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THE YAW STABILITY OF TRACTOR-SEMITRAILERS DURING CORNERING

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FINAL TECHNICAL REPORT

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16. Abstract

The yaw stability of tractor-semitrailers in steering-only maneuvers is examined by means of computer simulation and full-scale tests. The tests included as-designed vehicles as well as vehicles modified with frame and front suspension stiffening elements installed. Results show that while tractor yaw instability can occur well below the rollover threshold for certain vehicles, modified stiffness parameters can eliminate such premature yaw instability.

Simulation study of the influence of design and operating variables on tractor yaw stability served to classify the relative importance of different suspension stiffness options, as well as tire mix, fifth wheel placement, and trailer loading practices. Results show that remarkably low levels of tractor yaw stability are possible with certain combinations of design and in-use variables.

A set of measurements of tractor-semitrailer ride vibrations is also reported as an add-on task to this study.

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This study examined the yaw stability of tractor-semitrailers under pure cornering conditions, with no braking. The particular stability mode which was of interest could be called the jackknife mode, although the divergent response which can occur is much less violent than that which occurs when tractor rear wheels are braked to the point of lockup.

The objectives of this study were to:

1) identify the maneuver conditions in which tractor-semitrailer yaw stability is challenged,

2) examine the propensity for unstable yaw behavior among different vehicle combinations,

3) identify the primary mechanisms by which vehicle design and operating variables combine to degrade yaw stability,

4) modify actual vehicles in a fashion which defeats the destabilizing mechanisms, demonstrating stable performance in full-scale tests.

This project was intended to follow up findings made in a previous NHTSA-sponsored study [1]. In that project, a long-standing myth was exposed, namely, it was discovered that the usable maneuvering range of heavy commercial vehicles was not merely constrained by the low levels of rollover threshold which are typical, but rather that a premature yaw instability could further reduce the maneuvering range by as much as 30%. Additionally, it was crudely established that the unstable yaw behavior could be largely explained through the fact that commercial vehicles possess a very heavily rear-biased distribution of suspension roll stiffness.

(Continue on additional pages)

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In this research project, the issue of tractor-trailer yaw stability, as a determinant of vehicle controllability, was studied by means of laboratory measurements and full-scale tests in the context of a generalized basis of understanding as achieved through supporting mathematical analysis and computerized simulations. Laboratory measurements concentrated upon the direct evaluation of design parameters affecting the manner in which the combination vehicle reacts the roll moment that arises during cornering. The basic questions, here, pertained to the manner in which the normal loads on the tires change when a vehicle is in a turn. A long understood axiom in passenger car design has been that the distribution of lateral load transfer among tires on fore- and aft-located axles will determine the yaw stability of the vehicle at higher maneuvering levels. On the heavy trucks and road tractors examined in this study, conventional measurements of suspension roll stiffness and roll center heights were supplemented with measurements of the torsional stiffness of the vehicle frame to permit calculation of load transfer distributions and, ultimately, yaw stability characteristics.

Two tractors and two semitrailers were measured and prepared for full-scale testing. Each vehicle combination was outfitted with an instrumentation package to provide recordings of dynamic response variables. In addition, each tractor was fitted with an additional structure and mechanical elements which provided a major rear-to-front redistribution of effective roll stiffnesses. Full-scale tests were then conducted to determine the levels of yaw stability yielded by a "baseline" and "modified" vehicle.

Through mathematical analysis, the groundwork was laid for applying certain classical presentations of passenger car yaw response to the articulated commercial vehicle, including configurations with tandemized axle sets. Computerized simulations were employed to identify the range of vehicle configurations which can exhibit yaw instability prior to reaching the rollover threshold. Additional simulation runs were devoted to exploring the extent to which roll stiffness modifications serve to eliminate the potential for such yaw instability.

In line with each of the objectives, the following results were obtained:

1) Tractor yaw stability in an articulated combination was found to be most challenged in steady-turn maneuvers at elevated levels of lateral acceleration. In transient maneuvers, the potential for an unstable yaw response is reduced as the duration of the transient is reduced. In quick transients, the lagging response of the semitrailer delays and reduces the roll moments being borne by the tractor—thus promoting a stable response.

2) Many types of tractor-semitrailer combinations exhibit a yaw instability prior to reaching their rollover thresholds. This instability occurs primarily with loaded vehicles and is degraded by the following conditions:

   a) A forward bias in the distribution of tire cornering stiffnesses. (Radial construction-front mixed with bias construction-rear or rib tread type-front and lug tread type-rear are combinations tending to degrade yaw stability.)

   b) Rearward placement of the fifth wheel coupling.

   c) High c.g. location of the trailer payload.

   d) Low roll stiffness of the trailer suspension.

   Design parameters of the tractor seen to degrade yaw stability were:

   a) Excessively rear-biased distribution in suspension roll stiffness.

   b) Torsionally compliant frame.

   c) Short wheelbase configuration.

   d) Single drive axle (rather than a tandem axle arrangement).

3) The most significant vehicle characteristic promoting yaw instability, by far, is the rear-biased distribution in suspension roll stiffness. The mechanical properties of the pneumatic tire are such that the rear-mounted tires (which typically bear the largest transfer of load during cornering) experience a greater net reduction in lateral force, thus providing for a destabilizing yaw moment to be developed.

4) Full-scale tests in this study confirmed that large increases in (a) front roll stiffness and (b) frame torsional stiffness can, indeed, eliminate the possibility of an unstable yaw response occurring below the rollover limit.

The significance of these findings lies in their potential application to the improvement of vehicle design and operating practices. Knowing that vehicle response can be improved through certain suspension modifications, it remains to be established whether such modifications can be practically implemented, given the host of other considerations which actual vehicles must satisfy.

Concerning the current state of traffic safety, it can be simply said that the potential for divergent yaw behavior occurring within the rollover limit constitutes a factor which degrades the controllability of heavy commercial vehicles. Further, it is clear that such behavior can be mitigated in the short term through the adoption of favorable operating practices and, perhaps, eliminated in the long term through the development of practicable design modifications.

TABLE OF CONTENTS

1.0 INTRODUCTION

This document reports on a research study entitled "Truck and Trailer Yaw Divergence and Rollover" which was conducted by the Highway Safety Research Institute of The University of Michigan. The study was supported by the National Highway Traffic Safety Administration of the U.S. Department of Transportation under Contract Number DOT-HS-7-01602.

It should be emphasized that this research effort addresses a specific aspect of the dynamic behavior of tractor-semitrailers, namely, the condition of pure cornering with no braking. If the overall domain of cornering responses can be usefully broken down into "normal" and "emergency" maneuvers, the subject study has addressed maneuvers of the high level variety which occur only rarely and which represent an emergency domain of operation. Particularly, it is under conditions of high lateral acceleration that a potential arises for losing control due to a "yaw instability," which phenomenon is the special interest of this study.

Although the possibility of a yaw instability is a pertinent safety issue with any class of pneumatic-tired vehicle, a recent NHTSA-sponsored project [1] has revealed that there is a special basis for concern in the case of heavy trucks and tractor-trailers. In that project, it was found that a yaw instability could arise at a maneuvering level which was well below the rollover threshold. Thus, it is possible that the usable maneuvering range of the heavy truck and tractor-trailer could be much narrower than that already-narrow range which had been previously accepted as the status quo with such vehicles. Further, it was established (in the earlier study) that certain peculiar aspects of the construction of truck frames and suspensions were the primary determinants of vehicle yaw stability in the emergency domain of operation. Whereas the previous work explored these relationships for the case of straight trucks, the present study has expanded the findings to tractor-semitrailers.

In the presently reported study, tractor-semitrailer yaw stability has been examined, primarily, in two ways, namely, through full-scale testing and through computer simulation. The selection of vehicles for full-scale testing was guided, in part, by a preliminary survey of accident data files, in which particular attention was given to incidents involving jackknife or rollover.

The subsequent testing activity sought not only to measure the behavioral properties of certain contemporary tractor-semitrailer combinations, but also to demonstrate the extent to which the yaw response of such vehicles could be stabilized through modification of suspension- and frame-stiffness characteristics. Such modifications were guided through supporting simulation activities which predicted the levels of parametric change needed to effect a stable behavior.

In order to simulate the actual vehicles to be tested, vehicle design parameters were measured in the laboratory. Additionally, existing parametric data were used to define a matrix of vehicle types, such as represent the bulk of the tractor-semitrailer population in the U.S.

Computerized calculations were then performed to identify the range of yaw stability levels which could be anticipated through modification of the test tractors. Simulations were also run to establish the extent to which combination vehicles (of the types making up the U.S. tractor-semitrailer population) exhibit a potential for unstable yaw behavior below the rollover limit.

In order to define or express yaw behavior so as to evaluate the potential for instability, it was necessary to adapt to the case of the tractor-semitrailer techniques used to diagram the yaw response of passenger cars. A mathematical basis for this adaptation was developed in this study and provides a common format for reviewing all test and simulation data.

Sections 2.0 through 6.0 of this report contain technical discussions, with conclusions and recommendations presented in Section 7.0. The report also includes seven appendices, which present, in detail, results from (a) an accident data analysis, (b) vehicle parameter measurements, and (c) test and simulation exercises.

As a supplemental task to the yaw stability study reported herein, a ride vibration measurement activity was also conducted. This exercise involved the measurement of cab and frame vibrations on heavy truck-tractors in both their bobtail configuration and in combination with a loaded semitrailer. Since this task was added by the sponsor only because of the efficiency accruing from the utilization of available vehicles and instrumentation, but was otherwise unrelated to the yaw stability investigation, it is reported separately in Section 8.0 of the technical volume.

2.0 SURVEY OF ACCIDENT DATA

Accident data gathered through the Bureau of Motor Carrier Safety (BMCS) of the U.S. Department of Transportation were analyzed to determine the relationships between tractor-semitrailer type and the frequency of involvement in jackknife- and rollover-type accidents. These data were examined so as to guide a rational selection of test vehicles for use in this study.

A computerized file of 17,291 accidents occurring with tractor-semitrailers (in 1976) was interrogated to find the jackknife and overturn involvement of the following types of vehicle combinations:

two-axle tractor pulling single-axle trailer (2/1),

two-axle tractor pulling two-axle trailer (2/2),

three-axle tractor pulling single-axle trailer (3/1), and

three-axle tractor pulling two-axle trailer (3/2).

As shown in Table 2.1, the 3/2 combination dominates the sample, accounting for 73% of all accidents, with the 2/2 combination appearing second with 11%. While Table 2.1 is provided to illustrate relative sample sizes, Tables 2.2 and 2.3 constitute individual summaries of percentage involvement in jackknife and overturn accidents by axle configuration of the combination. Concerning jackknife involvement, the 2/2 configuration appears to be significantly over-involved. Although we can hypothesize no reason for a relationship between the number of trailer axles and jackknife proclivity, per se, the higher jackknife involvement of the <u>two-axle tractor</u> does confirm observations to be presented later in this report showing decreased yaw stability of two-axle tractors.

Concerning rollover, Table 2.3 presents a mixed picture. We see the two-axle tractor rolling over most frequently when coupled to the smaller (single-axle) trailer, while the three-axle tractor rolls over with disproportionate frequency when coupled to the larger (two-axle) trailer. In general, it would be expected that two-axle trailers, capable of carrying greater load, would exhibit a higher center of gravity in the loaded condition and thus would be more involved in overturning accidents.

Table 2.1. Illustration of Jackknife and Overturn Accidents, by Axle Arrangements on Tractor-Semitrailers.

Axles Tractor/Trailer		Jackknife	Overturn	Other	No. Accidents
2/1	(Total)	35	93	1104	1232
	(Row %)	2.8	7.5	89.6	100.0
2/2	(Total)	103	76	1755	1934
	(Row %)	5.3	3.9	90.7	100.0
3/1	(Total)	5	7	153	165
	(Row %)	3.0	4.2	92.7	100.0
3/2	(Total)	433	936	11208	12577
	(Row %)	3.4	7.4	89.1	100.0
All Combin.	(Total)	622	1200	15469	17291
	(Row %)	3.6	6.9	89.5	100.0

Table 2.2. Jackknife Accidents by Axle Arrangement on the Tractor-Semitrailer Combination.

Axle Arrangement Tractor/Trailer	Number of Jackknifes	Percent of All Jackknifes	Percent Normalized by File Size for All Accident Types
2/1	35	5.6%	2.8%
2/2	103	16.6	5.3
3/1	5	0.8	3.0
3/2	433	69.6	3.4
Others	46	7.3	3.3

Table 2.3. Overturning Accidents by Axle Arrangement on the Tractor-Semitrailer Combination.

Axle Arrangement Tractor/Trailer	Number of Overturns	Percent of All Overturns	Percent Normalized by File Size for All Accident Types
2/1	93	7.8%	7.5%
2/2	76	6.3	3.9
3/1	7	0.6	4.2
3/2	936	78.0	7.4
Others	88	7.4	6.3

The data of Table 2.4 support the intuitively-reasonable notion that rollover is a phenomenon primarily associated with loaded tractor-trailers. Indeed, the incidence of loaded vehicle rollover is 13 times more frequent than rollover of the unloaded combination. The converse sensitivity is seen in the jackknife data; revealing that the unloaded combination jackknifes 4.4 times as frequently as the loaded tractor-trailer. This result can be interpreted to support the contention that jackknife is primarily an anomaly prevailing when tractor rear axles are unloaded and overbraked to the point of lockup. [The thrust of this overall study, however, was to determine the manner in which yaw instabilities, such as jackknife, develop during cornering maneuvers, which instabilities are most aggravated in the fully-loaded case. It remains to be determined, then, whether overturn accidents are being properly treated by the accident investigation and classification schemes which exist or whether certain rollovers of loaded vehicles were not, in fact, precipitated by a tractor yaw divergency. Perhaps certain rollover accidents might have been better assigned the "jackknife" classification, from a causality point of view. Clearly, however, it is difficult to investigate for the presence of a precipitating yaw divergency which may have culminated in rollover.]

Other data from the BMCS file are presented in Appendix I, illustrating by axle arrangements the accident involvement of vehicles characterized by

- -trailer type
- -trailer cargo
- -tractor manufacturer

One result of the data pertaining to trailer type is that, for the most numerous (3/2) combination, van trailers exhibit 1.7 overturning accidents for every jackknife while flat-bed trailers exhibit 4.8 overturning accidents for every jackknife. This result is somewhat surprising since flat-bed trailers are not typically loaded to the c.g. heights which are common with van trailers.

Table 2.4. Illustration of Jackknife and Overturn Accident Rates for Unloaded (24 through 30K GCW) and Loaded (60 through 80K GCW) Tractor-Semitrailers.

GCW		Jackknife	Overturn	Other	Total No. of Accidents
24-30K (Unloaded)	No. of Unloaded Accidents	182	23	2519	2724
	% of All Unloaded Accidents	6.7%	0.8%	92.5%	
60-80K (Loaded)	No. of Loaded Accidents	113	850	6568	7531
	% of All Loaded Accidents	1.5%	11.3%	87.2%	

In examining the accident data trends observed in the BMCS file, it is useful to reflect on the reporting and exposure biases which can tend to create apparent "findings" which may not apply to the U.S. vehicle population as a whole. Insofar as the BMCS reporting system excludes certain trucking operations within so-called "commercial zones," the file is biased by whatever nonrepresentativeness exists in either the operating conditions or vehicle configurations of the excluded trucks. Since the BMCS holds jurisdiction only in interstate truck transport, the data file may also represent an unusually concentrated exposure to rural (long-haul) rather than urban truck operations. While such concerns as these may have a bearing on the distribution of accident type among the vehicle configurations, 2/1, 3/2, and so on, they are not expected to have biased other results such as the relationship between overturn and jackknife accidents for loaded versus unloaded trailers.

3.0 EVALUATION OF VEHICLE PARAMETERS

In order to permit computerized simulation of the mechanical response of specific vehicles, a number of design parameters must be evaluated. The parameters of interest include vehicle geometry, inertial properties, the kinematic and compliance properties of suspension and steering systems, frame compliance and tire properties. While the measurement of many geometric parameters requires equipment no more sophisticated than a tape measure for evaluation of wheelbase, track width, fifth wheel placement, etc., inertial, suspension, and frame properties are not so easily obtained and require facilities specifically designed for these types of measurements.

3.1 Special Experiments to Obtain Chassis Parameters

To obtain the vertical and longitudinal location of the center of gravity, as well as the moment of inertia in pitch, the Pitch Plane Inertial Properties Facility, shown in Figure 3.1, was used in this study. Height of center of gravity is determined on this facility through a static pitch experiment in which the restoring moment about a knife-edge pivot is measured to deduce the lever arm at which the known vehicle weight is acting. By subtracting the swing's tare component from the pitch moment and by locating the knife-edge height with respect to a "ground plane," the height of the total vehicle c.g. above the ground is obtained. The height of the sprung mass center is then determined through removal of the estimated contribution afforded by unsprung masses located at the height of the wheel spin centers.

Pitch moment of inertia is determined by treating the free-swinging system as a compound pendulum, whose mass center has been located in the previous experiment. The natural period of the pendulum is used to determine the principal moment of inertia in pitch. Details of these measurement methods are given in Reference [3]. In this study, yaw and roll moments of inertia were estimated, given measurements which had been made on similar vehicles and which could be scaled using knowledge of c.g. height and pitch moment of inertia.

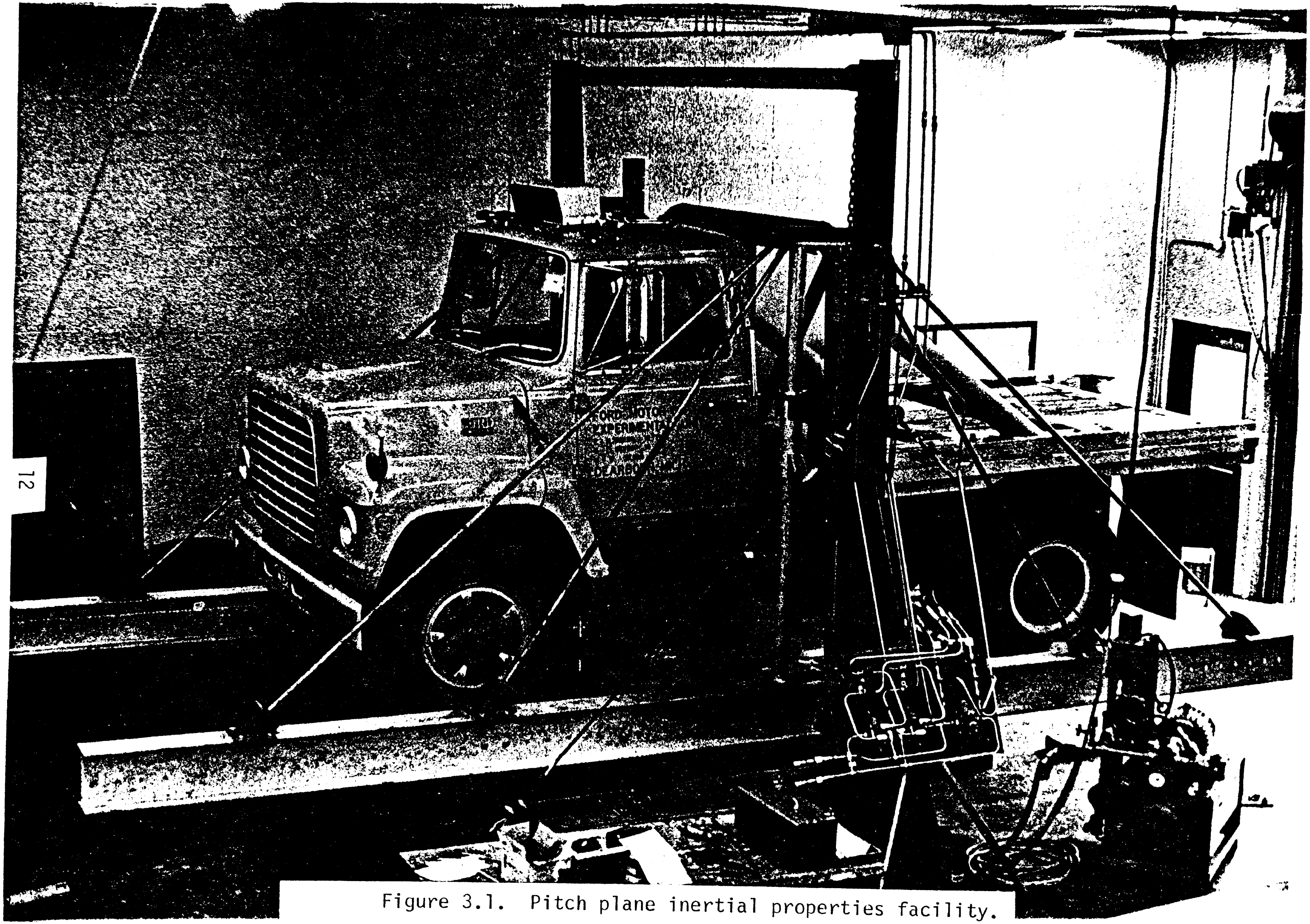


Figure 3.1. Pitch plane inertial properties facility.

Suspension and steering system properties were measured on HSRI's suspension parameter measurement facility illustrated in Figure 3.2. This facility consists of two vertically-oriented hydraulic cylinders which are used to apply load to the tires on the axle being measured. For this experiment, the vehicle frame is clamped firmly to ground. The cylinders are used to exercise the suspension in bounce and/or roll while transducers measure tire loads, vertical wheel motion and steer angles. These tests yield key suspension parameters such as spring rate, roll rate, roll steer, and coulomb friction measures. The facility also has the capability of applying aligning moments and brake force to the tire contact patch to measure compliance in the steering or suspension systems. A general discussion of the measurement of truck suspension parameters can be found in Reference [4].

Torsional compliance (about the x axis) of vehicle frames was considered to have a major influence on yaw stability and was therefore measured on each vehicle in this project. Figure 3.3 shows the method which was used to measure the torsional compliance of both tractors and semitrailers. A rolling moment is applied at one end of the vehicle and an inclinometer is used to measure the relative torsional deflections at several points along the length of the frame. This type of measurement provides data not only concerning the vehicle's total torsional stiffness, but also the distribution of this stiffness along the vehicle's length.

Listings of all vehicle parameters, both measured and estimated, are given in Appendix II.

3.2 Measurement of Tire Traction Parameters

The shear forces developed between a vehicle's tires and the pavement are the most important quantities governing the vehicle's directional control. The forces and moments developed by the tires vary in a nonlinear fashion with load and slip angle, thus requiring that tire behavior be measured over a broad range of conditions. Cornering force characteristics of the tires which were used in full-scale vehicle

Figure 3.2. Suspension parameter measurement facility.

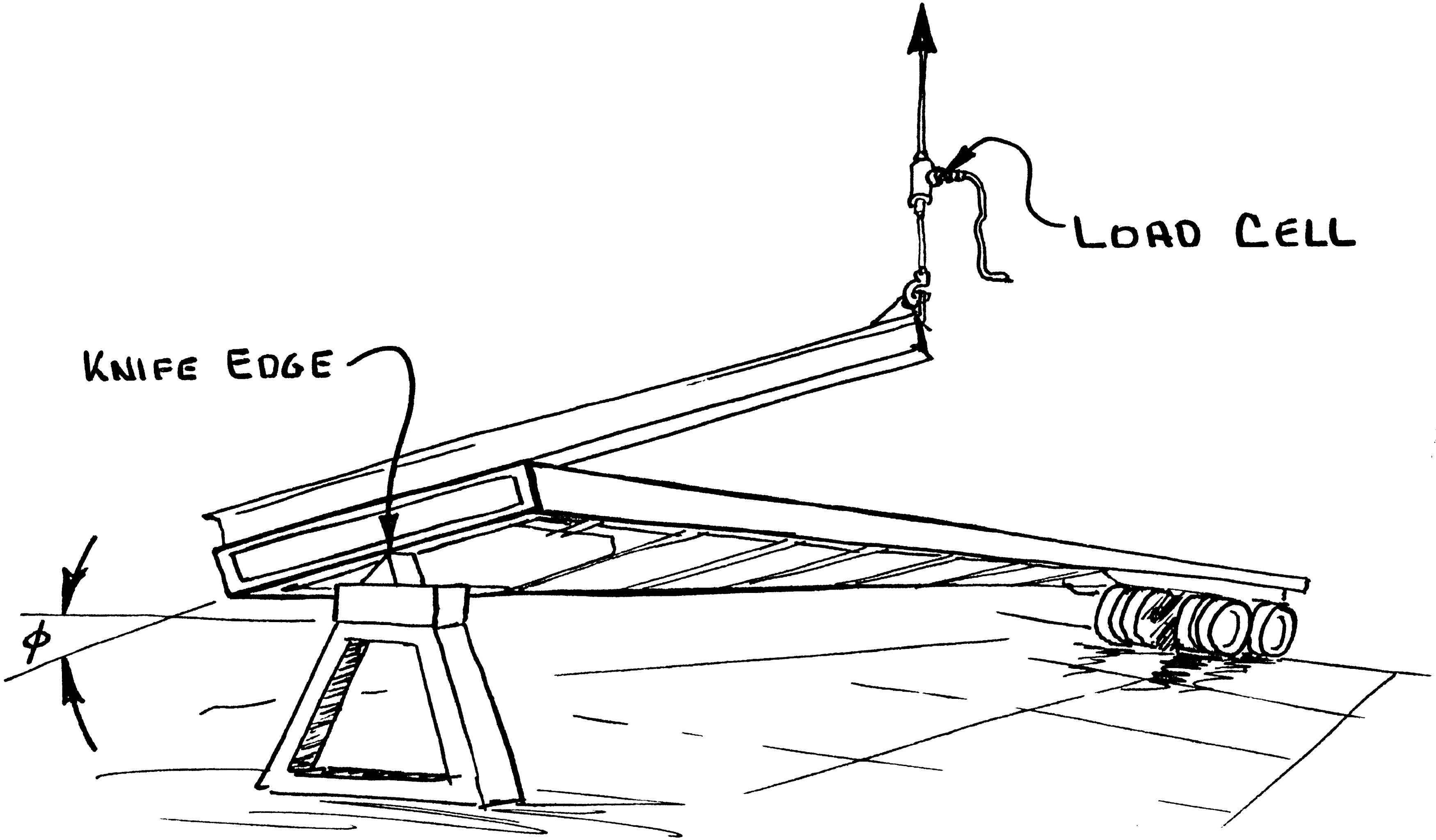


Figure 3.3. Measurement of frame torsional compliance.

tests were measured on the HSRI Flat-Bed Tire Tester shown in Figure 3.4. In the operation of this machine, the tire is mounted on a gimballed assembly which establishes the slip angle and camber angle orientations relative to the roadway which moves at a speed of 1.5 mph under the tire. Using the flat-bed machine, data relating cornering force and aligning torque to load and slip angle were generated for use in the simulation model.

Since test-induced tire wear had been found previously [1,5] to alter shear force properties, additional tire traction measurements were conducted to determine the effect of wear on the mechanical properties of the specific truck tires to be used in the full-scale tests of this study.

The HSRI Mobile Truck Tire Dynamometer, shown in Figure 3.5, was employed in a set of repetitive tests designed to establish whether test-induced wear would create a non-representative tire performance condition, perhaps necessitating frequent tire changing.

The slip angle control system of the mobile dynamometer was programmed for these experiments to provide a slip history which approximated that obtained from computerized simulations of a three-axle tractor/van semitrailer in trapezoidal steer maneuvers. Shown in Figure 3.6 is the slip angle waveform which was used in repetitive tests at a speed of 45 mph, with vertical load equal to 5000 lbs. Shown in Figure 3.7 are results of an 80-run sequence on each of two 10.00-20 tires, revealing that no significant trend in the level of maximum normalized side force, μ_y, prevailed with increasing test runs and thus test-induced wear. Accordingly, it was determined that no special procedures were warranted for changing tires due to an anomalous test-wear sensitivity.

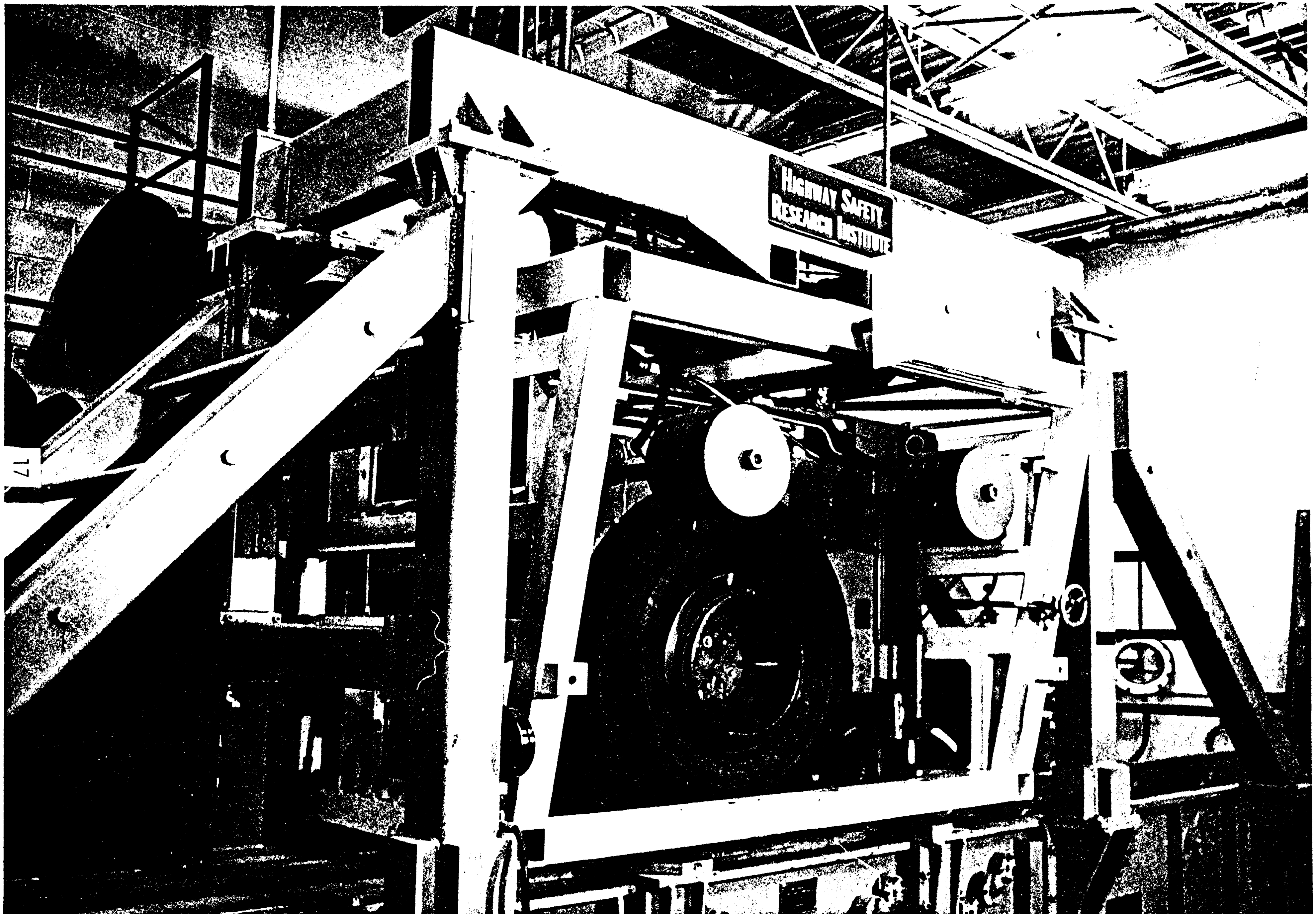


Figure 3.4. HSRI flat-bed tire tester.

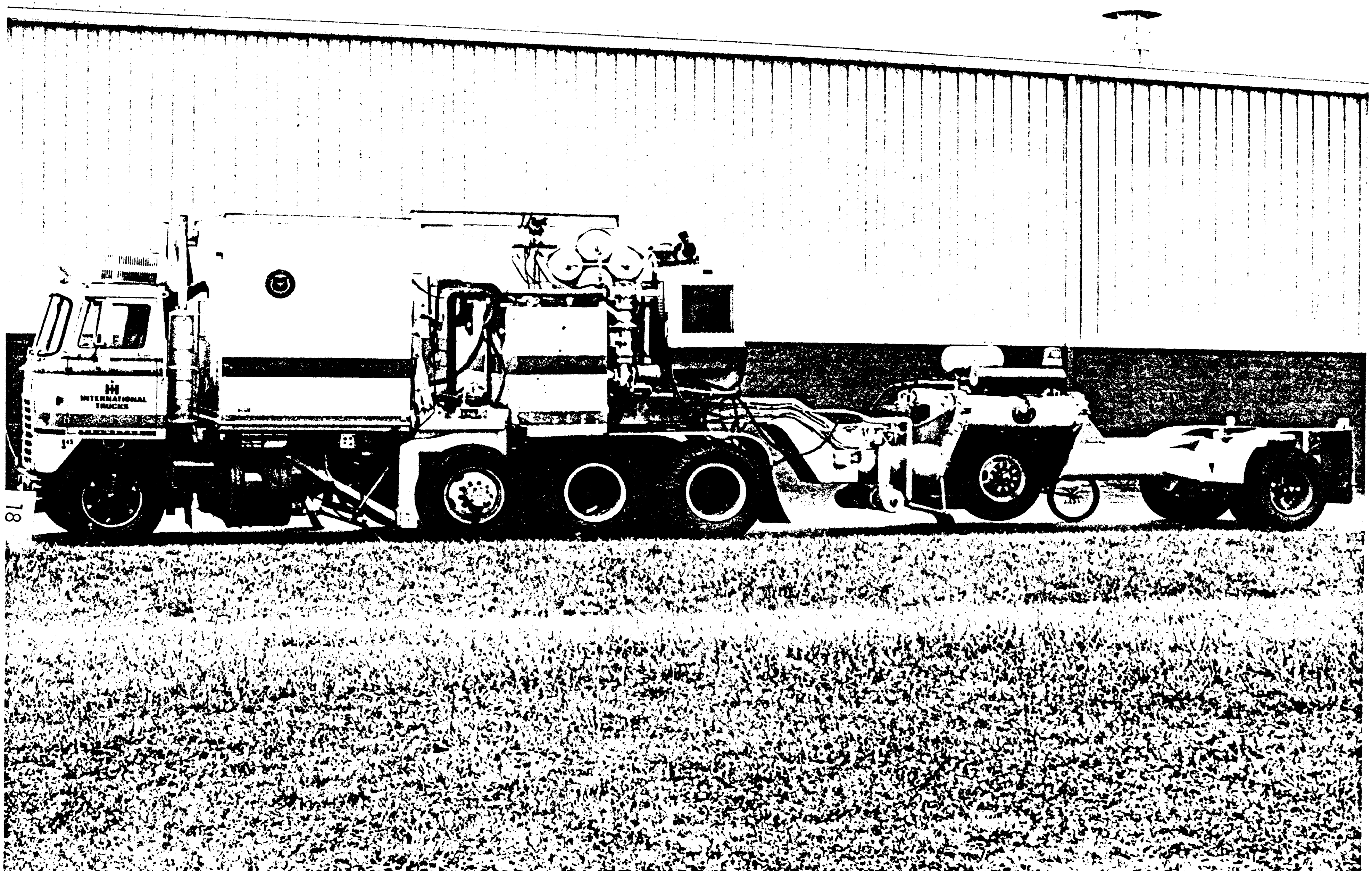


Figure 3.5. HSRI mobile truck tire dynamometer.

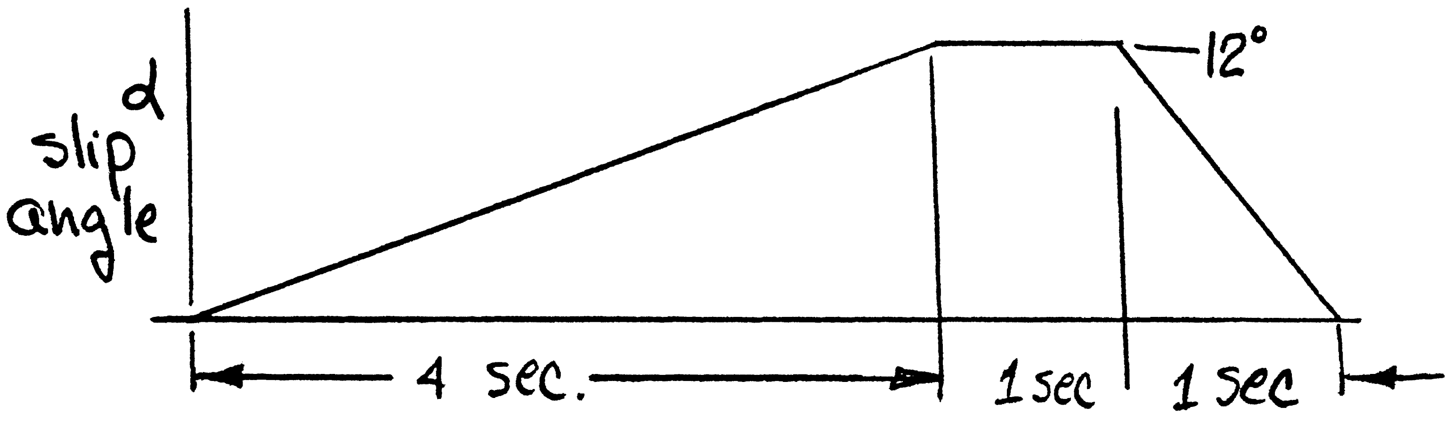


Figure 3.6. Slip angle history employed on mobile dynamometer to examine tread-wear sensitivities of test tires.

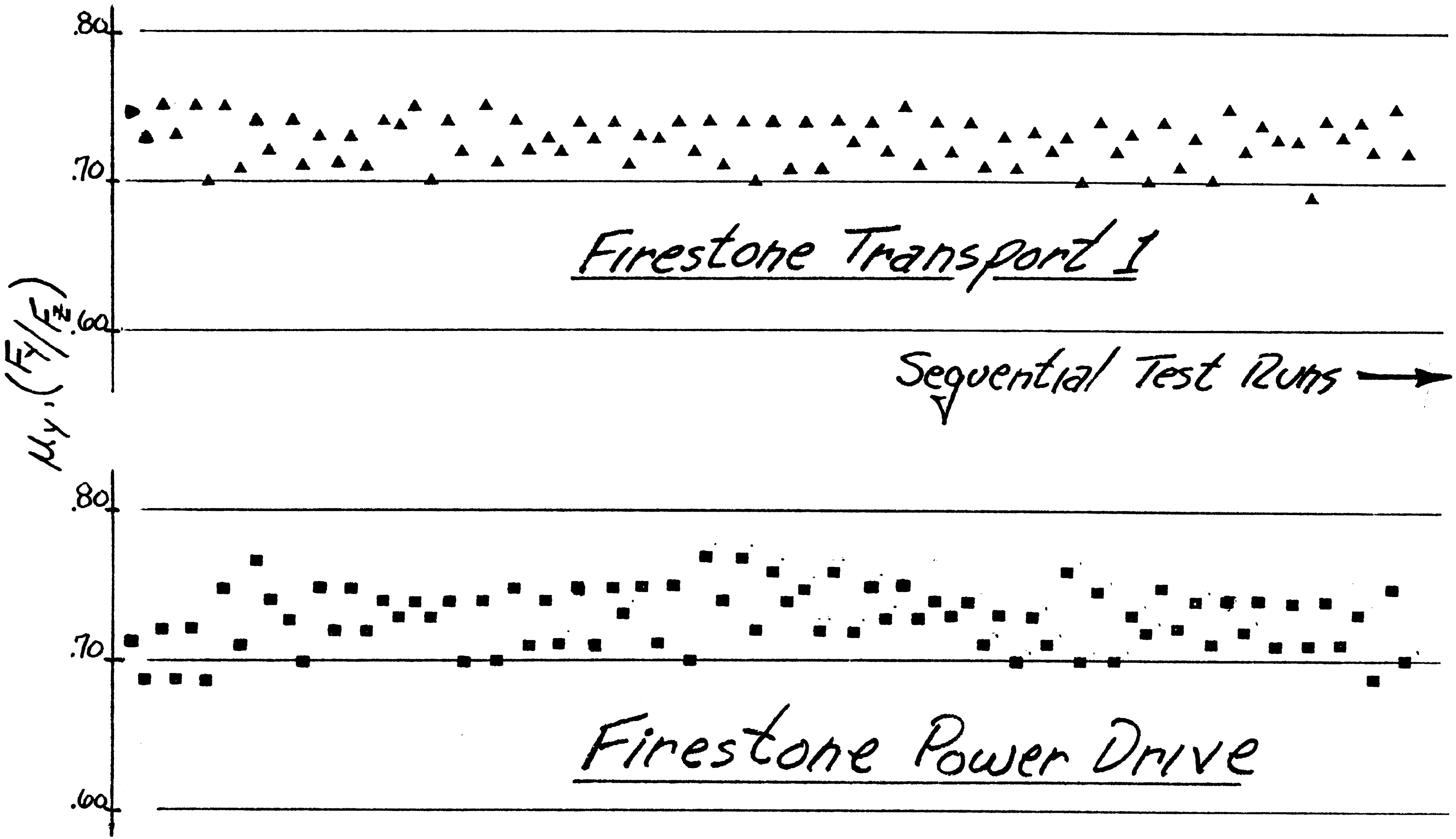


Figure 3.7. Repetitive runs, examining test-wear sensitivity in the μ_y response of two heavy truck tires, at a slip angle of 12°, given the slip angle test cycle shown in Figure 3.6.

4.0 INTRODUCTORY CONSIDERATIONS ON THE CHARACTERIZATION OF TRACTOR-SEMITRAILER YAW STABILITY

Appendix IV presents a general discussion and analysis of the yaw response of tractor-semitrailers to steering inputs. Although the interested reader is referred to that appendix for an in-depth treatment of the vehicle mechanics issues, the following brief discussion is provided as an introduction to the data analysis techniques which will be used in presented test and simulation results in the text of this report.

The primary concern of the test and simulation exercises has been the examination of tractor-semitrailer yaw responses so as to conclude whether an unstable yaw response is present. Analysis has shown (see Appendix IV) that the mode of instability which is of essential importance simply pertains to the yaw rate response of the tractor to steering. Although various types of limit yaw response can be identified for tractor-semitrailers over the range of operating velocities, the only truly divergent configurations of the combination vehicle require that the tractor itself be in a yaw divergent state.

Inspection of test and simulation data has shown, however, that tractor yaw instability often prevails with a relatively low rate of divergence, such that the existence, or not, of an instability (which is permitted to proceed for only a brief time interval) is not easily determined. In order to provide a means of clearly characterizing the presence of an unstable operating condition, it was necessary to adapt for tractor-semitrailers a technique developed to display the yaw behavior of the passenger car. This display, called the "handling diagram," was broadly developed by Pacejka [2] to facilitate the study of steady-state yaw response to steering, as described by the expression:

$$\delta = \frac{\ell}{R} + \frac{UV^2}{gR} \qquad (4.1)$$

where

δ = front-wheel steer angle, rad.

ℓ = wheelbase, ft.

R = path radius, ft.

U = understeer gradient, rad/g

V = velocity, ft/sec

g = acceleration of gravity, ft/sec^2

The handling diagram is simply a plot of this relationship, namely, the term (V^2/gR) is taken as the dependent variable while (δ-ℓ/R) is taken as the independent variable and the slope of the curve is described by the inverse understeer gradient, 1/U. If we portray the yaw behavior of a passenger car on this diagram, presuming its tires to behave linearly, we see that U is a constant over the entire range of centripetal acceleration, V^2/gR, such that diagrams of the understeer, neutral steer, or oversteer cases show up as illustrated in Figure 4.1.

For real vehicles with nonlinear tire properties, we find that the understeer gradient is not a constant, but rather changes as a function of lateral acceleration. Thus the typical two-axle truck, for example, yields a handling diagram such as shown in Figure 4.2, in which a transition from understeer to oversteer occurs (that is, from a negative slope to a positive slope), as lateral acceleration level increases. At some lateral acceleration level, a, the vehicle becomes neutral steer and thereafter becomes increasingly oversteer until the rollover threshold is reached. Cases of this type are of primary interest in this study since we wish to examine the conditions under which a yaw instability can actually be established at a lateral acceleration level which is below the rollover threshold. For any vehicle which exhibits an oversteer characteristic in some region of its handling curve, there does exist a critical velocity, V_c, above which the vehicle is yaw-unstable. Further, the handling diagram presents a convenient means for testing a given response curve so as to evaluate the conditions for such an instability. Critical velocity slopes can be superimposed upon the handling diagram using the relationship, $\text{Slope}_{crit} = V^2_{crit}/g\ell$, which derives simply

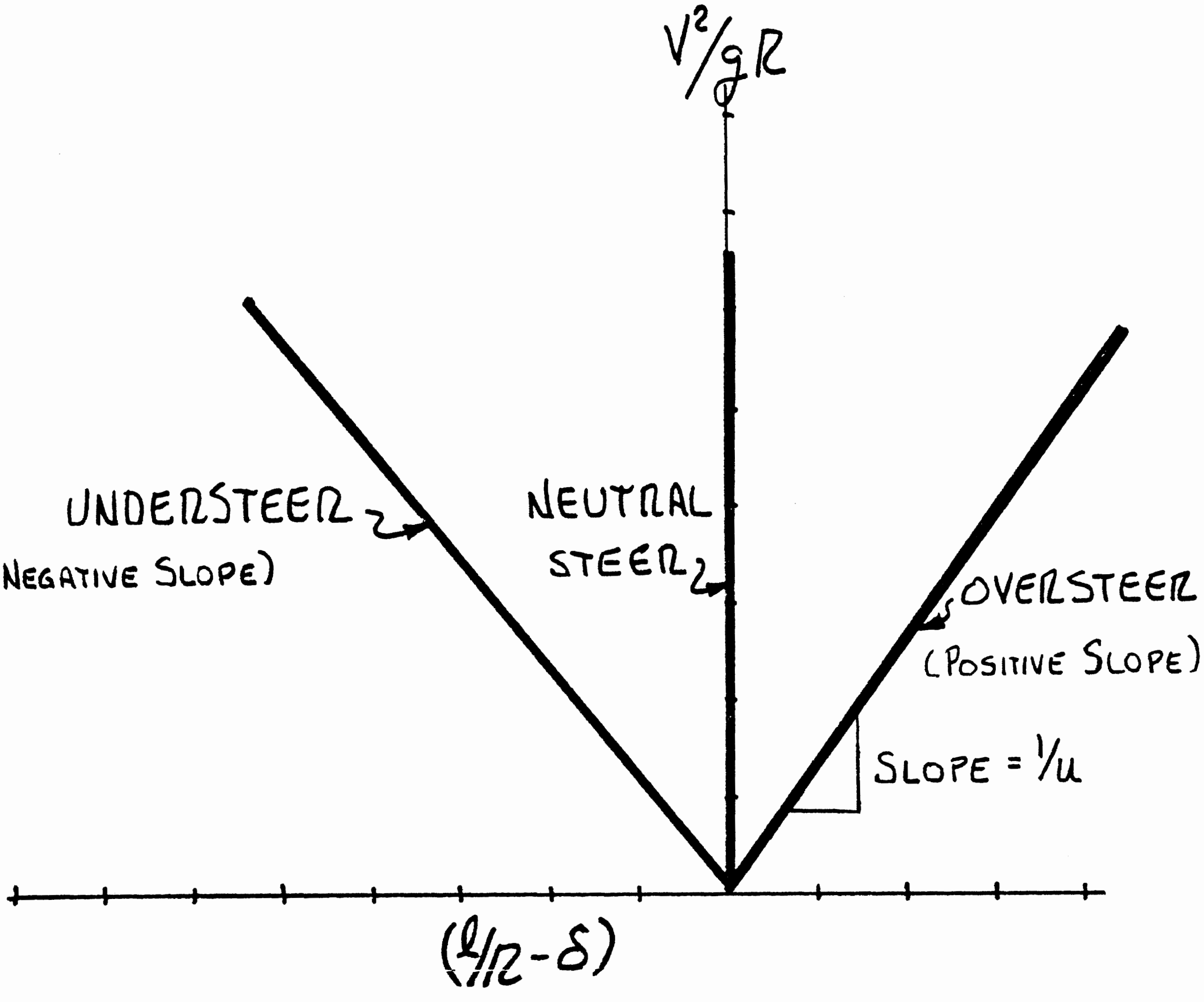


Figure 4.1. Example handling diagram showing three possible characteristics for a passenger car with linear tire properties.

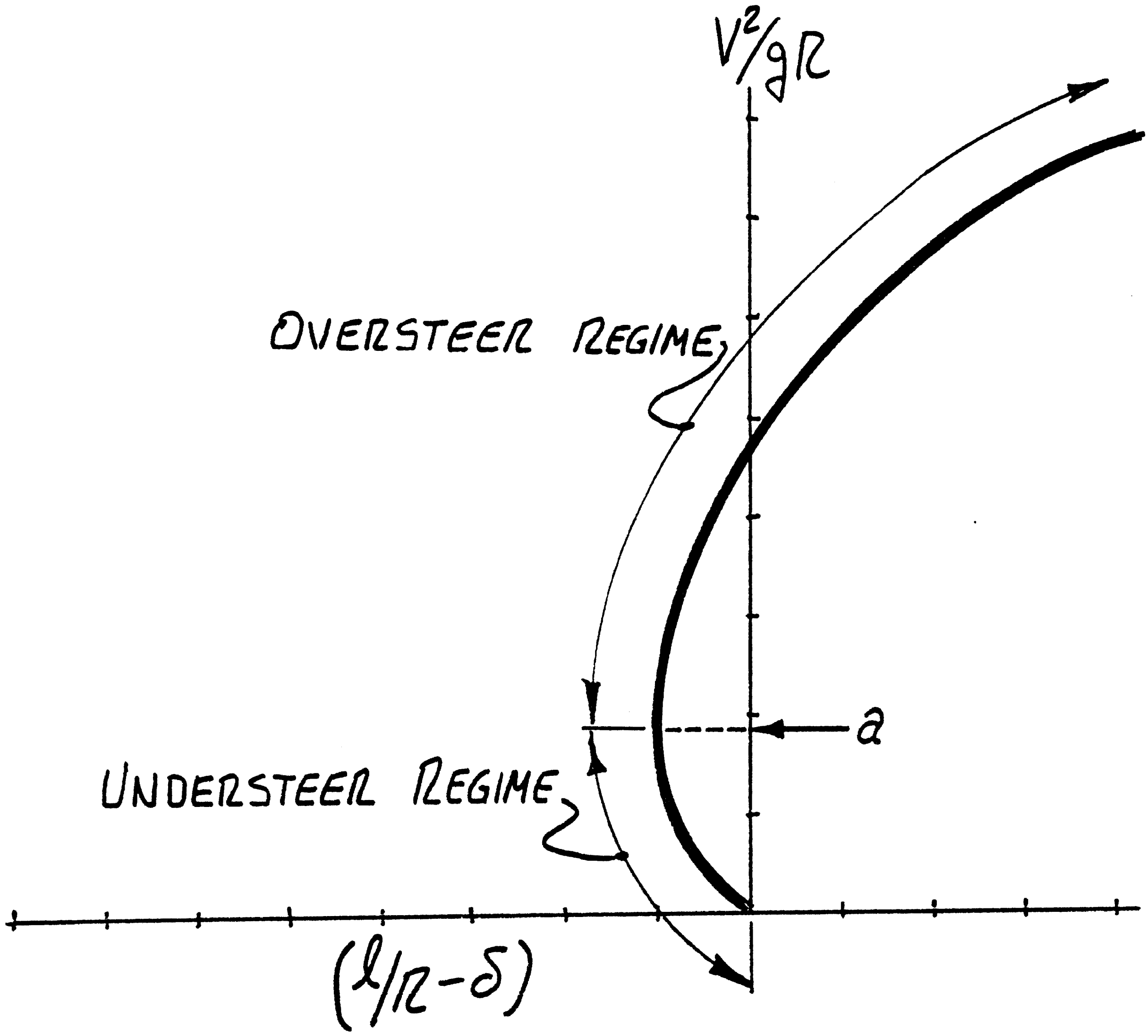


Figure 4.2. Example handling diagram, typical of heavy trucks, showing a transition from understeer to oversteer behavior at an intermediate level of lateral acceleration.

from the case when Equation (4.1) is solved for the limit condition in which path curvature gain $(\frac{1/R}{\delta})$ becomes infinite. Lines of critical slope, or critical velocity, are shown in Figure 4.3, together with an example handling curve, illustrating that a decreasing critical speed condition will produce yaw instability as the lateral acceleration level increases.

Note, again, that an oversteer polarity slope is needed for an instability to be possible. A nonlinear, but continuously understeer, response, such as shown in Figure 4.4, for example, will produce a yaw-stable behavior right up to the rollover limit.

It should also be noted that the abscissa variable on the handling diagram, $(\delta-\ell/R)$, implies that the diagram becomes peculiarly scaled to the individual vehicle's wheelbase. For the case of three-axle tractors, an analysis in Appendix IV shows that the relevant wheelbase is measured from the front axle to the mid-position of the rear tandem pair of axles. Additionally, when tandem axles are present, it is no longer possible to represent the vehicle by way of a single, unique handling curve. Rather, a family of handling curves is needed to represent the yaw response properties of a tandem-axle tractor, with one curve needed for each operating velocity, as shown in Figure 4.5. Since each curve is only valid, then, for the individual velocity, V_i, the vehicle's response can only be tested for stability against the corresponding critical velocity slope, with $V_{crit} = V_i$.

As shown for the example case of the 60-mph curve, the slope corresponding to the (V_{crit} = 60 mph) condition occurs at a lateral acceleration level of $A_{y_{c,60}}$. Thus the represented vehicle would be said to possess a yaw stability threshold at a lateral acceleration level of $A_{y_{c,60}}$ when operating in a steady turn at 60 mph. Of course, no realizable yaw stability threshold would apply in this example if the rollover threshold were encountered before the 60-mph handling curve had arrived at a slope equal to the critical value,

$$\left.\frac{V_{crit}^2}{g\ell}\right|_{V_{crit}=60}$$

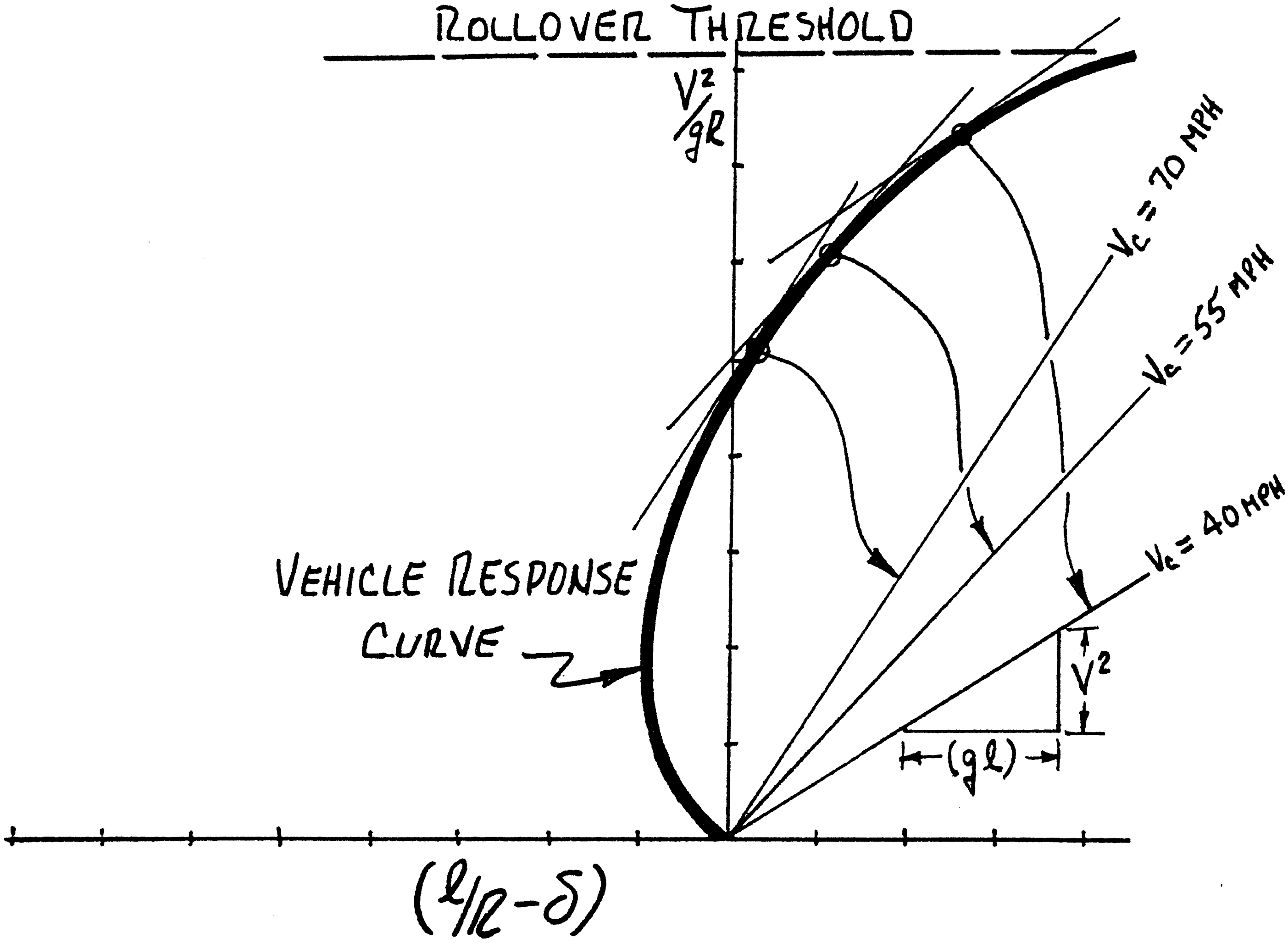


Figure 4.3. Example handling diagram showing three operating points at which the local slope has been evaluated and used to identify an equal-slope ray defining the critical velocity at that operating point.

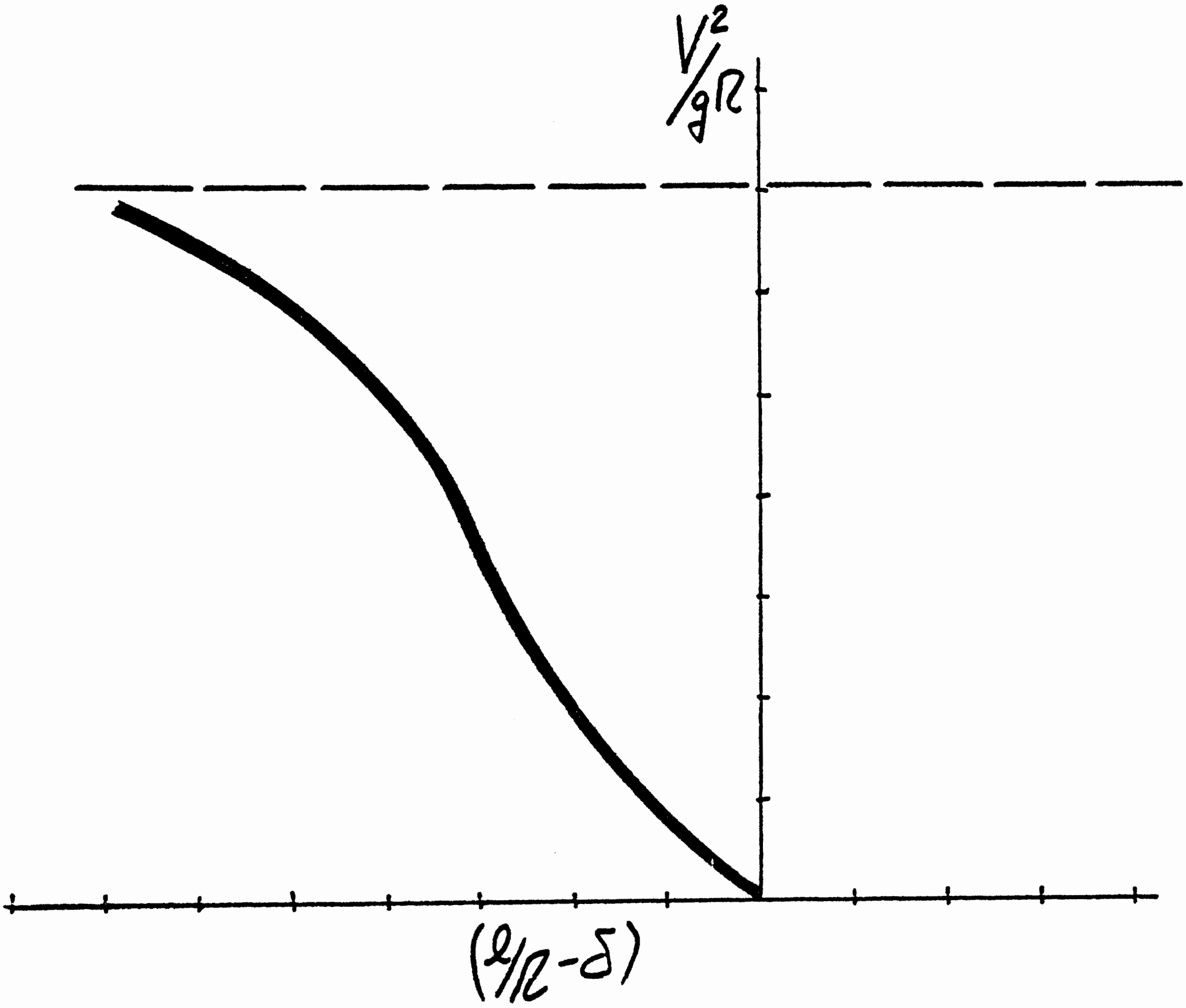


Figure 4.4. Example handling diagram showing a nonlinear but continuously-understeer behavior for which no yaw instability is possible.

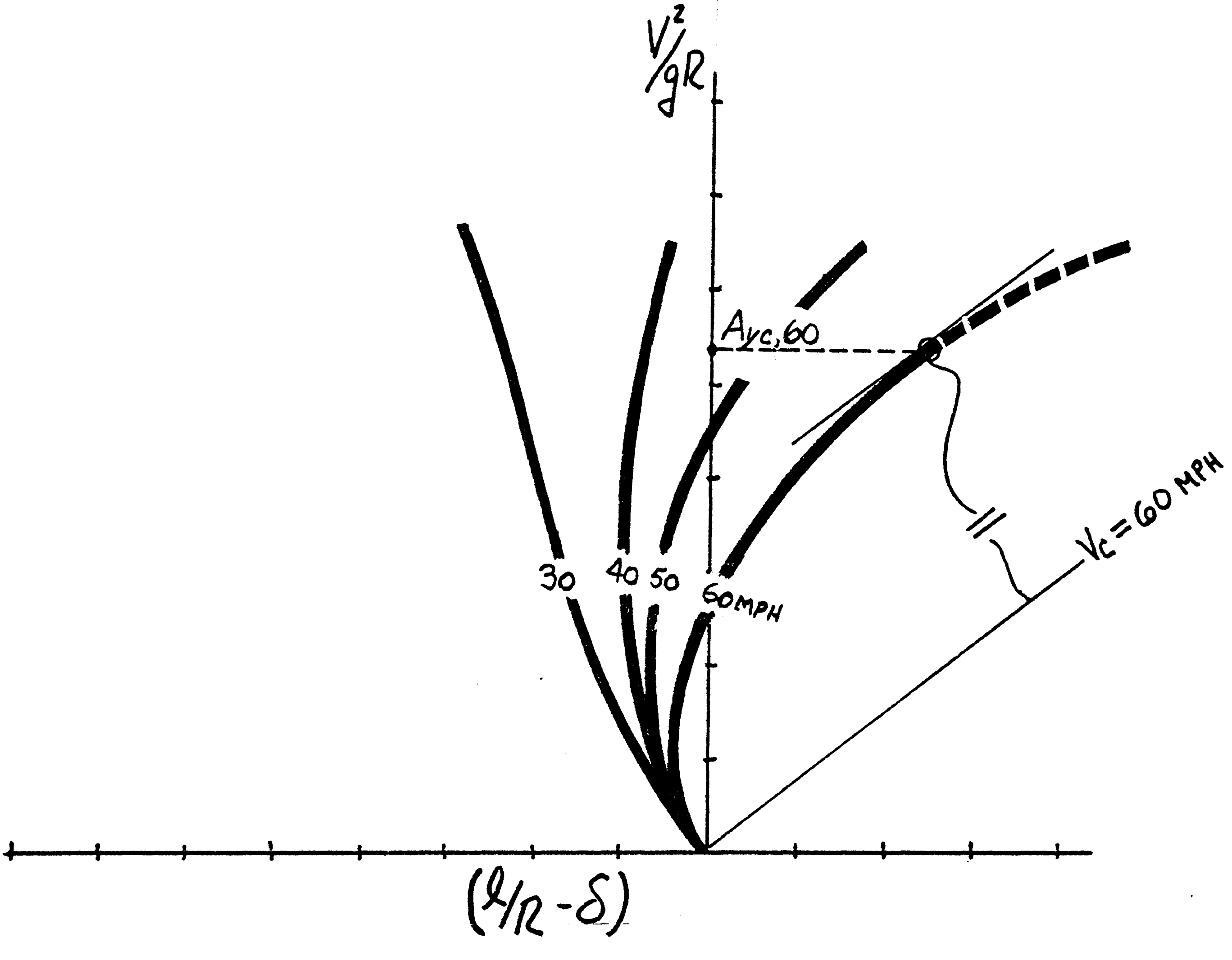


Figure 4.5. Example handling diagram showing a full family of curves for a tandem-axle tractor.

Also, it should be noted that the handling curve is valid, at a given velocity, only up to the A_{y_c} level of lateral acceleration. Beyond that level, the system is yaw unstable such that Equation (4.1) becomes meaningless. The yaw stability threshold numeric, A_{y_c}, will be employed in Section 6.0 as the primary measure for discriminating among the influences of various design and operating parameters on yaw stability.

In all handling diagrams which follow, presenting both test and simulation data, the abscissa variable $(r\ell/V - \delta)$, will be substituted for the previously-described form $(\ell/R - \delta)$. This substitution, while simply expressing an identity in the steady-state case, properly accounts for the quasi-steady conditions which actually prevail in full-scale tests and simulated maneuvers. For maneuvers in which some transient behavior is present, the variable, $1/R$, contains a component deriving from the instantaneous vehicle sideslip rate, $\dot{\beta}$, viz.,

$$1/R = r + \dot{\beta}/V$$

Inspection of the basic equations upon which the handling diagram is based reveals that the $\dot{\beta}$ content in $1/R$ would lead to an erroneous determination of vehicle understeer level, and thus stability, in the higher level maneuvers in which $\dot{\beta}$ is seen to be significant. Thus the term $(r\ell/V - \delta)$ permits interpretation of the handling diagram for quasi-steady maneuvers such as will be treated here.

5.0 FULL-SCALE TEST PROGRAM

A set of full-scale experiments was conducted in the study to evaluate the validity of computerized simulations and to provide empirical support to the examination of the influence of roll stiffness distributions on tractor yaw stability. Two tractors and two semitrailers were tested with the data being subsequently reduced to the handling diagram format. In the following subsections, the test vehicles, as well as the related vehicle-preparation steps and test procedures, are described. Test data are reviewed for the baseline tractor-semitrailer configurations and contrasted with data from the cases in which those vehicles were modified so as to alter roll stiffness distribution.

5.1 Test Vehicle Descriptions

Two tractors and two trailers were selected for testing on the basis of their representativeness of common configurations, as well as in consideration of design parameters expected to influence yaw stability. Both selected tractors were cab-over-engine units with similar wheelbases. One tractor possessed a single drive axle (i.e., a two-axle vehicle), whereas the other tractor was configured with tandem driving axles (a three-axle vehicle). The selected van-type and flat-bed trailer represented the most popular configurations of semitrailers and also represent extremes of the range of frame torsional compliance.

The selected two-axle tractor, shown in Figure 5.1, was a Ford W9000 COE with a sleeper compartment and a sliding (i.e., adjustable) fifth wheel. The front and rear suspensions consisted of steel leaf springs with a helper leaf on the rear axle. All test tires were 10.00-20 Firestone Transport 1, a rib-tread bias-ply tire. A complete vehicle description is provided in the parameter listings of Appendix II.

The second power unit was an International Harvester CO 4000, three-axle tractor, shown in Figure 5.2. This vehicle had a leaf-spring front suspension with an air-spring rear suspension. The rear suspension incorporated different levels of roll stiffness on each of the

Figure 5.1. Ford two-axle tractor with Fruehuaf van semitrailer.

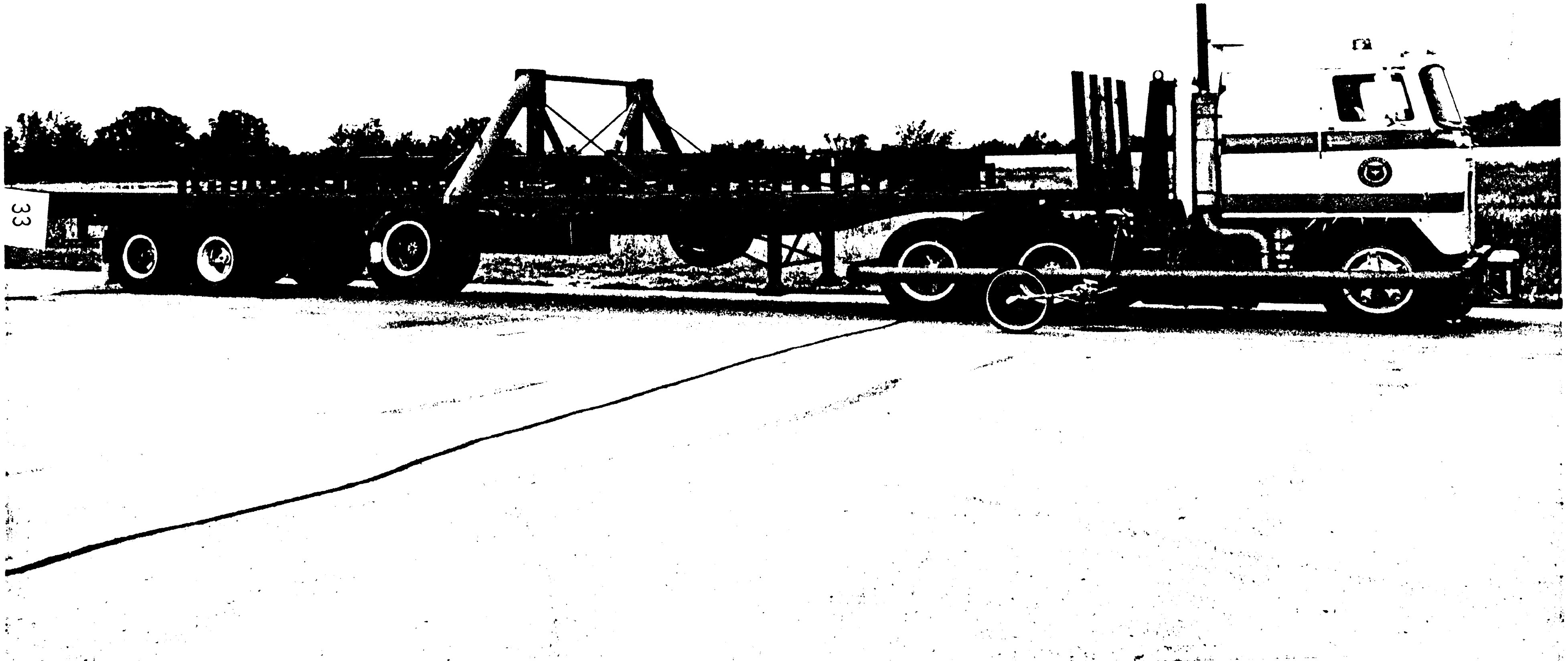


Figure 5.2. IHC three-axle tractor with Trailmobile flat-bed semitrailer.

tandem axles, with one axle providing approximately 85% of the total rear roll stiffness. This vehicle was also configured with a sleeper cab and a sliding fifth wheel. All test tires were 10.00-22 Firestone Transport 1.

The selected van-type trailer was the Fruehauf 45-foot van shown in Figure 5.1. The suspension on this trailer was a four-spring tandem incorporating taper leaf springs. The trailer was outfitted with Fruehauf 10.00-20 tires. This type of trailer provides a frame structure which is very high in torsional stiffness.

The 45-foot Trailmobile flat-bed trailer, shown in Figure 5.2, was also tested. In this case, the vehicle frame is highly compliant in torsion, providing a strong contrast to the van trailer. The flat-bed trailer possessed a four-spring tandem suspension incorporating multi-leaf springs and was equipped with 10.00-20 Firestone Transport 1 tires.

5.2 Test Methodology

Full-scale tests were conducted for the sole purpose of examining vehicle yaw stability in the vicinity of the yaw or roll stability limits. Accordingly, a tailored test methodology was designed, portions of which are peculiar to the narrow focus of interest in these experiments.

5.2.1 Test Site. All tests were conducted on the Vehicle Dynamics Area at the Chrysler Proving Ground in Chelsea, Michigan. This facility consists of an oval track with an 800-foot square skid pad between the straight-aways, as shown in Figure 5.3. The test surface is smooth asphalt with an ASTM (dry) skid number of approximately 86. All tests were run on dry pavement.

5.2.2 Instrumentation. The test vehicles were equipped with transducers to provide records of variables relevant to the analysis of yaw stability. The transducer signals were processed on-board to scale the signals for recording on FM tape. Selected signals were also displayed (in real time) on a strip chart recorder to provide immediate feedback on transducer operation while also guiding subsequent selection of steer input levels.

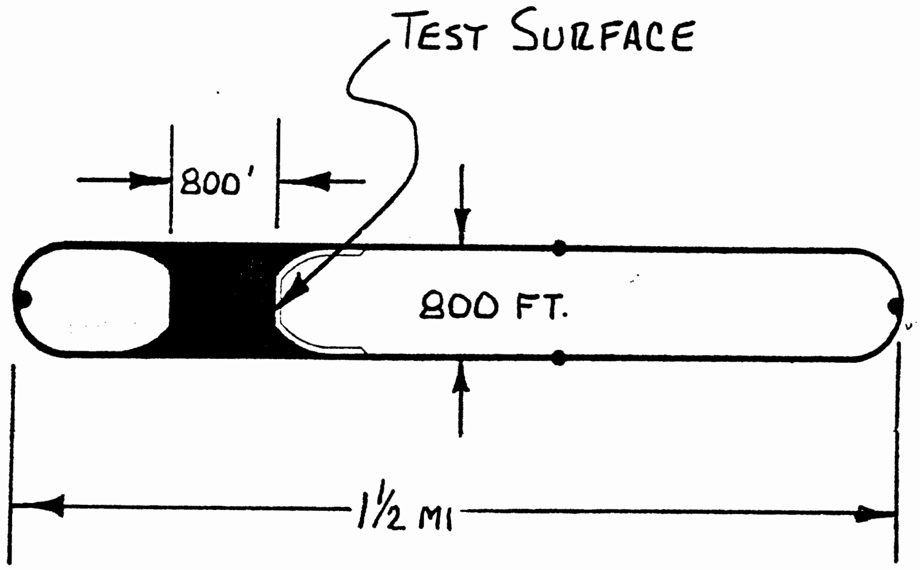


Figure 5.3. Vehicle Dynamics Test Area at Chrysler Corporation's Chelsea Proving Grounds.

The following variables were recorded in this test program:

-vehicle velocity, V

-steering-wheel angle, δ_{sw}

-tractor lateral acceleration, A_{y1}

-tractor yaw rate, r_1

-tractor roll angle, ϕ

-tractor-trailer articulation angle, Γ

-trailer lateral acceleration, A_{y2}

-trailer yaw rate, r_2

Transducers used to measure these variables are listed in Table 5.1.

The signal processing unit used to scale the transducer signals also provided calibration voltages for the various transducers, based on the results of bench tests.

5.2.3 Vehicle Preparation. To attain the desired test conditions while assuring test safety, extensive vehicle preparation was necessary. This preparation involved the installation of safety equipment, the instrumentation system, and modification hardware, together with the mounting of the payload weights.

Considerable attention was given to safety equipment in this project since it involved testing the vehicle at, or near, the limits of yaw and roll stability. Each tractor was equipped with a rollover protection bar, seat belts, and shoulder harnesses. Heavy chains were attached between tractor and semitrailer to limit the articulation angle to a maximum of 15° and thus prevent jackknifing. Trailers were also equipped with outriggers (as shown previously in Figures 5.1 and 5.2) to prevent rollover. The outriggers were constructed of heavy pipe sections with a telescoping lower strut to provide height adjustment. The outriggers were additionally restrained by chains attached fore and aft to the trailer structure. The rollover restraint is provided by the ground contact of dual truck tires which, on contact, introduce minimum extraneous disturbances to the vehicle.

Table 5.1. Transducers

Variable, Symbol	Transducer, Range
Vehicle Velocity, V	Fifth Wheel with DC Tachometer, 0-60 mph
Steering-Wheel Angle, δ_{SW}	Rotary Potentiometer ±360°
Tractor Lateral Acceleration, Yaw Rate, Roll Angle	Stabilized Platform Unit, Servo Accelerometer, ±1 g Rate Gyro, ±60 deg/sec Gimbal Angle Potentiometer, ±15°
Trailer Lateral Acceleration	Servo Accelerometer, ±1 g
Trailer Yaw Rate	Gas Rate Sensor, ±60%
Articulation Angle	Rotary Potentiometer with Parallelogram Linkage, ±15°

Two pieces of hardware were constructed to provide a means for altering the roll moment distribution on each test tractor. These devices comprised a structure for increasing the torsional stiffness of the vehicle's frame and an auxiliary roll stiffener for the front suspension. The frame- and front roll-stiffening devices were designed on the basis of simulation results obtained earlier (see Section 6.3) which showed that dramatic improvements in tractor yaw stability could be obtained through forward-biasing the tractor's distribution of lateral load transfer (or, roll moment reactions).

The frame stiffening structure diagrammed in Figure 5.4 consisted of two pieces of six-inch Schedule 40 pipe which were positioned outside the existing vehicle envelope, parallel to the frame rails. The pipes were welded to channels running laterally and providing attachments to the vehicle frame. A plate connected to the tractor's roll-protective structure was also welded to the frame-stiffening tubes so as to provide a "sandwiching" of the fifth wheel coupling on both sides of its mounting to the tractor frame.

To test the vehicle without any influence of the frame stiffener, the stiffening structure is made torsionally free by removing all front fastening bolts, except one bolt which serves as a pivot, thus permitting the tractor frame to twist without twisting the stiffener. When the additional stiffness is desired, the stiffener is secured to the tractor frame by eight additional bolts, thus rigidly coupling the frame and stiffener in torsion. The frame stiffener was designed to introduce an additional 80,000 in-lb/deg to the torsional stiffness of the vehicle frame.

Figure 5.5 illustrates the mounting of the auxiliary front roll spring to the vehicle frame and the front axle. Basically, the device constitutes the simple U-shaped element which is referred to in automotive parlance as a "sway bar." In this application, the two legs of the "U" can be considered rigid with all of the effective compliance being built into the straight torsional section. The "torquing" of this section as a result of vehicle roll motion affords relative rotations

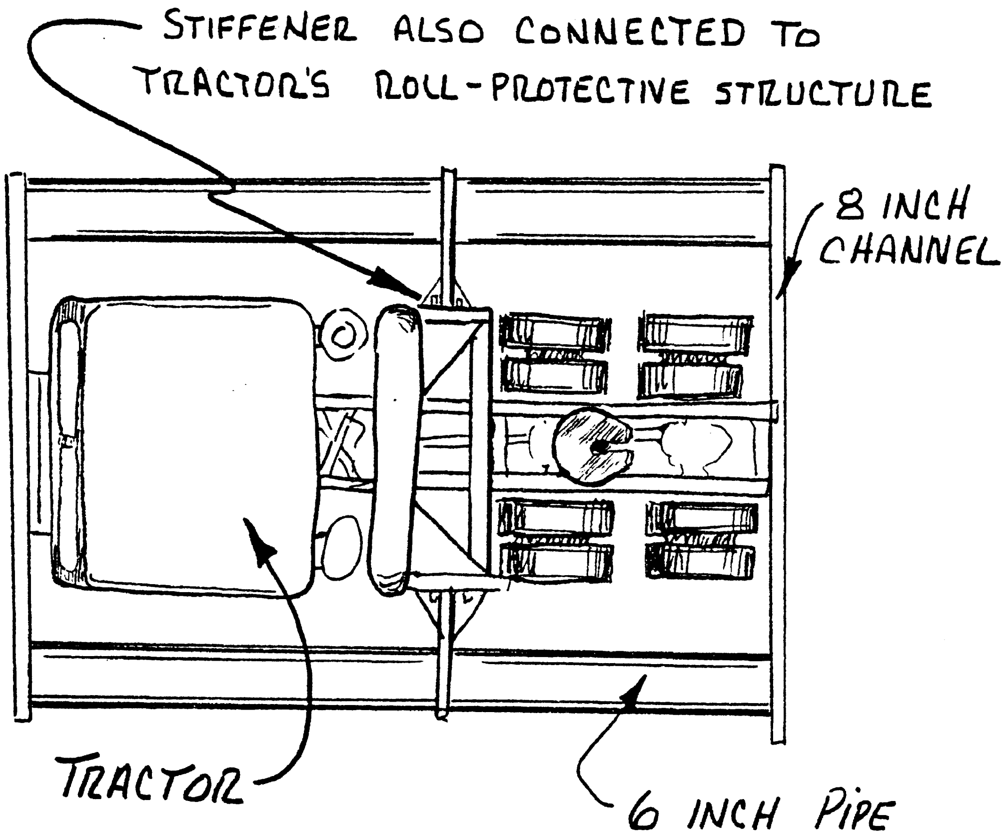


Figure 5.4. Plan view of tractor showing outboard-mounted frame stiffening device.

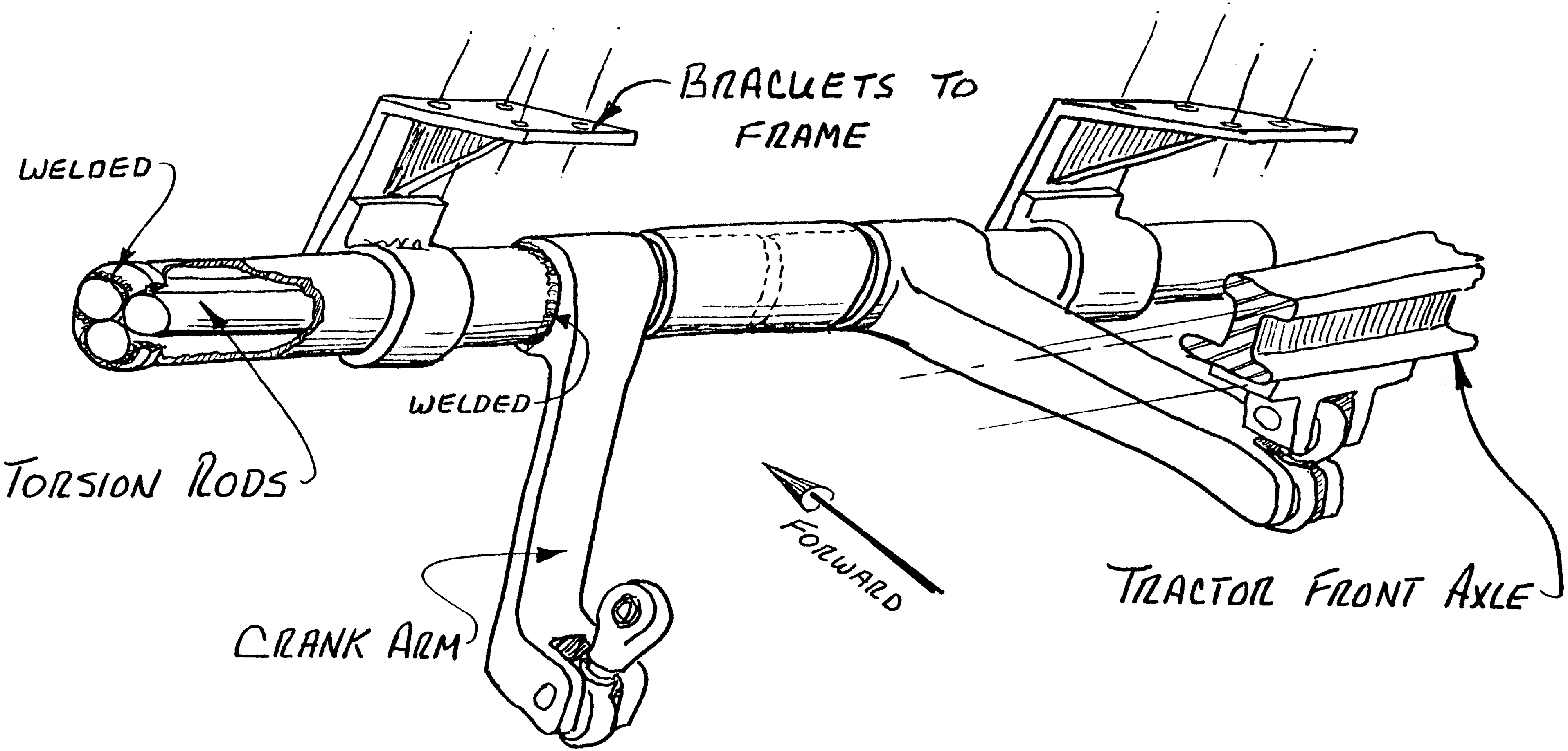


Figure 5.5. Auxiliary front roll-stiffening device ("sway bar") which fastens to vehicle frame, introducing a roll reaction moment as the sprung mass rolls with respect to the front axle.

of both ends of the "enclosure tube" within bearings which are mounted firmly to the tractor frame. The two crank-arm elements fasten to the front axle through short links which incorporate spherical bearing elements.

The enclosure tube consists of two six-inch pipes joined in the middle by a bearing sleeve which allows them to rotate independently. Three 2 3/4-inch diameter steel rods, 130 inches long, run through the enclosure tubes and are welded to plates at both ends. Crank arms welded to the enclosure tubes thus provide connection between the front axle and the long rod elements which comprise the primary torsional spring in the assembly. The auxiliary front roll stiffener was designed to introduce an auxiliary roll spring rate of approximately 100,000 in-lb/deg.

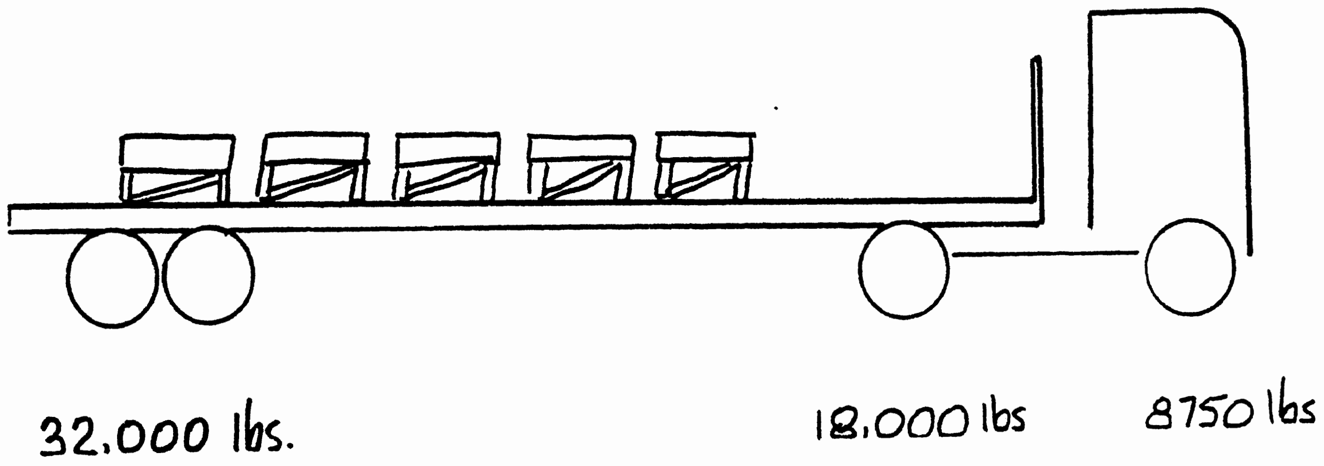
Trailers were loaded for all experiments using racks of cast steel blocks to provide nominal axle loads of 9000 lbs on front axles, 18,000 lbs on single, non-steering axles, and 32,000 lbs on all tandem axles. Loading schemes showing rack location and weight for both configurations are shown in Figure 5.6.

For all tests, the fifth wheel coupling was centered over the rear suspension; all tires were operated at a cold inflation pressure of 85 psi.

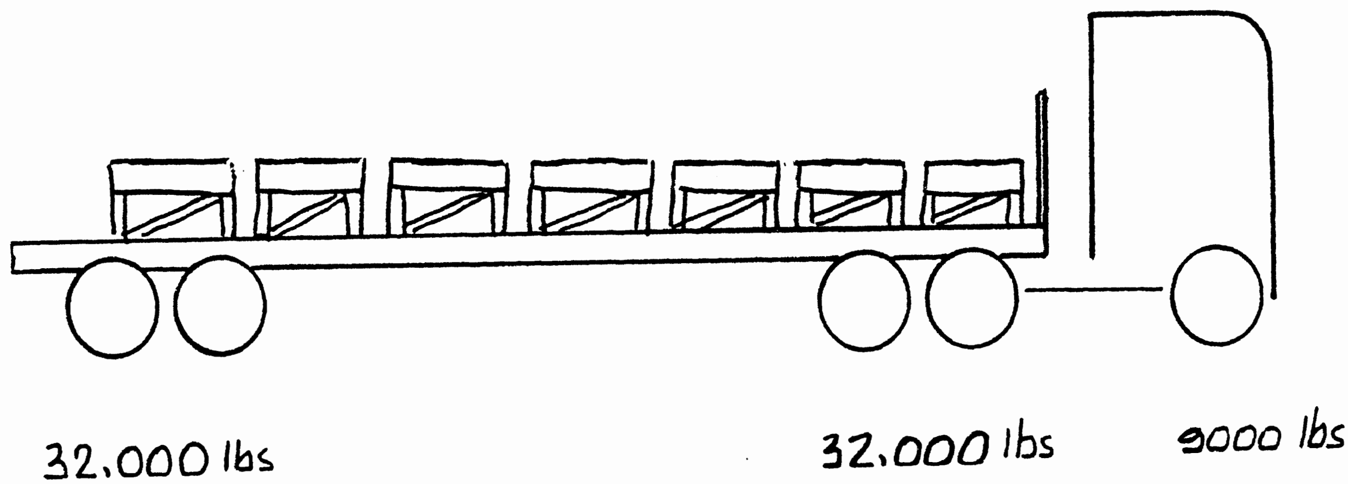
5.2.4 <u>Test Procedures</u>. Two test maneuvers were performed with the various vehicle configurations to validate the analytic activities and to examine vehicle response in the vicinity of yaw instability.

Both trapezoidal- and sinusoidal-steering waveforms, as shown in Figure 5.7, were applied to each vehicle under test. Both inputs were manually applied, but were rendered more precise with the aid of an adjustable steering displacement limiting device. As will be shown in the next section, the trapezoidal-steer maneuver was found to provide the only results directly meaningful to the investigation of yaw stability.

The trapezoidal-steer tests were conducted by applying the indicated steer input to the test vehicle traveling in an initially straight trajectory at 45 mph. Throttle was either maintained at a steady level or increased during the maneuver so as to sustain velocity—although a minimal

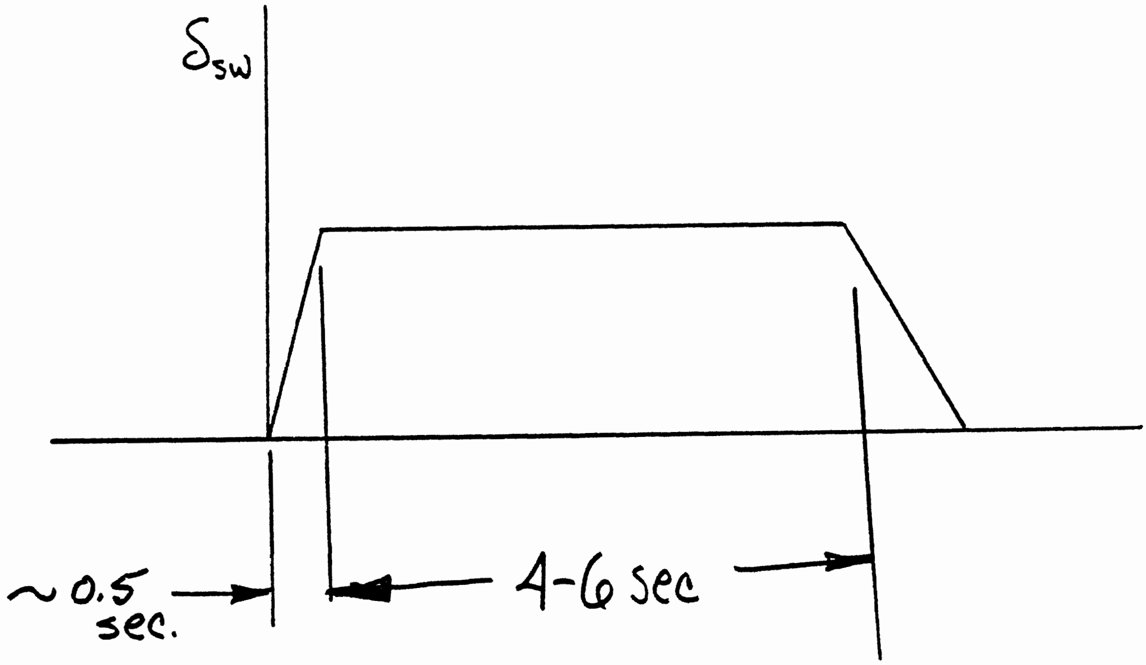


a) Two-axle tractor combinations

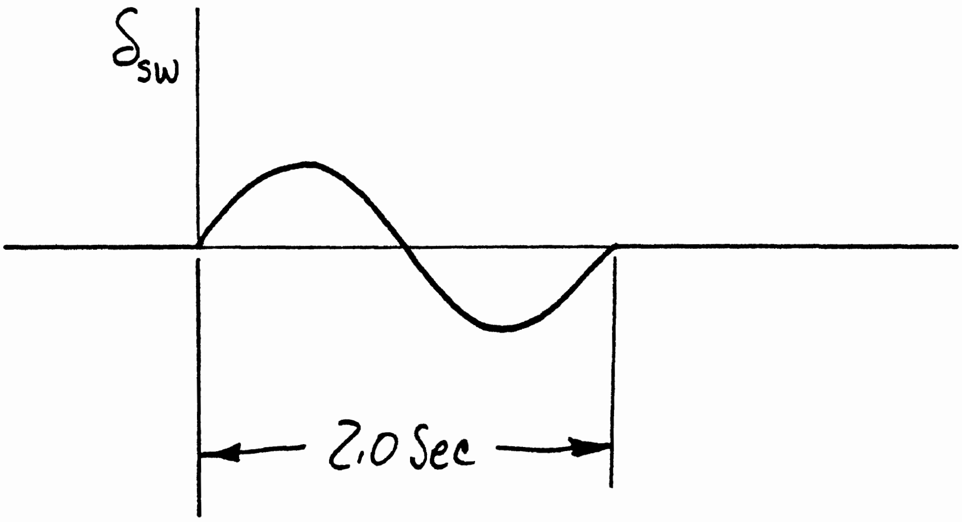


b) Three-axle tractor combinations

Figure 5.6. Load distributions employed with both trailer types for the cases of (a) combinations using the two-axle tractor and (b) combinations using the three-axle tractor.



a) Trapezoidal steer input form



b) Sinusoidal steer input form

Figure 5.7. Steering input waveforms employed in full-scale tests.

level of drive thrust was actually available in the high gear ranges being used. The magnitude of the steering input was constrained by the mechanical stop mechanism which yielded a rapid input without overshoot while also assuring a constant steering-wheel position throughout the maneuver. These tests were run with gradually increased steering inputs over a range of lateral acceleration levels varying from .20 g's up to the limit of the vehicle's capability as defined by the incidence of either yaw divergence, incipient rollover, or front tire side force saturation.

The yaw divergence limit was typically evidenced by the full extension of the articulation limiter mechanism. The incipient rollover point was judged on the basis of outrigger touchdown, with the outrigger height adjusted so that one or more of the vehicle's wheel sets were lifted off the pavement prior to contact. The tire side force saturation condition occurred in those cases in which the frame- and front-roll-stiffener devices produced a premature saturation in tire side forces on the front axle such that an asymptotic yaw response prevailed.

Sinusoidal-steer or lane-change maneuvers were also run in an open-loop fashion using the steering-stop device which was employed in the trapezoidal-steer tests. The sinusoidal input form, shown earlier, has a two-second period, with the timing of the waveform being controlled by the test driver. Repeat runs were typically made until each half period of the sine wave was within .1 second of the desired 1.0 second time for the half-wave. The sine-steer experiments were also run from an initially straight trajectory at 45 mph.

5.3 Summary of Test Results

The results of the full-scale test program are summarized below. A brief reference to the nature of results obtained in sinusoidal-steer tests will be followed by a more extensive review of the trapezoidal-steer results which directly involve yaw stability behavior.

5.3.1 Sinusoidal-Steer Tests. Shown in Figure 5.8 is a set of time histories characteristics of the highest level sinusoidal-steer maneuvers conducted on the two-axle tractor/van trailer combination. The

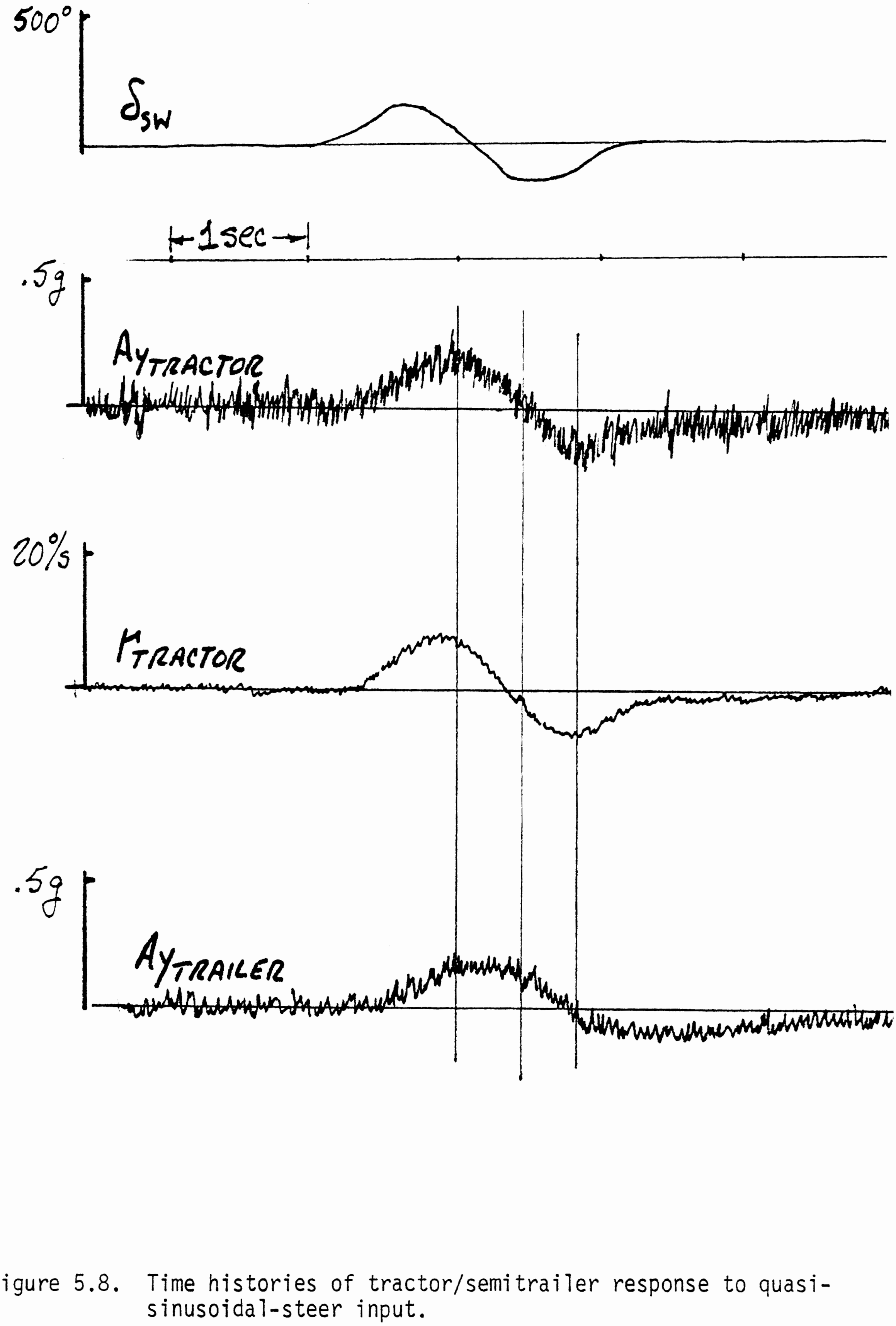


Figure 5.8. Time histories of tractor/semitrailer response to quasi-sinusoidal-steer input.

most significant feature of these time histories, hinging upon their relevance to the yaw stability investigation, involves the lag in trailer lateral acceleration response behind that of the tractor. Further, the trailer lateral acceleration level is attenuated in amplitude with respect to that of the tractor, particularly on the second half of the response wave. These response features are important insofar as they result in a reduced total level of roll moment, as well as lateral force, being reacted at the tractor's fifth wheel coupling during this transient maneuver. Since both reactions constitute the primary means for creating an unstable yaw response of the tractor, their reduction in the sinusoidal-steer tests were found to yield fully stable yaw behavior. As will be elaborated further in Section 6.2, the sinusoidal-steer results reinforce a general finding of the study, namely, that tractor yaw stability in an articulated combination is enhanced in transient maneuvers as the "quickness" of the transient is increased. Indeed, the "quickness" of the selected sine-steer maneuver was sufficient that no potential for unstable behavior derived. Since the trailer constitutes the major mass of the articulated vehicle system, its lag and attenuation features render a stable response at levels of tractor lateral acceleration which are seen to produce an unstable behavior in trapezoidal-steer or other quasi-steady turning maneuvers.

5.3.2 Trapezoidal-Steer Tests. Trapezoidal-steer maneuvers, conducted in sequences of increasing steer level, produced results showing an unstable behavior of the baseline vehicle as well as the respective influences on that behavior deriving from modifications in the roll stiffnesses of the frame and front suspension. Shown in Figure 5.9 is a pair of raw data time histories taken with the Ford two-axle tractor, showing both stable and unstable responses to trapezoidal-steer inputs. Both the yaw-rate and articulation-angle signals are seen as useful indicators of response. We see that neither case can be called a truly steady-state maneuver since an equilibrium velocity condition cannot be sustained with the limited drive torque available. In the divergent case shown, the maneuver concludes when the anti-jackknife device becomes engaged, at

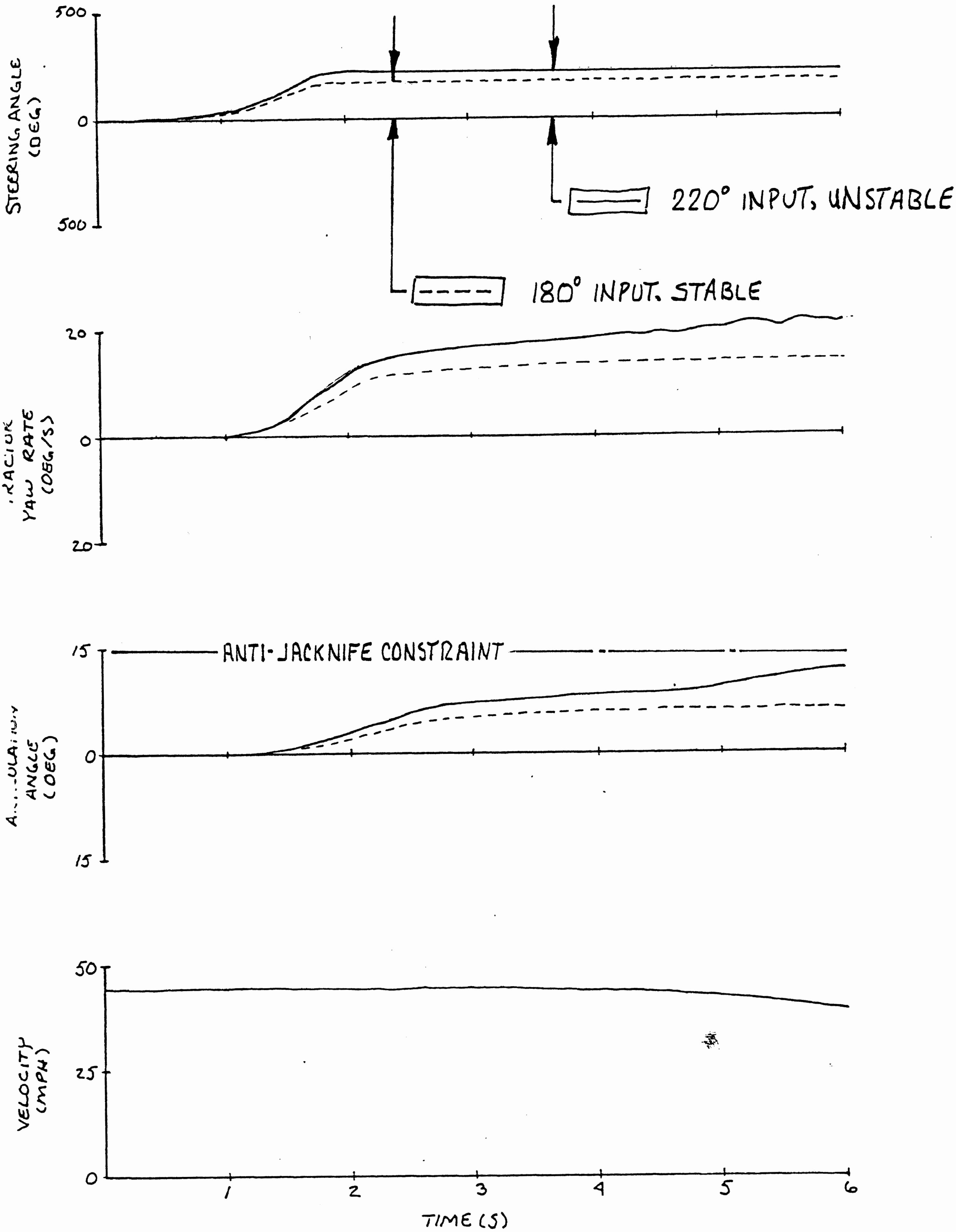


Figure 5.9. Unstable and stable yaw responses of the baseline two-axle tractor/van semitrailer combination.

$\gamma = 15°$. Note that the divergency rate, in terms of articulation angle rate (which is approximately equal to the rate of change of tractor side-slip angle), is about 3 deg/sec. Thus the indicated example could be looked upon as representing a marginally unstable vehicle.

Shown in Figure 5.10 is an overlay of the unstable response from Figure 5.9, together with a stable response obtained for the Ford tractor at nominally the same steer level, but with the frame- and front-roll-stiffening devices engaged. Note that the response is not only stable in the case of the modified vehicle, but also shows a shorter lag time. Time histories, such as these, were reduced to provide various summary measures of yaw response. The most simple measures of response, such as shown in Figure 5.11, are the quasi-steady-state values of steering-wheel angle versus yaw rate, or as shown in Figure 5.12, steering-wheel angle versus lateral acceleration. These data show the response of the Ford tractor/flat-bed semitrailer combination over the range of lateral acceleration levels from 0.15 to 0.50 g's. Figure 5.11 reveals only a subtle upward curvature in yaw rate response with increasing steer level such as would suggest a tendency to diverge. Thus the direct measures of yaw rate and lateral acceleration do not provide powerful discriminators of response, when the interest is in mild forms of yaw divergency such as prevail here. As shown in Figure 5.13, modifications to frame- and front-roll stiffness serve to reverse the curvature of the yaw-rate response, but the clear determination of proximity to an instability point is difficult to establish. A complete set of yaw rate and lateral acceleration response plots are presented in Appendix III.

Shown in Figure 5.14 are the same baseline data for the Ford tractor/flat-bed semitrailer combination plotted according to the measures of the handling diagram. We see in these data that the tractor exhibits an oversteer-polarity slope over the .15 to .5 g range of lateral acceleration. Evaluating the nominal (average right and left turn) response curve using the critical velocity slope for the 45-mph test speed, we see that the vehicle response approaches the critical slope condition at .42 g of lateral acceleration. At this operating condition, the tractor shows an oversteer-polarity gradient of $U = -4.8$ deg/g.

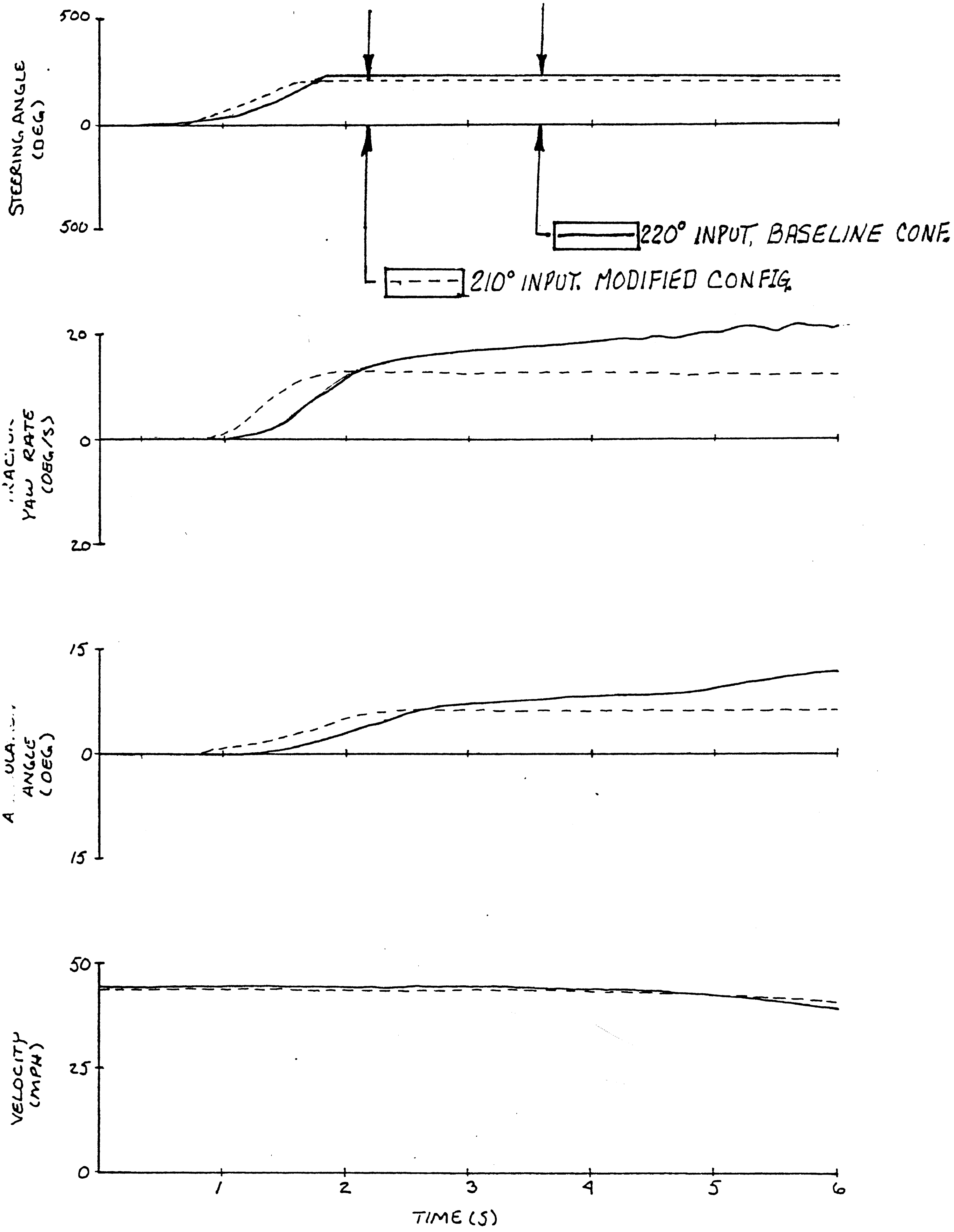


Figure 5.10. Unstable response of the baseline two-axle tractor/van semi-trailer combination as compared with stable response of the same combination, fully modified.

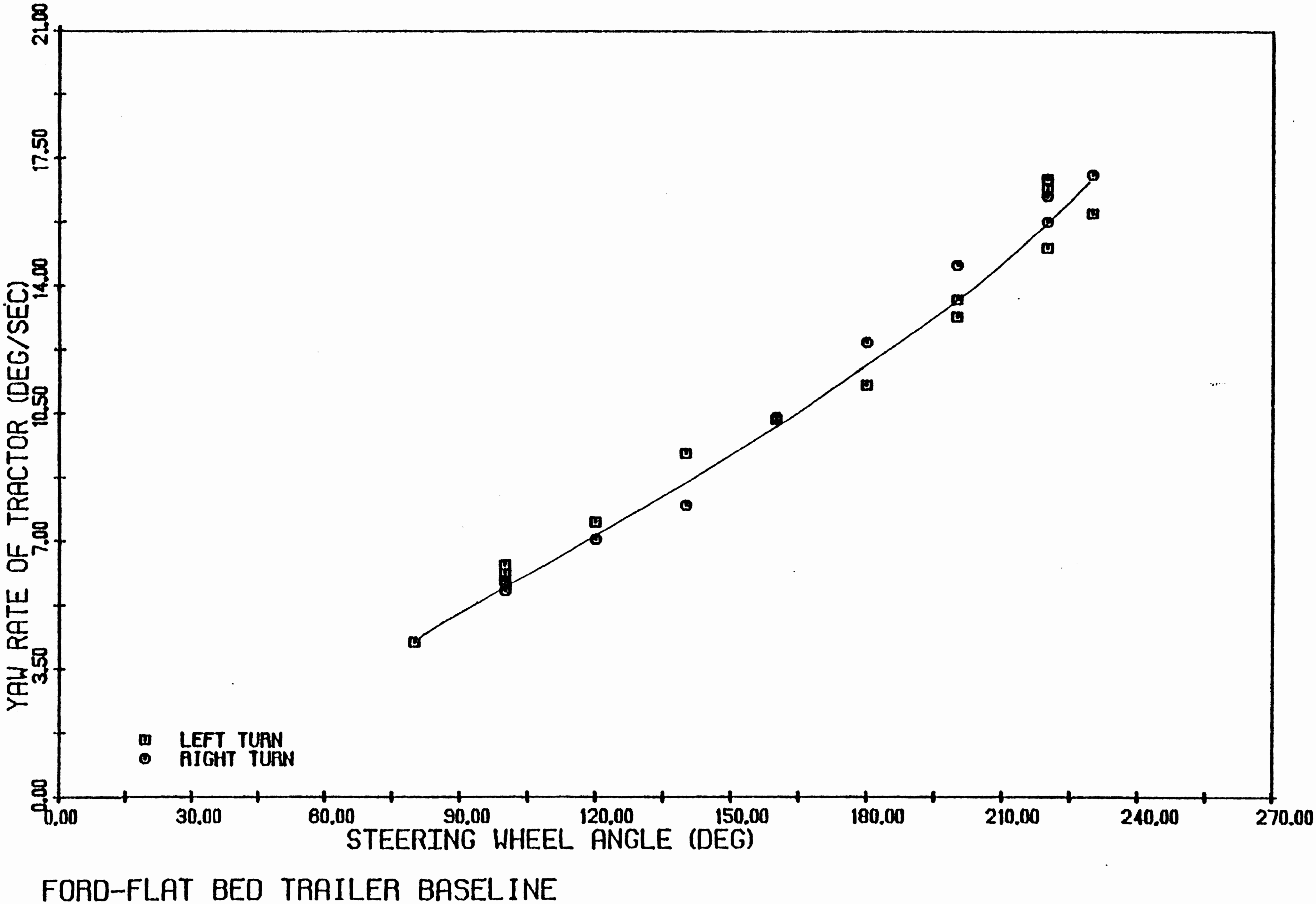
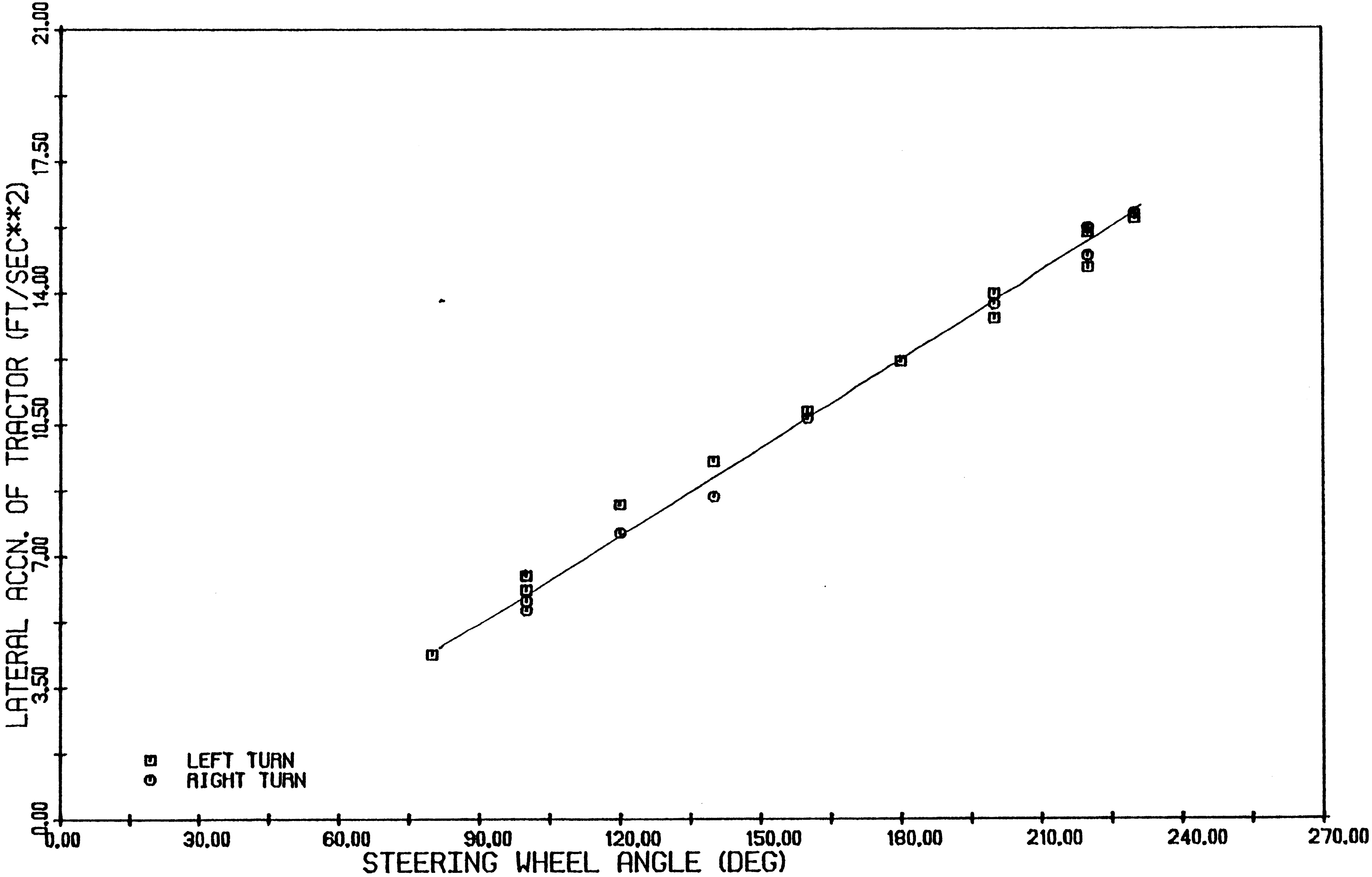
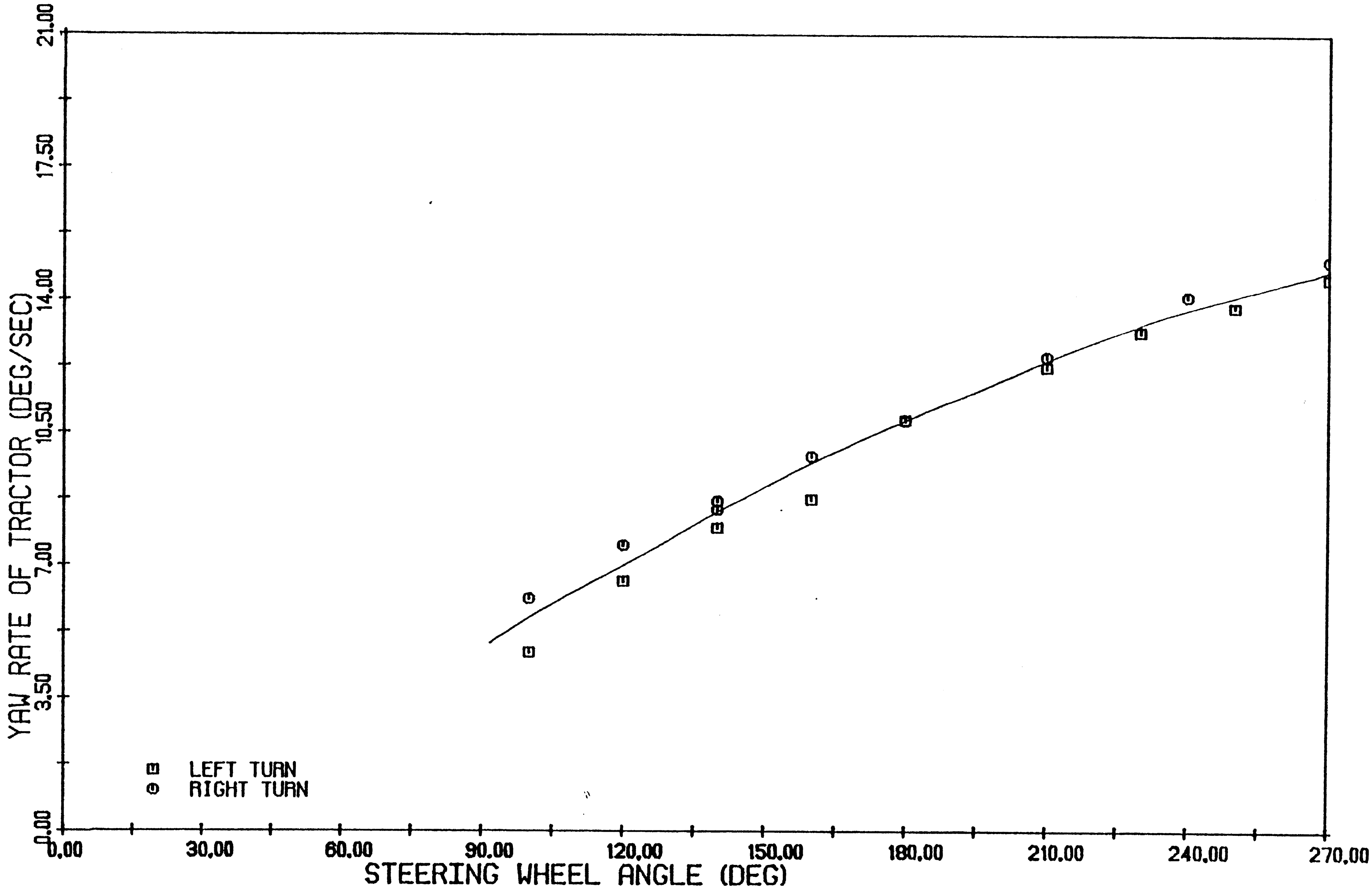


Figure 5.11 Quasi steady state values of yaw rate vs. steering wheel angle.



FORD-FLAT BED TRAILER BASELINE

Figure 5.12. Quasi-steady-state values of lateral acceleration vs. steering-wheel angle.



FORD-FLAT BED TRAILER MODIFIED

Figure 5.13. Quasi-steady-state values of yaw rate vs. lateral acceleration.

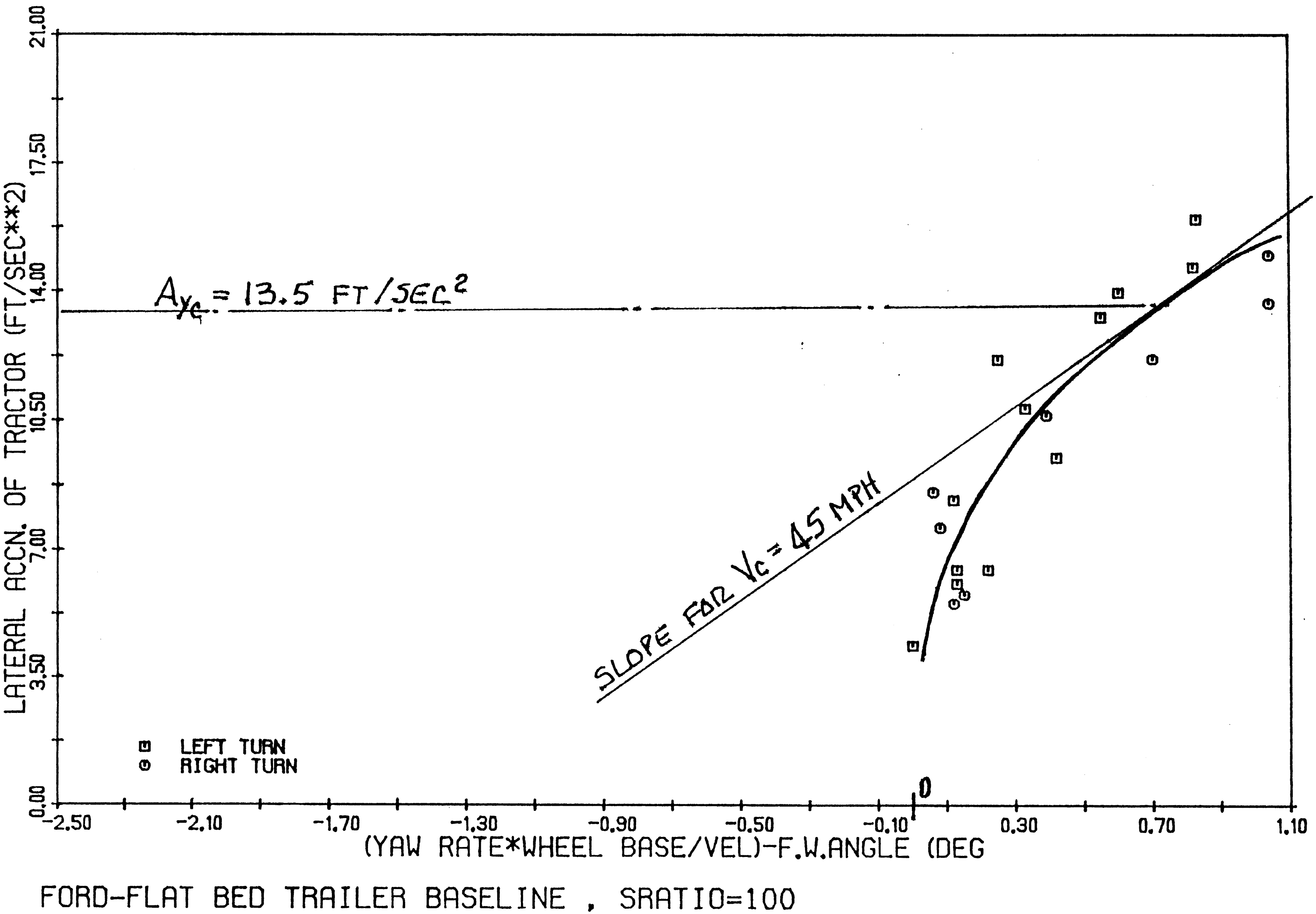


Figure 5.14. Handling diagram produced from trapezoidal-steer test data.

In Figure 5.15 the handling curves (using left-turn data only) obtained for the Ford tractor in its baseline and modified conditions are shown. The figure shows the performance achieved with single modifying devices engaged, viz., "frame alone" and "sway bar alone," as well as the condition in which both elements are engaged, cited as the "modified" condition. Some degree of anomalous lateral shifting of the respective curves derives from the unaccounted influence of steering lash on the abscissa variable—since the front-wheel angle was estimated using the steering-wheel angle signal together with an "effective steering ratio" which was established from other data taken on this vehicle using front-wheel angle transducers. Nevertheless, the slopes prevailing for each of the various vehicle conditions clearly reveal that the oversteer behavior is eliminated when the "sway bar" element is added. The vehicle's response is seen to be decidedly understeer ($U \sim +2.6$ deg/g) when both the frame- and front-roll-stiffening devices are engaged together. Indeed, these data show that the addition of the front-roll stiffener, alone, produces a sufficiently improved performance that the critical acceleration level was not approached at the 45-mph test speed. The engagement of the frame stiffener alone, however, was seen to only mildly reduce the oversteer gradient of the baseline tractor.

As seen in Figure 5.16, a summary of the data taken with the International Harvester tractor/flat-bed semitrailer combination show a similar set of influences. (Again, the reader is advised that the lateral spacing of the respective curves is largely an anomaly introduced by the unaccounted-for steering lash.) The baseline performance of this vehicle reveals a critical acceleration of $A_y = .44$ g at the 45-mph test speed. On examination of the data obtained for the modified vehicles, we see that the unstable behavior of the baseline configuration is eliminated by the addition of the "sway bar" and by the combined sway bar and frame stiffener, such that understeer behavior prevails throughout the maneuver range.

In the "modified" condition, we see very high levels of understeer, i.e., above $U = +15$ deg/g, indicating that the front tires have virtually saturated in side force capacity in the vicinity of $A_y = 0.5$ g. Also, as with the two-axle tractor, we see that the frame stiffener alone provides only a small reduction in the oversteer character of the baseline vehicle.

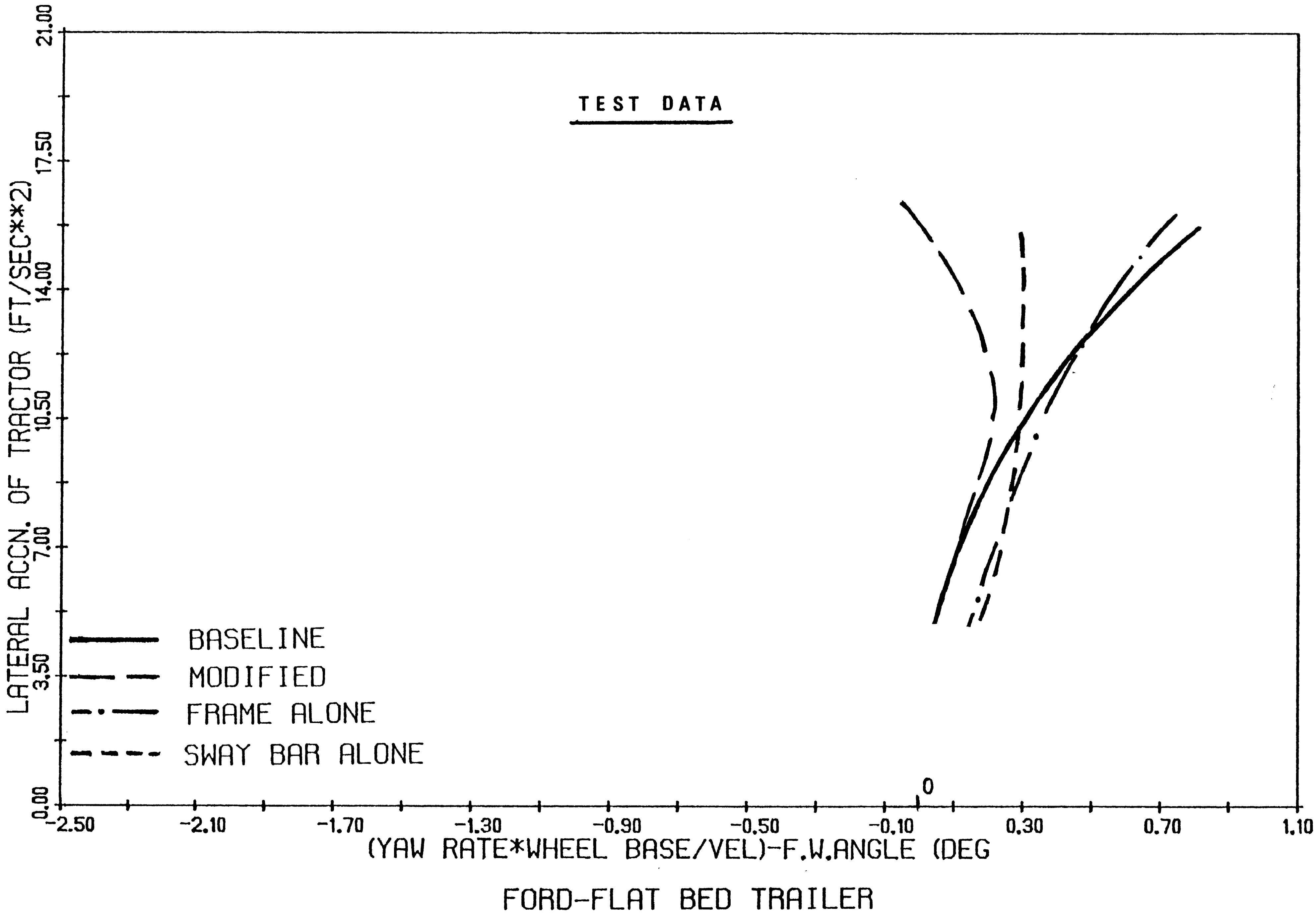
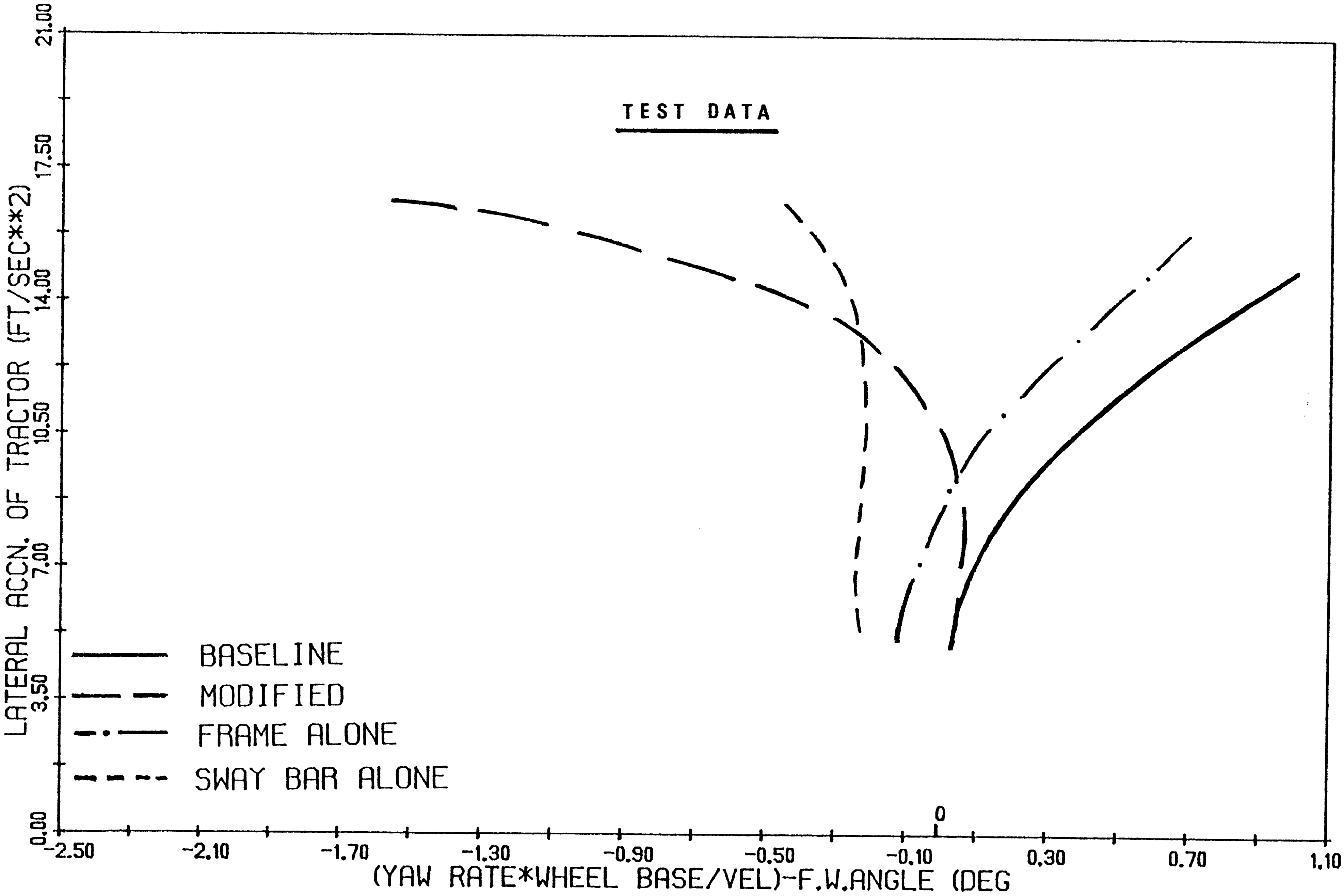


Figure 5.15. Handling diagram showing smoothed curves fitted to test data.



IHC-FLAT BED TRAILER

Figure .16. Handling diagram showing smoothed curves fitted to test data.

The corresponding handling curves obtained for the four cases of the International Harvester tractor/van semitrailer combination are shown in Figure 5.17, revealing that negligible changes in local slopes are observed due to the change in the type of semitrailer. A more generalized examination of the influence of trailer parameters, however (see, also, Section 6.4), shows that torsional stiffness of the trailer frame and the roll stiffness of the trailer rear suspension both combine to influence that portion of the trailer's overall d'Alembert roll moment which is borne by the tractor. Further, such influences can significantly alter tractor yaw stability, from one set of trailer parameters to another. Thus, the differences observed in the test data between tractor responses with specific samples of (a) flat-bed and (b) van semitrailers should not be interpreted as generally indicative of the influence of trailer type on tractor yaw stability.

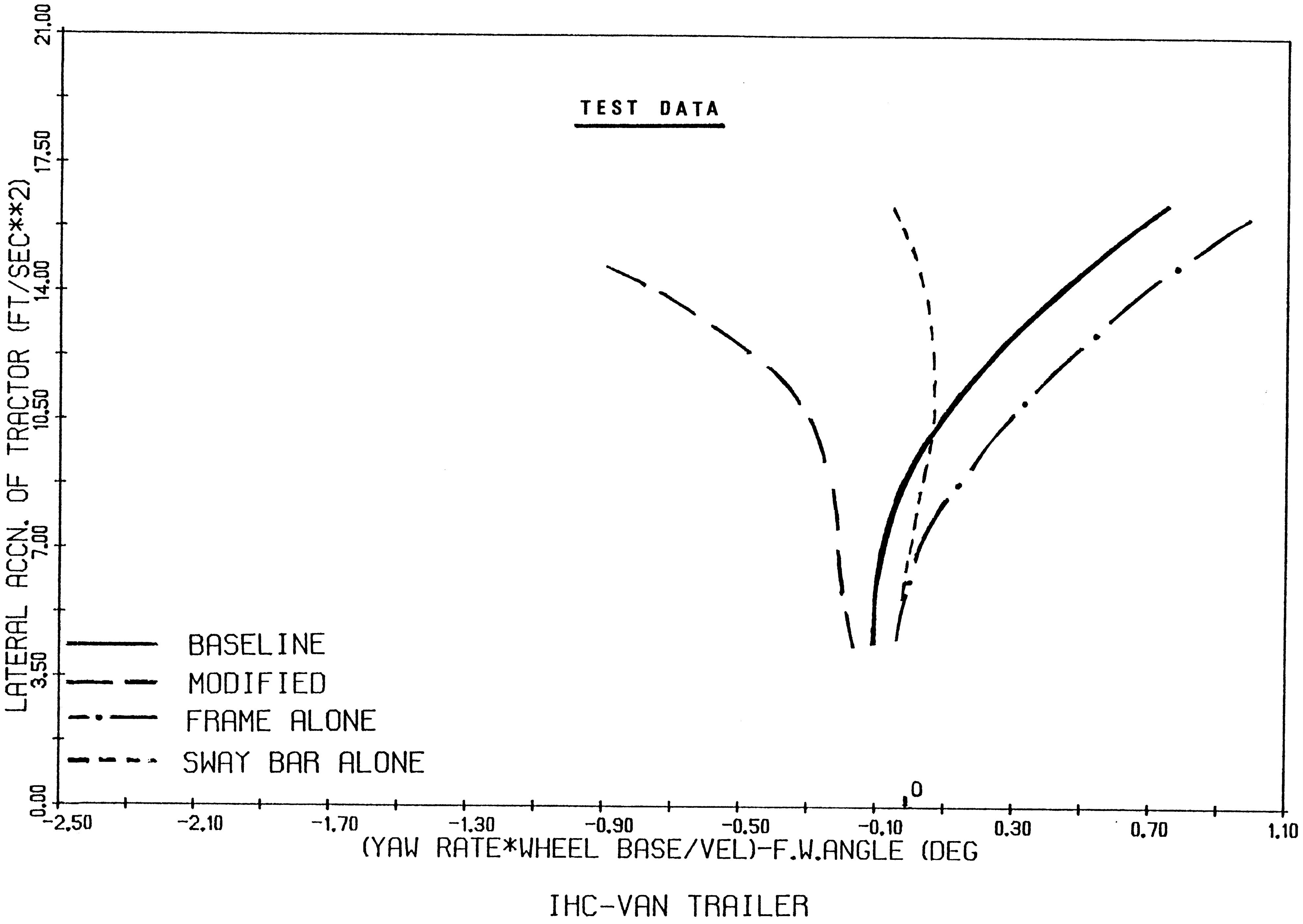


Figure 5.17. Handling diagram showing smoothed curves fitted to test data.

6.0 COMPUTERIZED PARAMETRIC SENSITIVITY STUDY

In this section the methods and results of a computerized simulation study are presented. The simulation exercise was undertaken to determine the sensitivities of tractor yaw stability to both design and operating variables. In the discussion to follow, the simulation model is briefly described and data are shown documenting its suitability for the prediction of tractor yaw stability. In Section 6.2, the use of a peculiar steering waveform as the simulated control input is discussed. In particular, this discussion treats the application of the handling diagram to vehicle response data obtained in simulations which are quasi-steady-state in nature.

Simulation techniques were applied to answer two questions, namely:

1) To what extent can tractor yaw instability be prevented through the modification of the tractor's fore/aft distribution of suspension roll stiffness? and

2) To what extent is tractor yaw instability aggravated through certain operating practices and equipment options which are commonly employed with commercial vehicles?

Accordingly, Sections 6.3 and 6.4 present simulation results providing answers to these two questions, respectively.

6.1 The Vehicle Model

The simulation program used in this study is a modified version of the "Phase II Directional Response Simulation" developed at HSRI [4]. The Phase II simulation is a comprehensive mathematical model capable of predicting the response of a truck or tractor-trailer to steering and/or braking maneuvers. The salient features of this simulation are:

1) A detailed semi-empirical tire model which, when made to fit measured tire data, is capable of predicting longitudinal and lateral tire forces under combined braking and sideslipping conditions.

2) Options for treating various suspension types, such as four-spring and walking-beam tandem axles, single-axle, and independent suspensions.

3) Provision for incorporating suspension nonlinearities such as coulomb friction.

4) A number of options for treating roll steer, steering compliance, and side-to-side differences in front-wheel angles.

In the original version of the simulation, the tractor and trailer masses were treated as rigid bodies, and the fifth wheel was modeled as a "stiff" spring-damper system which provides a nominally rigid coupling between the sprung masses of the tractor and trailer while permitting relative rotations in the yaw and pitch planes.

For use in calculations of yaw stability limits, the existing fifth wheel roll compliance, MC5 (see Figure 6.1), was augmented with the additional springs, TTC and TRSTF, representing tractor and trailer frame compliances, respectively.

The spring, TTC, is located at the elevation of the frame rails of the tractor while the trailer frame compliance parameter, TRSTF, is placed in series with the torsional spring, MC5, and is at the elevation of the fifth wheel.

These changes in the model introduce an additional roll degree of freedom, Y(45), of the fifth wheel structure. The complexities of adding the dynamics of the fifth wheel structure to the computer program was avoided by considering the fifth wheel as a massless member with its mass and moments of inertia being included in the parameters describing the sprung mass of the tractor. This simplification permits computation of the roll moments transmitted through the tractor and trailer frames by a solution of the equations defining static equilibrium of the massless fifth wheel structure. The equations corresponding to this modification can be found in Reference [6]. The method adopted to estimate the parameters TTC and TRSTF is given in Appendix II of this report.

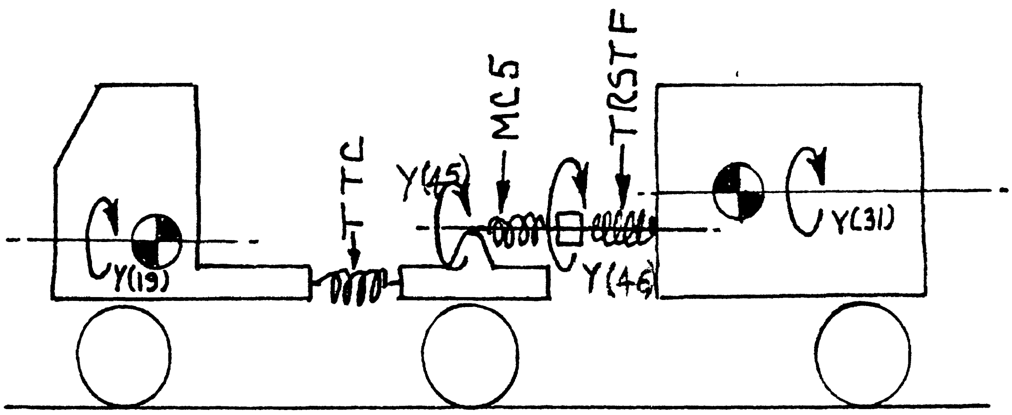


Figure 6.1. Revised simulation model incorporating frame torsional stiffnesses.

The modified simulation was employed in a set of calculations to verify its use in predicting tractor yaw stability, for cases representing two of the test vehicle combinations, namely,

1) Ford tractor/flat-bed trailer, and

2) International Harvester tractor/van trailer

For each vehicle combination, simulation results were compared with test results for each of the configurations listed below:

1) baseline vehicle,

2) frame stiffener attached to tractor,

3) auxiliary front roll stiffness ("sway bar") added to tractor,

4) both the frame stiffener and sway bar added to tractor (called the "modified" condition).

In these simulations, the forward velocity at the beginning of the maneuver was 43 mph. The tire loadings corresponded to the full-load test condition. No braking or accelerating torques were applied and the vehicle was allowed to coast freely during its response to a trapezoidal steering input. The loss of forward speed during the resulting quasi-steady turn is dependent upon the severity of the maneuver, and is seen to be small for lateral acceleration levels below 0.4 g.

Values of average yaw rate, forward velocity, and lateral acceleration were determined for each run during the quasi-steady portion of the response, thereby permitting calculation of those variables needed to produce the desired handling diagram.

Smoothed curves of test and simulation results are shown in the handling diagrams of Figures 6.2 and 6.3. Inspection of these figures shows that the primary influence of each vehicle modification on the test results is closely duplicated by the simulation results. On recognizing that yaw stability is a function of the slope of the handing curve which prevails in the range of elevated lateral acceleration, we see that, discounting absolute values of the abscissa variable, the simulation

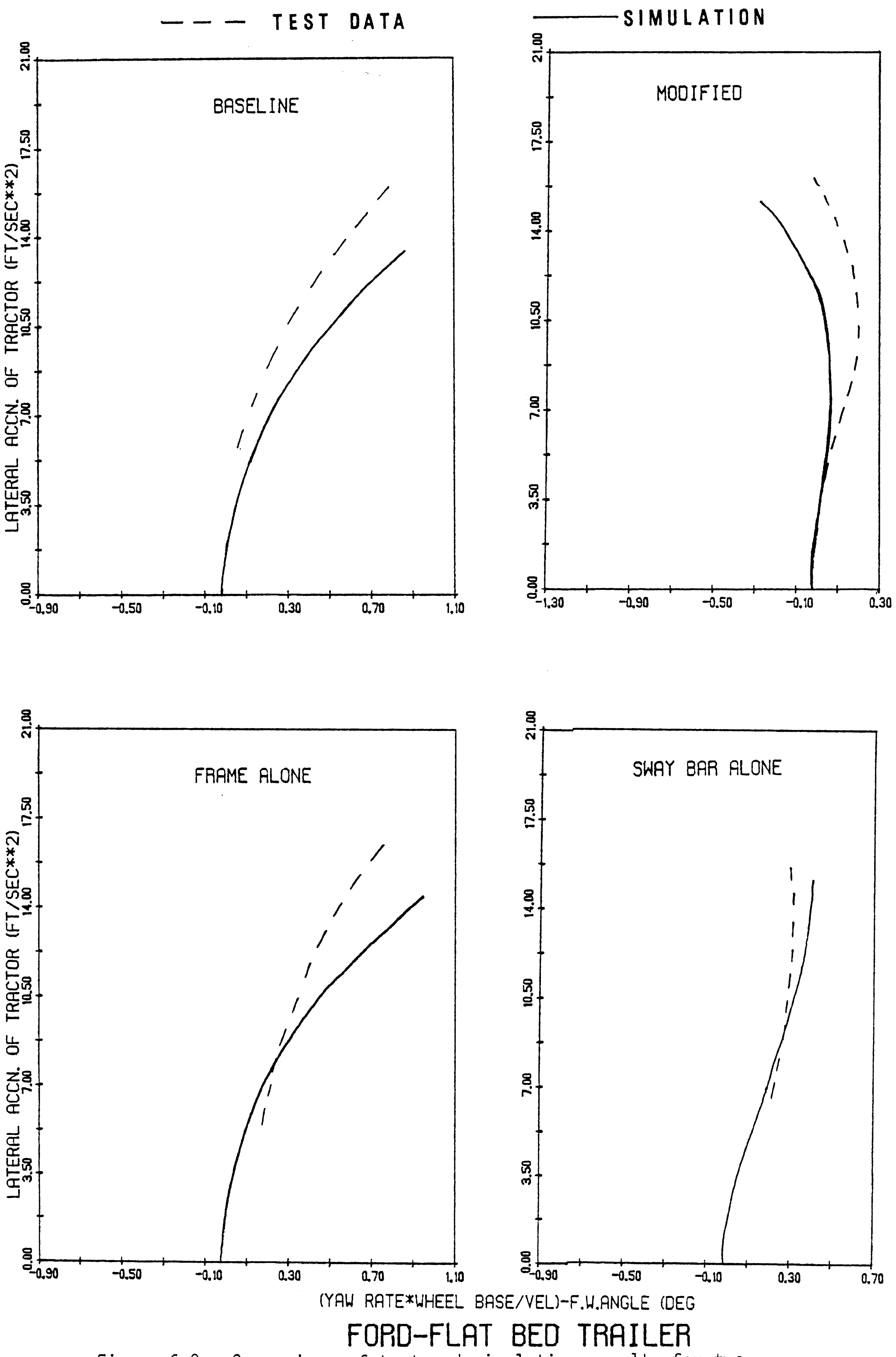
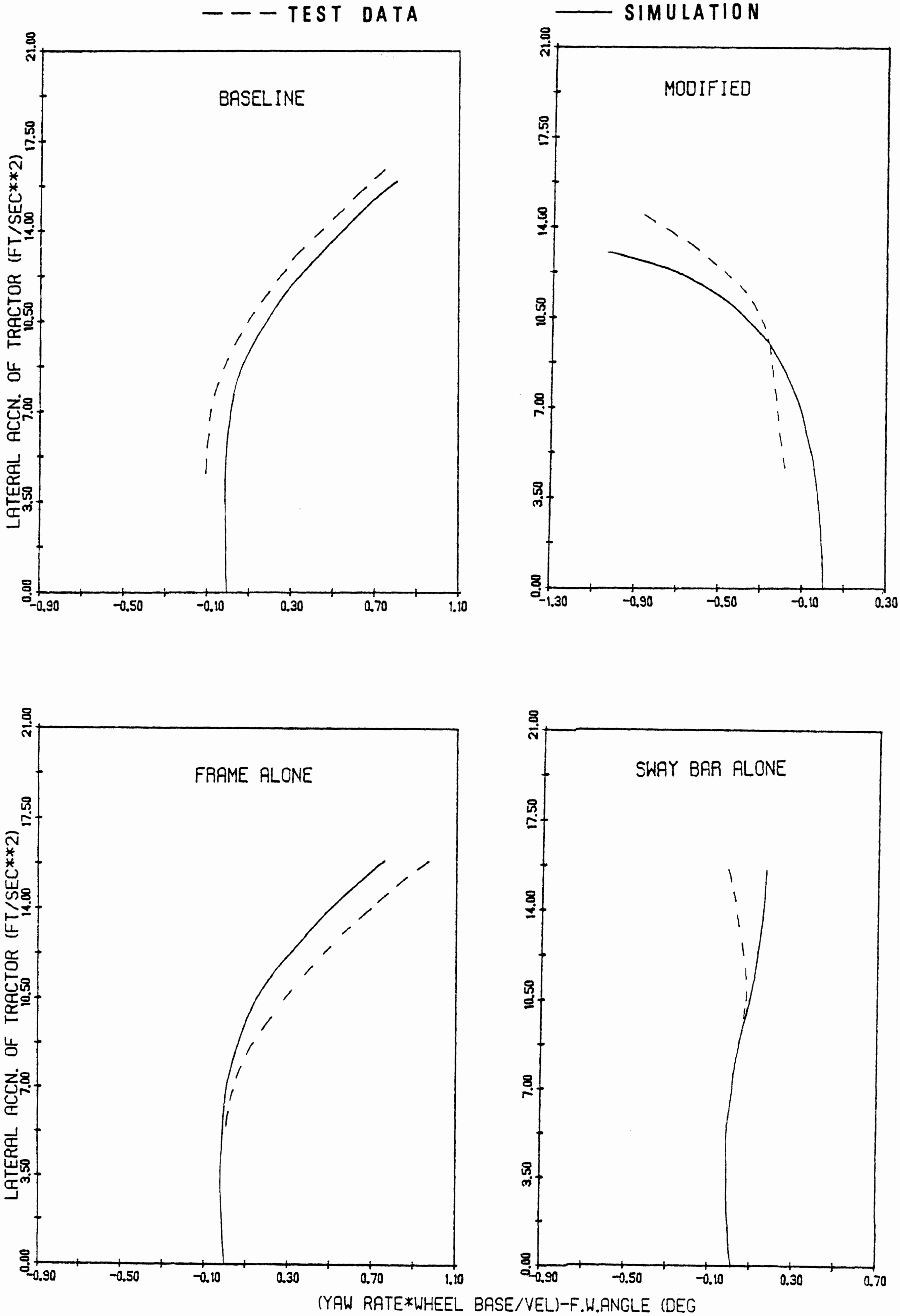


Figure 6.2. Comparison of test and simulation results for two-axle tractor/flat-bed trailer combination.



IHC-VAN TRAILER

Figure 6.3. Comparison of test and simulation results for three-axle

yields a rather good prediction of yaw stability. It is presumed that most of the disparity that is seen between the test and simulation results derives from an incomplete characterization of vehicle and tire parameters.

6.2 The Handling Diagram Obtained Using Simulated Responses to a Ramp Input of Steering Angle

To improve the efficiency of computer simulation in searching the boundaries of vehicle yaw stability, a ramp input of front-wheel steer angle was used for the control input form. This input waveform provided for a continuous sweep over the full range of lateral acceleration up to the rollover threshold, in a single computer run, as diagrammed in Figure 6.4. Although the ramp input rate, variously chosen as 1.5 deg/sec or 1.0 deg/sec, was sufficiently slow that the pitch, bounce, and roll transients of the sprung mass were well subdued, the rates employed do yield a distinctly non-steady-state yaw behavior of the tractor. Thus, while the ramp rates were selected to minimize the simulated time needed to reach rollover (thus also minimizing velocity fall-off during the maneuver), the results represent a special transient case and require a certain understanding for their interpretation.

The primary mechanism explaining the "specialty" of the ramp input transient maneuver is shown in Figure 6.5 in which the lateral acceleration response of the simulated semitrailer is seen to lag the response of the tractor to a steering input ramp rate of 1.0 deg/sec. Because of the trailer lag mechanism, the tractor at any moment in time experiences a smaller level of lateral load transfer than would have accrued at the same level of tractor lateral acceleration in a steady-state turn. Thus, the "trailer lag" aspect of this transient maneuver has been seen to contribute a net stabilizing effect for those tractors which show a potential for yaw instability in steady-state turn conditions.

Two other elements of the vehicle's response to a transient steer condition, which also influence the stability level, have been identified through a simple yaw plane analysis of the tractor-semitrailer (see Appendix VII). These two elements constitute the additional components

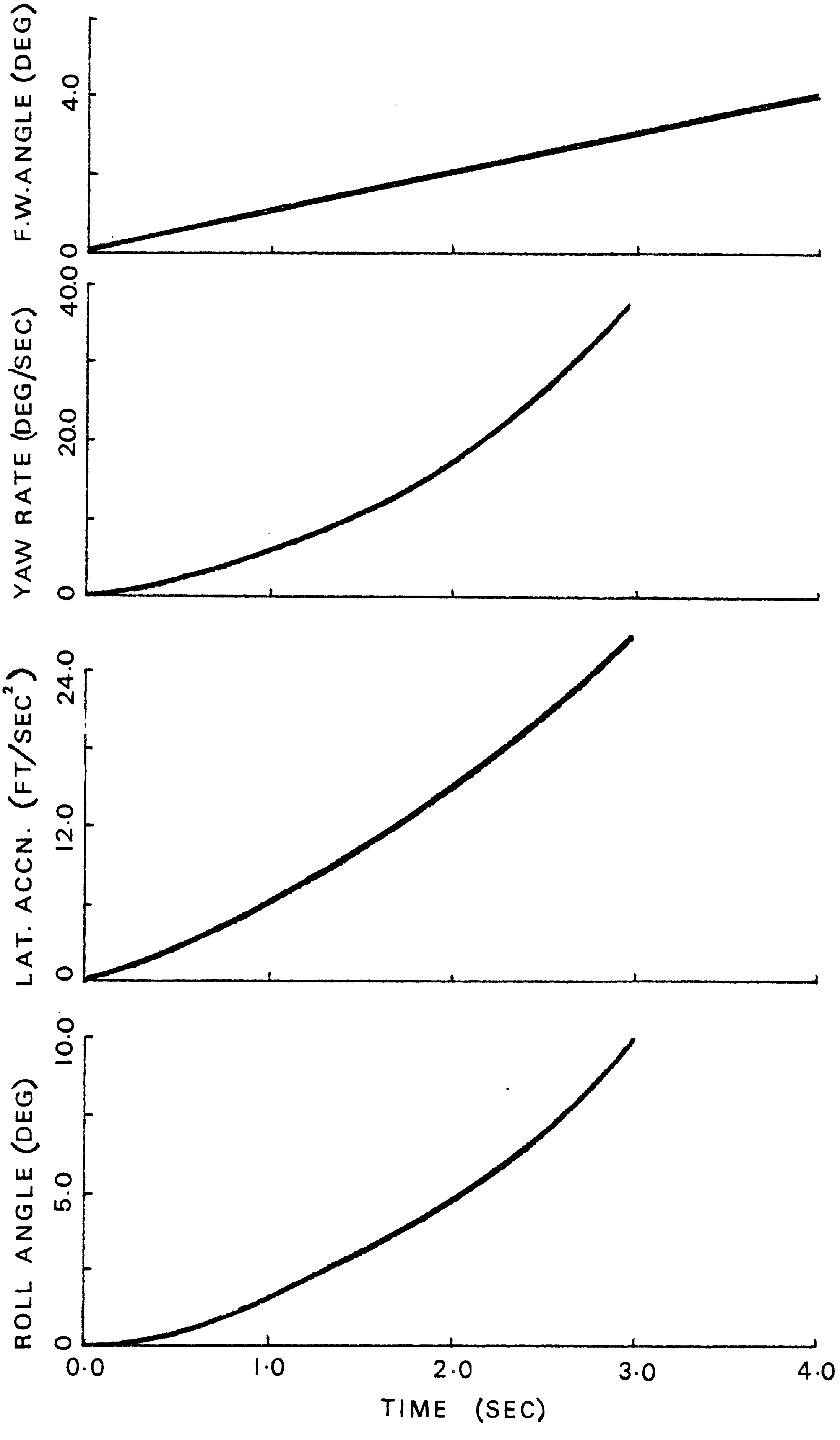


Figure 6.4. Typical input and response histories for the ramp-steer maneuver.

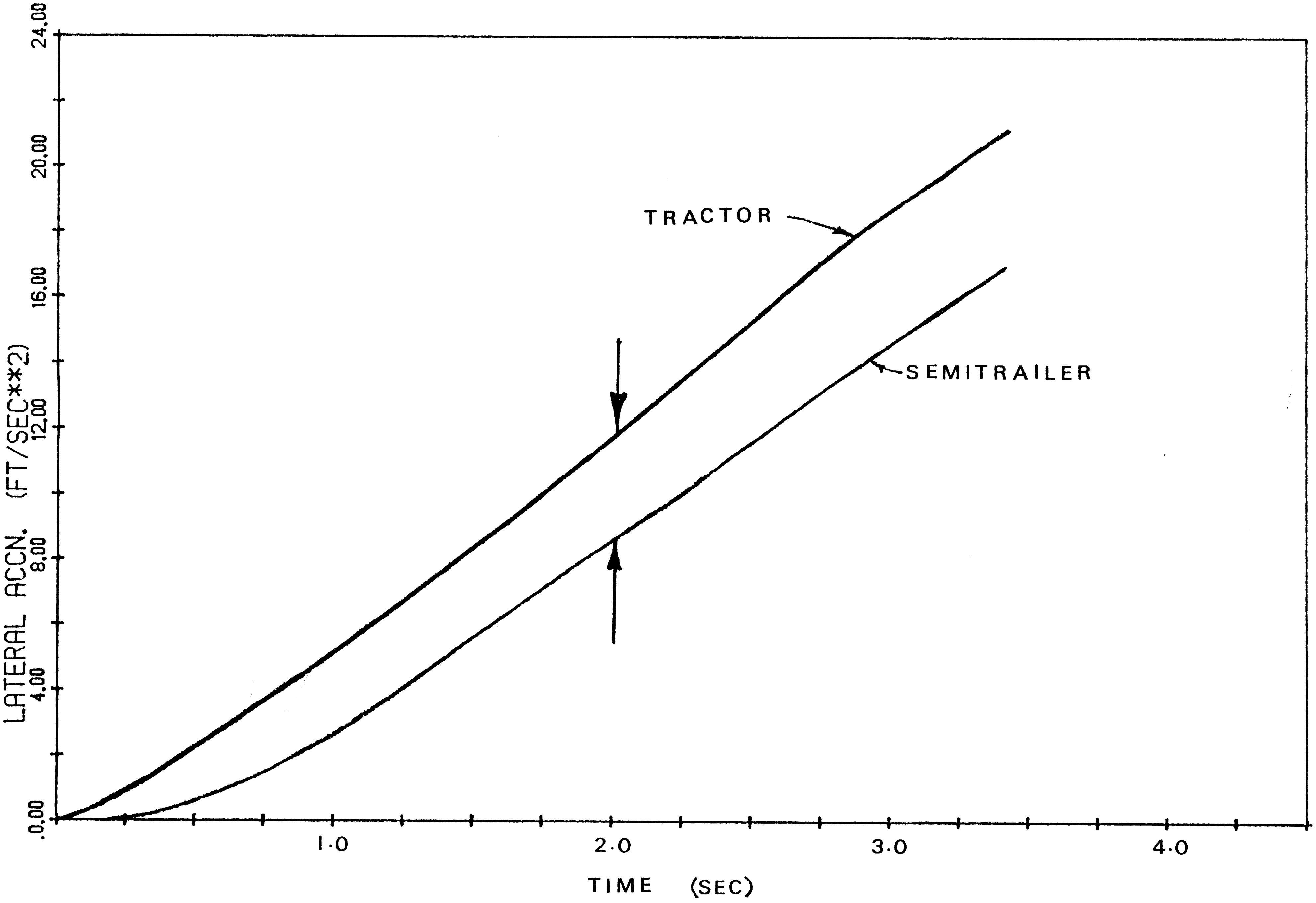


Figure 6.5. Lag in lateral acceleration of semitrailer with respect to tractor in ramp-steer maneuver.

of tire side force at each tractor axle which account for yaw acceleration of (a) the tractor and (b) the semitrailer. As is clear in the analysis presented in the appendix, the tractor tire side force components which produce tractor yaw acceleration act to aid tractor yaw stability (relative to steady-state behavior) while the components needed to produce trailer yaw acceleration represent a destabilizing influence on the tractor. Hand-reduction of a typical ramp-steer computer run reveals that these two opposing influences may introduce a net destabilizing contribution which is about 30% of the magnitude of the stabilizing contribution arising from the "lagging trailer" mechanism.

Moreover, it has been found that the transient character of the ramp-steer maneuver yields a greater level of vehicle yaw stability than that found in steady turns and likewise produces results on the handling diagram which can differ quantitatively from the steady-state curves. As summarized in Figure 6.6, the vehicle decreases in stability as the steady-state condition is approached. (That is, as the slope of the ramp-steer input is reduced toward zero. The curve marked "steady-state values" was produced from results of successive computer runs using step-steer inputs.) Note that this observation confirms the same general finding as was made in the case of sinusoidal steer test results in Section 5.3. Figure 6.6 also shows that increasing the steering rate produces an apparent increase in the "initial understeer level" accruing at zero lateral acceleration. This result is simply an anomaly of the start-up transient accompanying initiation of the ramp. The apparent understeer behavior derives from the fact that front-tire slip angle is developed before rear-tire slip angle, or put in terms of the plotted abscissa variable, the yaw rate response lags the input of front-wheel angle.

By way of interpretation of ramp-steer results, certain general comments are in order. First we observe that the handling diagram was originally developed as an aid in presenting steady-state properties of vehicles. Nevertheless, analysis shows that the stability determination which is facilitated by the handling diagram is valid for application to curves produced using ramp-steer data. Secondly we suggest that the

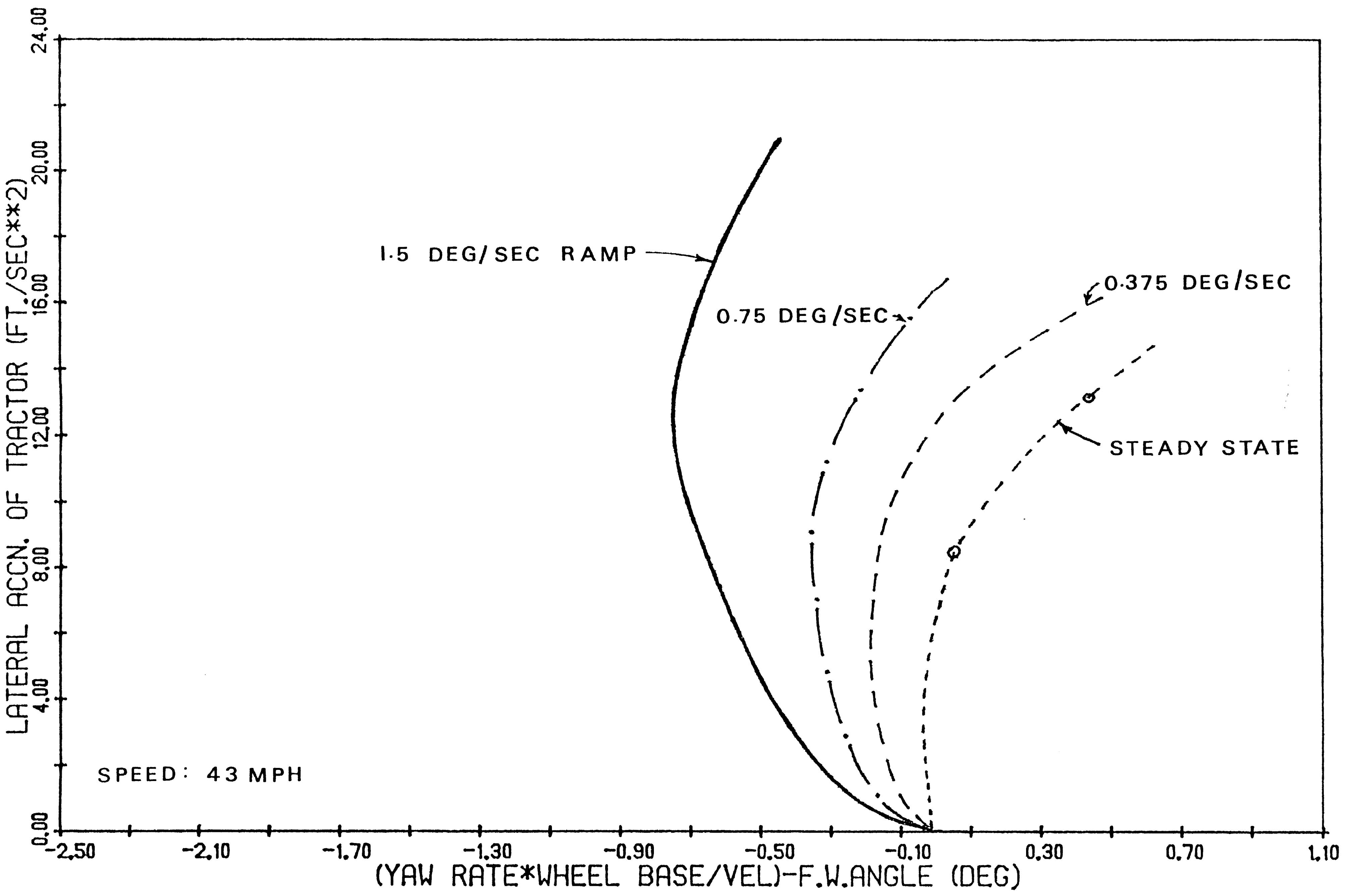


Figure 6.6. Increasing yaw stability with increasing steer input rate.

ramp-steer case is a special maneuvering case, just as is the steady-state turn. No single case, for articulated vehicles, can be looked upon as universally descriptive of a vehicle's operating characteristics. Nevertheless, since ramp-steer and steady turns produce similar parametric-sensitivity results (that is, similar patterns of the relative influence of design and operating parameters), the handling diagram based on ramp-steer results can be employed to demonstrate relative trends which can be presumed to apply rather generally. It is in this vein that the more efficient ramp-steer maneuver was applied to obtain the parametric-sensitivity data which follow in this section.

Also, as was discussed in general terms in Section 4.0, handling curves derived from ramp-steer results can be reduced to a lateral acceleration numeric, $A_{y_{crit}}$, which constitutes a yaw stability threshold. In the summary of simulation data which follows, ramp-steer results for both two-axle and three-axle tractors are evaluated at the V_{crit} = 50 mph condition to provide a tabulation of the performance measure $A_{y_{crit}}\Big|_{V_{crit} = 50 \text{ mph}}$. Although this numeric clearly discriminates among the various parameters by establishing the relative significance of parametric changes to the yaw stability threshold, the related measure concerning proximity of the yaw stability threshold to the rollover threshold is less straightforward. The difficulty which arises is due to the peculiarly elevated rollover threshold (expressed in terms of the lateral acceleration level of the tractor) which derives in the ramp-steer maneuver as a result of the trailer's lateral acceleration response substantially lagging that of the tractor. Just as trailer lag was seen to reduce the instantaneous level of lateral load transfer at the tractor, thus enhancing yaw stability, it also enhances the rollover immunity in such a transient maneuver (at least insofar as tractor lateral acceleration level is looked upon as the performance measure). Thus, while the tractor is truly capable of achieving a greater level of maneuvering severity in this transient case than that which is achievable without rollover in the steady-state turn, the difference in rollover threshold is not amenable to a simple analysis. Since the simulation model employed in this study was not suited for predicting rollover limits, ramp-steer results are presented in terms of yaw stability thresholds without reference to their proximity to a corresponding roll stability threshold. In all cases, however, the presented levels of yaw stability threshold are "valid"—that is, the levels were derived for vehicles which had not yet arrived at the rollover threshold.

6.3 Influence of Roll Stiffness Distribution

The first of the two major parametric sensitivity studies involved an examination of the influence of tractor roll stiffness distribution on yaw stability. Specifically, the parameter combinations, shown in Table 6.1, were employed in a matrix of simulated ramp-steer maneuvers. (Note that the selected parametric variations were all in the direction of creating a more front-biased distribution of tractor roll stiffness.) The initial velocity was 50 mph, and the ramp rate of front-wheel steer angle was 1.5 deg/sec. As shown in the table, calculations were performed for two tractors and two semitrailers, each of which represented the specific vehicle units employed in the full-scale test program. Roll stiffnesses of the frame and the front suspension were varied over a full 7x7 matrix for each vehicle combination. The lowest value of each stiffness parameter represented the baseline state of both tractors. Additionally, calculations were made for four selected conditions with tractors outfitted with lug tires, rather than the baseline rib tire, on their drive axles.

Subsequent to conducting the first set of calculations on the International Harvester three-axle tractor with flat-bed trailer, the simulation output format was changed to provide magnetic tapes which subsequently facilitated calculation of the variables comprising the handling diagram. Thus, although the responses of the International Harvester/flat-bed trailer combination will not be included in the summary of handling diagrams presented here, inspection of the time history data reveals that virtually no distinguishable differences in performance can be seen between the International Harvester/flat-bed and International

Table 6.1. Tractor Parameters Varied to Examine Influence of Roll Stiffness Distribution on Yaw Stability.

Condition	Frame Torsional Stiffness, in-lb/deg	Condition	Auxiliary Front Roll Stiffness*, in-lb/deg
1a**	20,000 (baseline case)	1b**	0 (baseline case)
2a	40,000	2b	25,000
3a	60,000	3b**	50,000
4a	80,000	4b	75,000
5a**	100,000	5b**	100,000
6a	120,000	6b	125,000
7a**	140,000	7b**	150,000

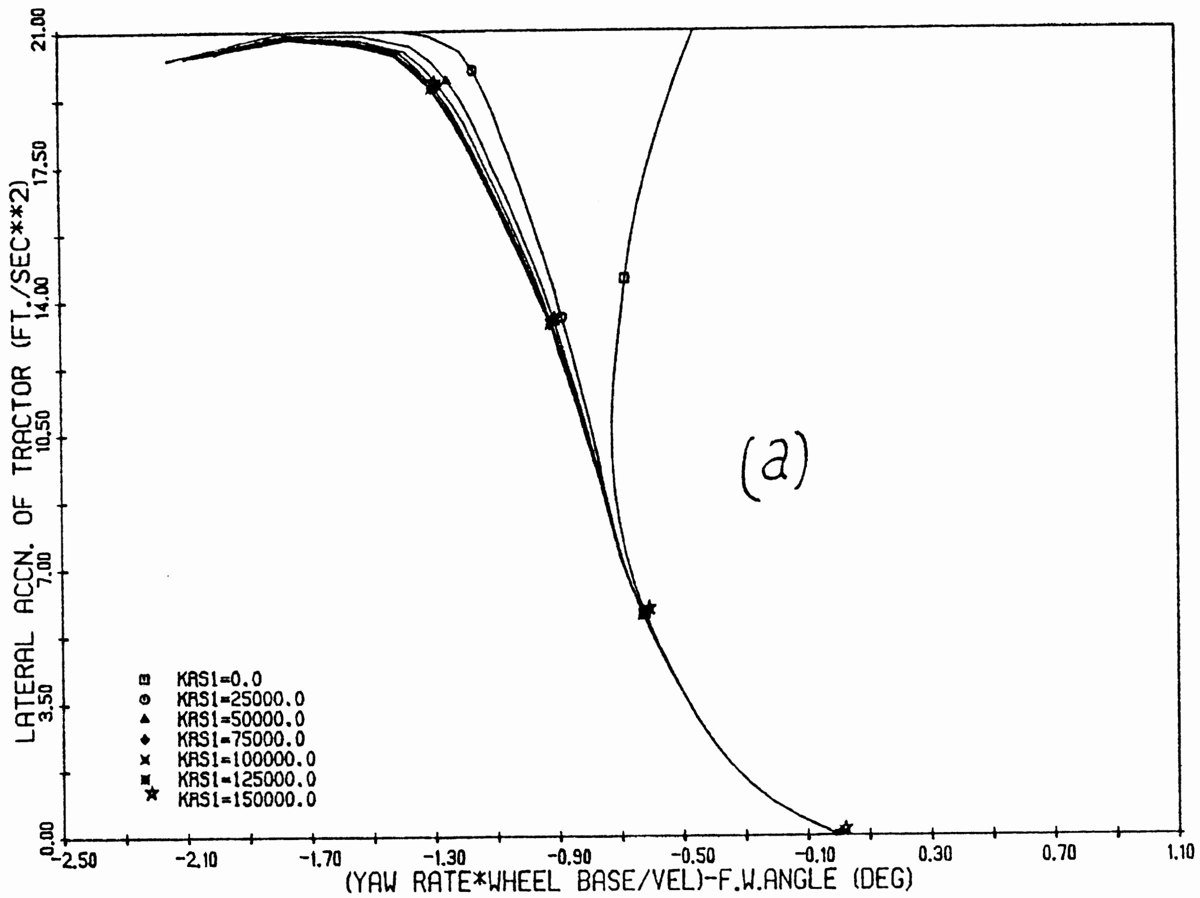
*Note [Baseline value of front suspension roll stiffness was: 9256 in-lb/deg (Ford two-axle tractor), 13,385 in-lb/deg (International Harvester three-axle tractor). The listed parameter is auxiliary, or in parallel with the suspension roll spring.]

**The indicated values were also employed in four additional calculations using baseline tractors equipped with lug tires on their drive axles.

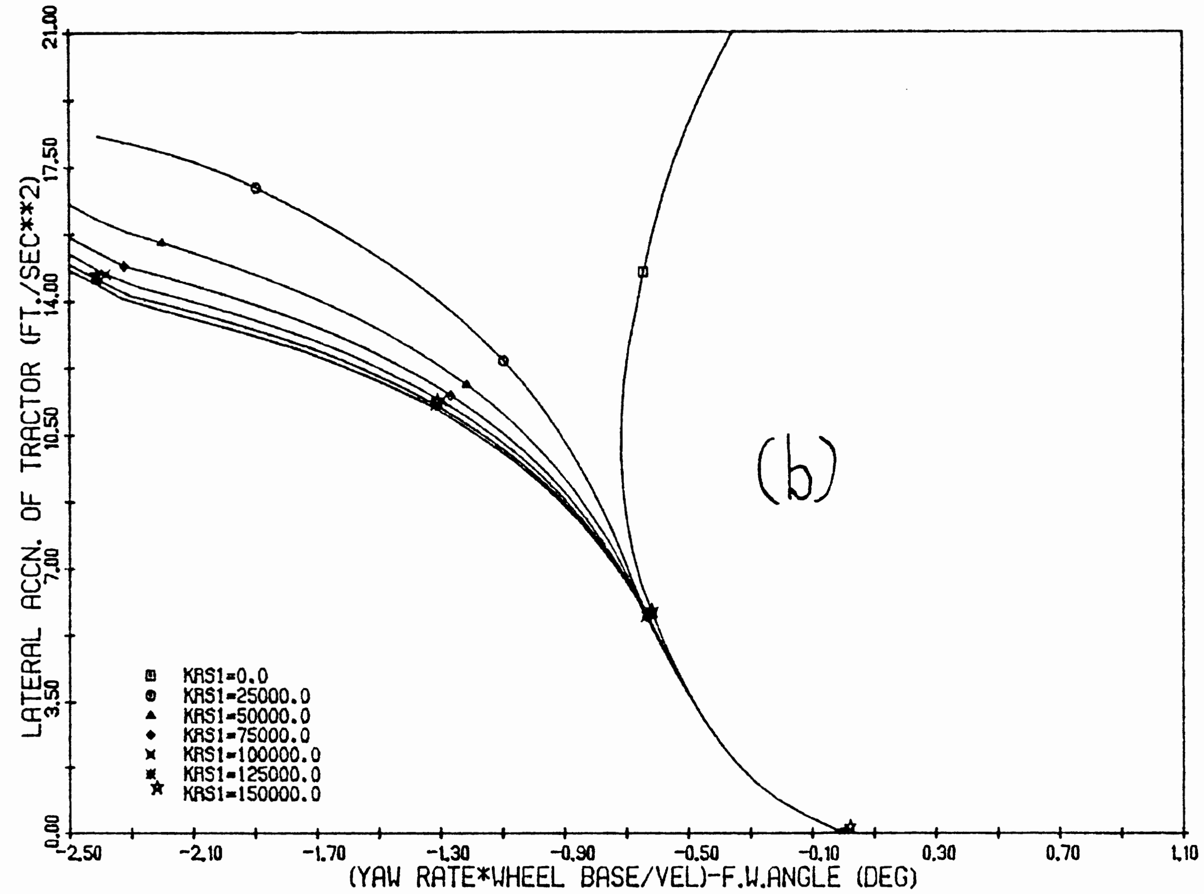
Harvester/van trailer combinations. This basic insensitivity to trailer type is also clearly evident in the results which are presented here for the Ford two-axle tractor in combination with each of the two trailers.

Shown in Figure 6.7 are examples of the range of handling curves obtained for the International Harvester/van semitrailer combination for the cases of (a) baseline frame stiffness and varied auxiliary front roll stiffness and (b) very high frame stiffness and varied front roll stiffness. Such individual displays of sensitivity to changes in frame and front suspension roll stiffness parameters have been summarized over the full matrix of cases for the International Harvester/van combination to produce the envelope of handling curves shown in Figure 6.8.

This figure shows the baseline handling curve plus each of the curves representing the greatest excursions in understeer that were observed when frame stiffener alone, front roll stiffener alone, and frame



IHC-VAN TRAILER,RIB TIRES,FRAME 20000 IN.LB/DEG



IHC-VAN TRAILER,RIB TIRES,FRAME 140000 IN.LB/DEG

Figure 6.7. Range of handling curves exhibited for (a) baseline frame stiffness and (b) very high frame stiffness, each with varied values of front suspension roll stiffness.

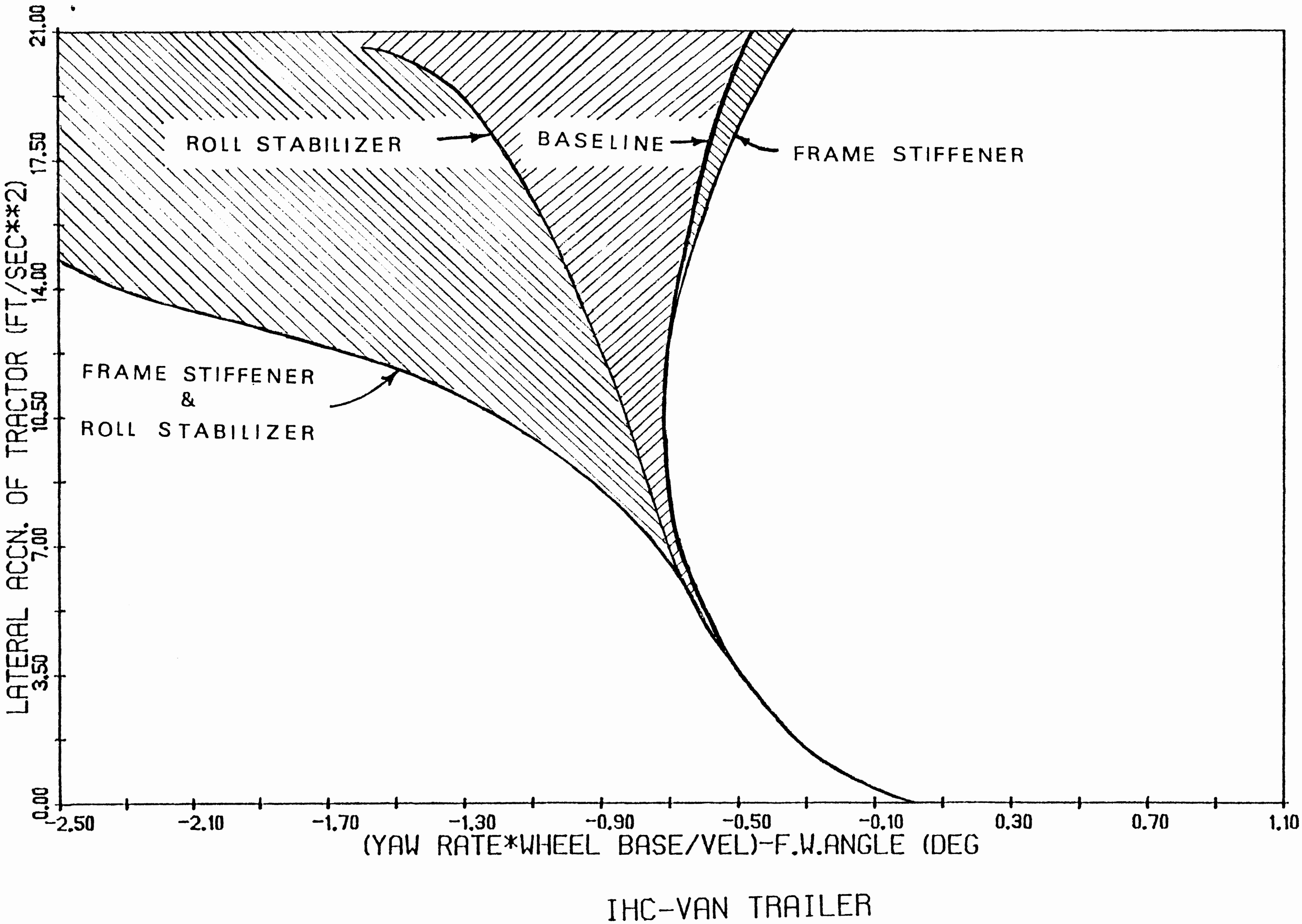


Figure 6.8 Envelope of handling curves obtained with varied stiffness parameters

and front roll stiffeners together were added to the vehicle. Note that the addition of a frame stiffener alone to this three-axle tractor introduces a slight change in the oversteer direction whereas the other modifications consistently produce increases in understeer level. (It should be understood that, theoretically, the influence of adding frame stiffener alone can cause either plus or minus changes in understeer level depending upon the relationship between the "front roll angle" and "rear roll angle" which accrue in the baseline configuration during a steady turn. In the case of Figure 6.8, apparently the addition of the frame stiffener served to reduce the front roll angle relative to the rear such that a greater portion of the tractor-mass-induced roll moment became reacted at the tractor's rear suspension.)

Figures 6.9 and 6.10 show similar gross trends (except for the frame stiffener-only case) for both combination vehicles involving the Ford two-axle tractor. Firstly, we see that the baseline two-axle tractor exhibits a substantially higher value of oversteer gradient than the three-axle tractor at high levels of A_y. Nevertheless, the first increment in front-suspension roll stiffness, alone, is sufficient to dramatically improve performance in the understeer direction. (Again, however, it must be noted that ramp-steer results produce higher apparent levels of understeer than will occur under steady-state conditions. Thus, the presented results should not be interpreted as providing estimates of absolute values of understeer for the "worst case," or steady-turn condition. Rather, these results describe relative scales of influence and provide a direct measure only for transient maneuvers that are suitably approximated by the 1.5 deg/sec ramp-steer input.)

By way of elaboration on this matrix of results which apply to tractors outfitted completely with rib tires (and, also upon considering the detailed diagrams, themselves, which are presented in Appendix V), we observe the following:

a) In every case, combined increases in frame and front-suspension roll stiffness effect an improvement in tractor yaw stability.

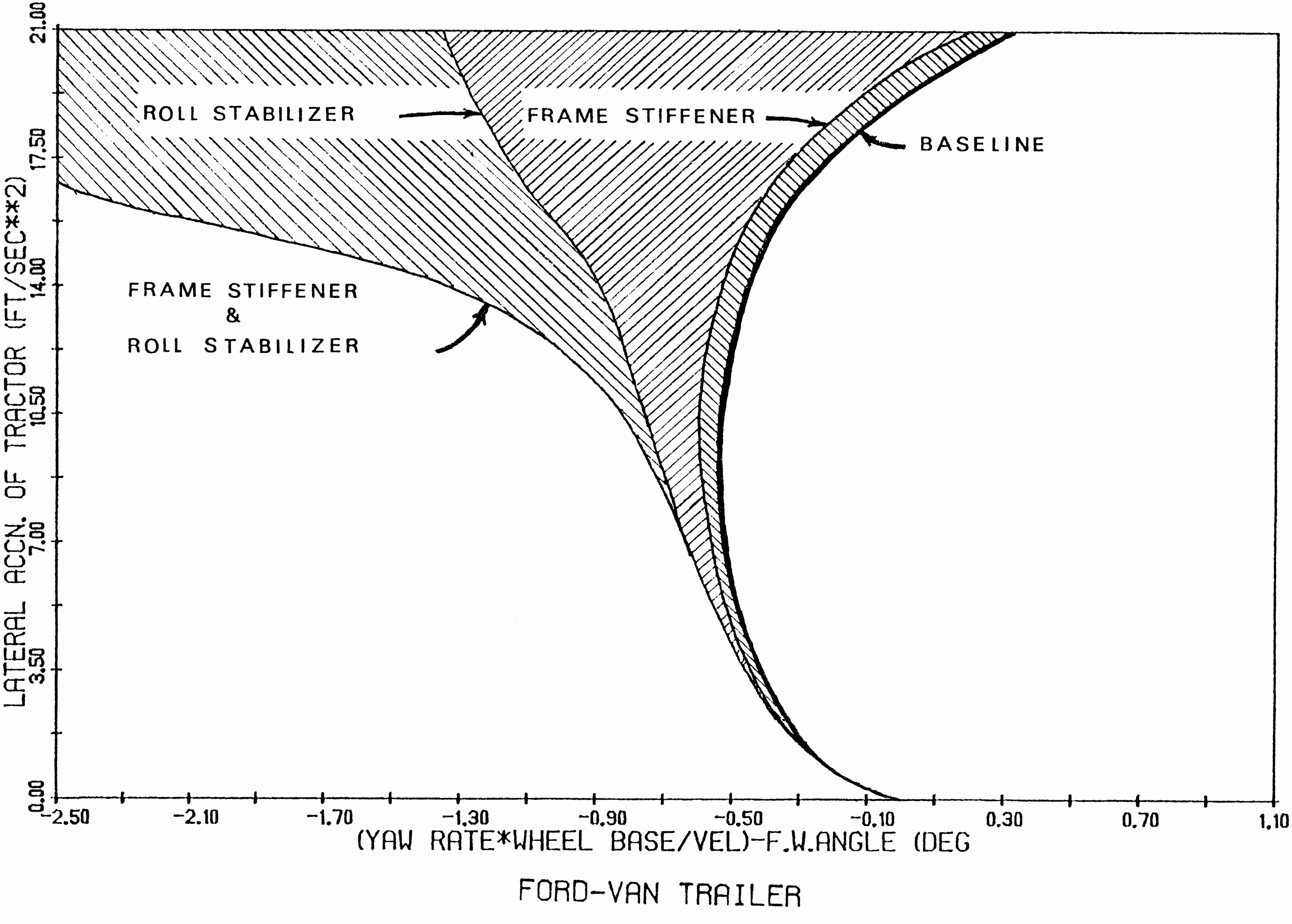


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Figure 6.9. Envelope of handling curves obtained with varied stiffness parameters.

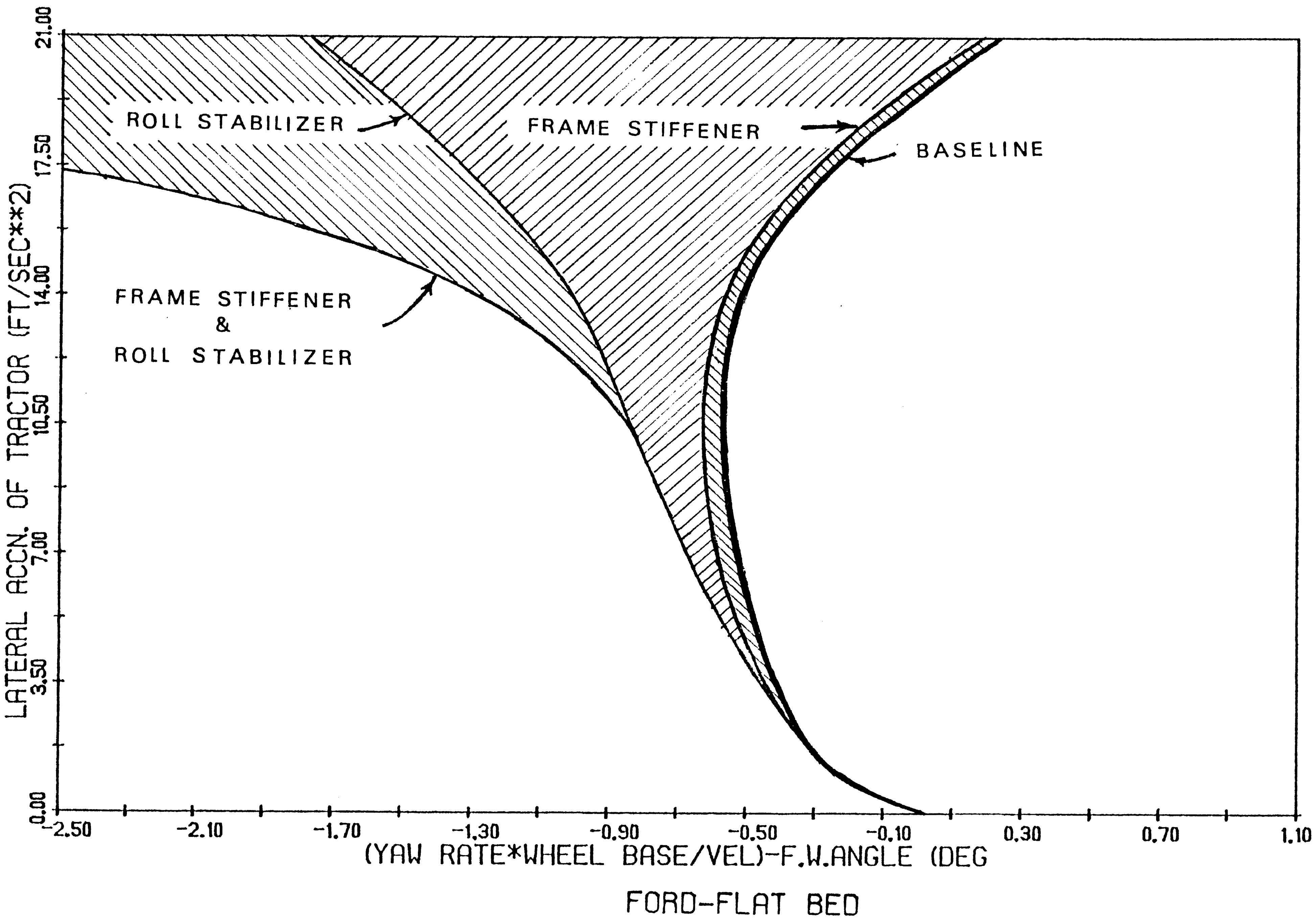


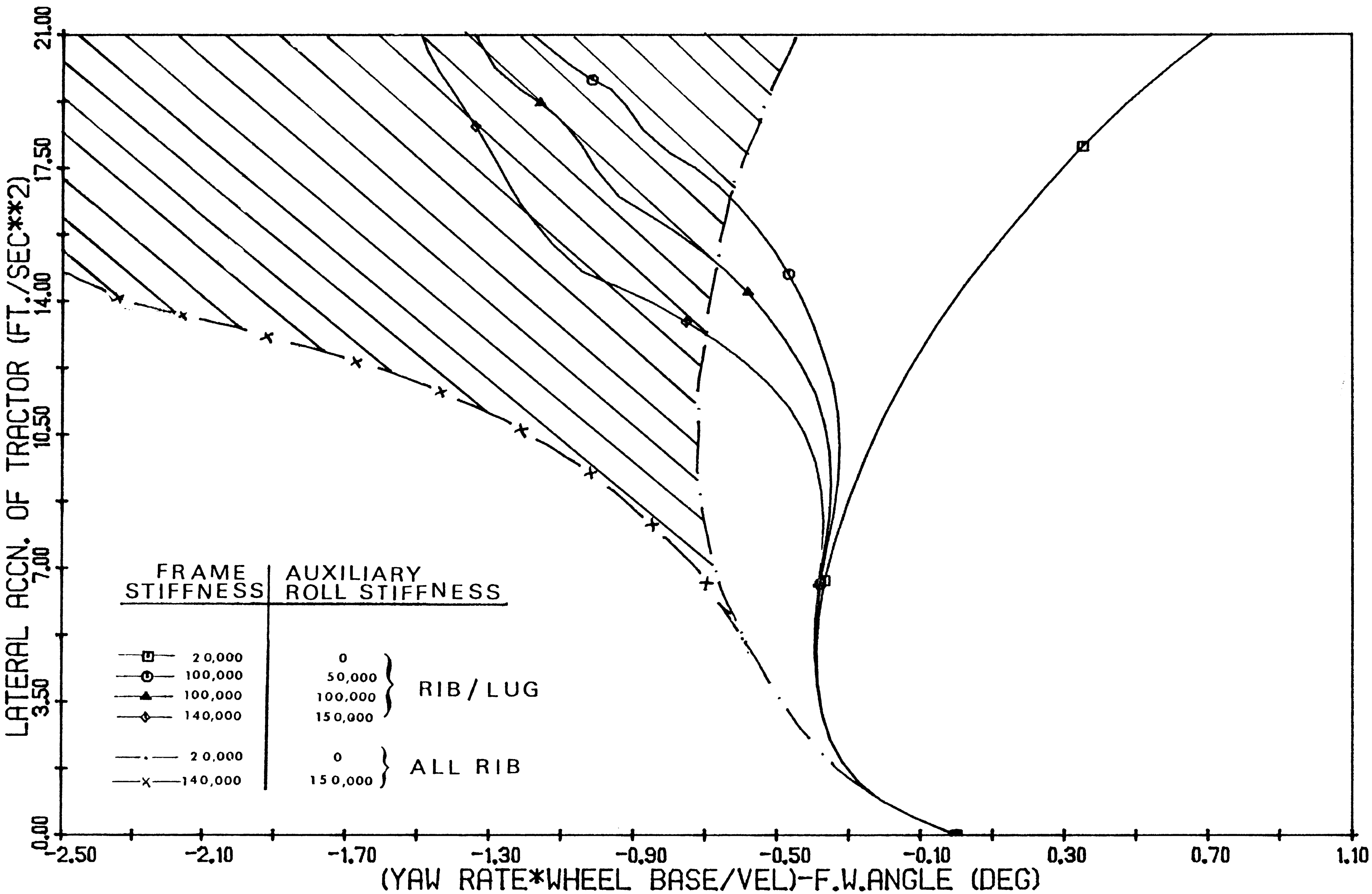
Figure 6.10. Envelopes of handling curves obtained with varied stiffness parameters.

b) Increases in front-suspension roll stiffness, taken alone, provide a much more powerful mechanism for improving yaw stability than do increases in frame stiffness alone.

c) Large increases in understeer level (at high levels of A_y) cannot be attained by increases in front-suspension roll stiffness alone. Rather, only combined increases in frame stiffness and front-suspension roll stiffness yield higher levels of understeer over the entire range of A_y.

d) The greatest portion of the possible increases in understeer level (tending to promote yaw stability) accrue within the first two or three levels of increase in both stiffness parameters examined in this study. The yield, in terms of increasing understeer level at high levels of A_y, most notably saturates in the case of the front-suspension roll stiffness parameter. A rapidly diminishing degree of further improvement is seen to obtain for levels of front-suspension roll stiffness above 50,000 in-lb/deg. When combined with a substantial increase in front-suspension roll stiffness, the further improvement afforded by increasing frame stiffness is seen to saturate such that 80% of the highest estimated level of understeer was obtained by a frame stiffness parameter of 80,000 in-lb/deg.

Shown in Figure 6.11 are four selected cases of the International Harvester/van trailer combination equipped with lug-type tires on the tractor's tandem axles. The use of lug-type tires on the drive axles is, of course, a very common practice in the U.S. and was shown previously [1] to seriously degrade the yaw stability of the tractor. These additional cases, which show the influence of increases in frame and front-suspension roll stiffness on a baseline vehicle that is less stable than that considered heretofore, are overlaid onto the overall envelopes shown earlier in Figure 6.8. Here we see that the expected trends in improved stability still accrue from the increases in the "stiffening"



IHC-VAN TRAILER

Figure C.11. Handling curves for tractor with mixed rib- and lug-tread tires, showing influence of selected variations in stiffness parameters.

parameters, although not without a peculiar transition zone of behavior in which the vehicle's yaw response goes from understeer to slightly oversteer and then back to understeer again.

The same basic behavior is seen in Figures 6.12 and 6.13, which show results obtained for the Ford two-axle tractor with lug tires on the drive axle, combined, respectively, with the van trailer and the flat-bed trailer. In these cases, the baseline vehicle is so heavily oversteer at high levels of A_y that a large degree of combined frame and front-suspension roll stiffness is needed to achieve even a marginally understeer condition.

Moreover, these calculations have served to "calibrate" the influence of changes in frame and front-suspension roll stiffness on relative tractor yaw stability. The simulation results generalize the finding which was demonstrated in the full-scale test data; namely, that combined increases in frame and front-suspension roll stiffness constitute a very effective mechanism for eliminating the potential for a yaw instability during cornering.

6.4 Simulation Study of the Prevalence of Yaw Instability Among Typical Tractor-Semitrailers

A second set of simulations was conducted to determine the extent to which different operating conditions and basic vehicle design configurations may render typical tractor-semitrailers capable of unstable yaw behavior in their fully-loaded condition. A matrix of runs was defined to cover the cases of two- and three-axle tractors coupled, respectively, to one- and two-axle trailers with which five selected parameters were varied to realistic degrees and in the directions which are hypothesized to degrade yaw stability. The baseline state of each vehicle was configured to incorporate typical values of those parameters which are expected to influence yaw stability. No parametric variations were made in the directions expected to improve stability since the goal was to identify the states in which stability is compromised. Thus, the results do not constitute an estimate of the yaw stability characteristics of the U.S. fleet of tractor-semitrailers but, rather, represent only

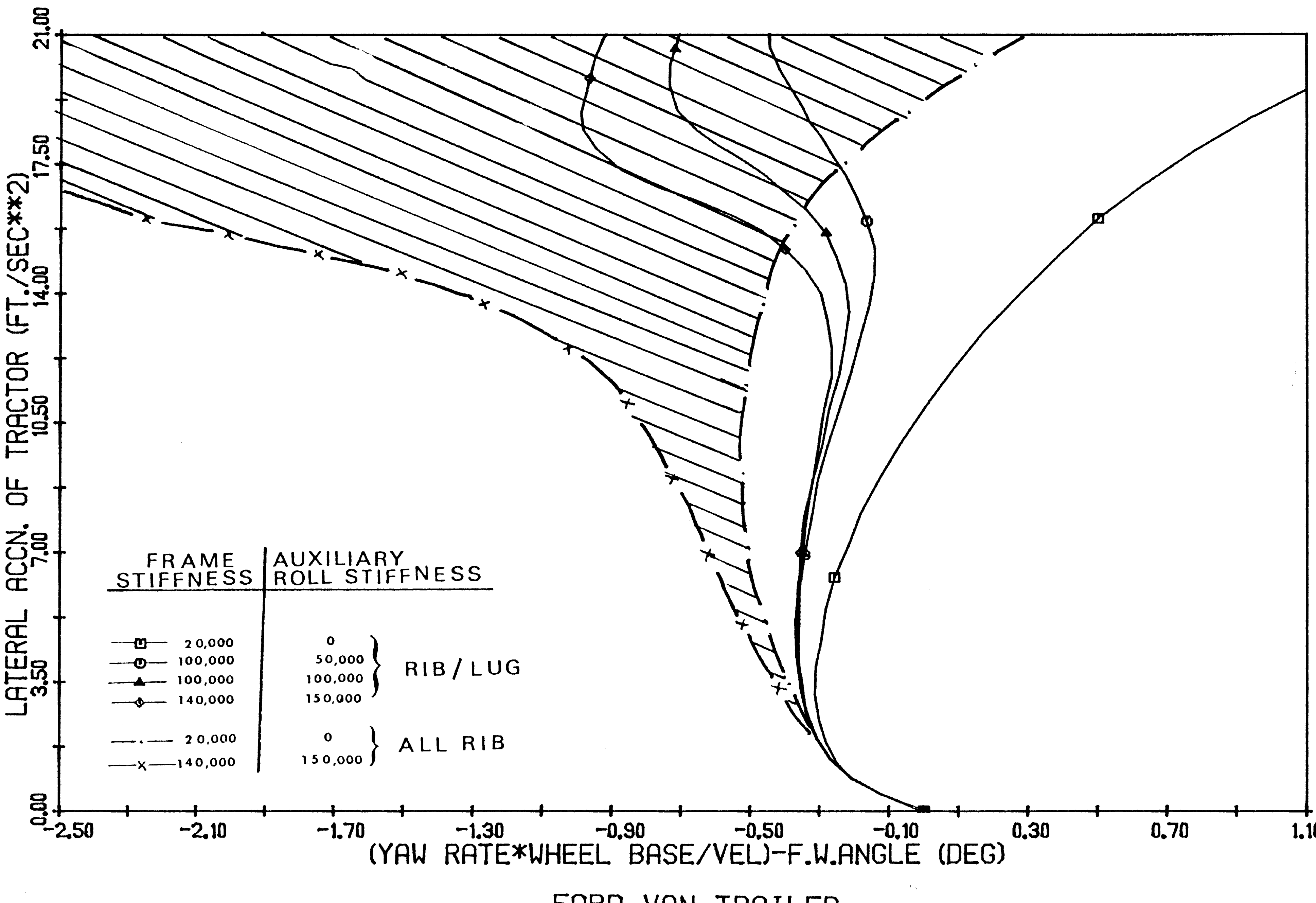
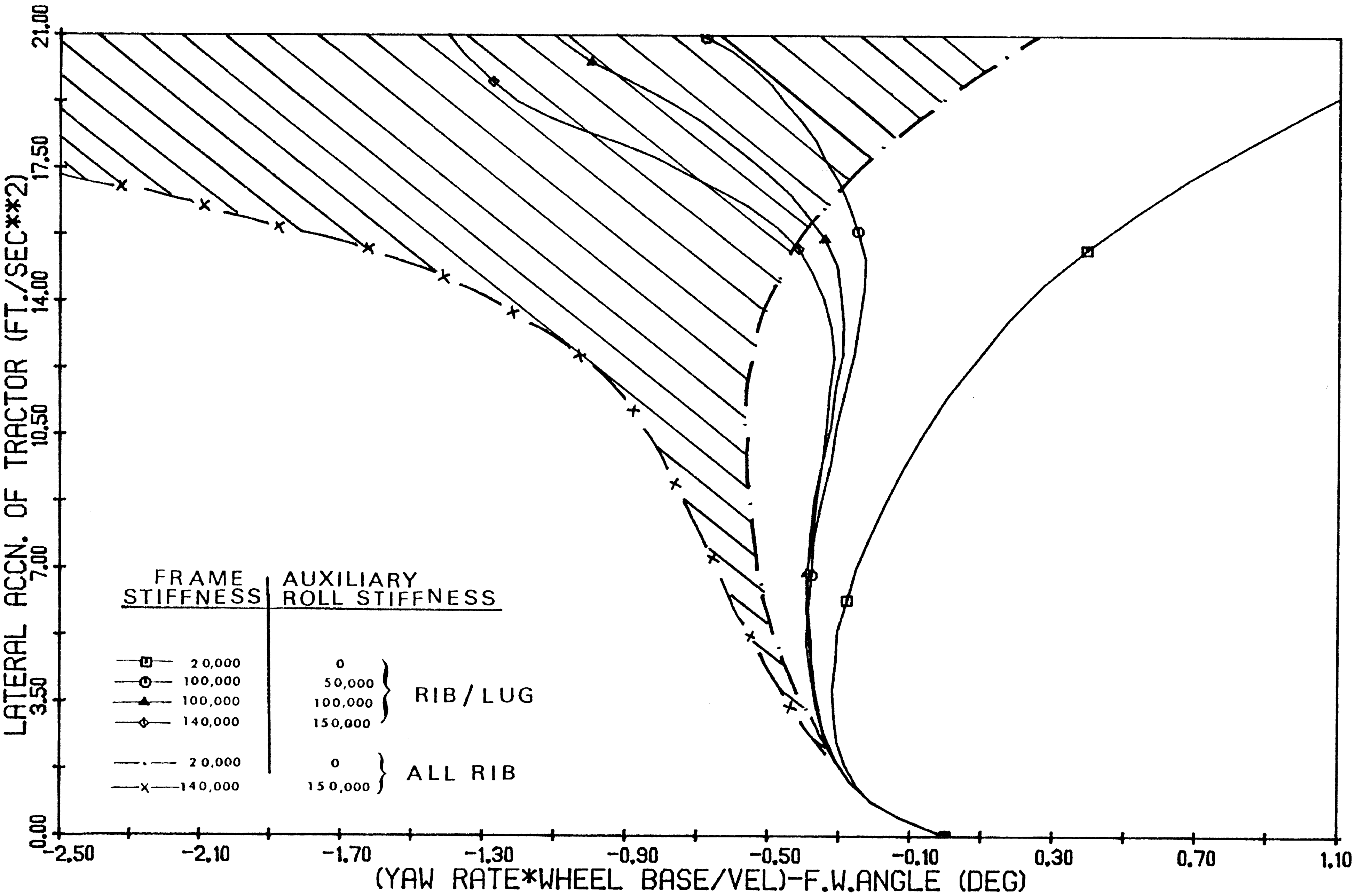


Figure 6.12. Handling curves for tractor with mixed rib- and lug-tread tires, showing influence of selected variations in stiffness parameters.



FORD-FLAT BED TRAILER

Figure 6.13. Handling curves for tractor with mixed rib- and lug-tread tires, showing influence of selected variations in stiffness parameters.

that portion of the fleet which lies on the "unstable side" of the more typical configurations (recognizing, again, that only fully-loaded vehicles are being considered so that the mechanisms most aggravating to rollover stability—especially the lateral load transfer mechanism—are emphasized). As will be shown, the results of this set of calculations suggest that the unexamined portion of the tractor-semitrailer spectrum can be presumed to be largely incapable of yaw divergence (in the described ramp-steer maneuver) prior to reaching the rollover limit, since the baseline vehicles are seen to be only marginally unstable at the examined condition of 50 mph.

Five baseline tractors were identified, namely, two two-axle cab-over-engine-type vehicles and three three-axle tractors, one of which was cab-over while the other two were configured as conventional cabs. The primary distinction between tractors having the same number of axles was the wheelbase dimension. As shown in Figure 6.14, the two-axle tractors had wheelbases of 110" and 140", whereas the three-axle tractors had wheelbases of 145", 165", and 200". The single- and tandem-axle semitrailers were described by parameters representing a torsionally rigid van-type construction.

The following variations in parameters were assumed:

a) Two arrangements of tractor tire properties, namely,

   1) a common installation of rib-tread bias-ply tires at all wheel positions (the baseline case)

   2) rib-tread bias-ply tires on the front wheels and lug-tread bias-ply tires on the rear wheels. (Tire properties were selected to represent a typical spread in the cornering force properties of rib- and lug-tread bias-ply tires.)

b) Two values of tractor roll stiffness distribution, namely,

   1) a typical front/rear distribution, assuming standard leaf-spring suspensions rated for the loads being carried (the baseline case)

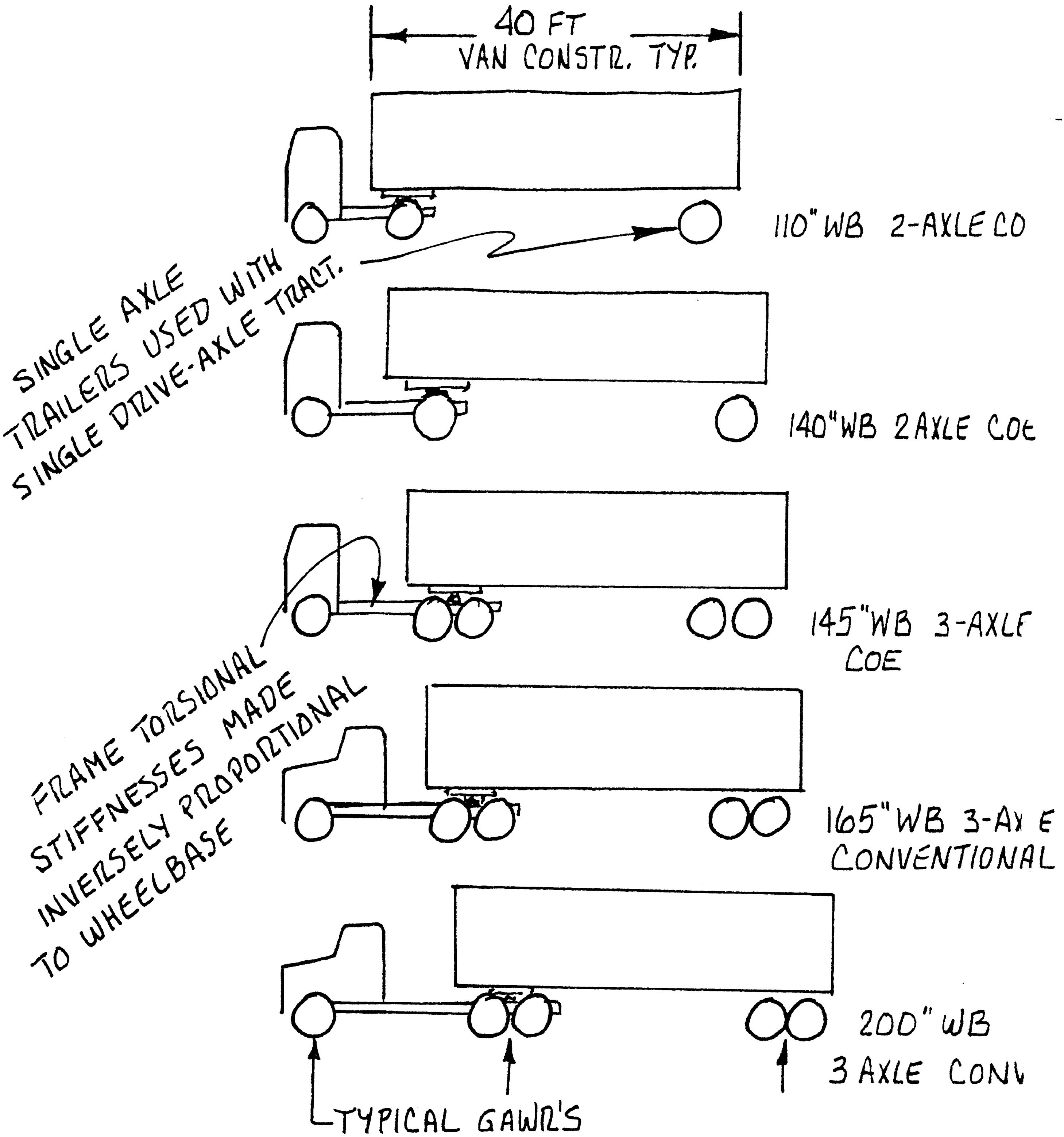


Figure 6.14. Vehicle combinations selected for "typical tractor-semitrailer" combinations.

2) a distribution more rear-biased than a standard configuration, as derives from the installation of a heavier-than-usual (although commonly employed) rear suspension.

c) Two values of fifth-wheel placement namely,

1) a forward position as yields a load distribution (on the two-axle tractor) of 10,500 lb. front/ 20,000 lb. rear and (on the three axle-tractor) a distribution of 12,000 lb. front/34,000 lb. rear (tandem) (the baseline case)

2) the fifth wheel located directly over the center of the rear suspension. [The trailer axle loads were selected to provide a constant level of either 20,000 lb. for a single-axle trailer or 34,000 lb. for a tandem-axle trailer. Since trailer payload weight was kept constant, the aft location of the fifth wheel represents an overload status on the tractor rear axles (per the load allowances that apply to the federal interstate highway system). This arrangement was selected, however, to represent the illegal, but rather common, circumstance which develops when the operator sets up to run legally but then shifts the fifth wheel aft to improve ride quality.]

d) Two values of trailer suspension roll stiffness, namely,

1) a typical value representing common leaf-spring suspensions used on trailers (the baseline case)

2) a low value representative of the roll stiffness found in some air spring suspensions employed on trailers.

e) Three values of payload c.g. height, namely,

1) 68" (the baseline case)

2) 78"

3) 88" covering cases in which trailers are loaded with low density freight.

A detailed listing of the parameters describing each vehicle configuration diagrammed in Figure 6.14 is given in Appendix II. The full matrix of parameter variations examined for each of the five baseline tractor-semitrailers is shown in Figure 6.15. Individual handling curves representing vehicle response for each of the above configurations are presented in Appendix V. These data have also been condensed into "summary handling diagrams" illustrating the total envelope of results obtained for each of the five basic vehicles. Figure 6.16 summarizes the parametric sensitivities examined for the 110"-wheelbase, two-axle tractor combination and Figure 6.17 does the same for the 140"-wheelbase tractor. The summary diagrams each show four individual curves. The curves define the performance of the "most stable" and "least stable" configurations for the cases of (a) tractor outfitted with rib tires and (b) tractor outfitted with the rib-front, lug-rear tire mix. Although at first glance the overall band of response characteristics appears rather narrow, it will be shown later that the range of critical acceleration levels, A_{y_c}, is indeed significant. The important aspect of the indicated sensitivities lies in the level of A_y at which the slope becomes sufficiently flat in the positive direction to yield an instability at the simulated speed of 50 mph. We see that the mix of rib and lug tires on the tractor constitutes the single most powerful mechanism for creating a low, positive slope at the lowest level of A_y. All other parametric variations (i.e., excluding the case of the rib/lug mix) bring about a lesser excursion in response away from the most stable case. In the calculations performed for each vehicle, the baseline configuration always constituted the most stable case.

Shown in Figures 6.18, 6.19, and 6.20 are the summary handling diagrams with curves defining the most and least stable configurations obtained with the two types of tire installations on each of the three-axle tractors with wheelbases 145", 165", and 200", respectively. On examining these three plots, it can be seen that very little difference in the handling curve is obtained as a function of wheelbase. Thus, any one of the three vehicle configurations represents a good approximation of any other. Clearly, each three-axle tractor produces a baseline behavior which is significantly more stable than that observed with

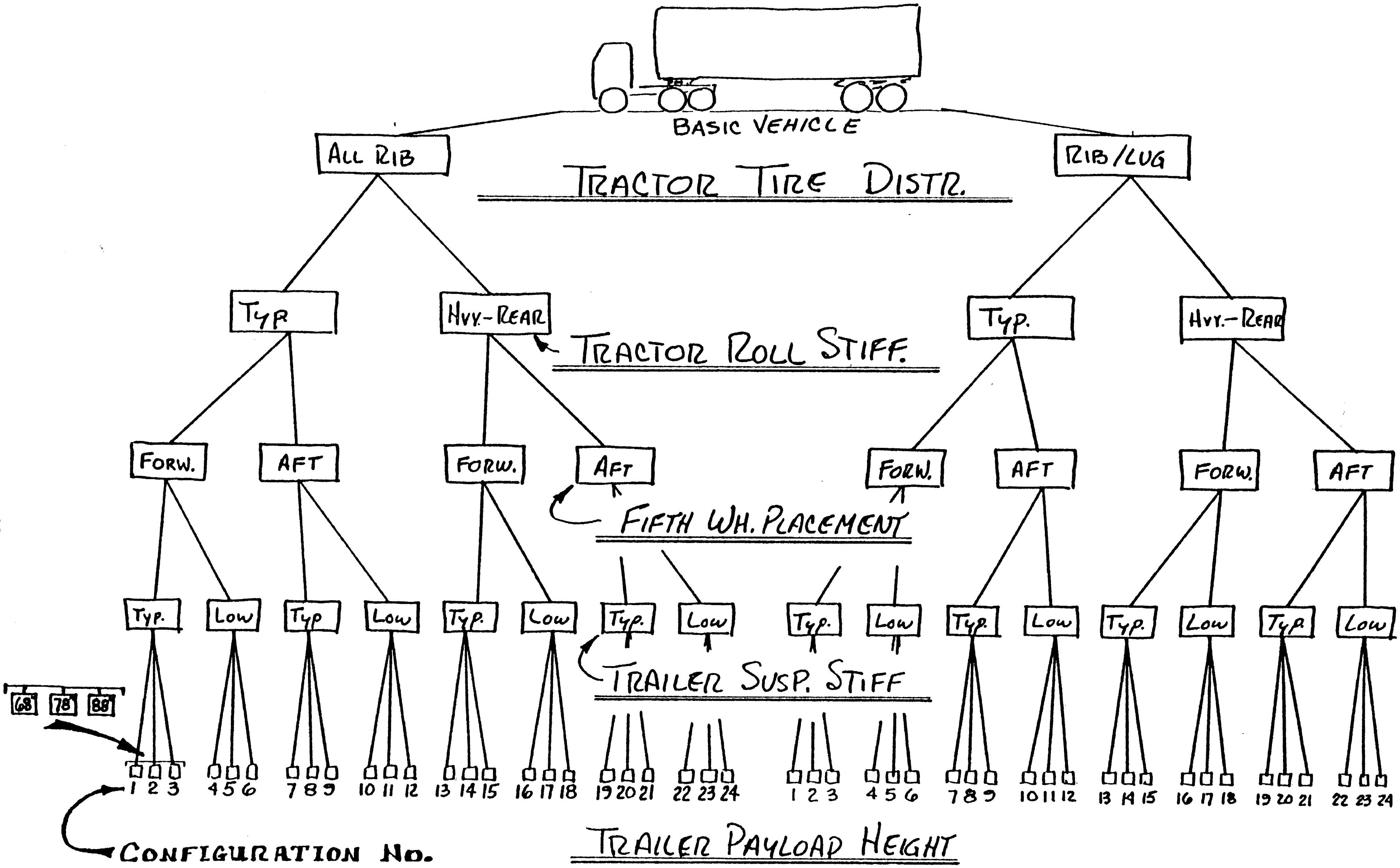


Figure 6.15. Parameter variations examined for each of the five baseline tractor-semitrailers.

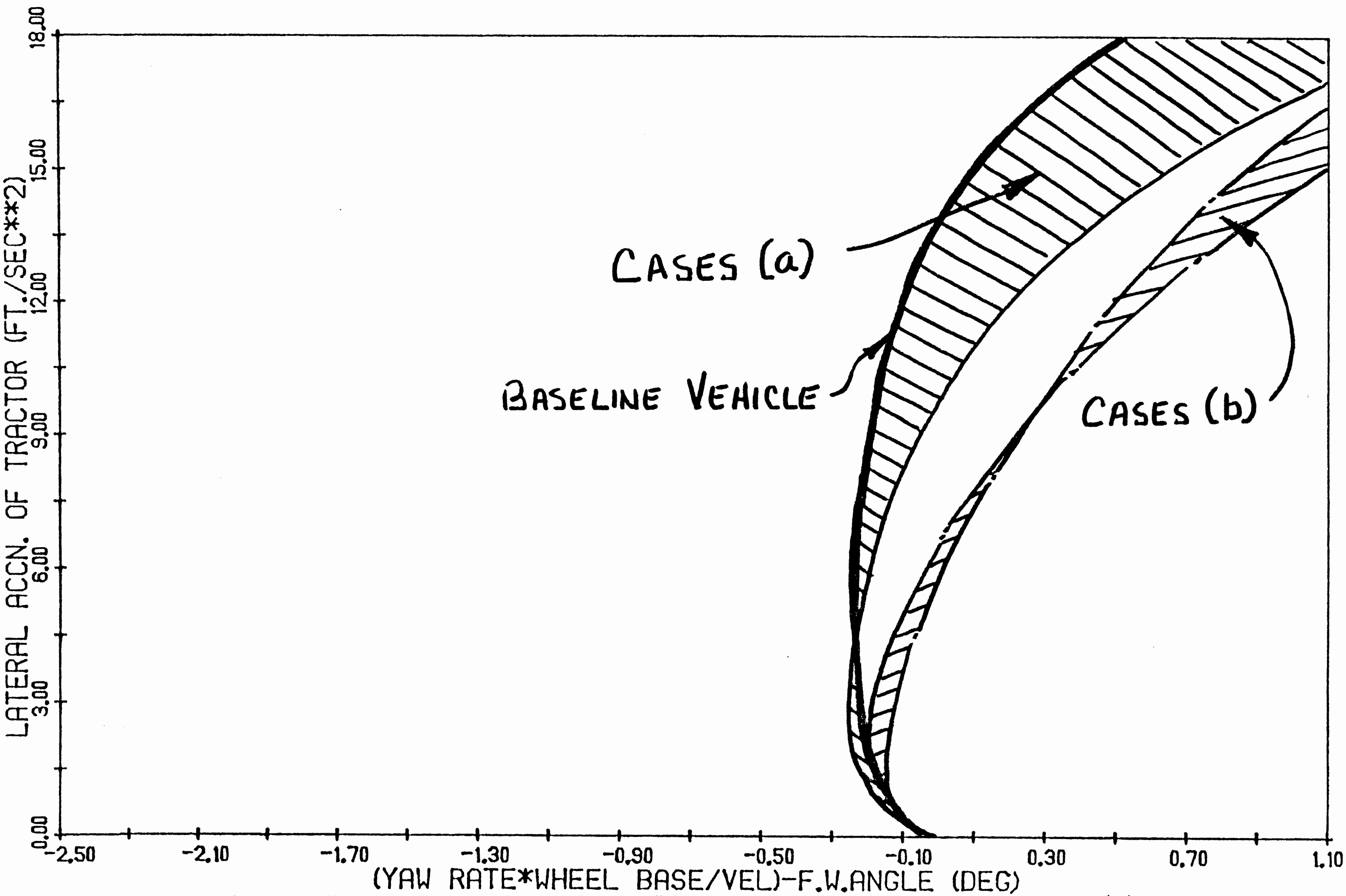


Figure 6.16. Handling curve envelopes for 110"-wheelbase two-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

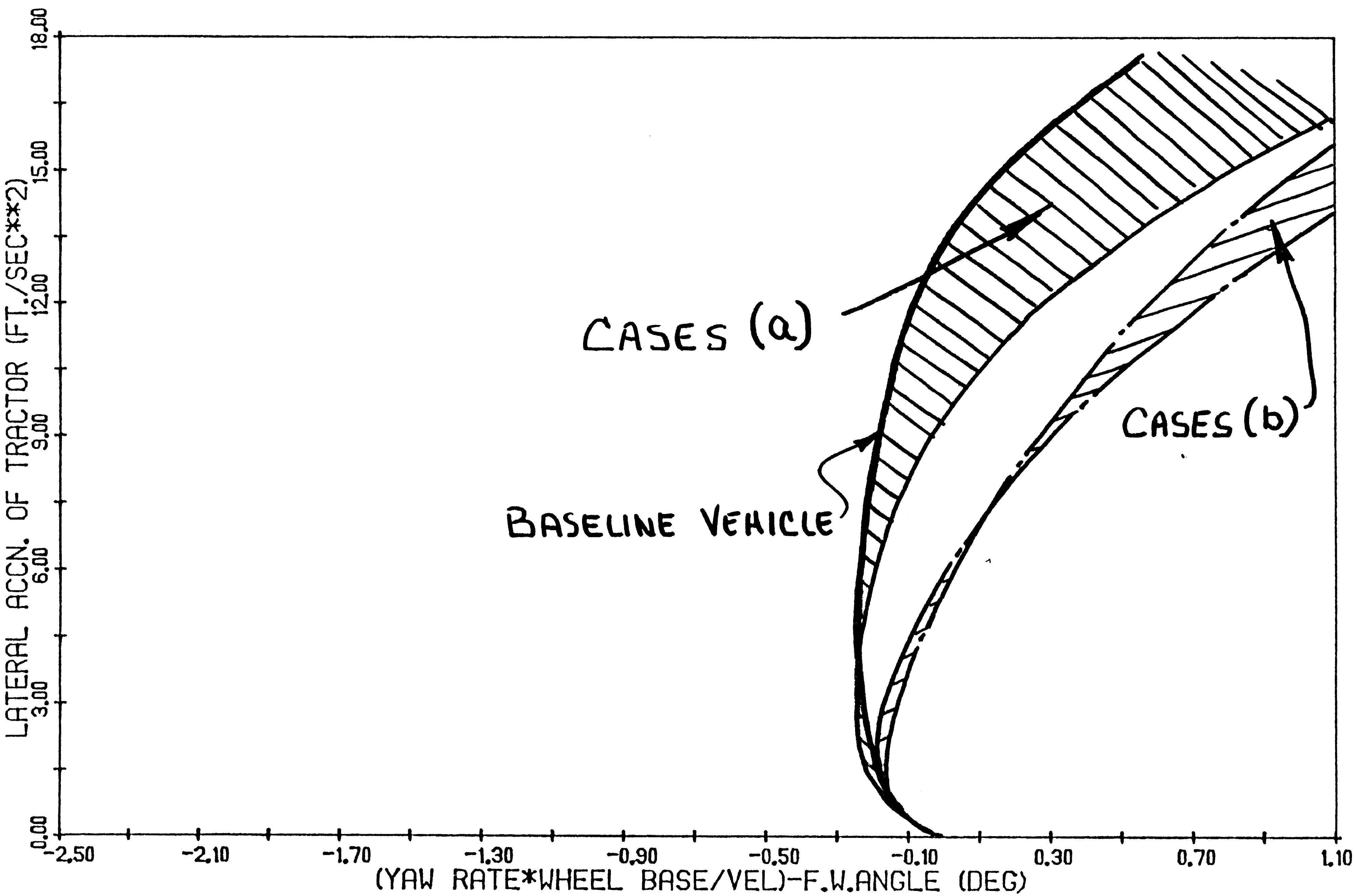


Figure 6.17. Handling curve envelopes for 140"-wheelbase, two-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

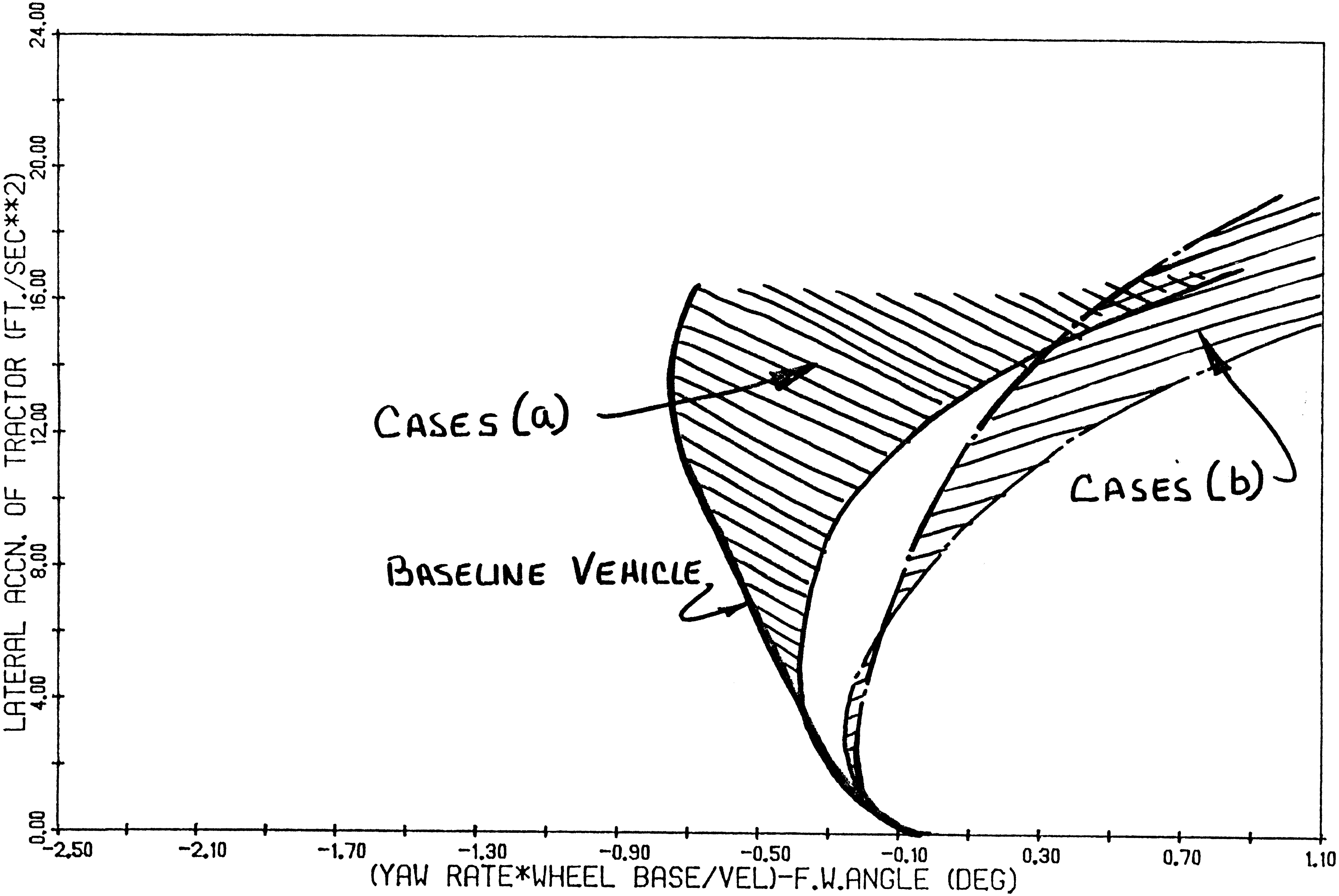


Figure 6.18. Handling curve envelopes for 145"-wheelbase, three-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

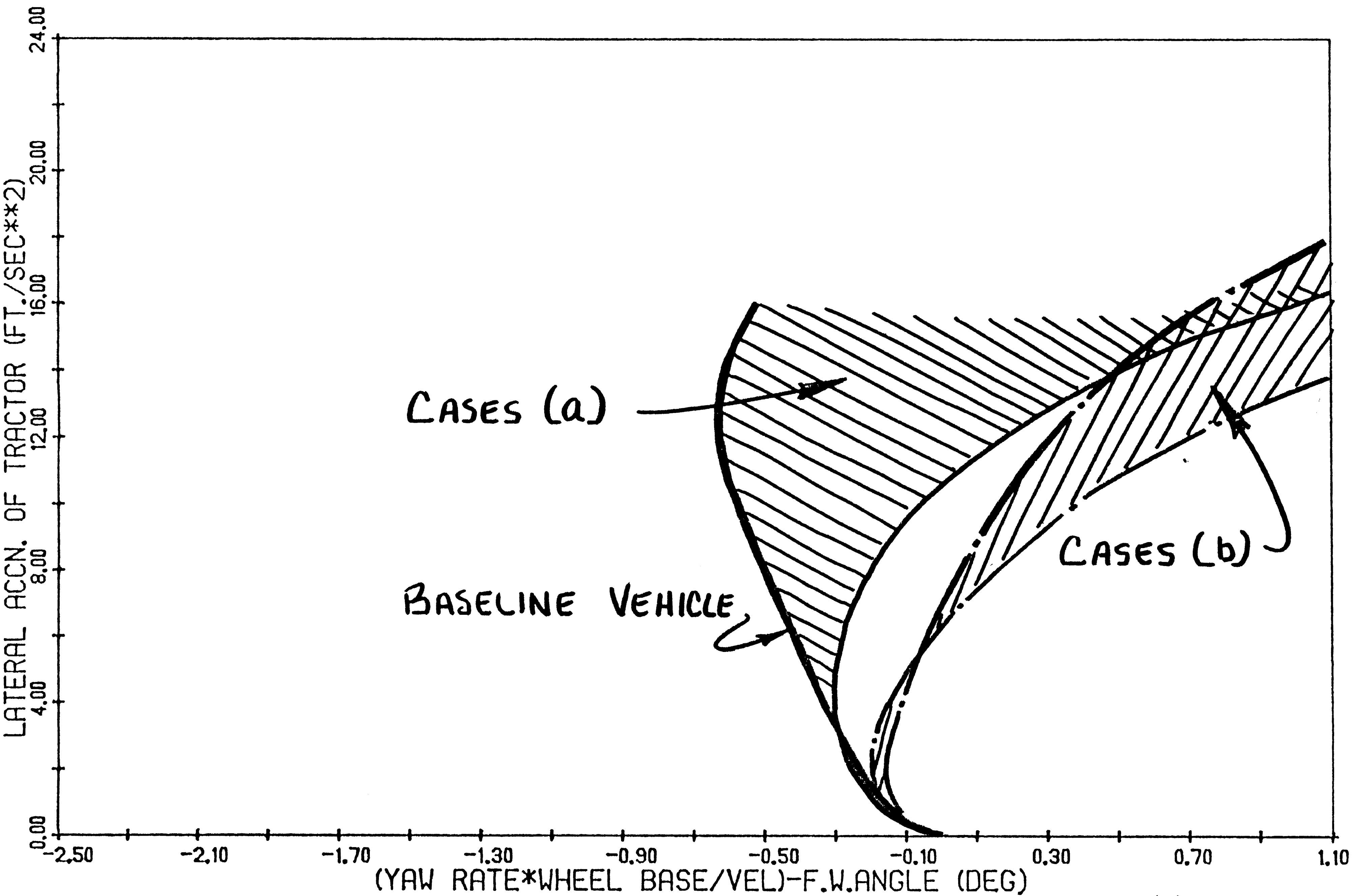


Figure 6.19. Handling curve envelopes for 165"-wheelbase, three-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

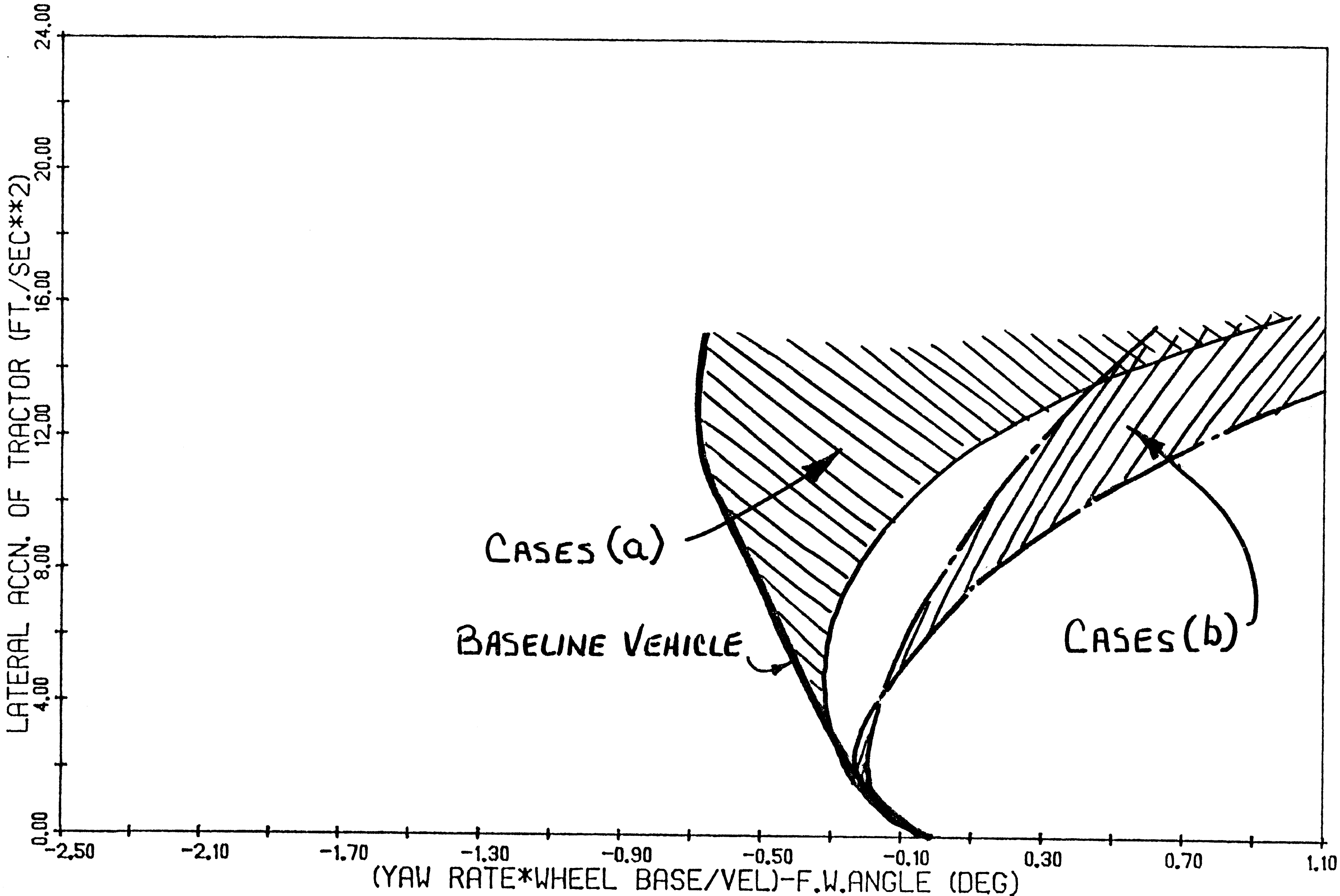


Figure 6.20. Handling curve envelopes for 200"-wheelbase, three-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

the two two-axle tractors. However, it is also evident that stability degrades more significantly, in the case of three-axle tractors, as a result of the combined parametric changes. Again, a rib/lug tire mix is the single most powerful parametric change degrading stability.

Perhaps the most direct means of summarizing the results of this parameter sensitivity exercise constitutes the tabulation of A_{y_c} values for each simulated condition. In Table 6.2, values of A_{y_c} are tabulated for each tractor operated in both the rib-tire and rib/lug-tire mix configuration. Twenty-four values of the critical acceleration measure are shown in each column, corresponding to the twenty-four conditions defined in Figure 6.15.

Examination of Table 6.2 reveals the following:

1) Stability performance levels range from a high of "fully stable" (that is, no potential for yaw instability in a 50-mph ramp-steer maneuver) in the case of the three baseline three-axle tractor combinations to a low value of A_{y_c} = 9.1 ft/sec^2 (.28 g) for the case of the short-wheelbase, two-axle tractor with multiple "degraded" parameter values.

2) The general observation is that degraded levels of stability did indeed prevail with each of the individual parameter changes, as hypothesized. In general, we see that A_{y_c} reduces as the "configuration number" increases, except that payload c.g. height causes a cycle that repeats every three configuration numbers. (Note that the parametric variation matrix, as diagrammed in Figure 6.15, was laid out such that configuration #1 would, presumably, constitute the most stable case whereas configuration #24, presumably, constituted the least stable case in each tire installation group.)

3) A change in an individual parameter is seen to cause a decrement in critical acceleration ranging from 0 to 4.4 ft/sec^2. Averaging the decrements in A_{y_c} which accrue from

Table 6.2. Values of A_{y_c} Obtained Through Parametric Variations (See Configuration No.'s in Figure 6.15).

Configuration Number	110"-W.B., 2-Axle		140"-W.B., 2-Axle		145"-W.B., 3-Axle		165"-W.B., 3-Axle		200"-W.B., 3-Axle	
	Rib	Lug	Rib	Lug	Rib	Lug	Rib	Lug	Rib	Lug
1	14.7	12.2	15.6	13.6	*	16.4	*	16.0	*	16.8
2	13.7	11.4	14.4	12.4	*	15.6	*	15.0	*	16.0
3	12.3	11.0	13.7	11.7	16.8	14.6	16.4	13.8	16.8	14.8
4	13.6	10.8	14.3	12.6	16.8	14.3	16.0	13.6	16.4	14.8
5	12.4	10.2	13.0	11.7	15.2	12.6	14.8	12.0	15.2	13.6
6	11.2	9.3	11.9	10.5	14.0	11.2	13.4	10.8	14.1	12.4
7	14.5	10.1	15.2	12.0	18.0	16.0	17.2	15.4	18.0	15.8
8	13.7	10.1	14.3	11.5	17.0	15.2	16.4	14.4	17.5	15.2
9	12.9	10.1	13.4	10.6	16.4	14.4	15.6	14.0	16.8	14.4
10	13.4	9.5	14.2	11.4	16.0	14.4	15.4	13.8	16.0	14.1
11	12.7	9.5	13.3	10.8	14.7	12.8	14.4	12.4	15.0	12.8
12	11.8	9.5	12.4	10.1	13.9	12.0	13.2	11.6	14.1	12.1
13	13.8	11.4	14.4	12.7	17.9	15.1	17.4	14.4	*	15.6
14	12.7	10.8	13.4	11.8	16.0	14.0	15.7	13.2	16.4	14.4
15	11.6	10.2	12.3	10.8	15.2	13.0	14.7	12.4	15.6	13.3
16	12.7	10.5	13.7	12.2	15.7	13.1	15.2	12.4	16.0	13.6
17	11.6	9.8	12.3	11.0	14.3	11.8	13.6	11.4	14.4	12.4
18	10.7	9.1	11.3	10.5	12.8	10.8	12.4	10.3	14.0	11.5
19	13.8	9.8	15.0	11.8	16.5	15.1	16.0	14.7	17.0	15.0
20	13.4	9.8	13.8	11.3	15.7	14.0	15.2	13.6	15.6	14.4
21	12.3	9.8	13.0	10.3	14.8	13.2	14.4	12.6	14.8	13.5
22	12.9	9.3	14.1	11.1	15.0	13.4	14.4	13.0	15.3	13.6
23	12.4	9.3	12.8	10.5	14.0	12.4	13.4	11.8	14.3	12.6
24	11.7	9.3	11.8	9.8	13.2	11.3	12.4	10.8	13.2	11.5

*$A_y)_{crit}$ was not reached within the 4-second simulation time.

individual parameter changes, over all five vehicle combinations, we obtain the following "average decrements," $\overline{\Delta A}_{y_c}$:

a) Distribution of tire types on tractor

$\overline{\Delta A}_{y_c}$ = 2.4 ft/sec^2

b) Roll-stiffness distribution on tractor

$\overline{\Delta A}_{y_c}$ = 0.8 ft/sec^2

c) Fifth-wheel placement

$\overline{\Delta A}_{y_c}$ = 0.2 ft/sec^2

d) Trailer roll stiffness

$\overline{\Delta A}_{y_c}$ = 1.6 ft/sec^2

e) c.g. height of trailer payload

$\overline{\Delta A}_{y_c}$ = 1.8 ft/sec^2

These variations in stability level cannot be conveniently normalized and must be viewed simply as the characteristic degradation in stability which can be expected to accrue, if and when vehicles (of the types described herein) are altered from their baseline condition by means of the parameter changes summarized in Figure 6.15.

4) Whereas changes in individual parameters, in general, produced a consistently monotonic decrement in stability (despite the cross-influence of other parametric combinations) a notable exception was fifth-wheel placement in combination with a change in tire distribution on the tractor. For the two-axle tractors, the placement of the fifth wheel <u>aft</u> virtually always <u>improved</u> the stability of the vehicle with rib tires installed while virtually always degrading stability for the same vehicle with mixed rib- and lug-type tires. Curiously, however, this anomaly is reversed for three-axle tractors, with the rib tire installation degrading stability and the rib/lug mix improving stability as the fifth wheel is moved aft.

Moreover, this particular parameter sensitivity study has revealed that the yaw stability of a tractor-semitrailer can be reduced to a remarkably low level through variations in operating conditions and design variables which are known to be relatively commonplace.

7.0 FINDINGS AND RECOMMENDATIONS

The focus of this study has been the yaw stability of tractor-semitrailers as can be challenged in steering-only maneuvers on dry surfaces. In this narrow context, findings will be summarized pertaining to the following topics:

1) the basic nature of the phenomenon of tractor-semitrailer yaw instability

2) the influence of design and operating variables on yaw stability

3) engineering methods suited to the evaluation of tractor-semitrailer yaw stability.

7.1 Findings Regarding the Basic Nature of Tractor-Semitrailer Yaw Stability

•The instability at issue involves the yaw response of the tractor alone. Analysis shows that a yaw instability of the tractor element is necessary for any truly divergent behavior of a tractor-semitrailer vehicle to be manifested in response to steering-only inputs. Although the semitrailer can exhibit high-speed off-tracking or can proceed through a divergent yaw motion as a consequence of a tractor divergency, no divergent yaw response of the semitrailer, alone, is possible (again, for the case of steering-only maneuvers on a horizontal surface).

•Yaw instability of the tractor can occur at a maneuvering level in which tire side forces are still well below saturation (i.e., friction-limited) conditions.

•*In such low-level maneuvers, tractor yaw instability is precipitated primarily through a mechanism which has long been recognized in the vehicle dynamics literature as applies to passenger cars. The mechanism is much more likely to be encountered by heavy trucks and road

*Note - this conclusion is basically a restatement of the finding of a preceding study [1].

tractors, however, because of the peculiar requirements imposed by the load-carrying mission of the commercial vehicle. Because of the large range of loads to be carried on rear suspensions, we see that such vehicles are characteristically designed with very stiff rear suspensions, giving rise to a very strong rearward bias in the distribution of roll stiffness. A destabilizing mechanism derives because, during cornering, the higher roll stiffness of the rear suspension yields a change in load between left and right tires which is greater at the rear than at the front, thus producing a greater net loss in the side forces being provided by rear tires. This "net loss," in turn, is due to the nonlinear property of the pneumatic tire by which changes in side force are caused by changes in vertical load at a given slip angle. While this mechanism is the primary means by which a road tractor can exhibit yaw instability at a low maneuvering level, other interactive mechanisms also exist which can either raise or lower the net destabilizing effect described above.

•The "premature" destabilization of the tractor can be looked upon as effectively narrowing the usable maneuvering range of the vehicle. Referring to Figure 7.1, the range of steady-state lateral acceleration over which the tractor-semitrailer may be controllably operated is limited either by the rollover limit or by a yaw stability limit. When a potential for yaw instability exists at a maneuvering level which is well below the rollover threshold, that characteristic could be presumed to degrade the overall safety quality of the vehicle.

•The maneuvering condition which most challenges the yaw stability of a tractor-semitrailer (in response to steering inputs only) is the steady-state turn.

•Transient steering maneuvers pose lesser demands on tractor yaw stability because of the lag in trailer response, thereby yielding attenuated levels of roll moment and side force being reacted by the tractor at the fifth wheel.

•The most likely sites for accidents to occur as a result of tractor yaw instability are short-radius curves on secondary roads, or

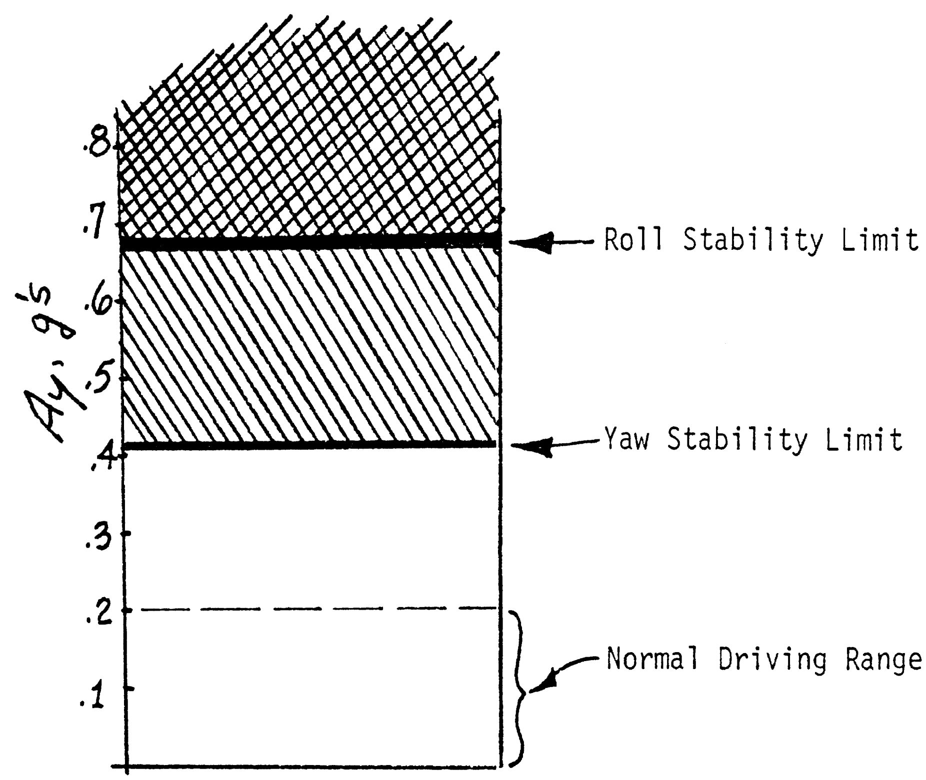


Figure 7.1. Example tractor maneuvering range.

exit ramps on interstate and other limited-access highways. A likely outcome in such accidents will be rollover of the vehicle combination, thereby rendering a complicated physical scene which may confuse the task of reconstructing the causal role that may have been played by a yaw instability.

7.2 The Influence of Design and Operating Variables on Tractor Yaw Stability

•Torsional deflection of the tractor frame and roll deflection of the front suspension can be looked upon as two spring functions which are in series between the fifth-wheel coupling and the front axle. Increases in the stiffness of both of these springs, in combination, constitutes a powerful means to improve tractor yaw stability.

•If the above two spring stiffnesses were increased one at a time, on a typical road tractor, the roll stiffness of the front suspension would contribute a many times greater improvement in yaw stability, per unit change in stiffness, in-lb/deg.

•Increased frame stiffness, alone, will produce small (and generally negligible) changes in the yaw stability of a typical road tractor. Such changes in yaw stability can be either "positive" or "negative," depending upon the other design parameters of the baseline vehicle.

•Great variations in tractor yaw stability are expected to occur within the existing commercial vehicle fleet as a result of:

a) Commonly selected options (including, at least, tractor suspension stiffnesses, mixed installation of tires of differing construction type, wheelbase dimensions, number of tractor axles, and trailer suspension roll stiffnesses).

b) Operating practices (including, at least, fifth wheel placement and trailer payload placement).

Certain combinations of vehicle design options and operating practices were seen to effectively reduce the "usable maneuvering range" to less than half of the range which would be otherwise available within the rollover threshold.

•Among the various vehicle options and operating practices, which are seen to degrade tractor yaw stability, a worst-case combination would include the following features:

1) two-axle tractor

2) short wheelbase (e.g., 105"-120")

3) low front suspension rating (e.g., 9,000 lb)

4) high rear suspension rating (e.g., 23,000 lb)

5) mixed tire installation resulting in high cornering stiffness tires at the front (e.g., radials front/ bias-ply lug-tread tires rear)

6) fifth wheel located in an aft position (e.g., directly over the rear-axle center)

7) trailer fully loaded with payload set high (e.g., payload c.g. height at 85" to 95" above the ground)

8) trailer with low roll stiffness suspension (e.g., certain air suspensions incorporating low levels of auxiliary roll stiffness).

7.3 Engineering Methods Suited to the Evaluation of Tractor-Semitrailer Yaw Stability

•The trapezoidal- or step-steer type control input constitutes a generally useful maneuver condition in which to evaluate tractor yaw stability.

•The operating state in which the most discriminatory evaluations of tractor yaw stability can be made involves the fully-loaded vehicle traveling at highway speeds.

•The "Handling Diagram" has been identified as a broadly useful means of displaying the quasi-steady yaw response characteristics of road tractors—up to their yaw or roll instability thresholds.

•For vehicles with tandem suspensions, a separate curve will be generated on the handling diagram for each operating speed. Accordingly, the test for yaw instability can only be conducted, for each curve obtained with such vehicles, using a "critical speed," V_{crit}, which corresponds to the operating speed for which the curve was generated.

•An existing digital simulation for articulated vehicles has been shown to be broadly capable of predicting the relative influences of design and operating parameters on the yaw stability of tractor-semitrailers. This capability derives largely from the completeness of the simulation model in its representation of distributed frame and suspension stiffnesses in roll, as well as the nonlinear properties of tires by which vertical load influences the side force versus slip angle relationship.

7.4 Recommendations

On the basis of the findings of this study, certain recommendations can be made for possible implementation by NHTSA and/or by various elements of the trucking industry. The previously stated findings define a heavy truck phenomenon which merits further examination as a factor potentially causing accidents. Three research initiatives are identified as a means of pursuing this potentially causal role.

1) It is recommended that a study be conducted of the extent to which the mass accident record shows accident types suggesting a tractor yaw divergence. The most likely maneuvering scenarios and vehicle configurations cited in the "Findings" section of this report should be particularly examined.

2) It is recommended that in-depth accident investigations be conducted to establish whether the underlying hypothesis behind the foregoing study can be substantiated in a case-by-case examination of certain tractor-semitrailer accidents. The particular accident types to be investigated would encompass loss-of-control events in which no braking was present. The in-depth approach would provide a detailed description of those vehicle design and operating parameters which are known to influence basic yaw stability while also assessing whether a yaw instability may actually have precipitated the accident.

3) Aside from the accident studies, it is recommended that human factors research be conducted to determine the extent to which the directional controllability of tractor-semitrailers is influenced by the open-loop yaw stability of the tractor. In pursuing this avenue of research, the closed-loop path-keeping performance of representative tractor-semitrailer drivers would be used as a direct measure of the significance of tractor yaw stability to vehicle control. A sample of drivers operating vehicles of varying degrees of yaw stability would be directed to

perform "limit-proximity" driving tasks in which divergent behavior is threatened. Such research would require a complete engineering characterization of the open-loop properties of each vehicle and would necessitate outriggers and other safety measures to protect the test drivers. Compliance in the tractor's steering system should also be considered as a test variable, since the currently-reported study only treated the vehicle in terms of response to front-wheel angle input.

In addition to these investigations, it seems appropriate that the NHTSA take some information initiatives to advise motor vehicle manufacturers and representatives of the truck operations community of the particular findings which bear upon vehicle design and usage, respectively. Motor vehicle manufacturers should be encouraged to consider the findings implicating roll stiffness distribution so as to determine the possible feasibility of future changes in chassis and suspension design. Fleet operators and private vehicle owners should be encouraged to review the list of effects which degrade yaw stability so as to take steps to "neutralize" any practices that might result in a grossly reduced level of yaw stability.

8.0 STUDY OF CAB VIBRATIONS

This section reports on a contract task which was added by NHTSA to obtain introductory information relative to the vibrational environment in truck cabs. The task, as conducted, was primarily a test exercise in which two tractor-semitrailers were instrumented to measure the ride vibrations caused by running over a road. Prior to reporting on the ride tests, however, a brief introductory discussion is presented to provide an overview of the nature of the ride vibrations occurring in commercial vehicles. The complete ride test data are in Appendix VI.

8.1 Introduction

The vibration of cabs in commercial motor vehicles has recently received increasing attention as an element of the working environment of the truck driver. The in-cab vibrations comprise the vehicle ride motions transmitted through the cab and the noise levels generated or transmitted by the cab. In general, the ride motions, which are felt rather than heard, consist mainly of vibratory motions with frequencies ranging from 1 to 20 Hz, as result from the gross motions of the vehicle, resonances of various vehicle components, and vibrations of engine/drivetrain components.

In-cab vibrations have been of concern to commercial vehicle manufacturers because of their influence on durability and overall customer acceptance. However, as will be pointed out in the following discussion, the ride vibrations of commercial vehicles are dependent not only on vehicle design, but on usage factors as well and the operational (highway) environment. Compounding these considerations is the cost competitive nature of the commercial hauling industry whose primary concern is ruggedness and durability. The de facto consequence has been that the ride vibration quality of commercial vehicles has suffered from the dilution of responsibility among manufacturers, users, and highway agencies, and the necessity of subjugating ride considerations to factors of more direct economic concern. Thus, the recent interest exhibited by agencies of the federal government in truck ride as a health and safety issue provides motivation (in lieu of the marketplace) for expanding the technology required to understand and treat "the truck ride problem."

8.1.1 <u>Excitation Sources of Truck Ride Vibrations</u>. The truck, like any dynamic system, vibrates only in response to inputs which excite the system. The inputs exciting the ride response of highway vehicles arise predominantly from (a) the roughness of the roadway and (b) non-uniformities in the rotating components of suspension systems and engine/drivetrain components.

The elevation profile of the roadway consists of random and periodic components which may be characterized by a spectrum of spatial frequencies (that is, cycles per unit distance traveled) or wavelengths. The random component is caused by bumps, undulations, potholes, etc., in the roadway. The periodic components arise from the design and construction methods employed (i.e., uniform lengths of concrete slabs, oscillations of the pavement finishing screed, etc.) and result in increased spectral amplitudes at the wavelengths associated with the construction process. Figure 8.1 gives examples of the spectral characteristics of the roughness profiles measured on actual highways [8]. The roughness is characterized by its Power Spectral Density representing the mean square value of the roughness content as a function of spatial frequency. All roadways, and for that matter, all sections of roadway, are different, although they tend to exhibit certain common characteristics throughout the world [9, 10]. The dominant characteristic is that the roughness amplitude increases with wavelength (i.e., large road features appear with longer spatial separations), such that the roadway excitation is strongest in the range serving to excite the low frequency vibration modes of vehicles.

As the truck or tractor-trailer drives along the roadway, the spatial frequencies are transformed, according to the velocity, to temporal frequencies. Accordingly, each wheel/axle is subject to an excitation containing a broad spectrum of frequencies. These inputs are then attenuated or amplified by the suspension to excite the various resonant systems and vibratory modes of the vehicle.

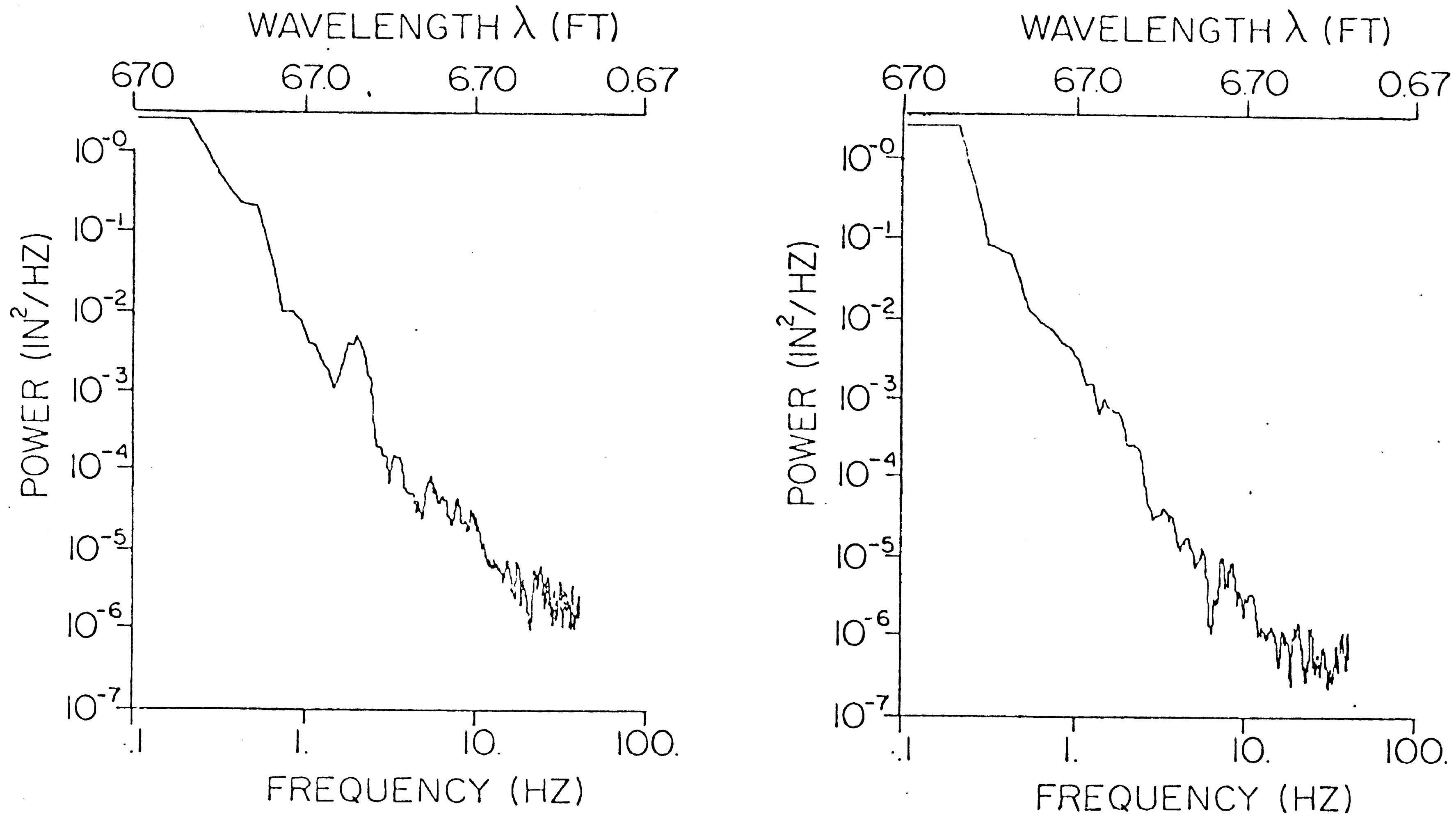


Figure 8.1. Power spectral densities for two examples of highway roughness.

In addition to the broad spectrum excitation existing at each axle, a second significant mechanism of roughness excitation on trucks and tractor-trailers arises from the time-phased relationship between a bump hitting the first and then each of the following axles. Figure 8.2 shows the typical range of dimensions between each of the axles on a tractor-semitrailer combination. A bump hitting the front and then the rear axle(s) on the tractor excites a pitch motion of the tractor. At a 55-mph road speed, this particular phenomenon is a common source of pitch motions (in the range of 3-4 Hz) on typical road tractors. That same bump then passing under the trailer axle(s) may add to the motion by its tendency to cause pitch of the trailer in phase with the tractor motion. This process frequently makes tractor-semitrailer combinations sensitive to highways of the concrete slab type of construction. Such highways are prone to exhibit roughness at the slab joints which may "tune in" at normal highway speeds to the trailer pitch mechanism.

In the same way that vehicle pitch is excited by the "wheelbase effect," load-equalizing tandem axles may be excited in pitch by the longitudinal spread between the axles. The unsprung mass of the axles on trucks typically have a vertical mode natural frequency on the order of 10 Hz. At normal highway speeds, a bump passing under a pair of tandem axles spread by a typical distance of 4 to 4 1/2 feet favors a 10 Hz mode in which the axles oscillate out of phase with one another, causing a vibration known as "tandem hop."

By virtue of their rolling circumference, tire and wheel non-uniformities may also be significant sources of ride motion excitation. Truck tires are approximately 10 feet in circumference resulting in a rotational speed of eight revolutions per second at a speed of 55 mph. The tires, wheels, hubs, and drums have a certain degree of imbalance and eccentricity. Furthermore, tires typically exhibit a circumferential variation in radial stiffness. The net result can be a strong excitation at 8 Hz which excitation is at, or near, the resonant frequency of the first frame-bending mode. (This particular phenomena is the primary reason why many tractor-trailers ride better at speeds in the vicinity of 70 mph and has been the source of some truck driver objection to the 55-mph speed limit.)

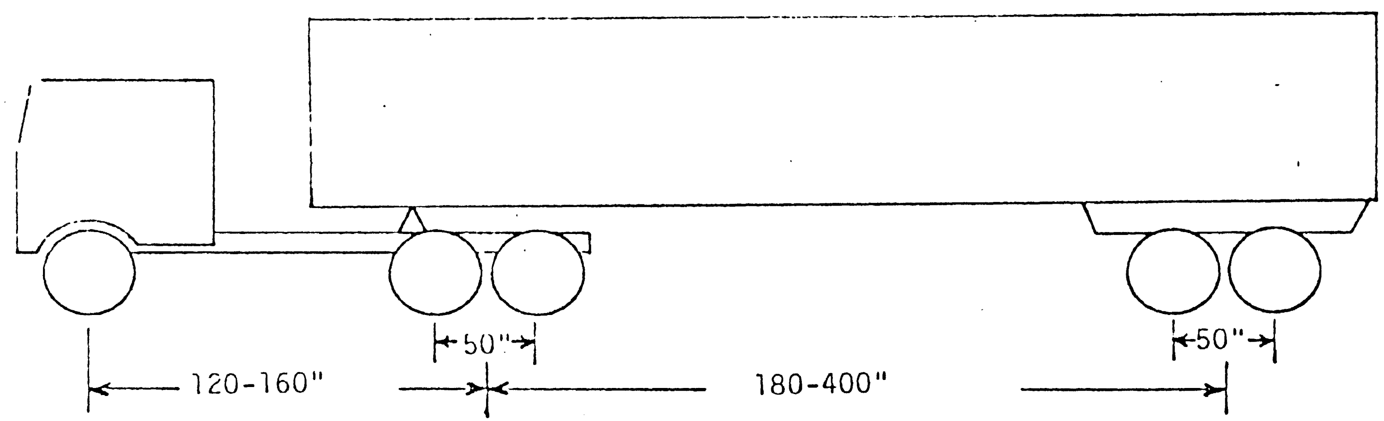


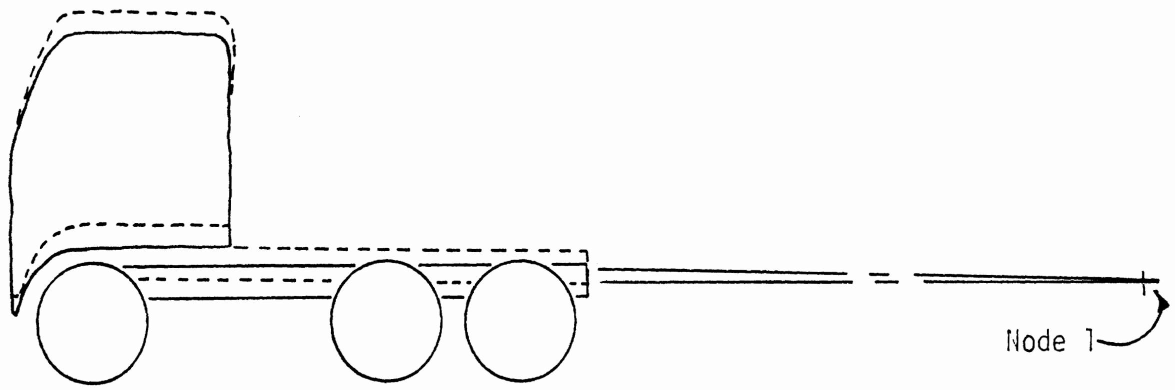
Figure 8.2. Typical axle locations on a five-axle tractor-semitrailer.

8.1.2 Mechanisms of Truck and Cab Vibrations. Although there is a tendency to consider trucks or tractor-trailers as singular types of vehicles, there are, in fact, a number of design types which exhibit differing characteristics in response to ride excitation. The excitation sources discussed in the previous section generally account for the random and phase-related forcing inputs to the vehicle frame through the suspension systems or the fifth-wheel connection. The roadway excitation may be dominantly vertical in direction or may include longitudinal force components phased with the vertical, due to the longitudinal tire forces generated by a road bump. The net result is a multidirectional spectrum of vibration inputs to the frame that may cause so-called rigid-body motions, bending motions of the frame, vibration of components attached to the frame, and, of course, motion of the cab itself.

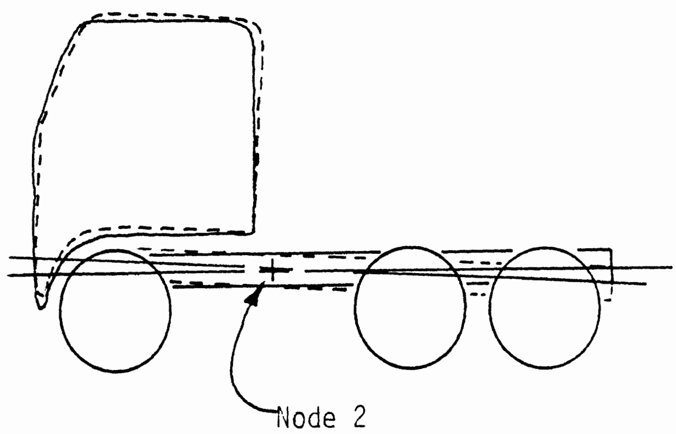
The lowest frequency motions (1-5 Hz) are rigid-body modes. For straight trucks, as with passenger cars, these may be characterized as bounce or pitch motions, though in practice these two degrees of freedom are rarely decoupled. Rather, they are the result of oscillations about two nodes [11] as illustrated in Figure 8.3—the oscillation about the most distant node being considered "bounce," with the oscillation about the nearest node being considered "pitch."

Tractor-trailers, because of the additional degrees of freedom introduced by the trailer, can exhibit four rigid-body modes of motion involving combinations of bounce and pitch of the tractor and trailer. Again, these modes are usually coupled in that they do not consist of pure bounce or pitch. Because these modes are determined by such factors as center of gravity location, radius of gyration in pitch, suspension (or tire) stiffness, and wheelbase, they tend to differ with each vehicle combination and loading condition.

The next set of resonant frequencies following the rigid-body vibration modes are dominated by the flexural modes of the frame (5-10 Hz). The strongest and most important mode tends to be the vertical bending (or beaming) mode, although torsional or lateral modes can occur



Bounce



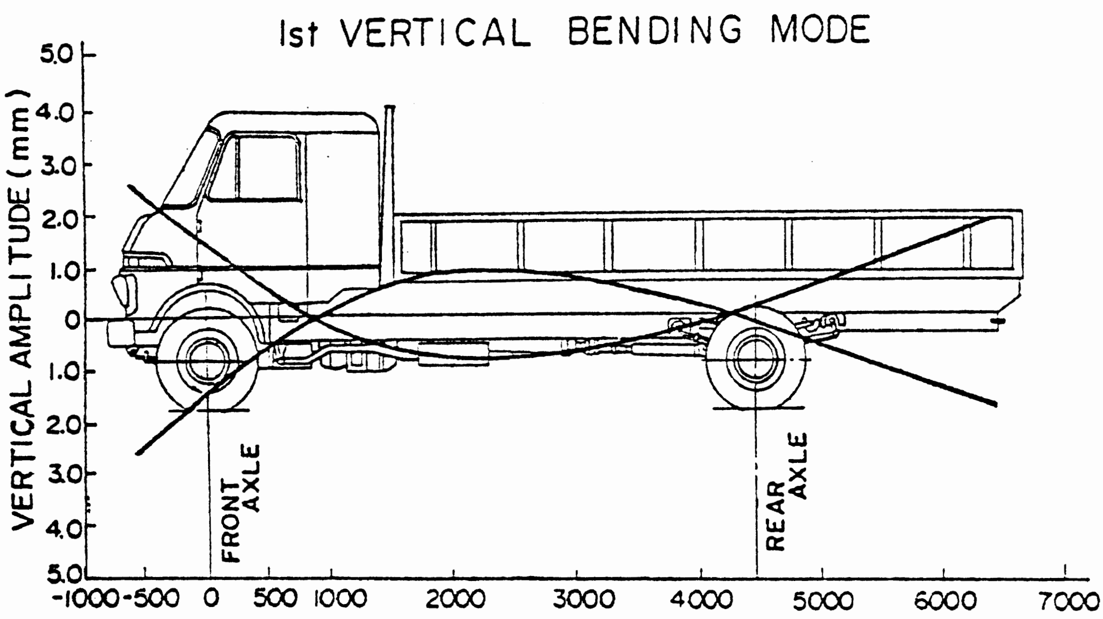
Pitch

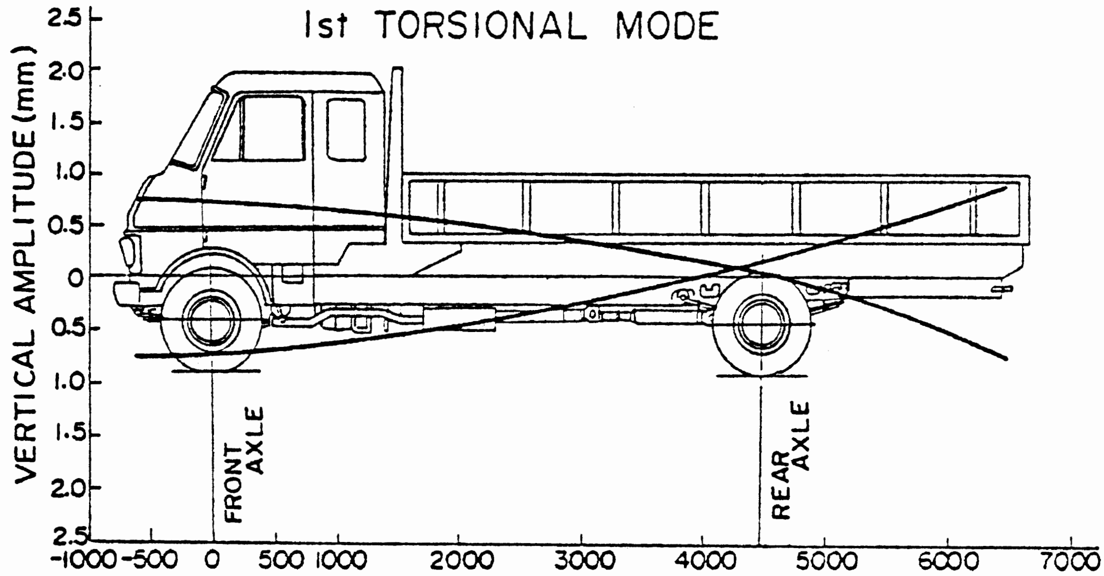
Figure 8.3. Rigid-body vibration modes of a straight truck.

as well (see Figure 8.4). On straight trucks, the flexural modes of the frame are influenced by the type and mounting arrangement of the body affixed to the frame. In the case of tractors, the frequency and degree of excitation of the flexural modes are influenced by the position of the fifth wheel.

At frequencies of approximately 8 Hz and above, any number of other vibratory modes specific to a vehicle may be excited involving higher order modes of frame flexure, axle, component (exhaust stack, battery box, etc.), and engine/transmission/driveline vibrations. In total, over the range of 0-20 Hz, as many as three dozen vibration modes may be occurring on a typical tractor-trailer. The significance of each of these vibratory modes depends on the degree to which it is excited and the degree to which the cab is effectively isolated from an individual mode.

The transmission of frame vibrations to the cab is dependent on the design of the cab mounting system. The designs tend to fall into the categories illustrated in Figure 8.5. The cab-over-engine (COE) configurations of the high-tilt highway tractor and the low-tilt local delivery types are rigidly attached to the forward end of the frame to facilitate "tilting," and have a second frame attachment at the rear of the cab consisting of a compliant isolation element. The "tilt-hood high conventional cab" is mounted only at the front and rear of the cab since the hood is separately suspended to allow it to tilt forward for maintenance. The third category is the more familiar arrangement in which the cab and front-end sheet metal are a unit structure permitting mounts at the front, rear, and intermediate locations of the body. Each category represents a different potential ride behavior because of the different positions of the rider in the vehicle, the different constraints in providing isolation in the mounts, the location of the mounts on the vehicle frame, and the spacing between the mounts that can be accommodated. Beyond that, the different categories of cabs generally represent different trucking applications which influence the ride behavior as a result of the types of loads that are carried and the types of highways that are most frequently traveled.





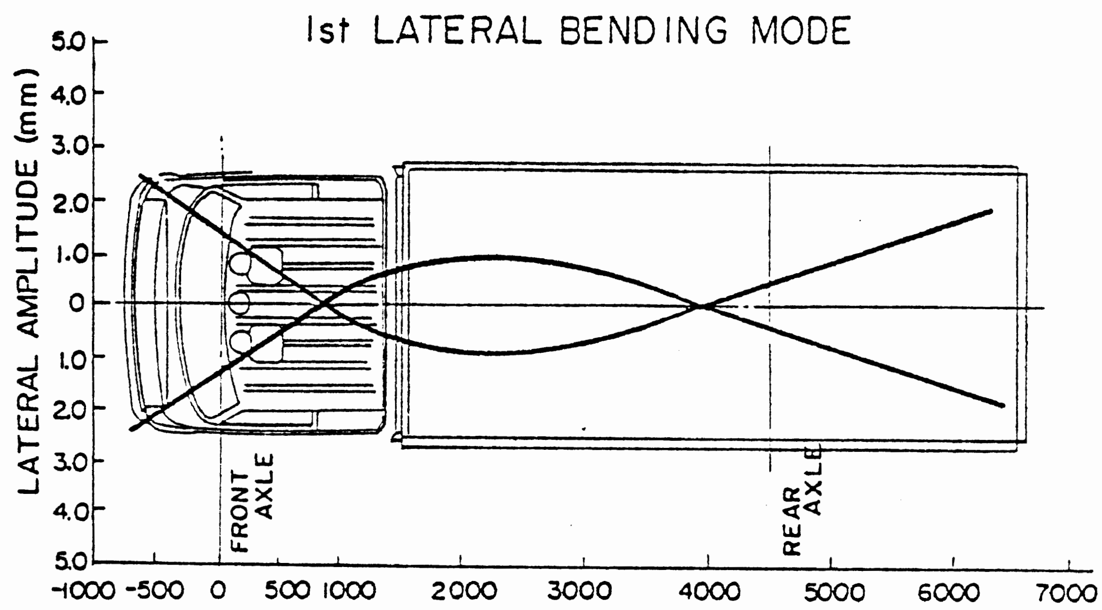


Figure 8.4. Frame flexural modes of a straight truck [12].

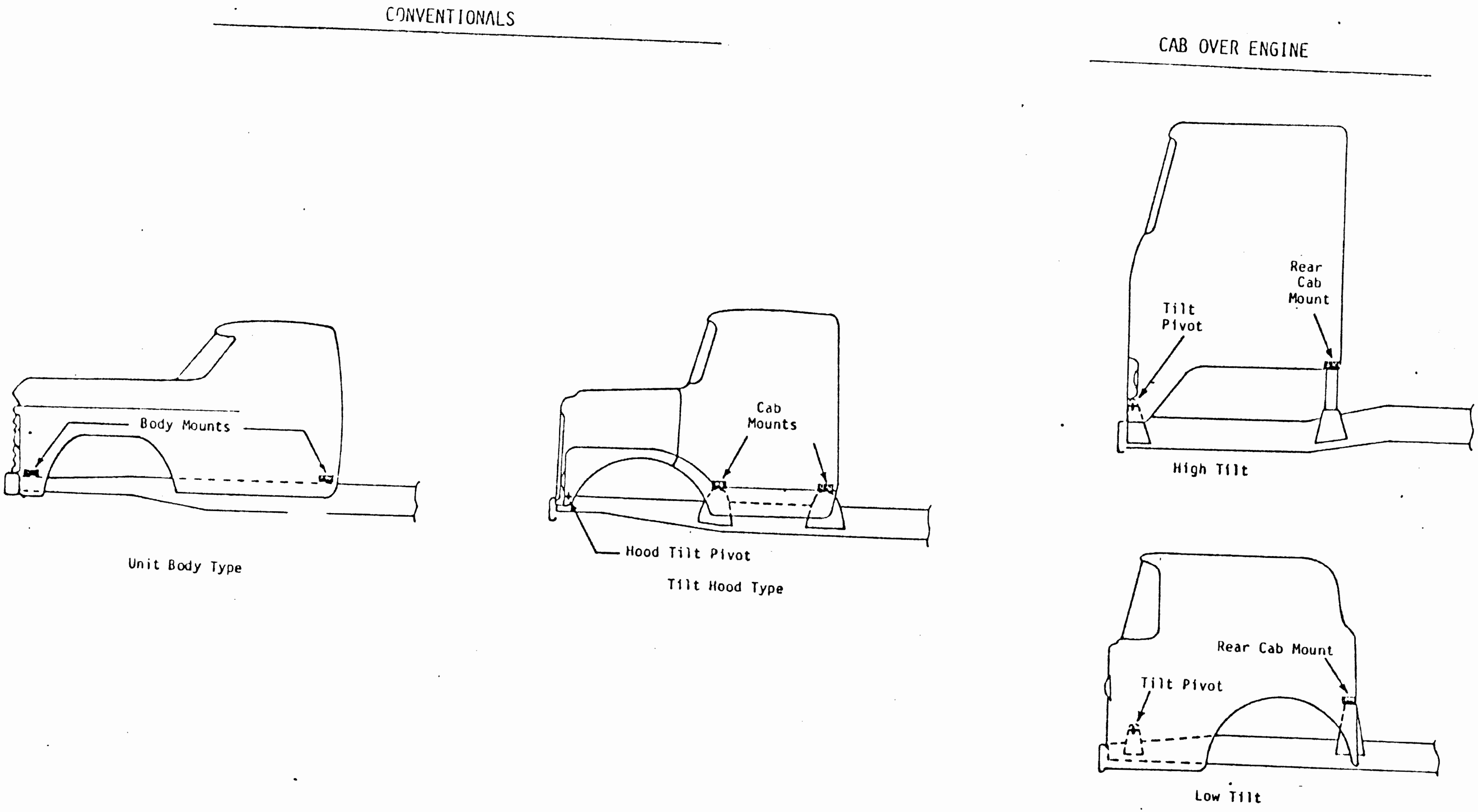


Figure 8.5. Illustration of truck body types.

Cab mounting systems may be characterized by vertical compliance, longitudinal compliance, and compliance in the direction of pitch rotations. Although compliance is naturally provided by the elastic isolation elements commonly used, the brackets and structural members making up the mounting system also make significant contributions to the compliance. Since the cab itself represents a lumped mass attached to a localized area of the frame, the frame, as well, can be considered to be a source of compliance.

Despite the potential complexities of the cab/frame dynamic system, certain simplifications arise from the design constraints that must be met. In particular, cab motions relative to the frame must be limited for functional reasons, with the consequence that the cab resonant frequencies are typically near 10 Hz and higher. Thus the low frequency vibration modes (i.e., less than 5 Hz) of the cab are essentially rigid-body motions of the cab/frame structure, which take the form of bounce and pitch motions of the vehicle as excited by the low frequency road roughness and "wheelbase effects." At higher frequencies (above 5 Hz), the frame and cab suspensions become dynamically active. The first vertical bending mode of the vehicle frame (occurring at 6 to 8 Hz) causes a fore/aft pitch motion of the cab. Likewise, at these frequencies and above, resonances of the battery box, fuel tanks, exhaust systems, and other components may occur causing localized frame motions which may have influence on the cab, depending on the relative motion produced at the cab mounting points. Thus, the higher frequency components of cab vibration arise from motions of increasing complexity as caused by increasing numbers of active components with more degrees of freedom among the coupled systems.

8.1.3 Methods for Ride Improvement. The truck or tractor cab has six degrees of freedom in which vibrations can occur—vertical, longitudinal, and lateral translation, and roll, pitch, and yaw rotations. Except in special cases, most of the vibration energy is concentrated in the vertical and longitudinal translation directions. Because of the complexity of the motion and the importance of identifying only those modes most commonly excited, cab accelerations are usually measured and reduced to the amplitude-frequency plots illustrated in Figure 8.6. On examining this type of plot, the resonant frequencies can be identified and oftentimes associated with particular resonant modes on the basis of experience or on the basis of additional measurements made on the vehicle.

Though only a limited amount of data characterizing truck ride vibrations has been published in the literature, the plots in Figure 8.6 are characteristic of the vibration environment to be found in a typical cab-over-engine highway tractor. The vibration spectrum normally includes three or four dominant resonances, in the range of 1-10 Hz, which include the tractor-bounce, tractor-pitch, and the first frame-bending modes. Above 10 Hz, additional resonances occur due to resonances of other components; but, in general, the amplitude of the vibrations reduce quickly with increasing frequency, such that the major ride vibrations are encompassed within a frequency band ranging from one to fifteen Hz.

The means most readily available to the vehicle manufacturer for reducing cab vibrations is through the design of the cab mounting system. By using compliant mounts it is possible to achieve isolation of frame vibrations at frequencies above the effective natural frequency of the cab/mount structure. Various articles in the literature (12, 13, 14, 15, 16) report on cases where ride improvement has been achieved on specific vehicles. Four to five Hz is normally the minimum resonant frequency practically attainable* using passive elastomeric mounts and such a design can substantially reduce higher frequency vibrations, as indicated in Figure 8.6. Note, however, that the dominant low frequency vibration modes associated with rigid-body motions are not affected by the cab mounting arrangement. Only by reducing the natural frequency of the cab's suspension to 1 Hz or less with active suspension systems [16] is it possible to achieve an attenuation of the rigid-body modes. Figure 8.7 illustrates the typical magnitude of attenuation that has been achieved on a commercial tractor with an active cab suspension

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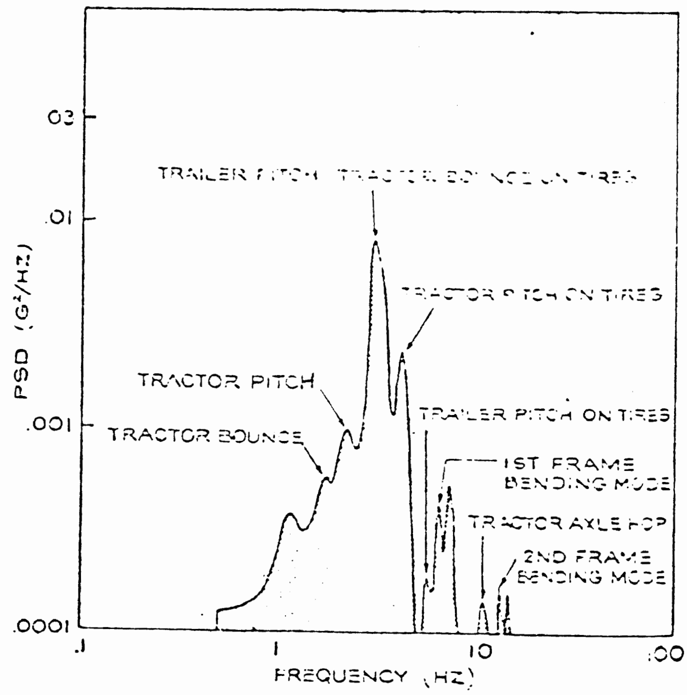
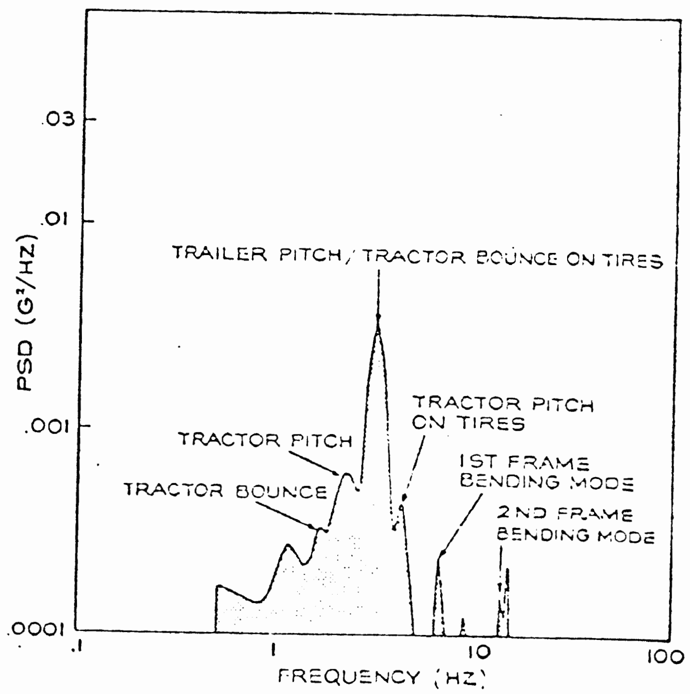
*The achievement of lower frequencies requires static deflections which are greater than cab mountings can typically accommodate.



Vertical Frame Acceleration at Front Cab Pivot



Vertical Frame Acceleration at Rear Cab Mount

Figure 8.6. Typical cab acceleration input spectrum on a COE highway tractor. [13]

system. Note that the vibration is given in terms of absorbed power [17] which implies it has been weighted as a function of frequency and is not directly comparable to the previous figure.

Looking beyond cab suspension systems as a means to improve tractor-trailer ride, Figure 8.7 illustrates another significant point. Using the absorbed power method of weighting rider sensitivity to ride vibration frequency content (absorbed power being qualitatively similar to other tolerance weighting functions such as the ISO standard [18]), shows that the dominant causes of ride discomfort are the low frequency, rigid-body modes (0-5 Hz). (These modes tend to dominate the tractor-trailer vibration spectra, as evidenced by data to be found in the published literature, and by data obtained in the tests to be discussed in Section 8.4 of this report.) The intensity of rigid-body vibration modes exhibited by typical tractor-trailers is, in part, a de facto result of the articulated vehicle configuration, and, in part, the result of the suspension systems commonly in use. The leaf-spring suspension systems commonly used exhibit very high spring rates and coulomb friction levels such that very little suspension motion occurs in normal highway travel. As a result, the tires become the primary suspension elements with very little damping present to control the resulting motions. In effect, the rigid-body motions occur through the action of the vehicle bouncing on its tires. Though this problem has been recognized in the literature [15], little research or results have been reported.

8.2 Experimental Method

Two Class 8 highway tractors were tested in this study—one having a conventional style cab with the other being a cab-over-engine style tractor—in both a loaded (i.e., pulling a loaded 45-ft van trailer) and bobtail (i.e., no trailer) condition. Each tractor was equipped with accelerometers to measure vibrations at various locations on the tractor chassis and on the seat in order to define (approximately) the primary response modes as well as describing the vibration environment of the driver's compartment. Five sections of public roadways in

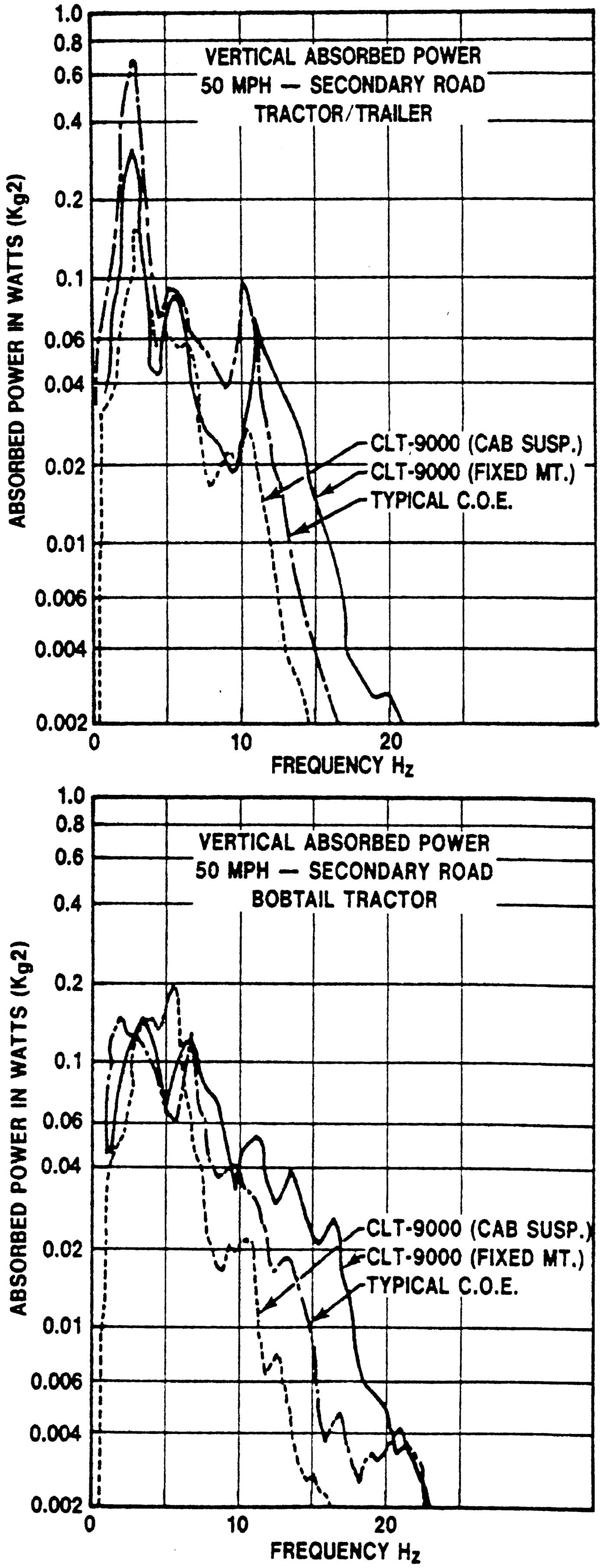


Figure 8.7. Vertical power absorbed by driver of a tractor-trailer with active cab suspension.

Washtenaw County, Michigan were selected as test sites to provide a cross-section of differing road surface types and states of repair. Data were recorded over the test sections at each of two speeds for each of the four vehicle combinations. The collected acceleration data were reduced to power spectral densities with the aid of a spectrum analyzer in order to define the frequency content of the vibration experienced at the point of attachment of each accelerometer.

8.2.1 Vehicles. The specific vehicles used in the study were the previously-described Ford two-axle COE (as used in the yaw stability test series) and a GMC (Model 9500) two-axle conventional cab tractor. Both tractors had gross axle-weight ratings of 10,860 lb. front and 19,040 lb. rear for a GVWR of 29,900 lb. The COE tractor had a wheelbase of 134.5 in. and a cab-to-rear-axle dimension of 80 in., compared to 151 and 84 in. for the respective dimensions of the conventional tractor.

Both vehicles incorporated multi-leaf-type spring suspensions on the front and rear axles, with the rear suspension on the Ford tractor incorporating a two-stage spring design in which a "helper" leaf became engaged under full load. Both vehicles also were outfitted with spring-type suspension seats, providing a measure of vertical isolation to the driver. A common set of tires were used on both tractors with individual tires being placed at the same respective wheel positions on both vehicles and with dual tire sets being clamped in the same relative angular orientations in an attempt to minimize variations in ride vibration that could be attributed to tire installation. Tires were inflated to a cold pressure of 85 psi prior to running tests with each vehicle.

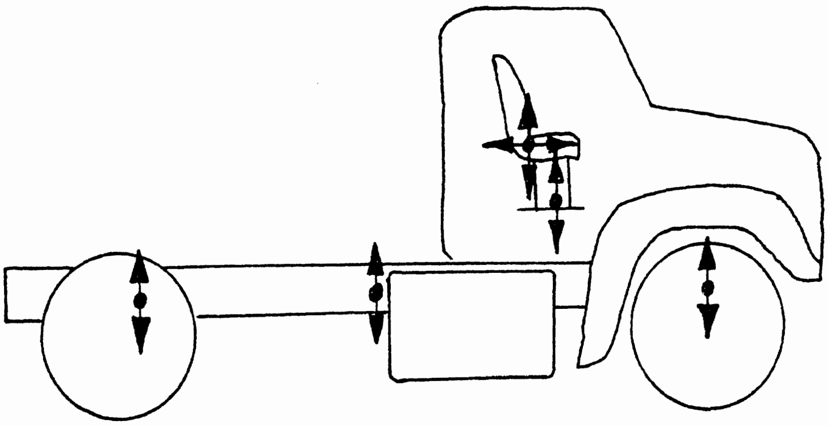
In these tests, the trailers were loaded in the same manner as the van trailers had been loaded for the yaw stability study (see Section 5.3.2).

8.2.2 Instrumentation and Signal Recording. Accelerations at the various points of interest were measured with servo accelerometers manufactured by Schaevitz Engineering. For measurement of the vertical components of acceleration at five points, accelerometers with a $\pm$ 10 g range and a natural frequency of approximately 140 Hz were used. These specifications provided for adequate transducer fidelity over the range of accelerations and frequencies believed to be important in ride measurements. A single measurement of longitudinal acceleration was made using an accelerometer having a range of $\pm$ 1 g and a natural frequency of 100 Hz (nominal).

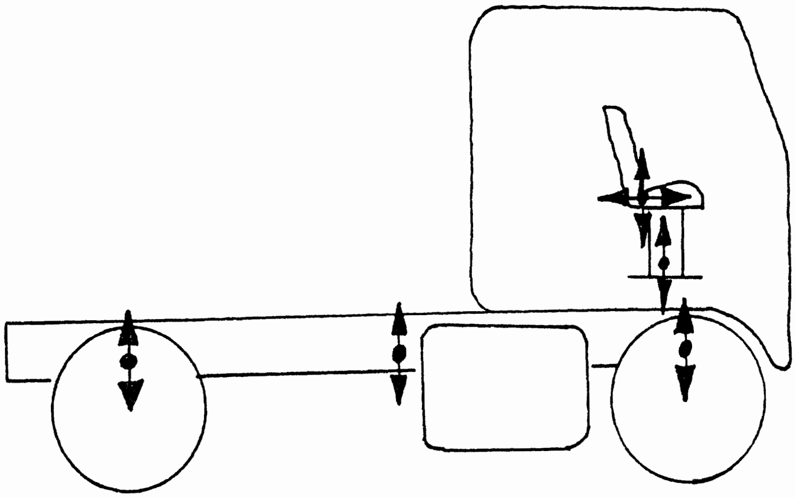
These six accelerometers were mounted on each test tractor at the locations shown in Figure 8.8. Mounting positions were chosen to (a) provide a characterization of the driver's vibration environment and (b) permit identification of the dominant ride vibration modes of the two test vehicles. As shown, two accelerometers were mounted directly on the driver's seat, one with a vertical orientation and one horizontal. Another accelerometer was mounted on the seat base, near the floor, to obtain the vertical acceleration of the cab itself. Three accelerometers were mounted directly to the frame with their sensitive axes oriented vertically. The frame mounting points were (1) directly over the front axle, (2) midway between the axles, and (3) directly over the rear axle. Using the amplitude-frequencies spectra and phase relationships of these three frame-located transducers, certain of the dominant ride modes of the vehicle can be identified. It should be noted, however, that a rigorous modal analysis was beyond the scope of the study.

These acceleration signals, along with a velocity signal generated by a standard fifth wheel measurement unit, were recorded on an on-board FM tape recorder. An on-board interfacing unit was used to provide an easy adjustment of signal gain with individual signal gains being adjusted to provide input levels to the tape recorder that were as high as possible without saturation, thus maximizing signal-to-noise ratio. Acceleration signals were continuously monitored on a chart recorder during the tests as a means of determining the needed gain adjustments on each test pavement.

8.2.3 Test Sites. The five test sites selected to represent typical surfaces to be encountered by tractor-trailer vehicles in normal operation were:



Conventional Tractor



Cab-Over-Engine Tractor

Figure 8.8. Location and orientation of accelerometers for vibration measurement.

1) I-94 Eastbound, between Mile Markers #181-182
New, smooth bituminous asphalt pavement

2) I-94 Westbound, between Mile Markers #175-174
Aging, cracked, patched Portland cement concrete

3) US 23 Southbound, between Mile Markers #47-46
Aging Portland cement concrete expressway (not as deteriorated as site #2)

4) M-14 Eastbound (east of US 23)
New Portland cement concrete expressway

5) Huron Parkway, Glacier Way to Geddes Road (City of Ann Arbor)
Aging urban street, bituminous asphalt

Sites 1 and 4 were in excellent condition whereas site 2 was in bad need of repair and constituted a very rough ride input. Sites 3 and 5 represent, more or less, typical expressway and urban surfaces, respectively, both showing signs of wear and cracking but not to the extent of site 2.

The test vehicles were run over each test surface at a nominal speed of 55 and 45 miles per hour. Each test section was selected to provide sufficient length and homogeneity as to assure an adequate data sample for processing and analysis. Trial runs were repeated at each site until the maximum gain for each channel had been identified and adjusted as described earlier.

8.3 Data Analysis

The data recorded on FM tape was processed using a Hewlett Packard 3582A Spectrum Analyzer. This analyzer is a two-channel device with a frequency range of .02-25000 Hz and a dynamic range of 70 db. The dual channel feature facilitiates the measurement of phase relationships and transfer functions between different input signals while otherwise evaluating the amplitude spectra of each signal. All processing was done over a frequency span of 0-25 Hz using a "Hanning" type sampling window function which provides a compromise between good amplitude and frequency resolution. The data was rms-averaged over eight samples, each sample being five seconds in length, to effect a smoothing of the response spectra and to eliminate extraneous information which might be contained within a single five-second sample. Further information concerning this analyzer and its operation can be found in Reference [7].

Data processing consisted of (1) reducing the input time histories to amplitude spectra for each accelerometer signal and (2) determining the phase relationships between signals. All signal phases were related to the signal from the accelerometer located over the front axle, with the exception of the "seat-vertical" which was related to the signal generated at the seat base so as to facilitate examination of the seat suspension system alone. Amplitude spectra generated by the analyzer with units of db, volts, were converted to power spectral densities, g^2/Hz, by a simple algebraic calculation involving the scale factor used in the signal processing unit and the bandwidth employed by the analyzer in generating the spectra.

By examining the power spectral density (PSD) of each signal, along with the relative phase angle, certain vibrational modes were identified. In general, it was found that the dominant low frequency peaks of the PSD could be related to the bounce, pitch, and beaming modes mentioned in a previous section. In theory, a pure bounce resonance is characterized by zero relative phase angles at all three locations on the frame. Alternatively, a pitch mode involves frame acceleration signals, front and rear, that are 180° out of phase while the midpoint signal is in phase with either the front or rear signal, depending upon the longitudinal location of the node. In the case of the fundamental mode of beaming of the frame, the midpoint signal is 180° out of phase with signals from both the front- and rear-located accelerometers. In practice, due to the phase lag inherent between the front and rear tires as they encounter road irregularities, and due also to the complex (i.e., coupled) nature of the modes of an actual vehicle, the analysis is not so simple and each PSD and phase relationship must be carefully scrutinized and compared to other tests in order to identify the vibrational modes present.

8.4 Summary of Ride Test Results

The processed ride spectra were machine-plotted as power spectral densities for each accelerometer involved. These PSD plots will be used in the following discussion first to demonstrate the salient features of the response spectrum of each vehicle. Identification of the dominant modes occurring in the low frequency (i.e., 0 to 10 Hz) portion of the spectrum will be based upon examination of the amplitude and phase information obtained for the frame-mounted accelerometers. The data will then be summarized according to the following comparisons:

1) bobtail configuration versus loaded combination

2) COE tractor versus conventional tractor

8.4.1 Basic Spectral Features. Shown in Figure 8.9 are the PSD of seat accelerations in the vertical and fore/aft directions, as measured at 55 mph with the COE bobtail tractor on the smoothest test surface, Number 1. Six peaks in the vertical acceleration spectrum have been identified as to the mode or source of excitation involved. Four of these peaks are, likewise, evident in the fore/aft acceleration spectrum.

Peaks labeled c, e, and f occur at 7.5, 15, and 22.5 Hz, respectively. These frequencies comprise the fundamental and first two harmonics of the wheel rotation frequency at the test speed of 55 mph. Presumably, the power being input at those frequencies derives from the combination of wheel unbalance, geometric runout, and tire stiffness nonuniformities. As shown in Figure 8.10, the identification of these peaks as deriving from wheel rotation phenomena is aided through the conduct of tests at a differing speed. Note that only the indicated "wheel rotation peaks" have moved along the frequency scale as a result of a speed reduction from 55 mph to 45 mph, although reductions in power level are apparent at virtually all points along the spectra. Peaks c, e, and f are seen in Figure 8.10 to occur at the 45-mph synchronous wheel rotation frequencies of 6.2, 12.4, and 18.6 Hz.

The peaks labeled (a) and (b) in Figure 8.9 can be identified as two of the approximately-decoupled rigid-body modes of vibration of the

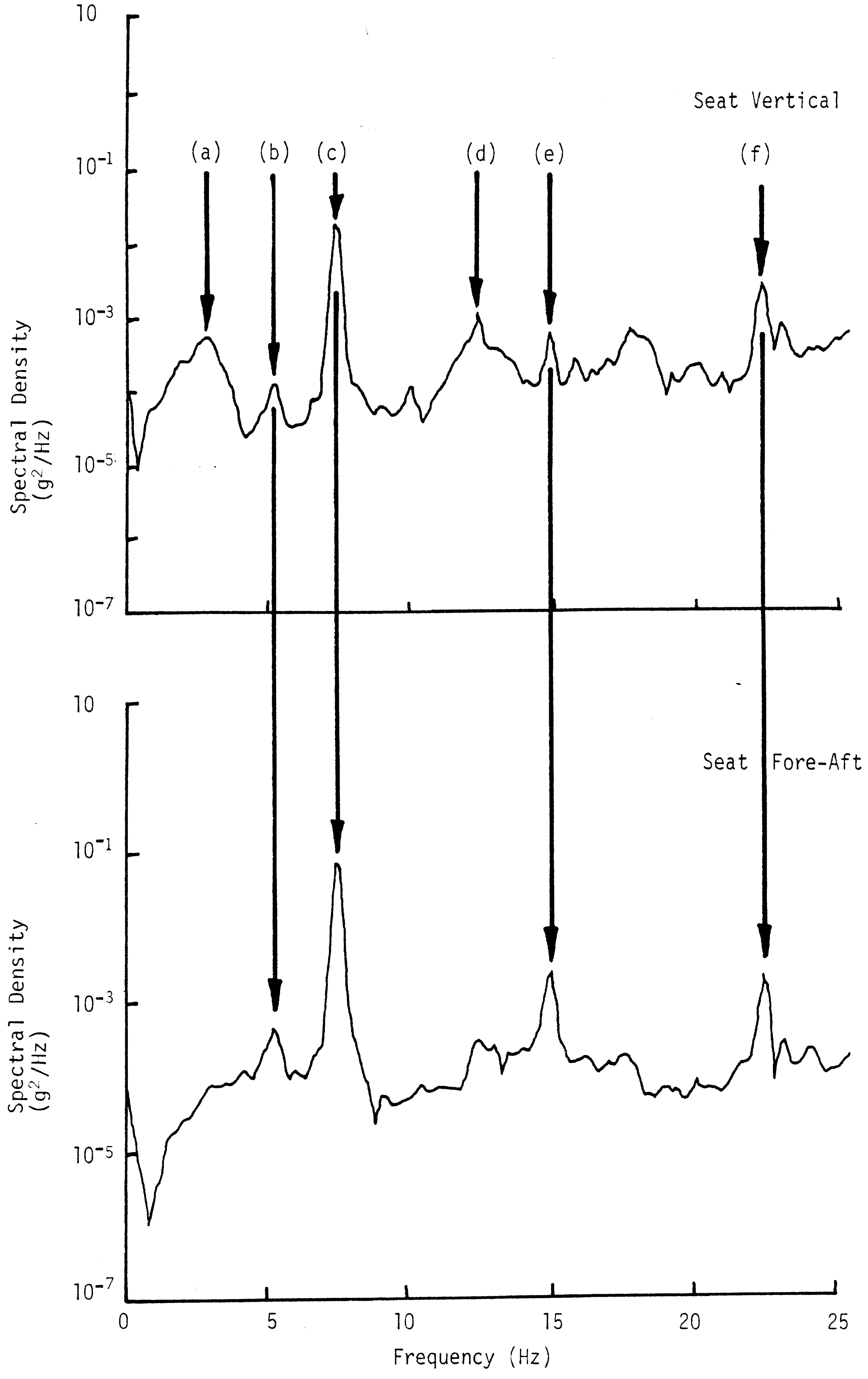


Figure 8.9. PSD's for vertical and fore/aft vibration of driver's seat - COE tractor, bobtail, 55 mph, test site No. 1.

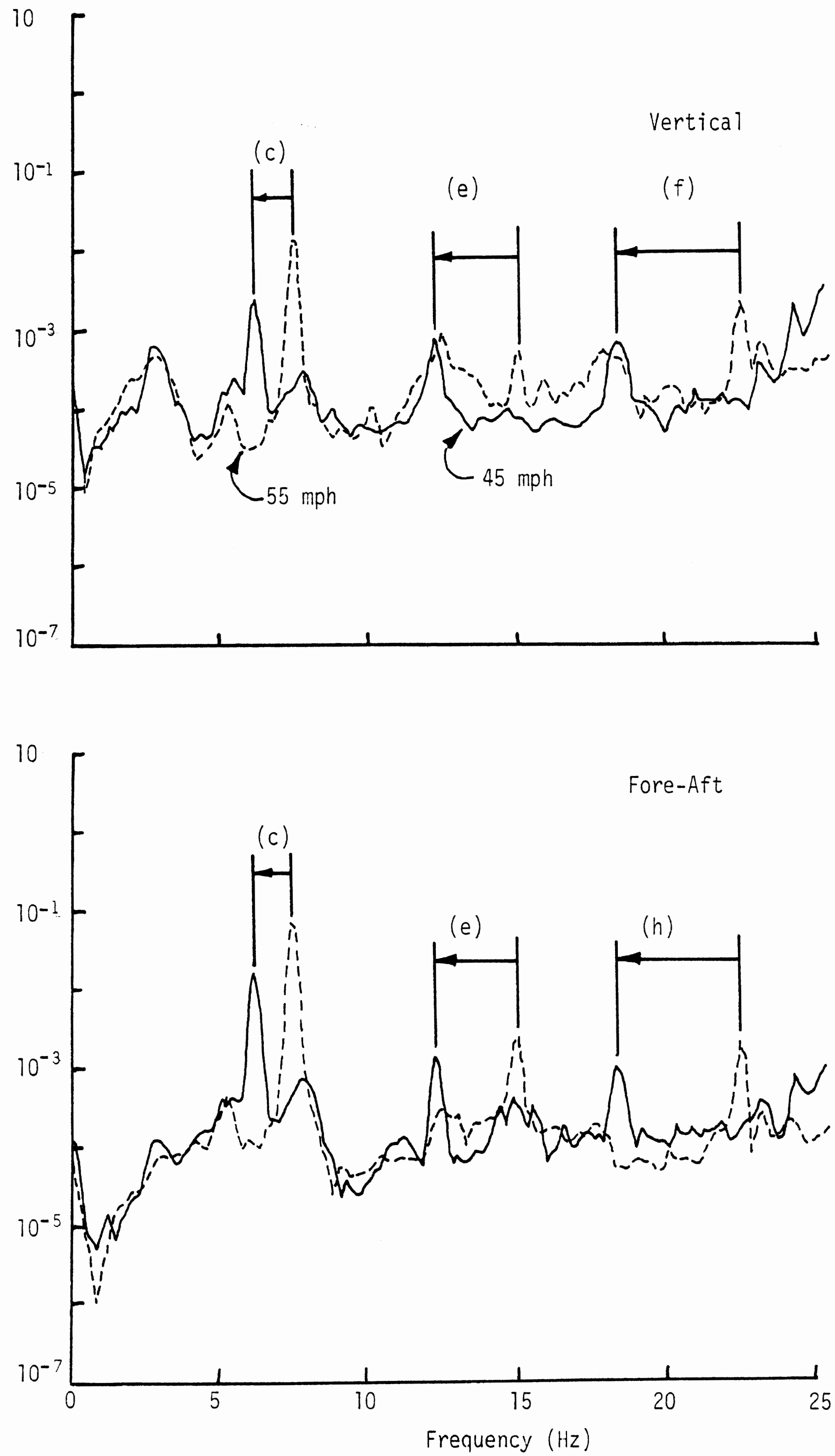


Figure 8.10. Spectral peaks c, e, and f are seen to derive from wheel nonuniformities showing proportionately reduced frequencies when speed is reduced from 55 mph to 45 mph.

tractor—bounce and pitch, respectively. These modes are identified (in the absence of an automated modal analysis) through inspection of the relative amplitude and phase spectra of the vertical accelerations measured at three points on the tractor frame. Acceleration signals from the frame-mounted accelerometers reveal the mode shape sketched in Figure 8.11, corresponding to a predominantly bounce mode (although some coupling with pitch is noted) for the peak (a) at 3 Hz. Similarly, Figure 8.12 shows a predominantly pitch mode shape for the peak (b) at 5.2 Hz. The mode shapes were deduced from the amplitude and relative phase data shown in Figures 8.13 and 8.14. The only portions of these figures which are relevant to determination of mode shapes for peaks (a) and (b) , of course, involve the specific values of amplitude and phase angle prevailing at the respective frequencies, 3 Hz and 5.2 Hz. Assuming that only rigid-body motions occur at those frequencies, the amplitude and phase information permits the approximation of the mode envelopes which were shown. Such approximations are hampered, however, by the built-in phase distortions deriving from the fact that a time delay exists between the front- and rear-wheel passage over road irregularities.

Using amplitude and phase spectra describing the motion of the cab floor with respect to the front frame position, a cab vertical bounce mode (that is, the bouncing with respect to the frame on its rubber mounts) was also identified at peak (d) in Figure 8.9. Many other significant modes of vibration are possible in the 0-25 Hz range, but were not identifiable through the very limited set of signals being gathered in the described tests.

8.4.2 <u>Comparative Response - Bobtail Versus Loaded Configuration</u>. The PSD plot in Figure 8.15 shows an overlay of the responses measured on the seat of the COE tractor for both the bobtail and loaded configurations on the smooth test surface (No. 1). The spectra reveal a complicated set of contrasts between the ride environments of the two vehicle configurations. Most generally, the bobtail vehicle shows a greater level of power over most of the frequency range above 6 Hz and a lower power level than the loaded vehicle below that frequency. Many of the

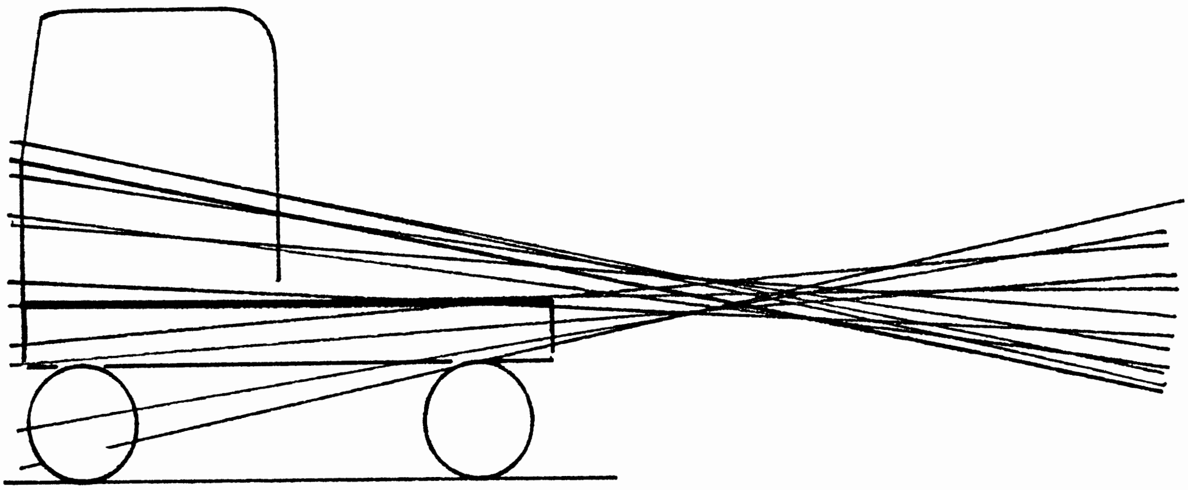


Figure 8.11. Predominantly "bounce" mode shape at 3 Hz (peak a).

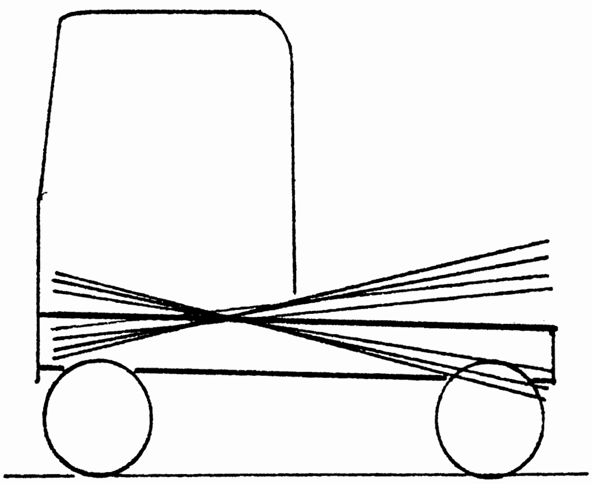


Figure 8.12. Predominantly "pitch" mode shape at 5.2 Hz (peak b).

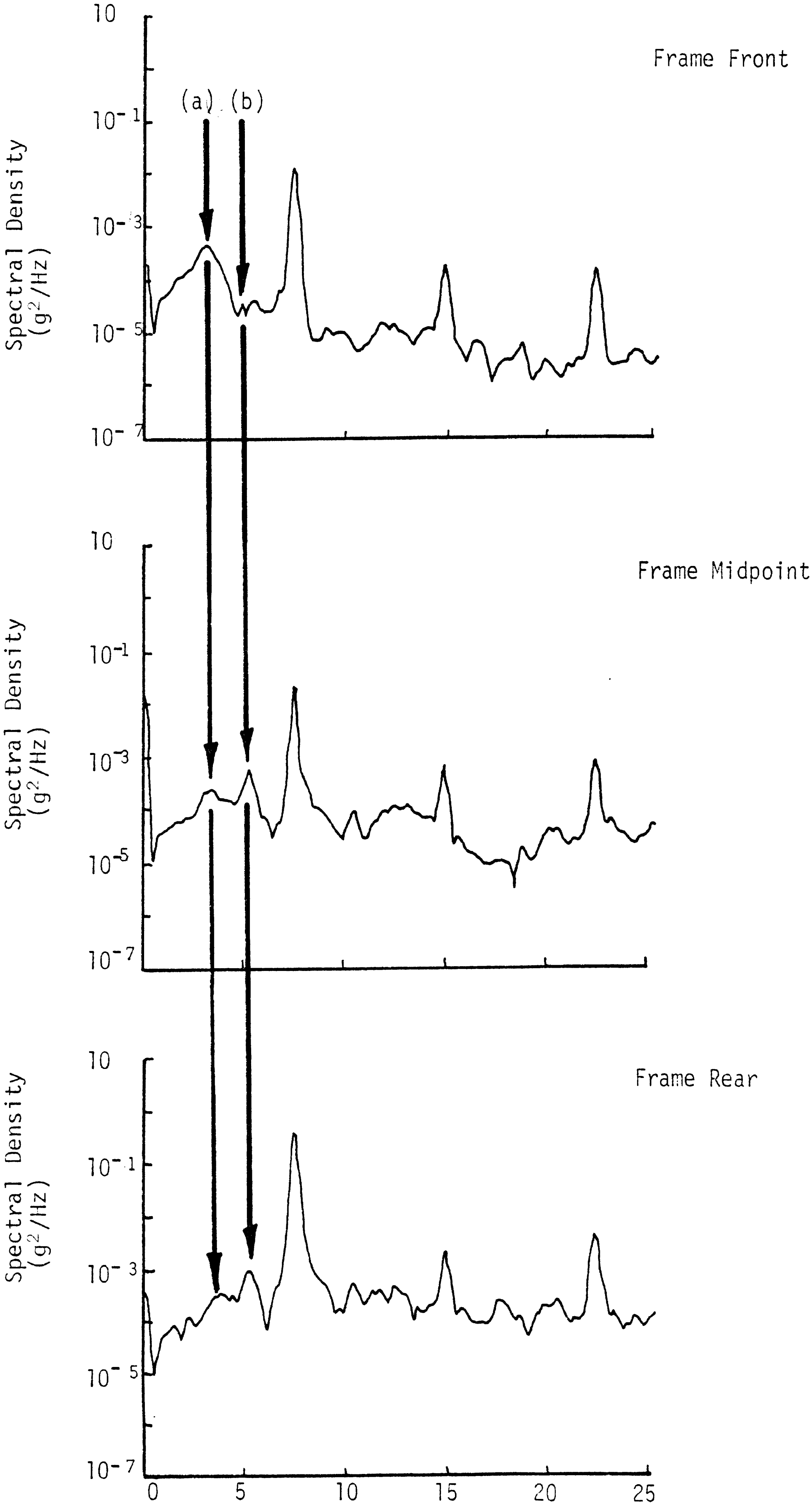


Figure 8.13. PSD's for frame-mounted accelerometers - COE tractor, bobtail, 55 mph, site No. 1.

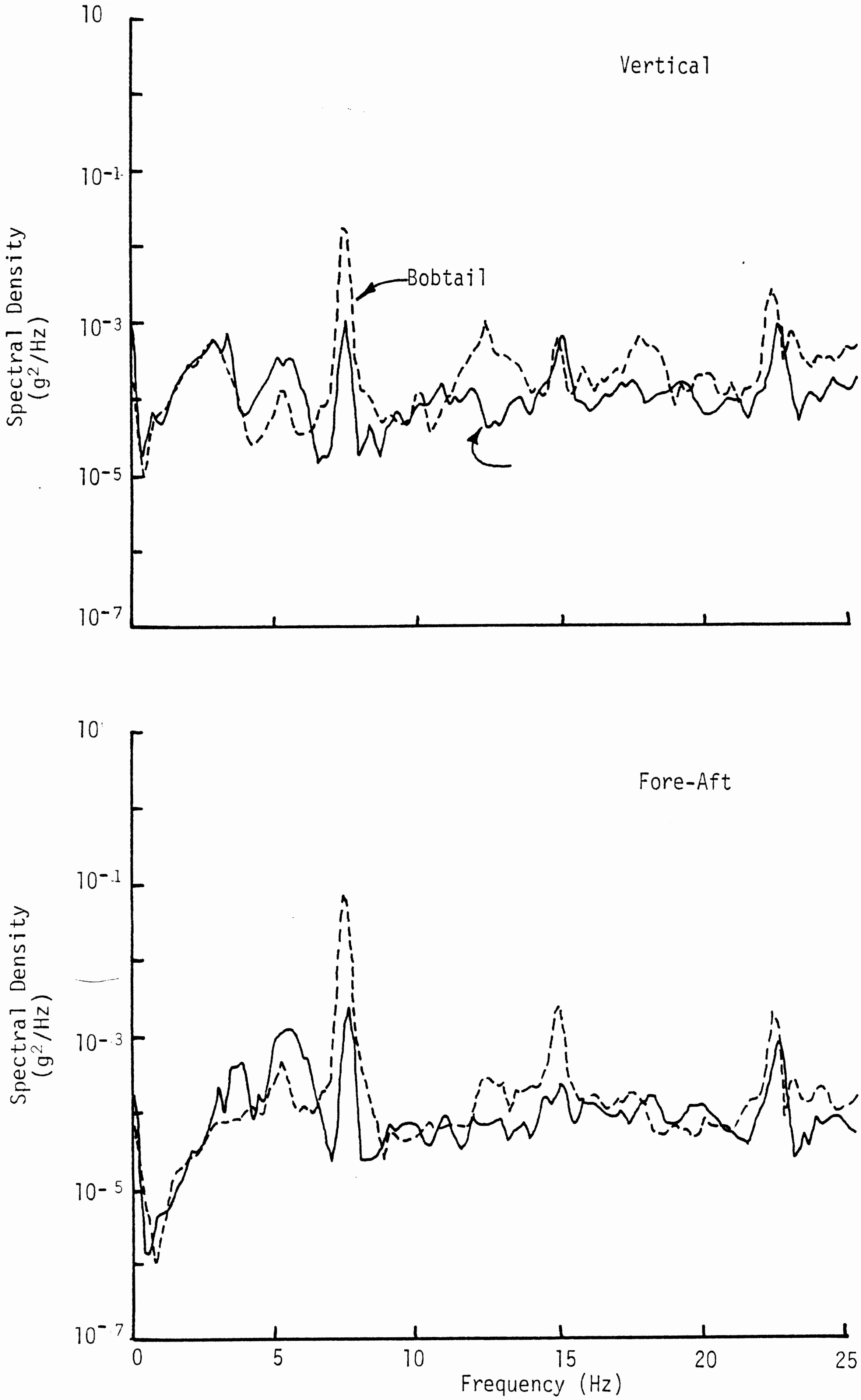


Figure 8.15. PSD's of bobtail vs. loaded combination - COE tractor, 55 mph, site No. 1.

same mode shapes can be deduced for the bobtail and loaded trailer configuration. The bounce mode with loaded trailer appears at 2.5-3.0 Hz and the two pitching modes are found at 3.5 Hz and 4.5 Hz. In both pitch modes, the node is located between the front axle and the frame midpoint. The interaction of these pitching modes with the trailer cannot be determined, however, because the trailer was not instrumented in the subject experiments.

On comparing these same spectra, as obtained on a rough road site (No. 2), the differences are much more dramatic. Figure 8.16 shows that the bobtail tractor experiences approximately five times as high a power level in the vertical direction as does the tractor with a loaded trailer. Further, the bobtail tractor shows a very high fore/aft peak at the 7.5 Hz synchronous wheel rotation frequency—whereas no similar feature appears in the spectrum for the loaded vehicle.

8.4.3 Comparative Response - COE Versus Conventional Tractor. As shown in Figure 8.17, the COE tractor when running bobtail exhibits a greater amount of transmitted power than the conventional tractor (bobtail) over most of the frequency range. The very high peak at 7.5 Hz in the fore/aft response of the COE tractor when running bobtail would appear to derive from a lightly damped resonance between the fundamental beaming mode of the frame and the wheel rotation frequency, producing a heavy fore/aft component at the higher seat position of the cab-over configuration. (Of course, the seat height, per se, accounts for a power amplification of only about 1.5, whereas the chance resonance of the beam mode accounts for the predominant level of power seen in the COE peak at 7.5 Hz.)

Again on the smooth (No. 1) surface, but in combination with loaded trailers, we see in Figure 8.18 that the COE and the conventional tractor show rather similar spectra. The COE does, however, show substantially higher power levels in its fore/aft response over most of the frequency range covered in the figure.

On the rougher pavement (No. 2), the contrast between the COE and the conventional tractor is seen to be rather different from the smooth road case. In Figure 8.19 we see no significant differences in the

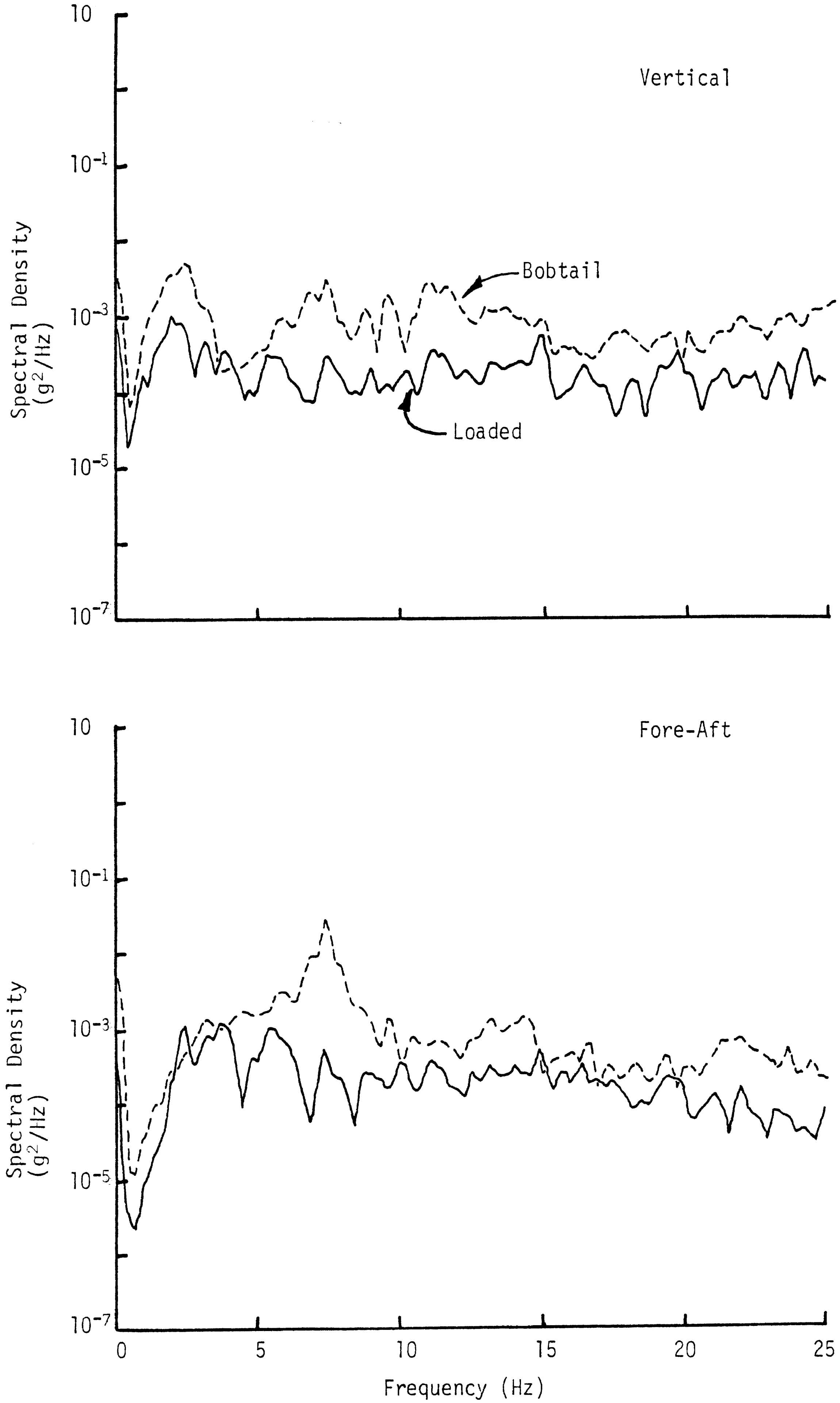


Figure 8.16. PSD's of bobtail vs. loaded combination - COE tractor, 55 mph, site No. 2.

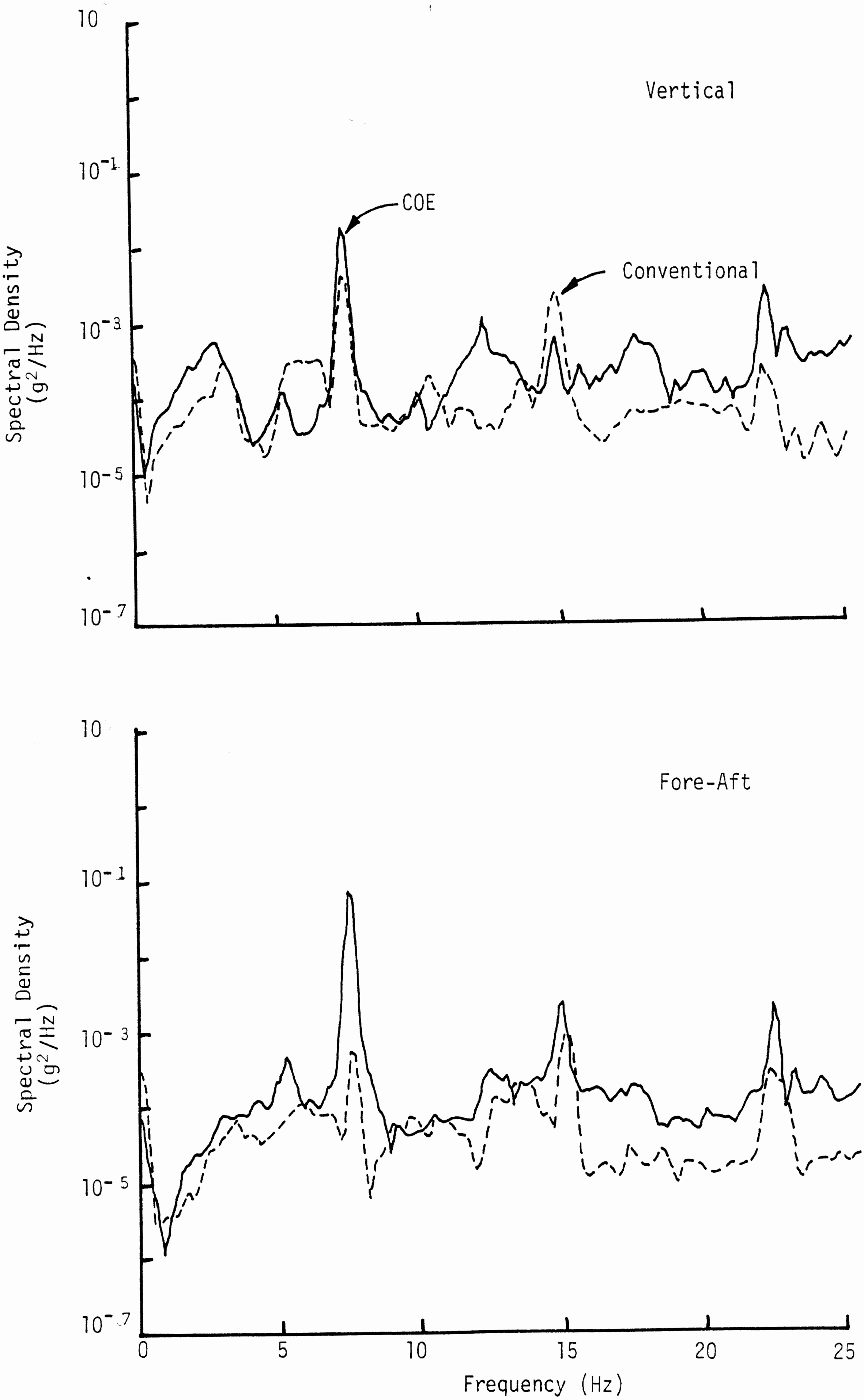


Figure 8.17. PSD's of COE vs. conventional tractors - bobtail, 55 mph, site No. 1.

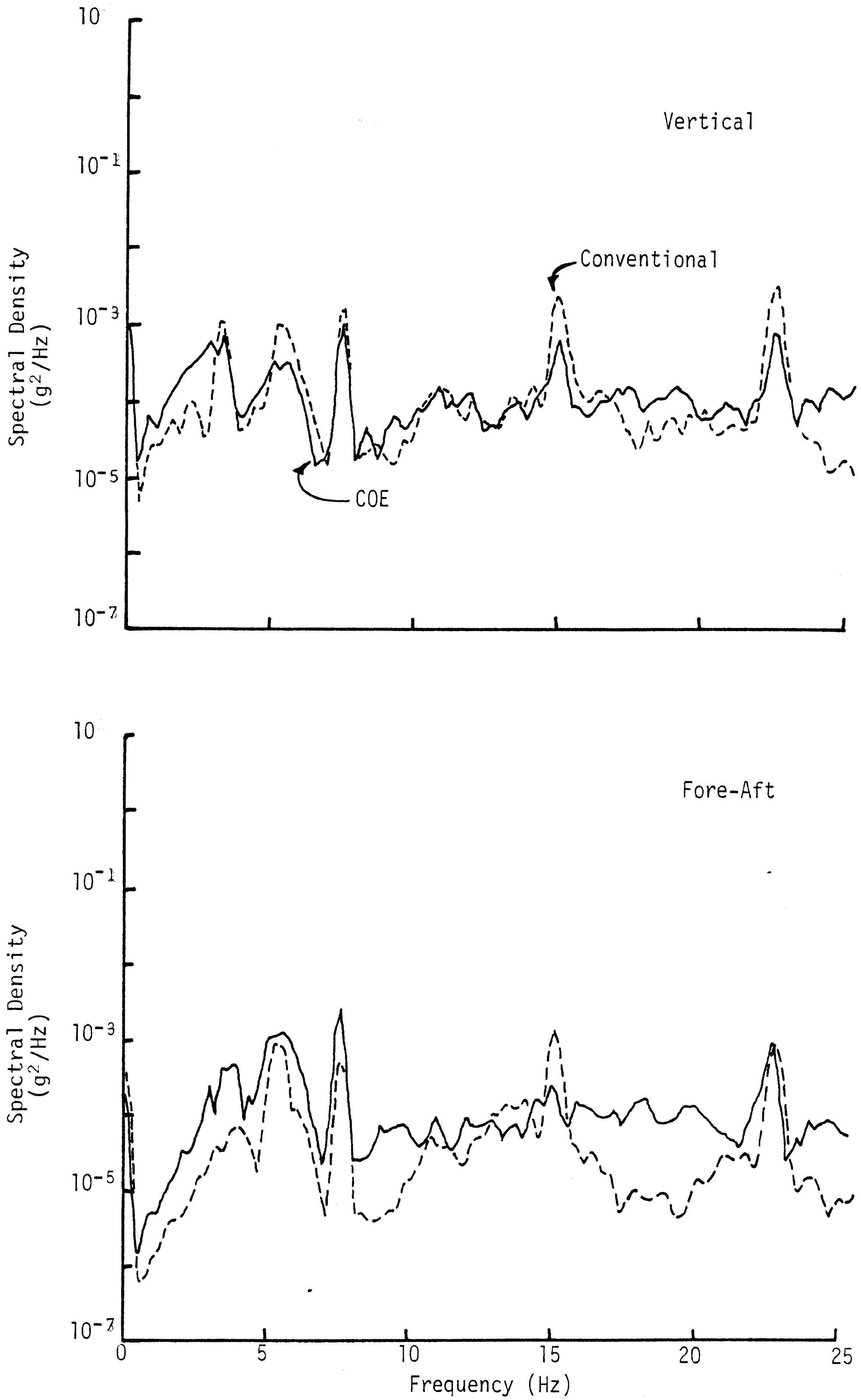


Figure 8.18. PSD's of COE vs. conventional tractor - loaded combinations, 55 mph, site No. 1.

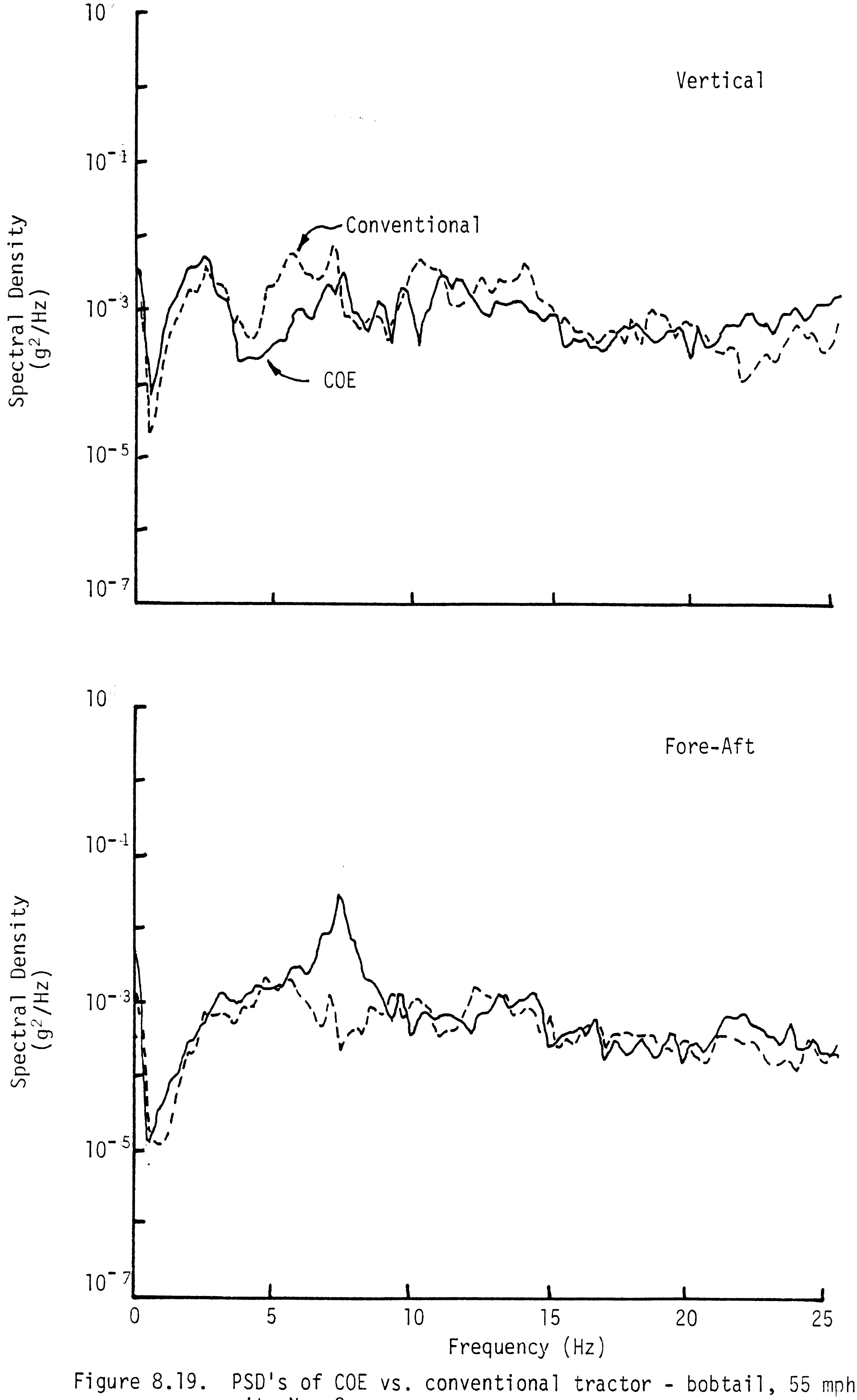


Figure 8.19. PSD's of COE vs. conventional tractor - bobtail, 55 mph, site No. 2.

spectra obtained for the COE and conventional tractors running bobtail on the rough surface, except for the large peak in fore/aft response of the COE at 7.5 Hz. As seen in Figure 8.20 for each tractor coupled to a loaded semitrailer, the COE tractor shows a response on the rough surface which is substantially better than that exhibited by the conventional tractor over virtually the entire spectrum of frequencies covered in this analysis.

It should be understood, however, that the above findings do not provide a representative view of the contrast in ride behavior between COE and conventional tractors since the differences in ride quality between vehicles is highly dependent upon the road surface and loading condition, at minimum.

8.5 Concluding Remarks

The foregoing discussion has addressed truck ride vibrations in only a very cursory fashion. Clearly, the vibrational phenomena which prevail on a tractor-semitrailer derive from a complex set of interactions between excitation input mechanisms, transmission mechanisms, and mechanical resonances.

The exercise conducted in this study serves, however, to provide NHTSA with data demonstrating the general nature of commercial vehicle ride. In particular, measurements have been made showing that the ride response of a heavy road tractor:

1) is comprised of responses whose energy level is highest in the 0 to 15 Hz range

2) is comprised of simple (though, perhaps, coupled) "rigid-body" modes of motion (e.g., pitch and bounce) in the range of 0 to 6 Hz, followed by more complex modes (including those involving structural flexing) at higher frequencies.

3) includes large, if not dominant, components of acceleration in the fore/aft direction at the driver's seat location

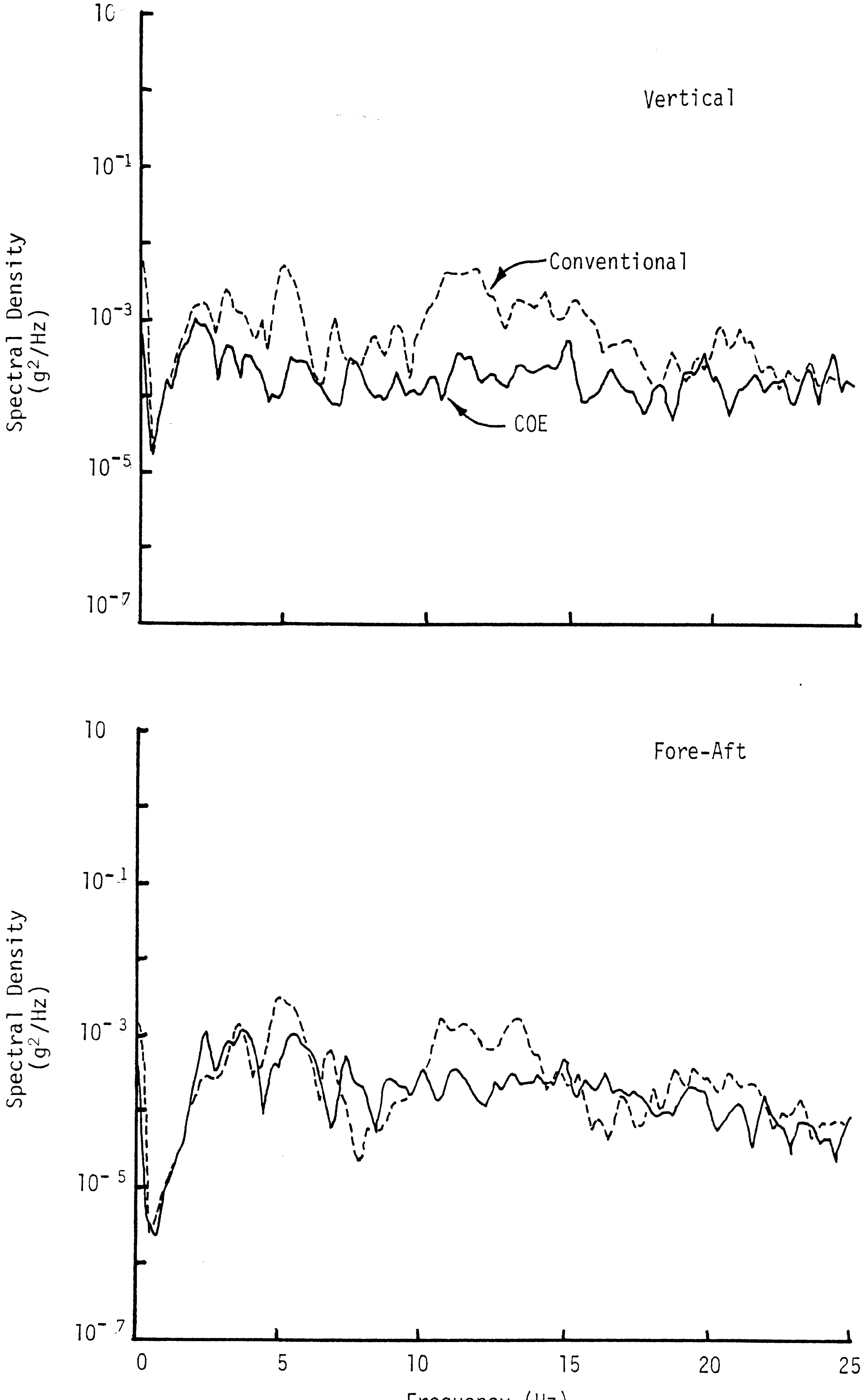


Figure 8.20. PSD's of COE vs. conventional tractor - loaded, 55 mph, site No. 2.

4) can exhibit high energy responses on smooth road surfaces, simply due to wheel and tire nonuniformities

5) is heavily influenced by road surface condition, both in terms of overall absorbed power level and in terms of the resonant match between the peculiar design properties of the vehicle and the spectral content of the road profile

6) is heavily influenced by road speed, especially insofar as the wheel rotation frequency can become matched with lightly damped natural modes of vibration

7) is heavily influenced by the loading condition such that, for example, the bobtail tractor configuration can exhibit much higher energy levels in certain modes of response while being devoid entirely of other modes that prevail when a trailer is attached.

9.0 REFERENCES

1. Ervin, R.D., Winkler, C.B., Bernard, J.E., and Gupta, R.K. "Effects of Tire Properties on Truck and Bus Handling." Final Report, Contract DOT-HS-4-00943, Highway Safety Research Institute, The University of Michigan, Ann Arbor, December 1976.

2. Pacejka, H.B. "Simplified Analysis of Steady Turning Behavior of Motor Vehicles. Parts 2 & 3." Vehicle Systems Dynamics, Vol. 2, No. 4, December 1973.

3. Summer Conference Notes on "Motor Vehicle Performance Measurement and Prediction." Highway Safety Research Institute, The University of Michigan, Ann Arbor, July 22-26, 1974.

4. Bernard, J.E., Winkler, C.B., and Fancher, P.S. "A Computer-Based Mathematical Method for Predicting the Directional Response of Trucks and Tractor-Trailers." Phase II Report, Highway Safety Research Institute, The University of Michigan, Ann Arbor, June 1, 1973.

5. Ervin, R.D., et al. "Vehicle Handling Performance." Final Report, Contract No. DOT-HS-031-1-159, Highway Safety Research Institute, The University of Michigan, Ann Arbor, November 1972.

6. Fancher, P.S., Mallikarjunarao, C., and Nisonger, R.L. "Simulation of the Directional Response Characteristics of Tractor-Semitrailer Vehicles." Final Report, MVMA Project 1.39, Highway Safety Research Institute, The University of Michigan, Report No. UM-HSRI-79-9, March 1979.

7. "Operating Manual - Model 3582A Spectrum Analyzer." Hewlett Packard Co. Loveland, Colorado, March 1978.

8. Healey, A.J., et al. "Measurement of Roadway Roughness and Automobile Ride Acceleration Spectra." Research Report No. 13, University of Texas at Austin, July 1974.

9. VanDusen, B.D. "Analytical Techniques for Designing Ride Quality into Automotive Vehicles." SAE Paper No. 670021, January 1967.

10. LaBarre, R.P., Forbes, R.T., and Andrew, S. "The Measurement and Analysis of Road Surface Roughness." Motor Industry Research Association, Report No. 1970/5.

11. Thomson, W.T. Mechanical Vibrations. Second Ed., Prentice-Hall, Inc., pp. 123-127.

12. Inoh, T. and Aisaka, M. "Tuning Techniques for Controlling Heavy-Duty Truck Shake - Vertical, Torsional and Lateral." SAE Paper No. 730650.

13. Cosby, M.J. and Allen, R.E. "Cab Isolation and Ride Quality." SAE Paper No. 740294, March 1974.

14. Baum, J.M., Bennett, J.A., and Carne, T.G. "Truck Ride Improvement Using Analytical and Optimization Methods." Conference Proceedings, Second International Conference on Vehicle Structural Mechanics, April 18-20, 1977.

15. Ribarits, J.I., Anrell, J., and Anderson, E. "Ride Comfort Aspects of Heavy Truck Design." SAE Paper No. 781067, December 1978.

16. Foster, A.W. "A Heavy Truck Cab Suspension for Improved Ride." SAE Paper No. 780408, February 1978.

17. Lee, R.A. and Pradko, F. "Analytical Analysis of Human Vibrations." SAE Paper No. 680091, January 1968.

18. "Guide for the Evaluation of Human Exposure to Whole-Body Vibration." International Organization for Standardization, Standard ISO 2631.