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SIMULATION OF THE BRAKING AND HANDLING OF TRUCKS AND TRACTOR-TRAILERS

SUMMARY REPORT:

MOTOR TRUCK BRAKING AND HANDLING PERFORMANCE STUDY

BY

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SIMULATION OF THE BRAKING AND HANDLING OF TRUCKS AND TRACTOR-TRAILERS

INTRODUCTION

The dynamic performance of articulated vehicles has been studied by many investigators [1]*. Both the braking performance and the turning performance of trucks and articulated vehicles have been examined analytically [2-7]. However, the non-linear equations describing vehicle motion in rapid, accident avoidance-type maneuvers are not amenable to analytic solution. To provide solutions to these non-linear equations, computer simulations have been developed [8, 9].

The current intensified interest in the braking and handling performance of trucks and tractor-trailers has resulted in the need for more sophisticated simulations than have been previously available. The simulations described herein contribute to the state-of-the-art in that careful analyses of tandem axle dynamics, tire shear force characteristics, and brake system operation have been included. In addition, many vehicle components such as truck tires, suspensions, steering systems, and fifth wheel couplings have been tested in the laboratory so that accurate descriptions of the operational characteristics of these components can be incorporated in the simulations. Laboratory equipment has been developed for measuring the inertial properties of large vehicles. Full-scale vehicle performance tests have been made so that computer predictions can be compared with experimental results to validate the simulations. In addition, new computational techniques have been employed in the computer programs to reduce costs by speeding up the solution rate.

This report presents a concise summary of the HSRI truck and tractor-trailer simulation development work.

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*Numbers in brackets designate references at the end of this report.

COMMERCIAL VEHICLE BRAKING AND HANDLING PROJECT

The Motor Vehicle Manufacturers Association (MVMA) Truck and Tractor-Trailer Braking and Handling Project was begun at HSRI in mid-1971 with the expressed purpose of establishing a digital computer-based mathematical method for predicting the longitudinal and directional response of trucks and tractor-trailers.

The work has been divided into three phases:

a) Phase I. Empirical and analytical work resulting in a validated digital computer program to predict braking performance of trucks and articulated vehicles.

b) Phase II. Empirical and analytical work resulting in a validated digital computer program to predict the directional response of trucks and articulated vehicles.

c) Phase III. Refinement and extension of the previous work including consideration of:

   1) development of a digital computer-based method for predicting the moments of inertia and center of gravity locations along the principal axes for various truck, tractor, and trailer configurations;

   2) refinement of tandem suspension models already developed and formulation of models for three additional suspension types;

   3) development of over-the-road equipment for measuring the longitudinal shear force characteristics of truck tires;

   4) development of models for typical truck anti-lock braking systems to be used with Phase I (Braking Performance) and Phase II (Braking and Handling Performance) simulation programs;

5) extension of the Phase I (Braking Performance) Program to include provision for simulating a doubles (tractor-semitrailer-full trailer) combination;

6) development of more complete models of mechanical friction brakes, which will predict the decrease in brake effectiveness as a result of fade;

7) development of a computer-based mathematical model for evaluating the acceleration and handling performance of trucks and tractor-trailer combinations;

8) development of a computer-based mathematical method to study the dynamics of air brake systems.

At the present time, Phase I and Phase II have been completed and Phase III is underway.

The Phase I braking performance program (and appropriate documentation [10]) has been distributed to the MVMA member companies and a seminar was held at HSRI to explain the use of this program. This program is now operational at several companies where it is being used to aid in the design of vehicle systems which are intended to meet impending federal motor vehicle safety standards pertaining to air brake systems for trucks, buses, and trailers.

The Phase II directional response program contains all the features of the braking performance program plus the possibility for lateral, yaw, and roll motion of the vehicle [11]. The Phase II program is to be used for predicting response to steering inputs and response to combined steering and braking inputs. The program will be useful in the analysis of the influence on directional stability of brake imbalance, split friction coefficient surfaces, and antilock braking systems.

The special features of these simulations and the test procedures used to complement the simulation are described in the remainder of this report. The analytical detail necessary for representing the mechanical complexities of trucks and tractor-trailers is described elsewhere in technical reports [10-13].

TANDEM AXLE DYNAMICS

To allow large payloads without unduly large axle loads, many trucks and articulated vehicles make use of tandem axle suspensions. These suspensions commonly have a mechanism for "load leveling," that is, an attempt to maintain equal loading on each of the tandem axles in the presence of road irregularities. This mechanism may also cause unequal load distribution during braking, which may, in fact, result in so-called "brake hop." Thus, since the normal force at the tire-road interface has an important effect on the braking process, a careful analysis of tandem suspensions is in order.

Two very common tandem suspensions have been modeled: the four spring with load leveler and the walking beam (see Figures 1 and 2). Experimental results show that the four spring suspension tends to transfer load from the leading axle to the trailing axle during braking. The opposite may be true for the walking beam suspension, which, in some cases, may transfer load from the trailing axle to the leading axle during braking. These phenomena are predicted by the computer simulation. Options are available in the computer programs to select the type of suspensions used on the vehicle to be simulated. Thus wheels unlocked braking performance can now be predicted realistically for each axle of a tandem axle suspension system.

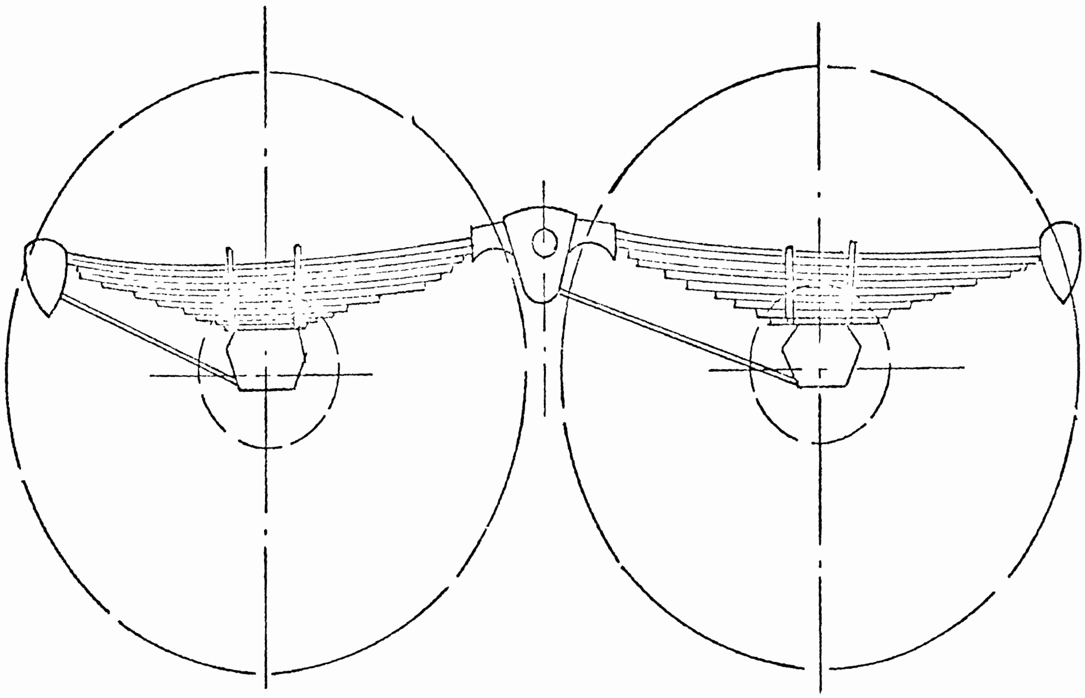

Figure 1. Four spring suspension

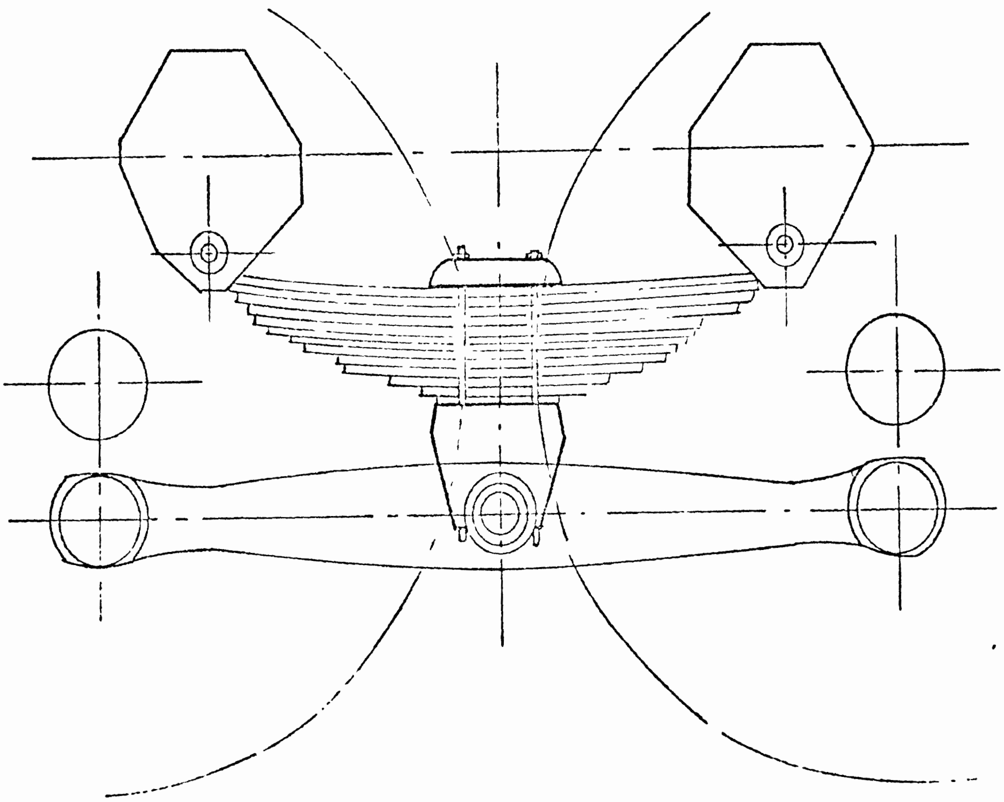

Figure 2. Walking beam suspension

BRAKE SYSTEM OPERATION

The brake system model may conveniently be divided into three sections. In a tractor-trailer air brake system the driver applies the brakes by operating a treadle valve which controls the air pressure at the brakes. In the first section of the model, the relationship between pressure at the treadle valve and the line pressure at the brakes on each axle is computed as a function of time. The time delay and the rise time characteristics of the air brake system are represented in the simulation. Typical simulation results which compare favorably with experimental measurements are shown in Figure 3.

In the second section of the brake system model the relationship between line pressure and brake torque is modeled. The program user has two options: either to input a table of brake torque for increasing line pressure, or to ask the simulation to calculate a torque versus line pressure relationship based on brake models which are contained in the computer program. The first option is quite convenient since the torque-pressure relationship may be readily available from dynamometer testing. However, since this table is not time dependent, brake fade is not precisely represented. The brake models, included in the computer program, do take into account the average effects of brake fade in a manner which yields accurate predictions of stopping distance. In Phase III, further work on representing fade as an instantaneous function of pressure, sliding velocity, and temperature has been started. Nevertheless, using the existing brake models, simulation predictions of stopping distance are very close to the experimental results as shown in Table 1.

The third section of the brake system model is now under development. When finished it will allow the user to select a variety of wheel antilock braking models. Preliminary results for a hypothetical antilock braking system have already been obtained from the Phase I simulation [12].

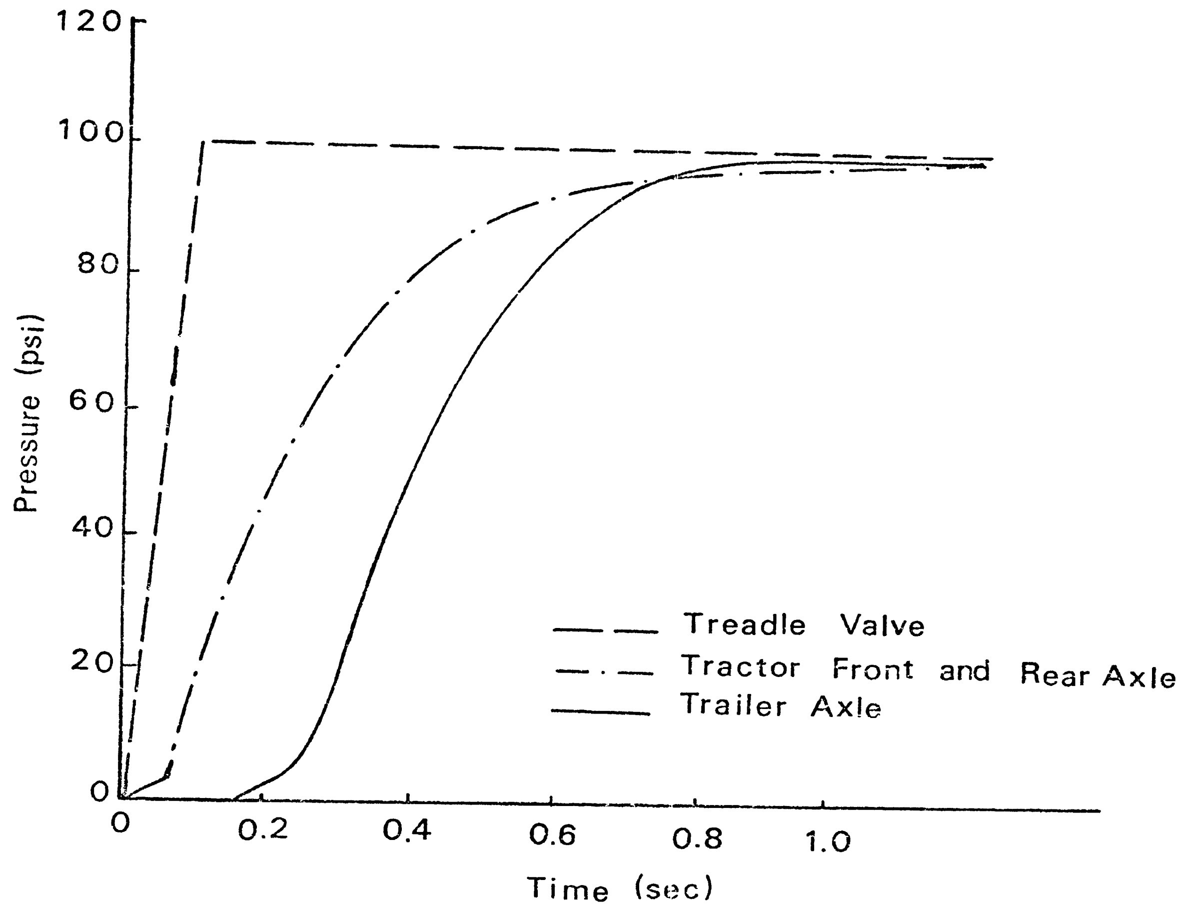

Figure 3. Simulated pressure-time relationship.

TABLE 1

TABULATED STOPPING DISTANCES

Tractor-Trailer

Brake Line Pressure, psi	Stopping Distance, Ft. Measured	Stopping Distance, Ft. Simulated	Brake Line Pressure, psi	Stopping Distance, Ft. Measured	Stopping Distance, Ft. Simulated
Empty, 60 mph, Dry Surface			Loaded, 30 mph, Wet Surface		
30	308	302	15	360	370
35	262	275	25	206	205
40	244	250	30	160	168
45	238	237	35	137	142
50	230	228	40	106	124
55	227	216	45	107	113
60	243	212	50	116	108
70	221	205			
80	197	198			
Loaded, 60 mph, Dry Surface					
40	517	490			
50	377	395			
60	358	325			
65	282/336	310			
75	284	290			
80	314	269			
Empty, 30 mph, Wet Surface					
10	321	304			
15	170	142			
20	132	120			
25	103	100			
30	104	91			
35	102	85			

VEHICLE COMPONENT MEASUREMENTS

One of the most costly, time-consuming, and difficult tasks connected with vehicle simulation is to obtain accurate values for the parameters used to describe the vehicle. This is particularly true for large, heavy vehicles because laboratory equipment for measuring component properties must be capable of applying and withstanding large forces and moments easily and without failure. Consequently, test techniques and equipment have been developed at HSRI for measuring the inertial and suspension properties of heavy highway vehicles [13] and for measuring the shear force performance of truck tires [14]. Also, techniques and equipment have been developed for the measurement of steering system deflection and compliance steer [11].

The inertial properties of significance within the simulations are:

(1) Weights of sprung and unsprung masses

(2) Center of gravity position of sprung and unsprung masses

(3) Yaw, pitch and roll moments of inertia of the sprung mass

(4) Yaw and roll moments of inertia of the unsprung masses

(5) Spin moment of inertia of the rolling masses

Methods have been devised to determine each of these properties. Special test fixtures consisting of knife edge supports, bearing pivots and swings were constructed. A typical example of the use of one of these test fixtures is shown in Figures 4 and 5. The arch members in the pendulum swings can be installed at the front and rear of the vehicle to make a swing for measuring the roll moment of inertia or they can be installed on either side of the vehicle to measure the pitch moment of inertia.

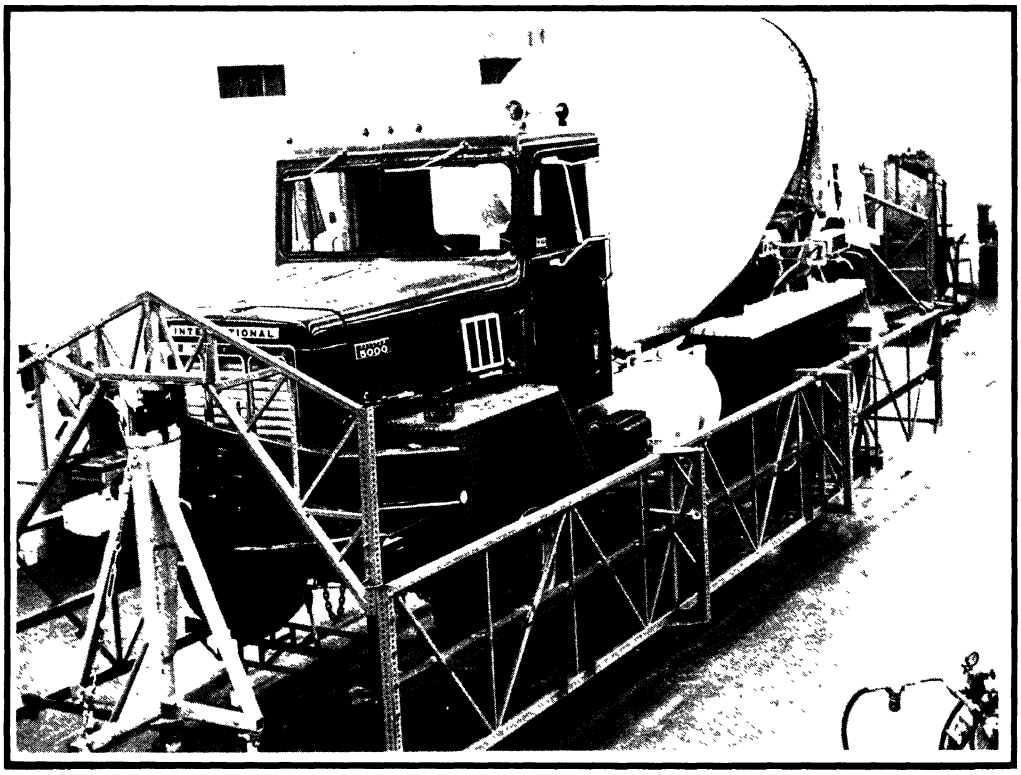

Figure 4. Roll moment of inertia testing using pendulum swing.

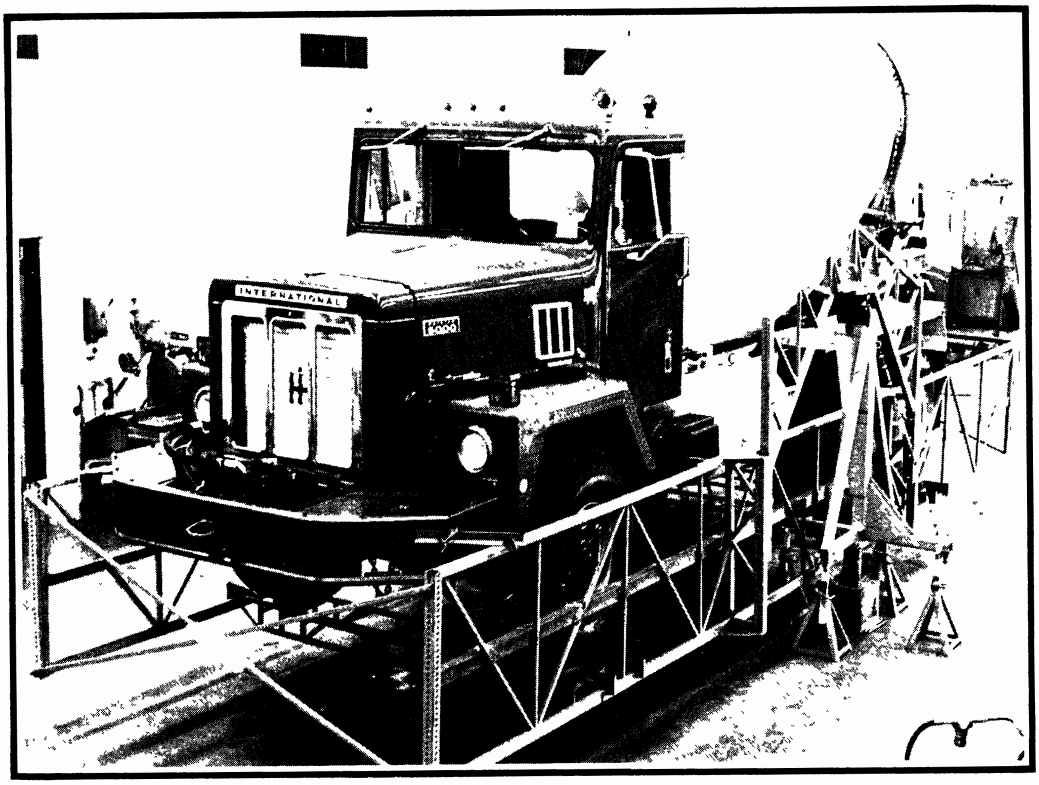

Figure 5. Pitch moment of inertia testing using pendulum swing.

The vehicle simulations require suspension spring rates and coulomb friction to be entered as input variables. Since this data is not readily available from vehicle manufacturers, the suspension properties of the vehicles to be used in the validation experiments were determined by applying and removing large vertical loads while measuring suspension deflection. Example results for a four spring suspension are shown in Figure 6. (In this case the measurements indicate an average spring rate of 5200 lb/in at each wheel and a coulomb friction which varies from 900 lbs/wheel at low loads to 1300 lbs/wheel at high loads.)

The successful prediction of motor vehicle performance depends heavily on the knowledge of tire shear force performance. The HSRI flat bed tire testing machine (shown in Figure 7) was modified to allow truck tires to be tested [14]. The tire testing machine can now be used to test tires up to 44 inches outside diameter with loads up to 10,000 lbs. For simulation purposes, two very important tire parameters can readily be obtained from the flat bed machine. These parameters are the cornering stiffness and the longitudinal (braking) stiffness of the tire. At the present time, HSRI is constructing an over-the-road device for testing truck tires at highway speeds. With the flat bed machine and the over-the-road device, HSRI will have two complementary pieces of equipment for measuring truck tire performance over a broad range of operating conditions.

As part of the Phase I and Phase II study, an extensive set of tire measurements were made with the flat bed machine. The influence of tire size, ply rating, inflation pressure, tread pattern, and wear were studied and reported on in reference 14. An example "carpet plot," which shows the dependence of lateral force capability on inflation pressure, is given in Figure 8.

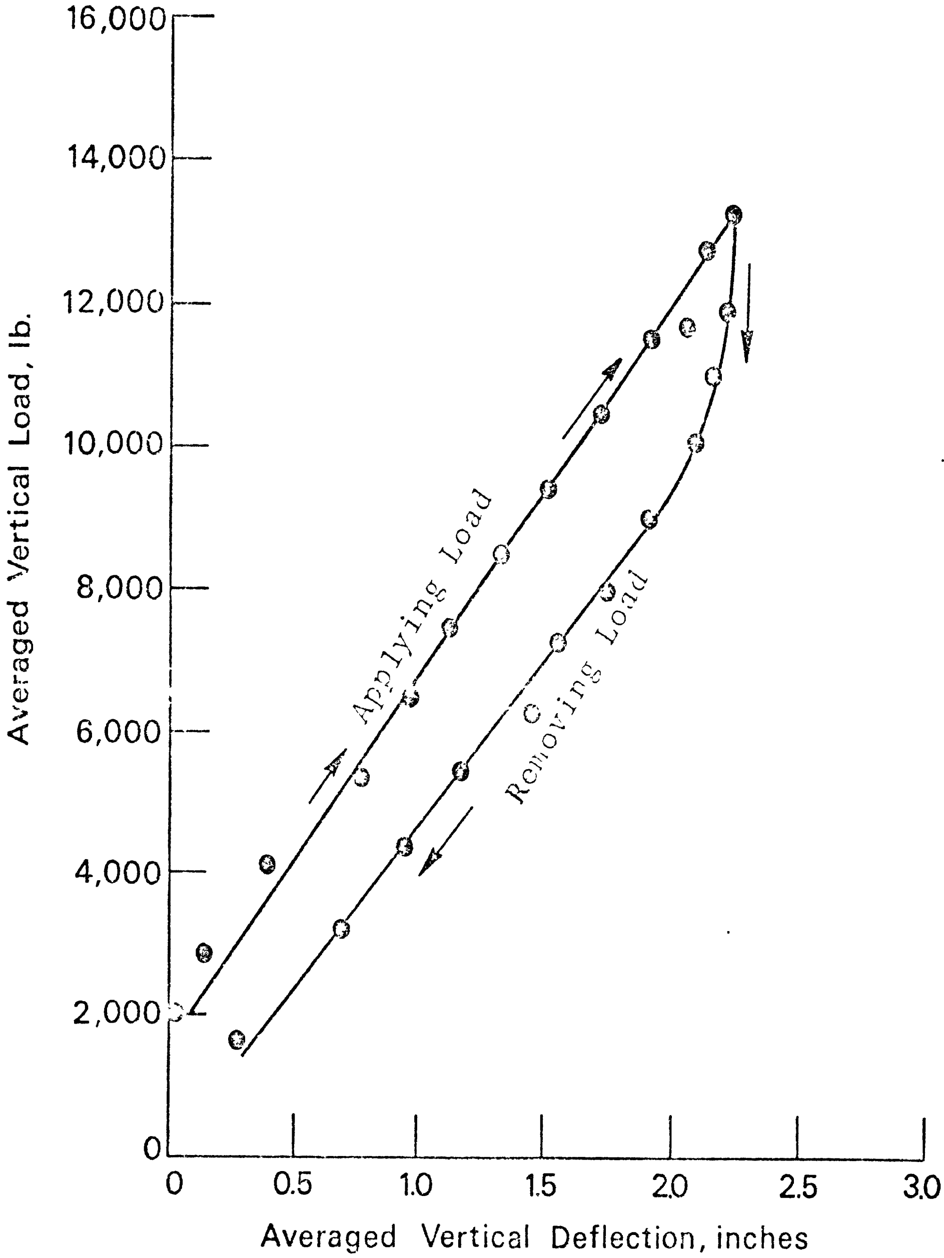

Figure 6. Averaged vertical deflection characteristics of a four spring suspension.

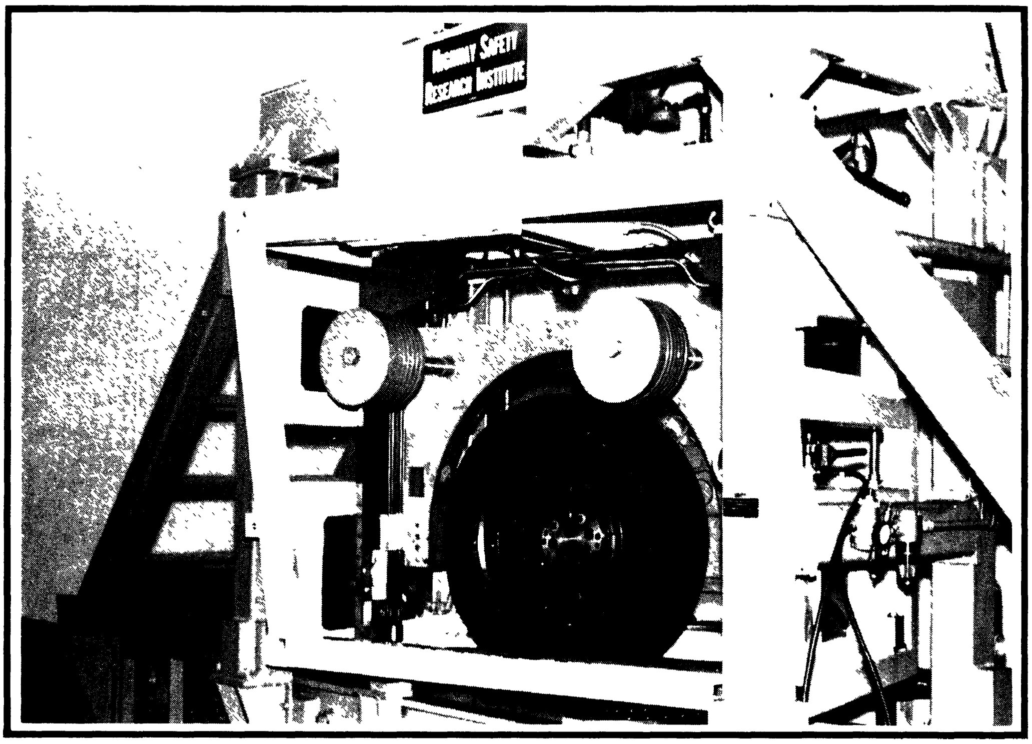

Figure 7. Dual tire assembly mounted in flat bed tire testing machine.

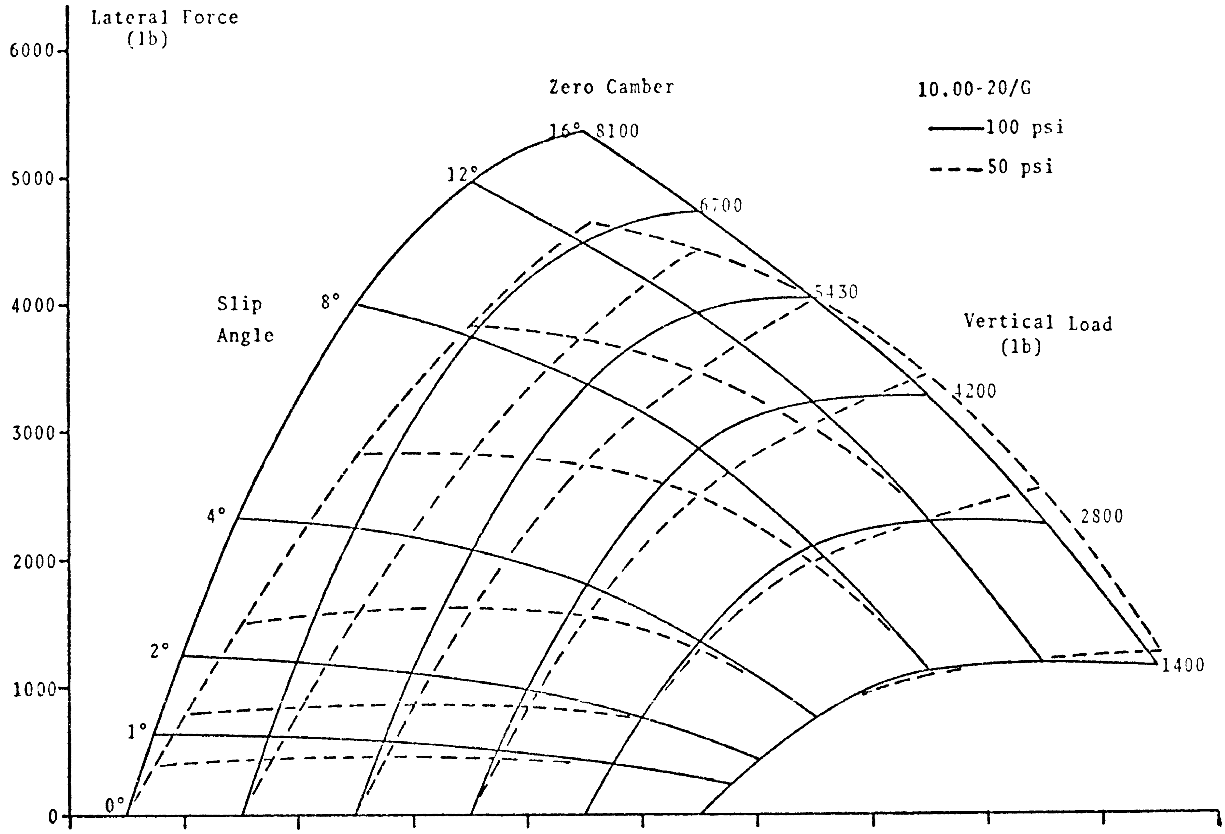

Figure 8. Lateral force versus slip angle and vertical load on 10.00-20/G tire at 100 psi and at 50 psi.

PERFORMANCE TESTS AND VALIDATION

In order to validate the computer simulations, an extensive program of vehicle testing was completed for the two vehicles shown in Figures 9 and 10. An example of the good correspondence between measured and simulated straight line braking results was given in Table 1. The Phase I braking simulation also does a very good job of predicting the point at which wheels lock up for the straight truck which was equipped with a walking beam suspension. For the tractor and trailer, both of which had four spring suspensions, the simulation predicted lock up of the leading tandem axles at slightly lower line pressures than were found experimentally. In the Phase III effort, the model of the four spring suspension has been extended to include friction at the frame-to-spring contact points; thus, the braking simulation now provides a much more accurate prediction of the level of braking at wheel lock up.

The validation of the Phase II directional response simulation is illustrated by a comparison between test and simulation results for a braking-in-a-turn maneuver. (See Figures 11 and 12.) In this maneuver, the steering wheel angle is rapidly increased to a fixed preset level, causing the vehicle to make a rapid turn. Once the vehicle has established a turn, the brakes are applied and the vehicle slows to a stop while still turning. The results shown for the truck (Figure 11) and for the tractor-trailer (Figure 12) indicate that the Phase II simulation accurately predicts both the steering response during the first part of the maneuver and the combined braking and steering response during the second part of the maneuver.

COMPUTATIONAL FEATURES

In the Phase I and II simulations wheel rotation degrees of freedom have been included for two important reasons: (1) the model of the rotating tire provides the best possible representation of the forces at the tire-road interface, and (2) control devices currently used in antilock braking systems require

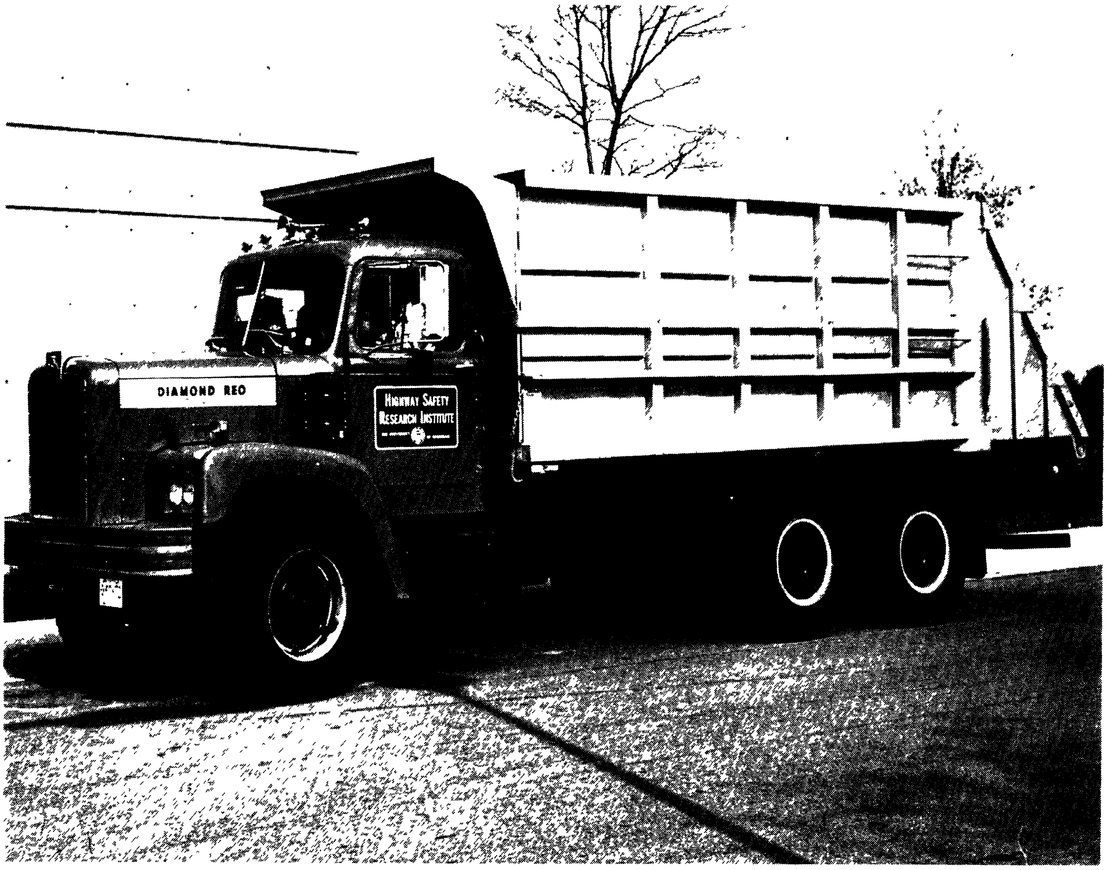

Figure 9. Straight truck used for testing.

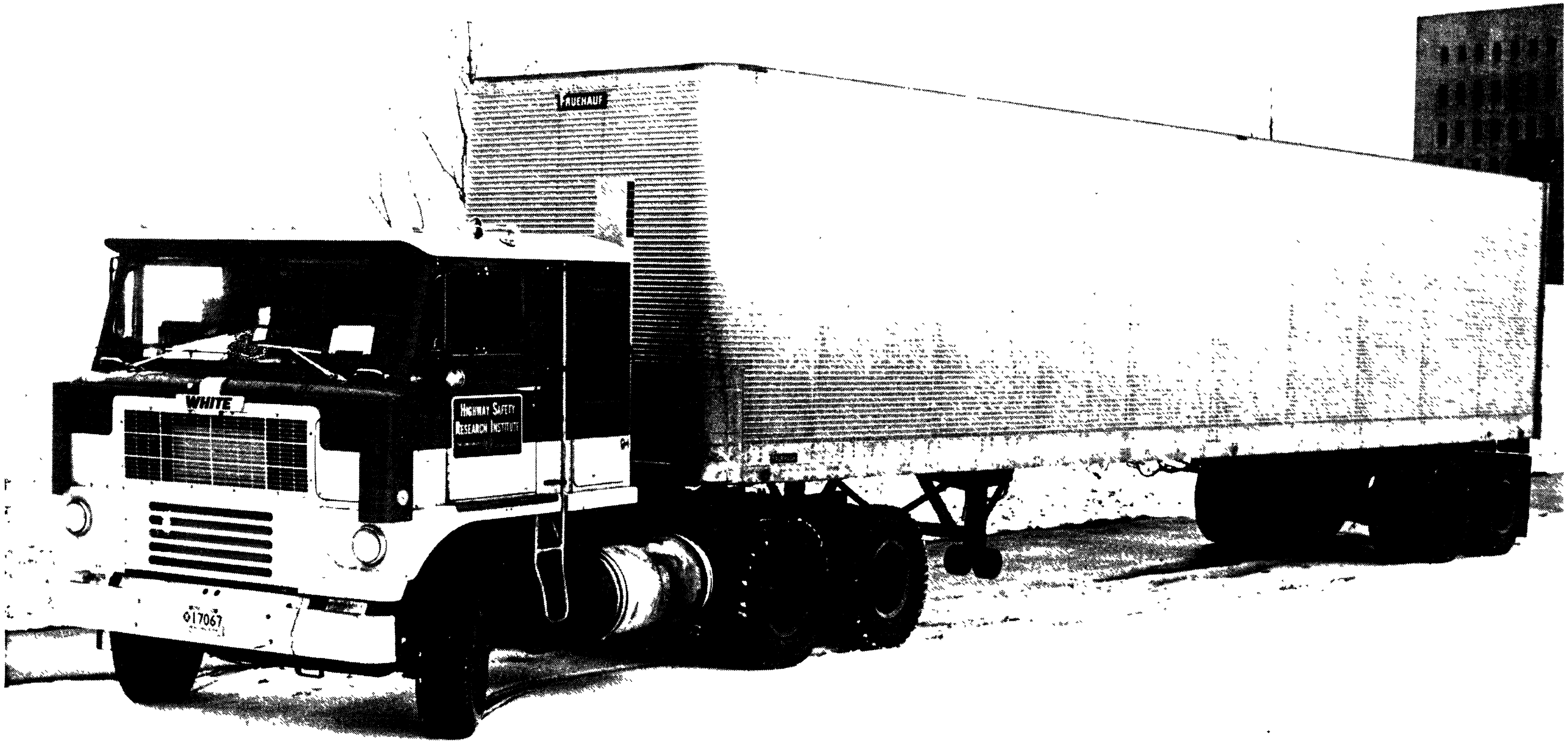

Figure 10. Articulated vehicle used for testing.

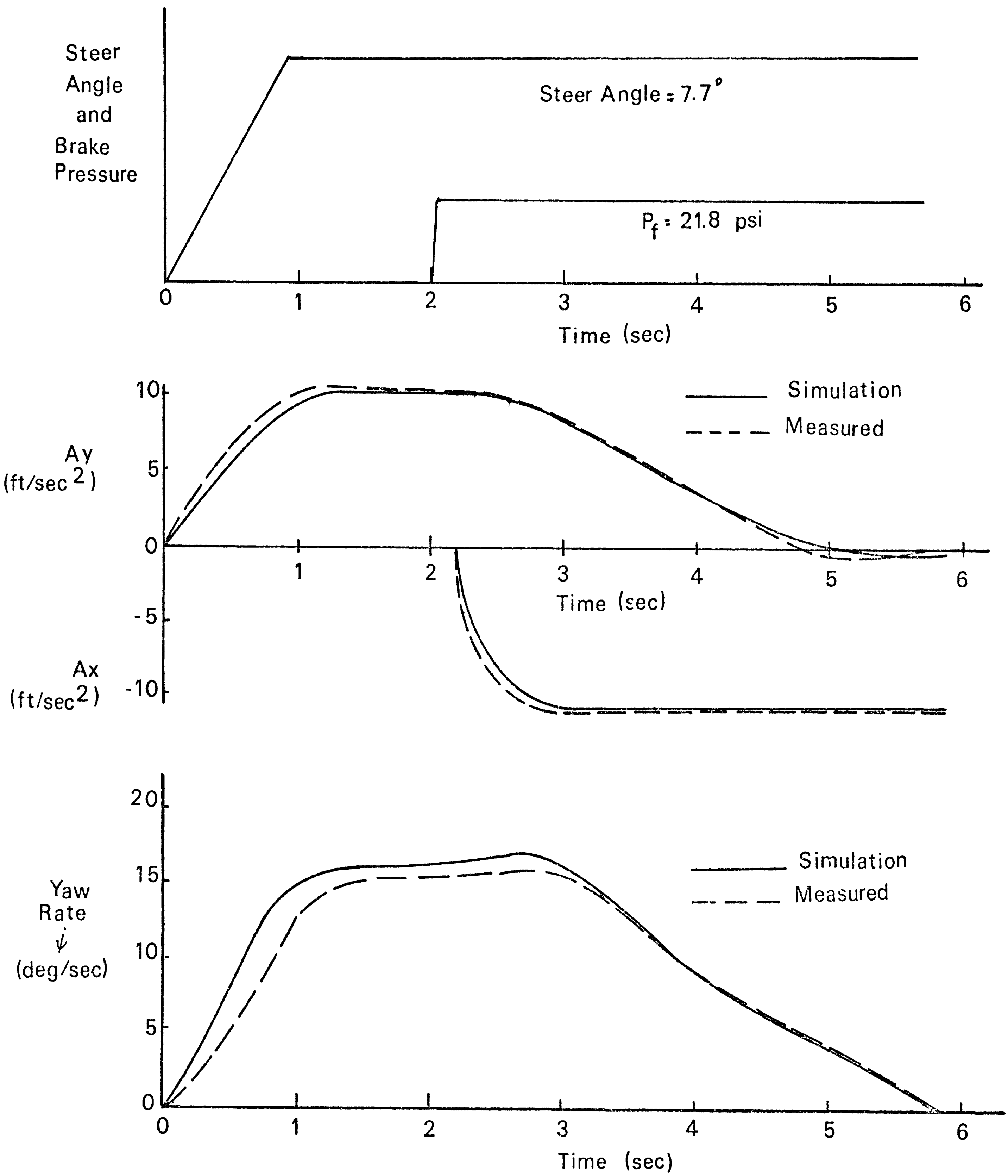

Figure 11. Validation of a braking-in-a-turn maneuver for a truck.

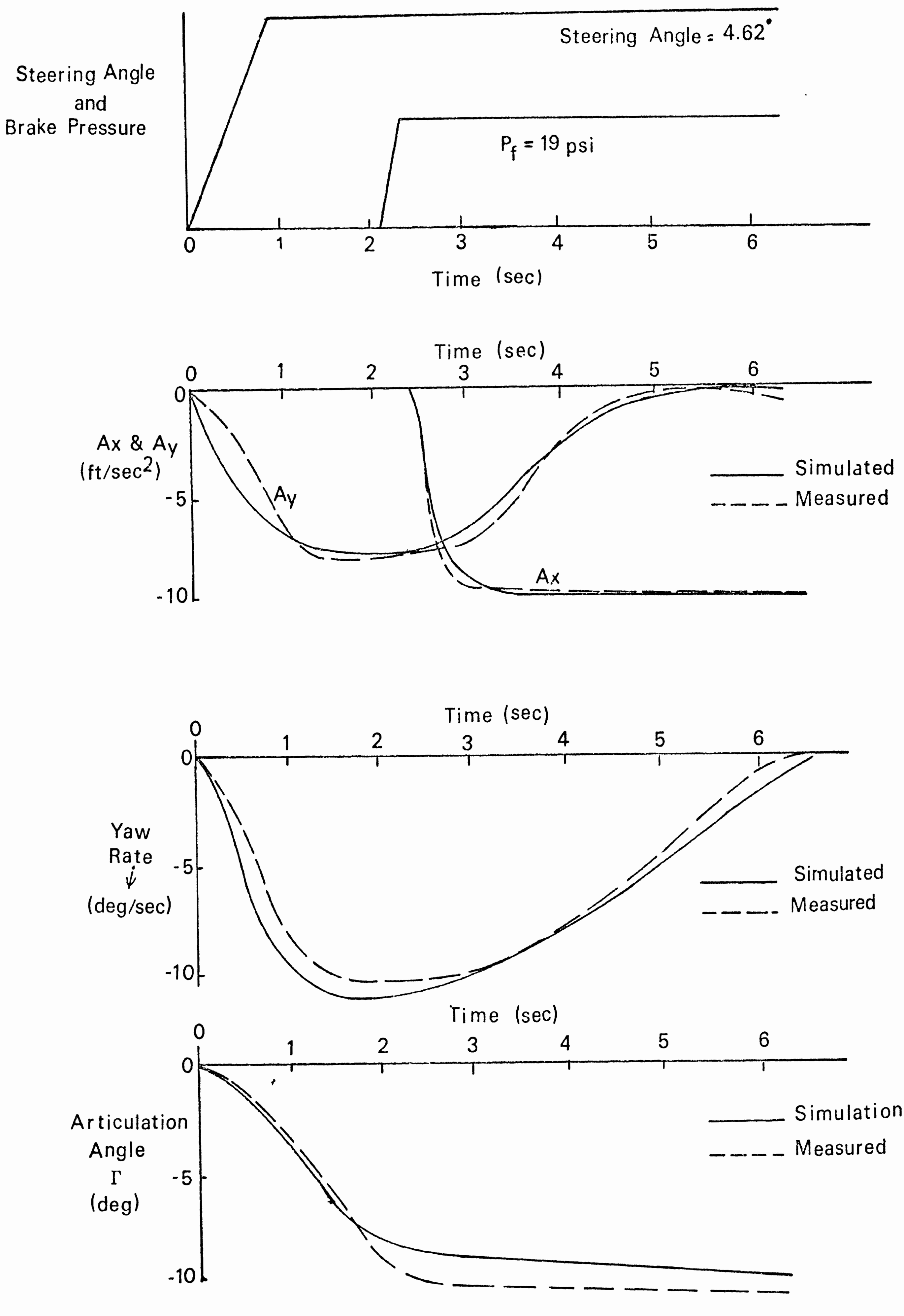

Figure 12. Validation of a braking-in-a-turn maneuver for a tractor-trailer.

information about the rotation of the wheels. Computer costs involved in digitally integrating the wheel rotational differential equations could be large enough to prevent extensive use of these programs. Consequently, a new solution technique, employing pre-programmed functions, was developed to allow the use of a rotational wheel model without a large increase in computer costs [12].

Previous articulated vehicle simulations [8, 9] have employed a rigid fifth wheel coupling constraint of one form or another. In the new program, developed at HSRI, there is no geometric constraint at the fifth wheel. There is rather a force and moment constraint in which tractor and trailer are subject to equal and opposite forces and moments dependent on the difference in the fifth wheel position and orientation as measured on the tractor and the semitrailer. This method allows for (1) new flexibility in the fifth wheel model and (2) readily available values of fifth wheel forces and moments.

The Phase I and Phase II programs are quite inexpensive to run even though they are very large. On the Michigan Terminal System, for the Phase I program, straight truck runs can be made for about one dollar per simulated second and articulated vehicle runs cost about three dollars per simulated second. For the Phase II program, the costs are about 4.5 dollars per simulated second for the straight truck and about 8 dollars per simulated second for the articulated vehicle.

CONCLUDING REMARKS

At this point in time, a simulation tool has been developed and it is ready for use. Its value is twofold; first, it allows the user to obtain numerical answers to very complicated problems, and second (and probably more importantly), it provides the user with a means for developing a better understanding of the physical characteristics of his particular problem.

Even though these simulations were developed for trucks and tractor-trailers, there is no reason why they cannot be extended for use in the study of other vehicles such as buses, motor homes, and recreational vehicle combinations. While the models for the suspensions and brake systems may require modification, the basic equations of motion will need no changes.

It is expected that these simulations will be applied in vehicle design studies which will lead to safer, and thus more economical, motor vehicle transportation.

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