CONSIDERATIONS OF TRACTOR YAW STABILITY

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SYNOPSIS:

The mechanisms giving rise to a strong tendency toward oversteer in loaded commercial vehicles is explained and experimental substantiation of the phenomenon is presented. The handling diagram is introduced as a useful display for illustrating this tendency as a function of lateral acceleration level and for identifying the point of yaw instability. Results obtained from both analytical and experimental methods are presented to show the sensitivity of tractor yaw stability to design and operating variables. The paper establishes that:

1) the tendency toward oversteer arises because of a peculiar nonlinearity of the pneumatic tire together with a) high payload centers of gravity in loaded trucks and b) the fact that rear suspensions on power units are much stiffer than front suspensions.

2) a critical speed value typically exists beyond which the oversteer truck is classically unstable in yaw.

3) the phenomenon is primarily influenced by suspension roll stiffness and roll center height, tire cornering properties, and load placement.

In addition to the steering gain issues associated with the oversteer tendency, certain transient lag aspects of the response of tractor semitrailers are discussed for their significance to the driver's task in closing the control loop on steering.

CONSIDERATIONS OF TRACTOR YAW STABILITY

The pneumatic tire behaves as a linear spring mechanism in producing cornering forces for small levels of perturbation about the zero slip angle condition and for a given value of vertical load. For the case of passenger cars, the changes in slip angle and vertical load occurring with increasing lateral acceleration, are such that the "linear regime" of tire and vehicle behavior is found to encompass all maneuvers within approximately 0.3 g lateral acceleration (A_y). When passenger cars do begin to exhibit a decidedly nonlinear yaw response to steering, it is the result, primarily, of the nonlinear relationship between side force and slip angle at a given load.

Heavy commercial vehicles, on the other hand, tend to exhibit linear yaw response to steering over only a very narrow range of lateral acceleration (say, $A_y < .1$ g). The primary mechanism giving rise to an "early" transition to nonlinear behavior involves the combined interaction of

a) the sensitivity of the tire's side force/slip angle relationship to vertical load and

b) the very strong "lateral load transfer" function deriving from the typically low value of the ratio of track width to c.g. height for loaded commercial vehicles.

Figure 1 illustrates the mechanism by which the change in vertical loads borne by left- and right-side tires on an axle causes a <u>net</u> change in the total side force produced on the axle in question. Considering the illustrated case of an assumed nominal slip angle condition of $\alpha = 4°$, we see that, for a left-hand turn in which $A_y = 0.35$ g's, the right-side tire increases in load from 5,000 lbs to 8,500 lbs, while the left-side tire decreases in load from 5,000 lbs to 1,500 lbs. Due to

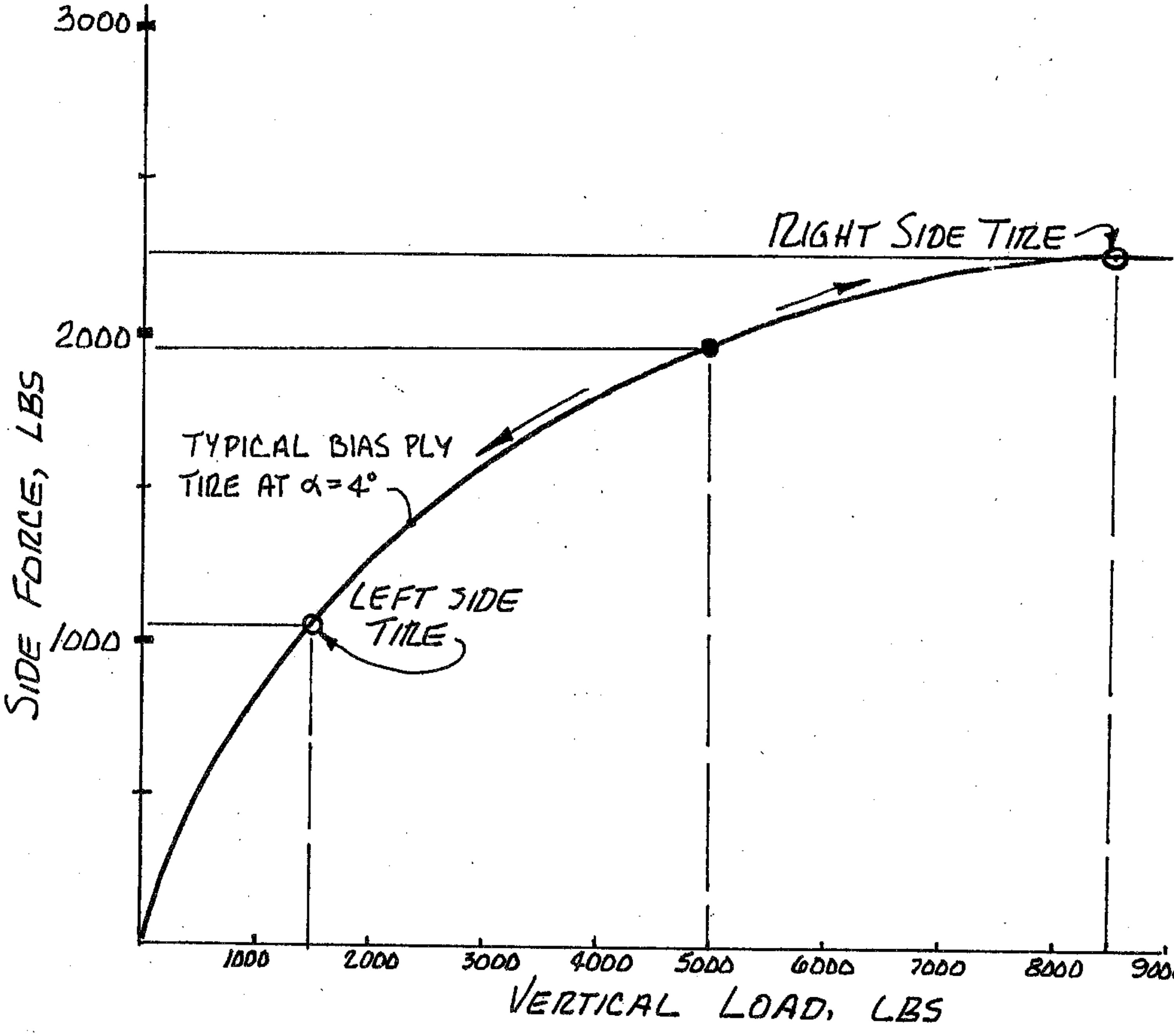
3000
2000
1000
SIDE FORCE, LBS
RIGHT SIDE TIRE
TYPICAL BIAS PLY TIRE AT α = 4°
LEFT SIDE TIRE
1000
2000
3000
4000
5000
6000
7000
8000
VERTICAL LOAD, LBS

Figure 1

the curvature of the F_y dependence upon F_z, the increase in side force experienced by the right-side tire is less than the reduction in side force experienced by the left-side tire. Accordingly, it can be said that the general influence of lateral load transfer upon the generation of side force at a given axle is to reduce the net force produced at a given slip angle, or to cause an effective increase in the cornering compliance, $\partial\alpha/\partial F_y$, at that axle position.

The "load transfer sensitivity" function becomes of special importance in the case of heavy commercial vehicles because of the fact that trucks and truck-tractors typically employ much stiffer suspensions at their rear-axle positions than they do at the front. The stiffness differences basically reflect the fact that payload weight is borne primarily at the rear axles such that an extra high level of rear-spring stiffness is needed to keep static deflections within reasonable bounds. As a result of this design feature, the roll stiffnesses prevailing at the rear-axle position are usually much larger than that prevailing at the front. When the roll stiffness levels at the front and rear axles of heavy trucks and tractors are ratioed to the number of tires employed on the respective axles (i.e., considering single tires installed at the front and duals at the rear), the front roll stiffness <u>per tire</u> is typically on the order of 40 percent of that found at the rear. Also, the front roll center height is typically 20 percent lower than that at the rear, causing the rear tires to carry a larger fraction of the total "kinematically-reacted" roll moment than is simply implied by the proportion, a/ℓ.* Further, since the typical heavy vehicle incorporates a frame having a rather low level of torsional stiffness, the rear portion of the frame (or the fifth wheel coupling) can, in certain vehicles, rotate through a considerably larger roll angle than does the front portion of the frame, thus causing the rear tires to bear an even larger fraction of the load transfer burden.

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*Consult general roll/yaw model of a unit vehicle to confirm that the roll moment "passed through" the rear roll center, having height, h_{rc}, is $WA_y h_{rc}(a/\ell)$.

Because of the exaggerated proportion of load transfer occurring at the rear axle, the rear tires tend to run at extraordinarily large slip angles as the level of lateral acceleration, and thus lateral load transfer, increases. Since the understeer level of the vehicle is directly expressed by the difference in front and rear slip angles, viz.,

$$\alpha_1 - \alpha_2 = \frac{\ell}{R} - \delta = - \frac{UV^2}{gR} \qquad (1)$$

where

α_1, α_2 = front and rear tire slip angles, respectively, rad.

δ = front-wheel steer angle, rad.

ℓ = wheelbase, ft.

R = path radius, ft.

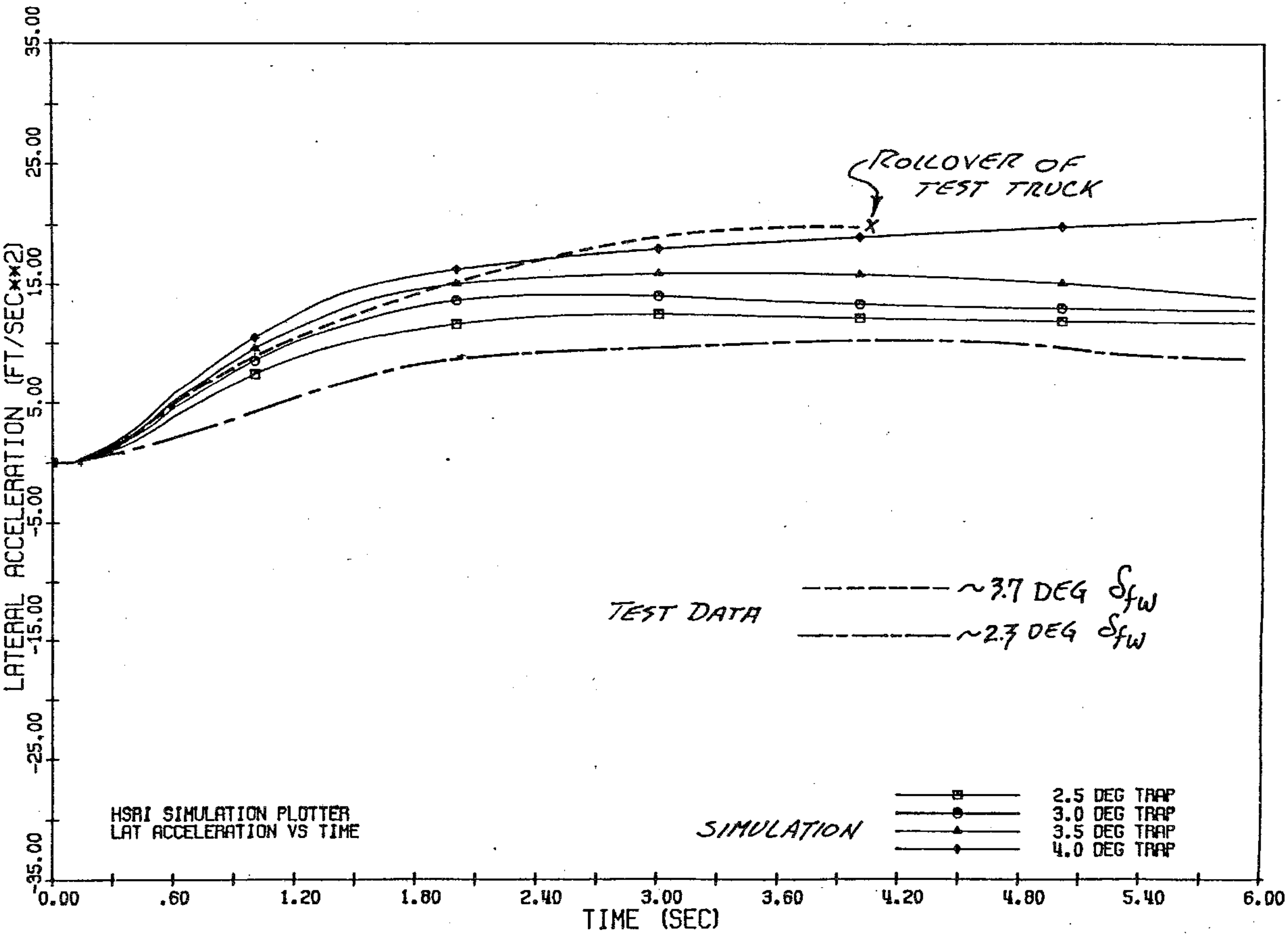
U = understeer gradient, rad/g

V = velocity, ft/sec

g = acceleration of gravity, ft/sec^2

it can be shown that growth in the rear tire slip angle, α_2, relative to the front tire slip angle, α_1, produces a more negative (i.e., more oversteer) value in the understeer gradient. (Note that U is now being considered as a variable, $U(A_y)$, rather than as a constant such as is presumed in linear analyses.)

Because of the declining understeer level with increasing lateral acceleration, it is quite possible that yaw instability will be encountered with typical trucks and truck-tractors at intermediate levels of lateral acceleration and at normal highway speeds. Shown in Figures 2 and 3, for example, are time histories of lateral acceleration and body sideslip angle, β, for a loaded two-axle heavy truck conducting a sequence of trapezoidal steer maneuvers at 50 mph. Figure 2 shows both simulated and test results leading up to a condition in which, in response to a steer input of 3.7 degrees of front-wheel angle, the vehicle exhibits a divergent response culminating (in the case of the test vehicle) in rollover within 4.0 seconds of maneuver initiation. Figure 3 shows, by means of



2-AXLE TRUCK: LOAD SENSITIVE TIRE RUNS AT 50 MPH

Figure 2

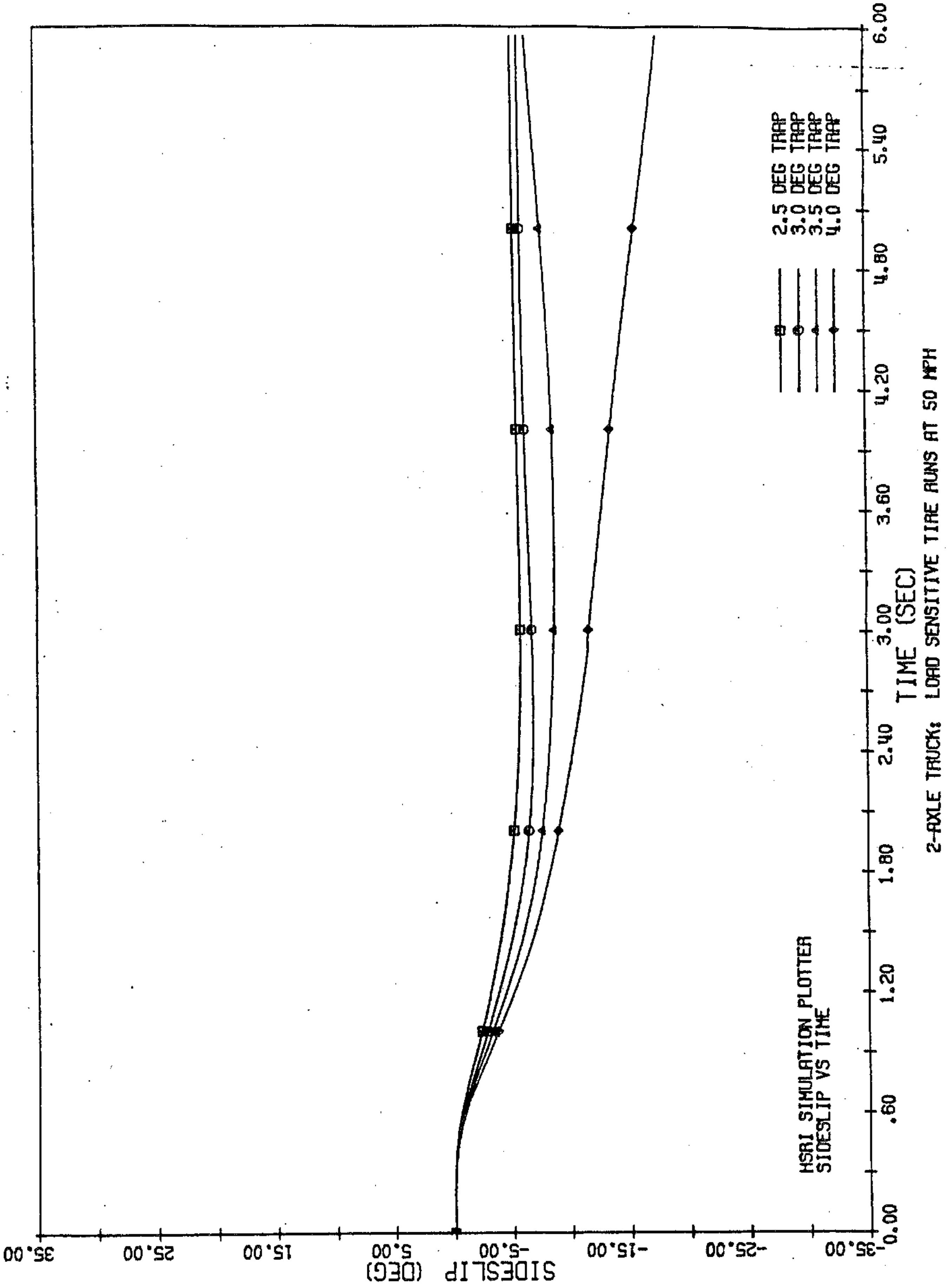

HSRI SIMULATION PLOTTER
SIDESLIP VS TIME
2.5 DEG TRAP
3.0 DEG TRAP
3.5 DEG TRAP
4.0 DEG TRAP
TIME (SEC)
0.00
.60
1.20
1.80
2.40
3.00
3.60
4.20
4.80
5.40
6.00
SIDESLIP (DEG)
-35.00
-25.00
-15.00
-5.00
5.00
15.00
25.00
35.00
2-AXLE TRUCK: LOAD SENSITIVE TIRE RUNS AT 50 MPH

Figure 3

simulation results, that the β time history diverges in response to a 4.0-degree steering input, although response to inputs of 2.5-, 3.0-, and 3.5-degree front-wheel angle were convergent (albeit, sluggish, in the case of the 3.5-degree input). These data illustrate a classic case in which the "usable" regime of lateral acceleration in steering maneuvers is not bounded simply by the rollover threshold, but rather by the occurrence of an unstable yaw response at an A_y level well below the rollover limit.

As a basis for general illustration of the steady-state cornering behavior of heavy vehicles, the following discussion presents the handling diagram and then employs it to summarize a number of the basic sensitivities of vehicle yaw stability to design and operating parameters.

Introduction to the Handling Diagram for Heavy Vehicles

One concern in the examination of truck or tractor-semitrailer yaw responses is to conclude whether an unstable yaw response is present. Analysis has shown (see Chapter 17) that the mode of instability which is of essential importance in the case of the tractor-semitrailer 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 truck and tractor yaw instability often prevails with a relatively low rate of divergence, such that the existence, or not, of an instability (which may be 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 is convenient to adapt a technique called the "handling diagram," which was broadly developed by Pacejka* to facilitate the study of steady-state yaw response to steering, as described by the previously-cited expression:

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*See references at end of chapter.

$$\delta = \frac{\ell}{R} + \frac{UV^2}{gR}$$

The handling diagram is simply a plot of this relationship, namely, the term (V^2/gR) is taken as the dependent variable while $\ell/R - \delta$ 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 unit vehicle 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.

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 5, 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 when one is concerned with yaw instability being 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.

For a vehicle exhibiting nonlinear directional behavior, the criterion for the directional stability of a steady turn is given by the inequality

$$\frac{\partial(\delta)}{\partial V^2/gR} > 0 \qquad (2)$$

(Note: The above condition can be derived from the linear differential equations which describe the motion of the truck or tractor-semitrailer

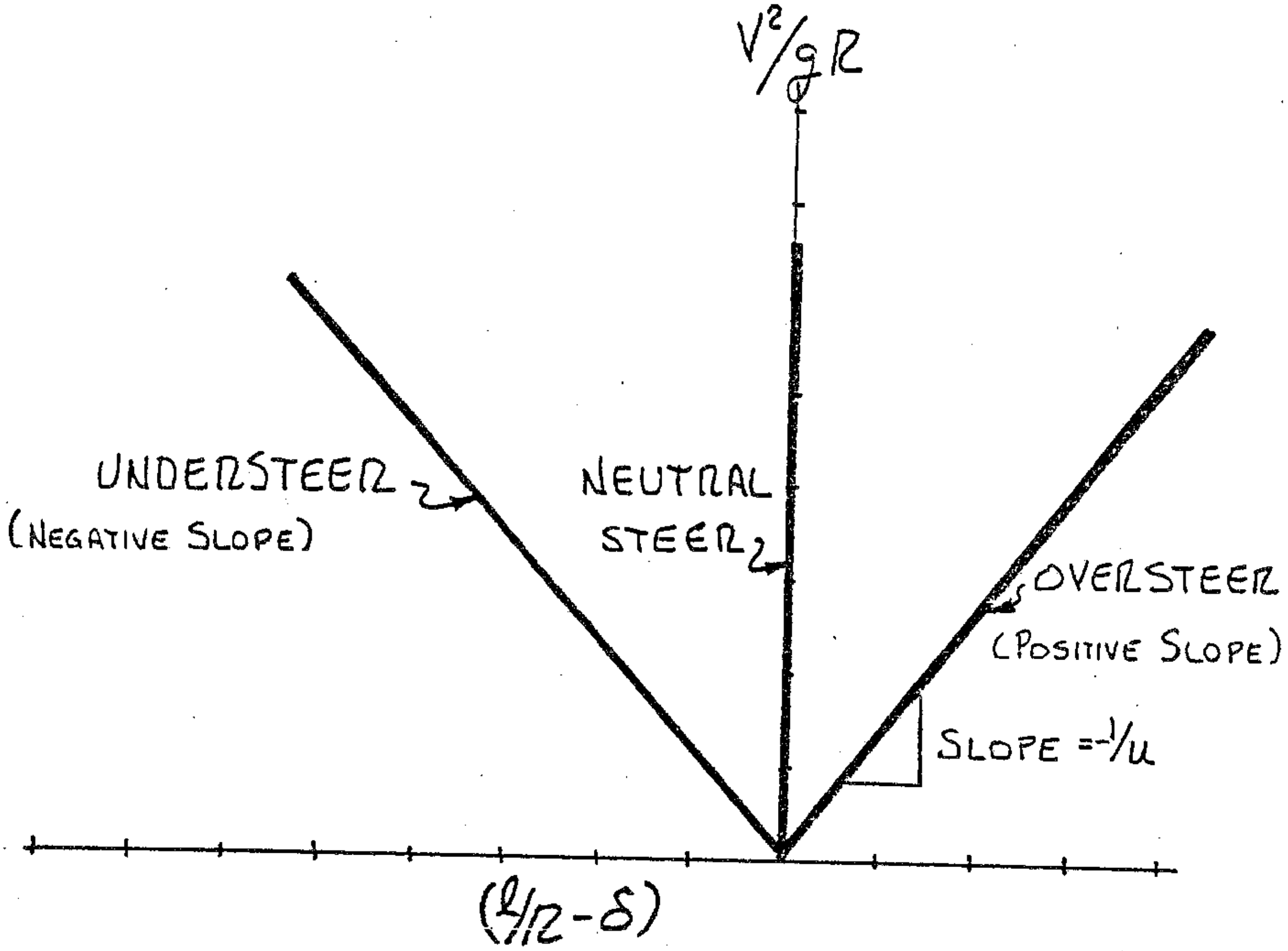

Figure 4. Example handling diagram showing three possible characteristics for a passenger car with linear tire properties.

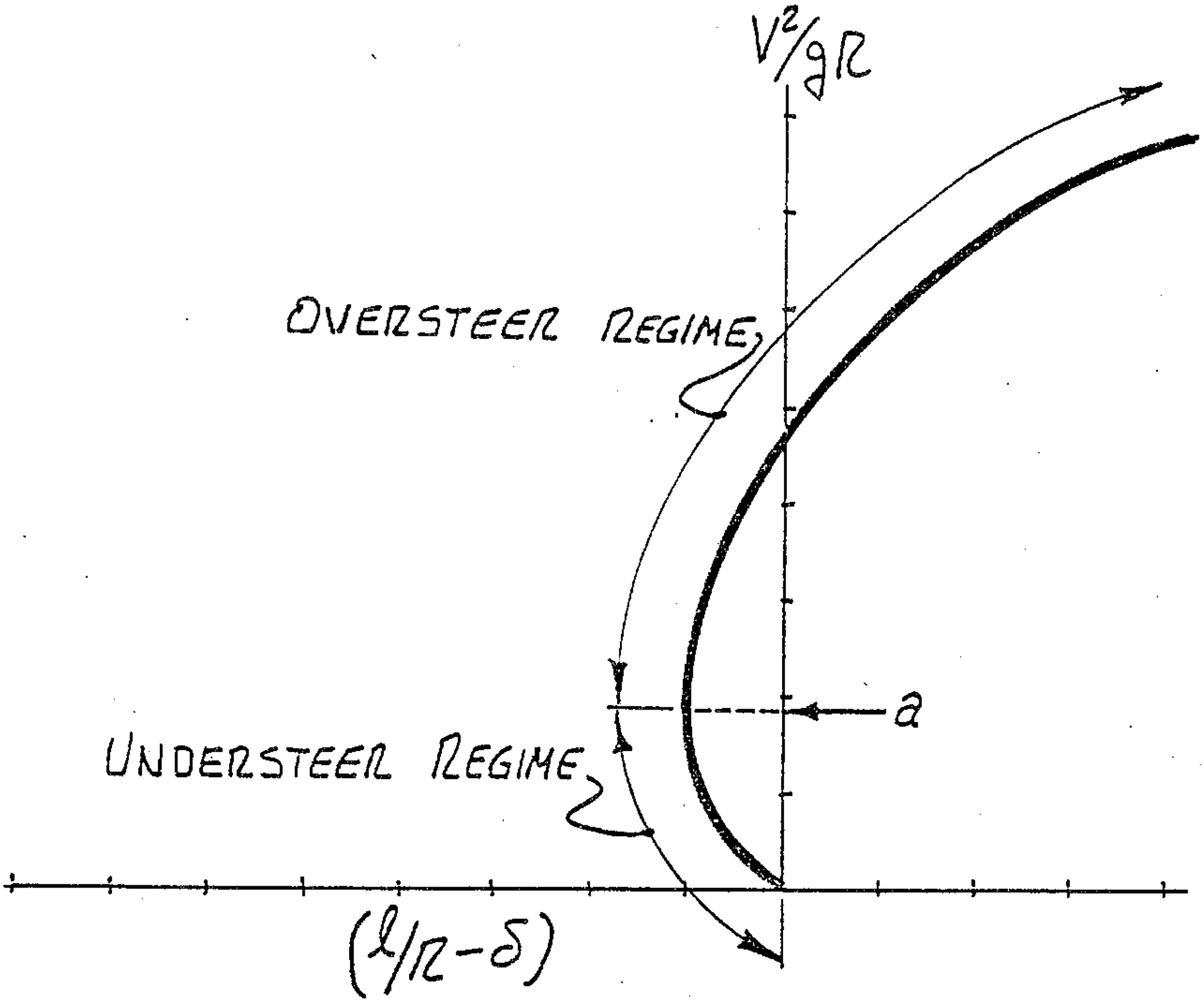

Figure 5. Example handling diagram, typical of heavy trucks, showing a transition from understeer to oversteer behavior at an intermediate level of lateral acceleration.

for small perturbations about a steady state.) The inequality (2) can also be written as

$$\frac{\partial(\ell/R - \delta)}{\partial V^2/gR} < g\ell/V^2$$

or

$$\frac{\partial V^2/gR}{\partial(\ell/R - \delta)} > V^2/g\ell \qquad (3)$$

When the stability condition is expressed in the form of the inequality (3), the left-hand side corresponds to the slope of the handling diagram and the right-hand side corresponds to the slope of the constant velocity line (which is superimposed on the handling diagram using (L/R, a_y) coordinates). Therefore, the condition for directional stability can also be stated as follows:

A two-axle tractor/single-axle trailer combination traveling with a forward velocity, V, is directionally stable (does not exhibit yaw divergence) at lateral acceleration levels for which the local slope of the handling diagram is steeper than the slope of the constant velocity line that corresponds to velocity, V. Lines of critical slope, or critical velocity, are shown in Figure 6, 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 7, 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, $\ell/R - \delta$, implies that the diagram becomes peculiarly scaled to the individual vehicle's wheelbase. For the case of three-axle trucks and tractors, with the wheelbase represented by the distance from the front axle to the mid-position of the rear tandem pair of axles, it is not possible to represent the vehicle by way of a single, unique handling

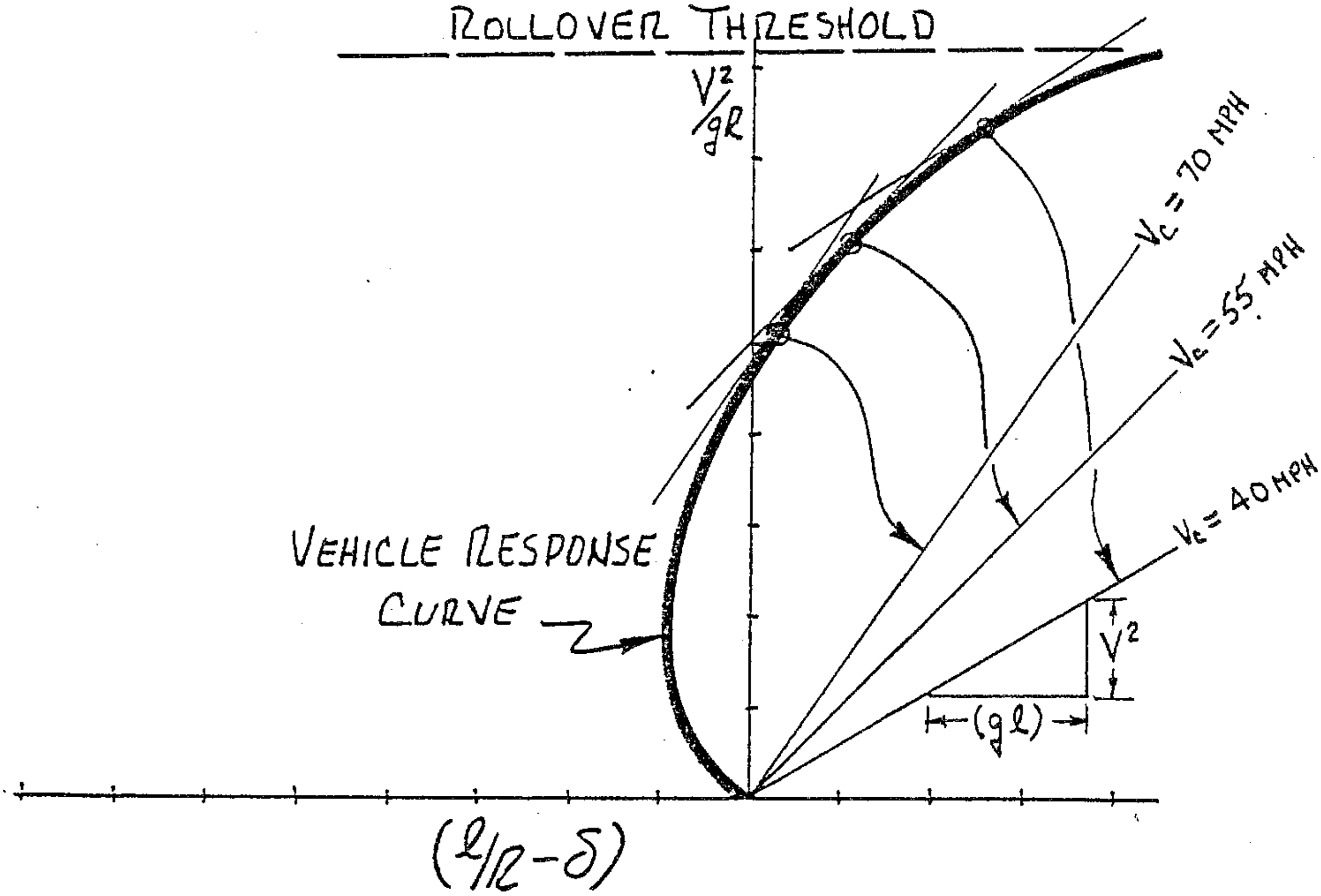

Figure 6. Example handling diagram showing three operating points at which the local slope has been evaluated and used to identify an equal-slope ray defining the critical velocity at that operating point.

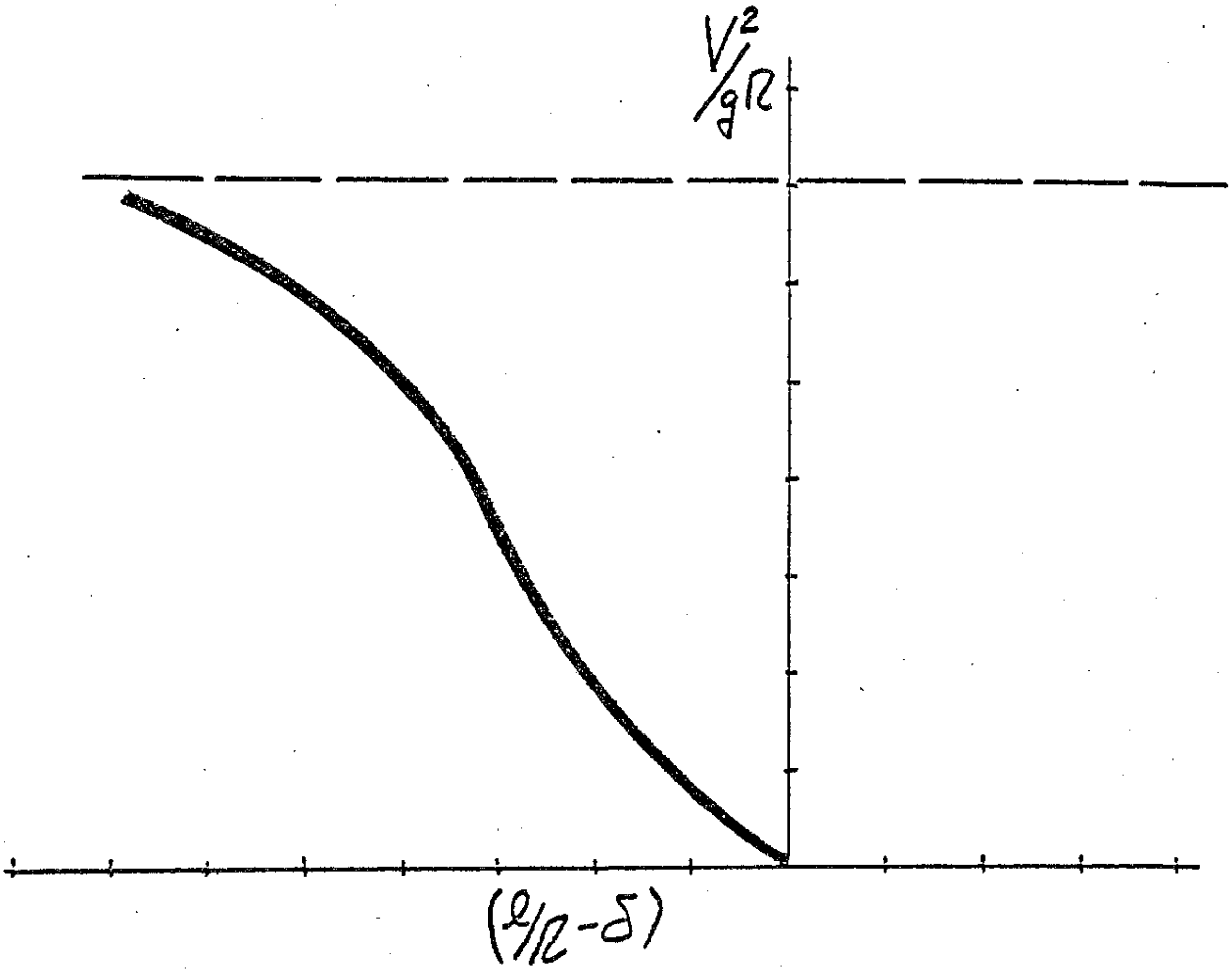

Figure 7. Example handling diagram showing a nonlinear but continuously-understeer behavior for which no yaw instability is possible.

curve. (Please note: A more detailed discussion of the application of the handling diagram to the case of tandem-axle tractors is appended to this chapter.) 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 8. 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 shope, 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,

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

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 (1) becomes meaningless. The yaw stability threshold numeric, A_{y_c}, will be employed later as the primary measure for discriminating among the influences of various design and operating parameters on yaw stability.

In all handling diagrams which follow, the abscissa variable ($r\ell/V - \delta$) is 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 <u>quasi-steady</u> conditions which prevailed in certain simulated maneuvers to be discussed. 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.,

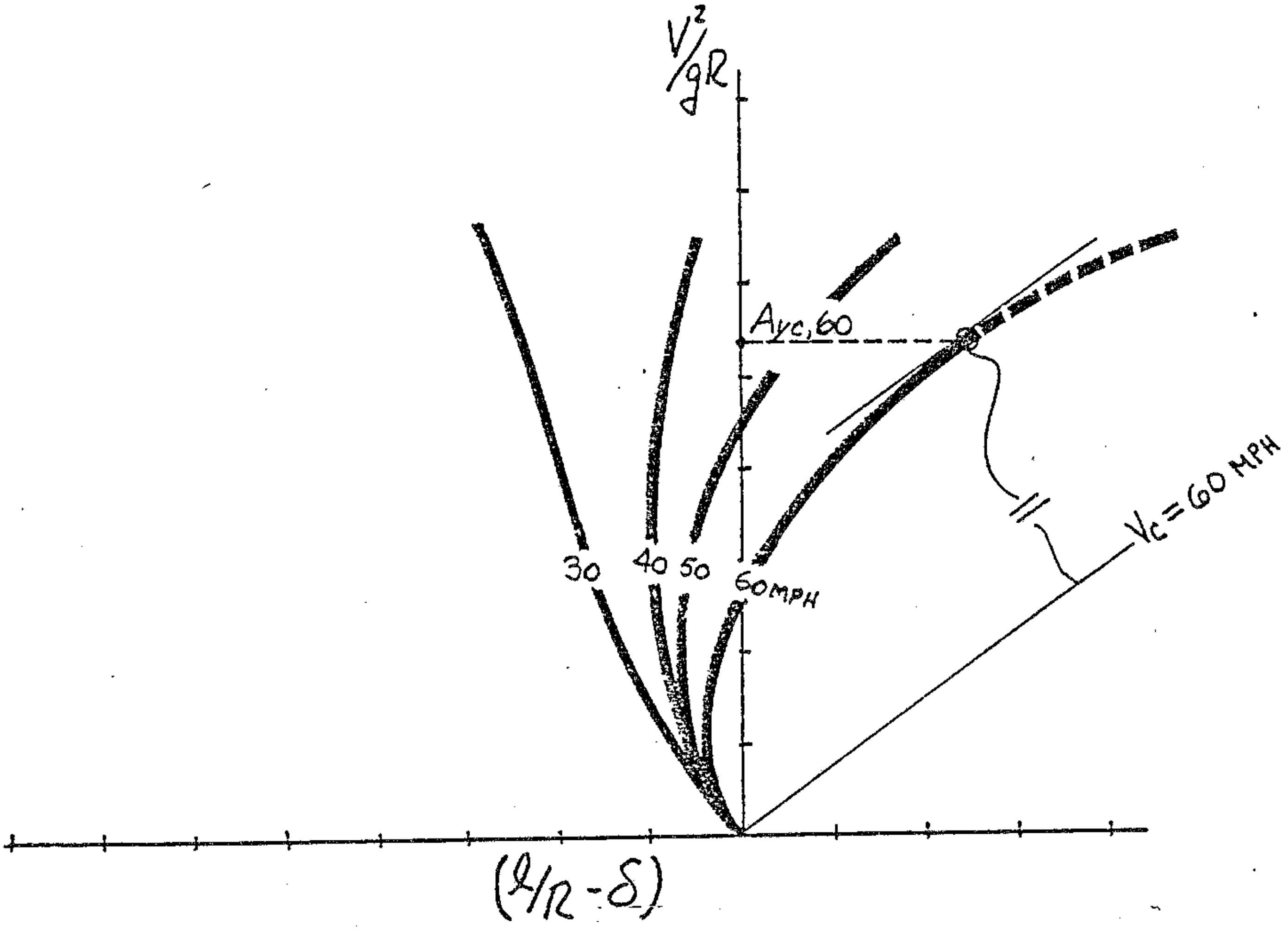

Figure 8. Example handling diagram showing a full family of curves for a tandem-axle tractor.

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

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.

Parametric Sensitivities—Steady-State Response to Steering

In this section, two categories of parametric sensitivities will be presented for cases involving various configurations of tractor-semitrailer. The indicated sensitivities, in most cases, can also be considered as qualitatively descriptive of the corresponding parametric sensitivities of straight trucks. Since concern for yaw instability has been shown earlier to stem primarily from mechanisms related to lateral load transfer, the sensitivities of interest all involve loaded, rather than empty, vehicles.

The parametric sensitivities to be shown here have been explored through computerized simulation using a quasi-steady-state maneuvering condition as an economical means of evaluating the relative significance of various system parameters. Using a slow ramp input of steer level, the entire lateral acceleration range is examined in a single computer run. In a subsequent section of this chapter, the transient content of the ramp-steer maneuver, as well as other transient maneuvers, will be specifically addressed.

Parametric sensitivities are presented below in the following order:

1) Influence of roll stiffness distribution as perturbed over a very broad range of values through changes in:
    a) torsional stiffness of tractor frame
    b) auxiliary front roll stiffness.

2) Combined influence of the above variations with an accompanying mix in tire types on front and rear axles.

3) Combined influences of typically-varied values of the following characteristics:

   a) tractor tire installations

   b) tractor suspension stiffnesses

   c) fifth wheel placement

   d) stiffness of semitrailer suspension

   e) payload c.g. height

Influence of Roll Stiffness Distribution

In this section, we will discuss the influence of tractor roll stiffness distribution on yaw stability. Specifically, the parameter combinations, shown in Table 1, have been employed in a matrix of simulated ramp-steer maneuvers. (Note that the selected parametric variations

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Table 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

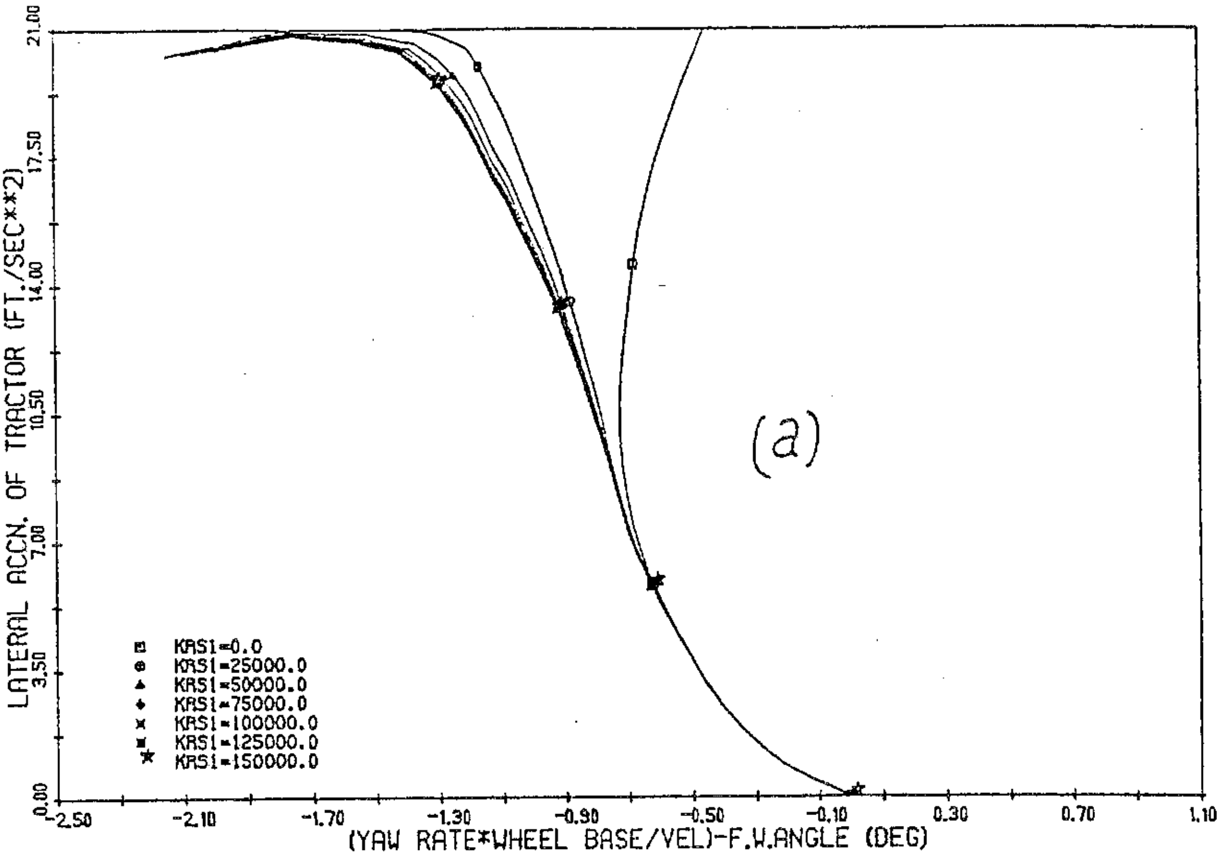
*Note [Baseline values of front suspension roll stiffness were 9256 in-lb/deg for the simulated two-axle tractor and 13,385 in-lb/deg for the simulated three-axle tractor. The listed parameter is auxiliary, or in parallel with the suspension roll spring.]

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

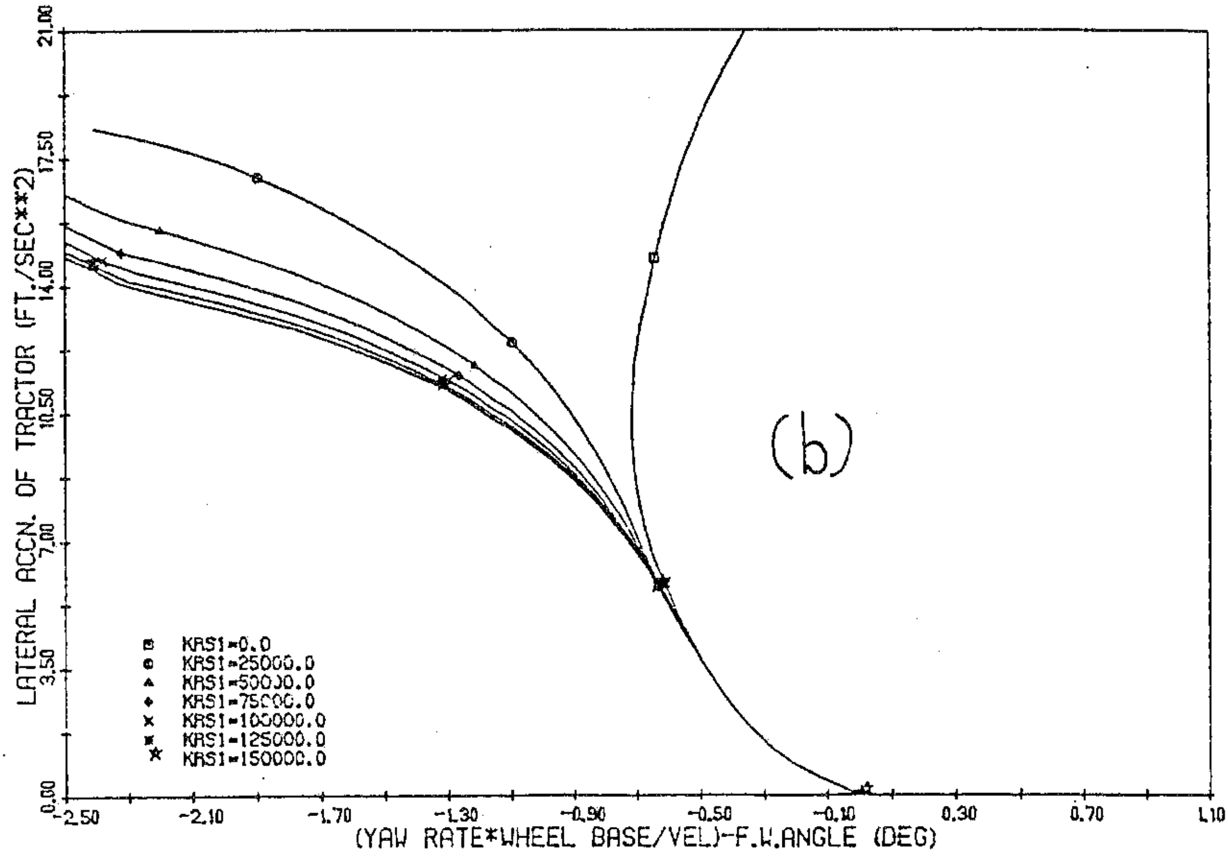
are all in the direction of creating a more front-biased distribution of tractor roll stiffness.) In the simulations to be discussed, the initial velocity is 50 mph, and the ramp rate of front-wheel steer angle is 1.5 deg/sec. As shown in the table, calculations have been performed for two tractors and two semitrailers. One tractor is a two-axle unit having a 10.8K rated front axle and a 23K rear axle. The second tractor is a three-axle vehicle having a 12K front axle and a 34K rear tandem rating. Roll stiffnesses of the frame and the front suspension have been varied over a 7x7 matrix for each vehicle combination. The lowest value of each stiffness parameter represents a baseline state found on actual heavy 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.

Shown in Figure 9 are examples of the range of handling curves obtained for the three-axle tractor/two-axle 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 this vehicle combination to produce the envelope of handling curves shown in Figure 10.

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 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 polarity of the torsional moment passed through the frame in the baseline configuration during a steady turn. In the case of Figure 10, apparently the addition of the frame stiffener served



RIB TIRES,FRAME 20000 IN.LB/DEG



RIB TIRES,FRAME 140000 IN.LB/DEG

Figure 9. 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.

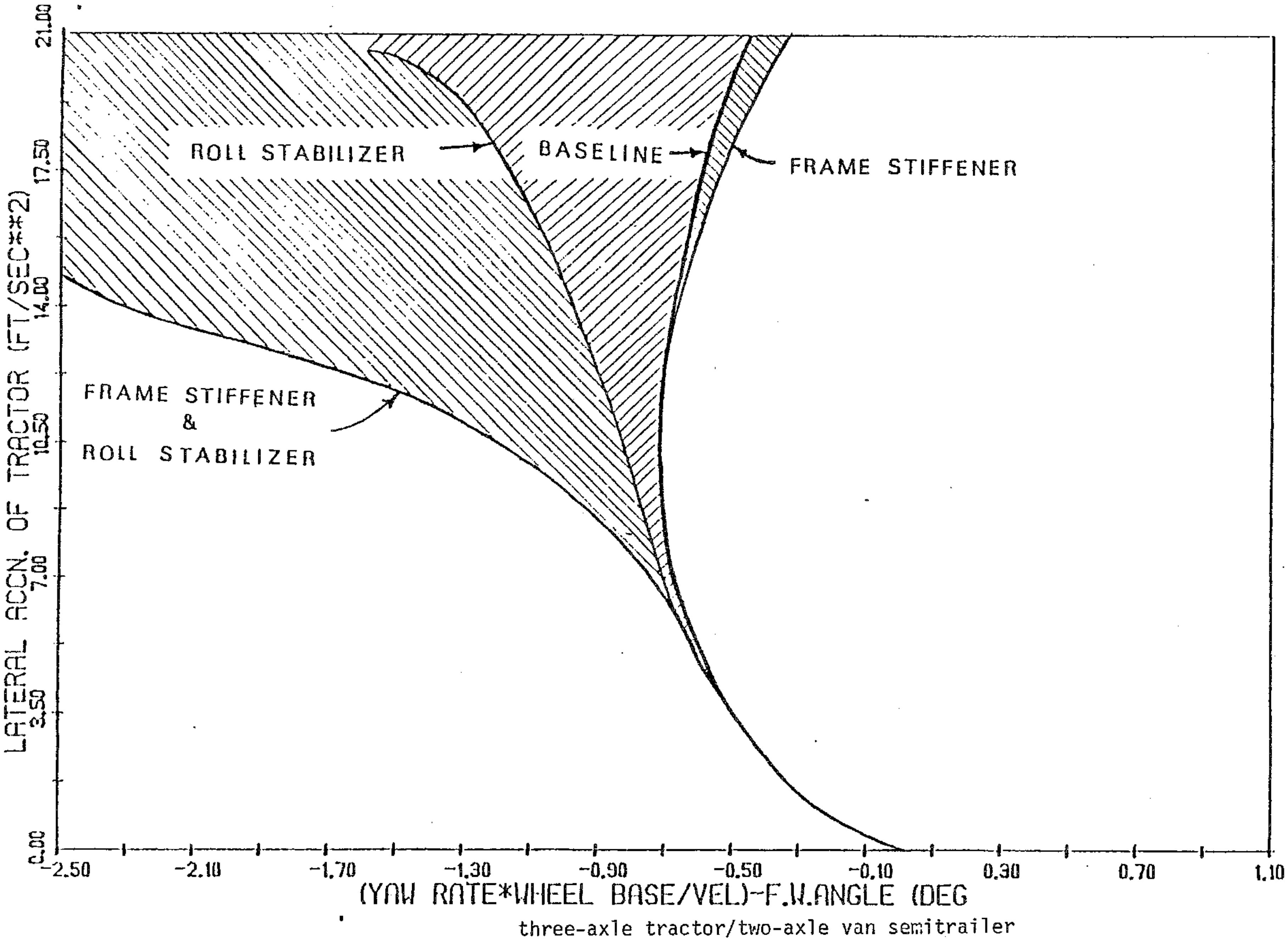


Figure 10. Envelope of handling curves obtained with varied stiffness parameters.

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.)

Figure 11 shows a similar gross trend (except for the frame stiffener-only case) for a combination vehicle involving the two-axle tractor with the same two-axle semitrailer. Firstly, we see that the baseline two-axle tractor exhibits a substantially higher value of oversteer gradient than the three-axle tractor a high levels of A_y. Nevertheless, the first increment in front suspension roll stiffness, alone, is sufficient to dramatically alter performance in the understeer direction.

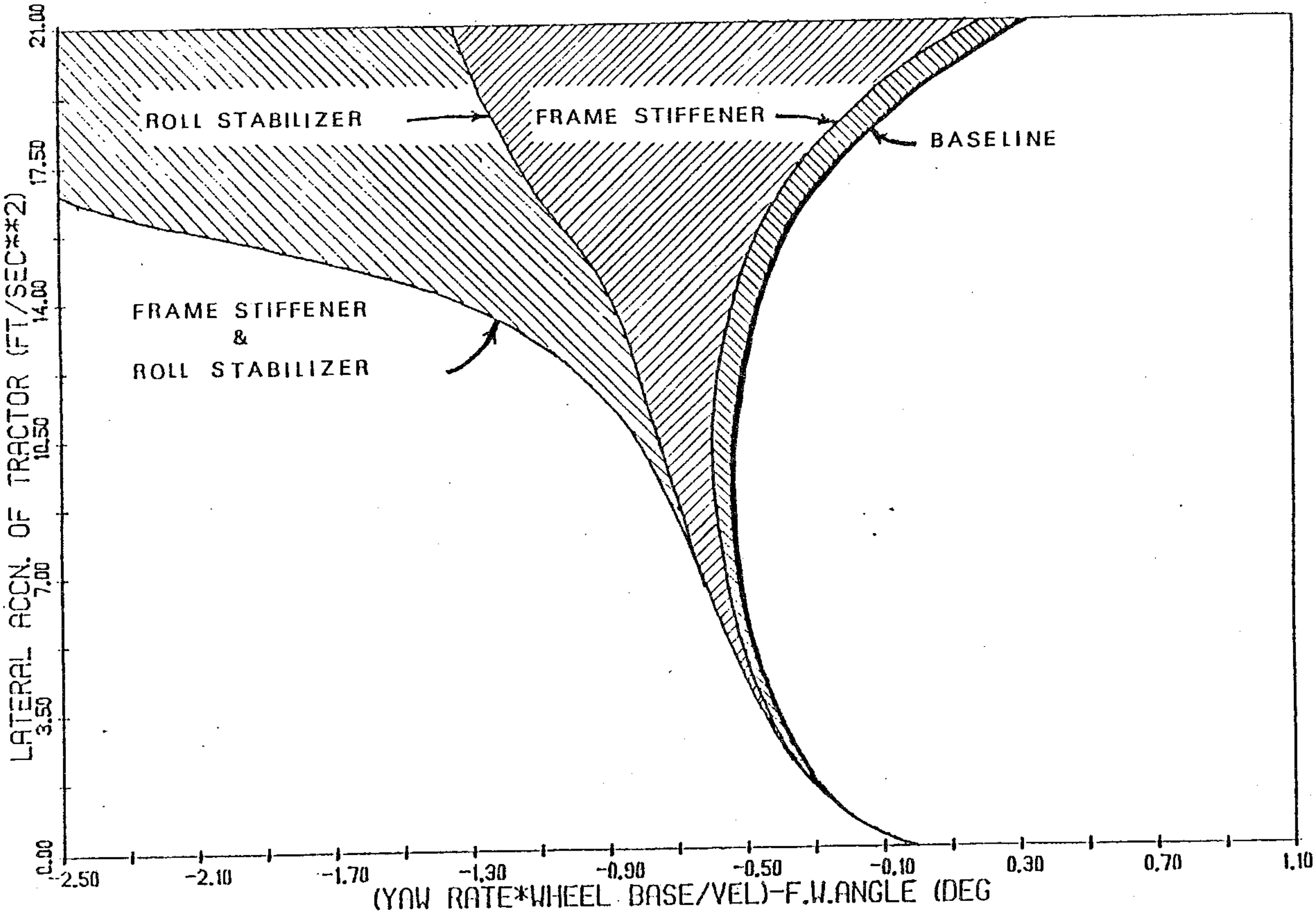
By way of elaboration on this matrix of results which apply to tractors outfitted uniformly with the same rib tires all around, one may observe the following:

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

b) Increases in front-suspension roll stiffness, taken alone, provide a much more significant means 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 presented. 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



two-axle tractor/two-axle van semitrailer

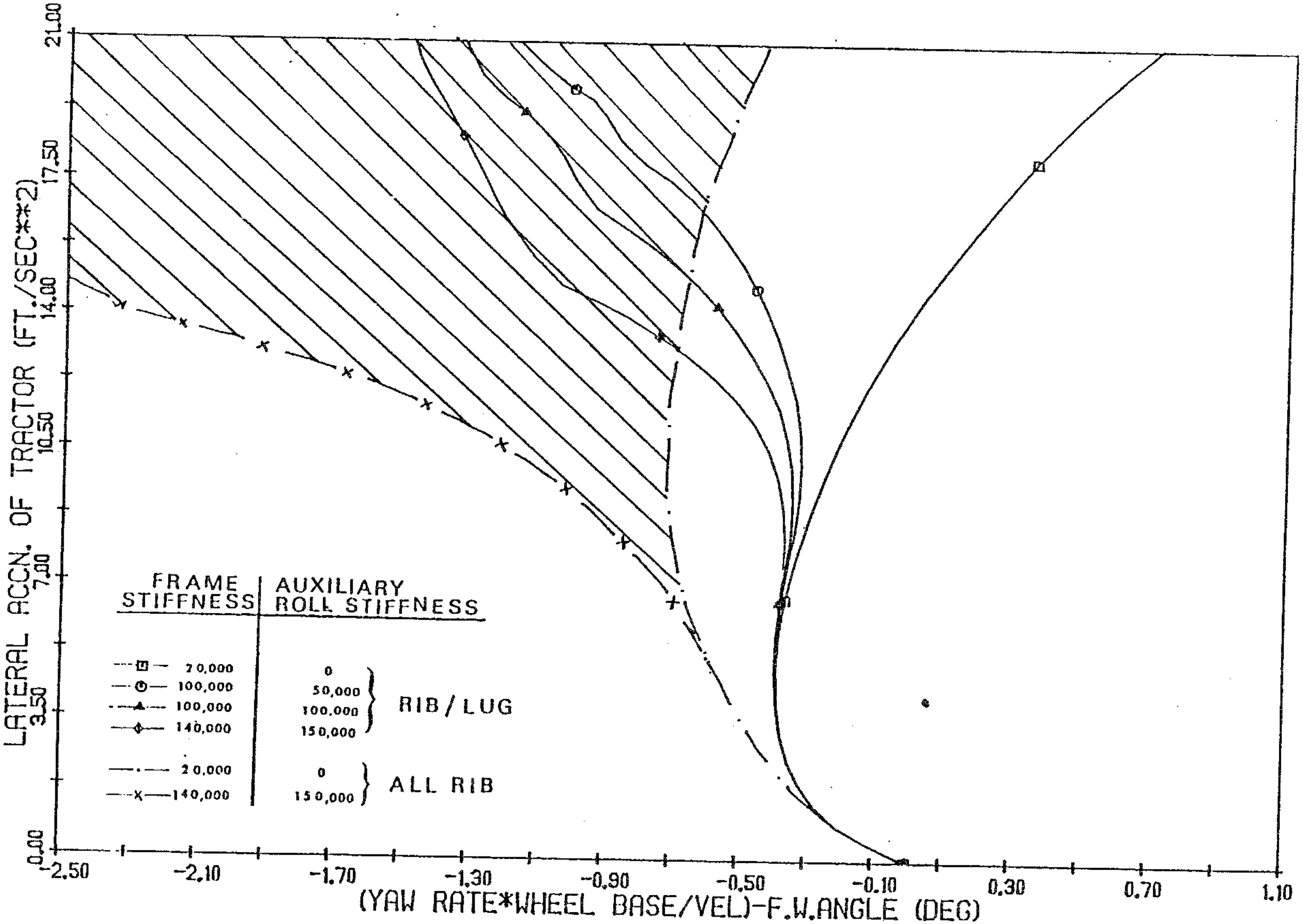
Figure 11. Envelope of handling curves obtained with varied stiffness parameters.

> 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 percent of the highest estimated level of understeer was obtained by a frame stiffness parameter of 80,000 in-lb/deg.

Shown in Figure 12 are four selected cases of the three-axle tractor/van semitrailer 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 is known to reduce 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 10. Here we see that the expected trends in improved stability still accrue from the increases in the "stiffening" 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.

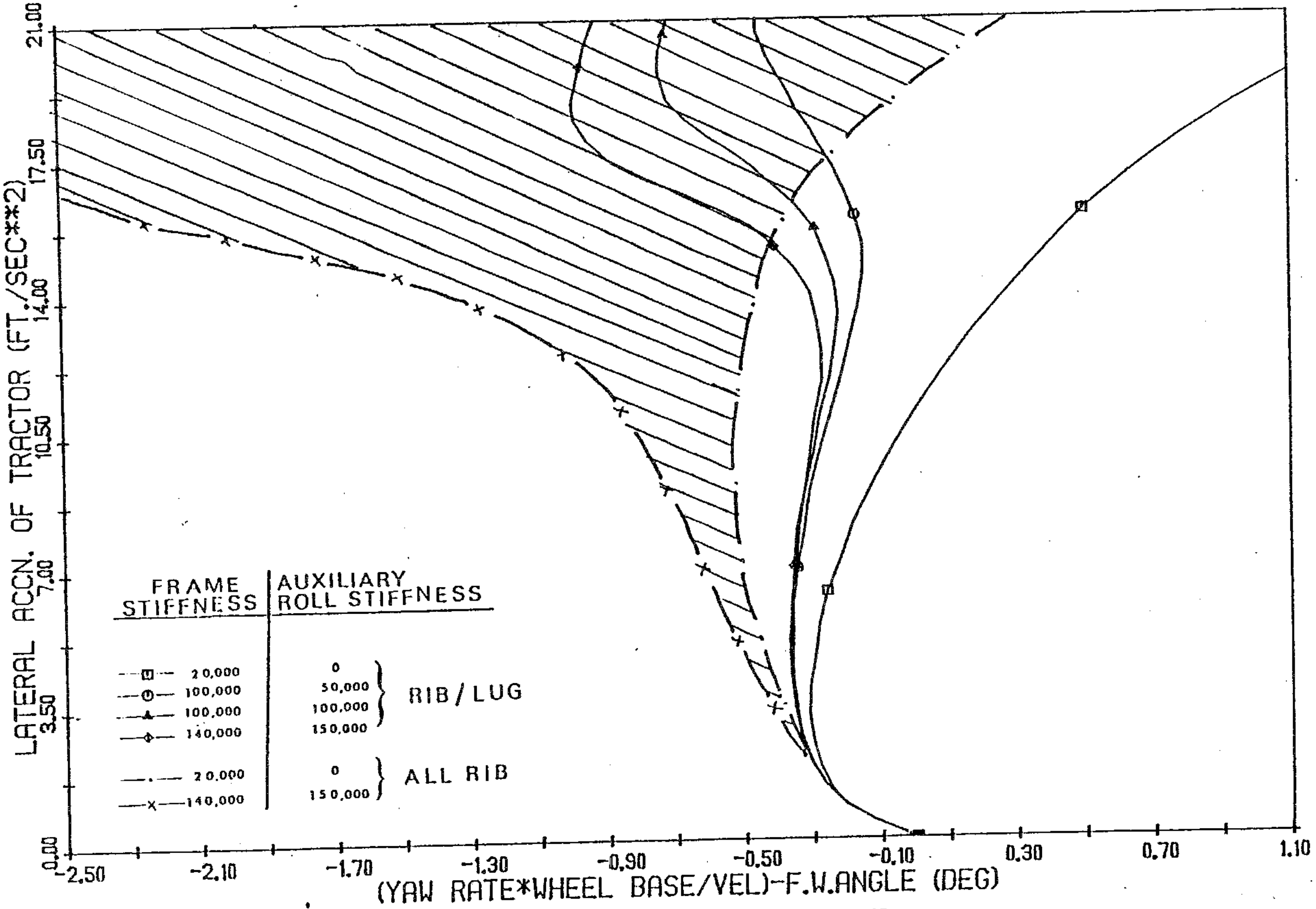
The same basic behavior is seen in Figure 13, which shows results obtained for the two-axle tractor having lug tires on its drive axle, and combined with the two-axle van trailer. In these cases, the baseline vehicle is sufficiently oversteer at high levels of A_y that a large degree of combined frame and front-suspension roll stiffness is needed to achieve even a marginally understeer condition.

Moreover, these calculations have illustrated that combined increases in frame and front-suspension roll stiffness constitute an effective mechanism for eliminating the potential for a yaw instability during cornering, although rather large increments in these parameters are needed to assure an understeer behavior over the entire range of lateral accelerations.



three-axle tractor/two-axle van semitrailer

Figure 12. Handling curves for tractor with mixed rib- and lug-tread tires, showing influence of selected variations in stiffness parameters.



two-axle tractor/two-axle van semitrailer

Figure 13. Handling curves for tractor with mixed rib- and lug-tread tires, showing influence of selected variations in stiffness parameters.

Effects of Typical Variations in Design and Operating Variables on Nonlinear Yaw Behavior

A second set of simulations are presented here showing 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 conditions. A matrix of runs is defined to cover cases of two- and three-axle tractors coupled, respectively, to one- and two-axle trailers upon which five selected parameters were realistically varied. Parameter variations have been chosen here such that nonlinear mechanisms in the vehicle's response tend to dominate. Particularly, the variations tend to increase the magnitude of the lateral load transfer at the tractor's drive axle(s) or to otherwise directly reduce the vehicle's understeer level.

Five baseline tractors are represented in the calculations, namely, two two-axle cab-over-engine-type vehicles and three three-axle tractors, one of which is of the cab-over style while the other two are configured as conventional cabs. The primary distinction between tractors having the same number of axles is the wheelbase dimension. As shown in Figure 14, the two-axle tractors have wheelbases of 110 inches and 140 inches, whereas the three-axle tractors have wheelbases of 145 inches, 165 inches and 200 inches. The single- and tandem-axle semitrailers are described by parameters representing a torsionally rigid van-type construction.

The following variations in parameters are covered:

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 are selected to represent a typical spread in the cornering force properties of rib- and lug-tread bias-ply tires.)

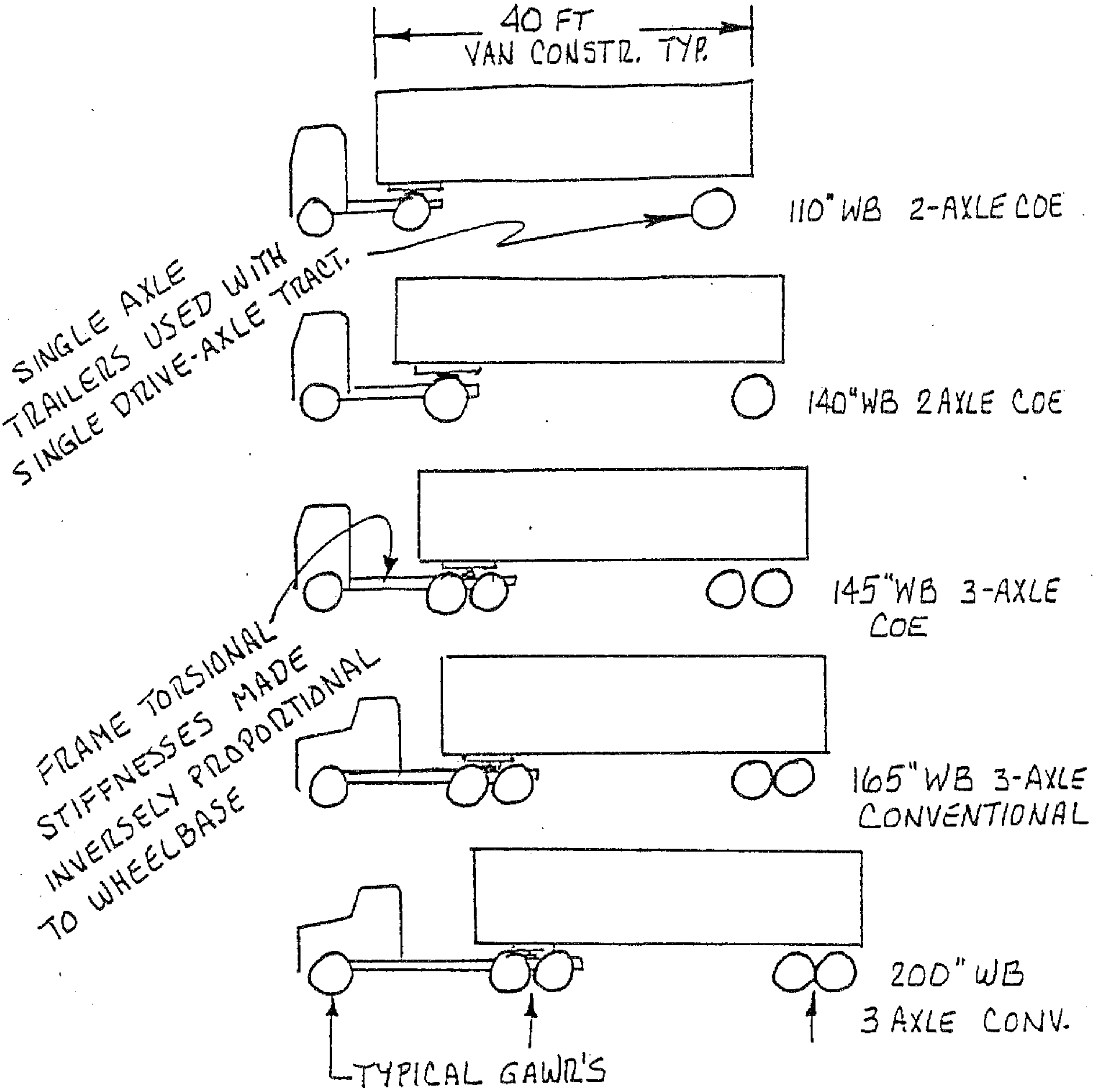


Figure 14. Vehicle combinations illustrating the parametric sensitivity of the nonlinear yaw behavior of "typical tractor-semitrailer" combinations.

- 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)
  - 2) a distribution more rear-biased than a standard configuration, as derives from the installation of an optional "heavy" 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.
- 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 inches (the baseline case)
  - 2) 78 inches

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

The full matrix of parameter variations examined for each of the five baseline tractor-semitrailers is shown in Figure 15. Response data representing each of the 48 indicated configurations have been condensed into "summary handling diagrams" illustrating the total envelope of results obtained for each of the five basic vehicles. Figure 16 summarizes the parametric sensitivities examined for the 110-inch wheelbase, two-axle tractor combination. The summary diagram shows 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}, appears 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 constitutes the most stable case. An almost identical "summary diagram" was obtained in the case of the two-axle tractor having 140-inch wheelbase. Indeed, wheelbase has been generally found to possess virtually no significance as a determinant of yaw stability.

Shown in Figure 17 is the summary handling diagram with curves defining the most and least stable configurations obtained with the two types of tire installations on the three-axle tractor with 145-inch wheelbase. Clearly, the three-axle tractor produces a baseline behavior which is significantly more stable than that observed with the two-axle

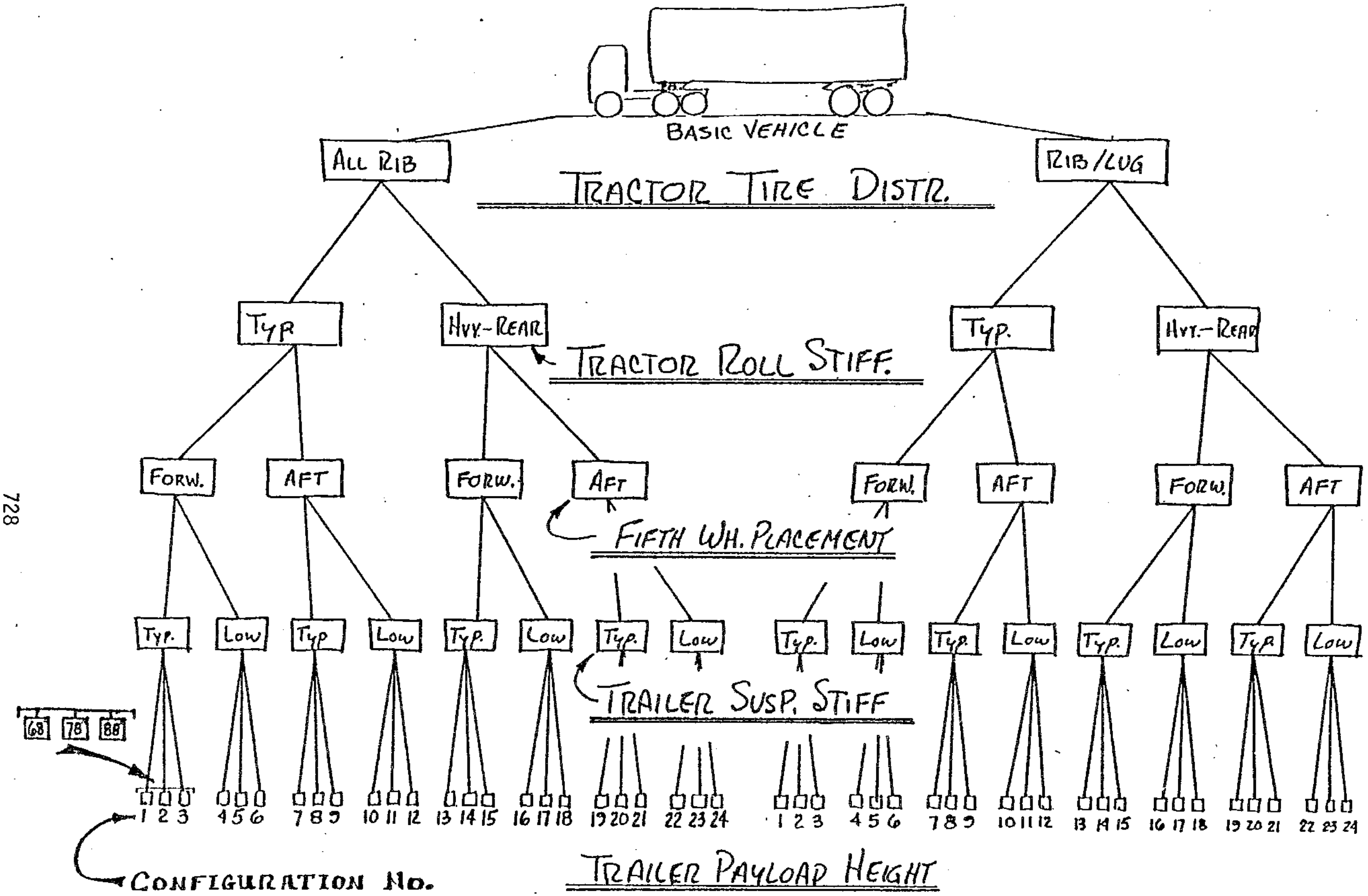


Figure 15. Parameter variations examined for each of the five baseline tractor-semitrailers.

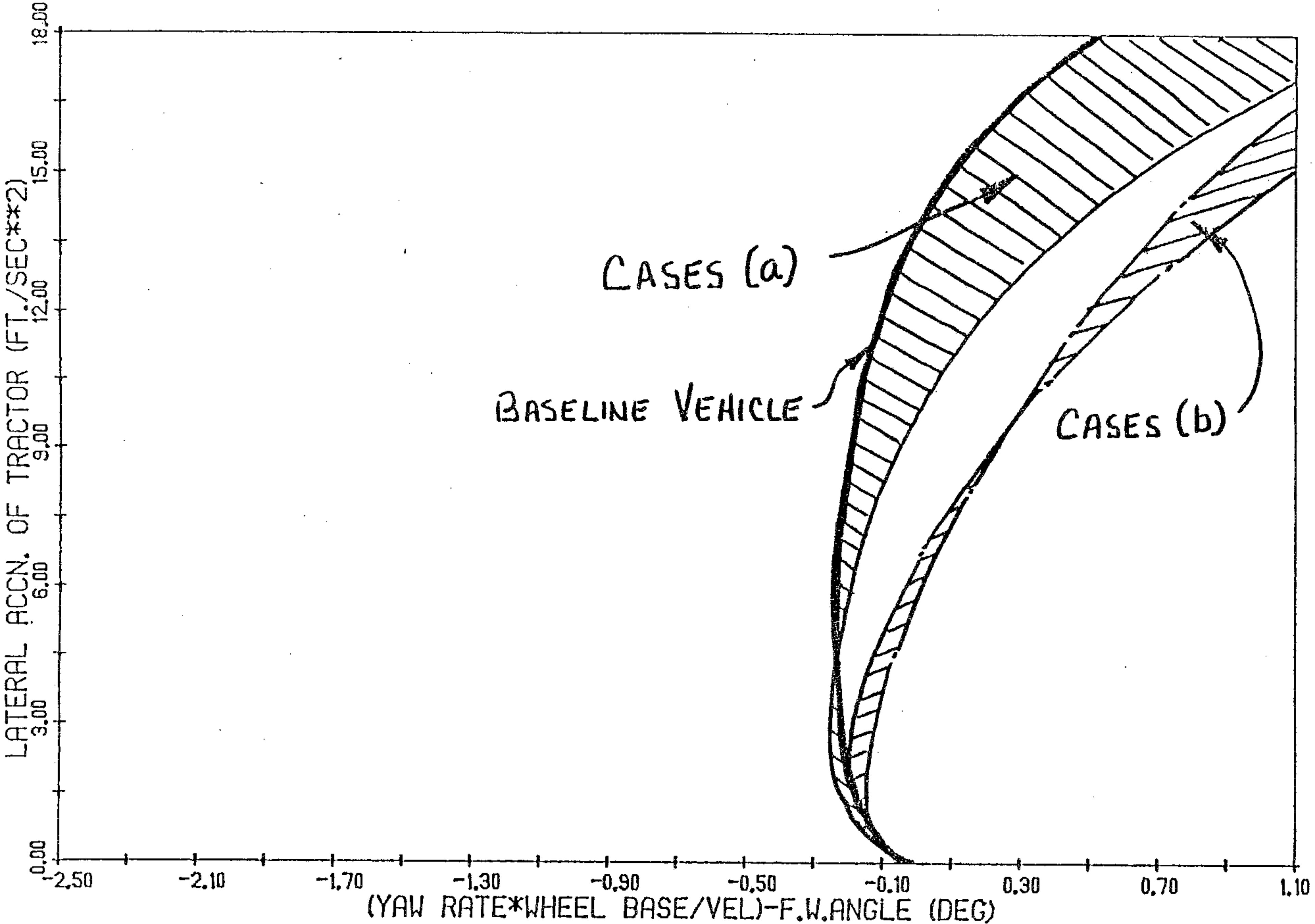


Figure 16. Handling curve envelopes for 110"-wheelbase two-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

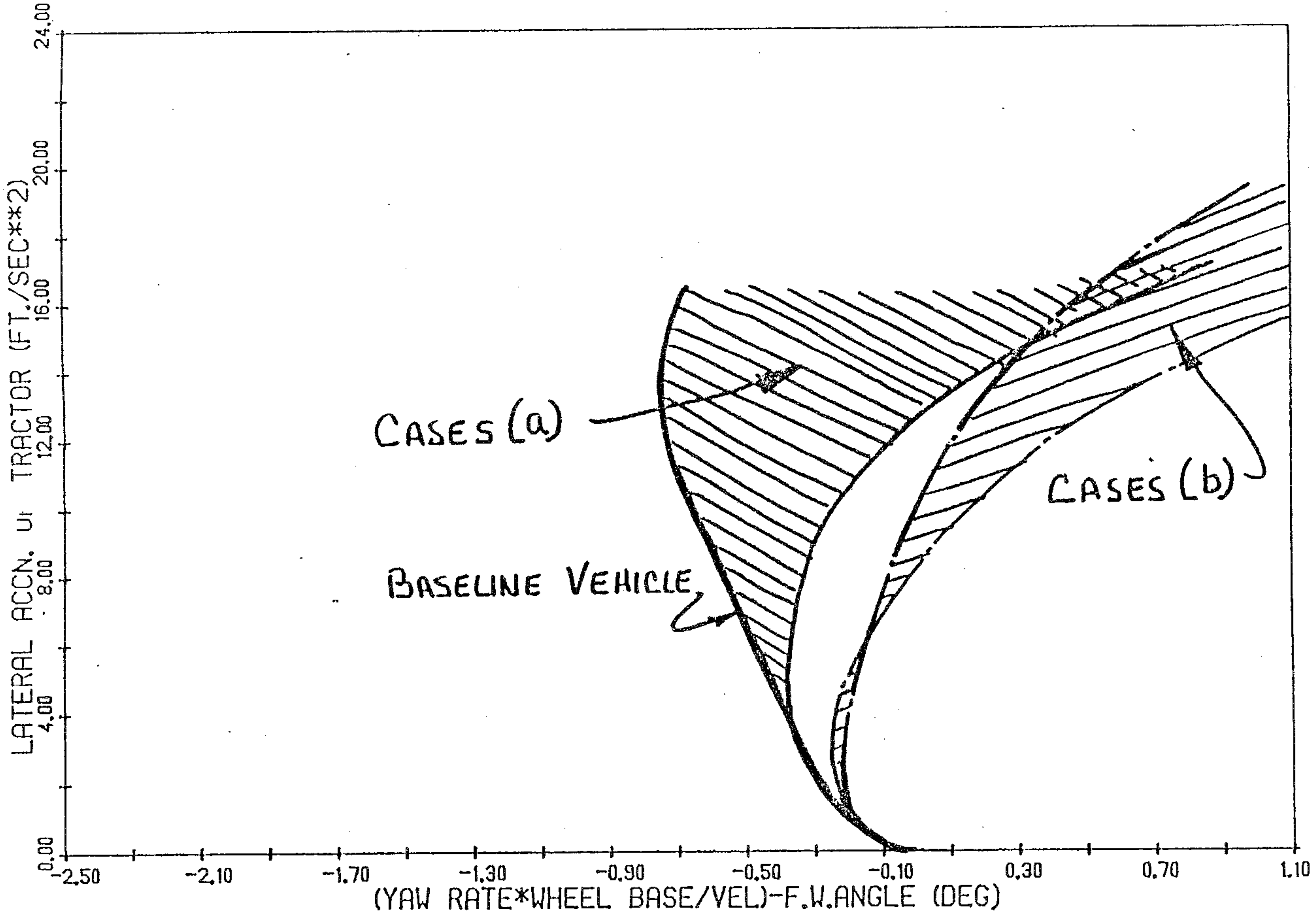


Figure 17. Handling curve envelopes for 145"-wheelbase, three-axle tractor for cases (a) all rib tires on tractor and (b) mixed rib and lug tires on tractor.

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

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 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 15.

Examination of Table 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) 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 15, was laid out such that configuration #1 would, presumably, constitute the most stable case whereas configuration #24 would constitute 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 2. Values of A_{y_c} Obtained through Parametric Variations (See Configuration No.'s in Figure 15).

Configuration Number	110"-W.B., 2-Axle Rib	110"-W.B., 2-Axle Lug	140"-W.B., 2-Axle Rib	140"-W.B., 2-Axle Lug	145"-W.B., 3-Axle Rib	145"-W.B., 3-Axle Lug	165"-W.B., 3-Axle Rib	165"-W.B., 3-Axle Lug	200"-W.B., 3-Axle Rib	200"-W.B., 3-Axle 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

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

b) Roll-stiffness distribution on tractor

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

c) Fifth-wheel placement

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

d) Trailer roll stiffness

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

e) c.g. height of trailer payload

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

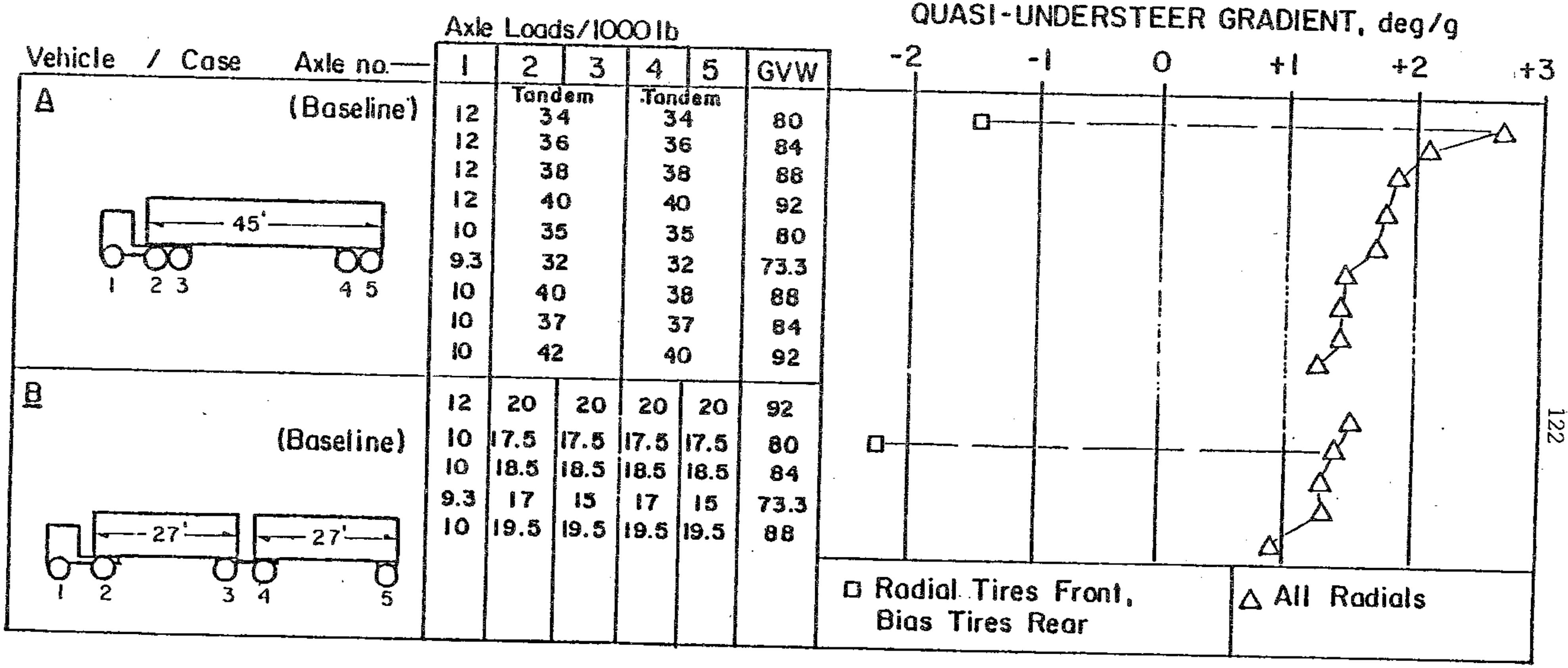
These variations in stability level cannot be conveniently normalized and must be viewed simply as the characteristic change in stability which can be expected to accrue, when vehicles (of the types described herein) are altered from their baseline condition by means of the (typical) parameter changes summarized in Figure 15.

Moreover, this particular parameter sensitivity study has revealed that the yaw stability of a tractor-semitrailer is markedly sensitive to variations in operating conditions and design variables which are known to be relatively commonplace.

It is also of interest to examine the influence of size and weight variables on yaw stability behavior. A study involving both simulation and full-scale experiments has produced results showing how vehicle load, center of gravity location, and wheelbase influence yaw stability. Numerical results of this research are expressed in terms of the understeer level observed at a lateral acceleration of 0.25 g's, in response to a ramp-steer input. For reasons that will be shown later in the lecture, the absolute level of the (quasi-) understeer gradient measured on a tractor at any A_y level in this test is greater than would be found in a steady-turn maneuver conducted at the same lateral acceleration level. Nevertheless, the results in this format provide a relative indicator of the yaw response degradations deriving from size and weight changes.

Shown in Figure 17b are the so-called "quasi-understeer gradient" measures for a tractor-semitrailer and a doubles combination under differing conditions of gross vehicle weight and axle load distribution. Also, the figure shows values for a baseline case of each vehicle configuration in which the tractor drive axle(s) are equipped with bias-ply tires while the steering axle is equipped with radials. The mixed tire condition, shown previously to be a powerfully degrading influence, is included here to provide a "benchmark" disturbance in the vehicle's yaw behavior so as to enable a rough scaling of the significance of the loading changes. The results show that increased gross weight and a more rear-biased distribution of load on the tractor cause the quasi-understeer level to decline. For a fairly wide range of load variations, however, the influence of loading is substantially less than the influence of the mixed tire condition.

Shown in Figure 17c is an illustration of the influence of the height of the payload center of mass on the understeer behavior of the same two vehicles, for two levels of gross weight. (Note that the distribution of load among the axle sets is shown in parentheses adjacent to each of the



Vehicle / Case	Axle Loads/1000 lb 1	2	3	4	5	GVW
A (Baseline)	12	Tandem 34		Tandem 34		80
	12	36		36		84
	12	38		38		88
	12	40		40		92
	10	35		35		80
	9.3	32		32		73.3
	10	40		38		88
	10	37		37		84
	10	42		40		92
B	12	20	20	20	20	92
(Baseline)	10	17.5	17.5	17.5	17.5	80
	10	18.5	18.5	18.5	18.5	84
	9.3	17	15	17	15	73.3
	10	19.5	19.5	19.5	19.5	88

Figure 17b. Influence of Gross Weight Variations on Understeer Level

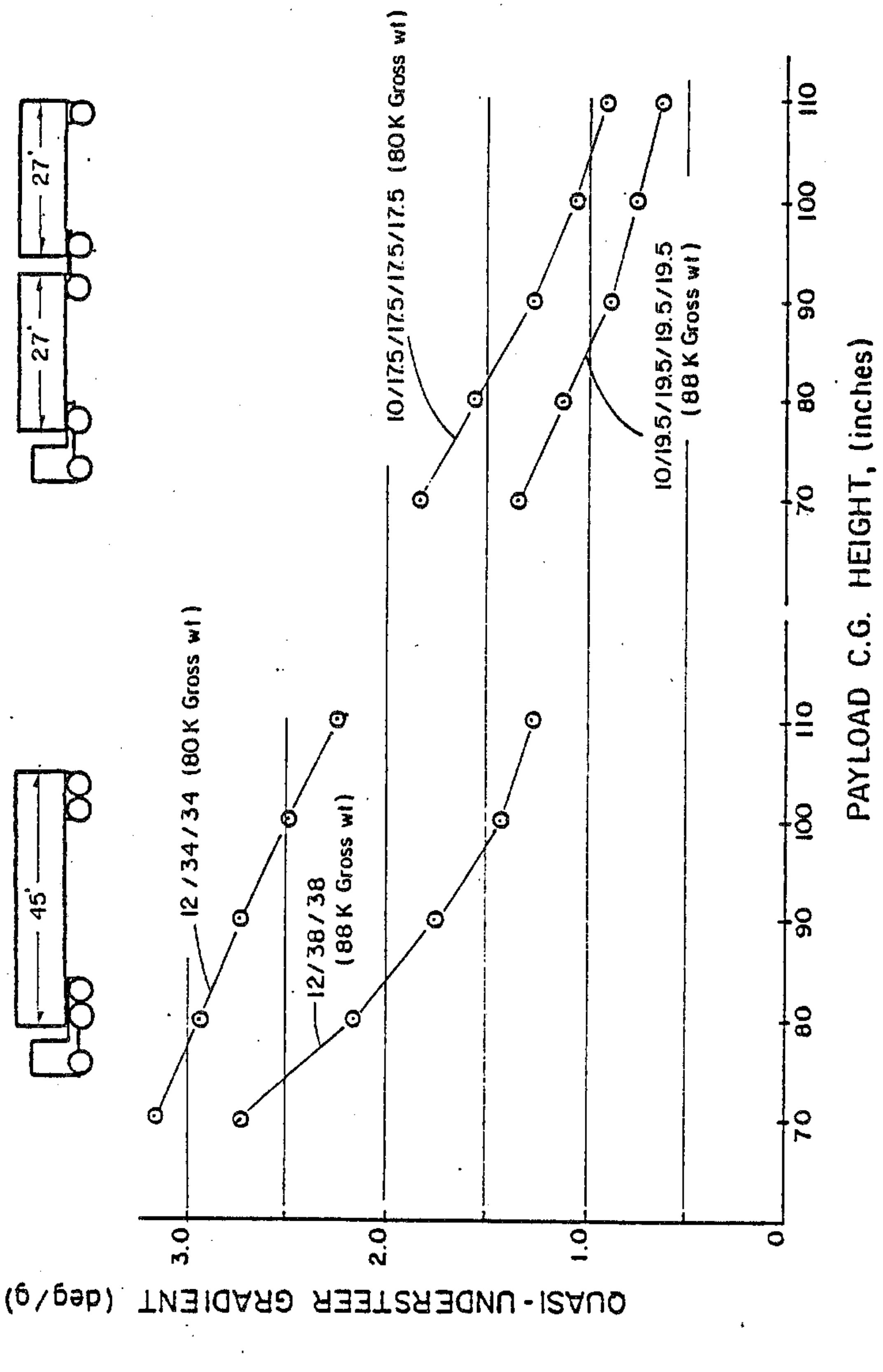


Figure 17c Influence of Payload C.G. Height on Understeer

gross weight designations.) The results show that increasing height of the payload mass center causes the understeer level to reduce. Clearly, this result is to be expected, given the proposition that the magnitude of the load changes on the left- and right-side tires during turning is directly related to the height of the vehicle's overall mass center.

Similarly, Figure 17d shows the influence of lateral offset in the placement of the payload mass center on the quasi-understeer gradient. We see that relatively small changes in lateral offset result in major reductions in understeer level (considering that the vehicle is turning such that the offset is toward the outside of the curve). Clearly, the yaw behavior of such a vehicle would be dramatically asymmetric in right- versus left-hand turns. In such cases, clearly the disturbance in the understeer property derives from the static transfer of vertical load between left- and right-side tires.

Figure 17e illustrates the general axiom that the wheelbase dimension is not instrumental in determining the understeer characteristic. As was shown in an earlier lecture considering the linear yaw behavior of a unit vehicle, however, the wheelbase dimension is a factor in determining the so-called "critical speed" of an oversteer vehicle, given the level of oversteer which prevails. Indeed, it was shown that increased wheelbase renders a first-order increase in the critical speed.

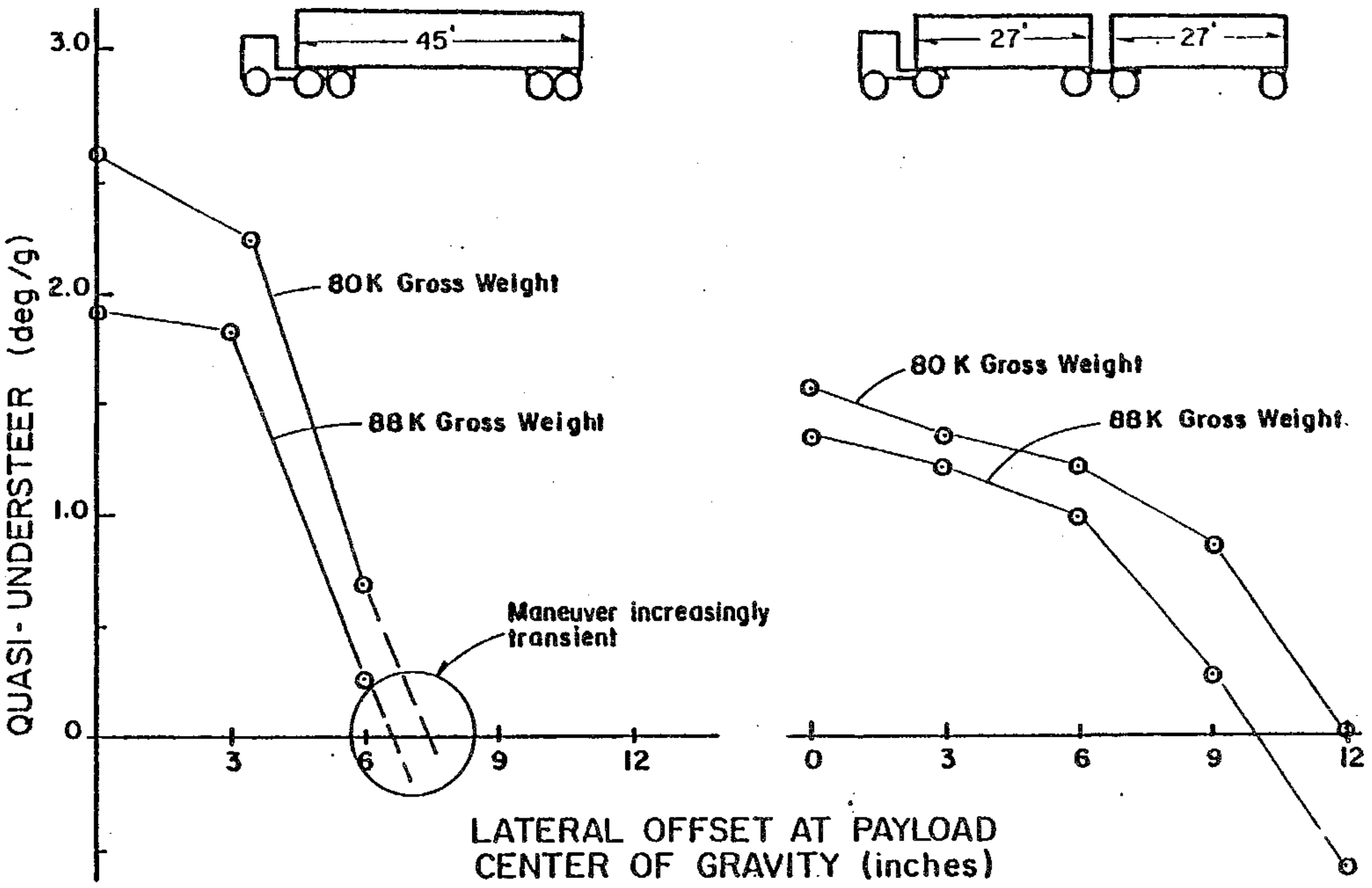


Figure 17d Influence of Payload Offset on Understeer Level

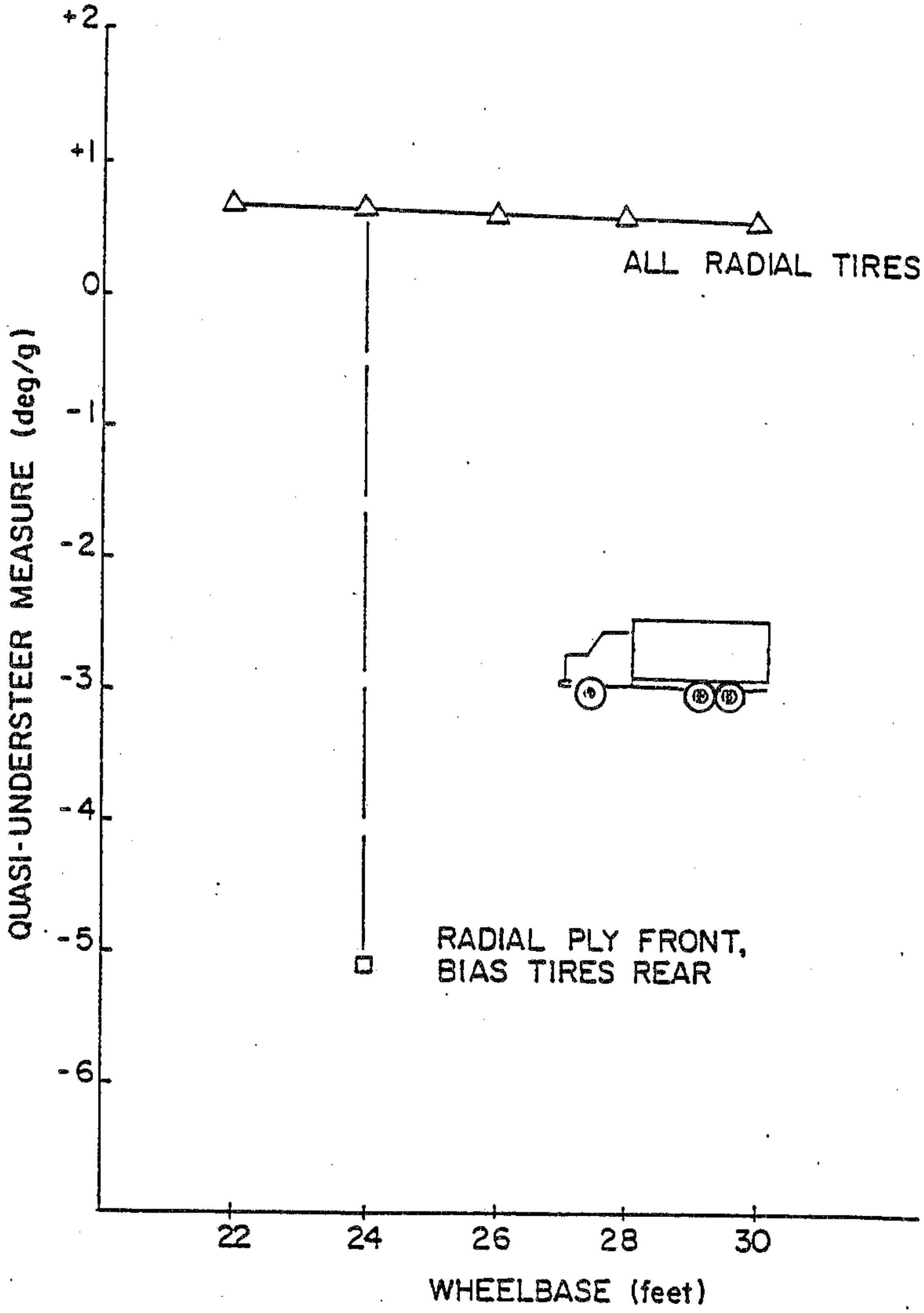


Figure 17e Influence of Wheelbase on Understeer for 3-Axle Truck, Fully Loaded.

Transient Steering Maneuvers in the Nonlinear Regime

As presented in Chapter 17, linear theory describing yaw response to steering establishes a specific set of relationships between steady-state and transient maneuvering characteristics. Thus it is to be expected that at least qualitatively similar relationships will be observed between the nonlinear tendency toward an oversteer steady-state yaw response of heavy trucks and the transient response of such vehicles to abrupt steering inputs.

For the oversteer vehicle operating below its critical velocity, for example, linear theory shows that relatively large values of response time-type measures will be found. Shown in Figure 18 are experimental data illustrating, for the nonlinear heavy truck, increasing levels of a yaw rate response time numeric* for successive step-steer test maneuvers conducted at increasing levels of lateral acceleration. The corresponding handling diagram constructed using raw data from the quasi-steady-state portion of the same step-steer-like experiments is shown in Figure 19. We see that the transition from understeer to oversteer in the handling diagram of the quasi-steady-state behavior corresponds to the increasingly elongated response times in the transient response to a step.

Interestingly, the data shown above for the case of a tractor-semitrailer responding to a step input of steering actually illustrate response times which are reduced below the level which would derive with a hypothetical <u>unit vehicle</u> having the same steady-state behavior shown in the handling diagram. The reason for this complicating feature of the yaw response of articulated vehicles is that the semitrailer typically introduces both a lag function and an attenuation in response to abrupt tractor motions. Accordingly, as we will see, the transient portion of the step-steer response of a tractor-semitrailer will typically reveal more understeer-appearing characteristics than will be found in the

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*The yaw rate response time measure is defined as the time for the yaw rate signal to attain 90 percent of its steady-state value, following the input of 50 percent of the steady level of steering input.

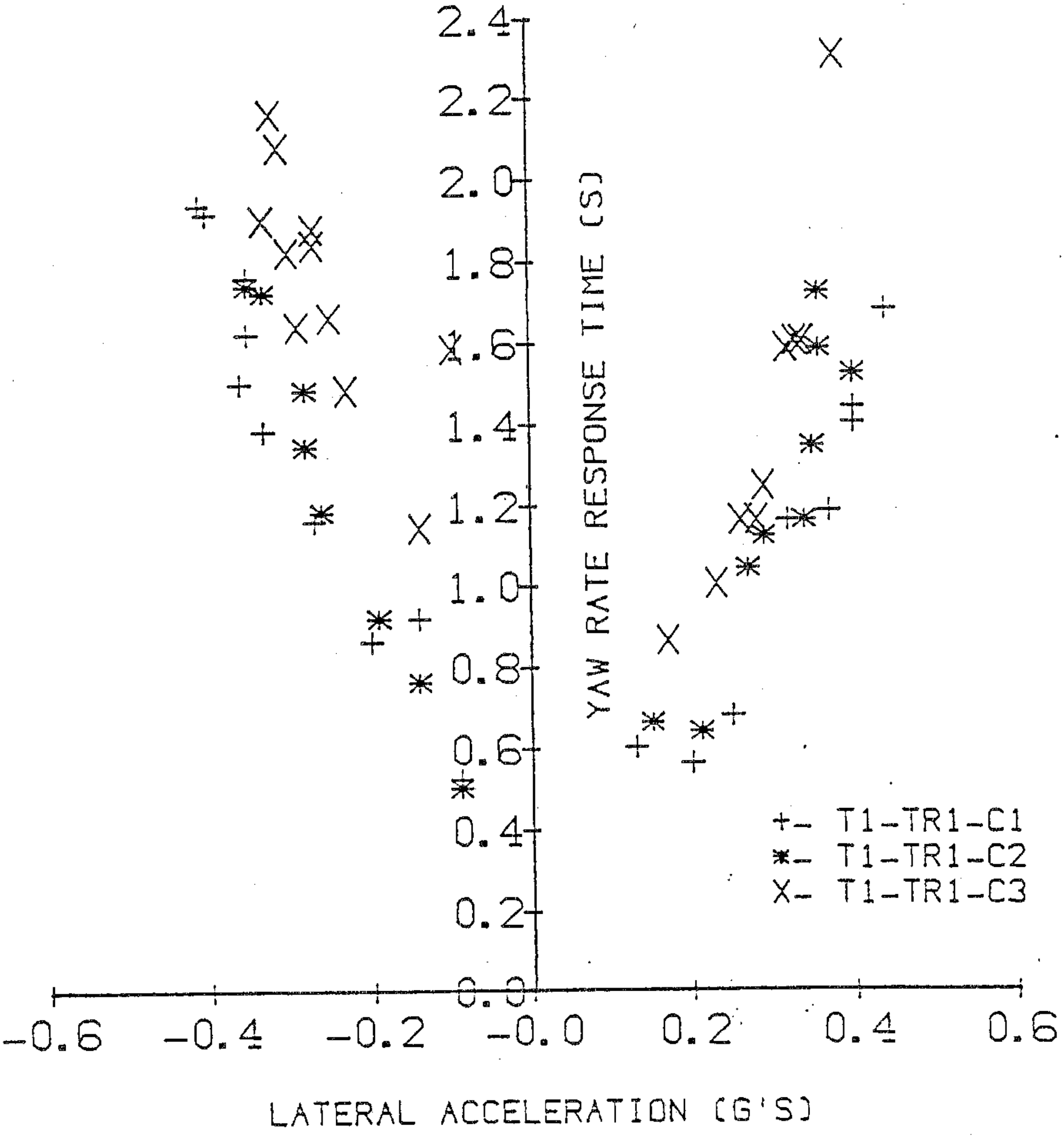
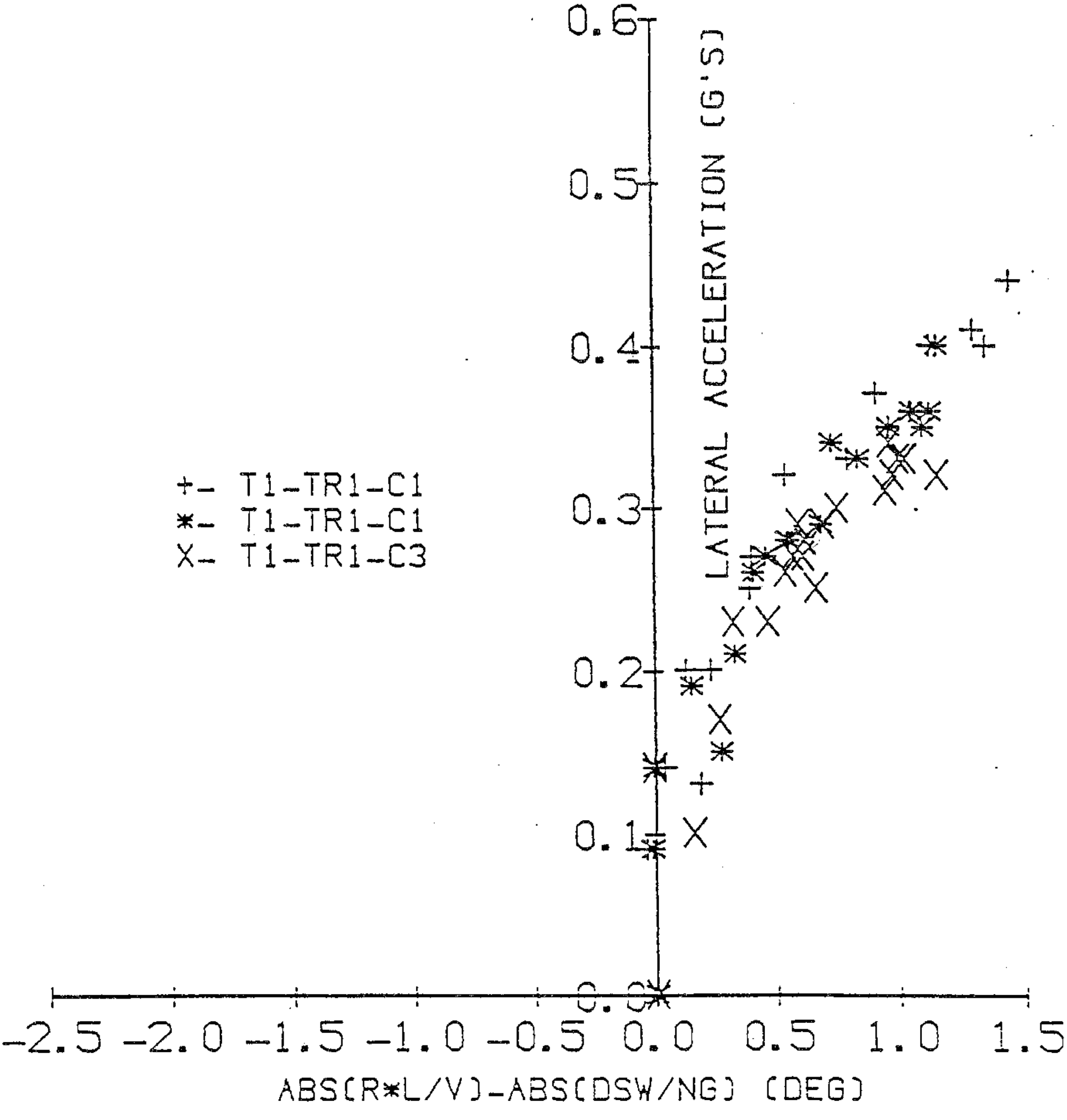


Figure 18. Yaw rate response times for 2S-1 combination in conditions as follows: C1 - load distribution 11K, 20K, 20K, same bias tire at all wheels; C2 - load distribution 11K, 22K, 22K, same bias tire all wheels; C3 - same as C2 but with radial tires on steering axle.



TWO AXLE TRACTOR- 27 FT VAN TRAILER

Figure 19

subsequent steady-state response. This anomaly can be easily understood upon considering, in Figure 20, the lateral acceleration responses of a tractor and semitrailer in a ramp-steer maneuver. We see that the lateral acceleration response of the trailer lags that of the tractor such that, at any moment in time, a substantially lower level of lateral load transfer is being borne at the tractor than would have been the case, at that value of tractor lateral acceleration, if the semitrailer's A_y response had been "in phase" with and of equal magnitude to that of the tractor. Since the trailer payload constitutes the major mass, and thus roll moment, to be reckoned with in accounting for the lateral load transfer occurring at the tractor, the lag in semitrailer lateral acceleration response is indeed significant as a mechanism serving to upgrade the effective yaw stability of tractor-semitrailers in any other than the slowest, or most steady-state-like maneuvers. Shown in Figure 21, for example, we see the results of differing ramp-steer maneuvers which are presented on the handling diagram as if they each pertained to legitimately steady-state cases. The illustrated handling curves indicate increasingly oversteer behavior as the ramp input condition approaches the steady-state case. Other transient maneuvers, such as a rapid lane change, for example, will likewise involve a significantly lagging and attentuated trailer response such that a higher level of apparent yaw stability prevails.

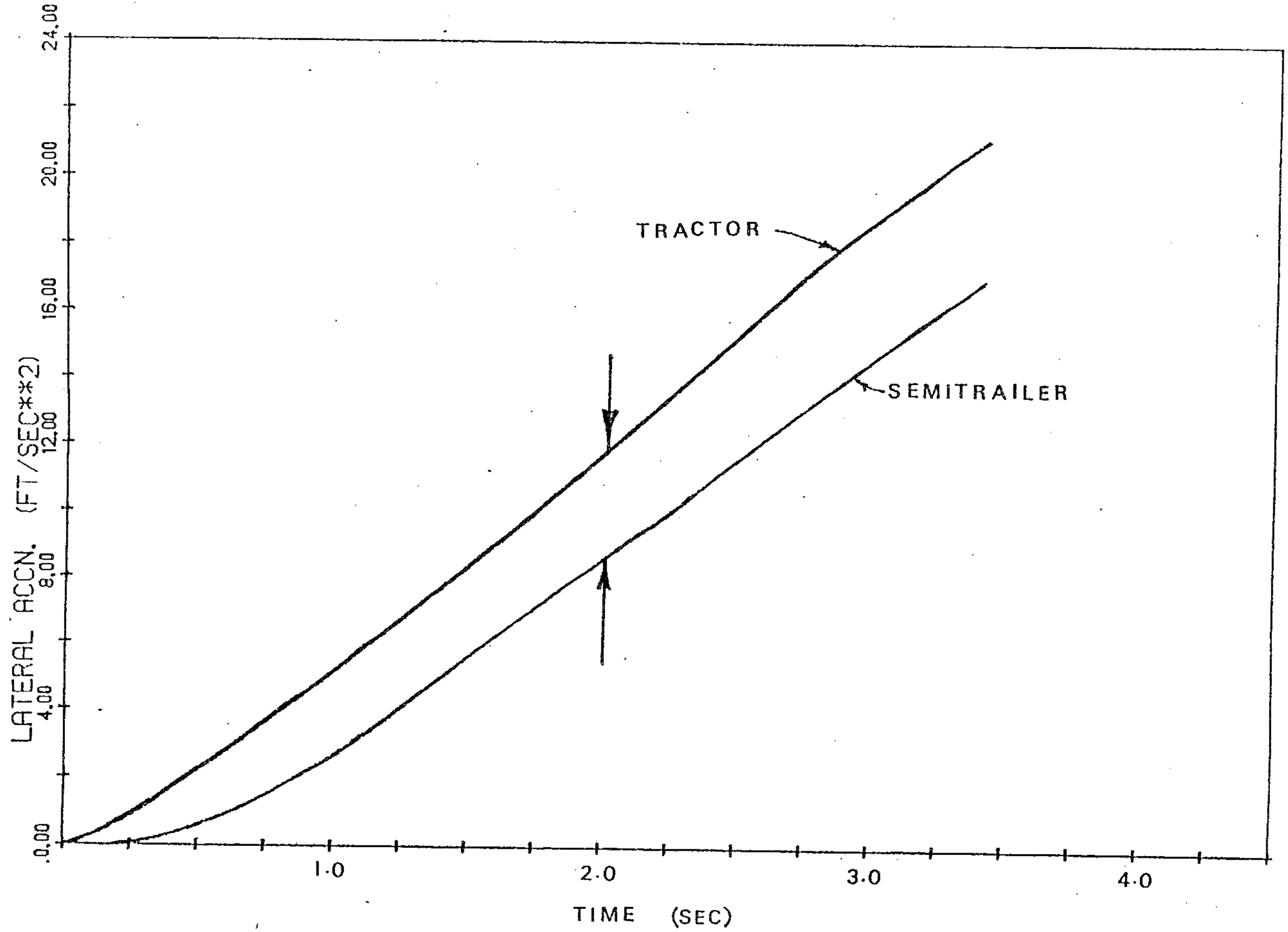


Figure 20. Lag in lateral acceleration of semitrailer with respect to tractor in ramp-steer maneuver.

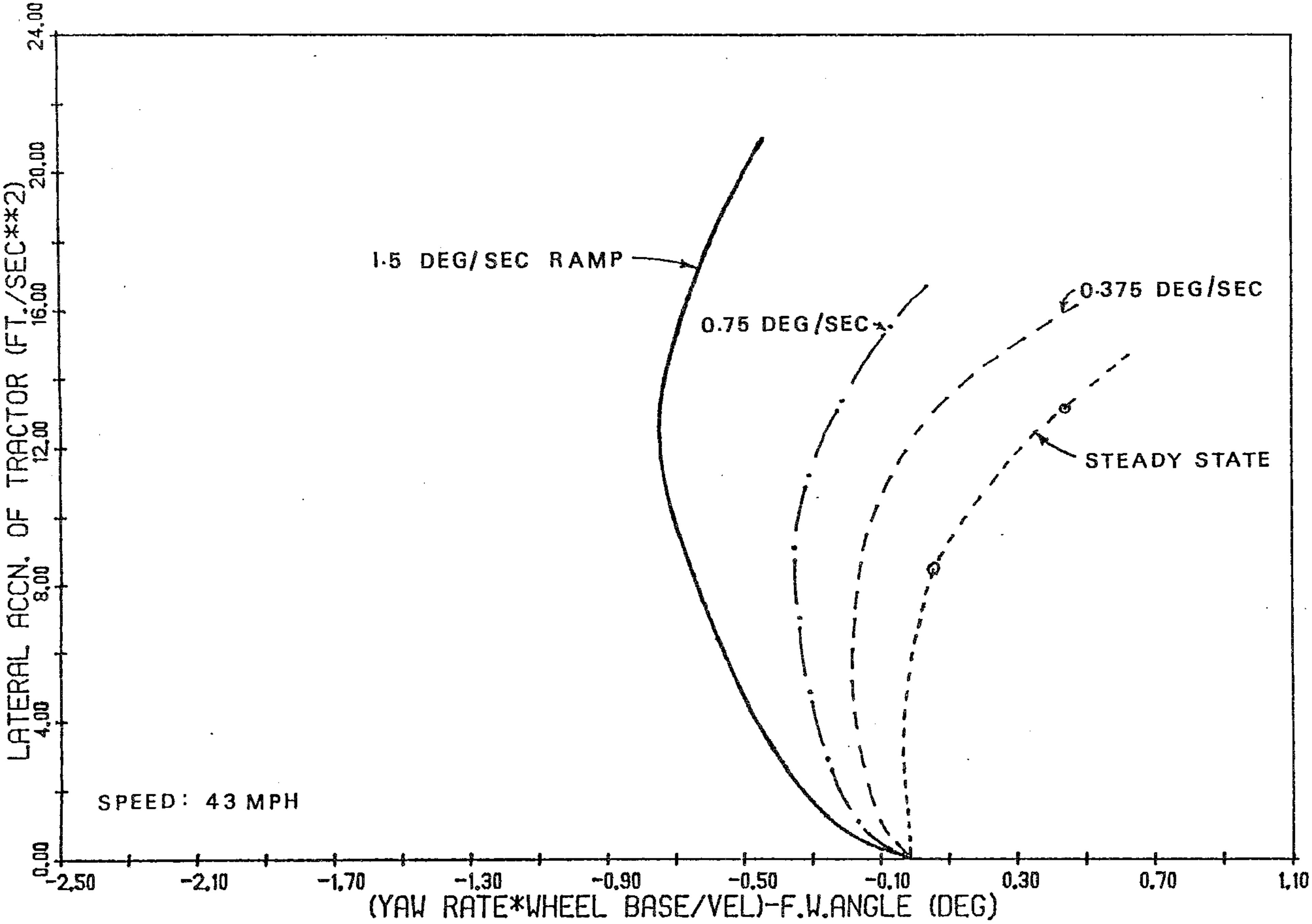


Figure 21. Increasing yaw stability with increasing stecr input rate.

BIBLIOGRAPHY

1. Ervin, R.D., et al. "Effects of Tire Properties on Truck and Bus Handling." Final Report, Contract DOT-HS-4-00943, Highway Safety Research Institute, University of Michigan, 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. Bundorf, R.T. and Leffert, R.L. "The Cornering Compliance Concept for Description of Vehicle Directional Control Properties." SAE Paper No. 760713, October 1976.

4. Segel, L. and Ervin, R.D. "The Influence of Tire Factors on the Stability of Trucks and Tractor-Trailers." Vehicle System Dynamics, Vol. 10, 1981, pp. 39-59.

5. Ervin, R.D., et al. "The Yaw Stability of Tractor-Semitrailers During Cornering." Final Report, Contract DOT-HS-7-01602, Highway Safety Research Institute, University of Michigan, Report No. UM-HSRI-79-21, June 1979.

6. Ervin, R.D., et al. "Influence of Size and Weight Variables on the Stability and Control Properties of Heavy Trucks." Final Report, Contract FH-11-9577, Transportation Research Institute, The University of Michigan, Report No. UMTRI-83-10, April 1983.

SESSION D - DISCUSSION PERIOD

D. Lambert

Question on regulation changes seeming to be going the wrong way.

B. Ervin

I think its a regulatory type of question. I don't think the 1 m difference in length would be very significant to any of the vehicles we are talking about here. So if you gave another metre to the A-Train and gave or took a metre from the B-Train it wouldn't make a significant difference in contrast between the B-Train or A-Train or the absolute performance of either one of them. I would assume that one of the regulatory postures that people might think of taking would be to take steps to look at what productivity means and try to provide some incentives to the industry to go towards better combinations in terms of stability and safety performance and productivity is either size or weight and if you add another metre to a vehicle that you want to promote because you think it has some advantages then that makes sense to me, at least it is rational, whether it is the right thing to do given all the trade offs I don't know. The comments you make are at least categorically true. If you have a vehicle that suffers in some kind of stability measure because it is short you would like to make it longer if you think it is a inherently desirable kind of thing. That's true. But if you could like to discourage its use altogether then you may be reluctant to add length because length happens to be productivity. Now one clever way to beat the thing might be to say alright if you guys are so concerned about safety why don't we keep the bed lengths the same and we require that you build on structures that stick the axles out further so as to lengthen the wheel bases without increasing the productivity but I don't think there would be many guys add on the extension. The regulation of trucking is a tough tough business and there are hard decisions. I'm a little nervous of a couple of things they are developing in which you get a couple of yokels like Peter and I who tell you a lot of things about what is good theoretically and what is bad theoretically and I don't think we say these things in vacuum because both of us have had input of a lot of actual experience that we wouldn't be telling you something here theoretically that apparently didn't have a very persuasive actual input to back them up. I certainly know that is the case in the things we are talking about. It is not just a theoretical issue but there are things to think about. There are versatility issues with the A- Train, there are some very serious in certain kind of usage. I want to comment on the dairy thing though because we recently did a slosh study for the Department of Transportation in U.S. and one of the vehicles that comes out is really having a problem with sloshing liquids is dairy pick up vehicles because they go from place to place and they pick it up on the way and they pass through these partial filled conditions during the day and one of the kinds of vehicles that we told the Federal DOT that they should be especially wary of is anybody who is in the slosh load business with a A-Train because it happens that the inherent slosh resinance frequencies are just about one half of a hertz, .5 cycles per second which is exactly the same frequency that rearward amplification tends to reasonate so if you have dairy tankers that are A- Trains and if they are partially loading the pop trailer, the full trailer, the slosh resinance is at the frequency as the rearward amplification resinance. Now maybe they never roll over their pop trailers but if they don't it must be just magic. It is something like being south of the equator again.

R. Law

The question I have concerns the stablity characteristics of the B-Train and in order to improve the low speed tracking if you position the 5th wheel on the lead semi aft of the tandem centre you improve the swept path. Where would you think would be a limit to the amount of aft setting of that 5th wheel?

P. Sweatman

I would have thought the limit would be any aft setting of the 5th wheel would be bad because you main problem there is under braking condition you have got much more tendancy to jack-knife so any aft position I think would be undesirable.

B. Ervin

If you are talking about a tractor, if you move it aft you mildly aggravate the yaw stability but if you were to generalise that comment and try to make it apply to the stinger steered car hauler that's not fair because that vehicle achieves a reasonable load distribution by another mechanism. I think the stinger steered thing can start to get into trouble when you have your payloads all goofed up. You have partially unloaded some things. Maybe an empty truck and loaded trailer, that's really bad news. Or if you unload the trailer completely and the back half of the truck so all the load is on the front axle that is screwy and that will aggravate the kind of thing Peter was talking about because you have an unloaded rear end and a heavily loaded front end. I don't think there is any magic to passing over the tandem centre with the 5th wheel. If you were to be a few inches forward or a few inches aft it is not that powerful a phenomenon.