UM-HSRI-79-85-2

SIMULATION OF THE EFFECTS OF INCREASED TRUCK SIZE AND WEIGHT

Final Technical Report

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Contract Number DOT-FH-119330
Contract Amount: $135,560

November 1979

Highway Safety Research Institute
The University of Michigan

Prepared for:

Federal Highway Administration
U.S. Department of Transportation

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Technical Report Documentation Page

1. Report No. UM-HSRI-79-85-2	2. Government Accession No.	3. Recipient's Catalog No.
4. Title and Subtitle SIMULATION OF THE EFFECTS OF INCREASED TRUCK SIZE AND WEIGHT Final Technical Report		5. Report Date November 1979
		6. Performing Organization Code
7. Author(s) T. Gillespie, C. MacAdam, G. Hu, J. Bernard, C. Winkler		8. Performing Organization Report No. UM-HSRI-79-85-2
9. Performing Organization Name and Address Highway Safety Research Institute The University of Michigan Huron Parkway & Baxter Road Ann Arbor, Michigan 48109		10. Work Unit No. (TRAIS)
		11. Contract or Grant No. DOT-FH-11-9330
12. Sponsoring Agency Name and Address Federal Highway Administration U.S. Department of Transportation Washington, D.C. 20590		13. Type of Report and Period Covered Final 9/77 - 11/79
		14. Sponsoring Agency Code
15. Supplementary Notes		

16. Abstract

A computer program for simulating the braking and directional response of heavy vehicles has been developed for the Federal Highway Administration as a tool for investigation of the effects of increased truck size and weight. Designated as the "Truck and Tractor-Trailer Dynamic Response Simulation - T3DRS:V1," the program is capable of simulating trucks, tractor-semitrailers, doubles and triples combinations. Modeling for the vehicle components has been adapted from earlier simulations produced under sponsorship of the Motor Vehicle Manufacturers Assocation. The T3DRS:V1 version consolidated all vehicle combinations into one program with improved input/output format, a new closed-loop path-following steering option, optional side-to-side differences on all paired components, a simplified tandem axle suspension model, and more versatility in the choice of output information. The program has been validated against analytical models, predecessor simulation programs and vehicle test data acquired separately by the Texas Transportation Institute and the Highway Safety Research Institute. The simulation program has achieved operational status on FHWA computer facilities and a training seminar was held to introduce users to the program.

Volume I is a summary report.

17. Key Words vehicle simulation, computer models, braking, directional response, trucks, tractor-trailers		18. Distribution Statement UNLIMITED	
19. Security Classif. (of this report) NONE	20. Security Classif. (of this page) NONE	21. No. of Pages 128	22. Price

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1.0 INTRODUCTION

This document reports on the research project entitled "Simulation of the Effects of Increased Truck Size and Weight," conducted by the Highway Safety Research Institute (HSRI) of The University of Michigan. The research was sponsored by the Federal Highway Administration (FHWA) under Contract Number DOT-HS-11-9330, extending over the period from October 1977-November 1979.

The Federal-Aid Highway Act of 1956 authorized the construction of the National System of Interstate and Defense Highways and placed certain limitations on the dimensions and weights of vehicles operating on the system. The limitations imposed were: 18,000 pounds (8165 Kg) on a single axle, 32,000 pounds (14,515 Kg) on a tandem axle, an overall gross weight of 73,280 pounds (33,240 Kg), and a width of 96 inches (244 cm). These limits were based on considerations of system capacity, strength of existing pavement and bridges, the need for maintenance and resurfacing, the highway geometrics required to accommodate larger vehicles, and the effects of large trucks on traffic operations.

The weight limitations have since been raised to 20,000 pounds (9,072 Kg) for a single axle, 34,000 pounds (15,422 Kg) for a tandem axle and 80,000 pounds (36,288 Kg) total gross weight for the vehicle combination. Increasing loads have broad implications on the operation of the highway system. For example, the 11% increase in single-axle load has significance because it is the basis for the structural design of pavements and is also a primary factor in vehicle control.

Proposals are being made to further increase the allowable loads, as well as raise the width limitations, and it must be anticipated that more such proposals will be submitted from time to time. Such proposals should be evaluated on a rational basis, and should consider the economic and social impacts as well.

With the emergence of high-speed computers, the simulation of complex processes has become a reliable and cost-effective method for investigating the performance of new concepts or the effects of modifications to existing systems. Computer codes enabling the simulation of the ride and handling of heavy trucks have been developed and are operational. It follows that such programs, together with some experimental and field data, offer considerable promise for aiding the Federal Highway Administration in conducting analyses that are needed for rational decision making.

To this end, HSRI has been supported by the FHWA in a program designed to modify and exercise computer simulation programs for investigating the dynamics of heavy vehicle trains, their response to control inputs, and their stability in the presence of disturbance inputs. The approach adopted in the study was that of selecting an existing simulation program for trucks and tractor-trailers and modifying it as necessary to meet the above stated objectives. To establish the veracity of this work, a separate project entitled "Validation of Truck Handling Simulation Results" was sponsored concurrently at the Texas Transportation Institute/Texas A & M University, to generate full-scale vehicle test data against which the computer simulation program could be validated. The research plan called for using the validated program in a prototype study of truck size and weight effects, and a copy of the program was to be vested with the Federal Highway Administration for use at their own computer facilities.

This report summarizes the development and the features of the computer code prepared to satisfy the needs and requirements of the FHWA. This report includes:

1) a statement of the background underlying the development of the program,

2) a description of the program,

3) a definition of the uses of the program,

4) a report on the validation of the program,

5) a summary of the available documentation on the program, and

6) a summary of the training seminar (Section 3.0).

The report discusses the application of the simulation program to the study of truck size and weight issues (Section 4.0), and concludes with a presentation of findings and recommendations with respect to follow-on use of the program by the FHWA.

2.0 DEVELOPMENT OF THE T3DRS:V1 PROGRAM

2.1 Background

Since 1971, the HSRI has conducted research under the sponsorship of the Motor Vehicle Manufacturers Association (MVMA) to develop computer-based methods for analyzing and predicting the directional and braking response of commercial motor vehicles. The initial phase of this research dealt with modeling the braking performance of commercial vehicles and was reported in Reference [1]* (Phase I). The second phase extended vehicle modeling to allow for directional response and was reported in Reference [2] (Phase II). The continuation of research into braking performance led to additional refinements in the braking simulation which were reported in Reference [3] (Phase III). In total, this research under the auspices of MVMA led to four separate computer simulation models:

- Straight Truck Braking Model (Phase I & III)
- Tractor-Trailer Braking Model (Phase I & III)
- Straight Truck Directional Response Model (Phase II)
- Tractor-Trailer Directional Response Model (Phase II)

Though all programs evolved from the same approach to vehicle modeling, separate programs were prepared and maintained.

Under this project, the requirements to add one or two full trailers (doubles and triples) to the tractor-semitrailer model were cause for reformulating the computer simulation model for the purposes of:

- Consolidating all vehicle combinations into one program
- Improving the input/output format
- Simplifying the model to include only the most relevant aspects as determined from the intervening research

The work led to a new simulation program using the same modeling approaches. The program, described here, is designated as "The Truck and Tractor-Trailer Dynamic Response Simulation - T3DRS:V1."

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*Numbers in brackets indicate references in Section 7 of this report.

2.2 Description of the Program

The T3DRS:V1 program is a time domain mathematical simulation of a truck/tractor, a semitrailer and up to two full trailers. The vehicles are represented by differential equations derived from Newtonian mechanics that are solved at successive time increments by digital integration. A more detailed description of the program is provided in the User's Manual [4] and Programmer's Manual [5] prepared under this project.

The program is written in a generalized fashion to allow simulation of a large number of vehicle configurations, as shown in Figure 1. The first vehicle is the power unit and may be a truck or tractor, both of which may carry payload. As a single unit with no payload, it is equivalent to an empty truck or bobtail tractor. With payload, it is a truck, which, with a semitrailer as well, simulates a car hauler, dromedary tractor, etc. The second unit is always a semitrailer (i.e., current models do not include a truck with full trailer). The third and fourth units are full trailers consisting of semitrailers on either a fixed or converter dolly. Separate payload may be specified for each trailer.

The truck/tractor unit is distinguished by the fact that it can have only a single front axle with single tires, and can be arbitrarily steered. All other axles on the vehicle combination can be represented as single or tandem axles with single or dual wheel sets.

The mathematical model effectively incorporates up to 71 degrees of freedom. The number of degrees of freedom are dependent on the vehicle configuration and derive from the following:

- Six degrees of freedom (three translational and three rotational) for the truck/tractor sprung mass
- Three degrees of freedom for the semitrailer (the three other degrees of freedom of the semitrailer are effectively eliminated by dynamic constraints at the hitch)
- Five degrees of freedom for each of the two full trailers allowed
- Two degrees of freedom (vertical and roll) for each of the 13 axles allowed

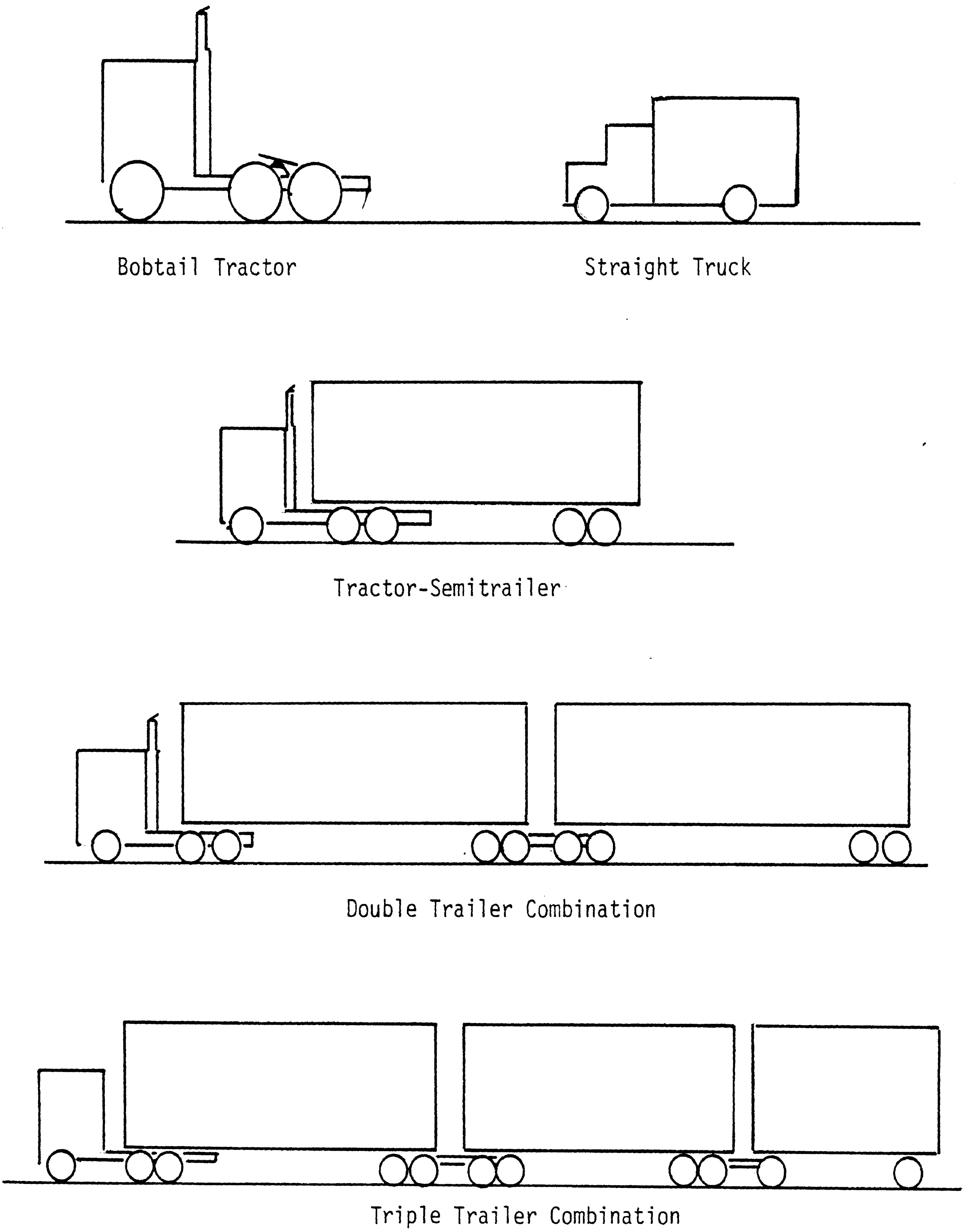

Figure 1. Vehicle configurations that can be simulated with T3DRS:V1.

•A wheel rotation degree of freedom for each of the 26 wheels allowed

The motion of each of the sprung masses is determined from the summation of forces and moments upon it arising from the tires (acting through the unsprung mass of the axle and suspension), gravity and the hitch point constraints. Small angle assumptions are made in the derivation of the mathematical equations so that the simulation can be validly applied up through the onset of rollover.

Operation of the T3DRS:V1 program is accomplished by submission of the necessary job control instructions followed by a list of input parameters. The specific job control instructions required are dependent on the user's computer system and whether batch or remote job entry is being used.

The list of parameters describes the vehicle being simulated and the maneuver being performed. The first group in the list is called the Simulation Operation Parameters and includes the following information:

- Title for the run
- Vehicle configuration
- Initial velocity of the vehicle
- Steer input (steering angles or path to be followed)
- Braking inputs at the treadle valve
- Simulation time
- Road description (flat, grades or user subroutine)
- Output (type and printing intervals)

The list next includes information to describe the truck or tractor, grouped in the order of sprung mass properties, front suspension and axle description, front tire and wheel properties, then rear suspension and axle, rear tires and wheels; and finally, the individual brake characteristics (if braking is used in the maneuver). If the vehicle configuration includes a semitrailer, the list of parameters continues with a description of its sprung mass properties, suspension and axle properties, tire and wheel properties, and brake characteristics. With doubles and triples combinations, the description of each trailer then follows in a similar fashion. The full trailers of a doubles or triples combination may be of the fixed or converter dolly type.

The sprung mass properties are described by the following types of parameters, as illustrated for truck/tractors in Figure 2:

-Wheelbase (the characteristic length)
-Front and rear curb weights (weight)
-Center of gravity height
-Moments of inertia in roll, pitch and yaw
-Payload (weight, location and moments of inertia)
-Hitch point location (fifth wheel or pintle hook)
-Fifth wheel roll stiffness (with tractors only)

The payload is an option that facilitates easy simulation of a vehicle under different loading conditions. Hitch point (and fifth wheel stiffness) information is required only when the unit being described is a tow vehicle for another trailer. The sprung mass information for a full trailer includes four additional parameters at the beginning of the list, which consists of a key for selecting fixed or converter dolly, and three dimensions that effectively describe the tongue length, location of the yaw articulation point, and location of the pitch articulation point.

The suspension and axle parameters describe the suspension and unsprung mass properties. These items are modeled as shown in Figure 3. Either a single or optional tandem axle may be specified at any axle location except the front axle of the truck/tractor unit. The descriptive parameters required are as follows:

-Suspension key (single or tandem)

-Tandem parameters (axle separation, static load distribution, brake torque load transfer effects)

-Spring rates

-Viscous damping and coulomb friction

-Axle mass and roll moment of inertia

-Roll center height

-Roll steer coefficient

-Auxiliary roll stiffness

-Lateral distance between springs

-Track width

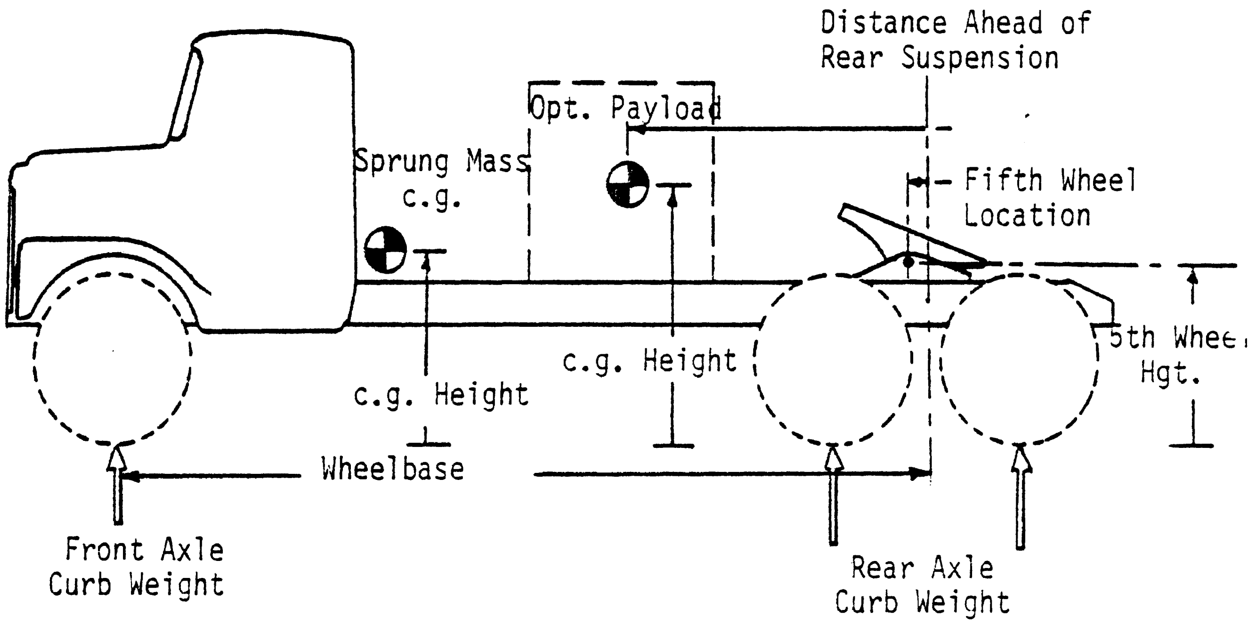

Figure 2. Modeling elements of the truck/tractor sprung mass.

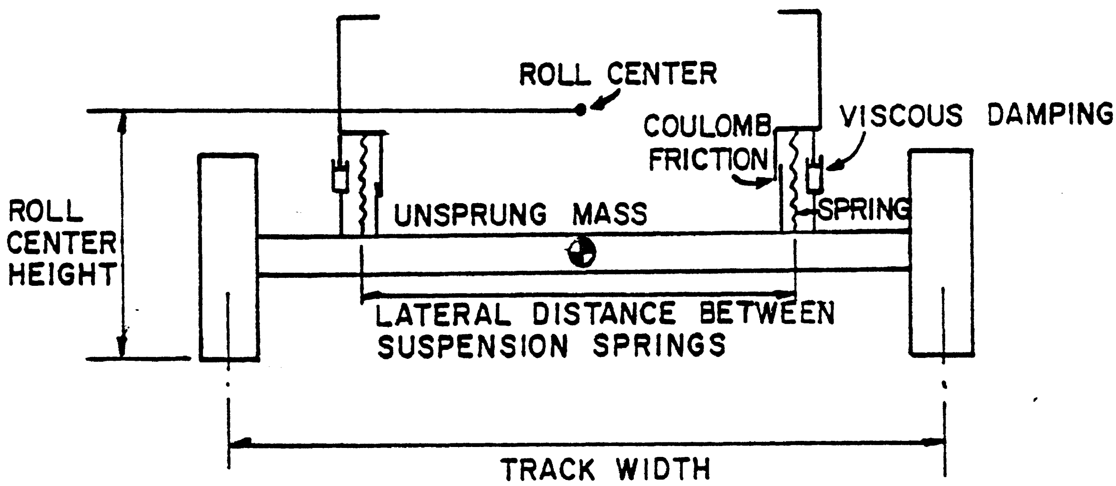

Figure 3. Model of suspension systems.

Tandem axles are modeled as two single axles with static and dynamic load transfer interactions. The suspension spring rate may be given as a single (linear) characteristic; or by use of a negative integer entry, the program is keyed to accept a multi-point table to define nonlinear characteristics. In addition, the suspension properties may be given different values on the left and right side of the vehicle in a side-to-side option exercised by making a double entry on the specified line.

The tires and wheels are described by parameters that represent:

-Dual tire separation (except truck/tractor front axle)

-Tire stiffnesses (cornering, longitudinal, camber, aligning moment, and vertical spring rate)

-Tire loaded radius

-Polar moment of inertia

The stiffness values represent the elastic properties of the tire and its frictional coupling to the road surface. The cornering and longitudinal stiffnesses are especially significant to turning and braking performance, respectively. Hence, either may be entered as a multi-point table by use of the negative integer keying method described above. Additionally, the side-to-side option may be used with any of the above parameters.

Brakes are described by three parameters—a time lag and rise time representing the brake pressure transmission characteristics of the pneumatic lines to each brake, and the brake torque characteristics. The last parameter may be given as a multi-point table if so desired. Each brake of the vehicle may be described individually by each of the above parameters. Further, an antilock brake control may be specified for each wheel position. The antilock simulation is a general purpose program which requires the user to define the operating characteristics of each antilock system being specified.

As input data is read, the data is normally "echoed" as the first pages of output. At the completion of the input read process, the program calculates necessary properties of the total vehicle combination and prints a page of output containing a summary of those vehicle properties. The program then "runs," solving the differential equations of motion for the vehicle until the vehicle reaches a full stop, a default stop (such as rollover), or until the designated maximum simulation time is reached. At various points during the run, simulation output is printed, which (at the option of the user) may include time-based values for the vehicle motion variables, tire forces at each axle, braking conditions on each axle, tire cornering conditions, and the suspension motions and forces.

The vehicle motion variables are given as instantaneous values of (translational and rotational) position, velocity and acceleration. Auxiliary information on the radius of turn, body sideslip angle and articulation angles (of tractor-trailer vehicles) is also provided. The tire forces include vertical, lateral and longitudinal components, the associated coefficients of friction being utilized, and the wheel operating conditions relating to steer and slip angles, brake torque, and wheel longitudinal slip. The suspension motions are defined by the vertical and roll positions and velocities. Suspension forces are those derived from spring deflections, damping effects and auxiliary roll stiffness.

2.3 Uses of the Program

The great versatility of the T3DRS:V1 program in representing commercial vehicle types and components in steering and braking maneuvers gives it great utility. It can be used to simulate the following vehicle configurations:

- Straight truck, empty and loaded
- Bobtail tractor
- Tractor-semitrailer (3 to 5 axles), empty and loaded
- Tractor-semitrailer-full trailer (5 to 9 axles), empty and loaded
- Tractor-semitrailer-full trailer-full trailer (7 to 13 axles) empty and loaded

For simulation of braking performance, the program incorporates representation of truck air brake systems, antilock wheel control systems and tire-road friction models. Typical examples of braking studies for which it can be or has been used are:

1) Stopping distance performance
2) Effects of brake timing
3) Dynamic behavior in braking
4) Comparisons of antilock wheel control logic
5) Influence of tire-road friction coupling
6) Split friction surfaces
7) Brake proportioning
8) Tandem-axle effects on braking limits

For simulation of cornering performance behavior, the program allows state-of-the-art representations of truck tire lateral force characteristics (with roll-off effects during combined braking), and vehicle suspension properties of significance to cornering behavior. Typical examples of studies involving cornering are as follows:

1) Understeer/oversteer properties of commercial vehicles
2) Determining cornering limits
3) Assessing the tandem-axle effects on cornering
4) Jackknife prediction
5) Effects of suspension properties on cornering and cornering limits
6) Accident simulation

In addition to the above, the program can be operated open-loop (defined steer angle inputs) or closed-loop (defined path input), on roads of specified grade or cross-slope, and on roads defined by the user.

2.4 Validation

The validity of T3DRS:V1, like any computer program, is dependent on the accuracy and execution of program statements, the capabilities of the simulation models, and the quality of the vehicle and maneuver descriptions defined by the input data.

The basic modeling methods used in T3DRS:V1 have evolved from the predecessor programs. A general discussion of the capabilities and validation of these programs was provided to the FHWA as a Task B report in this project, and is included in this report as Appendix B.

The methods reflect the most practical approaches to mathematical representation of commercial vehicles for general study of braking and handling performance. Over the years, modeling has grown more in sophistication than in detail. For example, early models for truck brake systems extending to mechanical details within the individual brakes have proven no more capable of predicting braking performance than the "black box" representation as a pressure-input, torque-output device. Hence the latter approach is used in T3DRS:V1, with a substantial saving in the complexity associated with understanding and using the simulation. With nearly every component model used in the simulation, there are instances where more modeling details would be appropriate for the study at hand; yet, provision for every instance would result in a simulation for which the input data requirements would be untenable. To some extent, these needs are provided for in T3DRS:V1 by allowing optional use of lookup tables, in lieu of a single numerical parameter, as a means to describe component characteristics in more detail when needed.

Validation of a new computer program is an essential step. For T3DRS:V1, the available possibilities included comparison against analytical models, other simulation programs on hand at the Institute, and vehicle test data acquired by the Texas Transportation Institute [6]. The validation plan, included as Appendix C, contained all these elements. Parametric data needed to describe the TTI test vehicle for the validation tests are given in Appendix D of this report.

2.4.1 Low-Speed Cornering. The simulation predictions of low-speed cornering behavior serve as a very fundamental check of programming accuracy, ensuring that kinematic equations and unit conversions are correct. Low-speed cornering can be modeled by closed-form analytical equations [7]. Figure 4 shows a comparison of the T3DRS:V1 predictions of low-speed cornering behavior for trucks against analytical models and experimental data from another project [7,8]. The vehicle is a three-axle

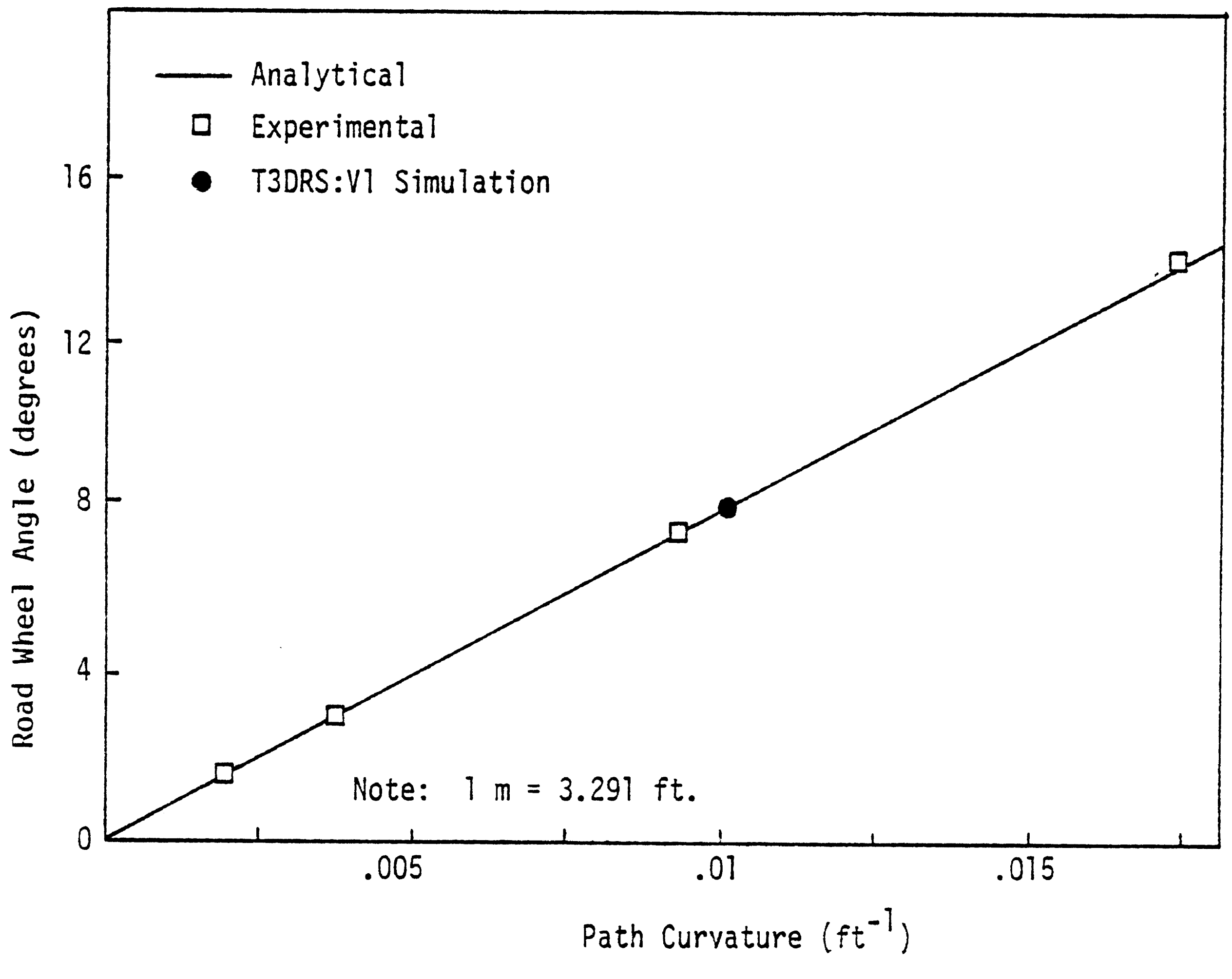

Figure 4. Comparison of analytical, experimental and simulation predictions for low-speed cornering of a loaded three-axle straight truck.

straight truck loaded to 44,500 lb (20,185 Kg) gross vehicle weight. At the selected eight-degree steer angle, the program predicts a path curvature in agreement with the analytical model and experimental data. Similar agreement may be expected at other steer angles.

The cornering of a three-axle vehicle differs measurably from a similar two-axle vehicle because of the tandem-axle sideslip that must occur. In addition, dual tires alter the cornering by the self-aligning movement they generate. These effects are included in this example, serving to verify the simulations representation of the effects.

2.4.2 High-Speed Cornering. As speed increases in a cornering maneuver, the lateral acceleration induces greater slip angles at the wheels, vehicle roll (and associated roll steer effects), and lateral load transfer on the axles. Steady-state cornering is characterized by the understeer gradient representing the change in steer angle with increasing lateral acceleration at a given radius of turn.

Figure 5 shows the change in steer angle with lateral acceleration for the same loaded three-axle straight truck as in the previous figure. The experimental data covers a range of speeds and radii of turn [8]. The Phase II simulation and the T3DRS:V1 differ slightly, but inconsequentially, in the predictions of steer angle change with lateral acceleration due to slight differences in behavior exhibited by the tire models; but both closely match the truck performance.

In addition, both simulations exhibit limit behavior, indicated in the simulation runs by inability to achieve a steady-state turn, at just over 0.3 g's lateral acceleration, as was observed on the test vehicle.

2.4.3 Transient Response. The transient response behavior of a vehicle is dependent, among other things, upon its inertial properties. A number of different validation tests were performed to assess accuracy in predicting transient behavior.

Figures 6, 7, and 8 show comparisons of the T3DRS:V1 simulation with the TTI tractor-semitrailer test vehicle in terms of the significant motion parameters in J-turn maneuvers. The first figure represents tests with the vehicle empty, whereas the second and third are for the loaded

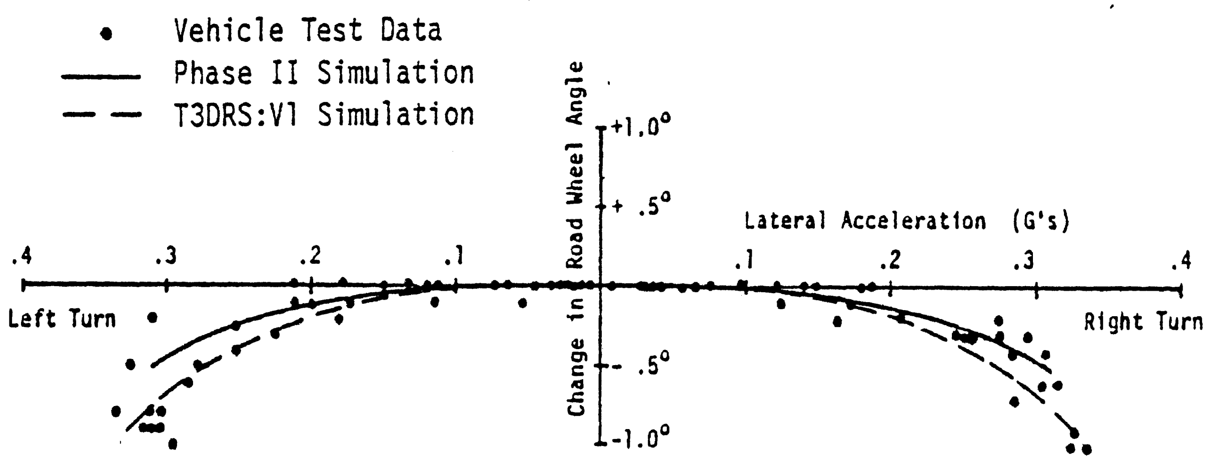

Figure 5. Comparison of experimental and simulated results for high-speed cornering of a loaded three-axle truck.

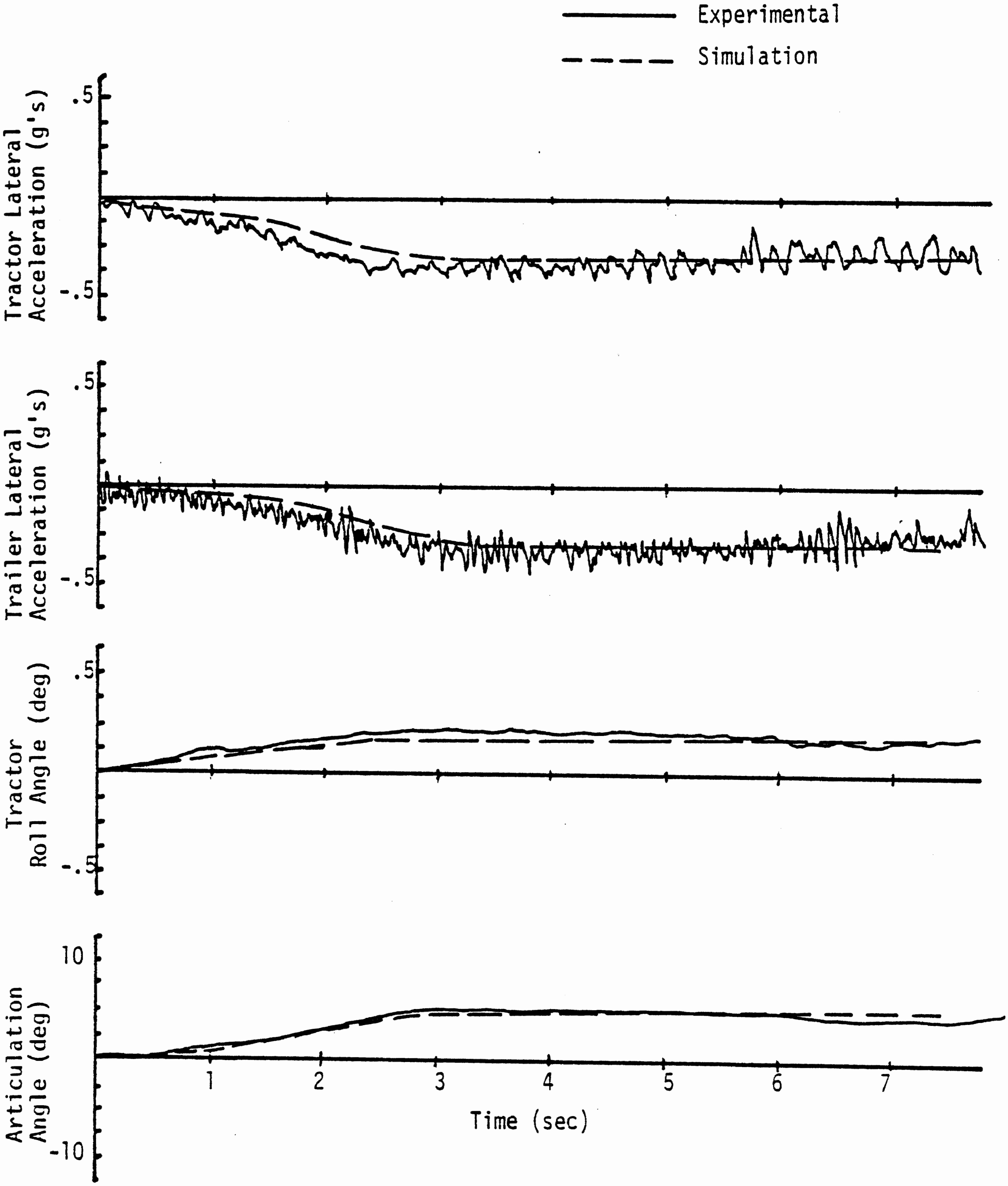

Figure 6. Comparison of simulation with experimental results in a J-turn maneuver (TTI Test #88, empty, fifth wheel-rear, trailer bogey-rear).

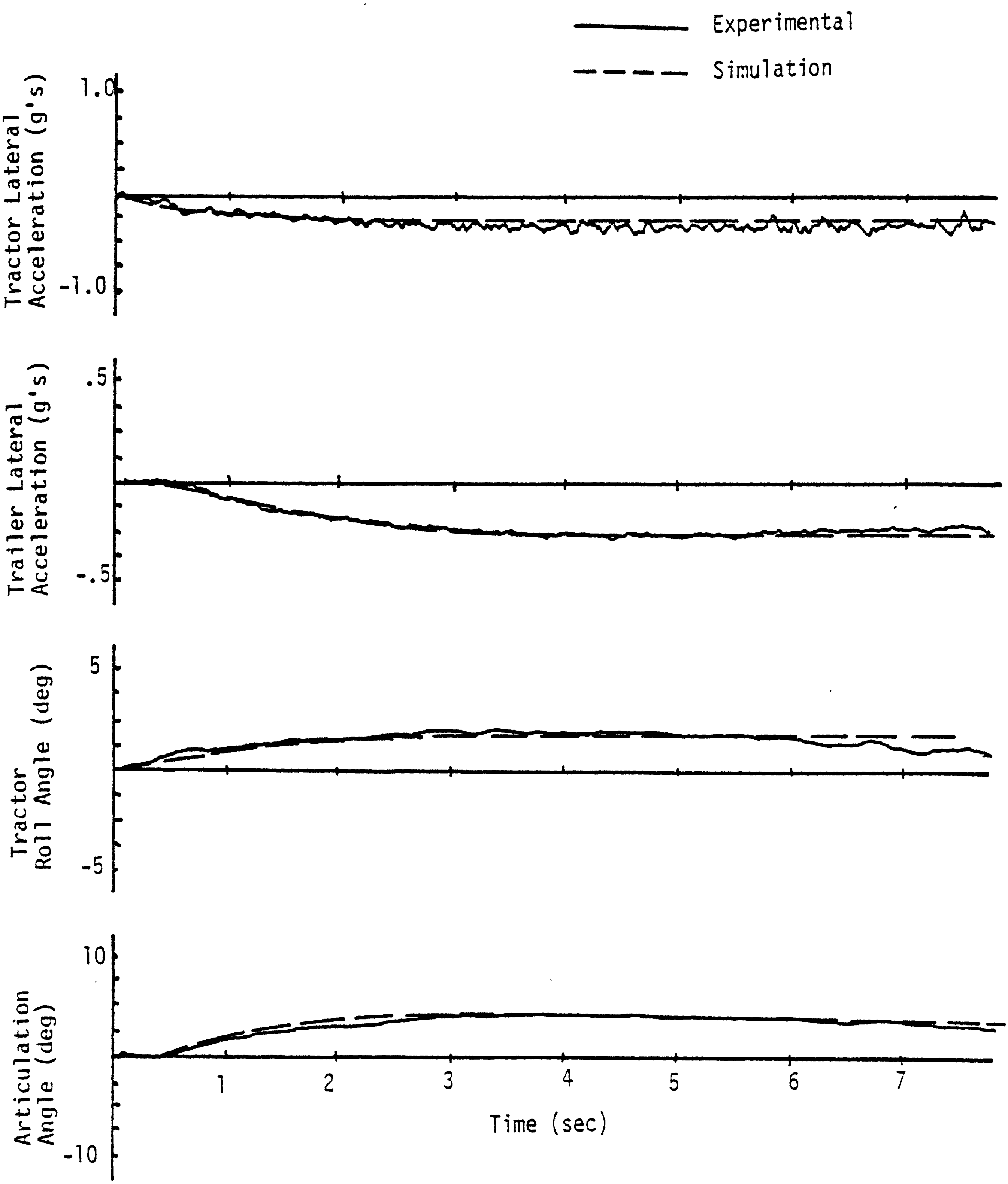

Figure 7. Comparison of simulation with experimental results in a J-turn maneuver (TTI Test #277, loaded, fifth wheel-rear, trailer bogey-mid-position).

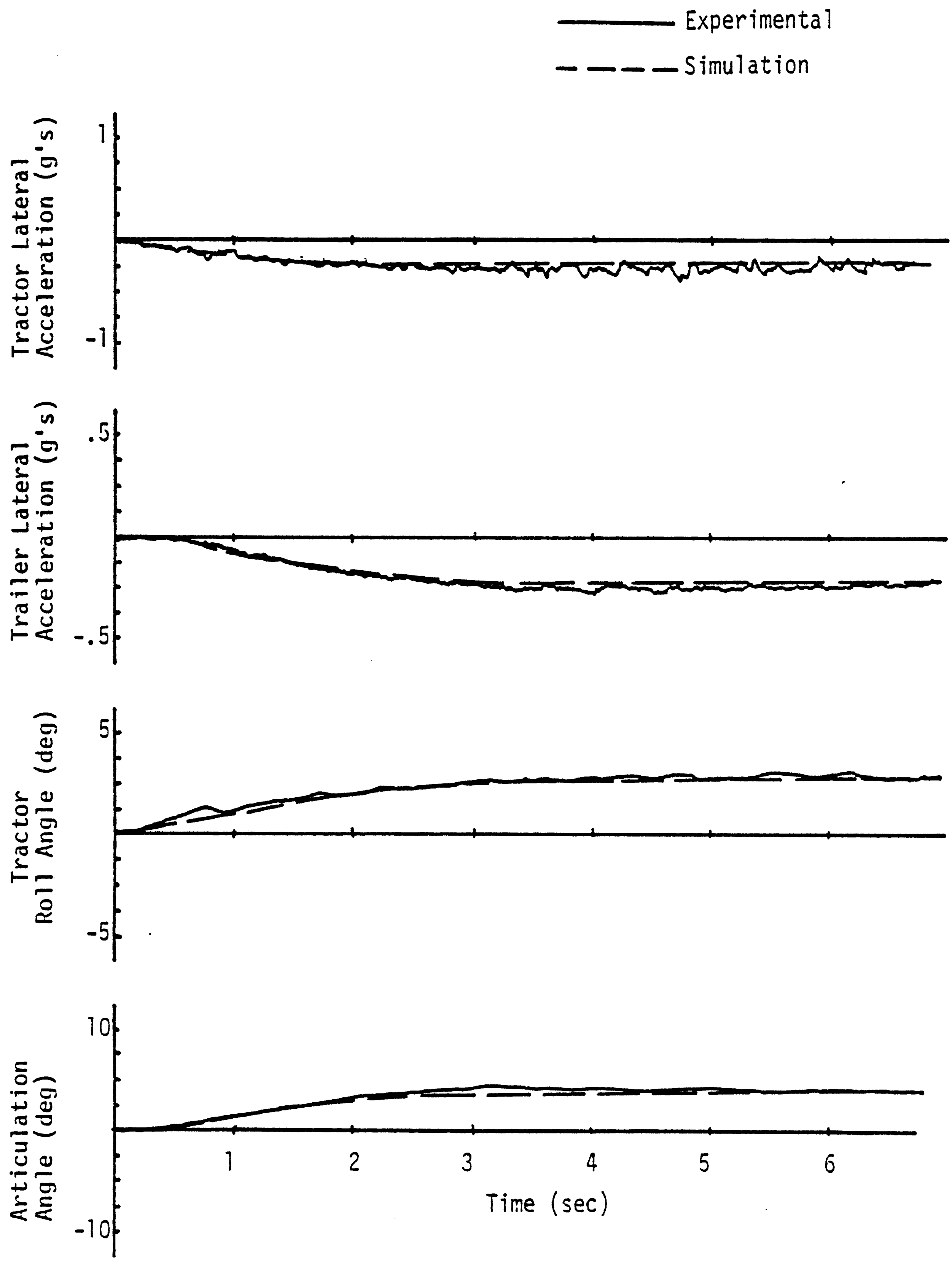

Figure 8. Comparison of simulation with experimental results in a J-turn maneuver (TTI Test #285, loaded, fifth wheel-rear, trailer bogey-forward).

condition with different locations of the sliding trailer axle bogey. In simulating these maneuvers, the actual tractor front-axle steer angles were used to define steer input tables for the simulation run. In all cases, good agreement between the test results and simulation is obtained. The magnitude of the differences observed is generally considered small for this type of validation procedure. In Figure 6, much of the difference is manifest as an apparent lag in the response of the simulation. However, that interpretation should not be applied because the start of the test maneuver is not defined that accurately in time. Hence, if desired, the registration of the simulation and test data could be validly altered to improve the agreement.

Figures 9, 10, 11, and 12 show comparisons of the simulation with the TTI test vehicle in double lane-change maneuvers. The first of these four figures is for the empty vehicle, while the last three are for the loaded vehicle with different trailer axle bogey positions. For simulation of the double lane-change maneuver, the lane-change path is defined in the simulation input allowing the path-follower model to steer the vehicle. The front-wheel steer angles taken by the simulation closely replicate those used by the TTI test driver in accomplishing the maneuver. Again, the motion response of the simulation closely follows that of the test vehicle, whether empty or loaded, and with the trailer bogey in any of the three positions.

Figures 13, 14, and 15 show a comparison of the step-steer response of a doubles combination predicted by T3DRS:V1 with that obtained from the HSRI linear doubles model [9] for equivalent vehicle parameters. The principal differences between these models in the linear range are (1) the linear model operates at constant forward velocity and (2) side-to-side load transfer due to roll is not accounted for in the linear model.

2.4.4 Straight-Line Braking. Example comparisons of straight-line braking performance are shown in Figures 16 and 17. The experimental data is obtained from the TTI tests with partial braking, both empty and loaded. The brake torque characteristics assigned to each axle in these simulation tests were calculated from performance of the vehicle in axle-by-axle braking tests incorporated in the TTI test program. Figure 16,

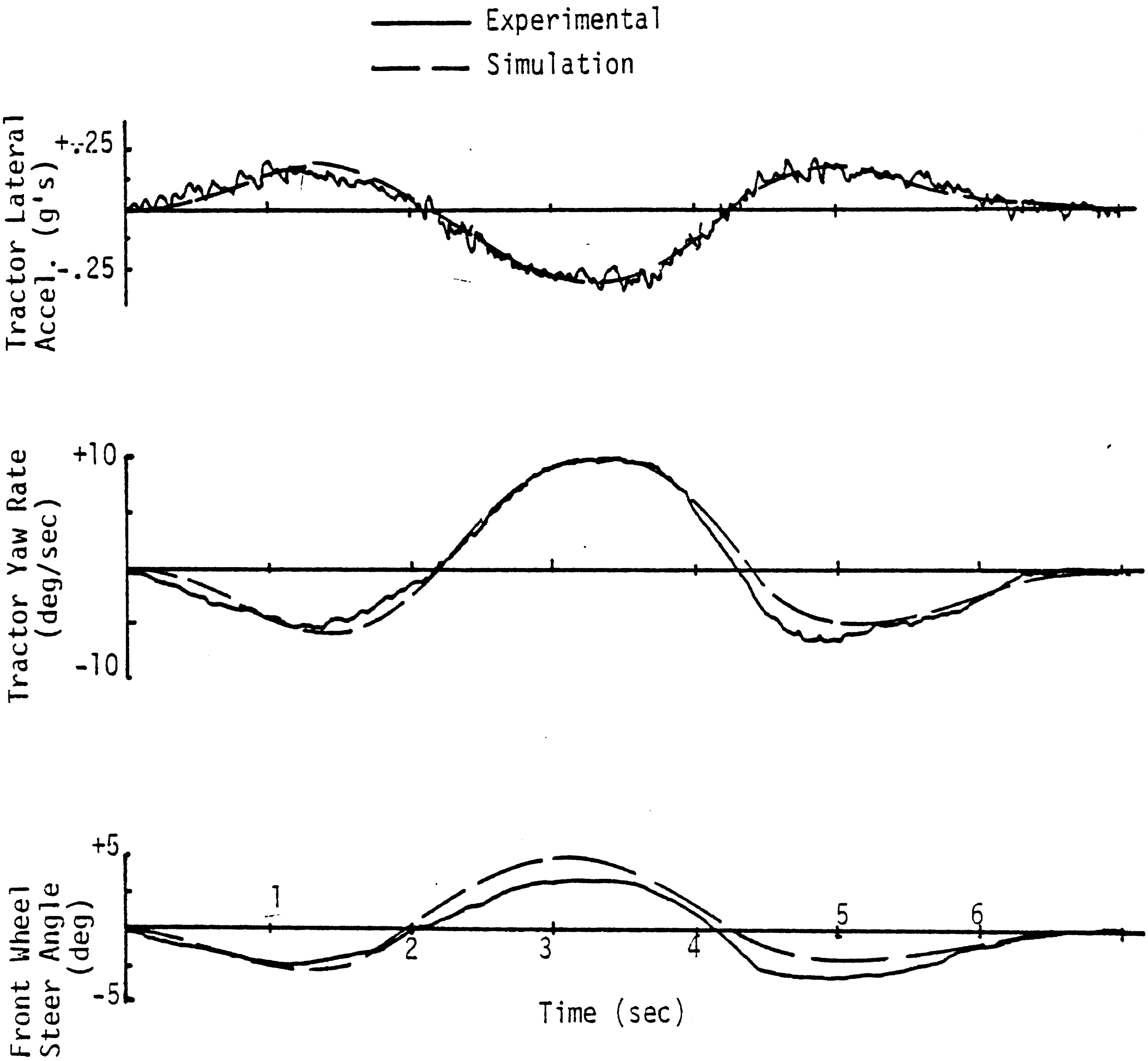

Figure 9. Comparison of simulation with experimental results in a double lane-change maneuver (TTI Test #199, empty, fifth wheel-midpoint, trailer bogey-rear).

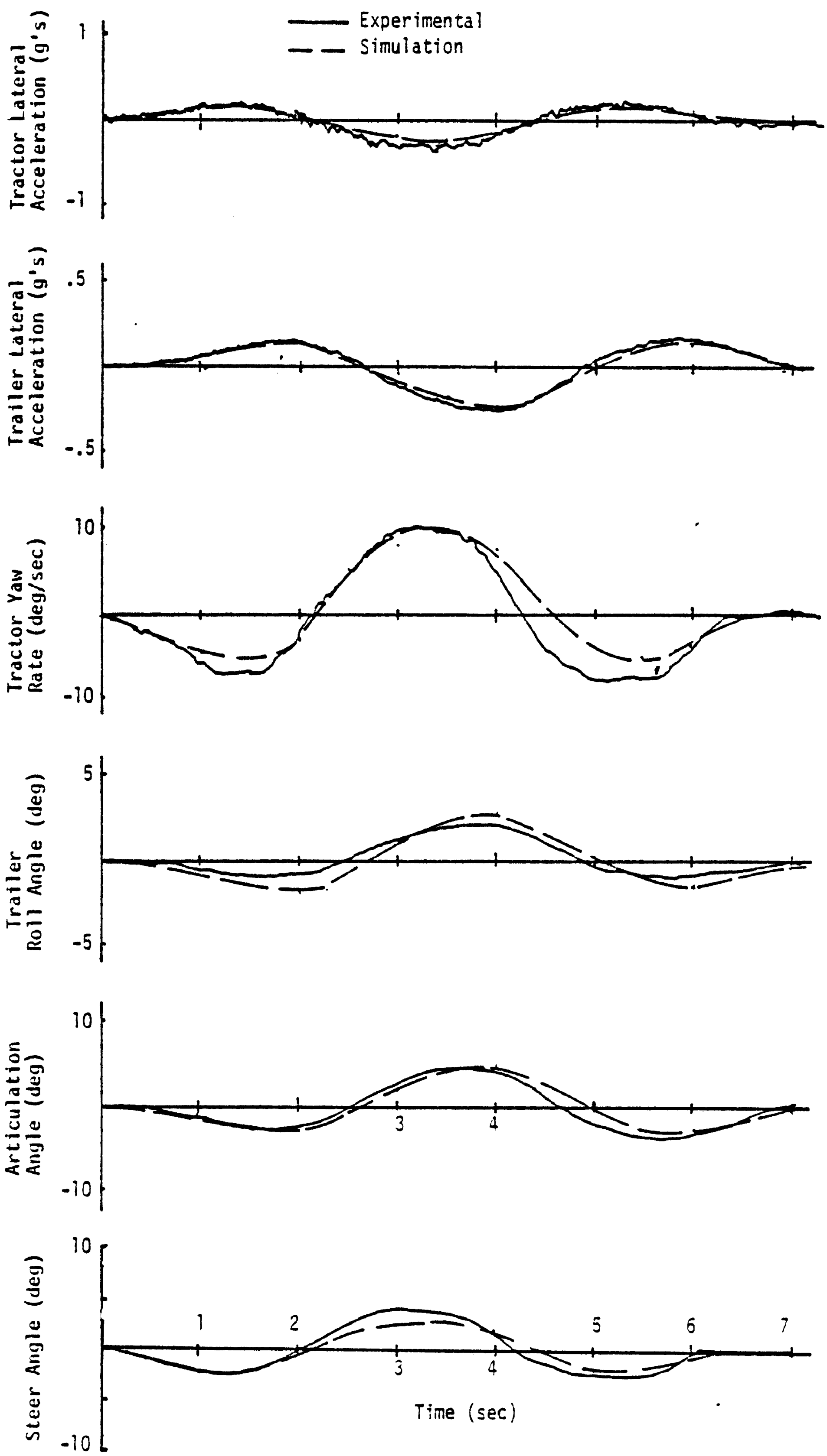

Figure 10. Comparison of simulation with experimental results in a double lane-change maneuver (TTI Test #316, loaded, fifth wheel-mid-position, trailer bogey-forward).

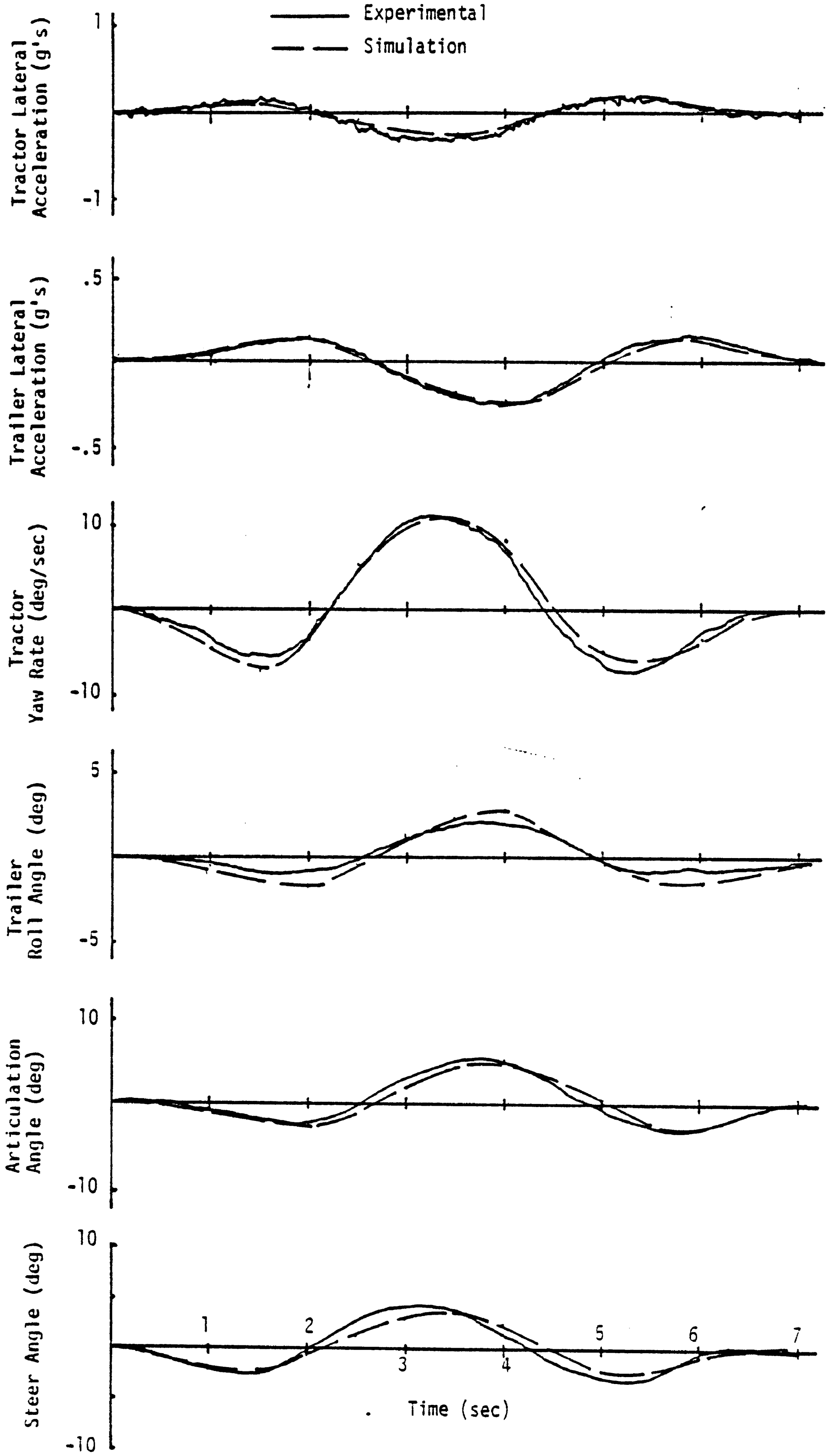

Figure 11. Comparison of simulation with experimental results in a double lane-change maneuver (TTI Test #320, loaded, fifth wheel-mid-position, trailer bogey-midposition).

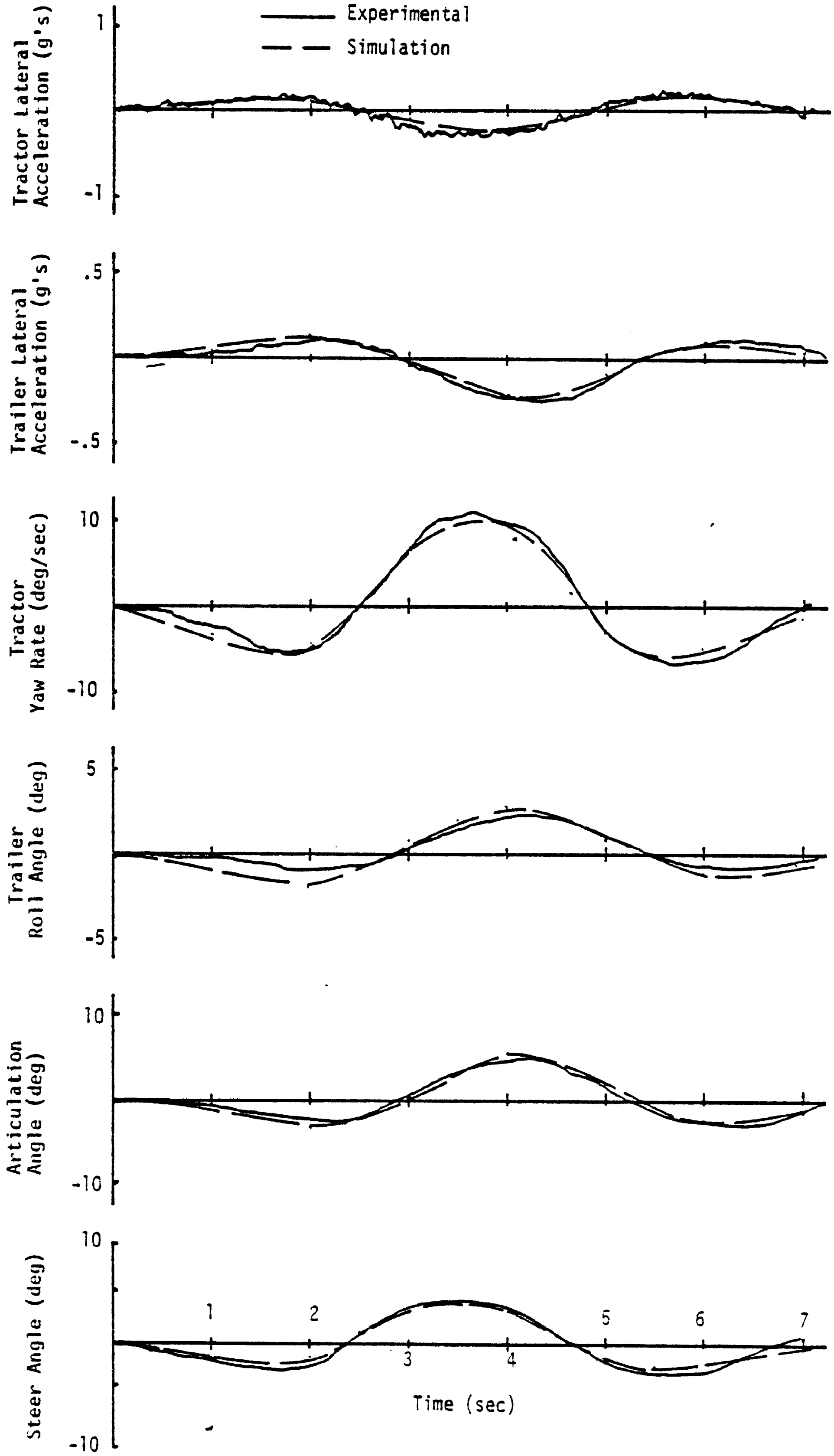

Figure 12. Comparison of simulation with experimental results in a double lane-change maneuver (TTI Test #322, loaded, fifth wheel-mid-position, trailer bogey-rear).

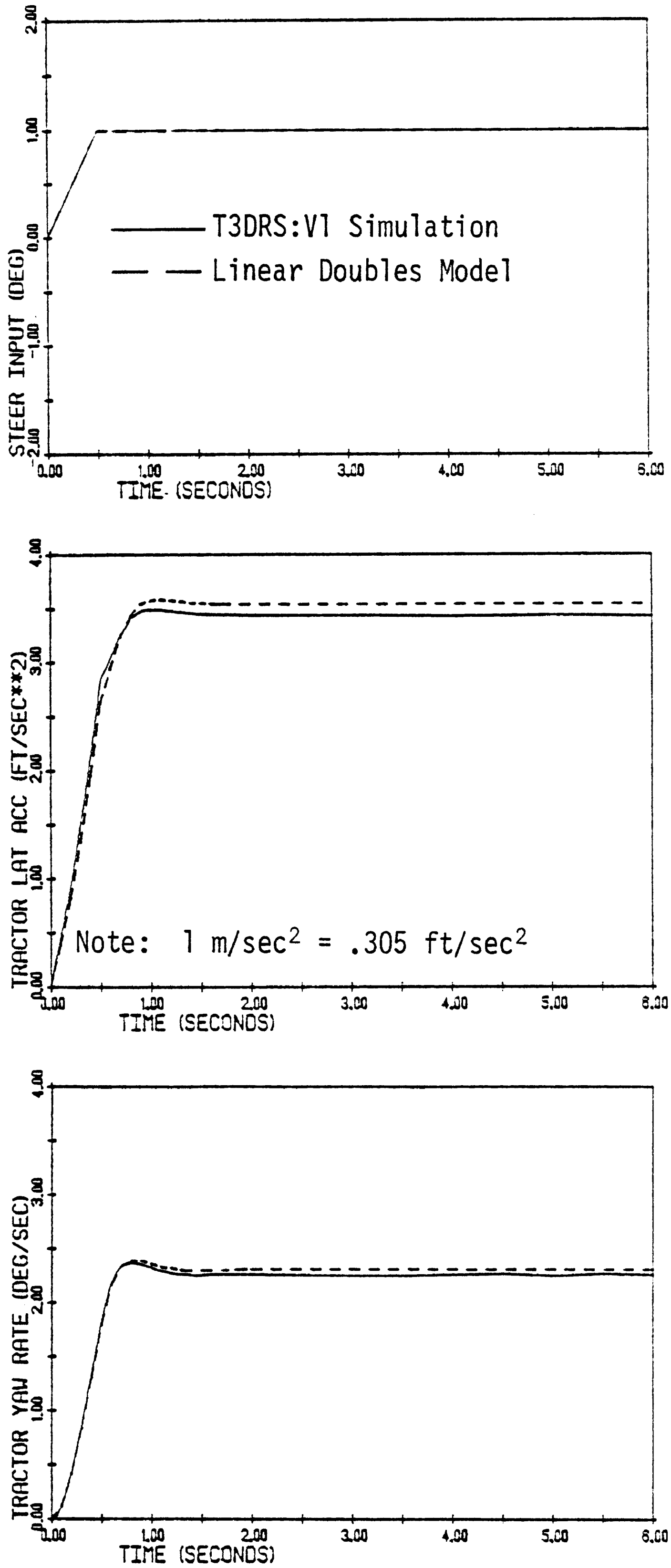

Figure 13. Tractor response comparison between linear doubles model and T3DRS:V1 simulation in a step-steer maneuver.

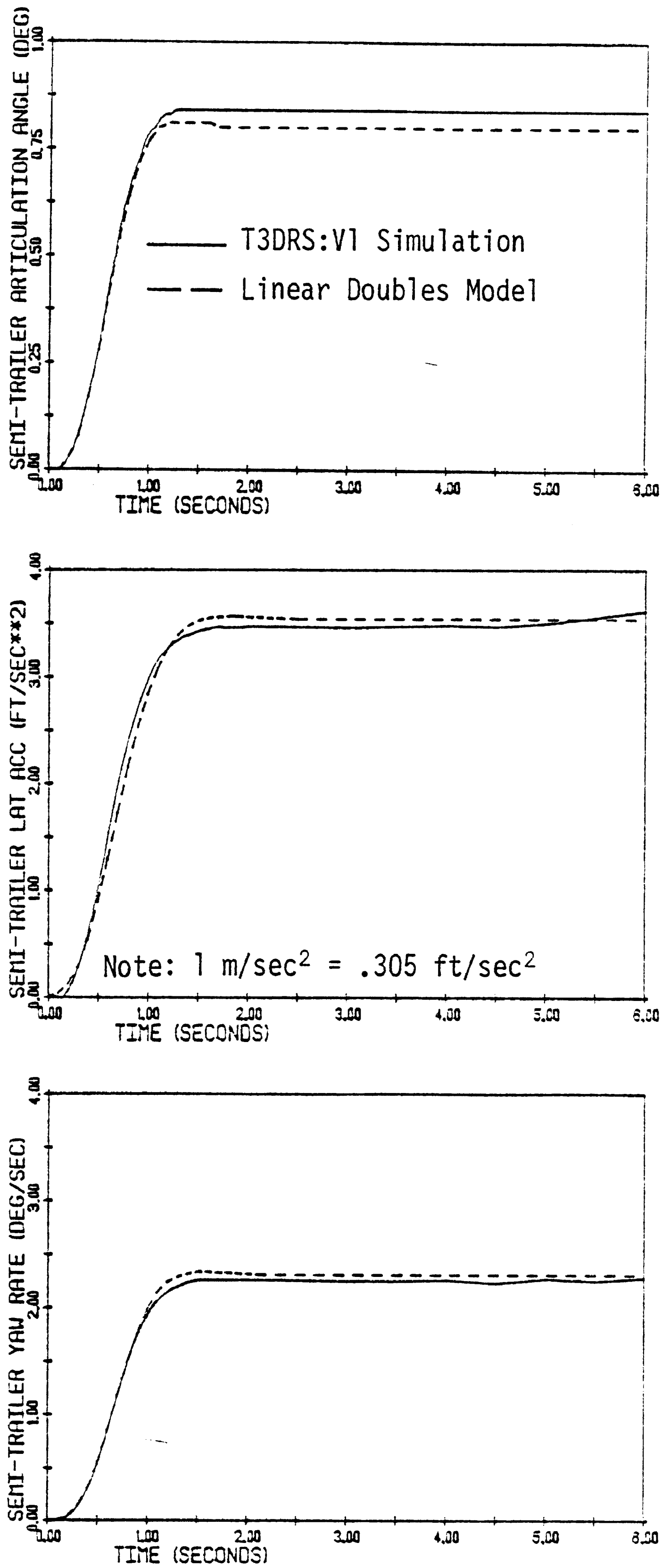

Figure 14. Semitrailer response comparison between linear doubles model and T3DRS:V1 simulation in a step-steer maneuver.

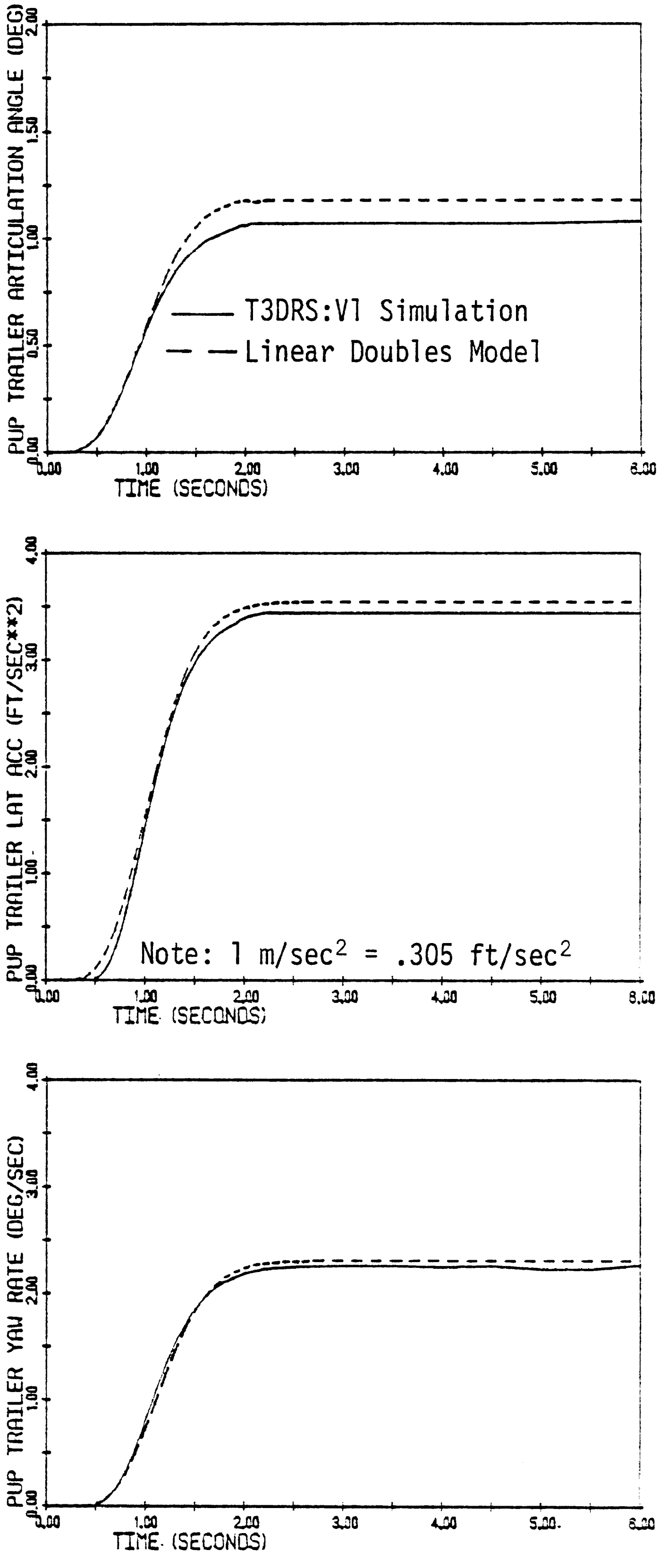

Figure 15. Pup trailer response comparison between linear doubles model and T3DRS:V1 simulation in a step-steer maneuver.

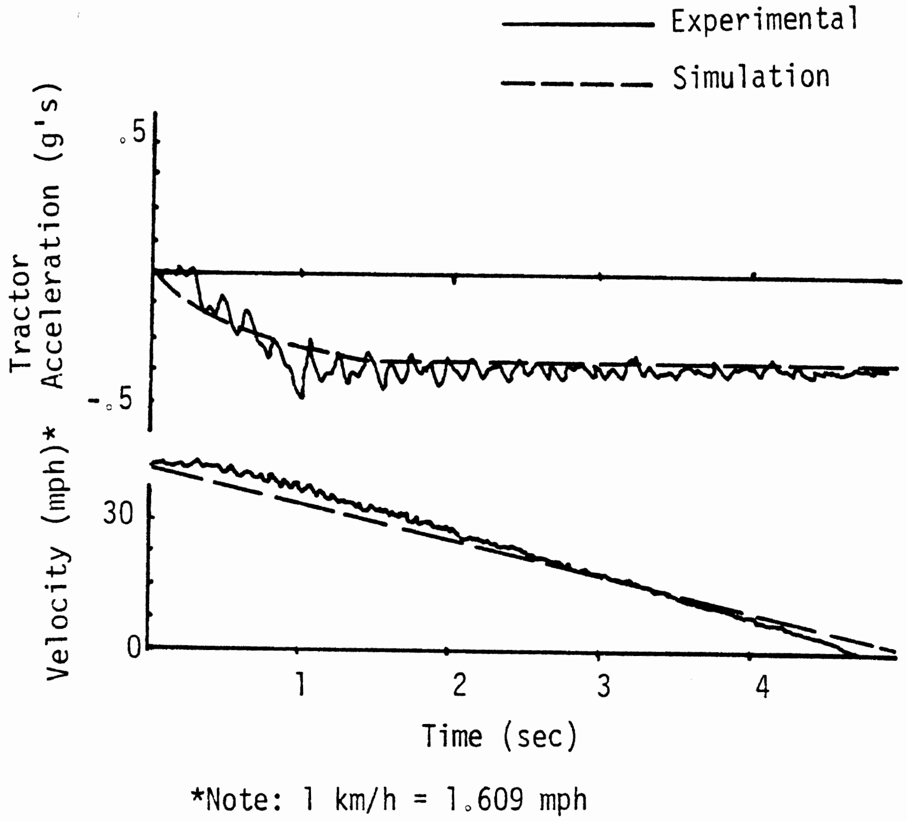

Figure 16. Comparison of simulation with experimental results in a straight-line braking maneuver (TTI Test #77, empty, fifth wheel-rear, trailer bogey-rear).

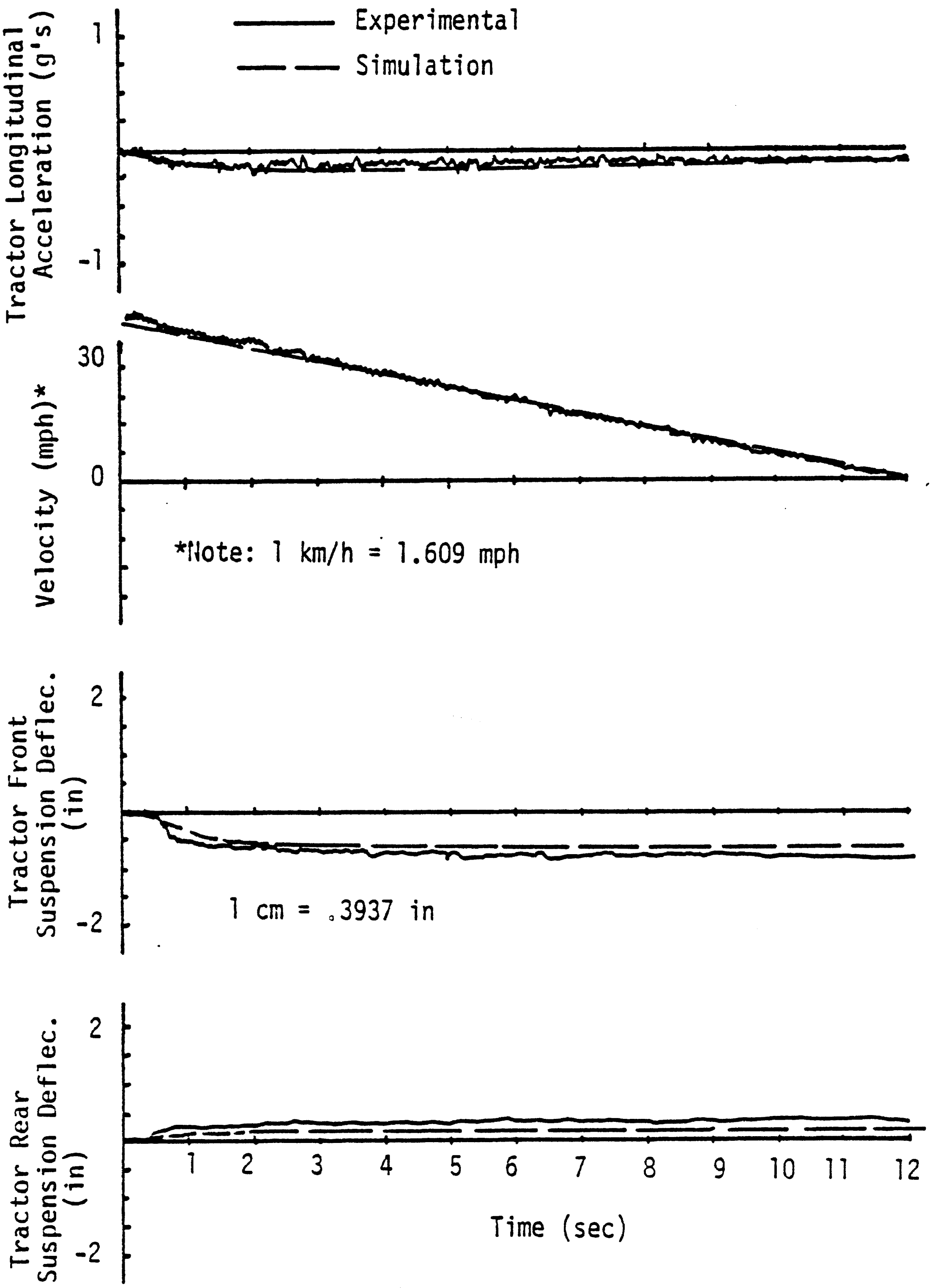

Figure 17. Comparison of simulation with experimental results in a straight-line braking maneuver (TTI Test #261, loaded, fifth wheel-rear, trailer bogey-rear).

for the empty tests, shows the agreement in terms of tractor longitudinal acceleration and velocity time histories. The loaded test of Figure 17 shows tractor front and rear suspension deflection, as well. Although these deflections are still relatively small, even with the loaded combination, they are included here to give an indication of the simulation's capability for predicting those parameters. Good agreement is evident in the case of front suspension deflection. The relative error is larger in the case of the rear suspension, although that error is on the order of tenths of an inch, undoubtedly reflecting hysteretic effects.

2.4.5 Braking in a Turn. Example comparisons of braking-in-a-turn maneuvers are shown in Figures 18, 19, 20, and 21. All examples are for a partial braking level equivalent to approximately 20 psi (140 kPa). The simulation input for these runs included the measured treadle pressure and measured steer angle of the front wheels.

Figure 18 is for the TTI test vehicle at the empty condition. Figures 19 and 20 are the TTI test vehicle loaded, with the trailer axle bogey at the midpoint and forward positions, respectively. With the exception of the tractor roll angle in Figure 18, the simulation's replication of the vehicle behavior in each of the primary motion variables shown is very good. The turning maneuver at the beginning of the simulation run is evidenced by the immediate response in tractor lateral acceleration and roll angle, with delayed response in trailer lateral acceleration and articulation angle. The braking action at approximately four seconds into the test is seen in the longitudinal acceleration, shown for the tractor only in the figures. Since the turn radius is nominally being held constant, the decreasing velocity after brake application results in decaying lateral acceleration for the tractor and trailer beyond this point.

The replication of tractor roll angles is an issue requiring some discussion, not so much because of serious shortcomings in the simulation as in ambiguities in its measurement. In the simulation, the vehicle sprung mass is treated as a rigid body. In practice, tractors have been observed to exhibit significant levels of roll compliance in their frames [10,11]. (This is less of an issue with straight trucks because

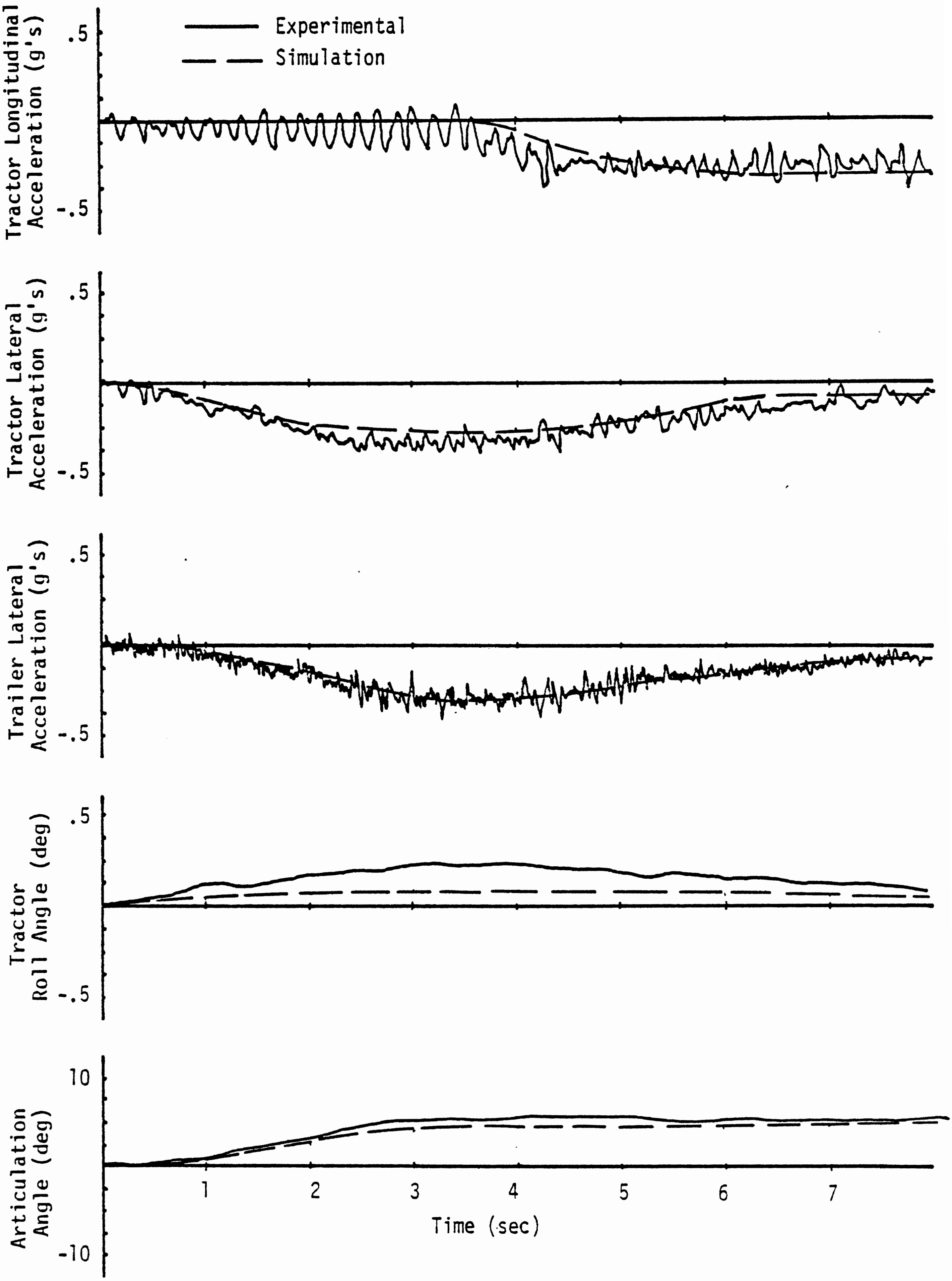

Figure 18. Comparison of simulation with experimental results for braking-in-a-turn maneuver (TTI Test #93, empty, fifth wheel-rear, trailer bogey-rear).

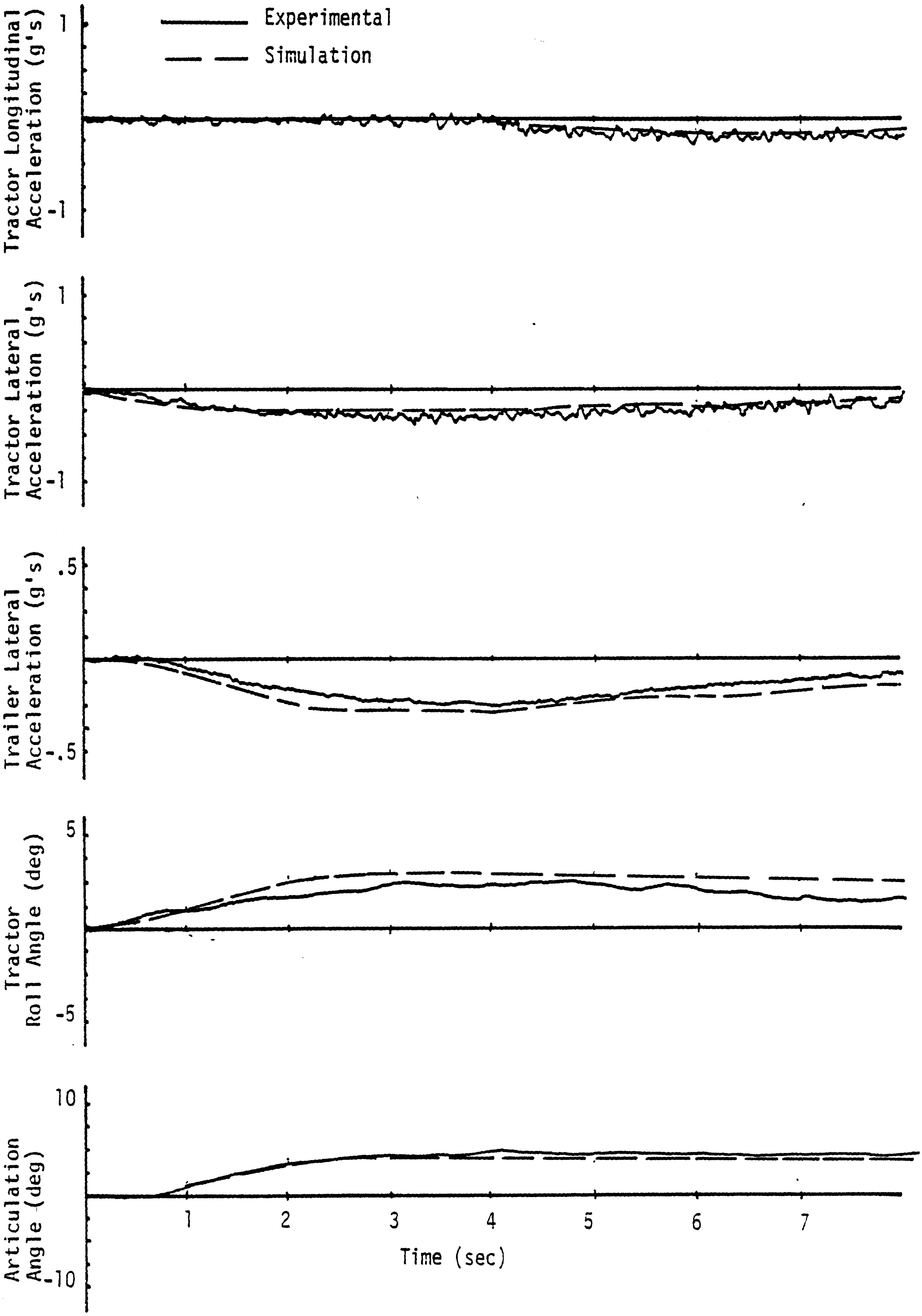

Figure 19. Comparison of simulation with experimental results in a braking-in-a-turn maneuver (TTI Test #281, loaded, fifth wheel-rear, trailer bogey-midposition).

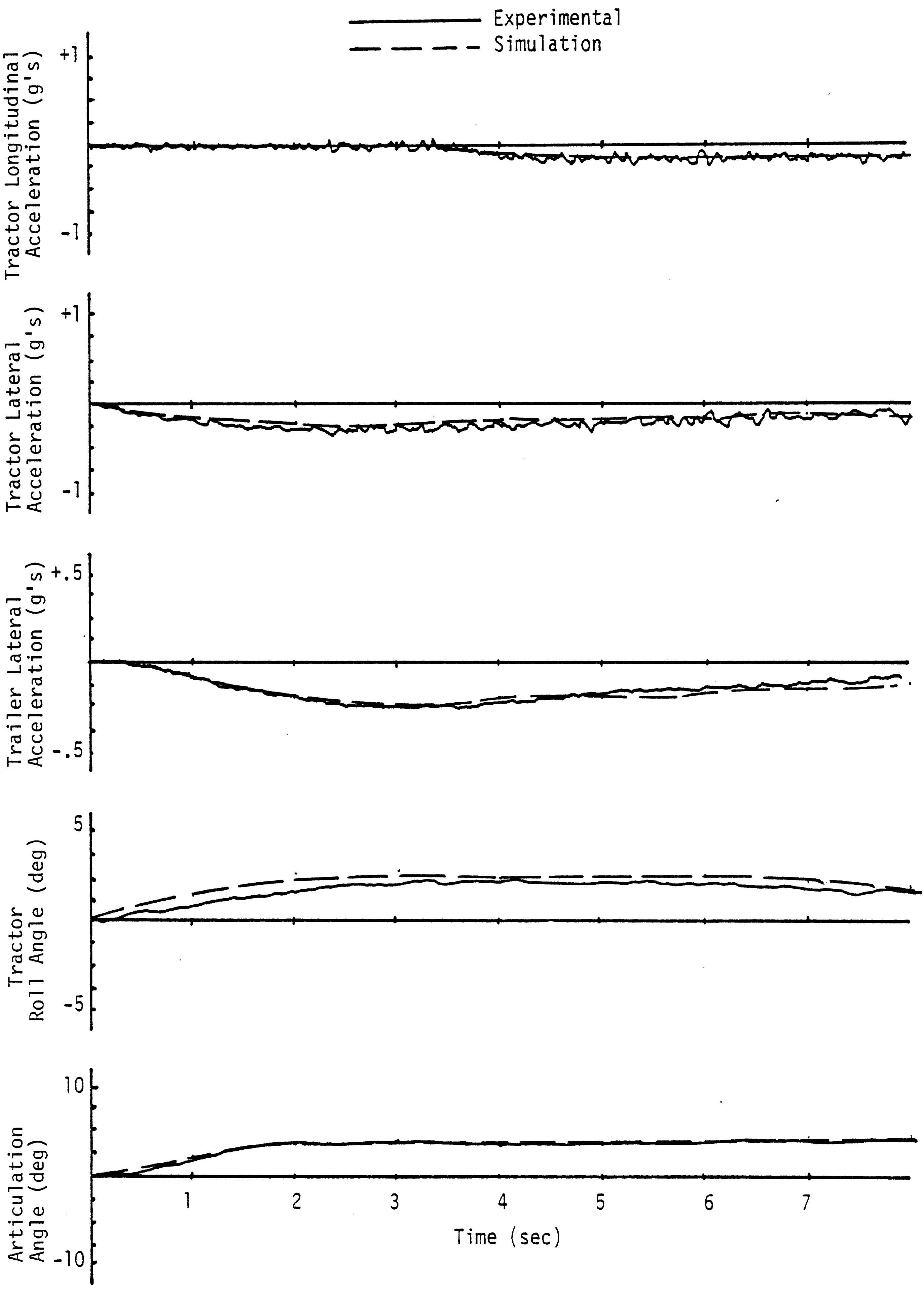

Figure 20. Comparison of simulation with experimental results in a braking-in-a-turn maneuver (TTI Test #287, loaded, fifth wheel-rear, trailer bogey-forward).

of the additional torsional stiffness normally contributed by vocational bodies.) In effect, the roll angle on the tractor varies with location. The variation in roll angle at different points on the vehicle is of little significance to performance in low-level cornering maneuvers and is significant only under certain circumstances in limit maneuvers [11]. However, the effect can contribute to poor agreement of roll angle data in validation. The roll angle data in the TTI validation tests was measured in the cab of the cab-over-engine tractor. The high cab with its compliant mount to the front of the vehicle frame is prone to roll somewhat independently of the rear of the vehicle. The result is most evident as an exaggerated roll angle, especially in the low center of gravity (empty vehicle) test condition. For example, comparing Figures 18 and 19 which are maneuvers of comparable severity, the measured tractor roll angle is effectively equivalent even though one is the test of an empty vehicle and the other is loaded. Hence, on this vehicle, cab roll is essentially independent of the loading condition. The simulation, however, shows the expected greater roll angle with the loaded vehicle since the simulated tractor is rigid and coupled to the loaded trailer. In effect, the roll angle predicted by the simulation would be expected to agree much more closely with frame roll angles measured near the fifth wheel, rather than in the cab.

Figure 21 is validation for a near limit braking-in-a-turn maneuver of a tractor-semitrailer unit tested by HSRI [12]. The semitrailer was a van-type loaded to achieve a 73,937-lb (33,538-kg) gross combination weight. In this example, measured brake treadle pressure and front-wheel steer angles were used as input to the simulation. Replication of the tractor yaw rate, lateral acceleration and longitudinal deceleration were found to be very good.

From the numerous examples that have been given here, it is concluded that the T3DRS:V1 computer simulation program is effectively free of errors; and with the models used, it is capable of validly predicting the behavior of heavy vehicles in braking and directional response. Testing the simulation up to the limits of cornering performance (as in Figure 5, Figure 21, and in other simulation tests that have been done at HSRI) indicates that the validity applies in maneuvers up to the limits

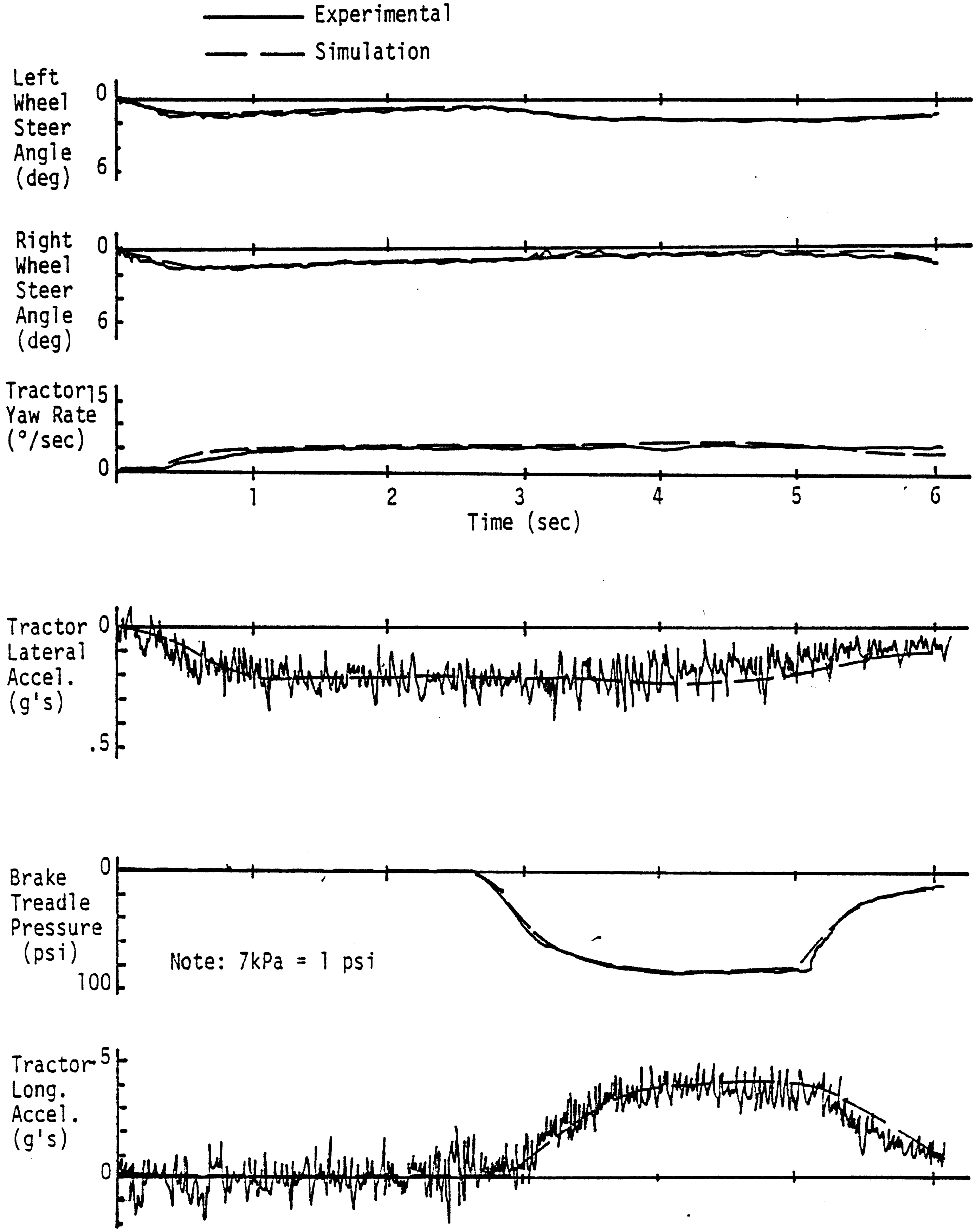

Figure 21. Comparison of simulation with experimental results in a braking-in-a-turn maneuver (HSRI test of a tractor-semitrailer [12]).

of rollover. The small angle assumptions used in the model representation, however, suggest that errors will increase when roll angles, slip angles or articulation angles approach 20 degrees. In general, this condition is well beyond the normal maneuvering limits of heavy vehicles. For example, rollover with a heavy vehicle is usually imminent once body roll angles approach 8 or 10 degrees.

Ultimately, the determinant of validity is the user-supplied input data and the interpretation applied to the results. In the special case where a direct comparison between a vehicle and simulation (i.e., validation) is intended, the acquisition of accurate experimental measurements and vehicle data is costly and time consuming. Fortunately, the usefulness of these simulation programs are not dependent on every user going through the same process. In most applications, the simulation is used for studying generalized performance and sensitivity of performance to vehicle parameters. In such cases, the user can assume, for example, a given tire characteristic and investigate vehicle performance with that tire, knowing that it is typical, but yet, not precisely equivalent to any specific tire on hand.

2.5 Documentation

The new documentation specific to the T3DRS:V1 program is contained in the User's Manual [4] and the Programmer's Manual [5]. While these, respectively, describe the external (input/output) and internal (program statements and flow) characteristics of the program, it is not practical to assemble the rationale, models, and execution of every aspect of the simulation in these documents. Over the years, a number of publications have been produced by this Institute describing details in the development of heavy vehicle computer simulations. Most of these publications are available in the libraries and may be referred to when specific questions arise. Where it is necessary to determine the exact details of execution within T3DRS:V1, the appropriate section of the Programmer's Manual should be consulted.

As an aid to program users, general areas of interest are discussed below with references suggested as a source of more detailed exposition.

2.5.1 Input/Output. The input/output parameters and format are unique to the T3DRS:V1 program. The User's Manual [4] gives a detailed description of each line of input, its location in the input stream, engineering units and the format required. Likewise, it provides a detailed description of the output available, indicating where each parameter is found in the output and the interpretation of its value. The inertial, body-fixed and tire coordinate systems are described as needed to interpret the output data.

2.5.2 Program Statements. The program statements and flow are unique to T3DRS:V1. The Programmer's Manual [5] describes the internal structure of the program in terms of:

-A map of subroutine calls

-Flow charts

-Discussion of each subroutine

-Tables of variables

-Programming convention

-Source list

2.5.3 Equations of Motion. The equations of motion for the sprung and unsprung masses are contained in the subroutine FCT1 of T3DRS:V1. Appendix E of the User's Manual [4] contains a general discussion of the equations of motion. The actual equations in their general form are given on page 167 of the Programmer's Manual [5] and are discussed in Section 2.3.4. A more extensive discussion of the methods used in formulating the equations, the axis systems used, and the Euler angle transformations is available in Chapter 2 and Appendix B of the Phase II program report [2]. The method of solving these equations involves certain approximations that have been developed as time-saving methods in simulation and are described in Section 3.3.2.1 of the Phase II report [2] and in Reference [13].

Time-saving methods have also been applied to calculations in the tire rotation degree of freedom that avoids the necessity of an integration step. The treatment is described in the Phase II report [2], Section 3.2.4, and in Reference [14].

The tire model has been changed from that used in earlier programs and the description given in the User's Manual [4], Section 3.3.3, should be applied.

The operation of the brake system is governed by equations given in the Phase III report [3], Section 2.3.1. The antilock system simulation is discussed in Appendix D of the User's Manual [4].

Calculations of the reactions within the suspension systems are rather straightforward with two exceptions: (1) coulomb friction is represented by a limiting function as described in Section 2.3.2 of the Phase I report [1] and in Reference [15] and (2) tandem suspensions may exhibit load transfer in braking. The representation of these effects occur in subroutine LINE of T3DRS:V1. For details, the user should consult the Programmer's Manual [5], Section 2.3.9 and page 216.

All hitch points (fifth wheel and pintle hook) in T3DRS:V1 are treated as spring connections as a time-saving method. Details are available in the Phase II report [2], Section 3.5.1 and in Reference [15]. Details of the full-trailer dolly hitch and modeling are contained in the User's Manual, Section 3.5.1.

3.0 TRAINING SEMINAR

At the stage in the project where the computer simulation program had been provided to the FHWA and had achieved operational status on their computer facilities, a Training Seminar was held by the project research staff. The Seminar was held on September 25 and 26, 1979, in the auditorium at the Fairbanks Highway Research Station in McLean, Virginia. Presentations by the HSRI staff were given on the first day. A working session on the second day provided opportunities for the FHWA staff and other attendees to exercise the program under the guidance of HSRI staff members. The attendees included representatives from the Federal Highway Administration, National Highway Traffic Safety Administration, the Johns Hopkins University/APL and industry. Table 1 is a list of those who attended.

The purpose of the Training Seminar was to provide vehicle dynamicists and potential users of the simulation program with:

- A general overview and description of the program and modeling used
- A familiarization with truck components, their modeling representation, and parametric values
- A discussion of applications for which the program is intended with associated limitations and an assessment of validity
- A detailed description of input requirements and interpretation of the output obtained
- Information on program diagnostics
- First-hand experience in program operation

A general overview of the content of the presentations on the first day is given in the program agenda shown in Figure 22. This portion of the Seminar was recorded by FHWA staff on an audio tape recorder.

The working session on the second day was conducted in the same auditorium using on-line terminals set up by the FHWA. Simulation runs

Table 1. List of Training Seminar Participants

Truck and Tractor-Trailer Dynamic Response Simulation T3DRS:V1, FHRS, September 25-26, 1979.

Name	Affiliation
David Solomon	FHWA, Office of Research
John Viner	FHWA, Office of Research
Michael Freitas	FHWA, Office of Research
Leonard Meczkowski	FHWA, Office of Research
Mort Oskard	FHWA, Office of Research
Steven Breslin	FHWA, Office of Research
Glenn G. Balmer	FHWA, Office of Research
Lloyd R. Cayes	FHWA, Office of Research
Thomas Krylowski	FHWA, Office of Research
Donald Gordon	FHWA, Office of Research
Rudy Hegmon	FHWA, Office of Research
Doug Simmons	FHWA, Office of Development
Lynn Runt	FHWA, Office of Development
Eric Munley	FHWA, Office of Development
Yvonne A. Clarkson	FHWA, Data Systems Division
Robert Clarke	NHTSA Office of Heavy Duty Vehicle Res.
Sid Williams	NHTSA Office of Heavy Duty Vehicle Res.
Paul Bohn	Applied Physics Laboratory
Mike Butler	Applied Physics Laboratory
Alec Chen	Ford Motor Company
Christopher Winkler	HSRI
Paul Fancher	HSRI
Thomas Gillespie	HSRI
Garrick Hu	HSRI
Charles MacAdam	HSRI

FHWA TRAINING SEMINAR

\- PROGRAM AGENDA -

Presenter	Items
FHWA	1) Introductions
	2) Description of the Simulation Program
P. Fancher	a) Background on HSRI truck research and computer program development
G. Hu	b) Modeling approach
	c) Internal program structure
T. Gillespie	d) Overview of input
	e) Overview of output
	3) Applications
C. MacAdam	a) Handling studies
P. Fancher	b) Braking studies
T. Gillespie	c) Validation
P. Fancher	d) Limitations
	4) Detailed Description of Input/Output
T. Gillespie	a) Sprung mass modeling
C. Winkler	b) Suspensions
C. MacAdam	c) Tires
P. Fancher	d) Brakes
C. MacAdam	e) Antilock
T. Gillespie	f) Output
G. Hu	g) Diagnostics
	5) Working Session, Second Day

Figure 22. Training Seminar Agenda

of straight trucks and doubles combinations were performed, examining high-speed step-steer maneuvers, double lane changes controlled by the path-follower model, and vehicle performance with steer of only a single front wheel. Simulation output was available on a local line printer so that results could be obtained and discussed in a timely manner. The working session served as an excellent forum for informal discussion between potential users of the program and the HSRI staff.

4.0 STUDY OF SIZE AND WEIGHT

The ultimate objective in developing this computer simulation program for the FHWA is its utilization as an engineering tool in assessing the potential consequences of changes in size and weight limits applied to heavy vehicles. The consequences of interest are those that have direct influence on highway safety and on the structural loadings affecting long-term performance on the highway. Methodology for using the simulation to investigate the influence of truck size and weight in this context was developed in the project. The methodology addressed the issues of how vehicle performance could be measured and how a prototype study might be designed.

4.1 Vehicle Performance Measures

The ideal measure of highway safety that would be used to assess the influence of changes in truck size and weight limits would be based on correlation of accident statistics with vehicle types and performance characteristics. A literature review was conducted to identify the performance measures that could be applied to heavy vehicles as a potential correlate of accident frequency. In general, specific relationships between vehicle handling characteristics and accident causation were found to be nonexistent. Rather, the state-of-the-art in measurement of vehicle performance by automotive manufacturers, research organizations and the Federal government is represented by measures that have only an intuitive link to highway safety.

Twelve major publications representing a cross-section of the organizations involved in vehicle handling research were reviewed, cataloging the various measures of performance that have been used or proposed. Five of these are specifically directed toward measurement of performance of commercial vehicles. The measures were analyzed to select those which would constitute appropriate measures of the dynamics of cornering and braking by which to discriminate the influences of truck size and weight. A summary report of this study is included as Appendix A of this technical report. Table 13, Appendix A, lists the proposed performance tests defined in terms of the maneuver to be conducted, the performance measures to be acquired, and the acceptance criteria that may be applied.

4.2 Size and Weight Study Plan

In the original statement of this project, a study of size and weight effects within the project was planned. That intent was reflected in development of a Phase II Study Plan included as a part of Appendix C to this report. The execution of that plan was later dropped as an activity within the project because of resource limitations. The plan itself, however, serves as an example of the method by which the effects of size and weight may systematically be investigated.

Seven vehicles are defined, beginning with a typical five-axle tractor-semitrailer combination. From that baseline, additional vehicles are proposed, reflecting the following changes:

1) An increase in front axle load
2) An increase in tractor and trailer tandem axle loads
3) An increase in both front axle and tandem axle loads
4) The same increase in axle loads with an appropriate increase in center of gravity height
5) An increase in axle loads with an appropriate increase in trailer length
6) An increase in axle loads with an increase in axle ratings (by appropriate parametric changes).

Five maneuvers are proposed in the Study Plan, similar to those proposed in the performance measures of Appendix A. Differences in the conditions of the test reflect limitations of the simulation at the time the plan was prepared (i.e., open-loop sinusoidal steer maneuvers rather than closed-loop lane change requiring the path-follower model, not yet available). Additionally, some of the differences reflect more severe test maneuvers that can be readily attempted in simulation rather than in actual vehicle testing.

5.0 CONCLUSIONS

Inasmuch as this project has been primarily concerned with development of tools and methodology without actually making assessments of the effects of increased truck size and weight on dynamic behavior, the conclusions are limited to summary statements relating to the simulation program and its application to the problem.

The conclusions are as follows:

1) The T3DRS:V1 computer simulation program provided to the FHWA, in the opinion of HSRI, is considered to be the most versatile and easily used simulation currently available for investigating the dynamic behavior of heavy vehicles.

2) The subject program has been made operational on computer facilities designated by the FHWA, and use of the program by FHWA staff has been demonstrated.

3) The T3DRS:V1 program is capable of validly predicting braking and directional response behavior of trucks, tractor-semitrailers, doubles and triples.

4) The program is a suitable tool for studying the effects of truck size and weight through its capability to characterize performance changes in any selected maneuver with variations in size and weight.

6.0 RECOMMENDATIONS

In order to fully utilize the products of this project, the HSRI recommends that:

1) The FHWA regularly use and maintain the computer simulation program at their facilities. The proper application of the program requires personnel with knowledge and experience in heavy vehicle simulation. Regular use will develop those skills. Failure to use a program usually results in its eventual relegation to a nonfunctional status.

2) The FHWA use the program in a systematic study of the effects of increased truck size and weight. The investigation should be directed toward identifying the dynamic performance changes associated with different size and weight limits as applied to vehicles of alternative design configurations. Specifically, the investigation can be used to identify vehicle design factors (such as tire or brake size) that should be upgraded as a condition for allowing increases in truck size or weight.

7.0 REFERENCES

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APPENDIX A

AN ANALYSIS OF HEAVY TRUCK CORNERING AND BRAKING PERFORMANCE MEASURES

Summary Report
for the FHWA Project

"Simulation of the Effects of Increased Truck Size and Weight"

Contract No. DOT-FH-11-9330

Highway Safety Research Institute
The University of Michigan
Ann Arbor, Michigan 48109

31 December 1977

INTRODUCTION

Proposals are frequently made to allow increases in the weight and size limits for trucks and tractor-trailer combinations operating on the national highway system. A rational evaluation of these proposals requires measures by which to weigh the relative advantages and disadvantages of such changes. The advantages, which may be largely economic, are advocated by the trucking industry itself. The disadvantages are mostly centered about concerns relating to compromises in the safety of highway travel, and to potential for accelerated deterioration of the highway pavement and structures.

The overall objective of this project, "Simulation of the Effects of Increased Truck Size and Weight," is to provide the Federal Highway Administration with an operational version of the MVMA/HSRI Directional Response Simulations for trucks and tractor-trailers, as analytical tools for study of the influences of size and weight on safety and highway structural loadings. In order to evaluate those influences, a methodology for use of the simulation programs is required. The methodology must consist of three elements:

1) Test maneuvers covering the range of highway maneuvers

2) Performance measures to quantify the vehicle behavior in the maneuver

3) Criteria by which to interpret the performance measures.

This report summarizes the methodology developed in other major research programs in which performance measures applicable either to actual test vehicles or computer simulation vehicles were required. Additionally, a proposed set of performance measures, appropriate in the context of this project, for evaluating the effects of truck size and weight are developed.

LITERATURE SURVEY

The frequency of accidents, injuries, and fatalities are the common yardsticks of highway safety. Therefore, it might be anticipated that the correlation of accident statistics with vehicle types and performance characteristics has been addressed in the literature. A recent publication, "A Methodology for Determining the Role of Vehicle Handling in Accident Causation," [1] provides an excellent summary of the state of knowledge with regard to that problem in the passenger car field. In general, that study concludes that specific relationships between vehicle handling characteristics and accident causation are difficult to establish because of factors such as the following:

1) Driver habits are thought to correlate with vehicle types such that accident statistics may reflect driver as well as vehicle characteristics.

2) It is difficult to establish at what level of performance to judge handling behavior deficient in contrast to roadway or driver deficiencies.

3) Studies by even the most competent researchers often lead to contradictory results and frequently reflect the search for only the one accident causative factor of interest to the researcher.

Although a few trends are noted in that report, their applicability to truck handling appears to be inappropriate and insupportable.

(It should be noted that the term "handling" is used here although it has no universally accepted meaning. In the broadest sense, "handling" encompasses the lateral and longitudinal behavior of the driver/vehicle/roadway combination. The interest in this project includes the same broad areas limited only in that the driver variable in vehicle "handling" is not directly treated; rather, specific vehicle maneuvers will be considered in which the driving task will be measured by the control inputs required to accomplish the maneuver.)

Despite the uncertain link between vehicle performance characteristics and accident causation, the subject of vehicle performance merits and measures is being actively investigated by the automobile manufacturers, automotive research organizations, and the Federal government. These investigations, centering around the NHTSA procurements for vehicle handling test procedures, the Experimental Safety Vehicle Program, and automotive industry research staff, have resulted in numerous publications [2-14].

The vehicle performance measures can take many forms. In the analytical sense the performance can be quantified by its dynamic properties—gains, natural frequency, damping ratio, etc. Alternatively, it can be quantified by measures of any of the large number of motion variables such as translational and rotational displacements and derivatives, or derived quantities such as understeer gradient. And at the extreme, limit conditions such as critical velocity, rollover threshold, and spinout threshold can be used.

The literature on vehicle handling performance has been reviewed with the objective of cataloging the various measures of performance that have been used or proposed. Tables 1-12 summarize the measures used in major publications selected to represent a cross-section of the organizations involved in vehicle handling research. The tables list the maneuver performed, the performance measure applied, and, when available, the criteria against which the performance measure is judged.

Table 1 lists selected performance measures appropriate to this project developed in the Research Safety Vehicle Program and its predecessor, the Experimental Safety Vehicle Program. The RSV Program represents the most definitive effort to develop passenger car handling criteria of any program, although the application to commercial vehicles would, in most cases, be inappropriate.

Tables 2-5 represent summaries of the latest published truck and bus handling performance analyses by the major independent research organizations in the United States, including:

Table 1. Selected RSV Handling Requirements [2].

Maneuver	Performance Measure	Criteria
Service Braking	Stopping Distance	190' @ 60 mph in a straight 12' lane*
		90' @ 40 mph in a 357' radius 12' lane*
		(*Max. steering inputs: 180° & 500°/sec.)
	Pedal Force	Graphical Limits
Parking Brake	Grade Holding	30% Grade
Steady-State Yaw Response •0.4 g's at 25, 50, & 70 mph	Steer Angle	Graphical Limits (Approx. equal to 0 to 6° understeer)
Transient Yaw Response •0.4 g's at 25 and 70 mph	Yaw Response Time	Graphical Limits
Cornering	Steady-State Lateral g's	0.6 g's Minimum
Control at Braking	Recovery Time	Recover Original Cornering Path Within 4 sec. After Braking
Crosswind Sensitivity 30, 50, and 70 mph	Course Deviation	Graphical Limits
Steering Control Sensitivity •2°/sec yaw rate at 30, 50, and 70 mph	Steering Input Torque	Must Exceed 5 in-lb
Pavement Roughness Sensitivity	Course Deviation over Oblique Bump	Max. of 1' Deviation 2 sec. After Contact
Slalom Course •1000' at 100' pylon intervals, max. run speed	Overturning Immunity	Freedom from Rollover
Drastic Steer & Brake •50 and 60 mph	Overturning Immunity	Freedom from Rollover

Table 2. Systems Technology Inc. - Truck and Bus Handling Tests [3].

Maneuver	Performance Measure	Criteria
Step Steer •Wet & dry, 30 mph •Empty & full load •Variable steer angle	Peak Heading Rate Peak Lateral Accel. Peak Sprung Mass Roll Angle	Comparative Limits at Plow-Out, Spin-Out, or Rollover
Trapezoidal Steer •Wet & dry, 30 & 50 mph •Empty & full load •Variable steer angle	Peak Heading Rate Peak Lateral Accel. Peak Sprung Mass Roll Angle	Comparative Limits at Plow-Out, Spin-Out, or Rollover
Sinusoidal Steer •Wet & dry, 30 mph •Variable steer angle and frequency	Ratio of Yaw Rate Peaks	Comparative Performance
Braking In A Turn •Wet, 30 & 50 mph •All & rear only brakes •Partial & full braking •Empty, partial & full load	Peak Heading Rate Peak Lateral Accel.	Comparative Limits at Plow-Out, Spin-Out, or Rollover

Table 3. Bendix Research Labs - Bus Stability Requirements [4].

Maneuver	Performance Measure	Criteria
Ramp Steer •40 & 60 mph •Empty, loaded & asymmetric load •Variable steer angle	Peak Sideslip Angle (Hazard Zone) Peak Lateral Acceleration Peak Yaw Rate Peak Roll Angle	Comparative Limits at Spin-Out and/or Rollover
Braking In A Turn •40 & 60 mph •Empty, loaded & asymmetric load •Variable steer angle	Peak Sideslip Angle Peak Lateral Acceleration Peak Yaw Rate Peak Roll Angle Peak Deceleration	Comparative Limits at Spin-out and/or Rollover
Sinusoidal Steer •40 & 60 mph •Empty, loaded & asymmetric load •Variable steer amplitude	Final Heading Angle Lateral Position Change Cross Road Vehicle Projection	Comparative Performance
Sinusoidal Steer with Braking •40 & 60 mph •Empty, loaded & asymmetric load •Variable steer amplitude	Final Heading Angle Lateral Position Change Cross Road Vehicle Projection	Comparative Performance

Table 4. HSRI Truck and Bus Handling Tests for the Effects of Tire Properties [5]

Maneuver	Performance Measure	Criteria
Straight-Line Braking •Empty & loaded •40 mph •Variable braking levels	Average Deceleration Wheel Lock-Up Occurrence Peak Lateral Acceleration	Comparative Performance
Trapezoidal Steer •30 & 50 mph •Variable steer angle	Peak Lateral Acceleration Peak Yaw Rate	Comparative Limits of Plow-Out, Spin-Out, or Rollover
Braking In A Turn •Wet, 30 mph, 0.2 g's lateral •Dry, 50 mph, .35 g's lateral •Variable braking levels	Average Deceleration Peak Lateral Acceleration Peak Yaw Rate Wheel Lock-Up Occurrence	Comparative Performance
Sinusoidal Steer •Wet, 30 mph •Dry, 30 & 50 mph •Variable steer angle and frequency	Peak Lateral Acceleration Peak Yaw Rate	Comparative Performance

Table 5. IITRI - Articulated Vehicle Handling and Stability Tests [6].

Maneuver	Performance Measure	Criteria
Straight-Line Braking •20, 35 & 50 mph •Variable tire-road friction •With locked wheels & with antilock	Stopping Distance	Comparative Performance
Perturbed Straight-Ahead Driving •0.2 ft. sudden lane change at 35 mph •Variable tire-road friction •Unloaded vehicles only	Trailing Unit Overshoot Trailing Unit Settling Time	Comparative Performance
Lane Change •12 x 160 ft. •Similar to sinusoidal steer •Variable tire-road friction •Loaded and empty •Without braking, with locked wheel braking & with antilock	Maximum Velocity without Jackknife, Trailer Swing, or Rollover Maximum Hitch Point Forces at Maximum Velocity	Comparative Performance

Systems Technology, Incorporated

Bendix Research Laboratories

Highway Safety Research Institute/The University of Michigan

Illinois Institute of Technology

Table 6 summarizes commercial vehicle handling tests developed by the National Swedish Road and Traffic Research Institute and includes specific criteria by which to judge the performance.

Tables 7-12 summarize handling performance measures reflecting the most recent publications from other major organizations concerned with vehicle handling. Since no corporate handling standards are published by the major automotive manufacturers, representative publications from individuals in the Ford Motor Company and the General Motors Corporation were selected for Tables 9 and 10.

TRUCK PERFORMANCE MEASURES

The listed references and others were reviewed as background for identification of a number of performance measures by which to evaluate the effects of increased truck size and weight. Increased size and weight affects performance in a broad spectrum of ways. The performance measures considered here are only intended to encompass the dynamics of cornering and braking and specifically exclude other aspects such as the following:

—Gradeability and traffic compatibility

—Maneuverability (low speed turning and off tracking)

—Emergency braking and grade holding

—Accident damage

—Durability

Of the many types of maneuvers and performance measures presented in the literature, many are similar in nature, differing only in name or by virtue of being open- or closed-loop. The review was conducted with the intent to select performance measures by the following criteria:

Table 6. Swedish Commercial Vehicle Handling Tests [7].

Maneuver	Performance Measure	Criteria
Double Lane Change .70 km/h	Lateral Acceleration	
	Sideslip Angles	Less than 8.6 Peak
	Overturning Risk	Less than Unity
	Lateral Deviations of Axles	Graphical Limits
	Rearward Risk Factor Amplification	Maximum of 2.0
Rollover Threshold (static test)	Lateral Acceleration at Rollover	.4 m/sec^2 Minimum
Low Speed Off Tracking	Deviations of Vehicle Extremities	Must track between two concentric circles 7.3 and 15 meters radius
High Speed Off Tracking •Steady-state cornering at 70 km/h, 2 m/sec^2 lateral acceleration	Deviation of Vehicle Extremities	Off tracking to outside of curve must not exceed 0.5 meters

Table 7. HSRI - Vehicle Handling Performance [8].

Maneuver	Performance Measure	Criteria
Straight-Line Braking •40 mph •Variable braking	Maximum Deceleration (wheels unlocked, average 35 to 10 mph)	Comparative Performance
Braking In A Turn •40 mph, 0.3 g's •Variable braking	Maximum Deceleration Peak Sideslip Rate Average Path Curvature	Comparative Limits at Spin-Out and Plow-Out
Roadholding In A Turn •30 mph, 0.4 g's •Variable Roughness Frequency	Peak Sideslip Rate Average Path Curvature	Comparative Performance
Trapezoidal Steer •40 mph •Variable Steer Angle	Average Path Curvature Peak Sideslip Angle	Comparative Limits at Plow-Out, Spin-Out, or Rollover
Sinusoidal Steer •45 and 60 mph •Variable Steer Amplitude	Lane Change Deviation Peak Sideslip Angle	Comparative Performance
Drastic Steer and Brake •50 and 60 mph •Variable steer angle •Variable braking intervals	Peak Roll Angle	Comparative Limits of Rollover

Table 8. Systems Technology Inc. - Automobile Controllability Requirements [9].

Maneuver	Performance Measure	Criteria
Normal Driving	Steady-State Yaw Velocity Gain at 50 mph	.2 to .4 deg/sec per Degree Steering Wheel
	Yaw Velocity Time Constant	Less Than 0.3 sec.

Table 9. Ford Motor Co. - Vehicle Handling Parameters for "Ease of Vehicle Control" [10].

Maneuver	Performance Measure	Criteria
Reverse Steer Ramp ·60 mph	Lateral Acceleration Gain at .67 g's	.007 g/deg (Ideal)
	Sideslip Response Time at .67 g's	.15 sec. (Ideal)

Table 10. General Motors Corp. - Vehicle Handling Measures Relating to Tire Intermix [11, 12].

Maneuver	Performance Measure	Criteria
Normal Driving 0-.3 g's Lateral accel.	Lateral Acceleration Response Time	Comparative Performance
	Lateral Acceleration Gain	Comparative Performance

Table 11. CALSPAN - Selected Vehicle Handling Measures [13].

Maneuver	Performance Measure	Criteria
Closed-Loop Avoidance •Severe level •Narrow lane	Maximum Speed	Comparative Performance
Cornering on a Bumpy Surface	Maximum Lateral Acceleration Amount of Steering Used Number of Steering Reversals	Comparative Performance
Straight-Line Braking •Smooth & irregular surfaces	Stopping Distance	Comparative Performance
Braking In A Turn •Smooth & irregular surfaces	Stopping Distance	Comparative Performance

Table 12. Dynamic Science - Handling Test Procedures for Light Trucks, Vans, and Recreational Vehicles [14].

Maneuver	Performance Measure	Criteria
Braking In A Turn •40 mph, 0.25 g's lateral acceleration	Longitudinal Acceleration Pedal Force Peak Sideslip Angular Rate Average Path Curvature Ratio	Comparative Performance
Trapezoidal Steer •30 mph •Variable steer angles	Peak Sideslip Angle Peak Roll Angle Normalized Path Curvature	Comparative Performance
Sinusoidal Steer •45 mph •Fixed steer amplitude •Variable period	Lane Change Deviation Peak Sideslip Angle Heading Angle Deviation	Comparative Performance
Rough Road Cornering •40 mph, 0.35 g's lateral acceleration •Variable roughness frequency	Peak Sideslip Angle Average Path Curvature Ratio	Comparative Performance
Trapezoidal Steer While Braking •0.4 g's longitudinal accel. •Steer input at 40 mph	Peak Sideslip Angle Peak Roll Angle Normalized Path Curvature	Comparative Performance
Crosswind Sensitivity •30 mph •Fixed steer angle	Lateral Deviation at 20 and 50 Feet After Leaving the Wind Disturbance	Comparative Performance

1) Sensitivity to truck size and weight

2) Meaningful interpretation

3) Representative of the range of typical highway maneuvers

4) Appropriate for evaluation by computer simulation or vehicle test.

In the selection process, many types of maneuvers and test conditions were eliminated for being inappropriate or dangerous with heavy trucks or for having no relevance to the size and weight issues.

Table 13 is a summary of the selected performance measures arranged by type of maneuver. Specific criteria are presented as available and appropriate; no attempt has been made to develop new criteria within the context of this project.

In the sections that follow, the items in Table 13 are discussed, presenting the rationale for their selection. The selection draws on the references cited previously as well as current concerns within the research community with respect to critical truck performance measures; and has the objective of encompassing the necessary and sufficient maneuvers to discriminate the influences on cornering and braking of truck size and weight.

BRAKING

Two aspects of braking performance are of interest with respect to accident avoidance—stopping distance and stability. With few exceptions, short stopping distance has been accepted as a desirable attribute of vehicle performance. The Federal Motor Vehicle Safety Standard No. 121 provides an accepted criteria for judgment of straight-line braking ability. Test conditions representing the extremes of performance are listed.

Stability in braking is not well defined in the straight-line braking situation. In computer simulation the instability may not

Table 13. Proposed Truck and Articulated Vehicle Performance Tests for Effects of Increased Size and Weight.

Maneuver	Performance Measure	Criteria
Straight-Line Braking Distance •Dry surface, 60 mph •Wet surface, 20 mph •Empty and loaded •Full braking with antilock-equipped vehicles •Max. braking without axle lockup on non-antilock vehicles	Stopping Distance Peak Wheel Loads	FMVSS 121 Requirements
Braking Stability with Fixed Steering •Same conditions as above •Pavement cross slope •Fixed steer angle - 0°	Sideslip Angles Lateral Acceleration Articulation Angles* Yaw Rates Heading Rate Path Deviation Roll Angles	Comparative Performance of Different Vehicles
Braking In A Turn •0.2 g's at 30 mph •Dry surface •Constant radius, 12' lane •Variable braking levels •Loaded	Maximum Deceleration Peak Articulation Angles* Path Deviations of All Axles Peak Lateral Accelerations Peak Yaw Rate Peak Sideslip Peak Roll Angle Peak Wheel Loads	Comparative Limits at Jackknife, Spin-Out, Plow-Out, or Rollover

*Articulated Vehicles Only

Table 13 (Cont.)

Maneuver	Performance Measure	Criteria
Trapezoidal Steer •30 & 50 mph •Variable steer angle •Dry surface •Loaded	Steady-State Path Curvature Peak Lateral Accelerations Peak Yaw Rates Peak Articulation Angles* Peak Sideslip Angles Peak Roll Angles Lateral Acceleration Gain Lateral Acceleration Response Time Sideslip Response Times Peak Wheel Loads	Comparative Limits at Jackknife, Spin-Out, Plow-Out, or Rollover
Lane Change •12 x 160' path •Variable velocity •Dry surface •Loaded	Entrance Velocity Peak Yaw Rates Peak Lateral Accelerations Peak Articulation Angles* Peak Sideslip Peak Roll Angles Peak Wheel Loads	Comparative Limits at Jackknife, Spin-Out, Plow-Out or Rollover
Ramp Steer •50 mph •Varying steer angle •Dry surface •Loaded	Maximum Lateral Acceleration Peak Sideslip Angles Peak Yaw Rates Peak Roll Angle Peak Articulation Angles*	Comparative Limits at Jackknife, Spin-Out, Plow-Out, or Rollover

*Articulated Vehicles Only

be excited, and in real world testing, driver skill influences the measure of stability obtained. Comparative measures of braking stability are best obtained in the situation in which no steering corrections are made. Thus a test of braking stability with fixed steering is included. In vehicle tests, disturbance factors such as brake imbalance, surface roughness, surface frictional variation, wind gusts, etc., will excite any vehicle instability in a random fashion and with an unknown amplitude. Since a computer simulation must also be given a disturbance, pavement cross-slope is suggested as a common, easily determined input which is independent of the specifics of vehicle design.

BRAKING IN A TURN

Braking in a turn provides a measure of the vehicle's combined braking and cornering capability. Among the various ways in which this maneuver can be performed, the closed-loop, constant path curvature method is most typical of highway maneuvers and most easily interpreted. The maneuver is performed with the vehicle coasting from a higher speed on a 300' radius, eaching 0.2 g's lateral acceleration at 30 mph. Brakes are applied at fixed values of increasing level until limit conditions are reached. In general, the best measure of performance is the maximum deceleration that can be achieved without loss of control allowing the vehicle to reach a limit condition such as jackknife, spin-out, plow-out, or rollover.

TRAPEZOIDAL STEER

Trapezoidal steer is the open-loop equivalent of a J-turn and provides a measure of both steady-state and transient performance of the vehicle. In this maneuver a high rate steer input (effectively, a step steer) is applied to various magnitudes up to the limits of vehicle capability. The maneuver excites the vehicle dynamic modes such as roll and the rearward amplified yaw responses of articulated combinations, to determine the limits of performance in a situation similar to an emergency avoidance maneuver by the driver. The maximum steady-state path curvature which can be achieved is the measure of best performance. The measures of the peak values for the various parameters indicate the limit achieved and the nature of the limit on the maneuver (i.e., jackknife, rollover, etc.). The measures of gain and response times are indicative of the control responsiveness of the vehicle in cornering performance.

LANE CHANGE

The single lane change is the closed-loop equivalent of a sinusoidal steer maneuver and provides a measure of response in a bidirectional transient maneuver. Using the closed-loop test, in which the vehicle follows a prescribed path representative of a typical highway maneuver, vehicle performance can be easily judged by the maximum entrance velocity at which the maneuver can be successfully accomplished. The measures of peak parameters indicates the mode in which the limiting condition occurs.

RAMP STEER

A slow ramp steer effectively measures the maximum cornering performance that can be achieved when the steer rate is low enough that near steady-state conditions exist. The primary measure of performance in this maneuver is the maximum lateral acceleration level that can be achieved at limit conditions; the peak values of other parameters being used to indicate the nature of the limiting conditions.

CONCLUSION

The performance measures described above, in most cases, can only be judged by a criteria of comparative performance between similar vehicles because of the absence of recognized performance levels for trucks. For purposes of evaluating truck size and weight effects, however, this method is appropriate and is commonly used.

It should be noted that in the proposed performance tests no rough road maneuvers are considered. Such maneuvers have been excluded due to the large number of variables that would influence the result and complicate any effort to evaluate performance. Specifically, it is a concern that many vehicle design variables (wheelbase, tandem axle spread, suspension type, suspension damping, etc.) would potentially influence rough road performance even more than the variables of road surface, speed, and vehicle weight.

The evaluation of dynamic wheel loads is included as a performance measure in four of the test maneuvers; those maneuvers representing the primary highway situations (excepting bumps) in which high wheel loads would occur. No separate tests are suggested because of the lack of a precedent, and the concern that the typical vehicle imperfections contributing substantially to such effects are not well known [15].

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APPENDIX B

"REPORT ON CAPABILITIES AND VALIDATION OF THE
MVMA-HSRI TRUCK DIRECTIONAL RESPONSE COMPUTER SIMULATIONS"

Summary Report
for the FHWA Project
"Simulation of the Effects of Increased
Truck Size and Weight"

Contract Number DOT-FH-11-9330

Highway Safety Research Institute
The University of Michigan
Ann Arbor, Michigan 48109

31 January 1978

1.0 INTRODUCTION

This document is submitted to the Federal Highway Administration (FHWA) as a summary of the capabilities and validation of the MVMA-HSRI Truck Directional Response Computer Simulation required in Task B of the project, "Simulation of the Effects of Increased Truck Size and Weight." The capabilities of the simulation program as described here are intended to reflect the content of the program as it will be provided to the FHWA in Task G of the project. Because of the continuing research on truck performance sponsored at HSRI by the Motor Vehicle Manufacturers Association (MVMA) and other organizations, simulations are periodically changed and improved to reflect the state-of-the-art knowledge of the best way to characterize braking and handling performance. The program to be provided to FHWA will be the latest update of the Directional Response Simulation. It may be noted that, as with all other users of the MVMA-HSRI programs, future expansions and improvements to the program will be routinely available to the FHWA.

The current MVMA-HSRI truck simulation capability is represented by two separate digital computer programs—one specializing in straight-line braking and a second, more general program, capable of simulating both braking and handling. The straight-line braking program was developed in 1972 as the Phase I program [1], and reflected the growing interest and concern by the MVMA in the braking problems of heavy trucks and tractor-trailers as they related to FMVSS 121. Since then, that original braking program has undergone significant revision, and in its current form is known as the Phase III program [3].

The current MVMA-HSRI braking and handling model, known as the Phase II Truck Directional Response Simulation [2], was developed in 1973 and represents the <u>general</u> model to be extended and used during the course of the FHWA project. Except for a few detailed model differences, its capability in representing the braking performance of trucks and tractor-trailers during straight-line braking duplicates its Phase III counterpart. Those few exceptions will be upgraded as part of the modifications to be made to the FHWA Phase II program during the course of this project.

The principal modifications to the MVMA-HSRI Phase II program will pertain to extending the current tractor-trailer model to allow for the simulation of doubles and triples. The revised model will also provide for an optional closed-loop steering control to permit the vehicle to follow a specified path.

In the sections that follow, the general features of the current Phase II model and the proposed modifications/extensions are summarized. Validation tests and results for the Phase II program are discussed in the final section.

2.0 MVMA-HSRI PHASE II SUMMARY

Current capabilities of the MVMA-HSRI Phase II simulations are discussed within this section. Planned modifications/additions to the program are outlined in Section 3.0.

2.1 Degrees of Freedom and Axis System

The current version of the Phase II program can represent up to 32 degrees of freedom (tractor-trailer configuration) distributed as follows: (1) six degrees of freedom for the tractor's sprung mass, (2) six degrees of freedom for the semitrailer's sprung mass, (3) a rotational degree of freedom for each of the ten wheels (the wheels may have dual tires), (4) vertical and roll degrees of freedom for each single axle, and (5) four degrees of freedom to describe the vertical and roll motions of a tandem axle pair.

For each unit (tractor/trailer) of the vehicle a moving axis system is associated with (1) the sprung mass (body axis system) and (2) the unsprung masses (unsprung mass axis system). Each of the body axis systems is related to the fixed or inertial coordinate axis system in orientation by conventional Euler angles (heading about the vertical axis of the inertial system, pitch about the lateral body axis, and roll about the longitudinal body axis). Each unsprung mass axis system translates with the sprung mass center but is only allowed to rotate in yaw. Each unsprung mass axis orientation is likewise related to the inertial coordinate system by a yaw angle (ψ) transformation. Figure 2.1 shows the relationship between each of the coordinate systems for the tractor-trailer.

2.2 Solution Method for the Equations of Motion

The equations of motion for each sprung mass provide the translational and rotational accelerations which are then integrated to obtain velocity and position. Each of the aforementioned coordinate

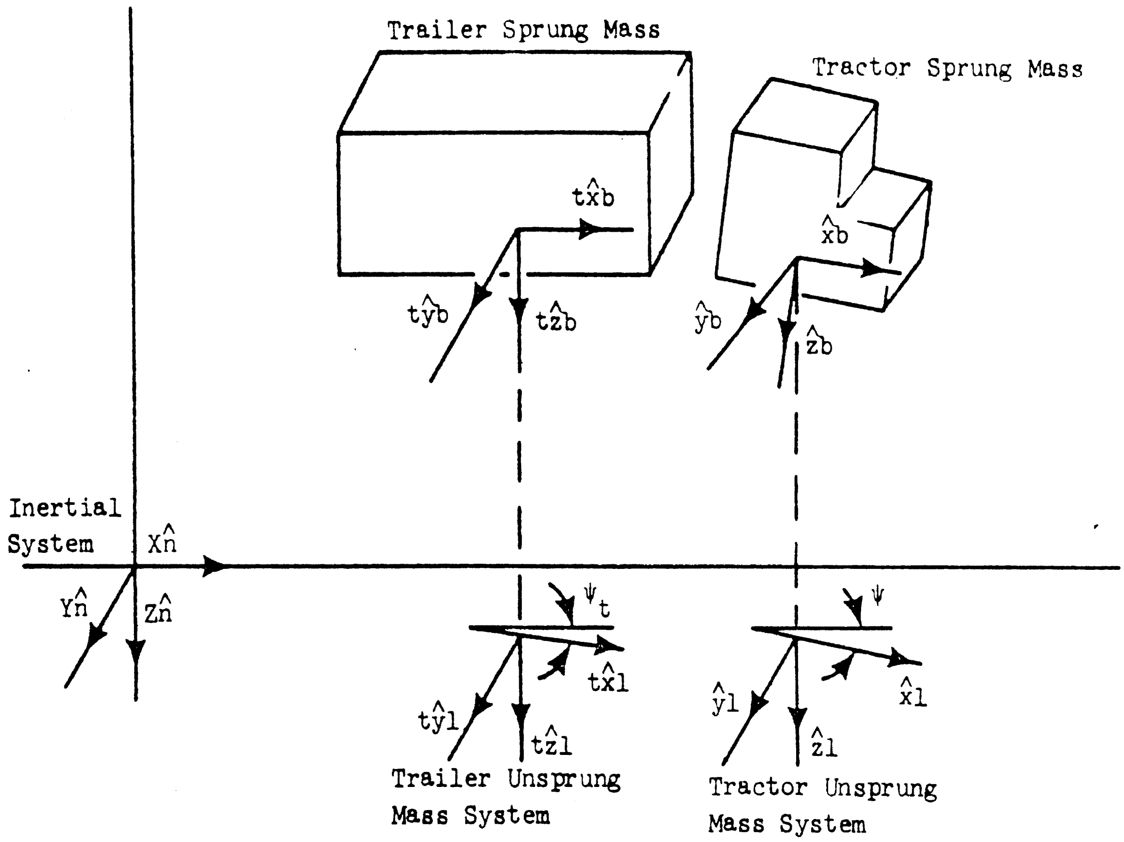

Figure 2.1. Coordinate system relationships.

Table 2.1. Summary of the Coordinate Systems Shown in Figure 2.1.

Coordinate Systems		
Name	Notation	Use
Inertial	XN, YN, ZN	Location of the vehicle. Observation point for accelerations and velocities.
Body, Tractor or Straight Truck	XB, YB, ZB	Convenient for calculation of rotational equations of sprung mass.
Semitrailer	TXB, TYB, TZB	
Unsprung Mass Tractor or Straight Truck	X1, Y1, Z1	Convenient for calculation of shear forces at the tire/road interface.
Semitrailer	TX1, TY1, TZ1	

systems is used to conveniently represent external forces (tire, suspension forces) within the equations of motion for each sprung mass. The unsprung masses are constrained to move with their associated sprung mass by appropriate constraint equations solved to provide the necessary constraint forces. Fifth wheel hitch forces are likewise included as external forces within the equations of motion but do not derive from conventional constraint equations but rather from a spring-damper restraint concept outlined in the next section.

2.3 Fifth-Wheel Hitch

The fifth-wheel hitch representation currently employed in the Phase II program consists of a linear spring/viscous damper coupling between the tractor and semitrailer, as shown in Figure 2.2. Initially, the fifth-wheel position of the tractor and the semitrailer are assumed to be identical. As the simulation run proceeds, forces developed by the tires will cause disparate paths for the fifth-wheel position of the tractor and the semitrailer; a distance, δ, will

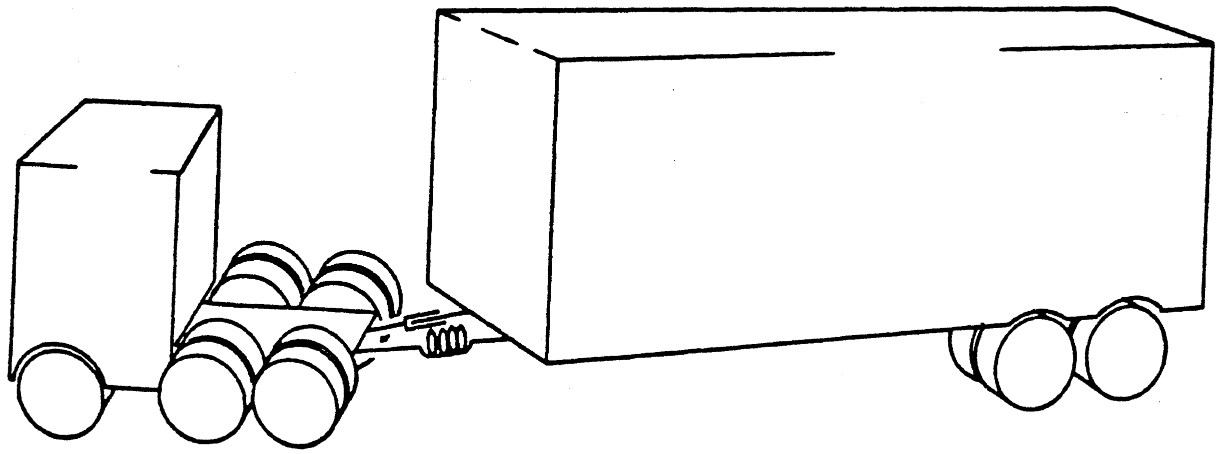

Figure 2.2. Fifth-wheel coupling model.

develop between them. The restraining hitch force developed by the spring/dashpot representations is proportional to δ and its rate of change. The direction of the hitch force is assumed to be along a line through the fifth-wheel location of the tractor and semitrailer. Numerical values for the spring stiffness and damping are selected to ensure that δ remains small and well damped.

A fifth-wheel roll moment retraint is similarly represented by a torsional spring between the tractor and semitrailer. The roll moment transmitted through the fifth wheel is assumed to be equal to the product of the torsional stiffness and difference in roll angles of the tractor and semitrailer.

2.4 Tire Model

The current Phase II program uses a semi-empirical model to generate combined longitudinal and lateral tire forces. The model uses parameters measured from tire data at two conditions: (1) longitudinal tire data at zero slip angle and (2) lateral tire data with no braking. Aligning torque is calculated based on tables of measured aligning torque versus steer angle and vertical load. Tire vertical deflection and normal load are based on a linear spring assumption at the tire/road interface. Figures 2.3 and 2.4 show an example tire model force calculation for various combined braking and sideslip conditions.

2.5 Wheel Rotational Dynamics

The Phase II program calculates rotational wheel slip by an efficient local linearization technique first introduced in the Phase I braking program [1]. Hence, the need and cost to integrate the conventional wheel rotational dynamic equations is eliminated. The wheel slip (or wheel speed equivalent) is used by the tire model/tables, brake fade, and antiskid models.

2.6 Suspension Models

An I-beam front axle model is used in the simulations. Any one of three different suspension options can be selected at the rear axle: (1) single-axle suspension, (2) four-spring suspension, or (3) walking-beam suspension. The solid single-axle suspension has both vertical deflection and roll degrees of freedom. The basic four-spring suspension model has four degrees of freedom (vertical deflection and pitch, each side) and is shown in Figure 2.5. The walking-beam suspension has the same four degrees of freedom and is shown in Figure 2.6. Both tandem axle models incorporate features for inter-axle load equalization and for describing inter-axle load transfer effects during braking.

2.7 Brake Models

Two options are currently available to represent brake torque in the Phase II program. First, any one of six brake modules can be represented: (1) S-cam with leading/trailing shoes, (2) dual

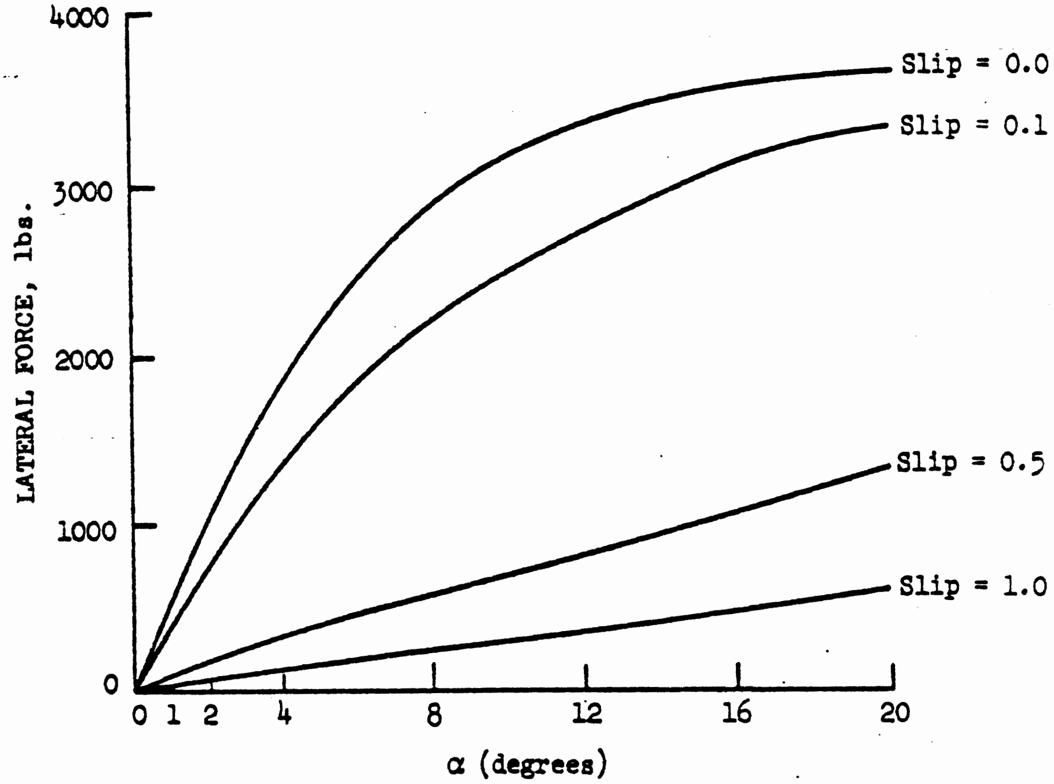

Figure 2.3. Cornering force vs. sideslip angle for various longitudinal slip values (HSRI semi-empirical tire model).

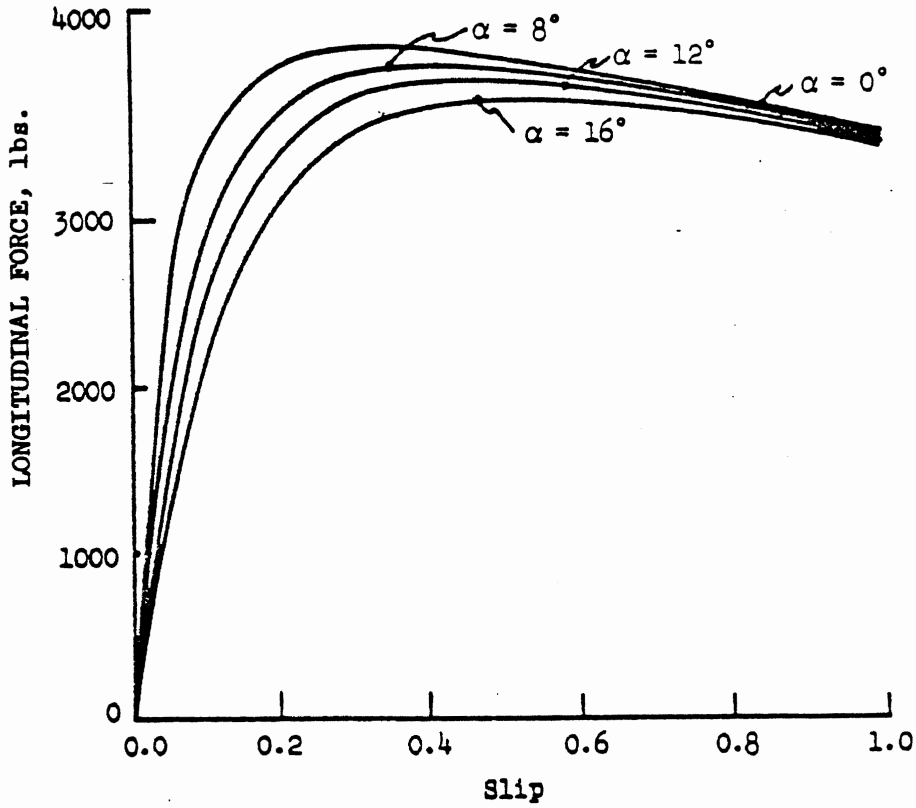

Figure 2.4. Brake force vs. longitudinal slip for various sideslip angles (HSRI semi-empirical tire model).

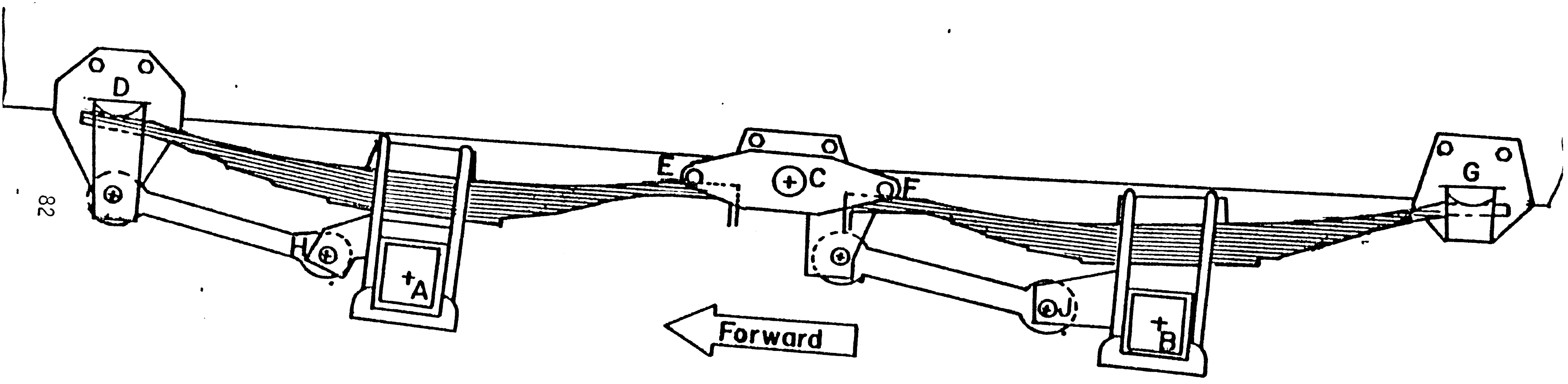

Figure 2.5. The basic four-spring suspension.

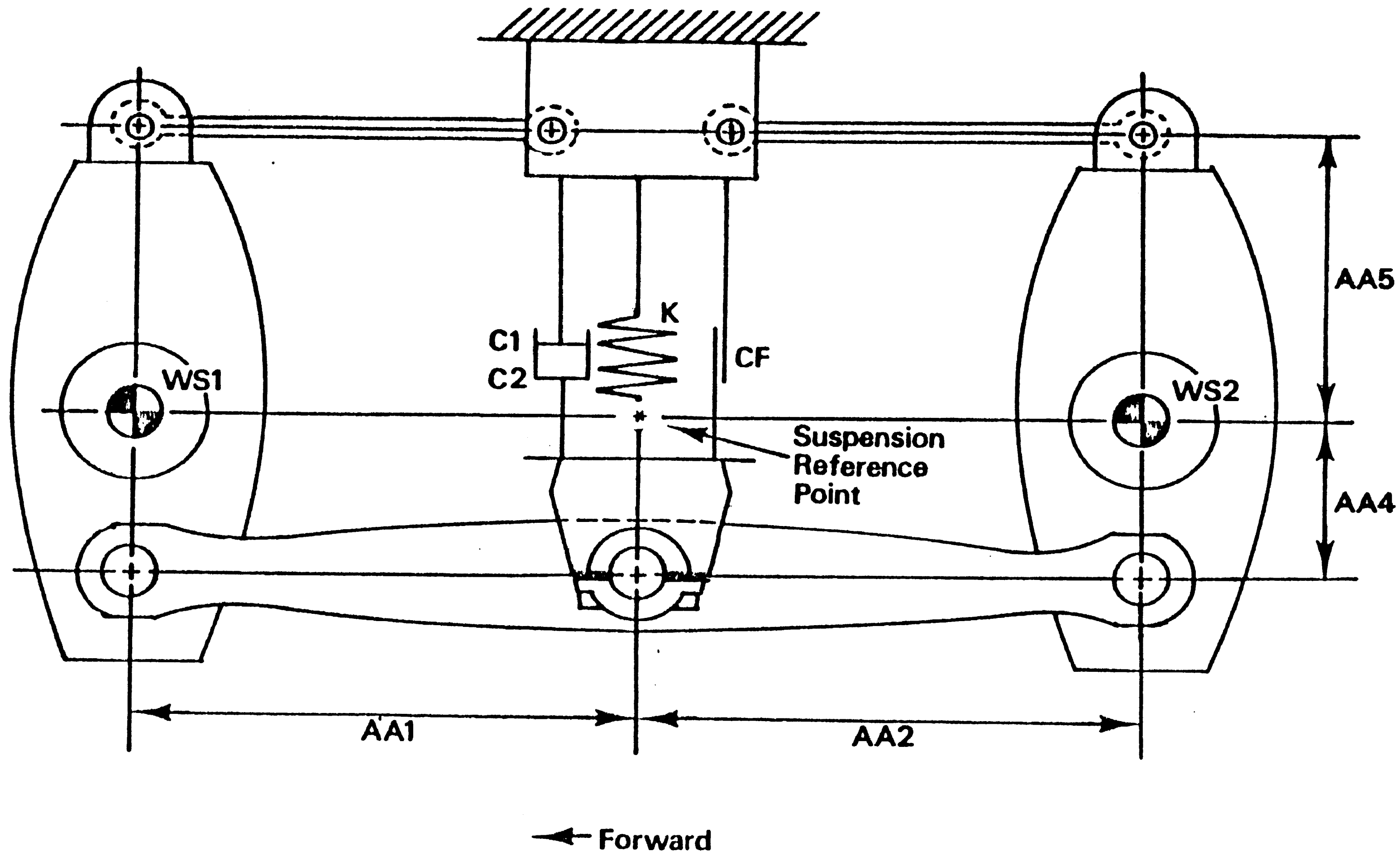

Figure 2.6. The walking-beam suspension model.

wedge with two leading shoes, (3) single wedge with leading/trailing shoes, (4) duo-servo self-actuating, (5) duplex, and (6) disc. Brake dimensions and lining friction coefficients are used as input by these modules to calculate brake torque. The second option is to use tabular data of brake torque versus chamber pressure usually obtained from brake dynamometer tests.

The air supply-brake system interaction is simulated by specifying transport line delays, exponential chamber pressure characteristics, and brake push-out pressures.

2.8 Antiskid Model

The Phase II program makes use of a general purpose antiskid model to represent most operational characteristics currently displayed by commercial antiskid systems. The model is quite flexible and provides the following basic features:

1) Separate detailed representations of an antiskid system's wheel speed sensor, computer logic, pneumatic logic, and air valve characteristics.

2) Allowance to specify different antiskid systems, axle-by-axle.

3) Side-to-side wheel select options covering (a) "worst wheel," (b) "best wheel," and (c) "average wheel" operation.

2.9 Steering System

Several different steering system options are available in the Phase II program and are listed below:

1) Single table steer input providing the same steer angle for both front wheels as a function of time.

2) Two table steer input providing separate steer angles at each front wheel.

3) Axle roll steer which combines either of the above options with a modifying influence of axle-frame roll.

4) Combined roll, pitch, and bounce steer similar in concept to (3), but more comprehensive by allowing pitch and bounce effects as well.

5) Steering system compliance model which permits steering compliance in addition to the features contained in the above options.

2.10 Inclined Roadway

An inclined planar roadway may be specified in the Phase II program for representing downgrade/upgrade, positive and negative superelevations, or any combination of the two.

2.11 Wind Loading

Aerodynamic forces may be simulated in the program by the provision for a user-written subroutine which calculates the wind forces and moments acting on the sprung masses. Hence, drag, lift, and side-loading may be represented to the degree of detail required by the user.

3.0 PHASE II-FHWA MODIFICATIONS

The principal modifications made to the Phase II program will permit the simulation of doubles and triples. While these specific modifications will largely take place during Task F, groundwork in the form of alterations of the basic program structure (indexing of sprung mass units, subroutining of external forces, etc.) is currently being laid to more easily accommodate the doubles/triples requirement. The dolly/pintle hook connection between units is planned as a massless, spring/damper restraint similar in concept to the current Phase II fifth-wheel hitch model.

In addition to the doubles/triples modifications, a closed-loop steering control option will be added which permits the vehicle to follow a user-input path. The steering control will incorporate preview or "look-ahead" strategy to steer along the desired path. The steering program will be flexible to allow different user-programmed strategies and steering response characteristics.

While Figures 2.3 and 2.4 of the preceding section adequately demonstrate the presumed nature of the interaction between combined longitudinal and lateral truck tire force generation, current HSRI plans are to improve the flexibility of the present tire force representation. A revised procedure for representing tire forces by tabular data is planned as part of the FHWA modifications under Task F. This revision will permit greater versatility in representing truck tire force characteristics and also allow improved format compatibility between the data used by the computer program and data commonly measured during tire tests. The revisions to the Phase II tire force calculation will require the use of tabular tire data. Presently, the tire model computes combined longitudinal-lateral tire forces based on longitudinal tire data at zero slip angle and lateral tire data with no braking. Under the planned revisions, the user will specify similar tables of tire data, but indicate the degree of interaction between longitudinal and lateral tire forces during combined braking and cornering by an additional "roll off" table. Combined braking and cornering tire data, if available, will also be able to be represented under this plan.

In recent years, there has been little use of the brake modules by most users of the Phase II program primarily because of the greater availability of brake dynamometer data. These modules were originally provided in the Phase I braking program to approximate torque-pressure relationships in the absence of suitable dynamometer data. At this time HSRI sees no compelling reason to carry along and maintain the brake module option in view of its current demand and usage status. Hence, the brake module option is not planned for inclusion in the FHWA and future MVMA versions of the Phase II program.

Finally, general improvements recently made to the Phase III braking program (extended antiskid model, improved suspension force measurement representations, brake fade, etc.), but not currently available in the Phase II program will be added to the final FHWA version.

4.0 MVMA-HSRI PHASE II VALIDATION

The Phase II program has been used by several vehicle manufacturers as well as HSRI over the past several years. The Ford Heavy Truck Division used the Phase II program to gain insight into the interaction between braking and steering of the front wheels on heavy trucks [4]. The program has also been used by HSRI in a recent DOT-sponsored study [5] to examine the mechanisms causing yaw divergence and subsequent rollover of a commercial vehicle during testing. While such use represents an implicit form of trust and certification of the results predicted by the Phase II program by experience, it does not constitute a formal validation as was undertaken during the course of the Phase II program development. The following sections serve to summarize the validation study performed during the Phase II program development. Chapters 6 and 7 of Reference [2] should provide any necessary additional details to this summary.

4.1 Test Vehicles

A straight truck and a tractor-trailer combination were run in a series of tests in both loaded and empty configurations. The tractor alone was also tested in its bobtail configuration.

The straight truck was a 4 x 6, 50,000-lb GVW, Diamond Reo vehicle with a 190-inch wheelbase and dump-type body equipped with a walking-beam suspension. The vehicle was in new condition with OE tires at the time of testing.

The tractor was a 4 x 6, 46,000-lb GVW, White vehicle with a 142-inch wheelbase equipped with a four-spring/load leveler suspension. The trailer was a 40-ft Fruehauf van-type equipped with a four-spring/load leveler suspension.

Tables 4.1-4.5 describe more completely the test vehicles and their loading conditions.

Table 4.1. Vehicle Specifications, Straight Truck.

General	4x6, 50,000 lb gvw, straight truck, 190 in. wheelbase	
Engine	V8-210	
Transmission	5 speed forward, 1 reverse with 4 speed auxiliary spicer	
Rear Axles	34,000 rated load with 7.8 ratio	
Steering Gear	19:24:19, hydraulic power	
Wheels	cast spoke	
Brakes	Front—dual chamber wedge type	Rear—dual chamber wedge type
Air chamber	type 9	type 12
Wedge angle	12°	12°
Size	15 x 5	15 x 6
Linings	RM-MA-417A	ABB-693-551-D
Lining area	314 sq in.	752 sq in.
Parking-emerg.	---	single swedge, spring actuated, 4 rear wheels
Axles	16,000 lb	34,000 lb
Suspension	leaf springs, 11 leaves, 7000 lb	rubber springs, RSA-340, 34,000 lb, aluminimum walking beam
Tires		
Size	highway tread, tubeless 15-22.5	highway tread, tube type 10.00-20
Load Range	H	F

Table 4.2. Loading Conditions for the Straight Truck.

Loading Condition	State Axle Loads front lb	rear lb	total lb
Empty	8,700	12,700	21,400
Loaded	13,000	32,200	45,200

Total Vehicle C.G. Position

Loading Condition	Inches Aft of Front Axle	Inches Above Ground
Empty	116	46
Loaded	137	55

Table 4.3. Vehicle Specifications, Tractor.

Model	4x6, 46,000 lb gvw, 142-in. wheelbase, COE (sleeper type)	
Engine	V-8, 335	
Transmission	5 speed forward, 1 reverse, 2 speed auxiliary spicer	
Rear Axle	34,000 with 4.11 ratio	
Steering Gear	28:1 constant ratio, lock to lock	
Wheels	Cast spoke	
Brakes	Front—dual chamber wedge type	Rear—dual chamber wedge type
Special equip.	limiting and quick release valve	relay valve and quick release valve
Air chamber	type 12	Type 12
Wedge angle	12°	12°
Size	15 x 4	15 x 7
Linings	RM-MR-417A	RM-MA-417A
Parking-emer.	---	single wedge, spring actuated, 4 rear wheels
Axles	12,000 lb	34,000 lb
Suspension	leaf spring	4 spring
Tires	highway tread, tube type	deep lug, tube type
Size	10.00-20	10.00-20
Load Range	F	F
Axle Weights		
Bobtail	8100 lb	6800 lb
Total Vehicle C.G.		
Position, Bobtail	67 inches aft of front axle 40 inches above ground level	

Table 4.4. Trailer Specifications.

Model	40 ft, van type, 2 axle, semitrailer
Suspension	4 spring (3 leaf)
Axles	34,000 lb
Brakes	S-cam, leading-trailing
Air chambers	type 30
Slack adjusters	6-inch length
Size	16-1/2 x 7
Linings	SAE friction code "EE"
Tires	highway tread, tube type
Size	10.00 x 20
Load range	F

Table 4.5. Loading Conditions for the Articulated Vehicle.

Loading Condition	Static Axle Load (lb)			
	Front	Rear	Trailer	Total
Empty	8,900	10,500	7,800	27,200
Loaded	10,500	32,000	31,800	74,300

Loading Condition	C.G. Position			
	Tractor		Trailer	
	Aft of front axle(in.)	Height (in.)	Aft of Kingpin(in.)	Height (in.)
Empty	67	40	265	56
Loaded	67	40	218	66

4.2 Vehicle Tests

The validation tests performed with the straight truck and tractor-trailer were fixed steer, steady-state turning and braking-in-a-turn maneuvers. The steady turning tests were conducted by ramping in a steer angle to a fixed level at constant speed, until a steady-state vehicle response was achieved. The tests were conducted at normal speeds of 25 and 30 mph using maximum steer angle levels corresponding to 25, 50, 75, and 100% of the maximum value considered safe for the particular load configuration.

The braking-in-a-turn tests were begun in the same manner as the steady-state turn test. However, once steady-state was achieved, a step brake application was made to a preset level determined by a limiter in the brake line. Tests were made from an initial velocity of 27 mph with steer angles and brake pressures selected to cover a broad range of lateral and longitudinal acceleration. These tests established performance limits above which one or more wheels locked.

Two special high-speed jackknife tests were also performed with the tractor-trailer combination.

4.3 Comparison of Simulation and Test Results

Steady turn data was taken for the straight truck and tractor-trailer in the empty and loaded conditions on the dry surface and in the empty condition on the wet surface. The bobtail tractor was tested in steady turns on the dry surface only.

Certain differences between the experimental procedure and the simulated procedure should be noted. The steady turn experimental results were taken at a steady speed; whatever drive torque necessary to maintain that speed was applied. In the simulation, on the other hand, no drive torque was applied. Thus the simulated vehicle speed drops during the course of the run as a result of the longitudinal component of the side force of the steered front wheels. Therefore, the initial condition of vehicle speed was chosen slightly higher than the speed for which the results were desired; the vehicle model would reach a quasi-steady turn condition in which it would gradually lose speed. When the speed dropped to the test speed, the simulated yaw rate and lateral acceleration predictions were noted.

It should also be noted that the measured steer angles were used in the simulation. These were, as one might expect, significantly different from side to side. (Since all the empirical results and simulation runs were left turns, the left steer angle was always larger than the right.) For the purposes of the following figures, average steer angles were plotted.

With very few exceptions, the measured results and the predicted results are in very close agreement. In all the steady turn figures, the simulated yaw rate and the simulated lateral acceleration may appear to be different only by a scale factor. This should be expected since, in the simulated "steady" turns

$$A_y \cong u \cdot \dot{\psi}$$

where

A_y is the lateral acceleration

u is the longitudinal velocity

$\dot{\psi}$ is the yaw rate

The yaw rate and the lateral acceleration were measured independently, however; thus, the empirical results conform to the above equation within the limits of accuracy of the instrumentation.

Figures 4.1 through 4.10 show the simulated and test results for the steady turn maneuver with each vehicle.

The measured results and the predicted results are in close agreement for the empty trailer runs, but in the case of the loaded vehicle, a marked difference is apparent between the experimental and

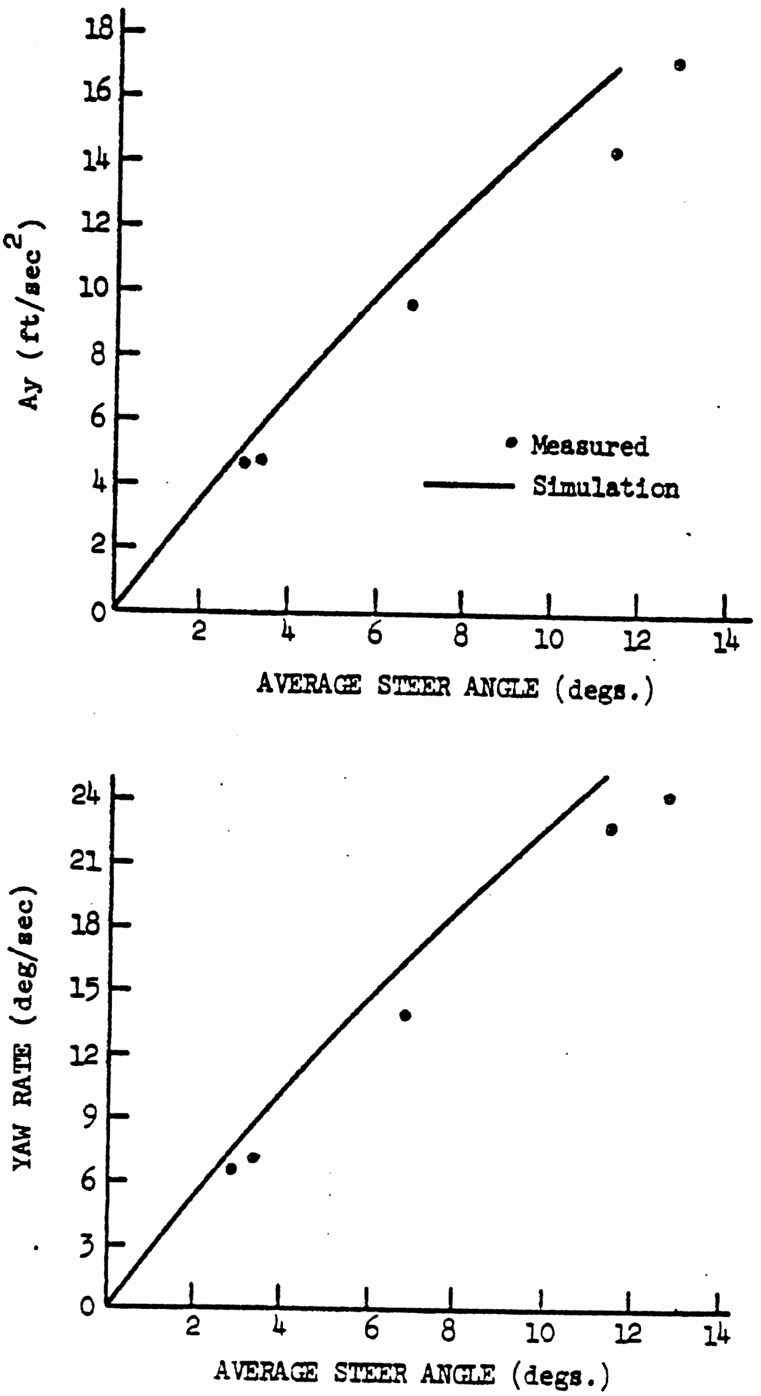

Figure 4.1. Steady turn, empty, dry, 39.5 ft/sec, straight truck.

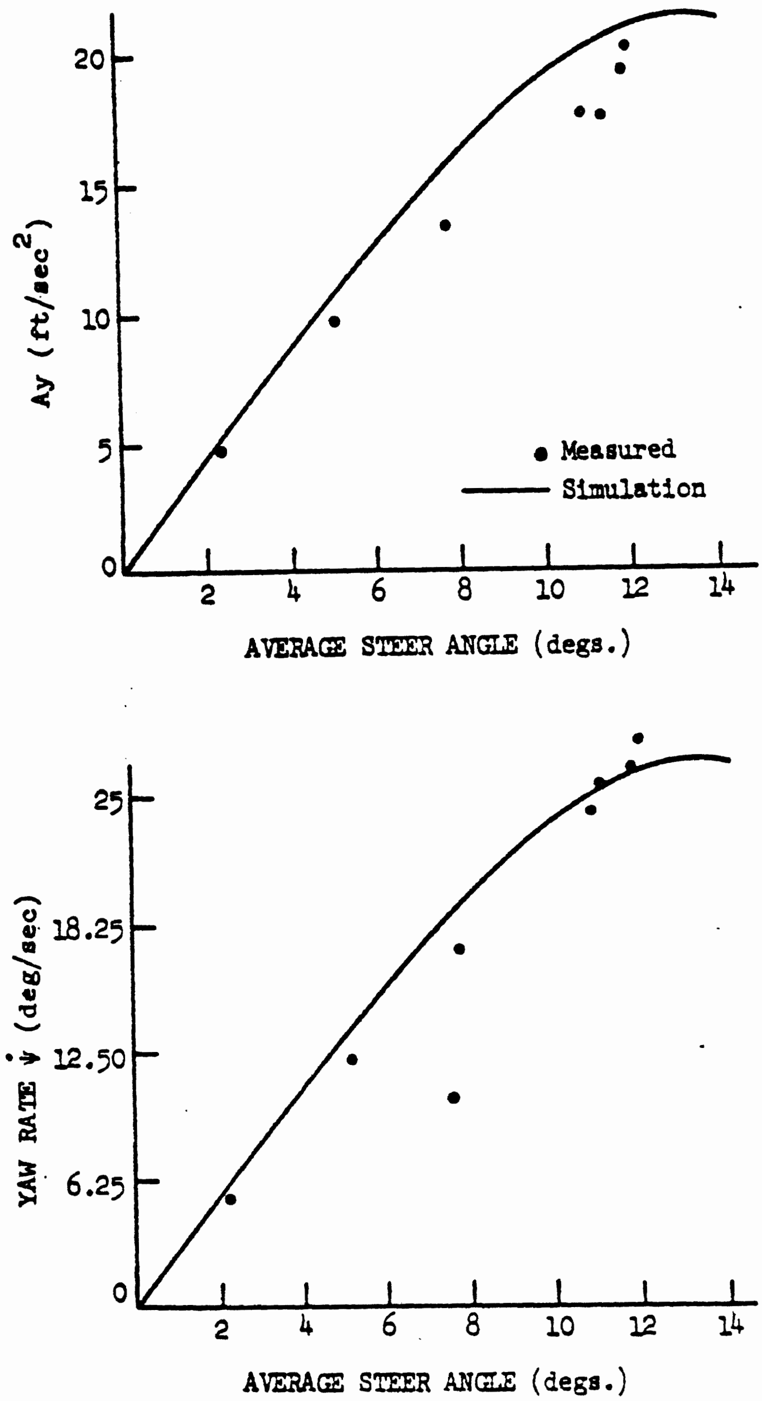

Figure 4.2. Steady turn, empty, dry, 47 ft/sec, straight truck.

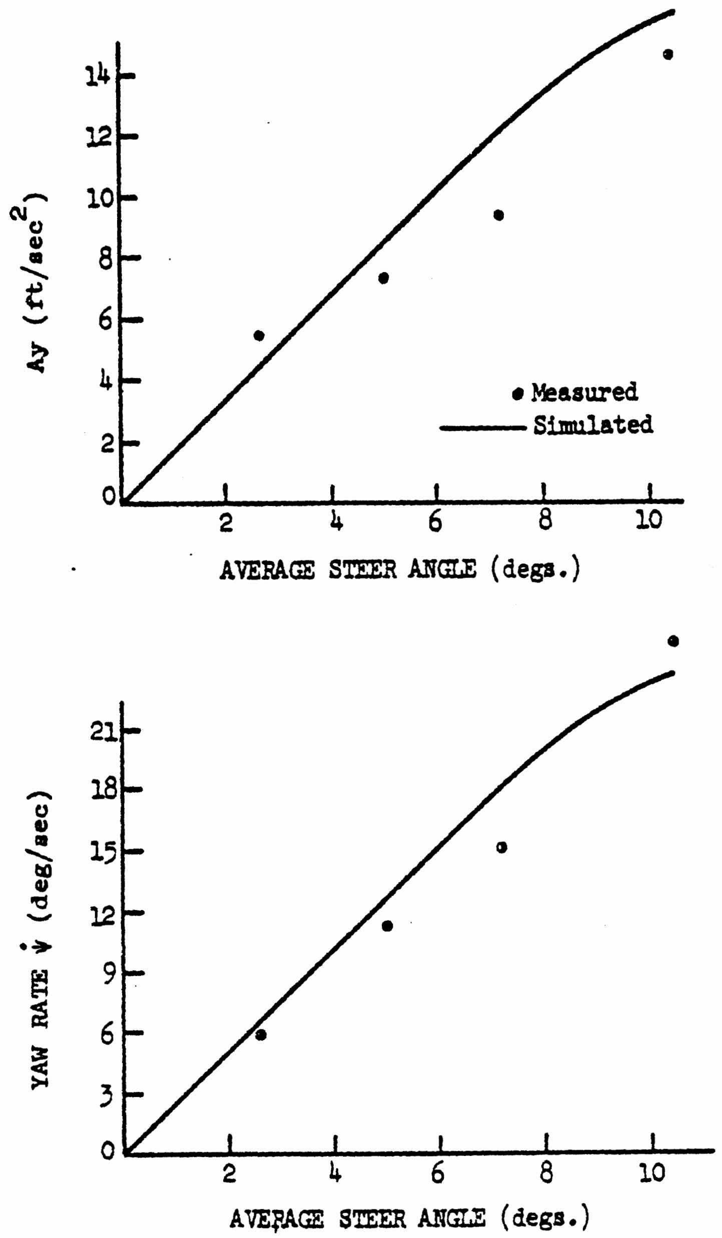

Figure 4.3. Steady turn, low c.g. load, dry, 39.1 ft/sec, straight truck.

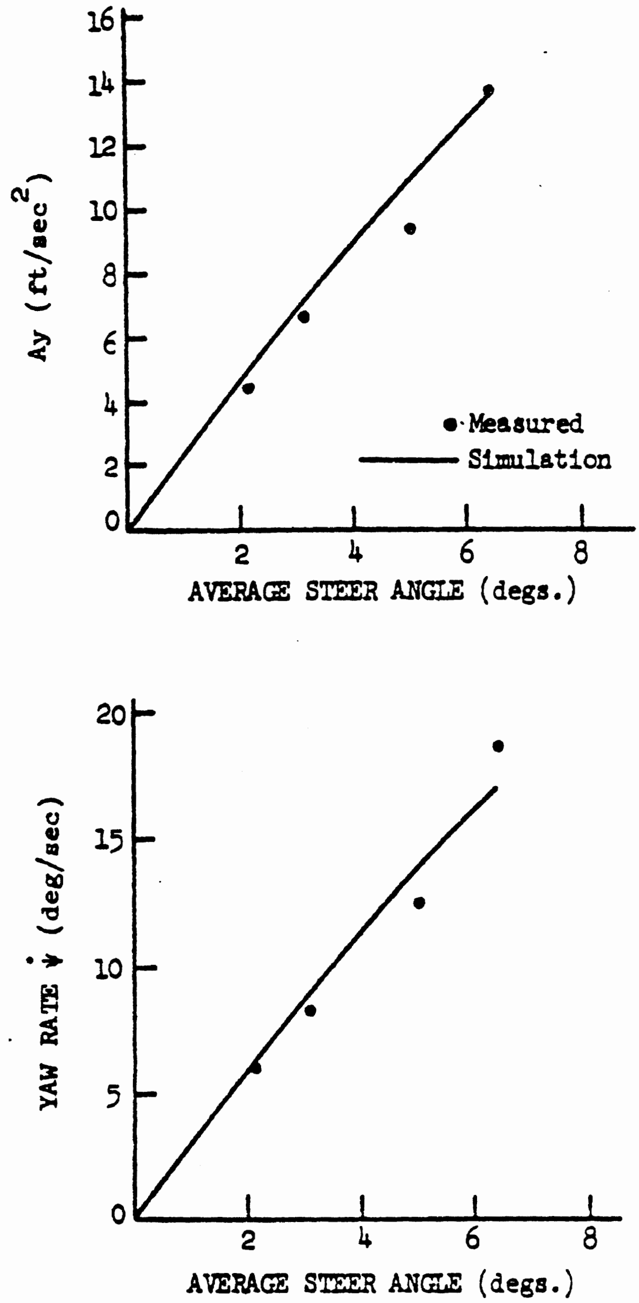

Figure 4.4. Steady turn, low c.g. load, dry, 45.6 ft/sec, straight truck.

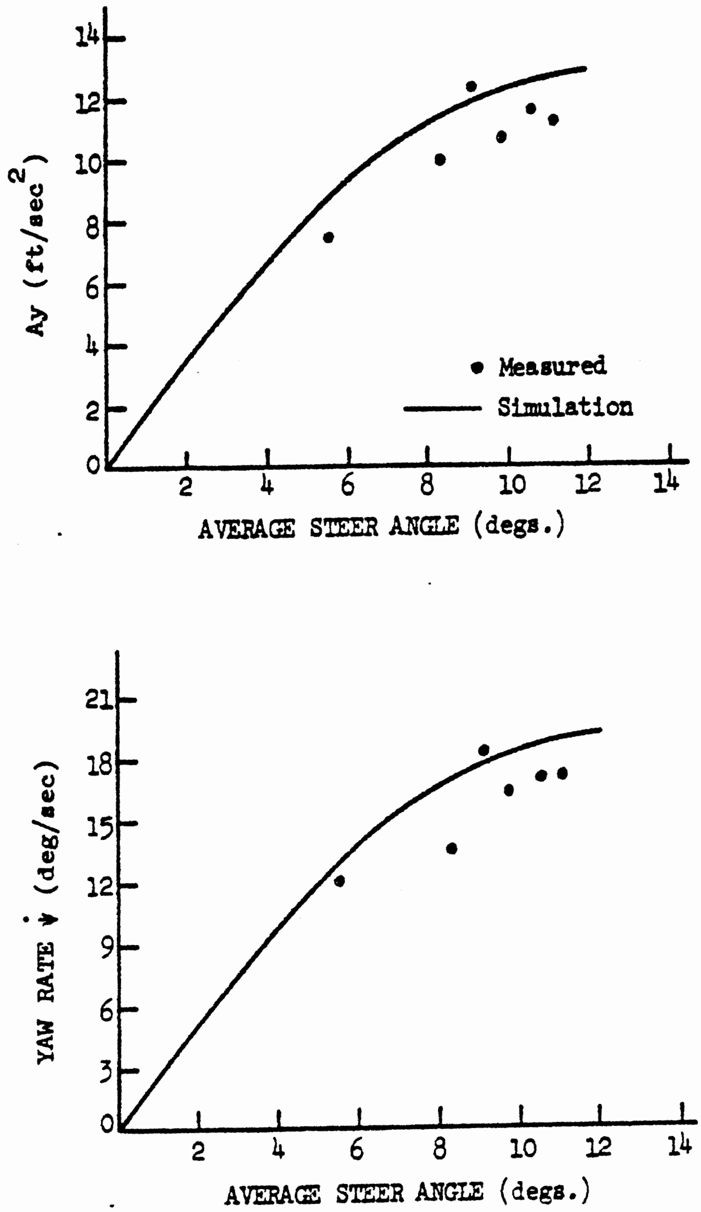

Figure 4.5. Steady turn, empty, wet, 39 ft/sec, straight truck.

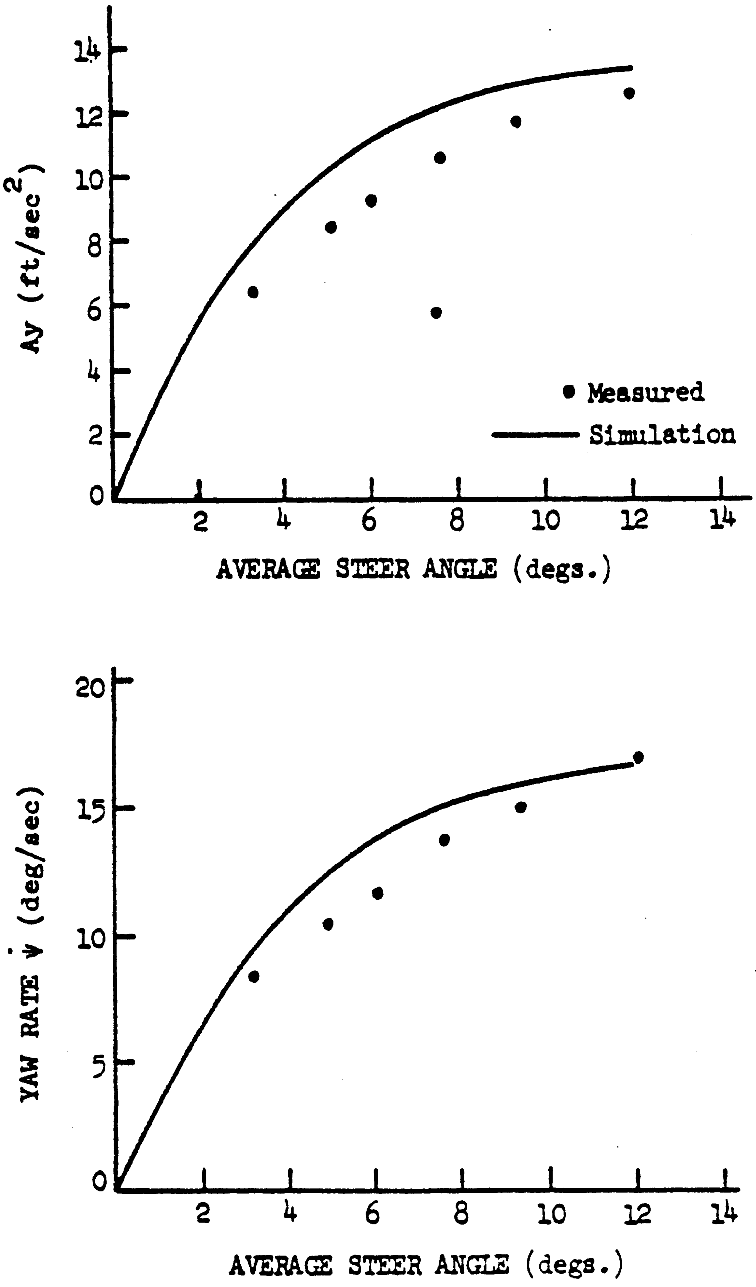

Figure 4.6. Steady turn, empty, wet, 46.8 ft/sec, straight truck.

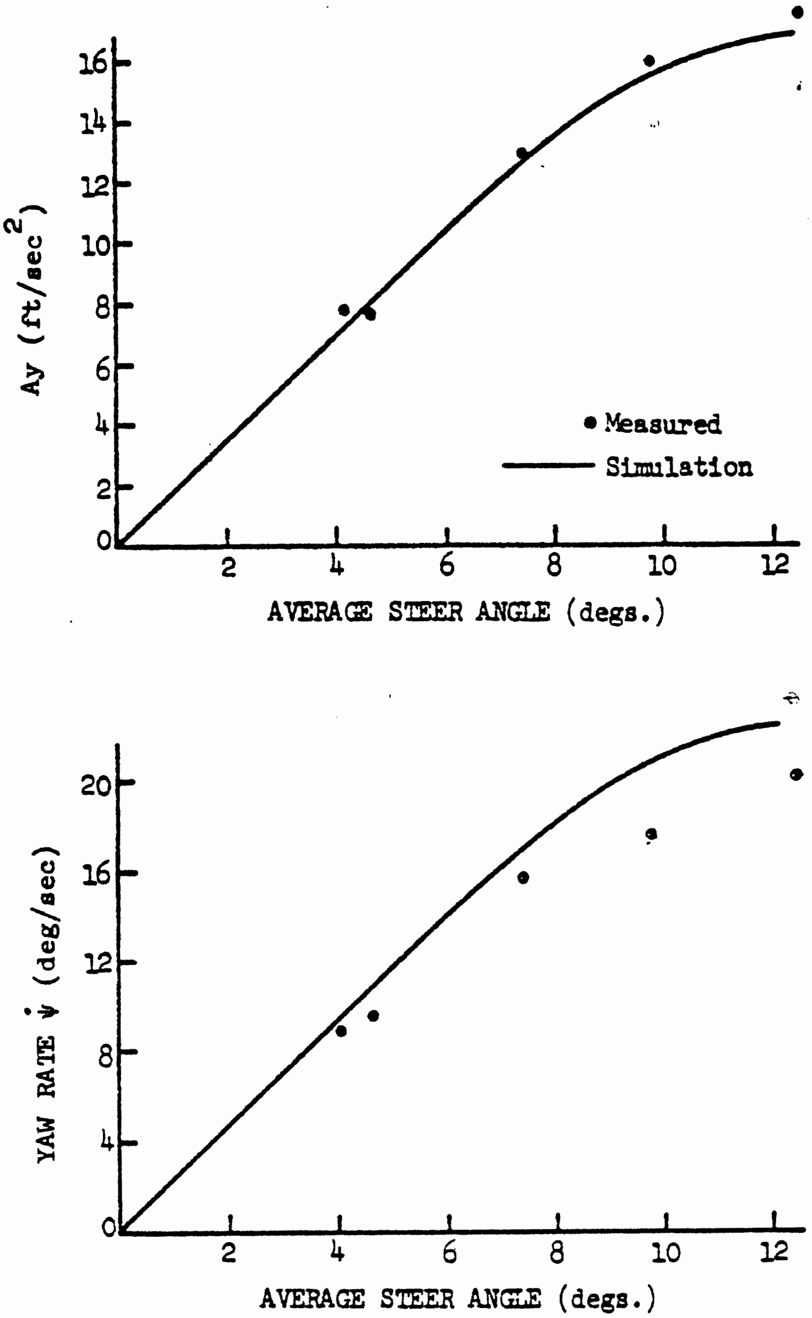

Figure 4.7. Steady turn, bobtail tractor; dry, 43 ft/sec.

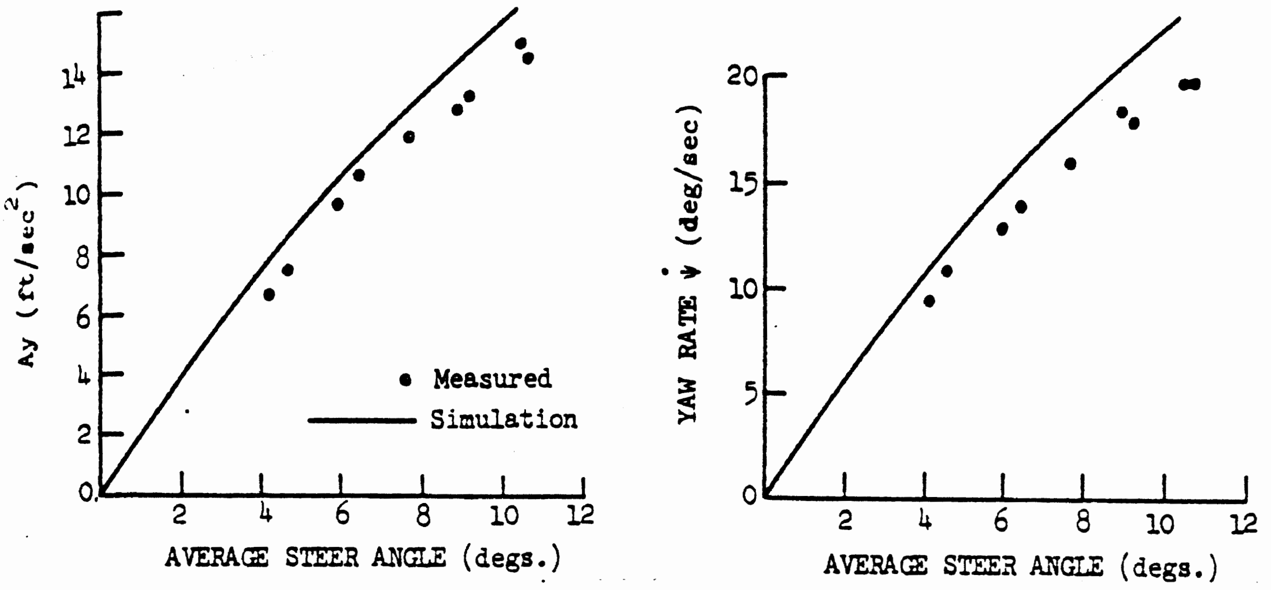

Figure 4.8. Steady turn, tractor-trailer, empty, dry.

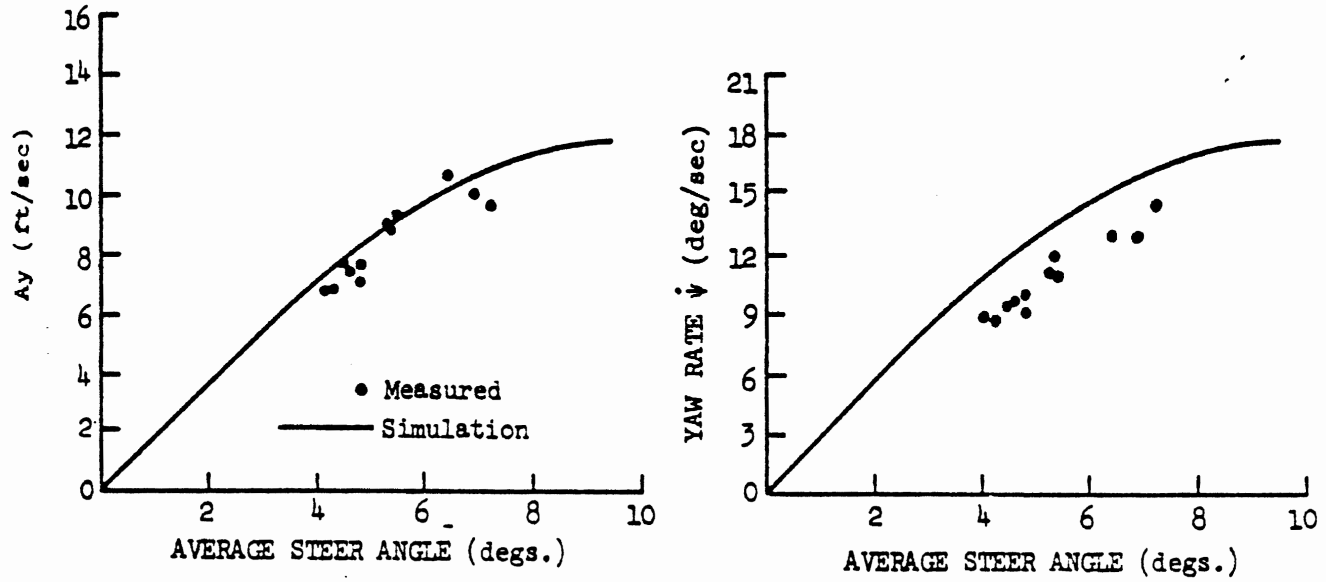

Figure 4.9. Steady turn, tractor-trailer, empty, wet.

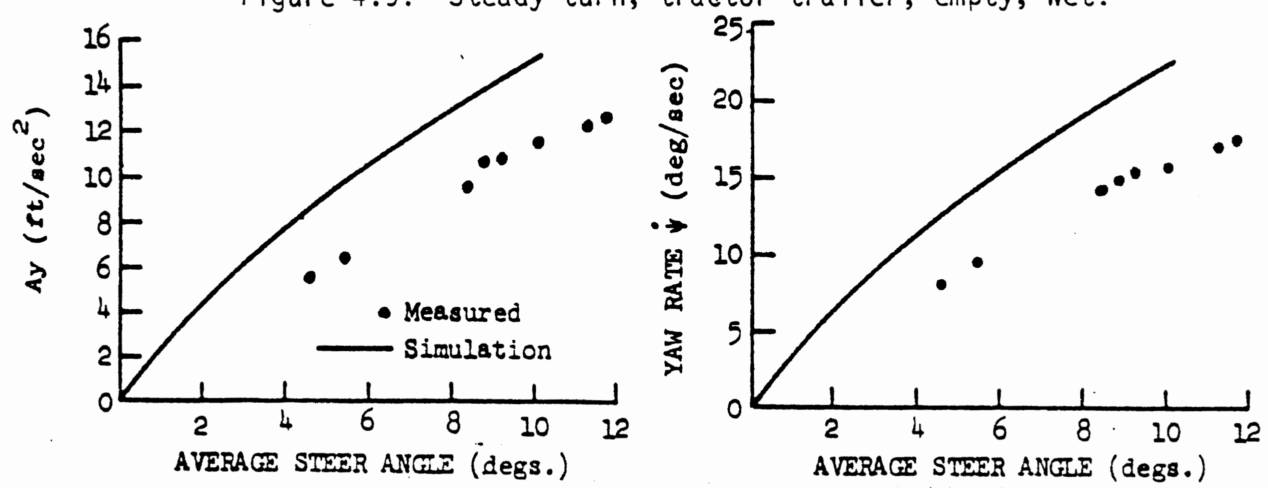

Figure 4.10. Steady turn, tractor-trailer, loaded, dry.

simulated results, since even at low lateral accelerations the simulation predicts higher lateral acceleration than the measured values. The current understanding attributes the loaded vehicle test-simulation difference to conservative load sensitivity in the tire representations and possible roll-steer effects not accounted for at the time of validation.

The experimental procedure for the braking-in-a-turn tests was outlined above. Some results from these tests are plotted in Figures 4.11 through 4.14. In these figures, steady-state lateral acceleration before the application of the brakes is plotted versus maximum longitudinal decelerations after the application of the brakes. The incidence of wheel lockup may be inferred from the manner of plotting of the point.

In the simulation runs, the actual steer and brake pressure data from the braking-in-a-turn tests was not used; rather, the simulation was used to predict the maximum longitudinal deceleration possible without wheel lockup when starting from a steady turn. Thus, for points in the area of the figures above the simulation line, the simulation will predict wheel lockup, and in the area below the simulation line, the simulation will predict that no wheels will lock. The simulated result splits the empirical data quite accurately; with few exceptions, the locked-wheel empirical results fall above the simulation line and the unlocked results below the simulation line.

Time histories of the important dynamic variables describing a braking-in-a-turn maneuver are given in Figure 4.15. In this maneuver, after entering a "steady" right turn, brakes were applied at time t = 2 seconds, and held until the vehicle stopped. Points taken directly from the empirical data were entered in the simulation for (1) the steer angle (right side steady-state 8.5°, left side steady-state, 7.0°), and (2) the applied brake pressure at the foot valve. At the time of brake application, simulated and measured speed were 36.5 ft/sec. Lateral acceleration, A_y, longitudinal acceleration, A_x,

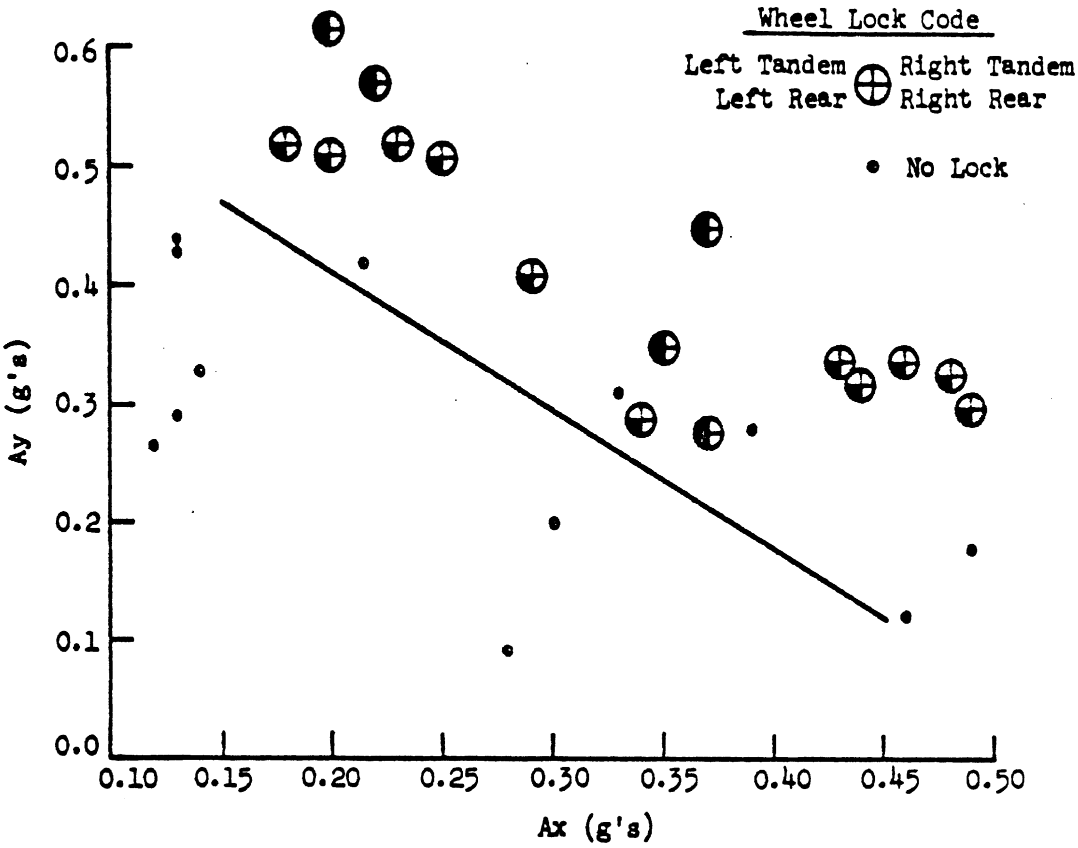

Figure 4.11. Braking in a turn; empty, dry, straight truck.

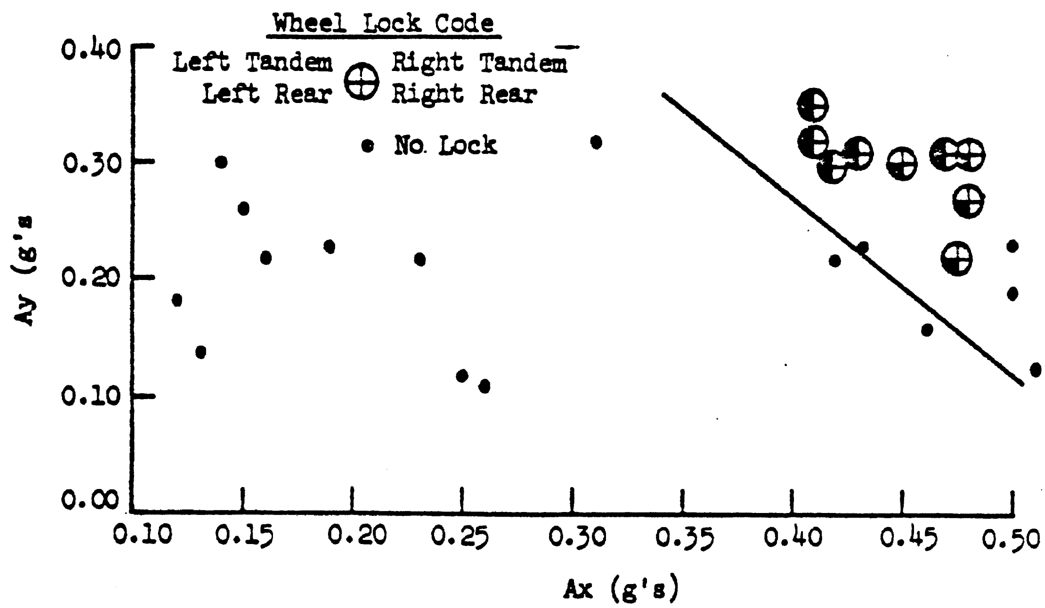

Figure 4.12. Braking in a turn; low c.g. load, dry, straight truck.

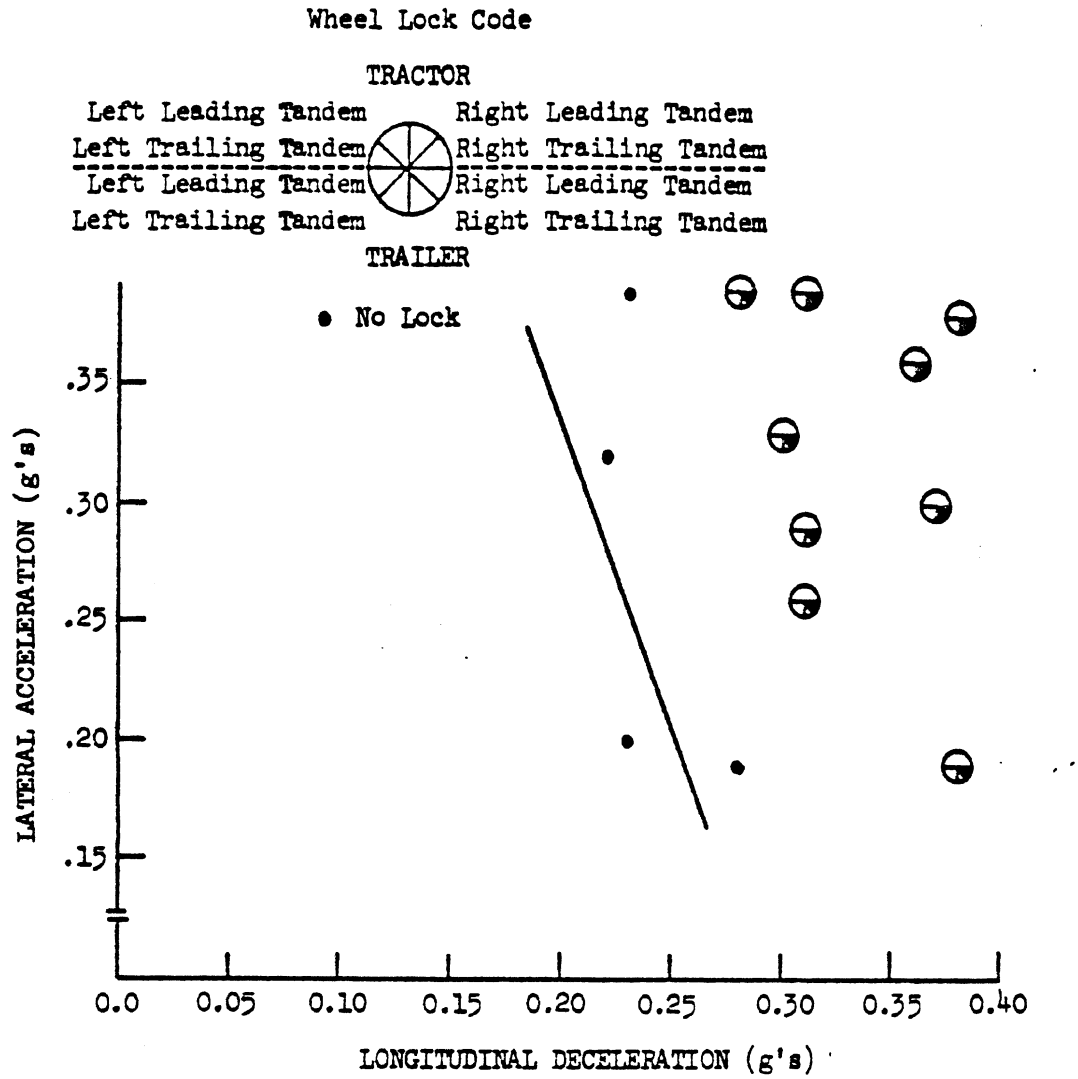

Figure 4.13. Braking in a turn; dry, empty, tractor-trailer.

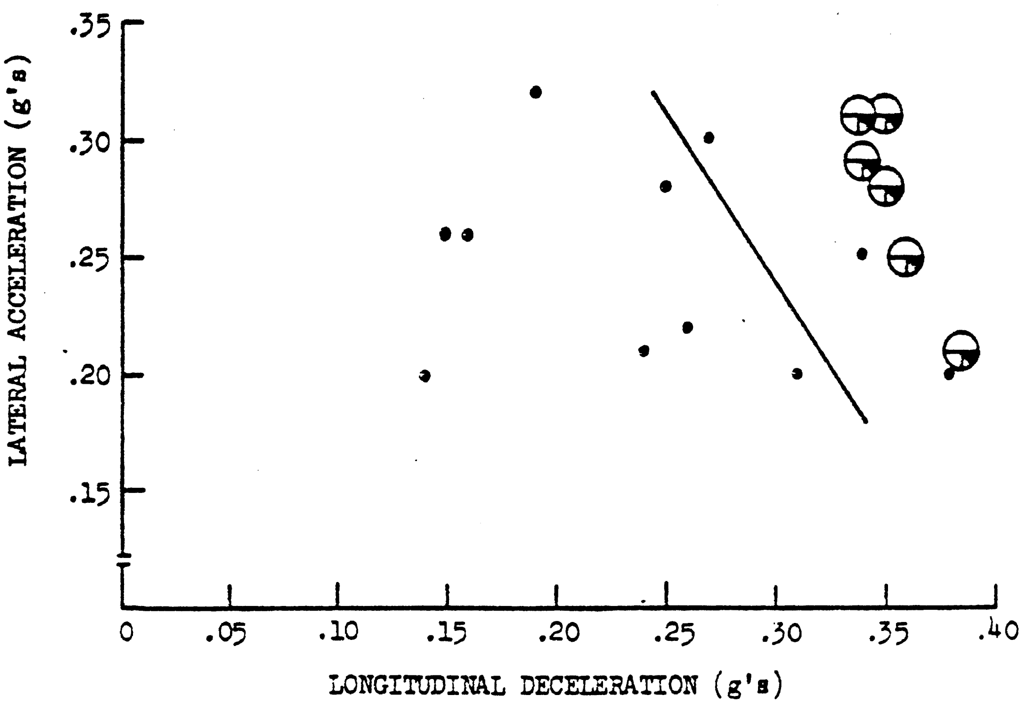

Figure 4.14. Braking in a turn; dry, loaded, tractor-trailer.

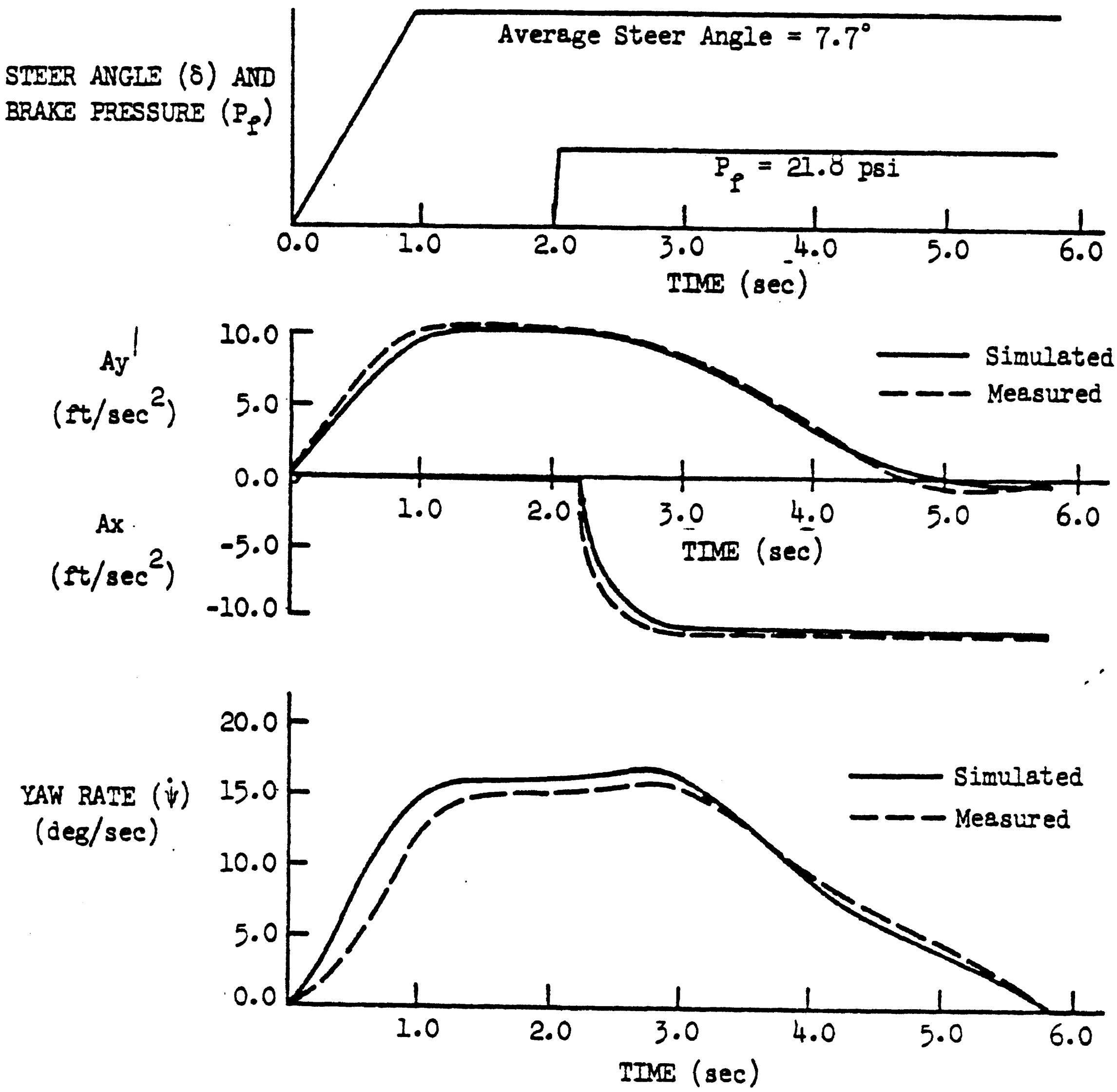

Figure 4.15. A time history of a braking-in-a-turn maneuver, straight truck.

yaw rate, $\dot{\psi}$, are plotted versus time. In this case, as in the majority of the straight truck runs, the correspondence between the empirical results and the predicted results is remarkably good.

Time histories of the important dynamic variables describing a braking-in-a-turn maneuver for the tractor-trailer are given in Figures 4.16 and 4.17. In this maneuver, a left turn with brakes applied at time t = 2.15 seconds, points taken directly from the strip chart data on board the articulated vehicle were entered in the simulation for (1) the steer angle (right side steady-state 4.73,

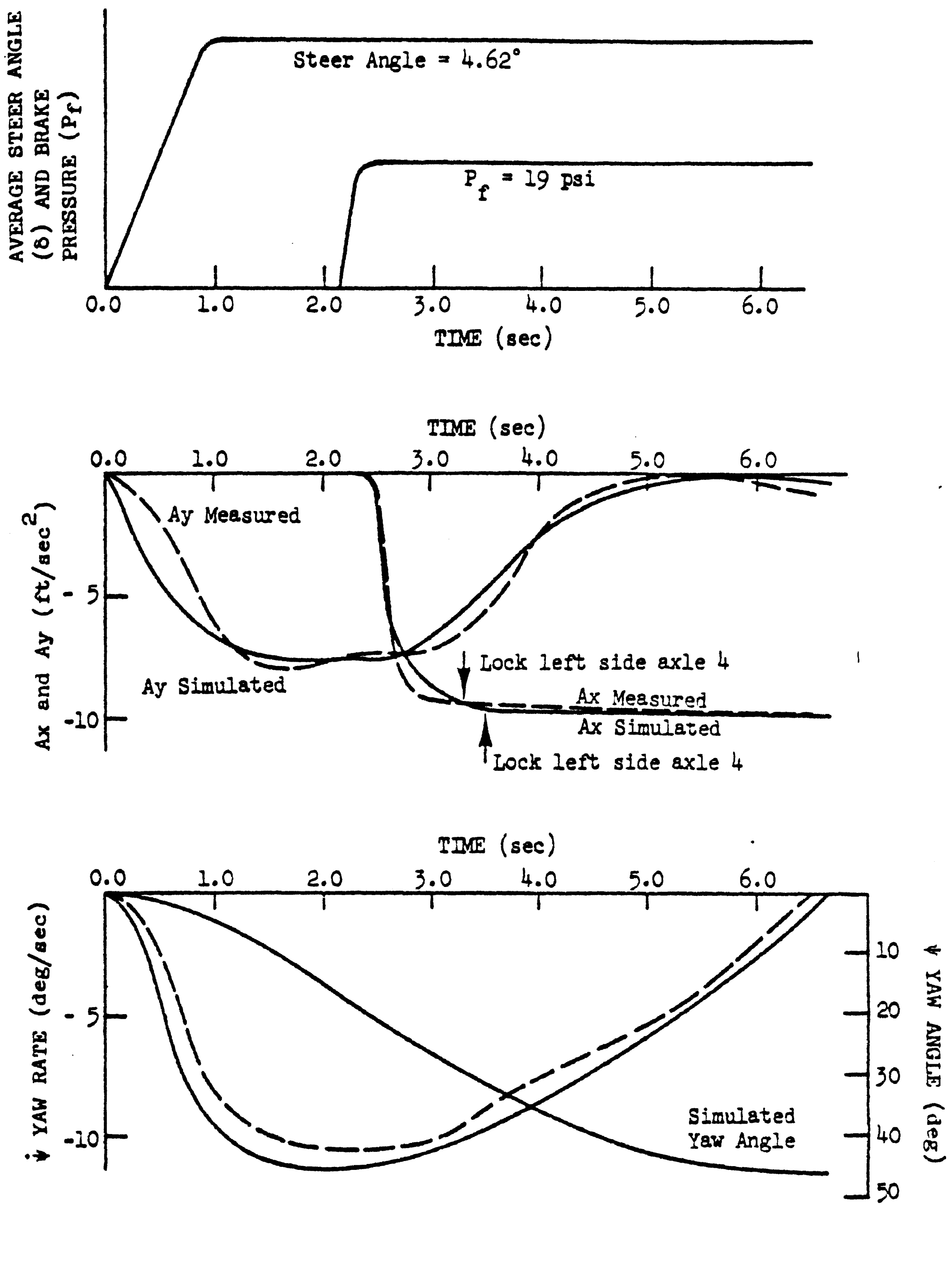

Figure 4.16. Time history of a braking-in-a-turn maneuver, tractor-trailer.

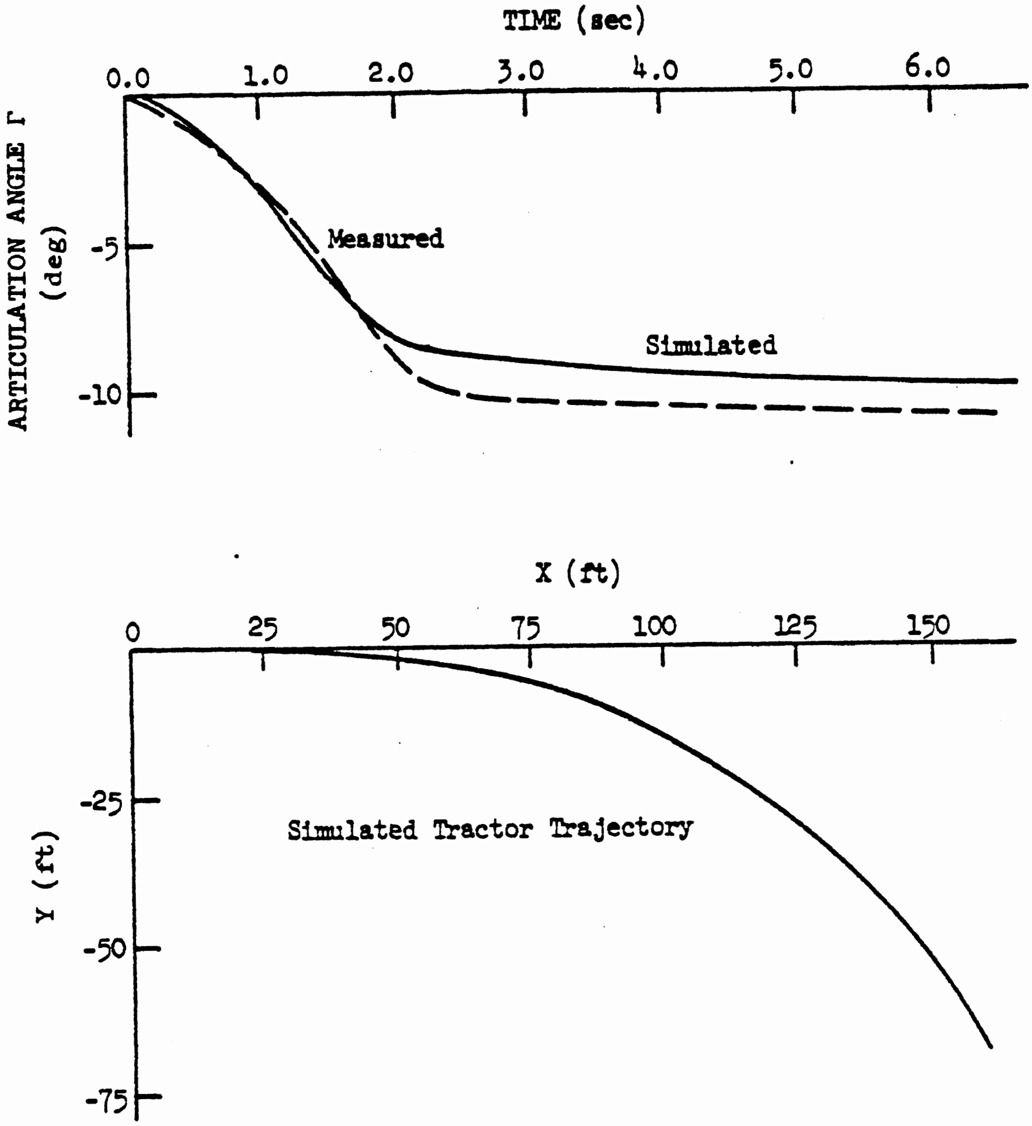

Figure 4.17. Time history of a braking-in-a-turn maneuver, tractor-trailer.

left side steady-state 4.47) and (2) the applied brake pressure at the foot valve. Lateral acceleration, A_y, longitudinal acceleration, A_x, yaw rate, $\dot{\psi}$, and the articulation angle, Γ, are plotted versus time, and the simulated trajectory is given. Predicted and measured incidence of wheel lockup are shown on the right side of the lead trailer tandem axle. Again, good agreement is seen between the experimental and simulated results.

Time histories of the important dynamic variables describing a high-speed jackknife test are given in Figures 4.18 and 4.19. In this maneuver, which starts with an initial longitudinal velocity of 60 mph, a step input is applied at the foot valve, causing line pressure to rise almost immediately to 88 psi. This was sufficient to lock all the tractor and trailer wheels in the test; this result was also predicted by the simulation. The empirical and simulated results prior to impact with the articulation angle limiter are given in Figures 4.18 and 4.19. It should be noted that, although the driver tried to maintain stability through the application of the steering maneuver shown in the figure, the simulated steer angle was held to zero.

Subsequent to the validation tests reported here, many additional informal studies have been conducted to checkout and improve various aspects of the models used. Studies in the performance of truck antiskid systems have resulted in models that can closely duplicate the pressure-cycling characteristics of antiskids on vehicles which have been studied. Recent tests have been conducted to ascertain that the simulations are accurately duplicating the influences on cornering due to tandem axles and dual tires. Likewise, studies are currently underway focusing on the accuracy with which articulated vehicle cornering is duplicated to determine the influence of frame compliance on the cornering performance of these vehicles.

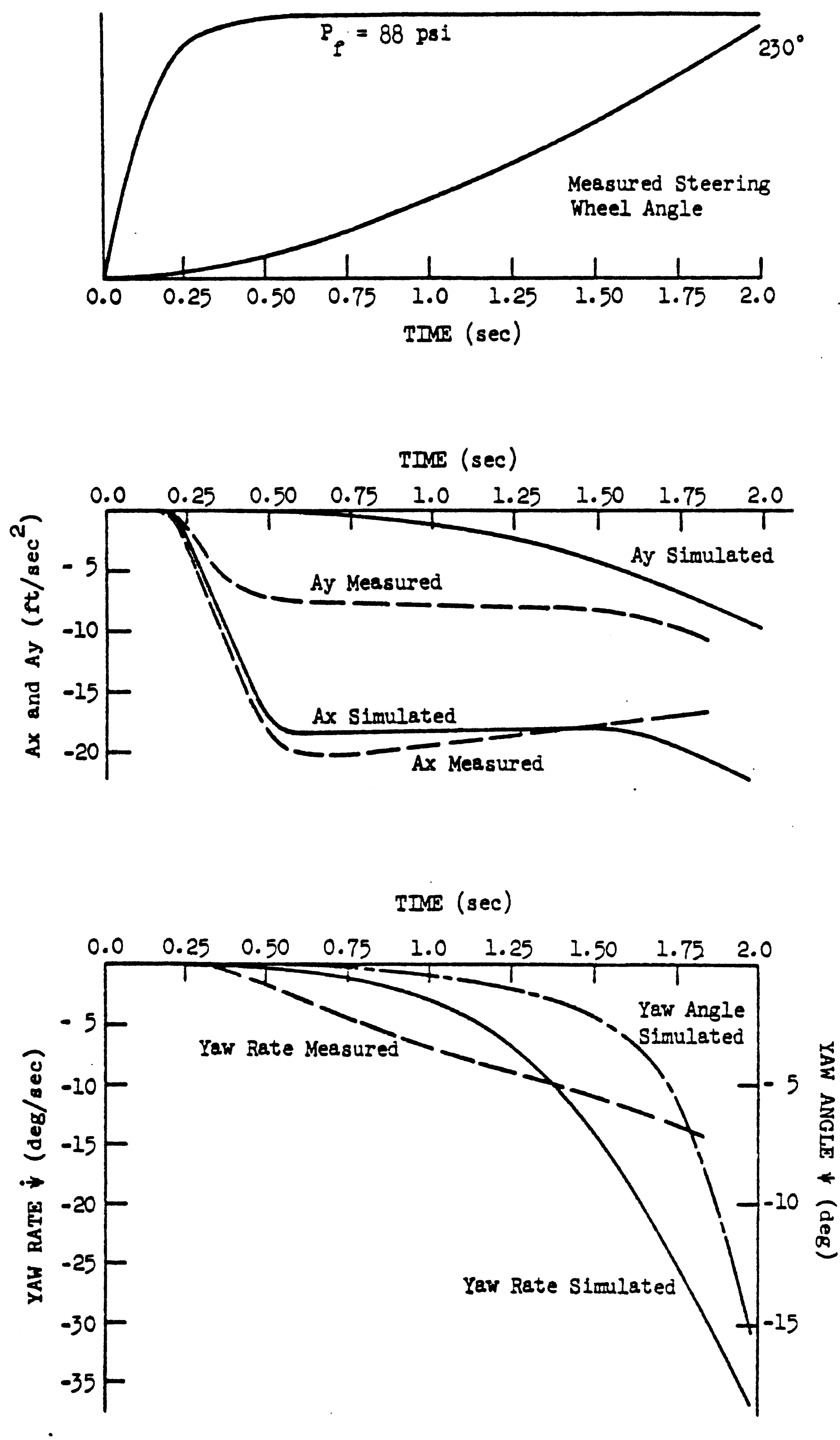

Figure 4.18. Time history of a jackknife maneuver, tractor-trailer.

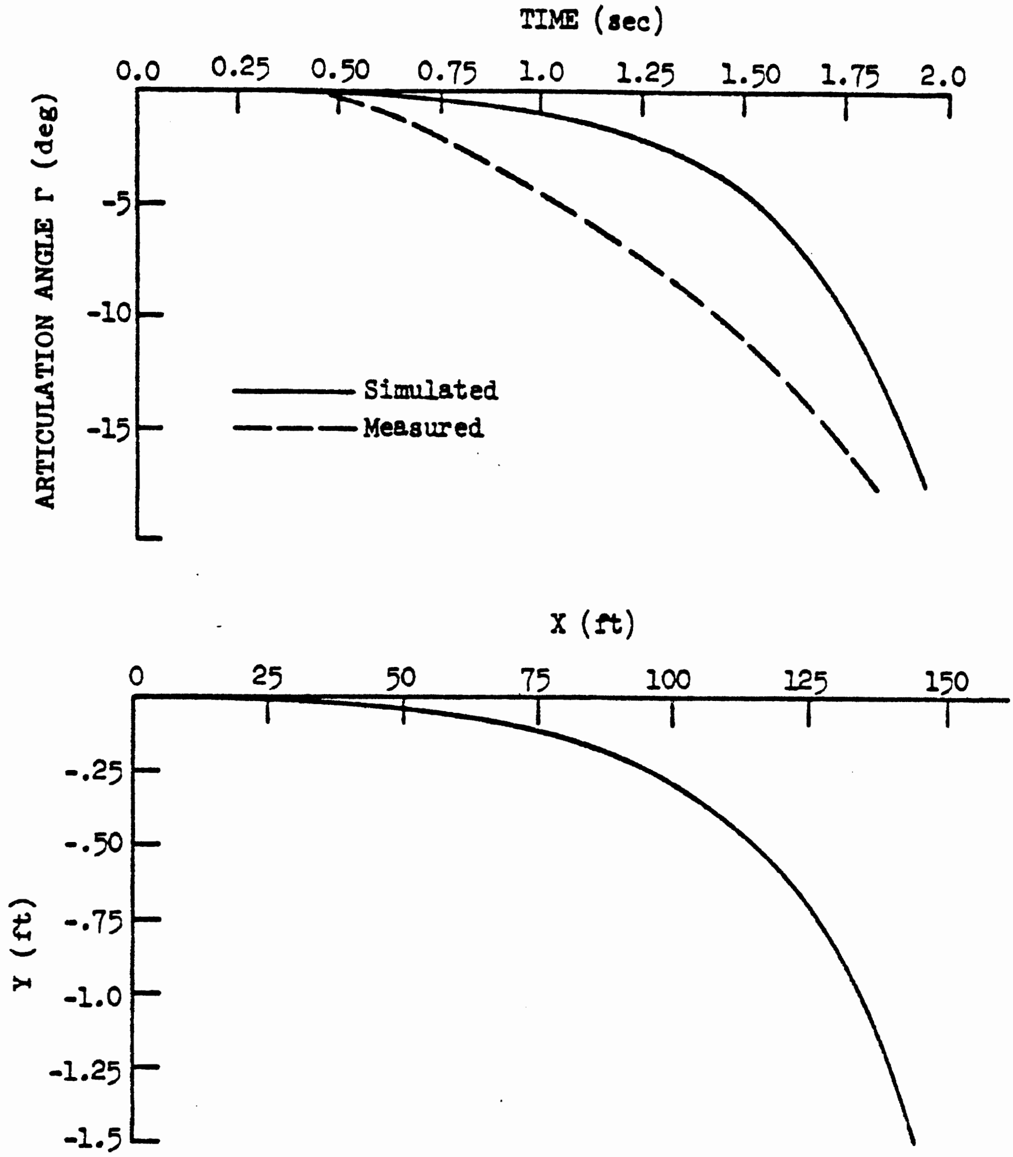

Figure 4.19. Time history of a jackknife maneuver, tractor-trailer.

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APPENDIX C

PHASE I AND PHASE II STUDY PLAN

"SIMULATION OF THE EFFECTS OF INCREASED TRUCK SIZE AND WEIGHT"

Contract No. DOT-FH-11-9330

Highway Safety Research Institute
The University of Michigan
Ann Arbor, Michigan 48109

June 1978

1.0 INTRODUCTION

This document constitutes the Phase I and Phase II study plans required under Task D of the FHWA Project DOT-FH-11-9330, "Simulation of Effects of Increased Truck Size and Weight."

Phase I addresses the subject of validating the simulation program being provided to FHWA. It draws on the extensive validation data available through other past and current projects at HSRI; and describes the manner in which data from the companion project, "Validation of Truck Handling Simulation Results," will be used to further validate the models.

Phase II represents a series of tests to investigate the effects of increased truck size and weight in the test maneuvers proposed in the report on Task A. Specific vehicles, representing some of the potential ways in which increased size and weight may be achieved, are selected for study. Because of the many variables involved, a complete evaluation of the many ways in which increased size and weight can be manifest in practice is beyond the scope of this project. Rather, this study plan is offered as a first exploratory effort to demonstrate appropriate ways for using the simulation program to evaluate the performance changes resulting from change of truck size and weight limits.

2.0 PHASE I - VALIDATION

The heavy vehicle simulation programs which will be supplied to FHWA under this program are under continuing development at HSRI. Since their origination in the early 1970's, they have experienced continuing revision, expansion, and refinement. The validation of these programs is a similarly continuous process. In this context, the full-scale testing activity being conducted as a companion to this project is seen as only a portion of the validation effort to be reported upon. Thus, we plan to supply documentation of validation efforts which greatly exceed, both in depth and breadth, that which would be available if only data from the companion study were available.

A review of some of the earliest validation efforts associated with these simulation programs was presented to FHWA in the Summary Report of 31 January 1978. The validation effort to which this plan applies will draw from the following data sources in addition to the companion study:

1. Straight truck antilock braking testing performed under MVMA sponsorship, October 1975, September, 1976 [1, 2].

2. Straight truck steering response testing performed under MVMA sponsorship, July 1977 [3].

3. Tractor-semitrailer steering response testing to be performed under NHTSA sponsorship, June-July 1978 [4].

4. Tractor-semitrailer steering response testing to be performed under MVMA sponsorship, June-July 1978 .

5. Tractor-semitrailer and tractor-semitrailer-full trailer (doubles) steering response testing performed under sponsorship by the State of Michigan, April 1978 [5].

The manner in which data available from these sources and the companion testing program will be employed is outlined in the following sections.

2.1 Braking Performance Validations: Straight-Line Braking

Perhaps the most difficult area in the validation of complex commercial vehicle simulations is braking performance validation. Commercial vehicle brakes are highly variable. Two, supposedly identical brakes, mounted on the same axle may produce substantially different brake torques in response to the same input brake pressure. Further, performance of an individual brake may vary substantially over time due to temperature, velocity, and hysteretic effects, as well as previous work history. Since these performance variations are not well understood, models of the mechanical friction brake are necessarily empirical in nature. Parametric data required by these empirical models are among the most difficult to obtain. The most useful method of obtaining such parametric data is through deduction based on actual vehicle braking test data.

This state of affairs clearly makes the a priori prediction of stopping distance an inappropriate task for state-of-the-art simulation programs. It does not, however, render braking performance simulations non-productive nor negate the need for their validation. The braking process, particularly in severe braking, involves complex, dynamic interactions of the tire friction mechanisms, rotating mass effects, unsprung masses, the sprung mass, and antilock and pneumatic systems. Braking simulations are invaluable in arriving at an understanding of the manner in which all these mechanisms interact in the braking process. The use of simulation in parameter sensitivity studies related to braking performance is one natural extension of this implied understanding. Validation of simulations with regard to the interaction of all these mechanisms, as well as stopping distance, is of paramount importance.

HSRI has completed an indepth validation study of straight-line braking using a three-axle straight truck as a subject vehicle. Although complete documentation has not been published, this activity was reported on briefly in [1, 2]. This activity will be fully documented as part of the validation effort of this study.

Braking validation will be extended into the tractor-trailer regime via data to become available through the companion study. Due to expected limitations of the test data, i.e., the lack of wheel speed and brake chamber air pressure data and the limited number of brake application levels, this activity will necessarily be shallower than that described above. Suspension deflection data from testing will be analyzed to the extent possible, to derive vehicle pitch and bounce information for comparison with simulation results. This same information, along with longitudinal acceleration data, may be useful for deriving antilock cycling data. Stopping distance tests will be used for validation within a sensitivity study context. That is, vehicle parameter changes (load and fifth wheel position) might be expected to alter stopping distance in the "full application" braking tests. The simulation will be evaluated with respect to its ability to predict these changes.

2.2 Handling Performance Validations: J-Turn, Lane-Change, and Braking-in-a-Turn

HSRI has available a substantial body of heavy vehicle test data deriving from a variety of sources upon which may rest the bulk of the handling performance validation effort. Additionally, data from the companion study will be employed.

Data for straight truck handling validation derives from work conducted by HSRI under MVMA sponsorship. In this activity, a short wheelbase, three-axle truck with air suspension rear tandem was tested in both steady turning and J-turn maneuvers of an open-loop nature. Portions of the resulting data were reported in [3]. This data will be analyzed and used for validation of steady-state and dynamic yaw plane response of the straight truck portions of the simulation program.

A major tractor-semitrailer testing program will be conducted by HSRI during summer of this year. Test vehicles in this program include the vehicle mentioned in the preceding paragraph (used as a tractor) and a short wheelbase, two-axle tractor. Each will be in combination with a 40-foot van (two-axle) and a 40-foot flatbed (two-axle) trailer. The testing is being conducted for two purposes, viz.:

1) Validation of handling simulation (under MVMA sponsorship)

2) An examination of yaw divergence and rollover tendencies, particularly as they are affected by frame compliance mechanisms (under NHTSA sponsorship.

In relation to (2) above, frame compliance of both tractors will be significantly reduced during portions of testing, thus providing for "different" tractors for validation purposes.

Testing in this program will include open-loop steady-state turning, J-turns, and lane-change maneuvers. A small amount of testing will be conducted with the vehicle empty, while the majority of the effort will be accomplished with a payload of approximately 42,000 pounds with a c.g. height of approximately 68 inches. Fifth wheel position will be varied.

The validation effort associated with this testing will be incorporated into this study.

Although it was not expected at the time of our proposal, we are now able to present validation evidence with respect to handling of double trailer combinations. Under the sponsorship of the State of Michigan, a testing program has been conducted on double trailer fuel tankers [5]. The bulk of the testing involved open-loop lane-change maneuvers of the fully-loaded vehicle. It is important to note that the test maneuver was specifically designed to excite the distinctive "snaking" behavior of the double. Thus, for validation purposes, we are able to examine the dynamic handling performance of the double which most uniquely distinguishes this vehicle from the tractor-semitrailer.

At the time of the Michigan project, the subject simulation was not yet adapted for the study of doubles. During that project, a linearized analog computer simulation was assembled to investigate handling dynamics, and a dynamic roll simulation model was developed to investigate roll motion response. Both models yielded accurate predictions of behavior over their expected range of validity. Validation of the doubles model now being prepared for FHWA will include comparison against these two simulations.

All the vehicle handling testing discussed above has, or will have, covered a broad input range. In each case, vehicle equipment includes rollover preventing outriggers so that testing can be conducted up to, and often beyond, the yaw divergence and/or rollover limit. Thus, validation may be conducted not only in the linear range, but well into the nonlinear range where heavy vehicle test data has been largely unavailable to date.

Test data to be provided by the companion project to this effort is seen as augmentation to the validation effort described above. The closed-loop nature of the handling tests described in the test plan for that project is expected to cause some difficulty in the validation effort, however, the resulting test data will remain very useful if addressed from the proper perspective. Repeatability of steering input freely applied by the driver is of obvious concern. The relatively "noisiness" of steering inputs is also a concern. When drivers superimpose small, relatively high frequency steering correction over the intended steering waveform, they are operating as a complex feedback mechanism in a closed-loop system. Thus, these corrections are favorably phased with vehicle reactions continually throughout the maneuver. If such a steering input is recorded and input to an open-loop simulation, very small errors in the frequency response of the vehicle may accumulate large phase shift errors between input and output as the simulated maneuvers proceed in time. We would note at this time that in using such data, it may be desirable to artificially smooth the steering data time histories for input to the simulation and to make comparisons between simulated and test vehicle responses only in terms of the lower frequency portion of those responses.

We also feel compelled to note that trajectory data must also be considered from the proper perspective. The test plan for the companion study quite rightly takes note of "the inaccuracies inherent in determining exact trajectory through double integration of measured dynamic parameters." It is important to recognize that simulations also determine trajectory by double integration of the same dynamic parameters. By its nature, this process accentuates very small errors in acceleration made early in time as well as accumulating phase errors. Thus, trajectory data is not particularly appropriate for validation of the open-loop simulations. (More valuable validation information can be expected to derive from comparison of the time histories produced by on-board instrumentation with those obtained from simulation.)

Conversely, trajectory data will be useful in testing the path-follower functions which will be developed for this simulation for FHWA. The ability of this feedback control algorithm to reproduce the vehicle trajectory obtained in the vehicle testing program will be evaluated. Comparison of the general quality of the steering time histories produced by the path-follower and the driver will be made.

3.0 PHASE II PLAN: AN EXAMINATION OF THE EFFECTS OF INCREASED SIZE AND WEIGHT

The following paragraphs outline the matrix of simulation runs which are planned for an examination of the effects of increases in size and weight on the braking and handling performance of highway tractor-semitrailer vehicles. The magnitude of this study is such that it cannot be viewed as definitive at all. Rather, it is a broad, brief scan that may help to (1) indicate specific areas of concern which should later be examined in more depth and (2) provide an indication of the usefulness of simulations in examining and understanding trends in vehicle performance.

3.1 Study Vehicles

Seven (7) vehicles have tentatively been defined as subjects of the simulation study. The baseline vehicle is prototypical of common line-haul tractor-trailer combinations. The other six vehicles represent perturbation of the baseline based on alterations of axle load ratings and length, height, and width dimensions. The vehicles are:

1) The baseline: a typical 142-inch wheelbase, three-axle tractor hauling a 40-foot, two-axle trailer with load ratings of 12, 34, and 34 thousand pounds at the front, tractor tandem, and trailer tandem axles, respectively. Trailer payload and fifth wheel positions will be established in order to establish these axle load conditions. The payload c.g. will be located at a height of 68 inches. All appropriate brake, antilock, suspension geometrics, mass, and other parameters will be typical of those measured for this class of vehicle previously.

2) A vehicle similar to (1) with the payload weight, longitudinal payload position, and fifth wheel position altered to obtain axle loads of 14, 34, and 34 thousand pounds.

3) A vehicle similar to (1) with appropriate payload and fifth wheel changes to obtain a 12, 38, and 38 thousand pounds axle loading combination.

4) A vehicle similar to (1) with all axle loads increased to obtain 14, 38 and 38 thousand pounds at the front, tractor tandem, and trailer tandem, respectively, again with appropriate payload and fifth wheel alterations to achieve these loads.

5) A vehicle similar to (4) with the payload c.g. raised proportionately to its increased weight relative to that of vehicle (1). That is, assuming a payload floor at a height of 54 inches, previous payloads are centered 14 inches above the floor. Based on the same density, the increased payload would be centered approximately 16 1/2 inches above the floor or 70 1/2 inches above the ground.

6) A vehicle similar to (4) with the trailer length extended in proportion to the increased payload, that is, a trailer length of approximately 47 feet yielding a vehicle length of approximately 58 feet.

7) A vehicle similar to (1) with the front and rear axle ratings increased to 15, 38 and 38 thousand pounds, respectively. Payload and fifth wheel changes will be made to obtain 14, 38 and 38 thousand pounds axle loadings at the c.g. height determined for vehicle 5.

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In the case of vehicle 7, axle load increases will be accompanied by increases in brake torque effectiveness and suspension stiffnesses. When possible, these increases will be determined from parametric measurement data on hand. Otherwise, increases will be made in proportion to the increased axle loads. Antilock logic parameters will be maintained constant across vehicles. Moment of inertia parameters will be altered appropriately with respect to mass and geometric changes.

While these accompanying changes are rational, they are also somewhat arbitrary. No doubt, a serious design effect could result in parameter values for such items as brake effectiveness and antilock logic which would produce a better performing, larger vehicle than what will appear in this study. Clearly, however, this study is not of the magnitude necessary for accomplishing such an optimization task.

3.2 Simulation Test Maneuvers

Each vehicle will be simulated in five (5) different types of test maneuvers, viz.:

1) straight-line braking
2) braking in a turn
3) step steer
4) sinusoidal steer
5) slow ramp steer

Each maneuver and the response parameters to be employed in comparing vehicles are discussed in the following paragraphs.

3.2.1 Straight-Line Braking. Each vehicle will be simulated performing stops on a high friction, level surface from 60 mph. One stop will be at full brake application and approximately four additional stops will be made at varying levels to identify maximum

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performance prior to antilock cycling. Comparisons will be based on stopping distance, maximum wheel loads, and number of antilock cycles.

Additionally, two stops will be made with each vehicle on the same high friction surface, however, with a significant cross-slope. One stop will be made at full brake application; the second will be made at the highest deceleration level not expected to produce antilock cycling on any vehicle. Comparisons of response variables reflecting on vehicle stability will be made. Comparison variables will include peak tractor heading angle, peak tractor yaw rate, and peak articulation angle.

3.2.2 Braking in a Turn. Approximately five (5) braking-in-a-turn simulation runs will be made with each vehicle. In these runs, a steady turn at 60 mph and .25 g lateral acceleration will be established at which time brake applications will be made to attain a desired longitudinal deceleration level, beginning at .3 g's and increasing in .05 g increments in successive runs until maximum stable performance is attained. Comparisons between vehicles will be made based on maximum deceleration obtainable within yaw stability limits, maximum dynamic wheel loads, peak tractor yaw acceleration, and peak tractor slip angle velocity. (The latter measure reflects on the level at which the yaw response of the vehicle is sensitive to the application of braking.)

3.2.3 Step Steer. Approximately six (6) step-steer maneuvers will be conducted with each vehicle. Steering angles will be selected such that the resulting steady-state lateral acceleration at 60 mph will increment in .1 g steps from run to run, beginning at an initial level of .1 g. Additional runs will be made at higher level lateral accelerations to more closely establish maximum performance. Measures of interest derived from this maneuver include maximum steady-state lateral acceleration, tractor yaw rate gain, response time and overshoot, lateral acceleration amplification of the trailer relative to the tractor, and peak wheel loads.

3.2.4 Sinusoidal Steer. Sinusoidal steer is the open-loop equivalent of a lane-change maneuver. Each vehicle will be subjected to approximately ten simulation runs of this type at a speed of 60 mph. Initially, five runs will be made with sine wave steer angle inputs varying in periods from 1.5 to 3.5 seconds in .5-second increments. Steering magnitude will be chosen to obtain lateral acceleration peaks at the tractor of approximately .25 g. These tests will provide information concerning the frequency sensitivity of the vehicle in this maneuver. A particular period will then be chosen and five additional runs will be made to establish maximum performance of the vehicle in this maneuver. Comparisons between vehicles will be made based on maximum lateral acceleration at the tractor, final heading angle, lateral acceleration amplification at the trailer and maximum wheel loads.

3.2.5 Slow Ramp Steer. Each vehicle will undergo one slow ramp steer simulation run. In this run, initial velocity is 60 mph and (front wheel) steer angle is slowly (1°/sec) increased from an initial value of zero. The measure of interest is the maximum lateral acceleration attainable within stability limits and the nature of the limit (directional or rollover stability).

The preceding paragraphs have described a simulation program of approximately 210 runs. This number may vary somewhat, depending upon maximum achievable performance and the number of iterative runs required to determine that level in the various simulation run types. The program will result in a broad view of the relative performance of the seven selected subject vehicles.

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APPENDIX D

SUMMARY OF TEST VEHICLE SIMULATION PARAMETERS

Test Vehicle: 1977 GMC Astro COE Tandem Tractor, Trailmobile 40-Foot Tandem Axle Van Trailer.

Weights

Bobtail tractor front axle (static)	9437 lb.
Bobtail tractor rear axle (static)	7953 lb.
Trailer rear axle (static)	8650 lb.
Kingpin weight (static)	2815 lb.
Tractor front axle unsprung weight	1450 lb.
Tractor rear axle unsprung weight	4925 lb.
Trailer rear axle unsprung weight	3060 lb.

Inertial Properties

Tractor pitch moment of inertia	105,493 in-lb-sec^2
Tractor c.g. height	40.13 inches
Trailer pitch moment of inertia	542,486 in-lb-sec^2
Trailer c.g. height	74.82 inches
Tractor roll moment of inertia	36,757 in-lb-sec^2
Tractor yaw moment of inertia	241,479 in-lb-sec^2
Trailer roll moment of inertia	66,224 in-lb-sec^2
Trailer yaw moment of inertia	
Forward bogey position	769,718 in-lb-sec^2
Midpoint bogey position	828,161 in-lb-sec^2
Rear bogey position	892,866 in-lb-sec^2
Tractor front axle roll moment of inertia	5,307 in-lb-sec^2
Tractor rear axle roll moment of inertia	12,230 in-lb-sec^2
Trailer rear axle roll moment of inertia	4,746 in-lb-sec^2
Tractor front wheel polar moments of inertia	245 in-lb-sec^2/axle
Tractor rear wheel polar moments of inertia	458 in-lb-sec^2/axle
Trailer wheel polar moments of inertia	425 in-lb-sec^2/axle

Suspension Parameters

Tractor front axle spring rate	1,380 lb/in
Tractor front axle coulomb/viscous damping	670 lb.
Tractor front axle roll center height	20 inches
Tractor front axle auxiliary roll stiffness	9,900 in-lb/deg
Tractor front axle roll steer coefficient	0.17 deg/deg

Tractor rear axle spring rate	3,880 lb/in
Tractor rear axle coulomb/viscous damping	1,741 lb.
Tractor rear axle roll center height	29.62 inches
Tractor rear axle auxiliary roll stiffness	30,000 in-lb/deg
Tractor rear axle roll steer coefficients	.007 deg/deg
Trailer rear axle spring rate	7,818 lb/in
Trailer rear axle coulomb/viscous damping	2,600 lb.
Trailer rear axle auxiliary roll stiffness	35,490 in-lb/deg
Trailer rear axle roll steer coefficients	-.004 deg/deg

Brakes

Tractor front axle	- Transport delay	0.02 sec.
	- Pushout lag	0.07 sec.
	- Pushout pressure	7 psi
	- Rise time constant	0.17 sec.
	- Torque	763.5 in-lb/psi
Tractor rear axle	- Transport delay	0.05 sec.
	- Pushout lag	0.08 sec.
	- Pushout pressure	5 psi
	- Rise time constant	0.40 sec.
	- Torque	1463 in-lb/psi
Trailer rear axle	- Transport delay	0.14 sec.
	- Pushout lag	0.08 sec.
	- Pushout pressure	10 psi
	- Rise time constant	0.17 sec.
	- Torque	1166 in-lb/psi

Geometry

Tractor wheelbase	150 inches
Trailer wheelbase	
Forward bogey position	347 inches
Midpoint bogey position	365 inches
Rear bogey position	383 inches
Tractor front axle track width	79.5 inches
Tractor rear axle track width	72.0 inches
Trailer axle track width	71.5 inches

Tractor front axle spring spacing	36 inches
Tractor rear axle spring spacing	40.75 inches
Trailer axle spring spacing	38 inches
Tractor dual tire spacing	12.8 inches
Trailer dual tire spacing	12.4 inches
Fifth wheel position	
Forward position	23.5 inches
Midpoint position	13.2 inches
Rear position	0.0 inches
Fifth wheel height	47.5 inches
Payload weight	36,887 lb.
Distance ahead of rear suspension	
Forward bogey position	145 inches
Midpoint bogey position	163 inches
Rear bogey position	181 inches
C.G. height	75.5 inches
Roll moment of inertia	79,891 in-lb-sec^2
Pitch moment of inertia	2,015,800 in-lb-sec^2
Yaw moment of inertia	2,250,655 in-lb-sec^2