UM-HSRI-PF-74-7

AN ANALYSIS OF THE BRAKING AND STEERING PERFORMANCE OF THE AM GENERAL TRANSBUS

FINAL REPORT

SUBCONTRACT NUMBER 9073-054

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MAY 10, 1974

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THE UNIVERSITY OF MICHIGAN

May 15, 1974

Mr. Robert Scheider
AM General Corporation
32500 Van Born Road
Wayne, Michigan 48184

Dear Mr. Scheider:

Enclosed are seven copies of HSRI's final report to AM General Corporation entitled "An Analysis of the Braking and Steering Performance of the AM General Transbus."

One copy of the computer results is contained in the grey binder.

If you have any questions, please feel free to call me.

Sincerely,

James E. Bernard

James E. Bernard
Associate Research Engineer
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1.0 INTRODUCTION

As part of its Transbus development program, the AM General Corporation has sought the assistance of the Highway Safety Research Institute (HSRI) in predicting the steering and braking response of a transit bus characterized by a number of new design features. One of the most innovative aspects of the Transbus, as designed to meet the specifications set forth by the Urban Mass Transportation Administration, are its tires which were specially designed for this project and utilize a construction that departs radically from standard design practice. Since the tire is the major determinant of the steering and braking performance of any motor vehicle, HSRI was asked to measure the mechanical characteristics of the Transbus tires and to utilize this information in performance simulations to judge the controllability and safety of the Transbus design.

In performing this study for AM General, HSRI has produced three preliminary reports. They are entitled:

1. Ride and Handling Analysis of the AM General Transbus,
2. Computer Prediction of the Braking and Steering Performance of the AM General Transbus (an Interim Report), and
3. Computer Prediction of the Braking and Steering Performance of the AM General Transbus (a Summary Report).

The first report contained discussions of (1) the applicability of limit maneuver measures to the motor coach, (2) a simplified procedure for predicting vehicle braking performance, (3) an initial analysis of the steady turning behavior of the projected Transbus, (4) the factors involved in an analysis of ride quality, and (5) the simulation programs available at HSRI which may be used to predict the performance of the Transbus. Preliminary computer predictions of the braking and steering performance of the Transbus in severe maneuvers were presented in the second and third reports. It was pointed out in these latter reports that the results should be expected to be only qualitatively correct, as estimated tire properties were used in the computations. Accordingly, it was agreed that, as soon as tire development problems were solved, the Transbus tires should be tested on the HSRI flat-bed tire test machine to obtain data that would enable a more accurate prediction of the braking and steering performance of the Transbus.

In the next section of this report, the test results obtained for the Goodyear J50 x 19.5 and the Firestone J50C x 19.5 tires are presented in summary form. In the section following, the changes that have taken place in the design of the Transbus since the last simulation effort are identified and quantified. The results summarized in these two sections were used to compute the performance of the Transbus in various steering and/or braking maneuvers and these findings are summarized in Section 4.0. (A complete set of the results obtained in the digital-computer simulations are being delivered to AM General under separate cover.) This report concludes with some comments on the implications of the simulation findings.

2.0 TESTING AND SIMULATION OF THE AM GENERAL TRANSBUS TIRES

2.1 THE TIRE TEST DATA

Lateral forces are created at the tire-road interface when the translational velocity of the tire is not aligned with the plane of the tire, as is indicated in Figure 1. In this figure, the longitudinal axis of the Transbus is indicated by the u_1 vector and the center plane of the tire is indicated by the arrow labelled "wheel heading." The wheel is shown steered with respect to the heading of the bus by the angle, δ. However, it is the slip angle, α, which causes a lateral deformation of the tire-road "contact patch" and thereby gives the motor vehicle its particular behavior.

The HSRI flat-bed tire test machine is shown in Figure 2. It should be noted from the figure that the tire may be set at an angle α with the flat plank which moves under the tire thereby producing the lateral forces to be measured. (The plank may be seen at the upper right of Figure 2.)

The J50 x 19.5 tire (Goodyear) was tested on the flat-bed test machine at several angles* and loads. The measured lateral force and aligning moment caused by lateral slip are tabulated in Tables 1 and 2 and the lateral force caused by inclination (camber) is tabulated in Table 3. In addition, the data presented in Table 1 are shown in Figure 3 in carpet plot form.

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*We had previously planned to take test data up to $\alpha = 30°$. However, the high force levels generated threatened the integrity of the components of the flat-bed, thus α was limited to 16° for the Goodyear tire and 8° for the Firestone tire.

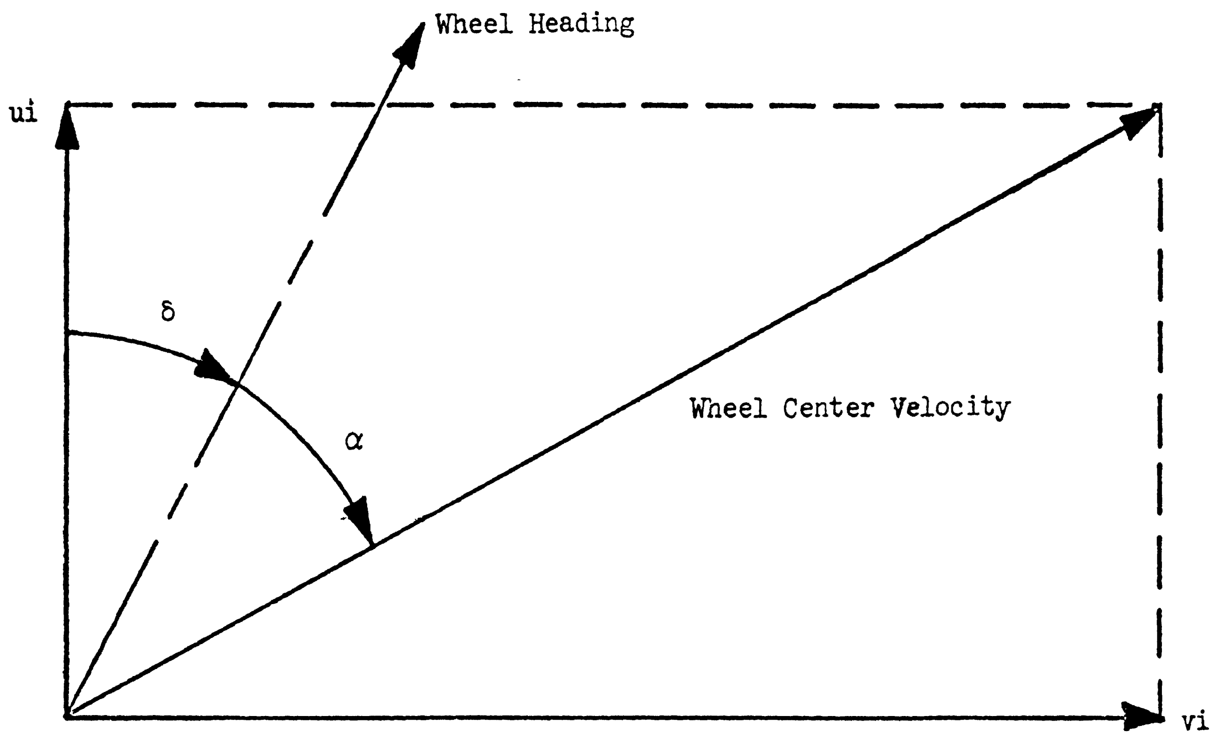

Figure 1. Tire-road interface kinematics

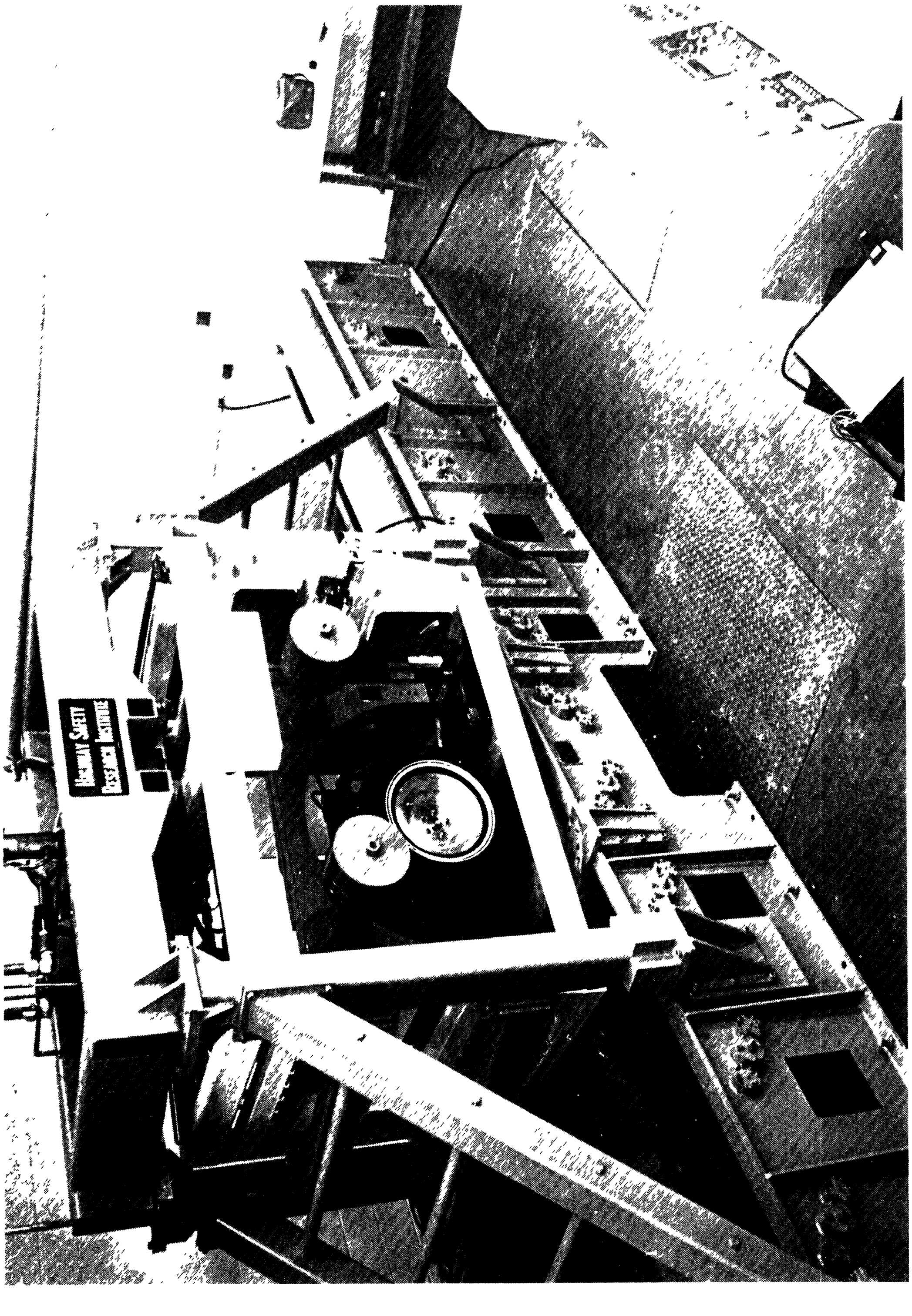

Figure 2. Flat Bed Tire Tester

TABLE 1

LATERAL FORCE VS. STEER ANGLE AND VERTICAL LOAD

TIRE: Goodyear J50-19.5

RIM: 19.5

INFLATION PRESSURE: 85 PSI

VERTICAL LOAD (LB)	LATERAL FORCE (LB.) AT INDICATED STEER ANGLE (DEG)					
	1	2	4	8	12	16
1800	299.3	542.2	919.6	1356.9	1577.9	1646.3
3600	532.5	999.3	1713.8	2588.9	3059.9	3380.5
5430	713.4	1340.1	2374.3	3593.4	4336.0	4885.9
7200	751.1	1577.5	2867.4	4495.7	5352.4	5994.7
9000	844.4	1726.4	3118.6	4628.1	6169.9	6970.3

TABLE 2

ALIGNING TORQUE VS. STEER ANGLE AND VERTICAL LOAD

TIRE: Goodyear J50-19.5

RIM: 19.5

INFLATION PRESSURE: 85 PSI

VERTICAL LOAD (LB)	ALIGNING TORQUE (LB-FT) AT INDICATED STEER ANGLE (DEG)					
	1	2	4	8	12	16
1800	19.8	27.8	34.9	24.7	10.9	9.3
3600	54.4	86.9	111.5	79.7	55.4	47.0
5430	100.2	163.8	176.7	187.6	133.6	106.8
7200	136.1	257.8	373.0	331.1	251.8	192.8
9000	203.9	380.9	558.5	532.7	410.0	249.3

TABLE 3

CAMBER THRUST VS. CAMBER ANGLE AND VERTICAL LOAD

TIRE: Goodyear J50-19.5

RIM: 19.5

INFLATION PRESSURE: 85 PSI

VERTICAL LOAD (LB)	CAMBER THRUST (LB) AT INDICATED CAMBER ANGLE (DEG)				
	1	2	3	4	5
1800	33.2	36.3	44.8	62.1	80.5
3600	66.3	81.3	165.8	219.8	266.5
5430	96.7	202.9	294.9	386.1	478.8
7200	129.4	264.3	401.6	525.7	646.6
9000	169.5	316.6	477.4	618.1	767.1

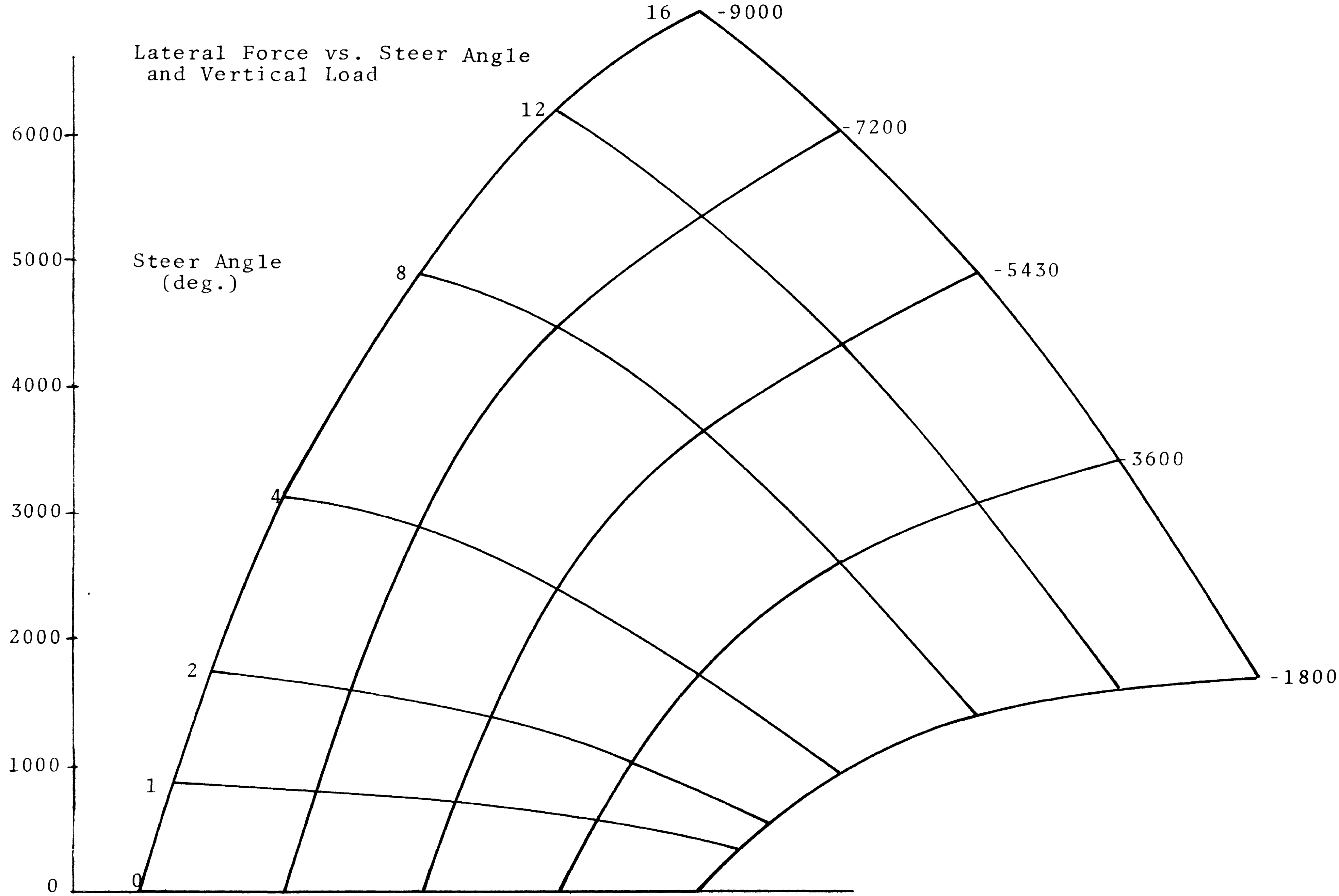

Figure 3. Lateral Force vs. Steer Angle and Vertical Load, Goodyear J.50-19.5

It should be noted that Goodyear had previously estimated the lateral force that would exist at $\alpha = 1°$ for use in the preliminary simulations. Goodyear had predicted that their tire would produce 824 lbs. at one degree when loaded at 5430 lbs. Examination of Table 3 also shows that Goodyear's prediction that the camber thrust at one degree of inclination would be about 10% of the lateral force at $\alpha = 1°$ was a reasonably accurate estimate.

The Firestone J50C x 19.5 tire was similarly tested, producing the data tabulated in Tables 4, 5 and 6. The force levels measured for the Firestone tire are seen to be quite similar to those obtained with the Goodyear tire.

2.2 USE OF THE DATA IN THE SIMULATION

For directional maneuvers conducted in the normal driving range, it is the cornering stiffness, C_α (defined as the rate of change of lateral force with sideslip angle at $\alpha = 0$) which has a first-order effect on the handling characteristics of the vehicle. Since C_α is a function only of the stiffness of the carcass and tread of the tire, this flat-bed result may be used in the simulation with no consideration being given to the nature of the surface on which the measurement was made or to the surface on which a maneuver is assumed to occur. Similar arguments apply to the inclination (camber) stiffness and aligning stiffness of the tire. However, the forces and moments produced at the tire-road interface in the presence of large slip angles are surface dependent. Since no "over-the-road" tire measurements were made or were available, HSRI employed the procedure outlined below for purposes of predicting realistic shear forces as would occur on real road surfaces. The ratios of the braking force (both peak and slide) to normal load as would occur on a dry surface with an ASTM skid number of 85 were estimated by Goodyear. These estimates are presented in Table 7. Goodyear also provided data

TABLE 4

LATERAL FORCE VS. STEER ANGLE AND VERTICAL LOAD

TIRE: Firestone J50C-19.5

RIM: 19.5

INFLATION PRESSURE: 85 PSI

VERTICAL LOAD (LB)	LATERAL FORCE (LB) AT INDICATED STEER ANGLE (DEG)					
	1	2	3	4	6	8
1800	322.7	553.7	750.9	906.1	1139.8	1281.1
3600	560.3	984.0	1362.6	1676.4	2090.7	2323.7
5480	737.6	1344.5	1867.9	2348.5	2966.9	3254.6
7200	849.9	1593.8	2193.5	2875.5	3632.2	3988.5
9000	915.2	1731.2	2372.2	3228.4	4167.5	--

TABLE 5

ALIGNING TORQUE VS. STEER ANGLE AND VERTICAL LOAD

TIRE: Firestone J50C x 19.5

RIM: 19.5

INFLATION PRESSURE: 85 PSI

VERTICAL LOAD (LB)	ALIGNING TORQUE (LB-FT) AT INDICATED STEER ANGLE (DEG)					
	1	2	3	4	6	8
1800	15.4	22.9	28.0	27.5	26.4	29.9
3600	52.2	77.3	95.8	88.3	90.2	83.8
5480	98.9	154.8	192.3	182.3	190.2	170.8
7200	144.2	240.7	298.9	325.5	327.0	290.1
9000	192.1	329.6	412.1	424.4	477.7	--

TABLE 6

CAMBER THRUST VS. STEER ANGLE AND VERTICAL LOAD

TIRE: Firestone J50C x 19.5

RIM: 19.5

INFLATION PRESSURE: 85 PSI

VERTICAL LOAD (LB)	CAMBER THRUST (LB) AT INDICATED STEER ANGLE				
	1	2	3	4	5
1800	126.9	229.8	313.0	377.6	440.9
3600	169.9	326.6	465.9	603.1	720.6
5480	186.1	358.7	518.7	689.1	857.6
7200	197.6	384.9	562.5	743.8	918.8
9000	213.7	405.1	594.8	786.7	977.6

TABLE 7

ESTIMATED PEAK AND SLIDING COEFFICIENT OF FRICTION, μ, FOR THE GOODYEAR J50 x 19.5 TRANSBUS TIRE

Speed = 60 mph

Load	μ_{peak}	μ_{slide}
1800	1.04	.661
3300	.892	.567
4770	.756	.480
5430	.71	.450
6900	.547	.347

that define the degree to which the "slide" coefficient varies with speed at a fixed vertical load for their tire on this same surface (see Table 8).

TABLE 8

ESTIMATED VARIATION OF SLIDE COEFFICIENT WITH SPEED FOR THE GOODYEAR J50 x 19.5 TRANSBUS TIRE

Load = 5430 lbs

Speed (mph)	μ_{slide}
10	.9
20	.775
30	.653
40	.548
50	.481
60	.450
70	.415

With the above information, namely, the cornering stiffness as measured by HSRI (as a function of vertical load) and the specific traction properties (Tables 7 and 8) estimated by Goodyear, HSRI was able to utilize a model of the tire traction process [1] to generate lateral forces for any combination of lateral and longitudinal slip and for any combination of speed and vertical load. The tire model also generates the longitudinal force that will be produced by the given tire at any combination of longitudinal and lateral slip and any combination of speed and vertical load. For example, the tire algorithm used in HSRI's tire-vehicle system simulation generates the $|F_x|/F_z$ versus longitudinal slip (i.e., μ-slip curves) shown in Figure 4 when the Goodyear tire is operating at 20, 40, and 60 mph with a vertical load of 5430 lbs.

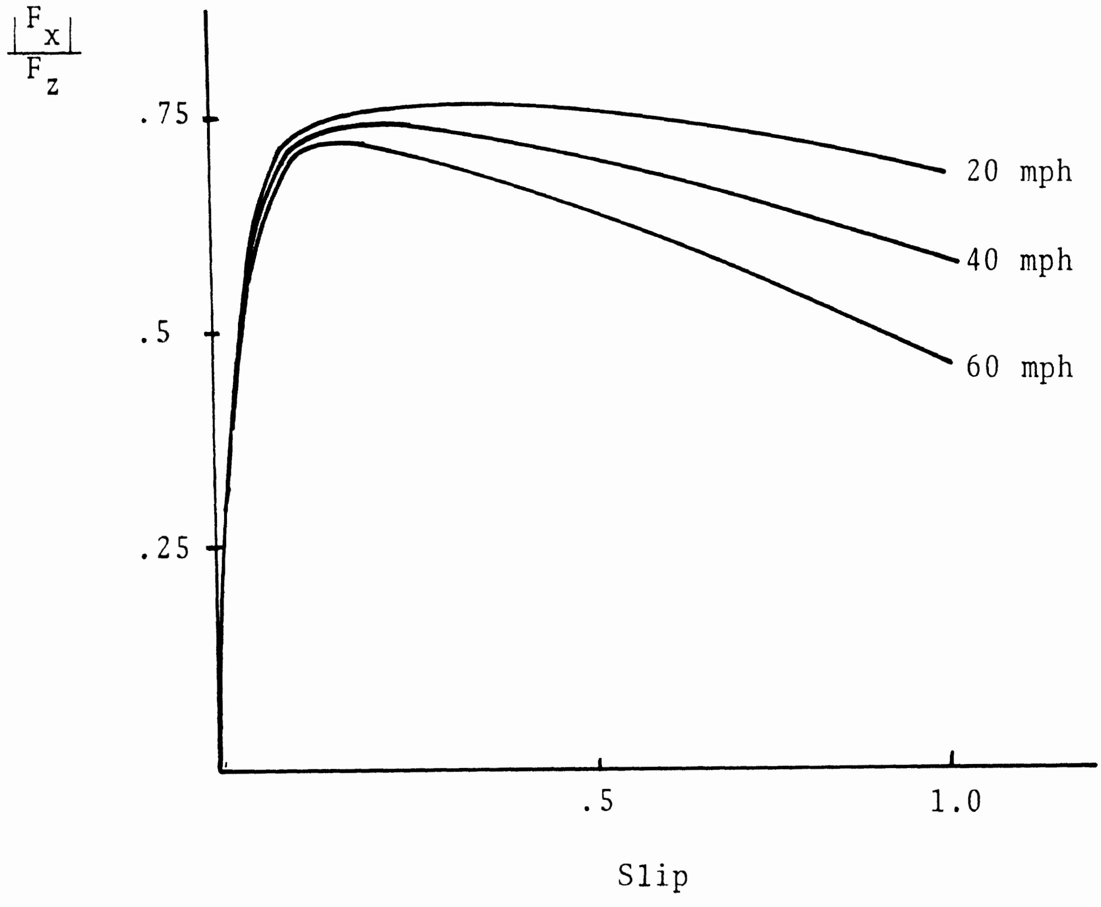

Figure 4. μ-Slip Curve, Load = 5430 lbs.

3.0 UPDATED PARAMETER DATA FOR THE TRANSBUS

In addition to updating the input data required to describe the shear force characteristics of the tires developed for the Transbus, HSRI modified previously supplied input data to reflect the latest design changes made by AM General. These design changes involved revised layouts of the steering linkage and of the rear suspension.

Examination of the layout of the front suspension geometry and the geometry of the steering linkage showed that roll understeer was present. For purposes of these calculations, a Taylor series approximation of the prevailing kinematics was utilized to derive the steer angle per unit roll angle of the chassis that applies for small displacements about the trim condition. This calculation indicated that the current design of the Transbus possesses a front roll understeer of 0.09 deg/deg. Since the maneuvers to be simulated, in nearly all instances, involved large slip angles, the Ackerman steer designed into the steering layout was ignored and all computations were performed with the right- and left-wheel steering displacements assumed equal.

Examination of the rear suspension drawings provided by AM General showed that roll understeer was present. No attempt was made to account for the actual steer of the axle that would occur over the full range of roll displacement, but, rather (as was done for the front suspension) the linearized roll steer was evaluated and found to be 0.05 deg/deg of roll understeer.

Brake dynamometer information supplied to AM General by Bendix Corporation was used in the simulation. The line pressure-torque data was taken directly from the final effectiveness test given on page 24 of Reference 2. These data are tabulated in Table 9. It should be emphasized that the results obtained in the braking simulations have a first-order dependence on this data.

TABLE 9

LINE PRESSURE VS. BRAKE TORQUE FOR THE AM GENERAL TRANSBUS BRAKE

Pressure (psi)	Torque (in lbs)
0	0
200	6666
400	15000
800	28200
1000	35000
1200	43300
1400	50000
1800	60000

4.0 PREDICTIONS OF TRANSBUS PERFORMANCE

Four maneuvers have been simulated to examine the steering and braking performance of the Transbus. These maneuvers are described below under the titles "straight-line braking," "trapezoidal steer" (J-turn), "sinusoidal steer" (lane change), and "braking-in-a-turn." The levels of steering and braking inputs used in these simulations were selected to represent control actions which might be taken by a driver in a severe, emergency maneuver.

Although the simulated maneuvers are derived from a limit-maneuver methodology developed in previous research projects [3, 4, 5] concerned with passenger vehicles, it has not been our intention to find the limit performance of the Transbus. Rather, maneuvers have been defined and simulated to examine whether or not the emergency response of the vehicle will be acceptable. As pointed out in the first report from HSRI to AM General, limit performance methods, as applied to passenger cars, may not be entirely satisfactory for the study of motor coach performance. Accordingly, we have selected maneuvers that should provide meaningful information on emergency maneuvering performance without imposing overly demanding requirements for the initial testing.

4.1 STRAIGHT-LINE BRAKING STOPS

Simulation runs were made at various brake line pressure levels. All lags and delays in the brake actuation system were assumed to be negligible. However, the driver is assumed to displace the foot valve in a manner such that 0.5 seconds is required to develop the desired pressure level (see Figure 5). Straight-line stops from 60 mph were simulated for three payload conditions. The computed results, which are summarized in Table 10, indicate that more than adequate brake torque is available to

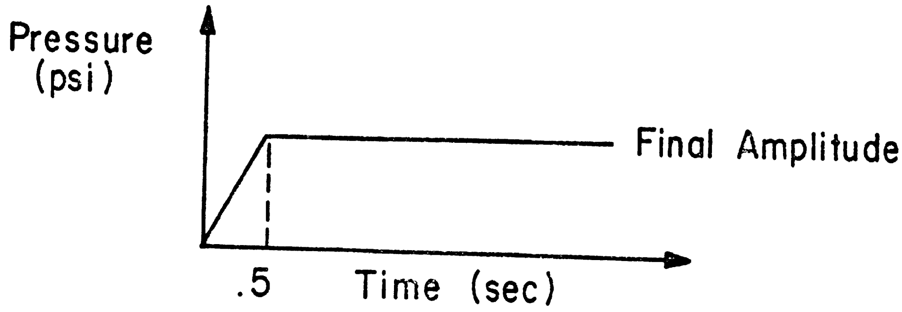

Figure 5. Pressure Vs. Time at the Foot Valve for the Straight Stop Simulations.

TABLE 10

Dry (60 mph)

Vehicle Weight Condition	Brake Line Pressure (psi)	Stopping Distance (ft)
Underloaded (25,550 lbs)	1400 (no lock)	184.0
	1450 (axle 3 locked)	183.1
Normal (32,000 lbs)	1650 (no lock)	200.2
	1700 (axle 3 locked)	202.0
Overloaded (35,300 lbs)	1650 (no lock)	217.6
	1700 (axle 3 locked)	218.7

produce deceleration levels in compliance with FMVSS 121. Further, these results indicate that the shear force capability at the tire-road interface will easily allow enough brake force to meet the standard. It should be emphasized that these computations have first-order dependence on two factors, namely (1) the brake dynamometer data produced by Bendix, and (2) Goodyear's estimate of the maximum shear force that is created on a dry road surface. Nevertheless, we feel quite confident that these estimations are reasonable enough to ensure that the Transbus will be able to meet stopping distance requirements on a dry surface as specified by FMVSS 121.

It is our understanding that the Transbus will be outfitted with an antiskid system on each axle. Since HSRI had no detailed information on this antiskid system, and since no test data were available to indicate the performance of the Transbus tires on a wet surface, we were unable to simulate straight-line braking performance on a specified wet surface with the anti-lock system operating. However, assuming a reasonable performance of the installed antiskid system, we expect the Transbus to meet the stopping distance requirements imposed by FMVSS 121 for decelerations performed on a wet (i.e., reduced friction coefficient) surface.

4.2 TRAPEZOIDAL STEER SIMULATIONS - 60 MPH

The purpose of these computer runs is to assess the performance of the Transbus in rapid turns. Five simulation runs were performed at an initial speed of 60 mph. In each, the front wheel steer angle was prescribed to have the time history shown in Figure 6.

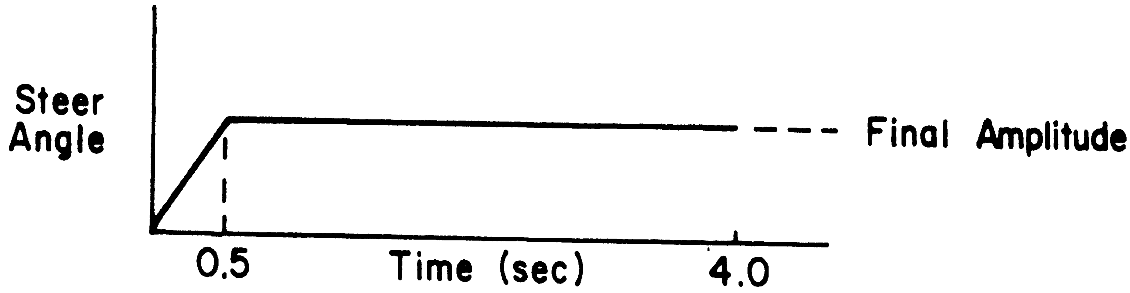

Figure 6. Steer Angle Vs. Time for the Trapezoidal Steer Simulations.

The trapezoidal steer results (see Table 11) show that the vehicle can make a "drastic" turn without exceeding the jounce/rebound travel limits in the front and rear suspension. (Bump stop contact occurs at approximately 5° roll angle.) An examination of the resulting time histories indicate that the vehicle will be able to perform sharp turns without exhibiting a propensity to spinout or roll over.

TABLE 11

TRANSBUS TURNING RESPONSE
(4.0 Seconds After the Initiation of Steering)
($V_{initial}$ = 88 fps)

Amplitude of Front Wheel Angle	Lateral Acceleration (ft/sec^2)	Turn Radius (ft)	Yaw Rate (deg/sec)	Longitudinal Velocity (ft/sec)	Max. Roll Angle (deg)	Max. Sideslip Angle (deg)
2	10.2	732	6.7	86.4	2.7	- 2.3
4	16.4	418	10.8	82.3	4.3	- 5.6
8	18.5	318	11.6	74.2	4.9	-12.0
16	18.0	297	8.6	71.4	4.8	-12.0
24	14.8	360	9.9	72.5	4.65	- 8.7

4.3 SINUSOIDAL STEER

A steering input having the wave form shown in Figure 7 was used to produce an approximate lane-change maneuver starting with an initial speed of 30, 40, and 50 mph. The peak value of steer angle (δ_p) was selected such that the lateral displacement produced during the lane change would be about the same for each speed (viz, between 10 and 15 feet).

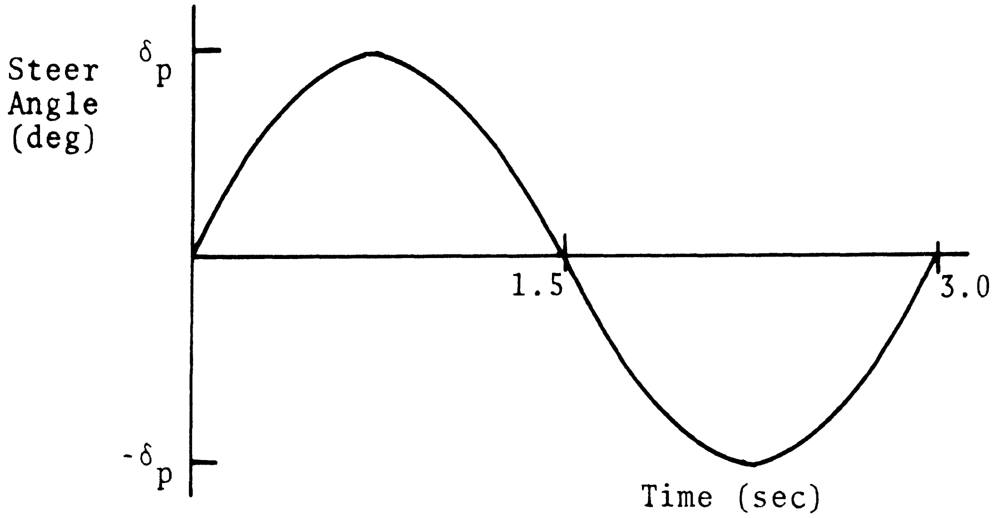

Figure 7. Steering Waveform for the Lane-Change Maneuver.

Figure 8 presents the vehicle path (i.e., the trajectory of the center of gravity) produced by the simulation. Note that at the time of the initiation of the simulated lane-change maneuver, the bus is heading in the "x" direction at y = 0. These results indicate that the Transbus should be able to perform rapid lane changes, wherein a symmetric steering input causes the vehicle to attain a final heading that is essentially equivalent to the initial heading of the vehicle.

4.4 BRAKING-IN-A-TURN

For the braking-in-a-turn maneuver, the front-wheel steer angle was applied as before, i.e., an 0.5-second ramp input starting at the beginning of the run. After three seconds elapse, the brakes are applied, assuming an 0.05-second ramp input. These inputs are shown schematically in Figure 9. Calculations were made for maneuvers in which the final or fixed values of steer angle and brake line pressure were as follows (initial velocity = 30 mph):

Final Steer Angle (deg)	Final Line Pressure (psi)
10	1400
20	1300
20	1800

In the first two of the above runs, brake line pressure was selected to cause the most severe braking possible without locking any wheel. (At higher line pressures, the antiskid system would be expected to come into play.) The trajectories produced in these simulations are presented in Figure 10. In each case, the deviation from the path of the steady-state turn without braking should be noted.

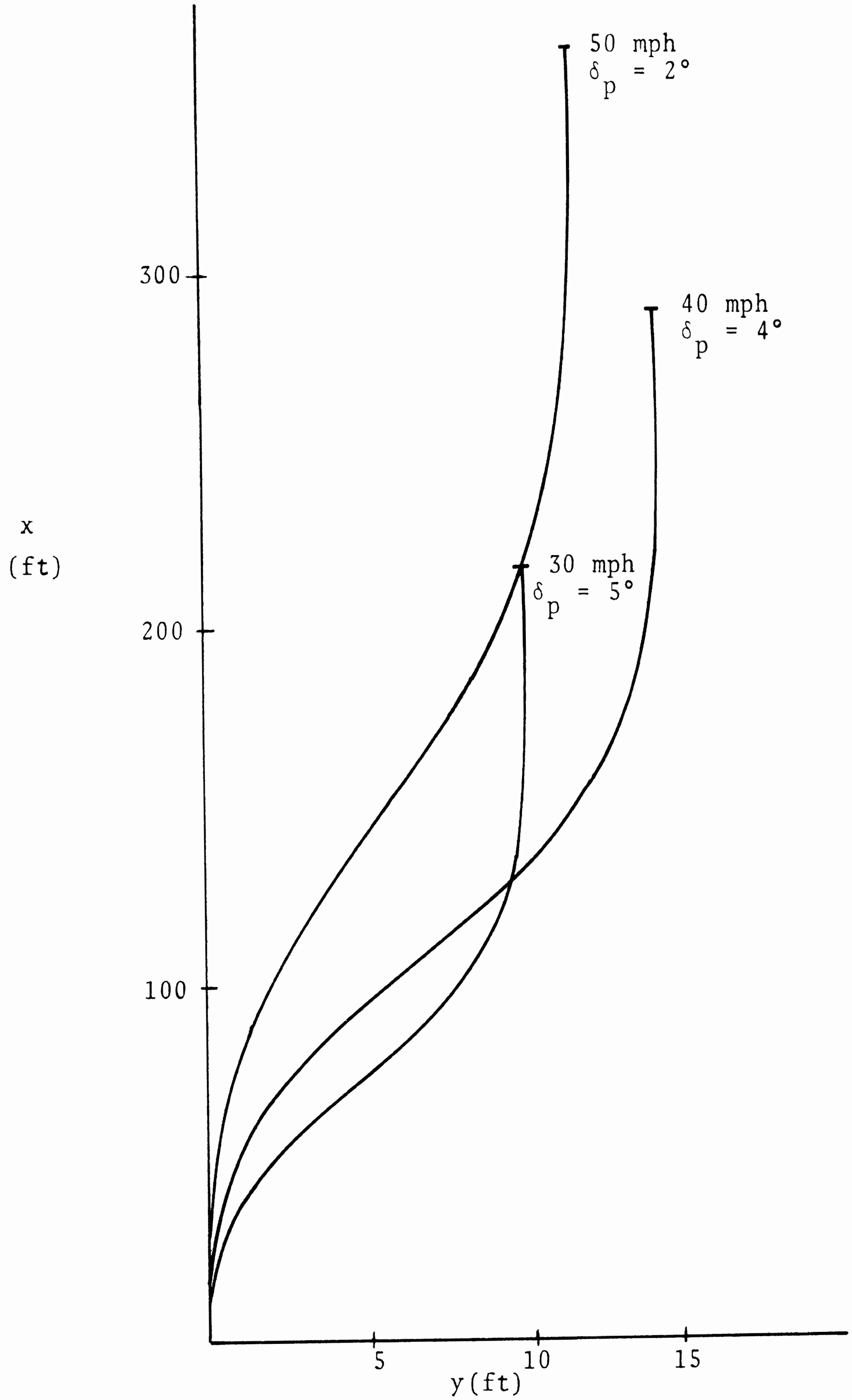

Figure 8. Trajectories During Lane Change Maneuver

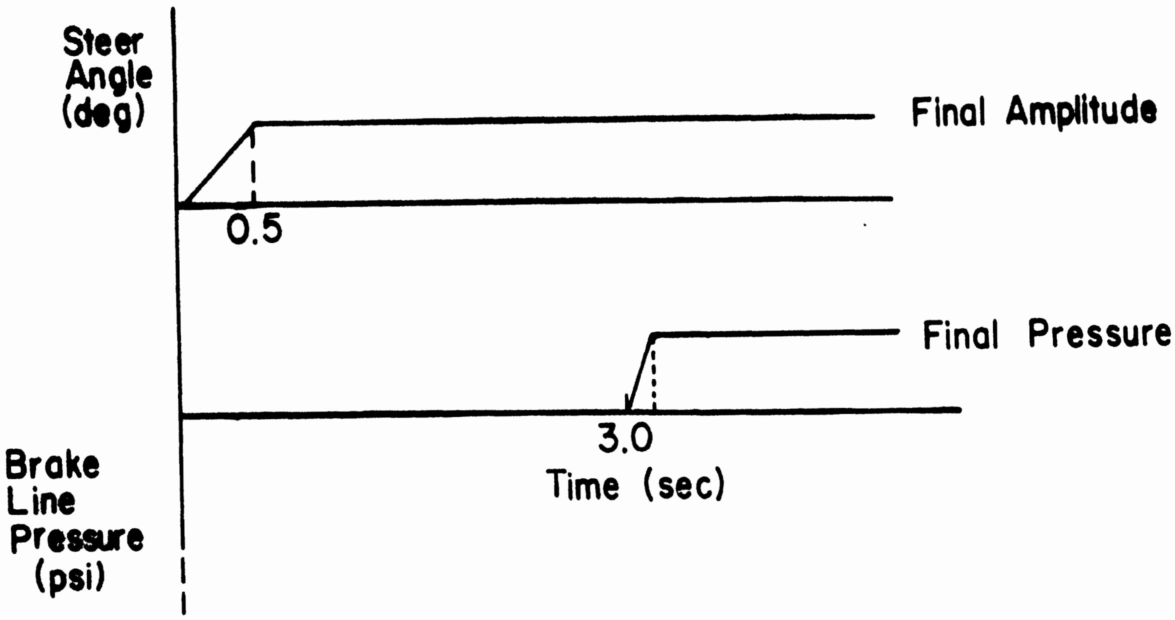

Figure 9. Steer Angle and Brake Line Pressure for the Braking-In-A-Turn Maneuvers.

On the basis of the results plotted in Figure 10, it would appear that the Transbus can be braked rather severely in a turn without causing a severe departure from the intended curved path. This statement must, however, be qualified by noting that the simulation assumes that the brakes are perfectly balanced, right and left, and that steer displacements caused by the changes in suspension geometry deriving from the existence of braking forces have been neglected in the simulation.

In the third braking-in-a-turn simulation, a full brake pressure (1800 psi) application is assumed to occur during the course of a severe turn. The trajectory of the latter part of this run is compared in Figure 11 to the trajectory computed for the 1300 psi run. The effects of rear-wheel lockup, when full pressure is applied, may be seen in the rapid divergence of the trajectory from the curved path. The antiskid system to be employed in the Transbus will clearly prevent this type of behavior.

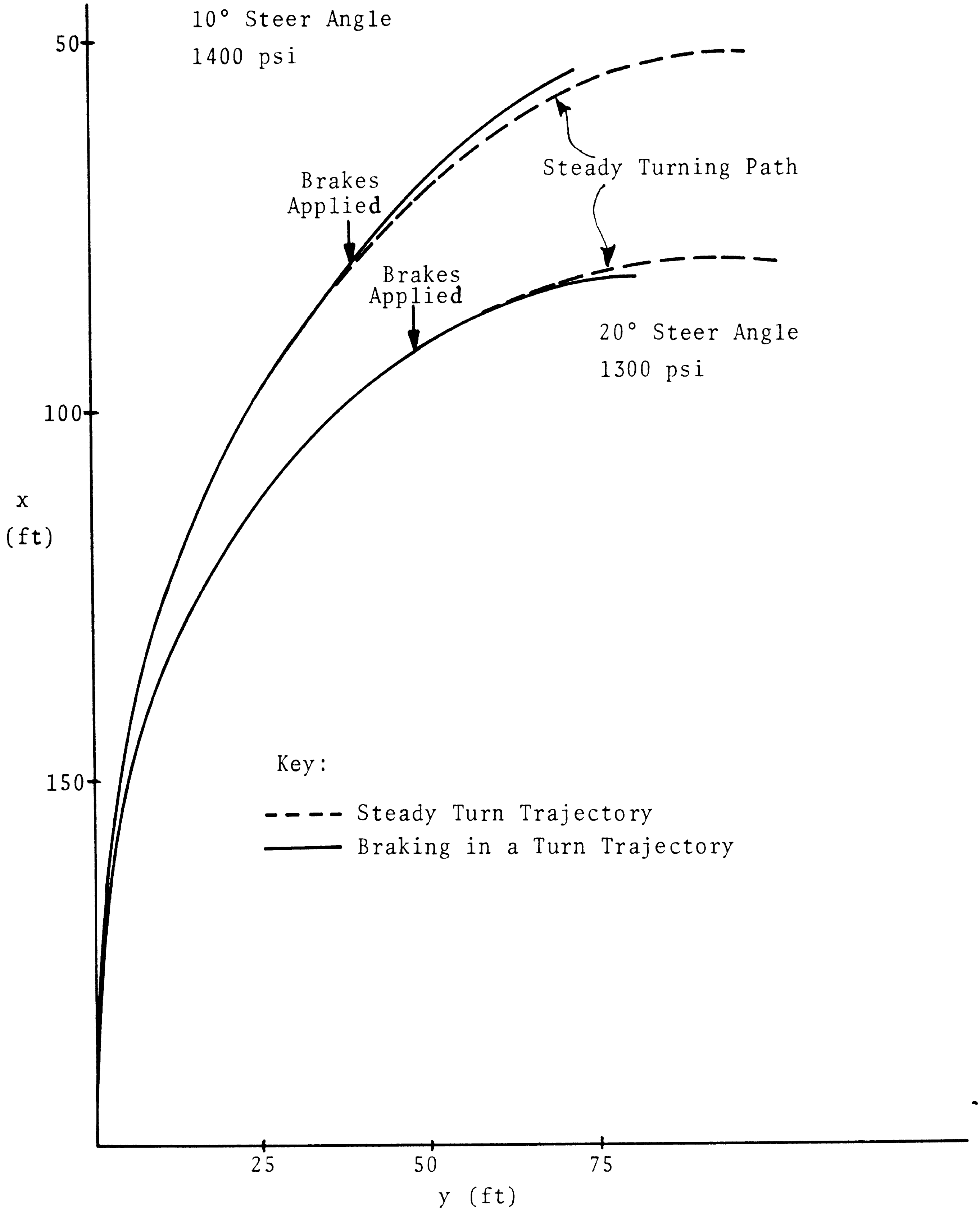

Figure 10. Trajectories During Braking-In-A-Turn Maneuver

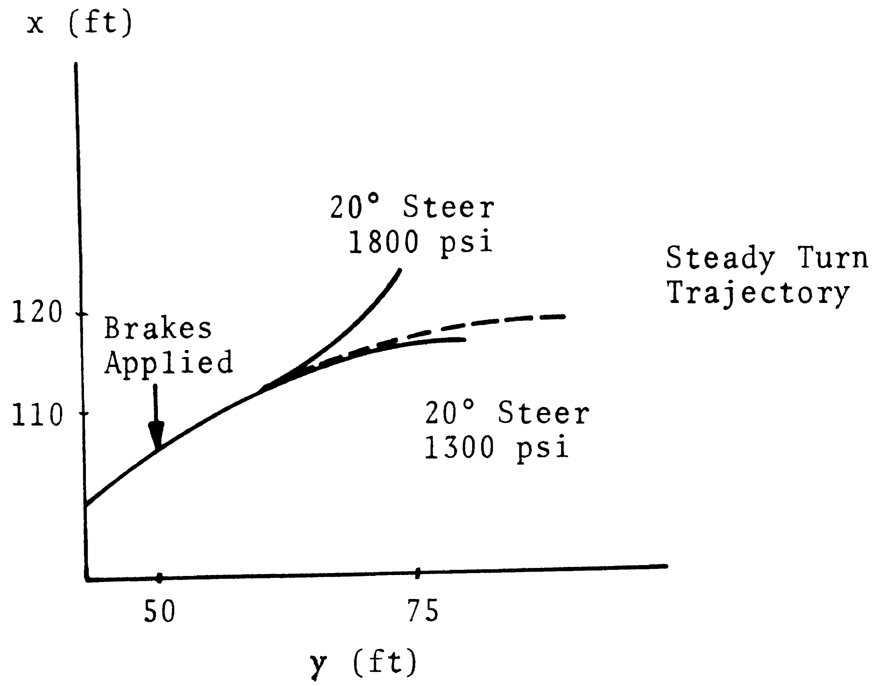

Figure 11. Trajectory During Braking-In-A-Turn Maneuver

5.0 CONCLUDING REMARKS

Performance predictions deriving from motor vehicle simulations must be viewed in a proper perspective. The validity of the simulation depends, of course, on (1) the completeness and correctness of the mathematical description of the system and (2) the accuracy of the parameter data that have been estimated or measured for the system. In this particular instance, HSRI believes that the equations used to describe the Transbus and their corresponding mechanization on a digital computer are valid in view of the considerable amount of checking and validation testing that has been performed under the auspices of the Motor Vehicle Manufacturer's Association. Accordingly, the findings of this study depend, in the last analysis, on the accuracy with which AM General and HSRI has estimated and/or measured the inertial, mechanical, and kinematic properties of the Transbus coach and its components.

Reasonable care has been taken in this latter regard and, in particular, the main effort (both in time and money) has been devoted to measuring the properties of the new and unique tires developed for the Transbus. Clearly, these tires provide a cornering stiffness that gives the Transbus directional response characteristics that are equivalent to those possessed by smaller motor vehicles. The simulation findings suggest that the Transbus is well behaved in severe steering and braking maneuvers and should be judged to be adequate or more than adequate in its overall pre-crash safety quality.

6.0 REFERENCES

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3. Dugoff, H., Segel, L., and Ervin, R.D., "Measurement of Vehicle Response in Severe Braking and Steering Maneuvers," SAE Paper #710080, January 11-15, 1971.

4. Ervin, R.D., Grote, P., Fancher, P.S., MacAdam, C.C., and Segel, L., Vehicle Handling Performance, Final Report, Contract DOT-HS-031-1-159, Highway Safety Research Institute, Univ. of Michigan, November 1972.

5. Ervin, R.D., Fancher, P.S., and Segel, L., "Refinement and Application of Open-Loop Limit-Maneuver Response Methods," SAE Paper #730491, May 14-18, 1973.