ(XCAR, XT, YT, YPATH, YPTHD, YPTHDD, U)
        YPATH = YPATH - VECM(2) * PSI0
        CALL GMPRD(DUMV11, G, S1, 1, 7, 1)
        S1 = S1 * TJI
        EP = T1 + S1 * DFWNOW - YPATH
        TSUM = TSUM + (T1 + DFWNOW*S1 - YPATH) * S1
        SSUM = SSUM + S1 * S1
        EPSY2 = EPSY2 + EP * EP * (TFF - TSS) / NTF
   60 CONTINUE
      EPSY = SQRT(EPSY2) / (TFF - TSS)
```

```
      DFWNOW = -TSUM / SSUM + DFW
      IF (ABS(DFWNOW) .GT. DMAX) DFWNOW = DMAX * SIGN(1.,DFWNOW)
      DO 70 J = 1, 2
        DO 70 I = 1, 99
   70 DMEM(101 - I,J) = DMEM(100 - I,J)
      DMEM(1,1) = DFWNOW
      DMEM(1,2) = T
      TTAB = T - TAUMEM
      DO 80 I = 1, 99
        IF (DMEM(I + 1,2) .LE. TTAB .AND. DMEM(I,2) .GE. TTAB)
     1      GO TO 100
   80 CONTINUE
      WRITE (6,90)
   90 FORMAT ('0', 'TAUMEM PROBABLY TOO LARGE.')
      CALL EXIT
  100 DFW = DMEM(I,1)
      TLAST = X
      TILAST = X
      RETURN
      END
```