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DYNAMIC BEHAVIOR OF THE B-TYPE CONVERTER DOLLY

Final Report

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16. Abstract

A B-type converter dolly is a device for coupling a semitrailer to a towing unit—normally a leading trailer in a vehicle combination. The B-dolly differs from the more conventional, or A-type, converter dolly in that it is connected to the lead unit by two pintle hooks and it employs a steerable, rather than fixed, axle. This study has examined the influence of B-dolly design characteristics on the stability and control behavior of double and triple trailer combinations. The investigation involved laboratory measurements and computer simulation of vehicle response. Stability and control behavior was evaluated in terms of low- and high-speed offtracking, rearward amplification in obstacle-avoidance maneuvers, and overshoot responses during transient steering. The strength requirements for pintle hitches used in B-dolly applications were also examined.

The B-dolly was found to be a very attractive concept for improving upon the stability and control properties obtained with conventionally coupled doubles and triples. Stability and control properties were found to be strongly sensitive to certain characteristics of the B-dolly's steering system, however, such that proper design of the device is seen as mandatory, from a safety point of view.

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CHAPTER 1

INTRODUCTION

This document reports on a study of the dynamic performance of self-steering B-type converter dollies, undertaken by The University of Michigan Transportation Research Institute (UMTRI) under contract to the National Research Council of Canada. In the context of this research, the terms "A" and "B" dolly distinguish two alternative converter dolly devices which can be used to hitch a semitrailer to a towing unit, thus making up a towed full trailer. The A-type dolly is the conventional device which connects to the towing unit at a single pintle hitch and which articulates about that hitch point in order to track a curved path. The B-type dolly couples to the towing unit by means of two pintle hitches which are spread laterally from one another. This form of coupling effects a yaw-rigid constraint which forces the dolly to remain directly in line with the towing unit. Tire scrub during cornering with the B-dolly is minimized by means of a self-steering dolly axle. The research effort involved parameter measurements on a commercially available B-dolly and simulation of various dolly and self-steering system configurations to determine the implications of the self-steering B-dolly on the dynamic performance of double and triple trailer combinations.

The parameter measurements were obtained for one sample of an Auto Steering Trailers Limited dolly, model SSD. The properties of the dolly's suspension and self-steering system were measured on the UMTRI suspension parameter measurement facility to provide basic information on the functioning of the dolly's self-steering system for use in computer simulation. The self-steering system of this particular unit was found to have a large amount of coulomb friction and a bilinear spring rate centering function. This form of the dolly steering system was used to guide parameter selections for subsequent simulations dealing with parametric variations of the self-steering system.

The installation of a B-type converter dolly with a self-steering system on a vehicle combination in place of a conventional (A-type) dolly introduces several basic changes in the vehicle system. The yaw constraint of the B-dolly frame in its attachment to a leading truck or trailer eliminates one of the articulation points present in a standard A-train. Also, the B-dolly couples the vehicle units by means of a rigid roll restraint providing improved roll stability for the subsequent, or "pup" trailer, such that it can gain a supportive roll moment through its connection to the preceding trailer. Both of these changes render a vehicle type which is analogous to the so-called "B-train," a vehicle configuration known to exhibit reduced rearward amplification relative to the standard A-coupled vehicle. It has been recognized, however, that the self-steering feature of the B-dolly axle introduces a potential for degrading certain of the performance qualities otherwise obtained with a B-train. The simulation study was performed to quantify the influence of this self-steering action on the vehicle's dynamic directional performance.

1.1 Organization of the Report

Chapter 2 of this report presents the results of the parameter measurement activity. Descriptions of the tested vehicle unit, the facility, and the test procedures are presented, along with the measurement results.

The simulation tools employed in the study are presented and described in Chapter 3. This chapter also includes the results of simulation validation exercises.

In Chapter 4 the results of the simulation are presented. The performance of combination vehicles employing self-steered B-dollies are compared to that of conventional A-trains and B-trains. The simulation results address four issues related to the self-steering B-dolly, namely,

1) Parametric variations in the self-steering system

2) Dynamic loading of the hitching mechanism

3) Influence of dolly length on vehicle performance

4) Application of B-dollies to triple-trailer combinations

Conclusions and recommendations from the study are presented in Chapter 5.

CHAPTER 2

SUSPENSION PARAMETER MEASUREMENT

The UMTRI suspension parameter measurement facility was used to characterize the response of the B-type converter dolly's suspension and steering system to vertical and shear forces applied at the tires. This facility consists of a movable table driven by hydraulic cylinders that exercise the suspension in bounce and roll. The vehicle tires are placed on instrumented pads that are able to transduce vertical and lateral forces, and aligning moment. Actuators within the pad assemblies apply shear forces and moments to the tire contact patch. The vehicle frame is secured to an overhead structure and serves as a rigid mechanical "ground" for testing. The general layout of the facility is illustrated in Figure 2.1.

The vertical motion of the axle is measured by string potentiometers attached to the wheel centers. Camber (roll) and steer motions of the wheels are measured with specially designed transducers that ride on an effective wheel plane surface. These transducers are also shown in the figure.

The vertical and roll motions of the table can be controlled to achieve one of three control schemes, namely, force control, axle control, and direct table position control. In the "force mode," the table is commanded to apply a specific total vertical load and side-to-side load differential. This mode is used in those tests requiring vertical or roll motion of the axle. The "axle mode" is used when performing tests that involve the response of the system to shear forces and moments. In this way, the influence of axle motions can be separated from compliance-related response.

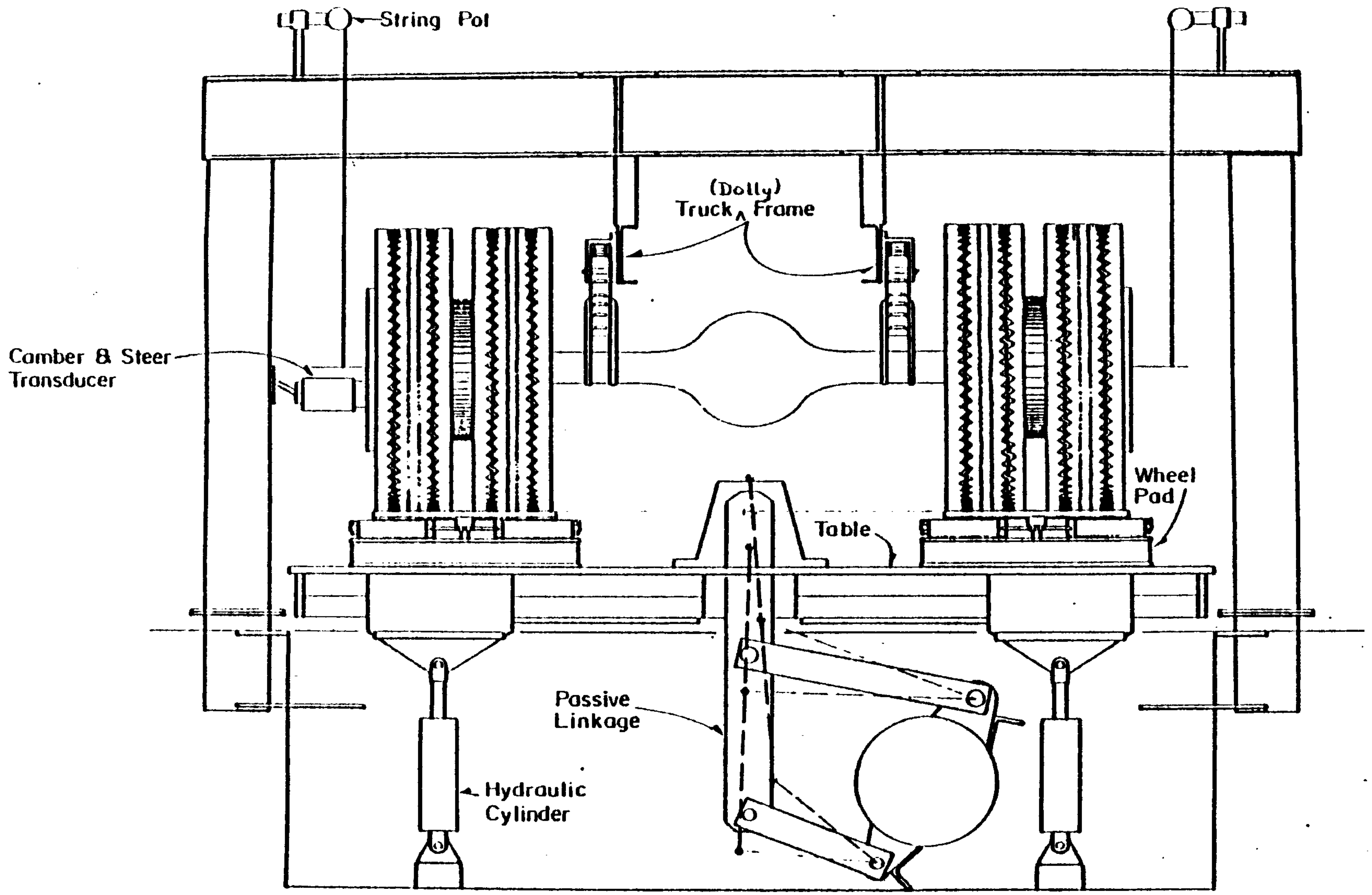


Figure 2.1. UMTRI Tandem Axle Suspension Parameter Measurement Facility.

A standard set of parameter measurement tests were performed on the ASTL B-type converter dolly. These tests are:

Vertical rate of suspension
Roll rate of suspension
Bounce steer kinematics
Roll steer kinematics
Roll axis location
Aligning moment compliance steer
Lateral force compliance steer
Brake force compliance steer

These measurements were made on the B-dolly with the steering locked, and with pressures of 483 kPa (70 psi) and 241 kPa (35 psi) applied to the ram which controls the compliance of the steering system. The two cited pressure values are the manufacturer's recommended settings for loaded and empty conditions, respectively. Parameter measurements were also performed on trailers which were used by the Ontario Ministry of Transportation and Communication in companion full-scale testing. These parameters were used in the simulation for validation purposes. Detailed parametric measurements are presented in Appendix A.

2.1 Suspension and Self-Steering System Description

The general layout of the B-dolly suspension and steering system is shown in Figure 2.2. The suspension is a typical single-axle layout with leaf springs riding in slippers on both ends and axle longitudinal location provided by torque rods. The steering system is composed of tie rods coupling the steering knuckles together at a bell crank mechanism. The bell crank carries a roller that is loaded against a cam by means of an air brake chamber, providing the system's centering function. The cam is profiled as shown in Figure 2.3. This profile is designed to provide a high stiffness at the center of its travel, while the roller is in the detent, with a softening spring rate at larger steer angles. The resistance of the cam to lateral motion of the roller is controlled by the angle that the cam face makes with the roller and the amount of force applied to the cam. Thus, the steering compliance

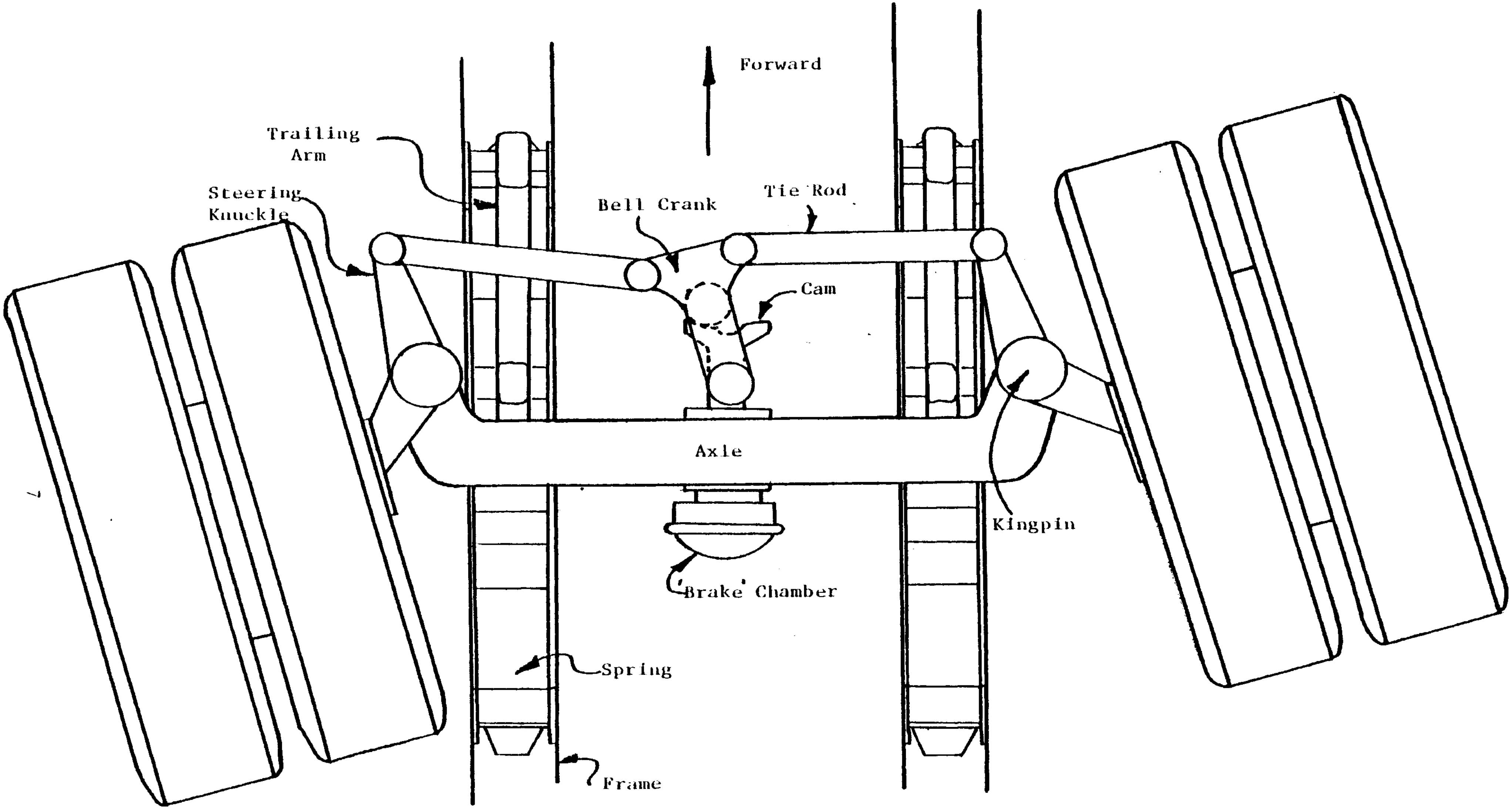


Figure 2.2. Suspension and Steering System Viewed from Below.

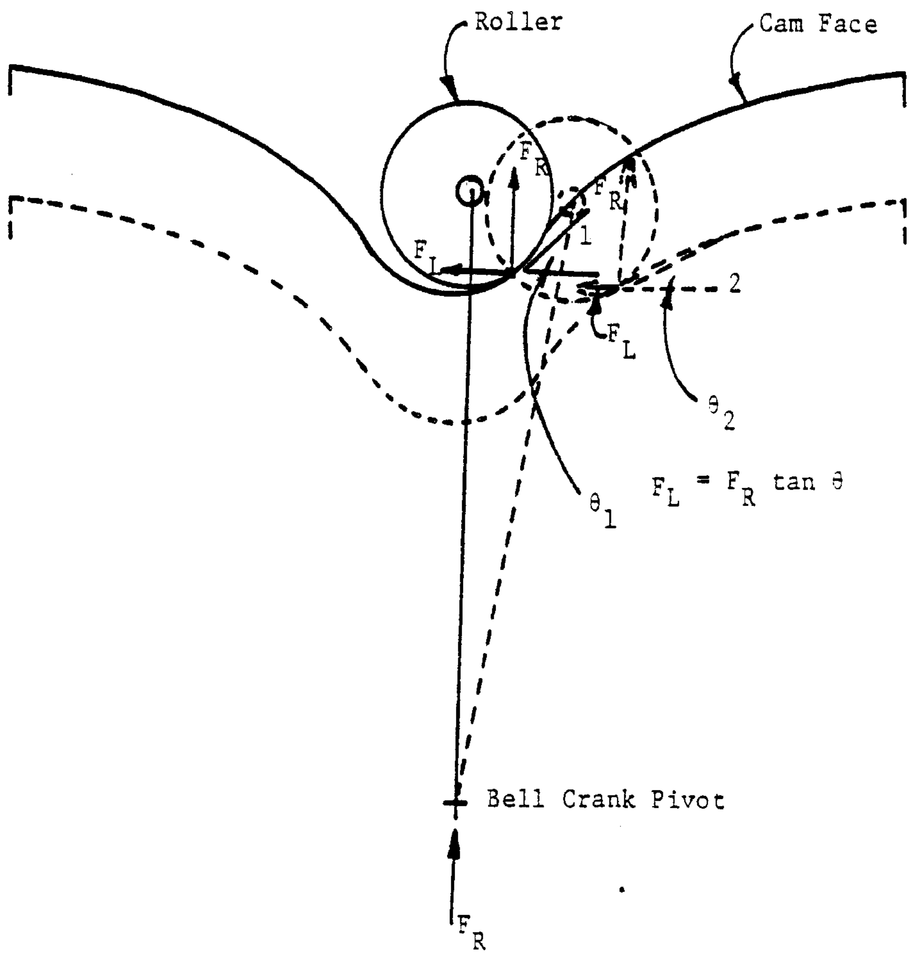


Figure 2.3. Development of Lateral Force at Cam Face.

of the B-dolly system can be increased by either increasing the applied pressure, or increasing the angle between the roller's line of motion and the cam face, as in the detent area.

The geometry of the steering layout provides a large mechanical trail at which the tire side forces act to apply a moment to the steering system. The geometry also has a very large kingpin offset that results in a substantial moment being passed to the self-steering mechanism under braking. The steering linkage is attached directly to the axle and steering knuckles such that roll steer due to the geometry of the steering linkage is virtually eliminated.

2.2 Measured Suspension and Steering System Properties

The basic properties of the B-dolly suspension with steering system locked are summarized in Table 2.1. The measured properties are typical of values seen on other single-axle, leaf-spring suspensions, with the exception of the roll steer parameter, which, due to the location of the steering linkage, is negligible at all axle loads.

Table 2.1. ASTL B-Dolly Suspension and Steering System Parameters—Steering Locked

	Axle Load (KN)		
	88.9	66.7	44.4
Vertical Rate (N/mm)	1400	1400	1400
Roll Rate (N-m/deg)	11900	9900	6800
Roll Steer (deg/deg)(L/R)	0/.04	0/.06	0/.08
Roll Center Height (mm below top of frame)	340	356	356
Aligning Moment Compliance Steer	nil		
Lateral Force Compliance Steer	nil		
Brake Force Compliance Steer (deg/KN)(L/R)	.016/ .018	.016/ .018	.025/ .020

The characteristic of most interest to this study is the compliance steer resulting from the application of lateral force to the tire. The lateral force compliance steer behavior for each of three values of axle load and for a ram pressure of 483 kPa (70 psi) are shown in Figure 2.4 for moderate levels of lateral force input. These data show a very small compliance up to a level of lateral force corresponding to the coulomb friction level. After breaking the friction, the steering system passes through the backlash in the linkage and again the system becomes quite stiff. The aligning moment compliance data in Figure 2.5 show a similar characteristic.

The lateral force level achieved in the tests illustrated in Figure 2.4 was limited by the available friction between the wheel pad and the table. That is, with an imposed lateral force at the contact patch, and with the test machine maintaining zero roll deflection across the suspension, one tire becomes unloaded in response to the resulting roll moment such that slippage of the wheel pad occurs. To allow the application of larger lateral forces without slippage of the pads, the test was rerun with the test machine holding tire loads constant, while permitting the axle to roll. In this control mode, the table "chases" the commanded vertical load by rolling the axle. This axle motion was found to be tolerable since the kinematic roll steer contribution to the total steer angle is negligible.

As seen in Figure 2.6, the larger lateral force achievable in this mode causes the steering system to be moved through the coulomb friction range, which measures approximately 21.1 kN (4,750 lb) at an axle load of 90 kN (20,000 lb), thus engaging the centering function of the steering system. The centering function presents a fairly high stiffness behavior, on-center, but as the roller moves out of the detent, the steer response becomes very compliant such that large increases in steer angle result from relatively small increments of lateral force. The stiffness observed for these conditions at a ram pressure of 483 kPa (70 psi) was 17.8 kN/deg (4,000 lb/deg) on-center and as low as 445 N/deg (100 lb/deg) at two degrees of steer angle.

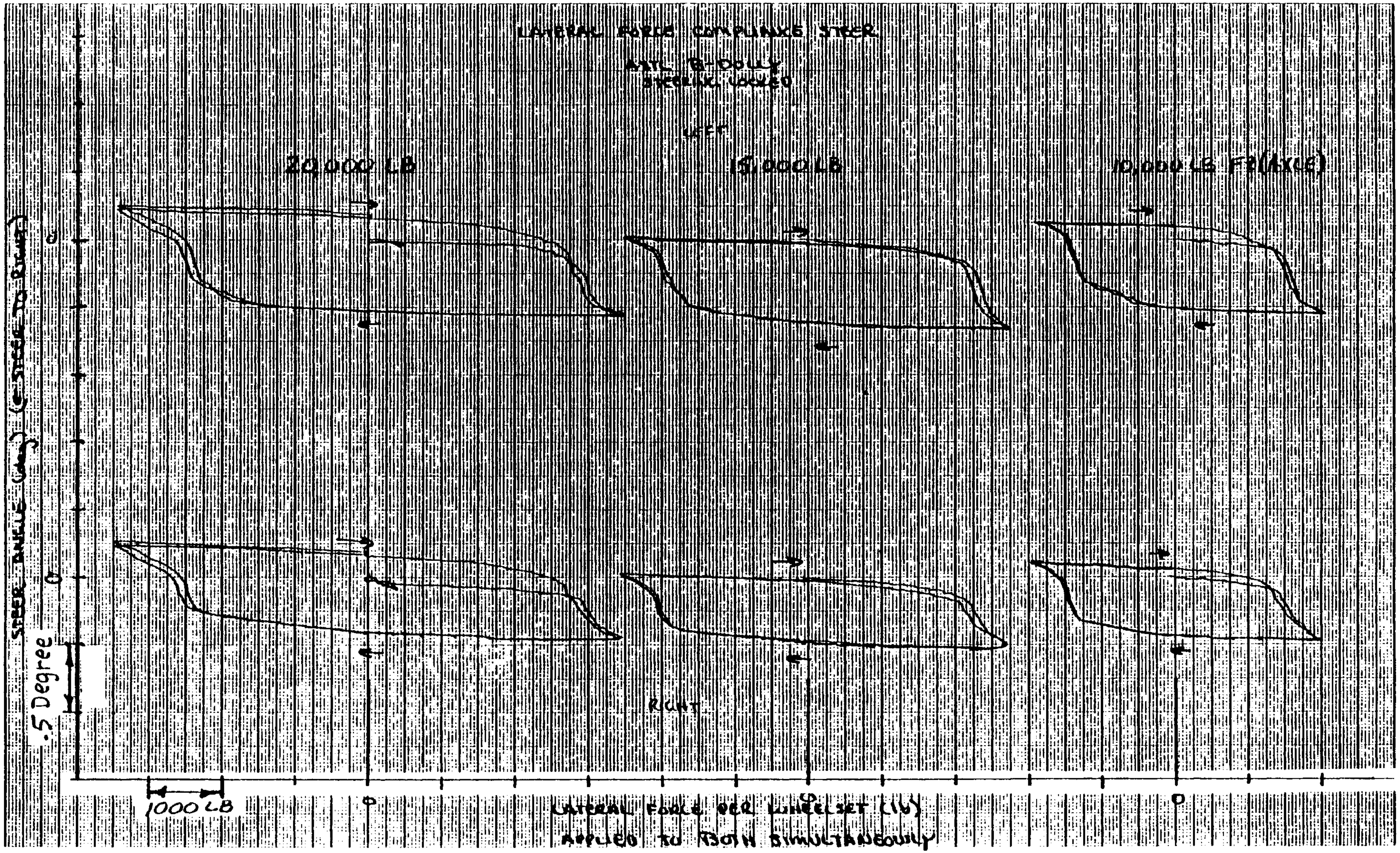


Figure 2.4. Lateral Force Compliance Steer with Ram Pressure of 483 kPa (70 psi).

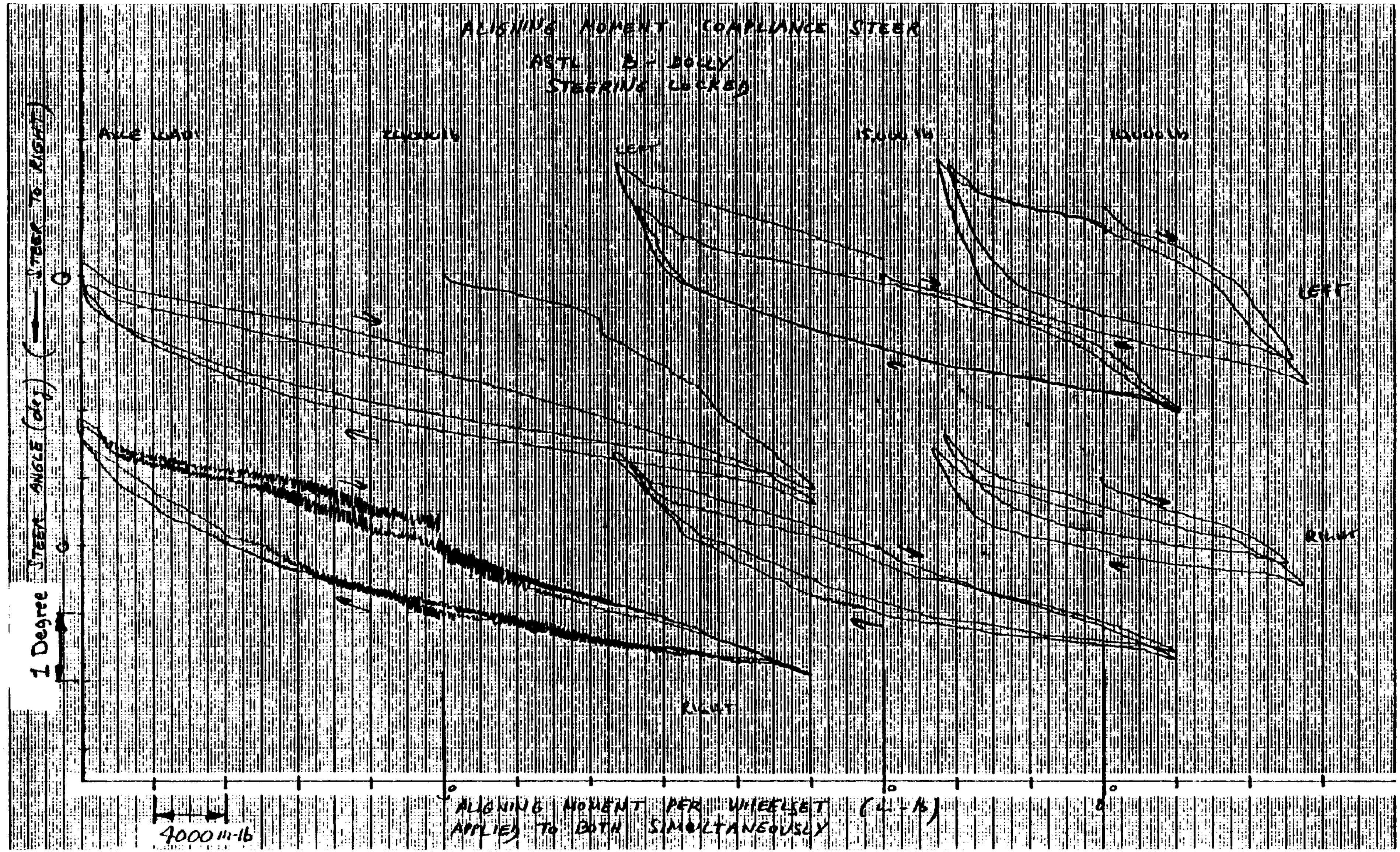


Figure 2.5. Aligning Moment Compliance Steer with Ram Pressure of 483 kPa (70 psi).

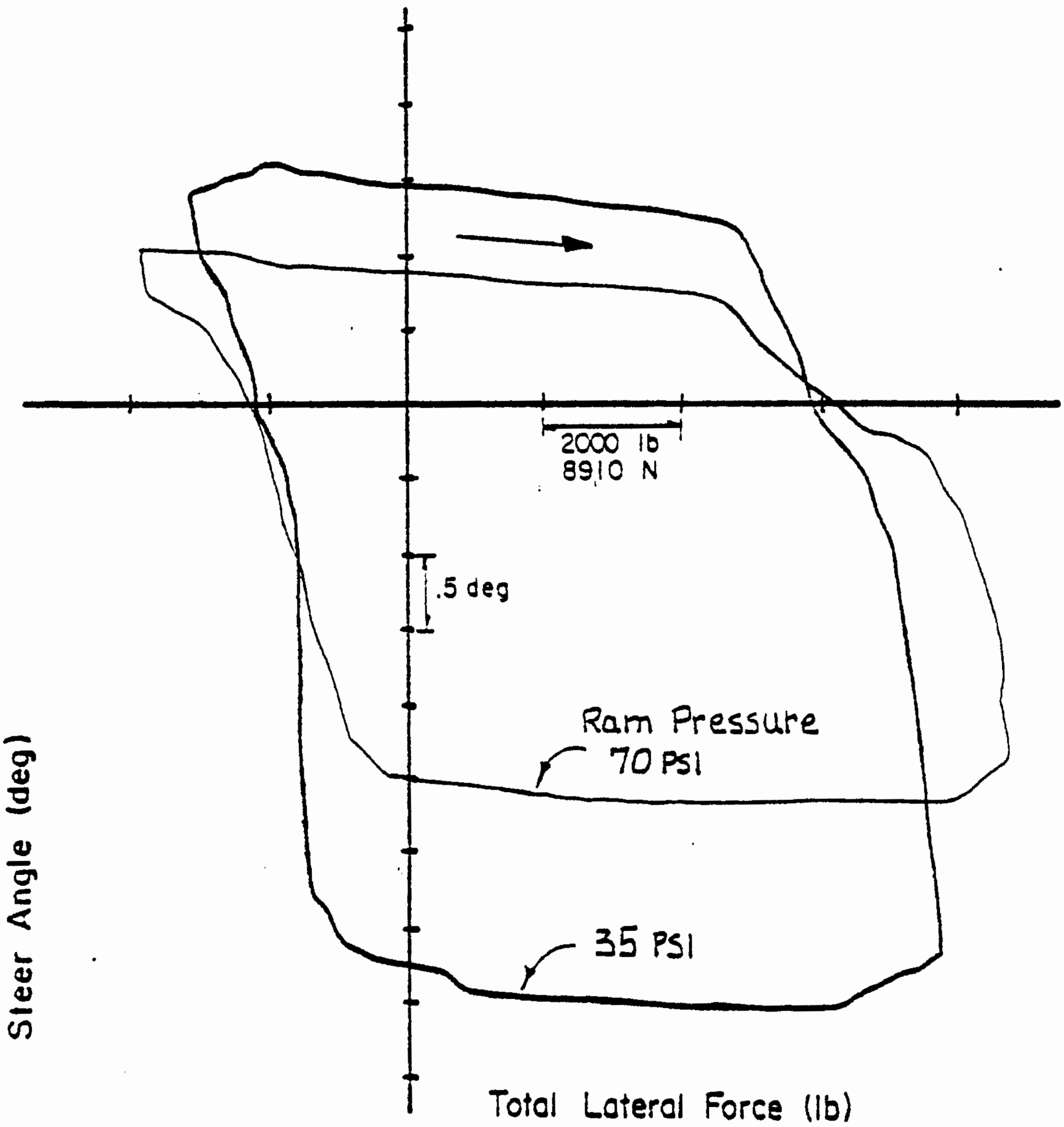


Figure 2.6. Steer Angle Response Over a Large Range of Lateral Force, as Measured while Maintaining Tire Loads Constant.

Compliance steer with a ram pressure of 241 kPa (35 psi) is also shown in Figure 2.6. The basic form of the response is the same in this case, although we do see the greater compliance which was anticipated with the reduction in ram pressure. The on-center stiffness of the lower-pressure case is 10.7 kN/deg (2,400 lb/deg), while the off-center rate is still 445 N/deg (100 lb/deg) at two degrees of steer. The effective width of the detent is slightly less than .5 degree in both cases.

The total friction in the steering system is seen to be a function of the vertical load imposed on the axle, rising from 11.1 kN (2,500 lb) at 44.5 kN (10,000 lb), to 16.7 kN (3,750 lb) at 66.7 kN (15,000 lb), and 21.1 kN (4,750 lb) at 90 kN (20,000 lb). This implies that the friction is a composite of some inherent friction level and a load-dependent friction increasing with axle load. The load-dependency arises, to a large degree, from the moment applied to the kingpin by the vertical load acting through the lever arm afforded by the large kingpin offset and mechanical trail dimensions.

The representation of the self-steering system in the simulation model used in this work considers the steering motion of the B-dolly to be constrained by coulomb friction, and by a bilinear torsional spring. This system is subjected to moments arising from tire forces acting at the mechanical trail lever arm, and from the aligning moment produced by the tires. The lateral force compliance steer data presented above were converted to a moment representation through calculation of the effective mechanical trail for each axle load.

For the sake of simplicity in the following discussions, steering system stiffness will be defined in terms of lateral force per degree steer angle, assuming a fixed mechanical trail. Normalizing these data to a mechanical trail of 12.5 cm (5 in) yields the steering system parameters listed in Table 2.2.

Table 2.2. ASTL Self-Steering System Properties.

	System Pressure (kPa)	
	241	482
On-Center Stiffness (N/deg)	16000	26700
Off-Center Stiffness (N/deg)	670	670

Axle Load (kN)	Coulomb Friction (N)
88.9	15800
66.7	11200
44.4	5550

CHAPTER 3

SIMULATION STUDY

Two UMTRI models were employed in this study of the directional dynamics of vehicles outfitted with the B-type converter dolly. The linear yaw plane model was used to provide a preliminary measure of the vehicle's directional response in both time and frequency domains. For a more precise examination of the vehicle's behavior, the yaw/roll model was utilized.

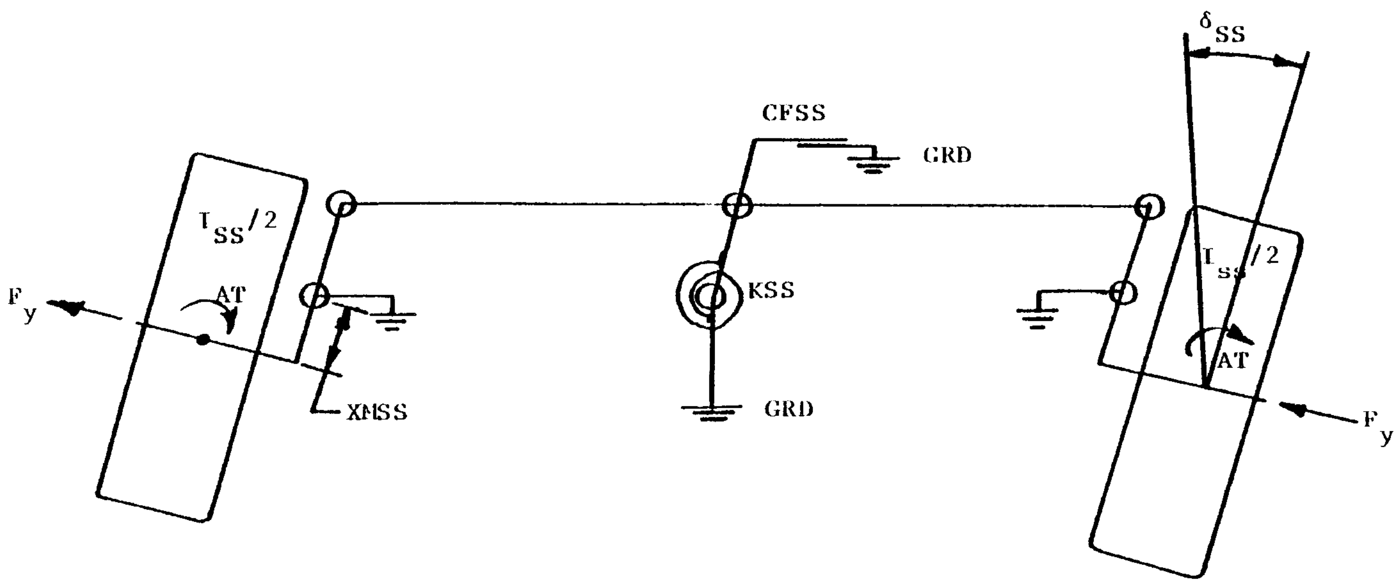
The linear yaw plane model treats the vehicle as a system of units which each possess two degrees of freedom and are acted upon by tire lateral forces proportional to local sideslip angle. The vehicle units are allowed freedom of lateral velocity and yaw rate. Tire forces are directly related to the slip angle prevailing at the axle according to the tire's cornering stiffness. Different versions of the model can be run to obtain specific measures of the vehicle performance. Response of the vehicle can be characterized by the system frequency response in terms of gain and phase angle, time response to a specific steer input, eigenvalues or poles and zeros. In this study, the linear plane model was used primarily to evaluate the frequency response of the vehicle combination to determine the basic nature of a given vehicle's directional response.

The more complex yaw/roll model was employed to conduct in-depth investigations of the vehicle response. This model represents the vehicle combination as a series of interconnected units, each with five degrees of freedom, and each coupled to the next unit by a set of constraint equations defining the hitch mechanism. Each vehicle unit can bounce, pitch, roll, sideslip, and yaw. The longitudinal velocity of the lead unit is constrained to be a constant. Axles are allowed bounce and roll degrees of freeom. Forces and moments generated by the tires are entered in tabular form as a function of load and slip angle. Spring forces are similarly entered as a function of deflection. The model is also capable of simulating the effects of lash and coulomb friction in the suspension. Other parameters describing the suspension are also included, viz., roll center location, auxiliary roll stiffness, and roll steer. A detailed description of the model can be found in Reference [1].

3.1 Self-Steering System Model

To accommodate the self-steering system of the B-type converter dolly, the yaw/roll model was modified to allow normally-fixed axles to steer in response to lateral forces and aligning moments generated by the tires. The form of the model was based on the measured parameters of the ASTL B-dolly. Based on these data, the model represents the steering motion of the dolly wheels as being constrained by a torsional spring, with varying rates on- and off-center, and by a coulomb friction function which is dependent upon the axle load. Experimental data suggested that a quasi-static representation of the dolly system would be adequate for systems of the type observed, but for systems with low levels of friction and stiffness, a dynamic model would be required. The model illustrated in Figure 3.1 was implemented for use in representing the extremely compliant, low friction cases, as well as the more representative cases in which higher levels of steering system stiffness and friction are present.

The static characterization of the self-steering system is shown in Figure 3.2, with experimental data to demonstrate how the model reproduces the behavior measured with the ASTL dolly. The parameters for the system are specified in terms of the off-center rate, the ratio of the on-center rate to the off-center, and the friction level. In addition to these parameters, the dynamic model also requires identification of the system inertia for steer motions and also allows for the introduction of viscous damping.



CFSS - Coulomb Friction

KSS - Self-Steering System Stiffness

XMSS - Mechanical Trail

GRD - Dolly Chassis Ground

F_y - Tire Lateral Force

AT - Tire Aligning Moment

δ_{SS} - Self-Steer Angle

I_{SS} - Self-Steering Inertia

Figure 3.1. Steering System Model.

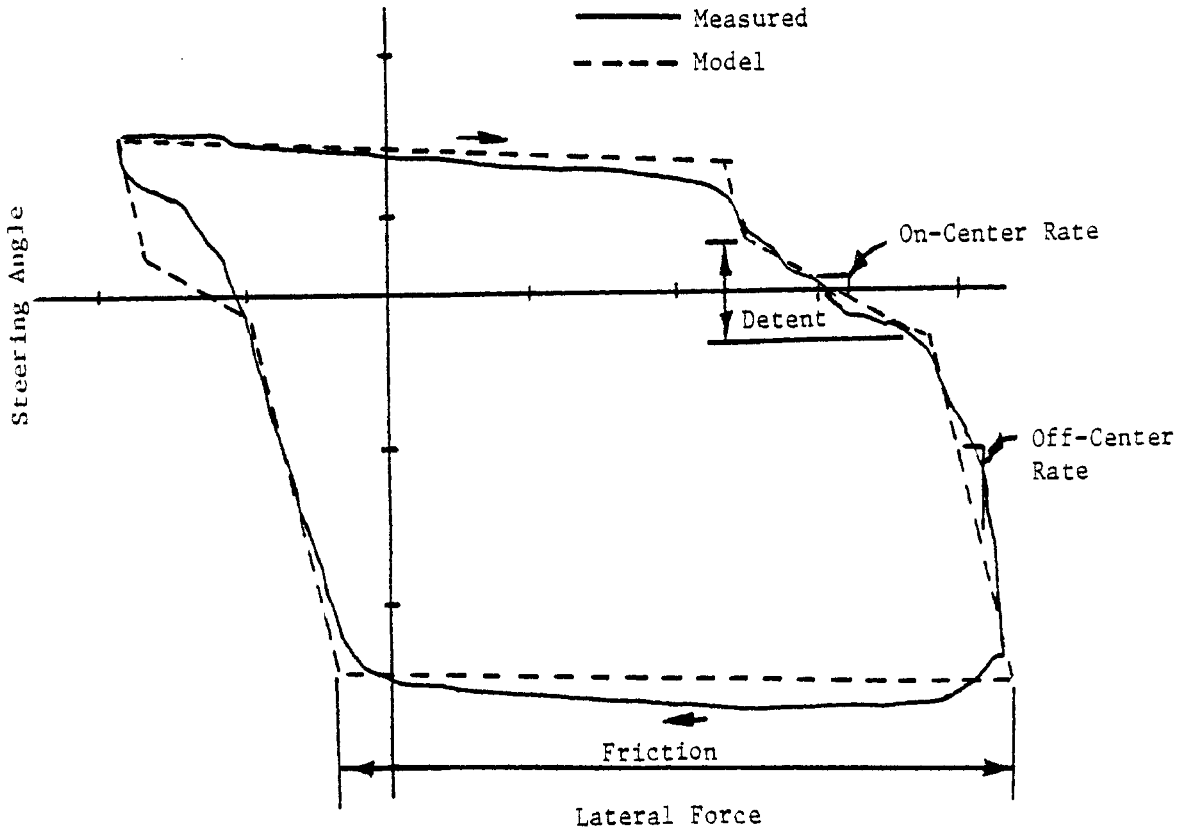


Figure 3.2. Steering System Characterization

3.2 Simulation Validation

To check the ability of the simulation to predict the dynamic directional performance of vehicles outfitted with B-dollies of the type described, simulation results were compared to test data taken by the Ontario Ministry of Transportation and Communication. Samples of such data corresponding to a ramp-step steer maneuver are compared in Figure 3.3. We see that the general nature of the vehicle response is well characterized by the simulation. Various suspension parameters of the MTC vehicle combination were also measured at the UMTRI facility and these data were used along with other parameters supplied by MTC in representing the test vehicle in the simulation.

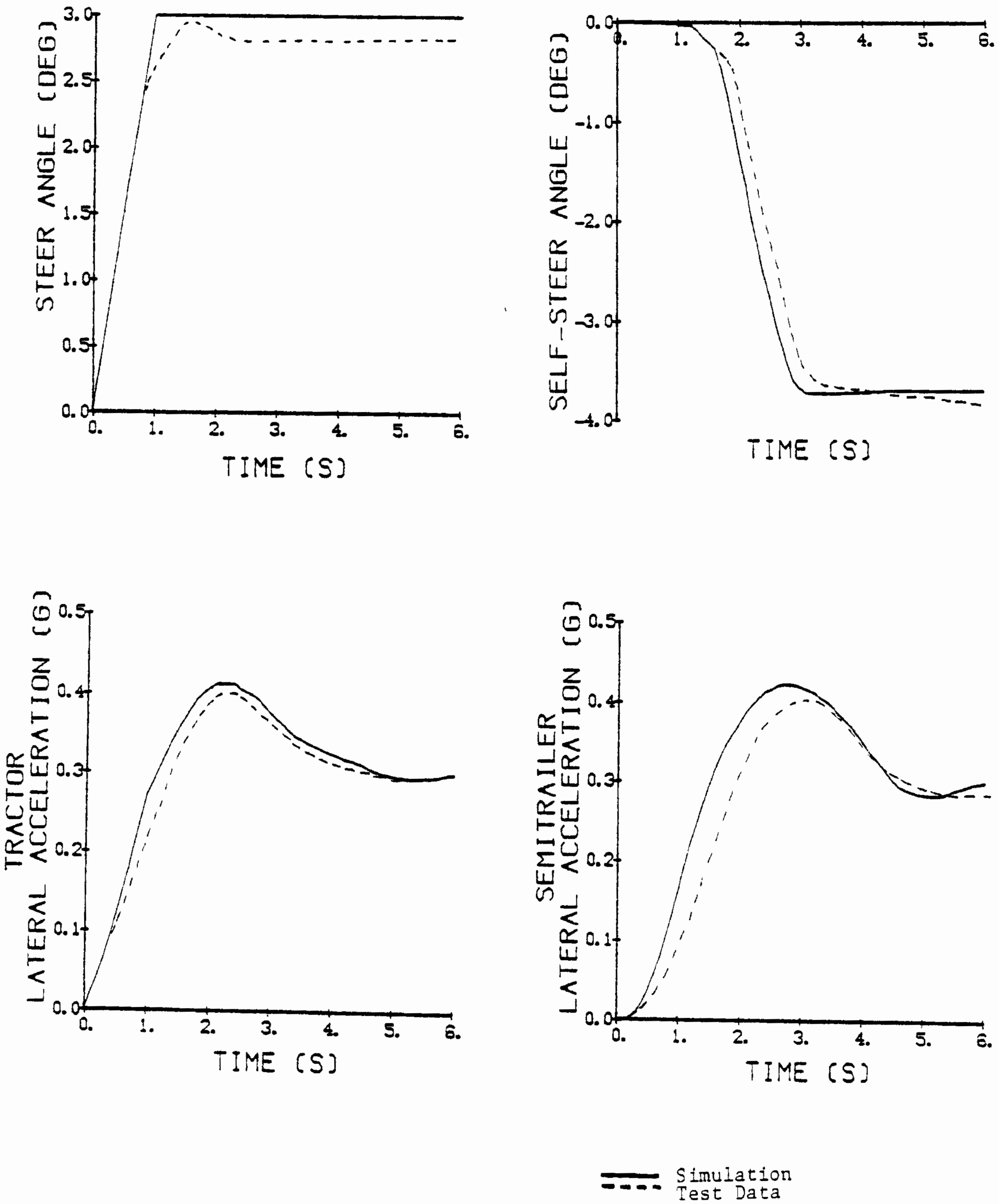


Figure 3.3. Simulation Validation Results.

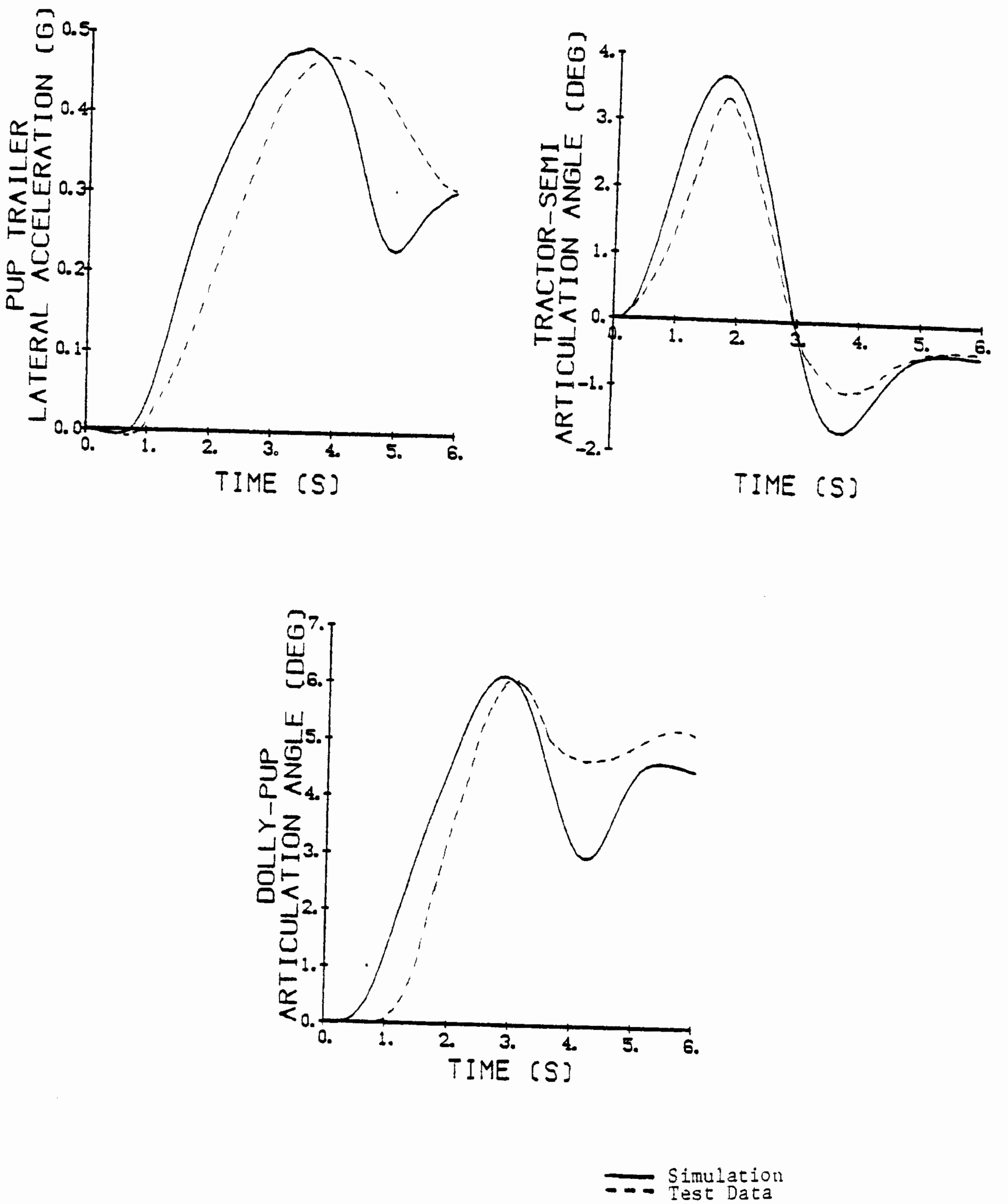


Figure 3.3. (Cont.)

CHAPTER 4

SIMULATION RESULTS

The simulation activity addressed several specific topics, namely:

1) The influence of varying levels of friction and stiffness in the self-steering system of the B-dolly
2) Maximum levels of force reaction at the pintle hitch couplings of the B-dolly
3) Influence of dolly length on the dynamic performance of the vehicle combination
4) The influence of B-dollies on the dynamic performance of triples

The major thrust of the investigation involved the first of these subjects, namely, the influence of friction and stiffness variations in the self-steering system on the dynamic performance of the vehicle combination. If the steering system of the B-dolly is locked or otherwise prevented from steering (such as when "stuck" in the coulomb friction), the vehicle combination behaves the same as a B-train. This condition defines an effective upper range for the stiffness and friction parameters. At the other end of the range would be a self-steering system that possesses no friction or stiffness, as would be the case if the steering system consisted merely of wheels castered on frictionless kingpings. This latter arrangement results in the dolly tires seeking the zero slip angle condition such that no lateral forces are produced. "Caster-steering" is a term used in the report to represent this other extreme variation in dolly steering parameters.

One type of double-trailer combination was used to examine the influence of the self-steering dolly on the performance of the vehicle train. A five-axle double with a gross vehicle weight of 36,300 Kg (80,000 lb) was selected as a likely candidate for the installation of a B-type converter dolly similar to that supplied by ASTL. This train consists of a single drive axle tractor having a 3 m (120 in) wheelbase and pulling two 8.23 m (27 ft) single axle trailers. The B-dolly is a single axle unit as well, with a 2.03 m (80 in) pintle-to-axle dimension, similar to the ASTL specimen. The vehicle combination carries 4,500 Kg (10,000 lb) on the tractor steering axle and 7,900 Kg (17,500 lb) on each of the other axles. By varying the constraints placed on the yaw and roll freedom of the dolly relative to the semitrailer and the parameters defining the steering action of the dolly axle, the vehicle can be configured to represent A- or B-trains as well as B-dolly trains having various self-steering systems.

The simulation was exercised to evaluate the performance of various configurations in maneuvers that represent real maneuvering scenarios that are considered challenging either to the commercial vehicle population as a whole or to this specific class of vehicles. Steady turning and the transient response associated with entering a turn were examined to evaluate the steady-state, high-speed offtracking of the combinations, as well as to explore the possibility of a degradation in dynamic stability due to the steerable dolly. Two steer input methods were used, namely, ramp/step steer inputs and closed-loop negotiation of a circular path which is approached from a tangent.

Another facet of multiply articulated vehicle performance that warrants close scrutiny is the response of the train during a rapid lane-change maneuver. Closed-loop negotiation of a specific course designed to produce a single sinusoid of lateral acceleration, of prescribed amplitude and period, was used for simulation of lane-change maneuvers. Other performance measures, such as low-speed offtracking, were obtained with specific tests.

4.1 Parametric Variations of Self-Steering System

The two properties of the self-steering system that are of interest in examining the dynamic performance of the overall vehicle combination are the coulomb friction and stiffness parameters. These two parameters define the level of lateral force at the dolly tires beyond which the vehicle combination ceases to respond like a B-train. Also, the steering stiffness parameter will determine an effective "cornering stiffness" that will prevail at the dolly axle after the coulomb friction has been "broken." That is, when the coulomb friction moment has been exceeded by the sum of the tire moments about the kingpin, a steer compliance is introduced in series with the tires on the dolly axle thus reducing the effective cornering stiffness at that axle. With the tires of the dolly axle steering in response to applied lateral force and aligning moment, the "apparent slip angle" (that is, the slip angle of the axle center) must be increased to achieve the same level of lateral force which would be obtained in the case of a fixed, non-steering axle.

The effect of combined friction and compliance in the self-steering system is illustrated in Figure 4.1. Assuming that the tire produces lateral force linearly with slip angle according to the cornering stiffness value, C_α, and assuming that the steering system is equivalent to another serial spring having the same level of stiffness, C_α, the force developed per unit of apparent slip angle is equal to C_α, while the wheel steer motion is locked by the friction, but becomes $C_\alpha/2$ once the friction has been broken, as shown in the figure.

Figure 4.2 illustrates the various force-deflection characteristics of the self-steering systems examined in this study. The B-train (or rigid dolly-axle) configuration is represented by the x-axis; that is, no steering is obtained regardless of the level of applied side force. The caster-steered version is represented by the y-axis, for which a zero level of side force is required to maintain any steer angle. The intermediate system configurations, except the ASTL case, were defined in terms of the "axle cornering stiffness" which follows from the steering stiffness and axle load level which determines the coulomb friction level. Friction level is defined here as half of the total width of the hysteresis loop determined by coulomb friction.

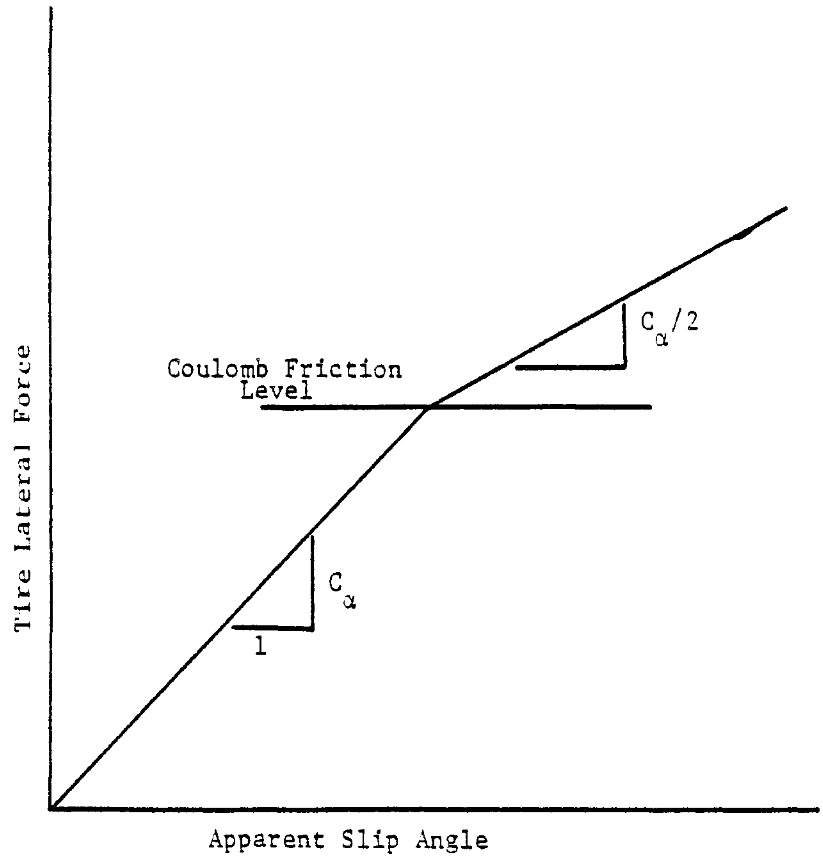


Figure 4.1. Influence of Friction and Compliance of Self-Steering System on Axle Lateral Force Production.

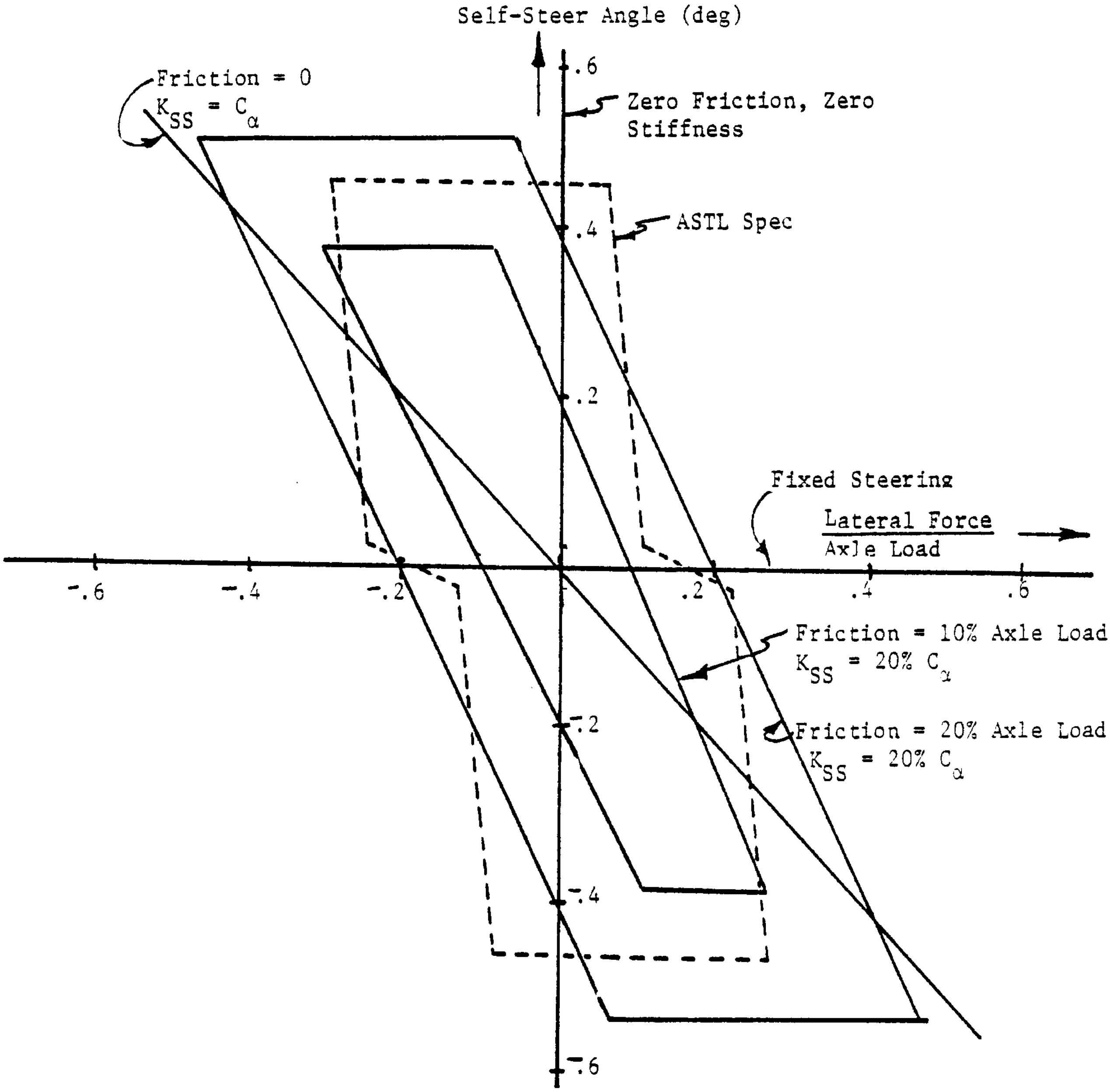


Figure 4.2. Self-Steering System Models. Various Schemes for Representing the Relationship between Lateral Force and Self-Steer Angle on B-Dollies.

4.1.1 Low-Speed Performance. The passive self-steering system of the B-dolly is intended to improve the low-speed maneuverability of the vehicle and to reduce the tire scrub and wear which spread axles cause in tight radius turns. The "scrub" or lateral slip condition would otherwise be incurred during low-speed cornering whenever the radius reduces such that the tires on the two axles of the spread set (namely, the semi and dolly axles) operate at significant values of slip angle. Since the slip angles would be of opposite sign, opposing forces are developed which provide for a yaw moment on the semitrailer. This moment, in turn, calls for a side force reaction at the tractor fifth wheel and, ultimately, at the tractor tires.

Offtracking on a 20 m radius curve at 10 Km/hr is used as a measure of the low-speed maneuverability of the train. The offtracking of a vehicle is defined as the maximum deviation of the center of any axle from the arc being tracked by the center of gravity of the tractor. This maximum deviation value generally occurs at the last axle of the combination. Offtracking results for the A-train, B-train, and caster-steered B-dolly train are shown in Figure 4.3. We see that the B-train offtracks considerably more than any of the other vehicles (3.8 m). The minimum offtracking is observed with the caster-steered B-dolly (note that this configuration offtracks even less than the A-train). As would be expected, the B-dolly with a compliant steering system shows performance between that of a B-train and caster-steered B-dolly train. In Figure 4.3, the ASTL dolly is not addressed directly since its offtracking performance is identical to that of the B-train. This result reflects the fact that, at this radius of turn, the lateral force at the dolly axle is insufficient to break the friction and cause steering of the dolly wheels.

The steady-state tire forces obtained in a 20 m curve at 10 Km/hr with a B-train and an equivalent vehicle incorporating a caster-steered B-dolly are illustrated in Figure 4.4. Despite the fact that the lateral acceleration is practically negligible at this turn condition (.04 g), the semi and dolly axles of the B-train are operating at

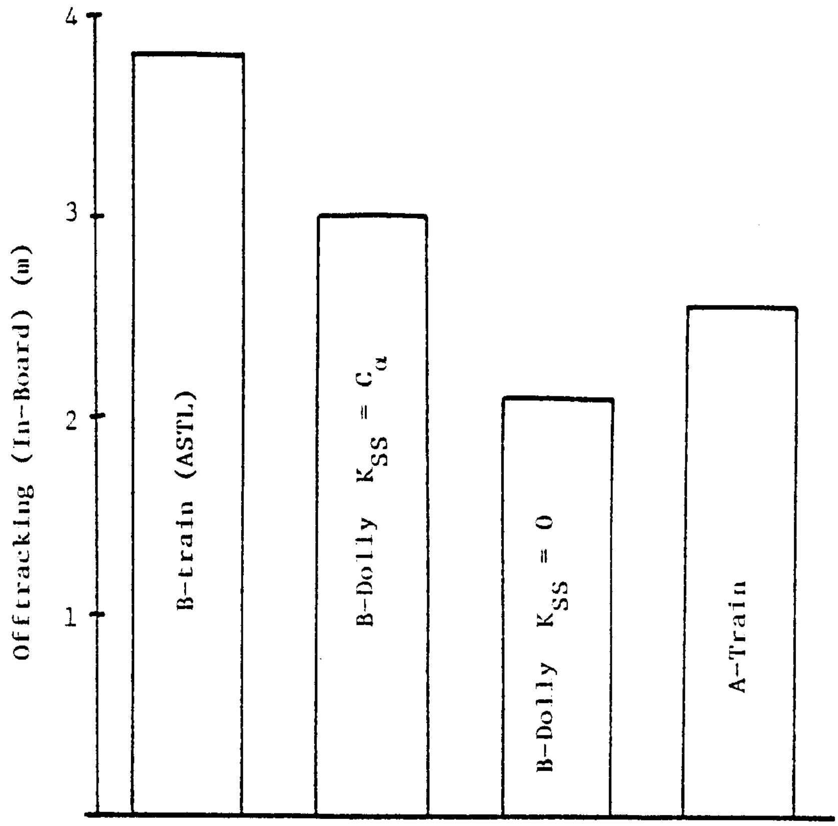


Figure 4.3. Offtracking 10 Km/hr, 20 m Radius Curve.

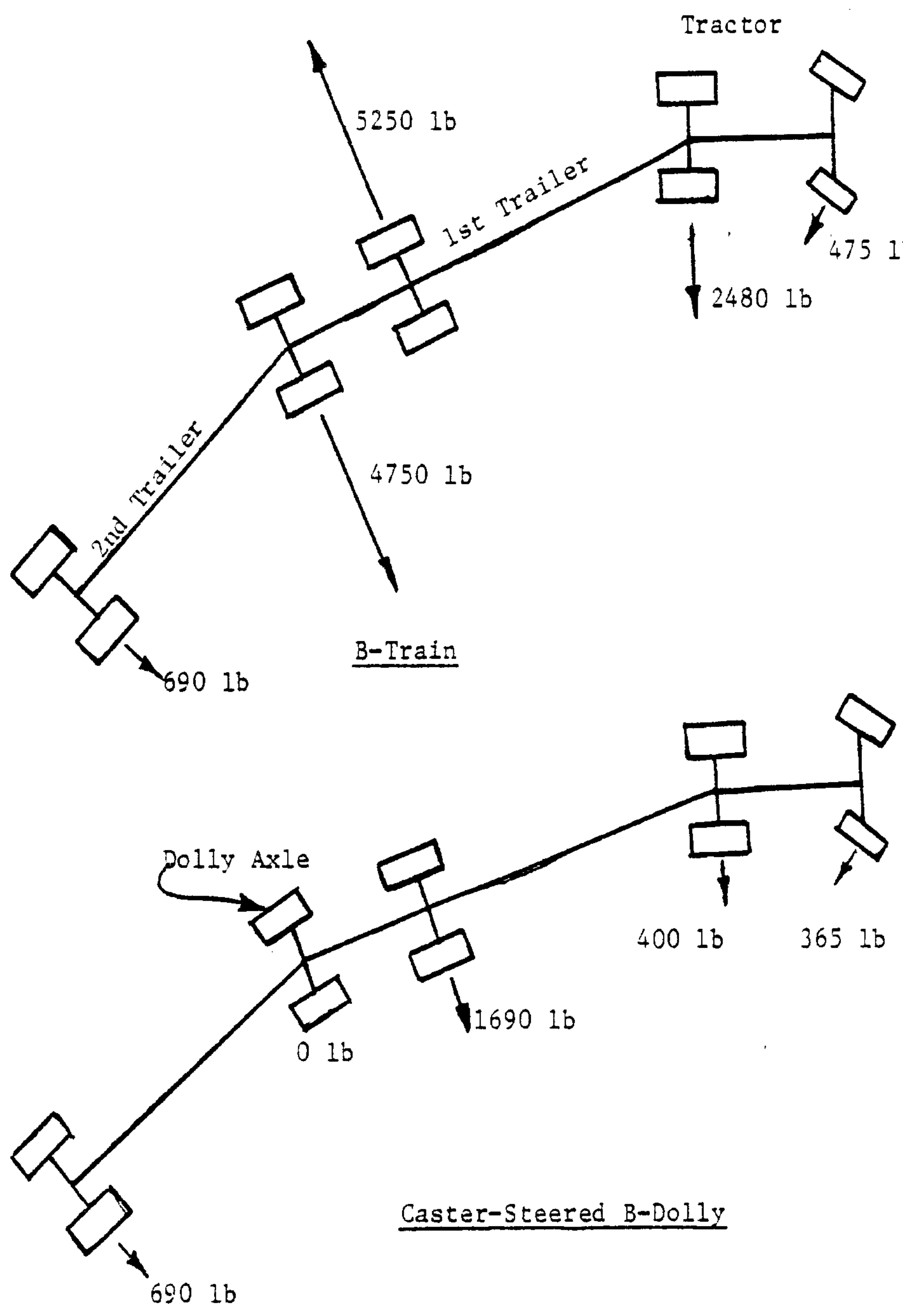


Figure 4.4. Tire Lateral Forces Obtained in a 20 m Radius Curve at 10 Km/hr.

lateral force levels which are between 25 and 30% of the axle load. As mentioned previously, for a small value of turn radius such as that employed here, the forces are of opposite polarity on these axles such that a large couple is produced, resulting in force reactions at the tractor fifth wheel and, consequently, at the tractor tires.

When the dolly is configured such that its tires produce zero lateral forces, the demand for side force at the two preceding axles (in a low-speed curve) is reduced, thus reducing tire scrub and wear. Moreover, the low-speed offtracking of a combination vehicle and the scrub of tractor, semi, and dolly tires is reduced with decreased stiffness in the dolly's self-steering system. Indeed, with a freely castering system, the offtracking performance of an A-train can be surpassed. The lower the stiffness of the dolly steering system can be made, the more favorable its low-speed performance will become.

4.1.2 High-Speed Curving Performance. During the negotiation of a curve at higher speed, the tires on each of the axles of the vehicle combination are called upon to produce the lateral forces needed to achieve force and moment equilibrium on the vehicle as a whole. When a decreased level of stiffness is introduced into the self-steering system of a B-dolly, the dolly's tires do not produce a lateral force response to "axle slip angle" which is in proportion to the cornering stiffness of the dolly tires. As a result, a lateral force "deficiency" will prevail at the B-dolly axle such that the axle on the semitrailer must "make up the difference" by itself, assuming a greater slip angle. Thus, the semi-dolly unit must, as a whole, adopt a more outward sideslip angle in order to enable the tires at the semi axle to produce the necessary reaction forces. The resulting vehicle attitude produces an increased outboard offtracking and a larger swept path for the combination.

This "softening" of the semi-dolly unit in yaw also can affect the transient performance of the vehicle when entering a curve. That is, the decreased stiffness in yaw leads to larger motions associated with the transient portion of the maneuver. To address these steady-state and transient characteristics, two separate maneuvers were simulated, namely, closed-loop negotiation of a fixed radius curve, and a ramp/step of steering input. Although both maneuvers produce a steady-state turn, the ramp/step input method lends itself better to the examination of the transient character of the response.

Steady-state offtracking for three B-type trains and an A-train are presented in Figure 4.5. These data are for negotiation of a 260 m (853 ft) radius curve at forward velocities of 50-90 Km/hr. The caster-steered case yields outboard offtracking which is substantially more than any of the other combinations. The A- and B-trains exhibit very little offtracking. The offtracking performance of the B-dolly combination whose steering system has a stiffness level approximating the sum of the tire cornering stiffnesses at the dolly axle falls between that of the locked and freely-steering variations of the same vehicle. The performance of the ASTL-spec dolly steering system is, again, identical to that of a B-train in this maneuver (since the dolly's tire forces are too low to overcome the friction within the steering system).

Vehicle response in the transient maneuver, involving an abrupt transition into the 260 m radius curve at 80 Km/hr, is shown in Figure 4.6 for the A-train, B-train, and the caster-steered B-dolly train. The large excursions in articulation angle for the caster-steered case point out a dynamic overshoot problem with this form of dolly. It is also worthy of note that the steering sensitivities of the three vehicles are different. While the A- and B-trains are seen to be very similar in terms of steering gain, the zero-stiffness case of the B-dolly indicates a reduced steady-state gain and, as a result, requires a greater steer input at the tractor in order to negotiate the curve. The lack of side force capability at the dolly axle of the caster-steered case eliminates the resisting couple associated with spread axles, and, as a consequence, reduces the side force demand on the tractor drive axle. The reduction of the slip angle at the drive axle is accomplished by altering the sideslip angle of the tractor such that more steer angle is required to properly balance the tractor forces.

The frictional qualities of the self-steering system present a more complicated influence on the directional performance of the vehicle

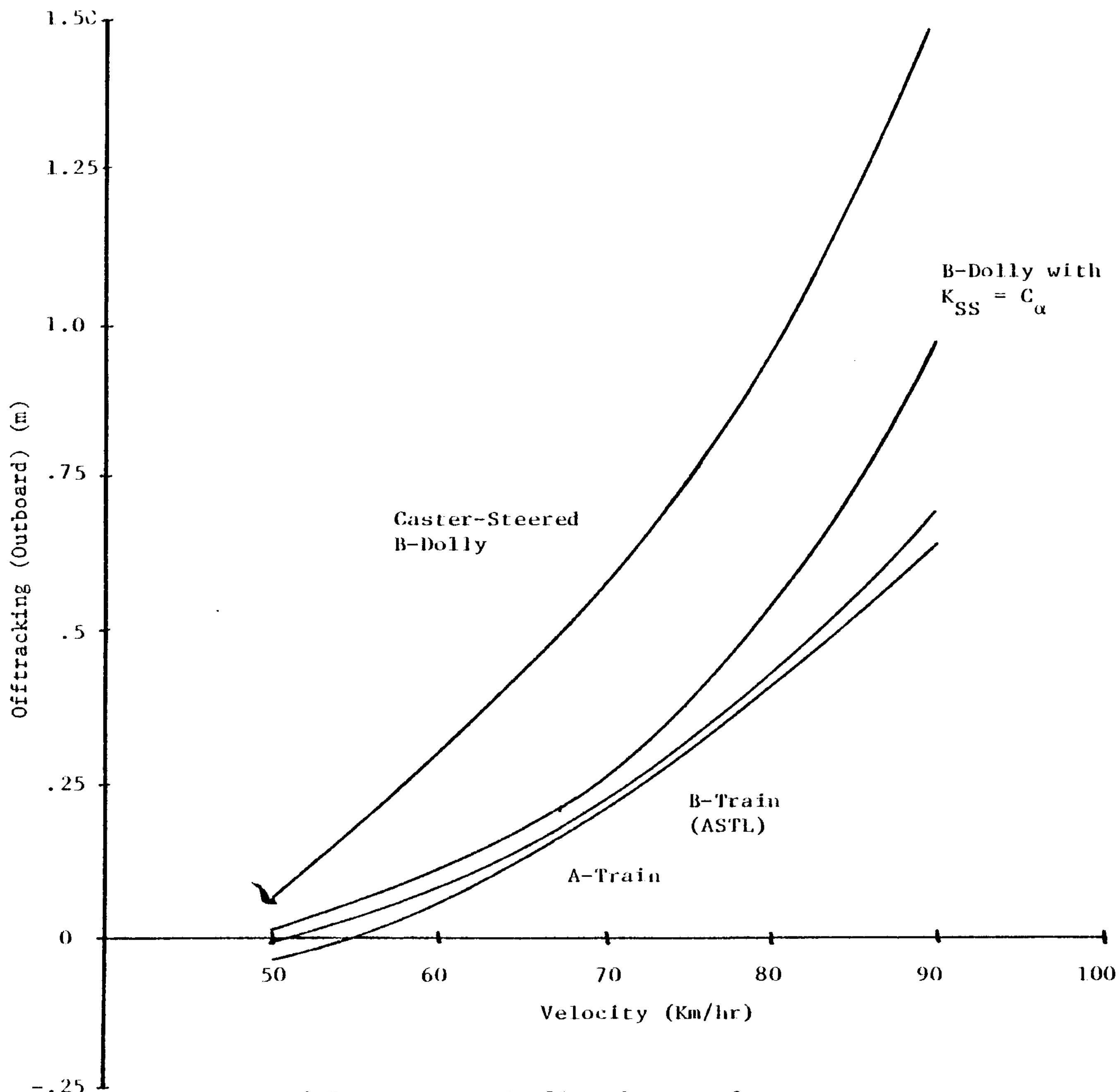


Figure 4.5. High-Speed Offtracking Performance.

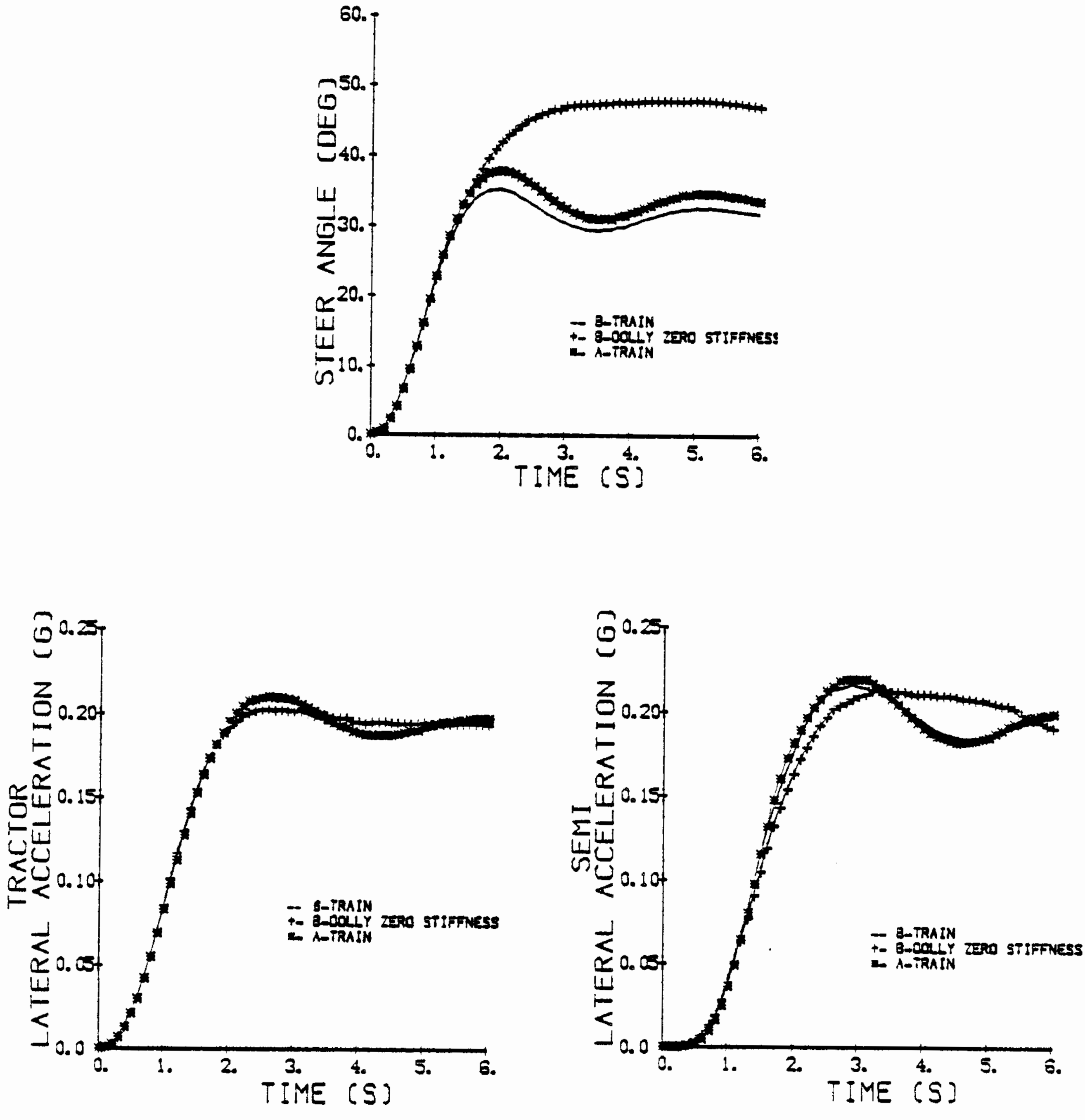


Figure 4.6. Closed-Loop Negotiation of 260 m Radius Curve - 80 Km/hr.

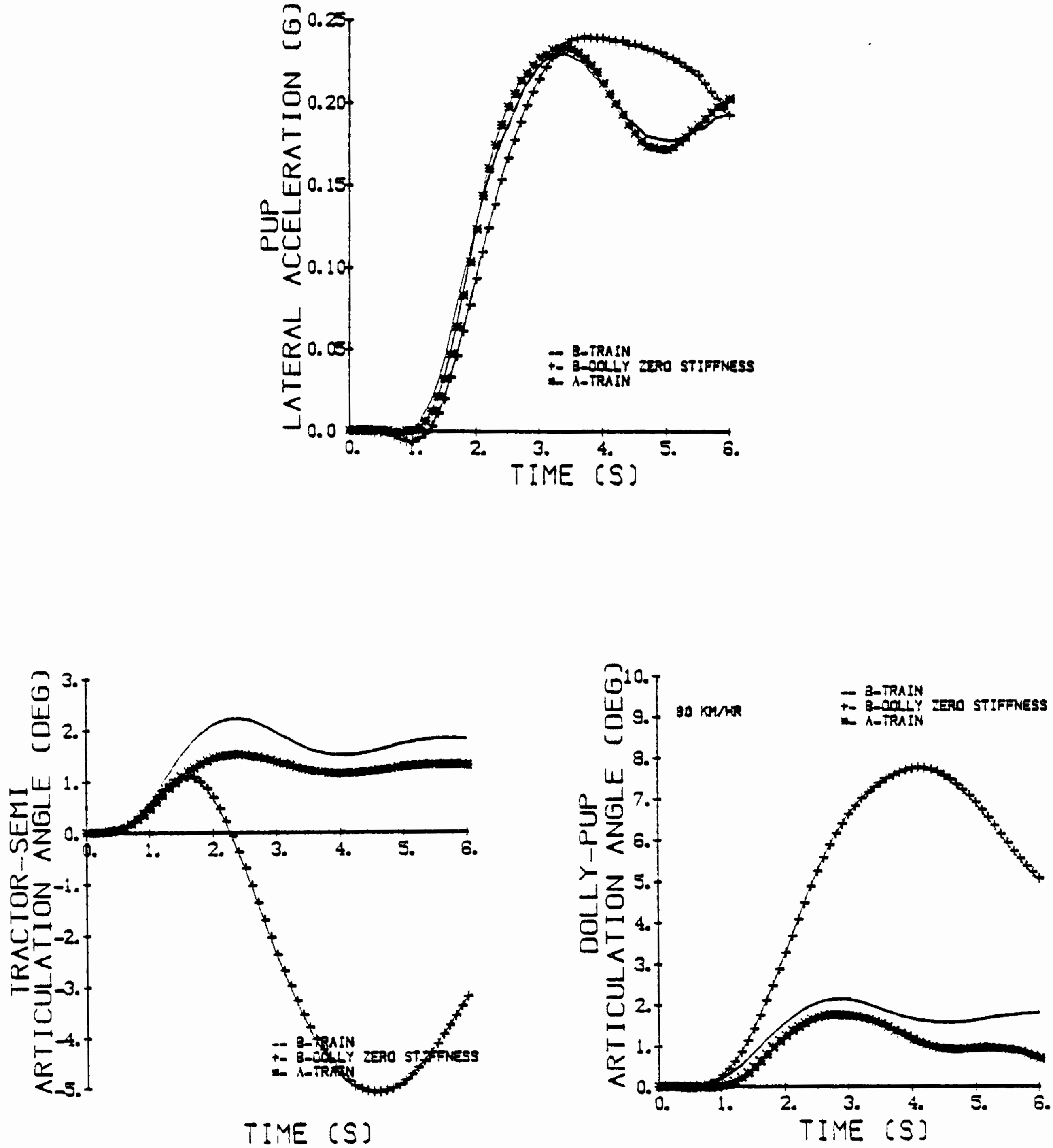


Figure 4.6 (Cont.)

system. When the friction is "broken" during the course of a maneuver, the operating parameters of the train can change discontinuously. This is particularly true if the stiffness of the steering system is low, such that a large effective reduction in the cornering stiffness at the dolly axle follows the "friction break." To illustrate this point, three B-dolly trains with varying levels of steering system friction were simulated in a ramp/step steering maneuver. Two of these vehicles have friction levels corresponding to 10% and 20% of the axle load, respectively, and both have a value of steering system stiffness which is on the order of 20% of the total cornering stiffness of the four dolly tires. These systems are illustrated in Figure 4.7. The third vehicle is a B-train, or any B-dolly train having sufficient friction to keep the dolly axle effectively locked.

As seen in Figure 4.7, the response of the three combinations is identical up to the time that the friction is broken in the respective steering systems. Once the friction is broken and the tires on the dolly axle begin to steer, the self-steered cases begin to deviate from the response exhibited by the B-train. The response of the B-train is seen to be heavily damped; that is, none of the units have any appreciable overshoot in lateral acceleration. Both of the self-steered cases have considerable overshoot, with the lower friction level being associated with the larger overshoot. At the onset of self-steering action, the semi-dolly unit is "softened" in yaw and swings out at a greater rate to achieve the slip angles necessary to balance the forces and moments on the unit. The semi yaws outward relative to the tractor, adopting an outboard articulation angle. In turn, the pup trailer articulates further relative to the semi. Note also that the steady-state steering gain of the overall vehicle combination is lowered somewhat by the self-steering action of the dolly. In this regard, also, we see that greater deviation from the B-train's response is shown by the steering system having a lower level of friction.

The large overshoot in the transient lateral acceleration response of the vehicles with self-steering dollies suggests that the rear trailer will experience an amplified roll moment tending toward rollover.

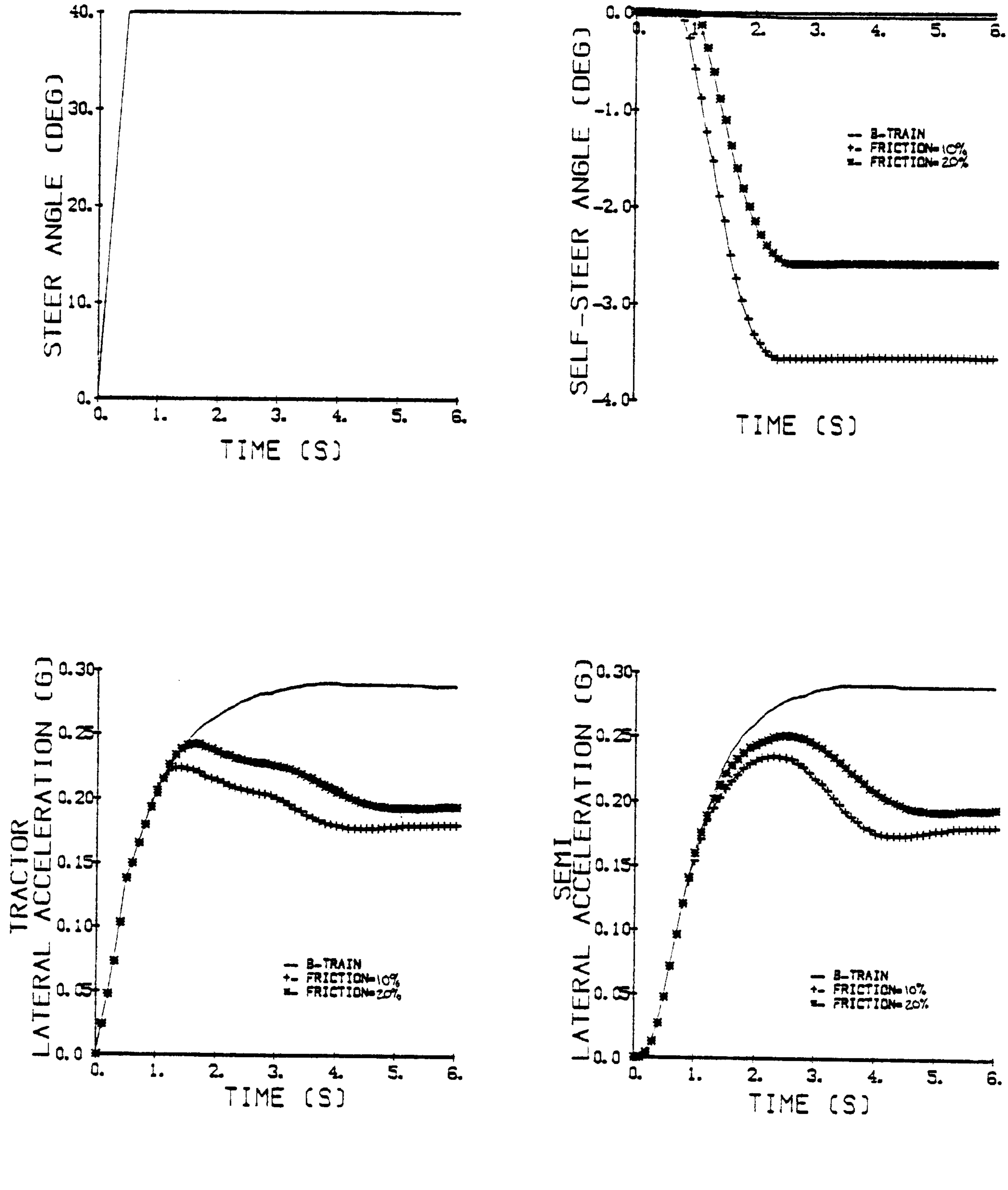


Figure 4.7. Influence of Friction Level on Step Input Response.

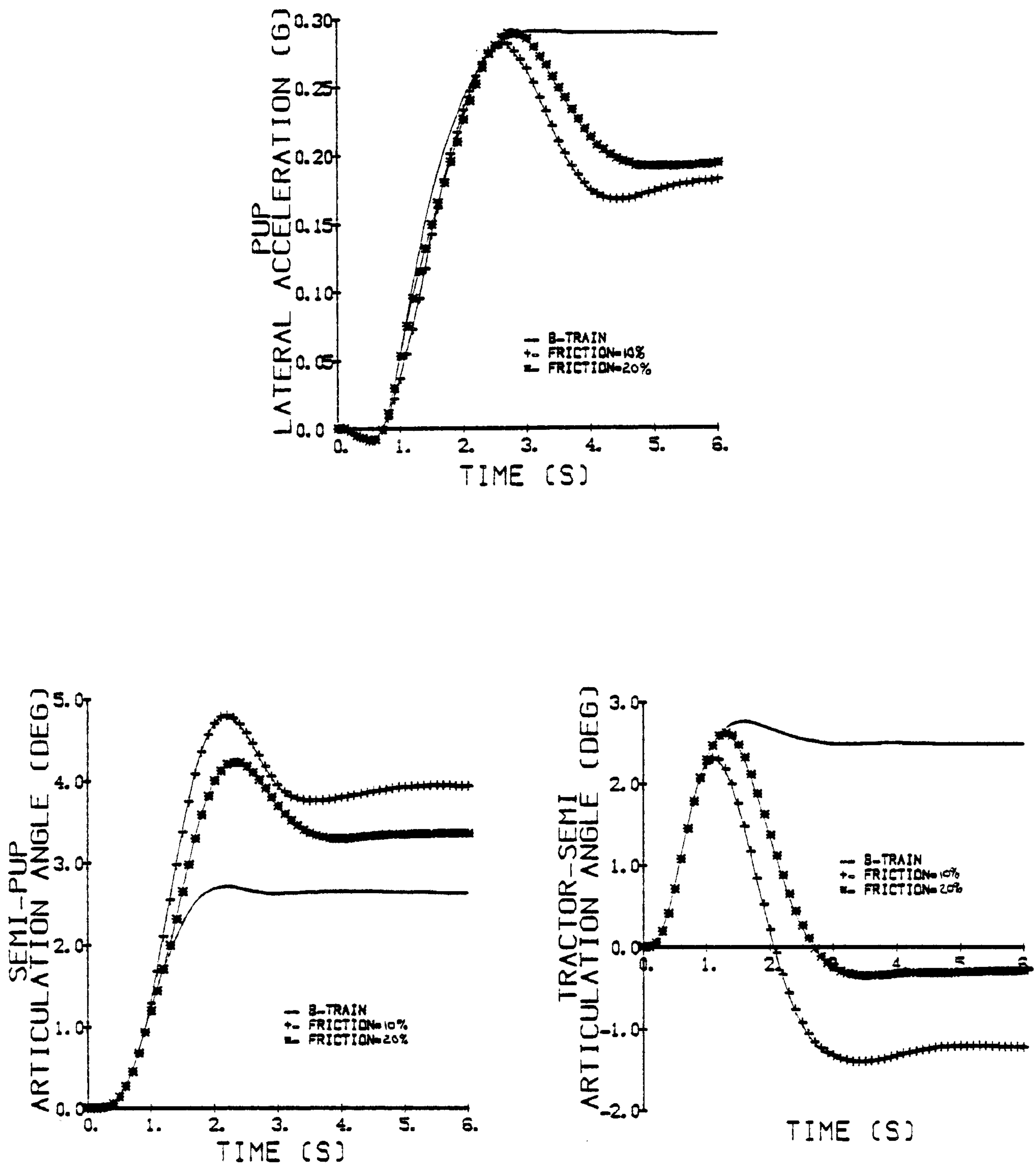


Figure 4.7 (Cont.)

In the case of the lower-friction steering system, above, the lateral acceleration of the pup trailer was seen to overshoot the steady-state value by 65% in response to the ramp/step steering input. With such a vehicle, an abruptly-applied steering input intended to establish a steady turn of .25 g could easily result in sufficient lateral acceleration at the pup to cause it to roll over, were it not for the roll coupling of the two trailers together.

Moreover, low levels of friction and stiffness in the dolly's self-steering system tend to increase high-speed offtracking and decrease the apparent yaw damping of the semi-dolly unit. Thus, the least objectionable behavior in steady turns and in response to ramp/step steering inputs is obtained with high levels of friction and stiffness in the dolly steering system—just those conditions that generally cause the low-speed performance to be degraded.

4.1.3 Rearward Amplification. Amplification of the tractor motion at the last trailer of a multiply-articulated vehicle during lane-change or similar maneuvers is a primary consideration in the active safety of multiply-articulated vehicles. The ratio of the peak value of lateral acceleration at the pup trailer to the peak experienced by the tractor is defined as the rearward amplification measure. Reduction of this amplification ratio is a primary benefit deriving from the design of B-trains. That is, elimination of one of the articulation points in the vehicle train, and the elimination of the dolly dynamics, results in a substantial reduction in rearward amplification. Since the magnitude of the amplification response increases with speed, the phenomenon is examined here for cases involving B-dollies at the highway speed of 100 Km/hr (62.1 mph).

Since rearward amplification is also sensitive to the frequency of the tractor steering input, the responses of an A-, a B-train, and a castered-steered B-dolly train are shown in Figure 4.8 as a function of steer input frequency. These data were generated by the linear yaw plane model and constitute numerics pertaining to the steady-state oscillation of the entire train. The peak amplification of the A-train is seen to

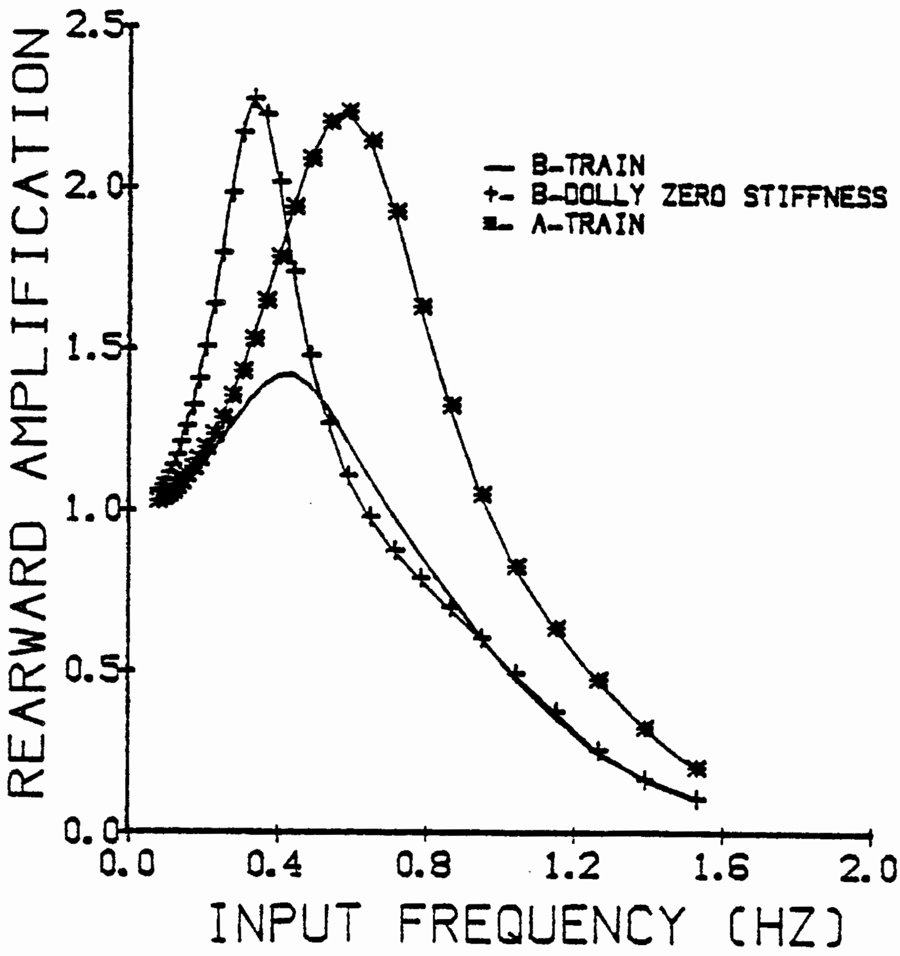


Figure 4.8. Rearward Amplification Frequency Response for Three Types of Vehicle Configurations (Obtained Using a Linear Systems Model).

be 2.2 at a steer input frequency of .6 Hz. Generally, significant steering inputs to commercial vehicles can be considered to be limited to the 0-.5 Hz frequency range, with the upper end of this range representing emergency avoidance maneuvers. The B-train exhibits a relatively lower amplification peak of 1.4 at .4 Hz. The vehicle combination employing a caster-steered B-dolly has a peak amplification of 2.25 occurring at a frequency of .33 Hz—well within the range of steering frequencies that drivers can readily apply. An amplification of this magnitude at such a low frequency is of concern regarding the active safety of a vehicle even during non-emergency operation. That is, since a three-second lane change (.33 Hz steering frequency) constitutes a maneuver of intermediate quickness such as might be undertaken to resolve a minor traffic conflict, it should be of concern that such maneuvers may occur more frequently than the "dire emergency" type of maneuvers needed to strongly excite rearward amplification in conventional A-trains.

The steady-state oscillatory behavior studied by means of a linear representation of a vehicle is useful for gaining insight into the basic nature of the vehicle behavior. To obtain a numerical value for the rearward amplification response in a real-world maneuver, however, it is more useful to perform a single lane-change or obstacle-avoidance maneuver, or to input a single sinusoid of steering. The single sinusoid approach can become complicated by the asymmetric behavior of the vehicle in response to first and second halves of the steer input wave. To avoid this asymmetry, obstacle-avoidance maneuvers were simulated here by using a closed-loop control of vehicle steering in order to follow a predetermined path. With this approach, a selected amplitude of lateral acceleration at the tractor can be obtained, as well as the nominal time period of the maneuver. This method results in improved symmetry between the initiation and recovery phases of the lateral acceleration response of the tractor [2], thus allowing direct comparison of the responses of differing vehicles. Additionally, this time-domain simulation incorporates all of the major nonlinearities needed to represent vehicle behavior in a moderately severe maneuver.

Using trajectories designed to provide .15 g lateral acceleration at the tractor with periods ranging from two to six seconds, a point-by-point representation of rearward amplification as a function of frequency (1/period) was generated for obstacle-avoidance maneuvers and is shown compared to linear results in Figure 4.9. The nonlinear results for the A- and B-trains show larger levels of amplification than were seen with the linear model, although both models yield amplification curves having similar shapes. The nonlinear results for the caster-steered case show reductions in both the amplification level and in the steering frequency at which peak amplification occurs. The specific values obtained with the nonlinear simulation show the caster-steered B-dolly train to have a peak amplification of 1.8 at a frequency of .25 Hz.

The apparent "disagreement" between the linear and nonlinear determinations of the rearward amplification response of the caster-steered B-dolly train is not entirely unexpected. That is, it is clear that the free-castering of the Number 4 (dolly) axle causes the Number 3 (semitrailer) axle to operate at much larger slip angles in order to produce the lateral forces needed to achieve equilibrium. Further, these lateral forces will be nonlinearly related to slip angle such that the total "yaw stiffness" and thus, yaw natural frequency, of the semitrailer-and-dolly unit is reduced below the level considered in linear analysis. As a result, the semitrailer reaches its peak amplification at a lower frequency than that needed to achieve peak response on the pup trailer such that the total amplification is reduced in comparison with the linear case.

Figure 4.10 shows the rearward amplification of the A-train, B-train, and the caster-steered B-dolly train as a function of lane-change period. Although these data present the same information as was shown in Figure 4.9, this format serves to better illustrate the relatively high rearward amplification of the caster-steered B-dolly train during the long-duration lane changes.

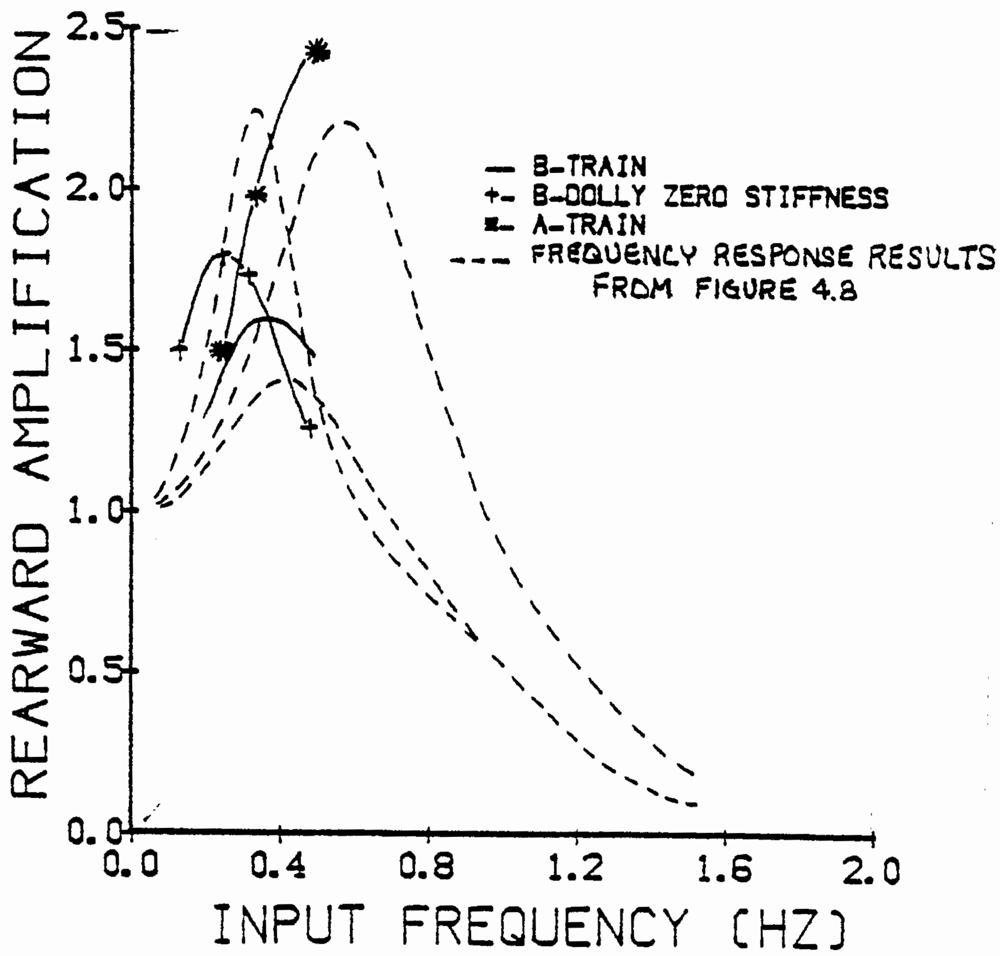


Figure 4.9. Comparison of Single Lane-Change Response to Linear Frequency Response.

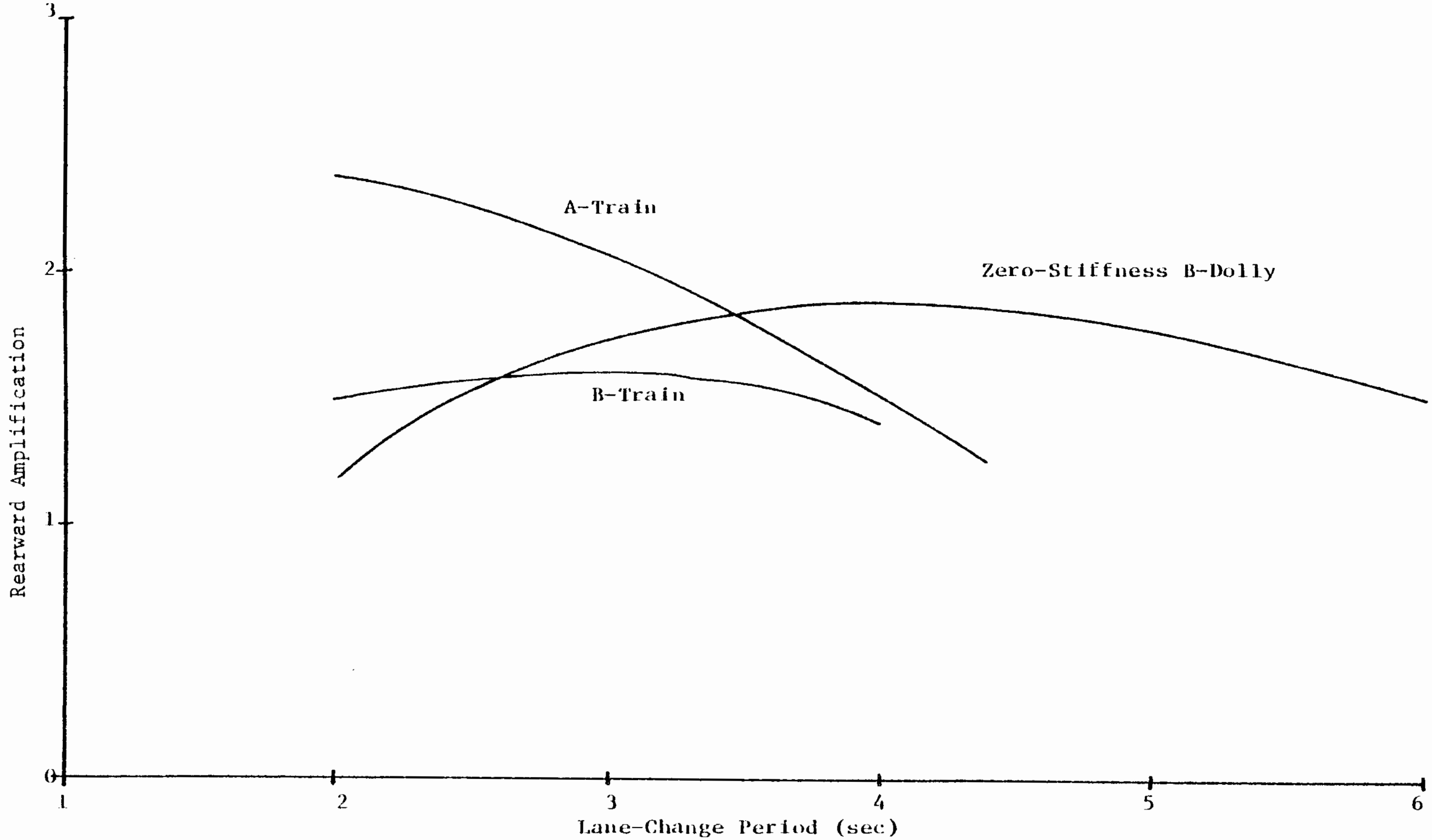


Figure 4.10. Rearward Amplification for Closed-Loop Lane Changes of Various Input Periods.

Another significant consequence of the rearward amplification responses is shown in Figure 4.11. In this figure, the lateral displacement overshoot for the combination is shown as a function of the nominal steering period employed in the maneuver. This overshoot is the amount that lateral displacement of the last trailer goes beyond the final, or steady-state, position during the lane-change maneuver. For obstacle-avoidance maneuvers of three seconds duration and longer (i.e., those in the more normal operating range), the caster-steered B-dolly combination overshoots its final position by substantially more than the other combinations. We see, for example, that the pup trailer on this vehicle shows more than a one meter overshoot in a four-second lane change.

In general, it can be said that excessively compliant self-steering systems installed on the dolly axle will increase the vehicle's rearward amplification response and lower the frequency at which peak amplification occurs. An intermediate level of steering compliance is represented in Figure 4.12, in which the linear frequency response of a B-dolly having a restoring stiffness approximately equal to the total axle cornering stiffness is compared to the response of the B-train and a vehicle outfitted with a caster-steered B-dolly. We note that even this intermediate (equal to axle cornering stiffness) case yields an increased amplification ratio with respect to the B-train.

The friction characteristic of the self-steering system has a more complex influence on the vehicle's lane-change performance. Figure 4.13, for example, shows time histories for the B-train and for two B-dolly systems having the intermediate levels of friction as introduced before. The three simulated responses are very similar despite the presence of measurable self-steer angles for both of the lower friction cases. The rearward amplification for the three vehicles is almost identical for this three-second lane change. The higher friction system (where the force needed to overcome the steering friction is 20% of axle load) actually has slightly less amplification than the B-train. We see that the steering system is moved out of the friction zone during the initiation phase of the maneuver and becomes "stuck" at .4 degrees of

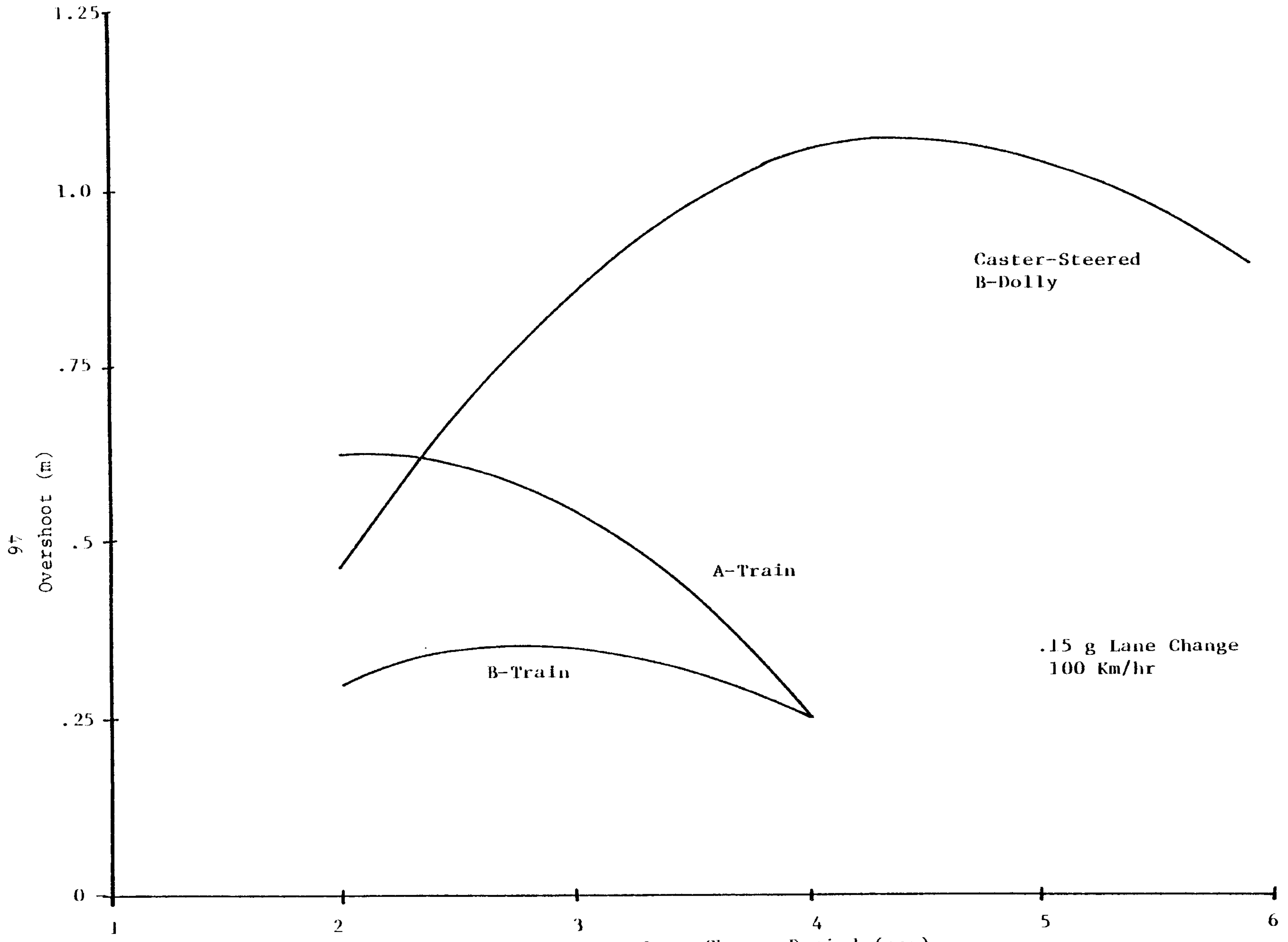


Figure 4.11. Lateral Displacement Overshoot During Closed-Loop Lane-Change Maneuvers, as a Function of the Nominal Steer Input Period of the Lane Change.

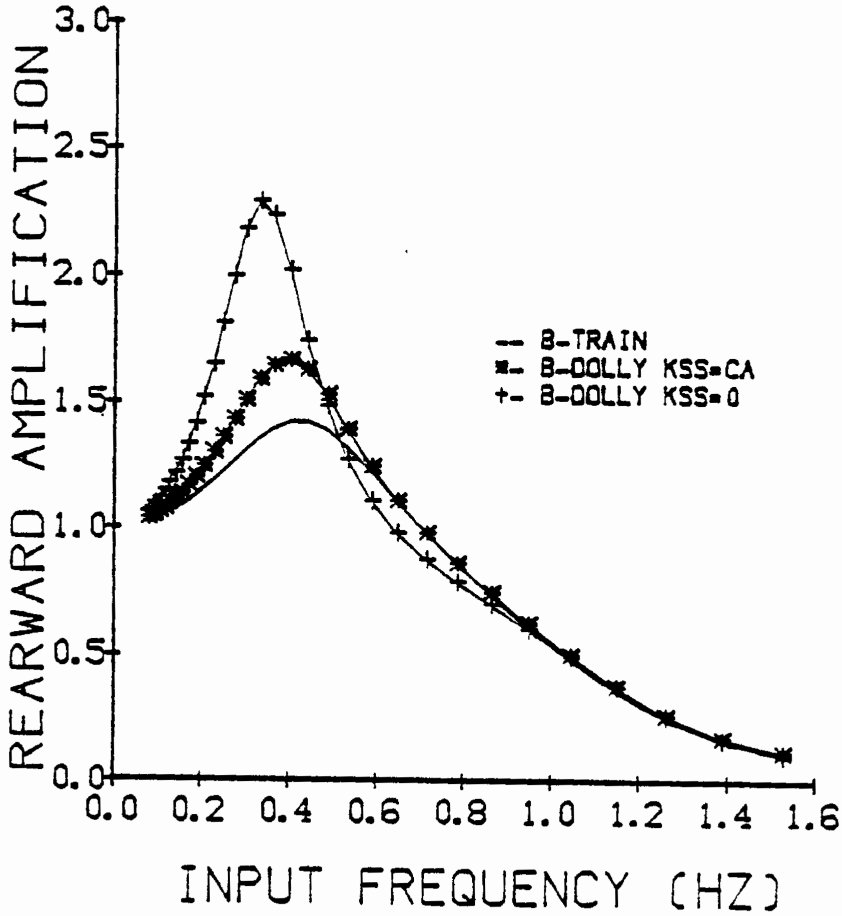


Figure 4.12. Linear Model Rearward Amplification for Self-Steering B-Dollies.

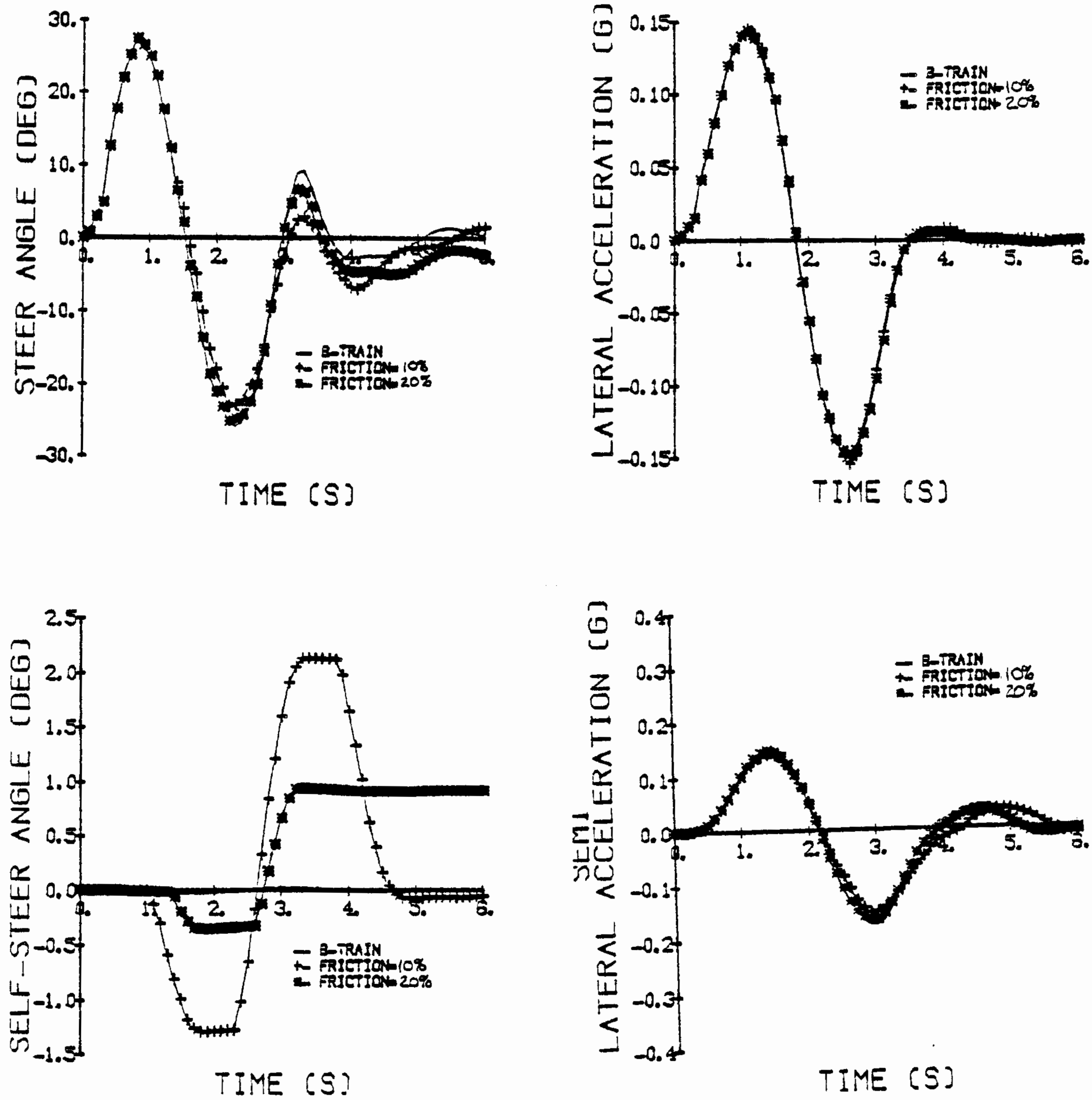


Figure 4.13. Influence of Friction in the Dolly Steering System on Lane-Change Response.

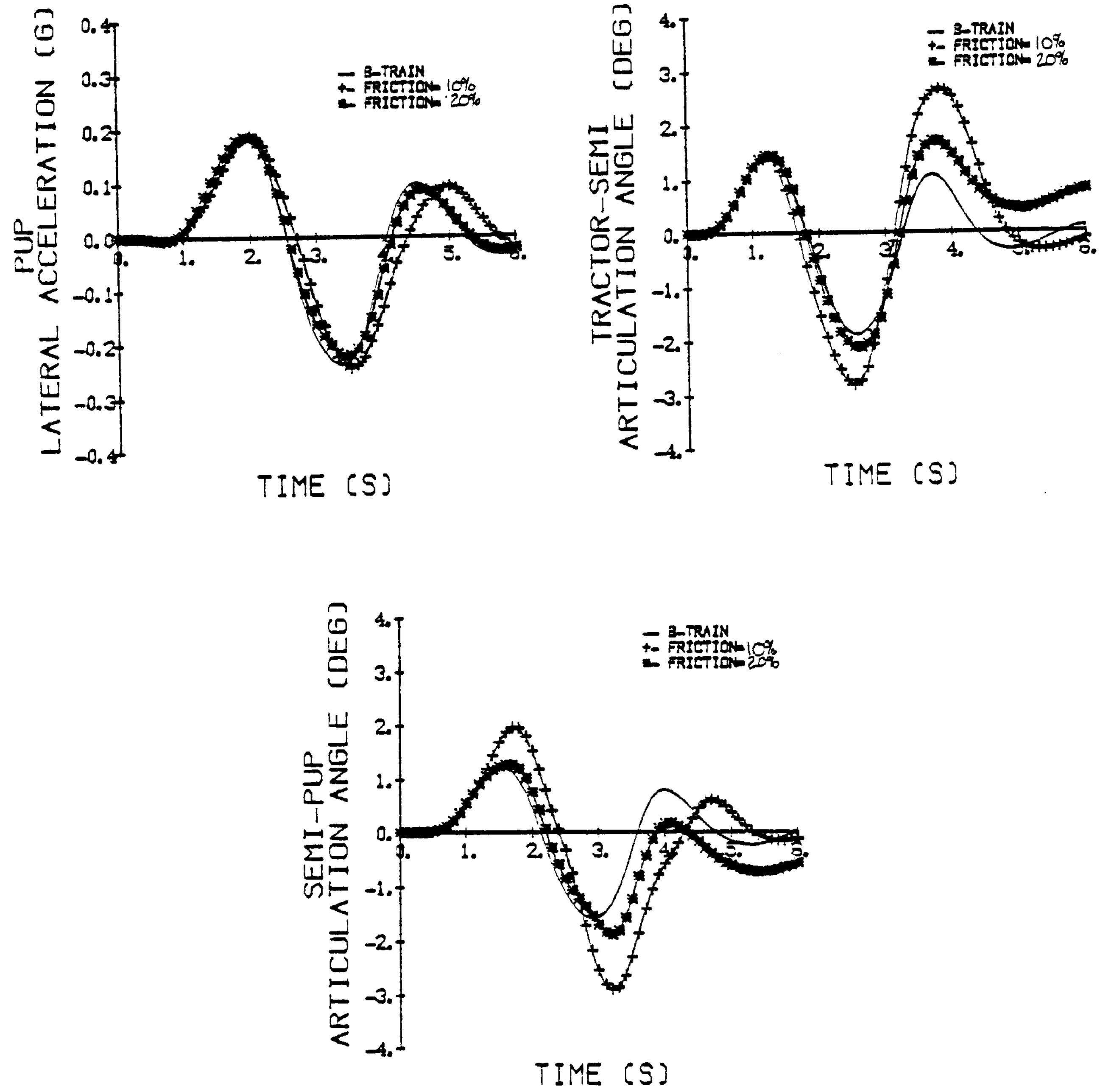


Figure 4.13 (Cont.)

self-steer angle as the lateral force is reduced and reversed. This steer angle serves to increase the lateral force opposing the motion of the semi-dolly unit during the recovery phase of the lane change, serving to reduce the second acceleration peak and thus, the rearward amplification. The friction is broken again during the recovery portion of the maneuver such that the dolly wheels steer in the opposite direction about .9 degrees. The self-steering angle becomes stuck at this angle and the vehicle exits the lane change with the semi-dolly unit "dog-tracking" relative to the tractor and pup trailer. The lower friction case (i.e., lateral force equivalent to 10% of axle load) exhibits nearly identical behavior, although the tires on the dolly axle take on larger steer angles (1.2-2.2 degrees). In this case, the period during which the self-steering system is stuck in the friction is reduced. Nevertheless, the 10% friction level is found to be sufficient to keep the rearward amplification level close to that of the B-train.

For the levels of coulomb friction examined here, the friction in the self-steering system is seen to have a minor effect on the vehicle performance during lane-change-like maneuvers. Nevertheless, a friction level approaching zero will result in the system acting simply as a compliant steering system, with the previously-described disadvantages. Extremely high friction levels assure that the vehicle combination reacts similarly to a B-train. Intermediate friction levels can result in behavior approaching that of a B-train, although this result depends on the interaction of the self-steering system friction and compliance with the motions of the vehicle unit to which it is attached.

4.2 Pintle Hitch Loading

Pintle hooks and lunette eyes used on commercial vehicles normally transmit loads only in the longitudinal direction. The concept of a B-type converter dolly changes the hitch load possibilities markedly since the dolly transmits roll and yaw moments between semi and pup trailers. Indeed, the dual pintle hitches of the B-dolly experience reaction forces in all three directions. Lateral forces developed at the dolly's fifth wheel and at its tires act to load the hitching system with a lateral force and yawing moment. The lateral force produces a lateral shear to the pintle hitches, while the yaw moment is reacted as longitudinal tension and compression forces at the left- and right-hand hitches. Roll moments transmitted between dolly and lead trailer produce vertical shear forces at the hitches. In conventional (A-type) dolly applications, the hitch is required to carry very small vertical loads due to the proximity of the dolly's fifth wheel to the axle centerline In fact, only during braking does the A-type pintle experience substantial vertical loads, and then the reaction load applied to the pintle hook is oriented downwards.

The most critical loading of the hitch in B-dolly installations is thought to involve the upward-oriented shear load that arises from the generation of a roll couple on the front connections to the dolly. The downward vertical loads imposed on the hitch are applied in the direction of the bottom of the hook, which is a fairly strong section, while the upward load is of special concern because it is borne by the latching mechanism. The ability of the hitch to remain latched under the application of this upward force is seen as the potentially critical factor.

In order to obtain an estimate of the maximum hitch forces which might prevail, a high-center-of-gravity version of the self-steering B-dolly train was simulated executing maneuvers approaching rollover. The center-of-gravity height of the trailers was placed at 2.8 m (110 in) above the ground to simulate the highest payload placement which is thought to occur in normal commercial service. This maximum reasonable c.g. height is used to generate the peak maximum value of roll moment during a lane-change maneuver. The lane-change maneuver produces a large differential roll moment due to the out-of-phase lateral acceleration responses of the two trailers during the transient maneuver.

The B-dolly train using the ASTL steering system model was simulated in a lane-change maneuver which produced a 0.25 g level of lateral acceleration at the tractor, for a steady speed of 100 Km/hr. During the recovery phase of this maneuver, the pup axle lifts and the roll couple across the hitch reaches its maximum value—approximately 51,900 N-m (450,000 in-lb). This causes an upward loading of 66.7 kN (15,000 lb) at one pintle hook and an equal downward loading of the other. The longitudinal and lateral loadings of each hitch in this near-limit maneuver are 20.5 kN and 5.3 kN, respectively. The most serious loading is the vertical load applied to the pintle hook mechanism. Special provisions must be made to assure that the hitch can transmit this load without failure.

4.3 Influence of Increased Dolly Length

To examine the implications of lengthening the B-type converter dolly, an increase of the dolly length from 2 m (80 in) to 3 m (120 in) was examined to assess its effect on the low- and high-speed performance of the caster-steered B-dolly and the locked-steering B-dolly (B-train). For the locked-steering vehicle, the increase in dolly length is analogous to an increase in wheelbase of the semi-dolly unit. The result of this increase in effective wheelbase is that the yaw stiffness of the unit is increased, and the oscillatory motion of the unit becomes more damped. The spread of the axles is increased as well and this brings with it a larger resistive moment which is inversely proportional to the turn radius. In the case of the caster-steered B-dolly combination, a longer dolly causes the force at the dolly fifth wheel to be applied at a longer moment arm from the c.g. of the semitrailer such that a greater lateral force is needed at the tires on the semitrailer axle in order to achieve force and moment equilibrium.

The long dolly combinations are compared to the standard vehicles for low-speed offtracking in Figure 4.14. The low-speed offtracking on a 20 m radius curve at 10 Km/hr is seen to be reduced for the caster-steered case, in comparison with the behavior exhibited with the shorter tongue. Clearly, with caster steering the longer dolly tongue constitutes the equivalent of an extended "stinger" which causes the dolly fifth wheel to track at a larger radius, with a lower total value of low-speed offtracking. In the case of a locked-steering dolly, the longer effective wheelbase implies, and results in, a greater amount of low-speed offtracking for the fixed-steering vehicle.

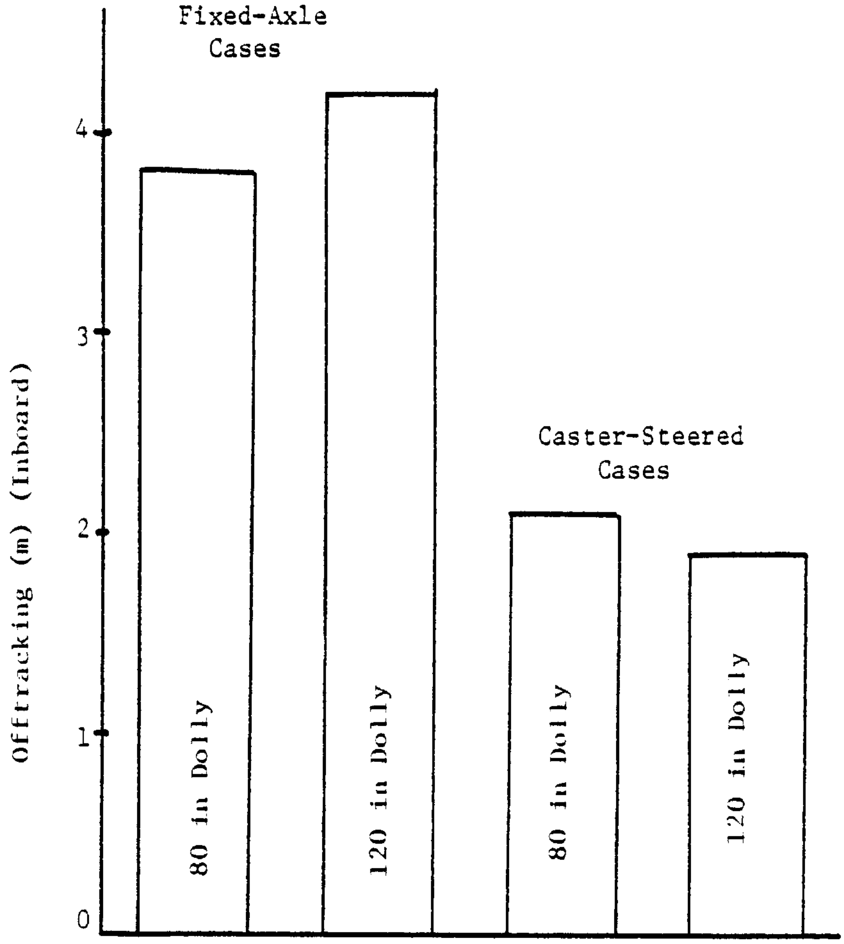


Figure 4.14. Dolly Length Influence on Low-Speed Offtracking - 10 Km/hr, 20 m Radius.

For the long dolly, caster-steered case, the high-speed offtracking is increased over that of the shorter dolly, as shown in Figure 4.15. At 90 Km/hr (56.3 mph) on a 260 m (853 ft) radius curve, the long dolly, caster-steered, vehicle offtracks 1.75 m to the outside of the turn. Also at 90 Km/hr, increasing the dolly length increases the high-speed offtracking of the fixed-steering (B-dolly locked) combination, with the outboard offtracking dimension increasing from .6 to .78 m. Below 75 Km/hr, this trend involving the B-dolly locked case is reversed such that the long dolly version exhibits marginally less offtracking than the standard dolly.

The rearward amplification obtained with the increased dolly length is only slightly different from that of the shorter dolly cases. For the fixed-steering case, the influence of dolly length is negligible. With the caster-steering version, the longer dolly results in a slightly increased rearward amplification. Figure 4.16 shows the rearward amplification of the vehicles as a function of input frequency. These data are from the linear yaw plane model and are used only as a rough indicator of the actual vehicle performance of the combinations.

Moreover, the influence of increased dolly length is not of great significance to the dynamic performance of the combination in the maneuvers examined here. Increased dolly length may change the loading of the pintle hitches, however, since the yaw moment resulting from tire and fifth wheel forces will be increased.

4.4 Triple-Trailer Combinations

It has been established [2] that triple-trailer combinations exhibit a greater level of rearward amplification than A-train doubles. Thus, the B-type converter dolly was seen as especially attractive as a possible means for controlling the rearward amplification tendencies of the triple. In this study, the linear yaw plane model was used to evaluate B-dolly installation in triple-trailer trains. The predicted rearward amplification of B-dolly triples with both locked and compliant self-steering systems are shown in Figure 4.17. The compliant steering system represented here has a stiffness level approximating

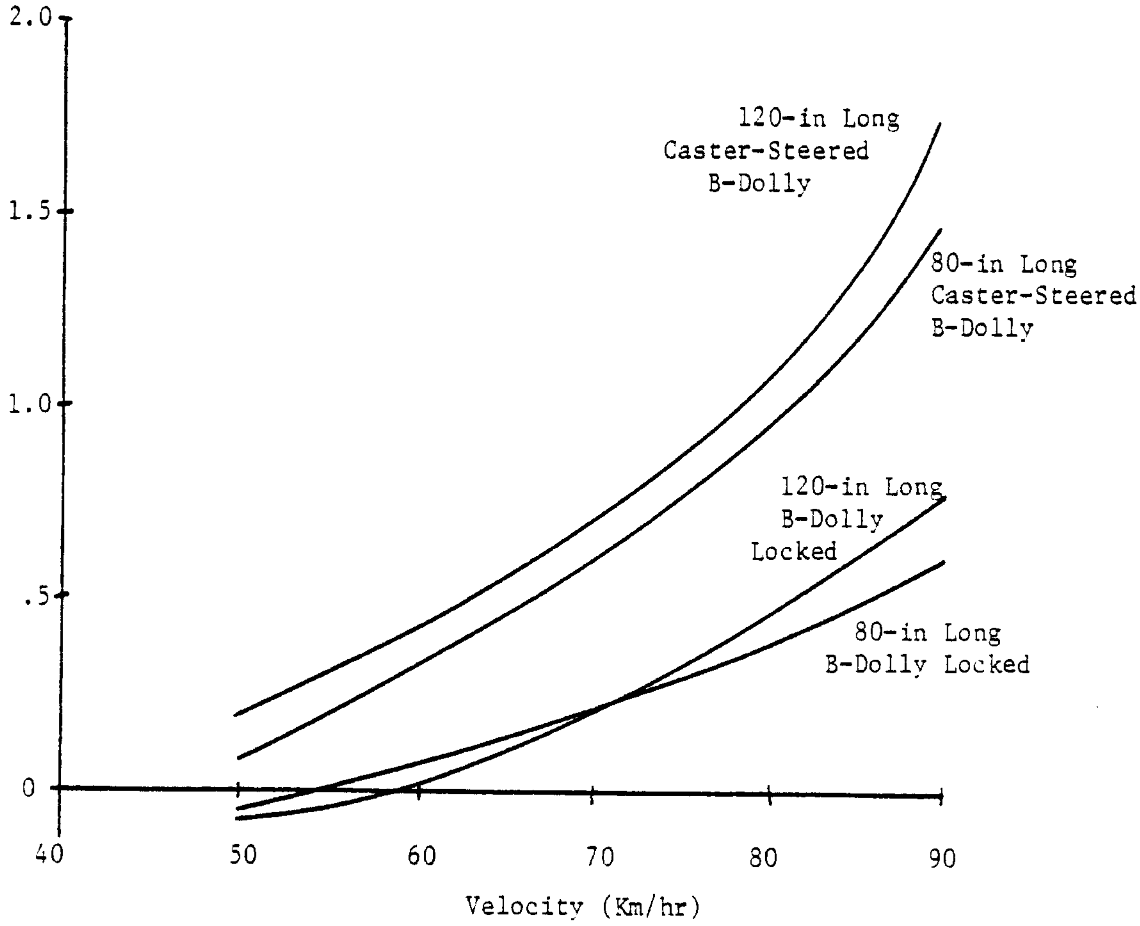


Figure 4.15. Influence of Drawbar Length on High-Speed Offtracking.

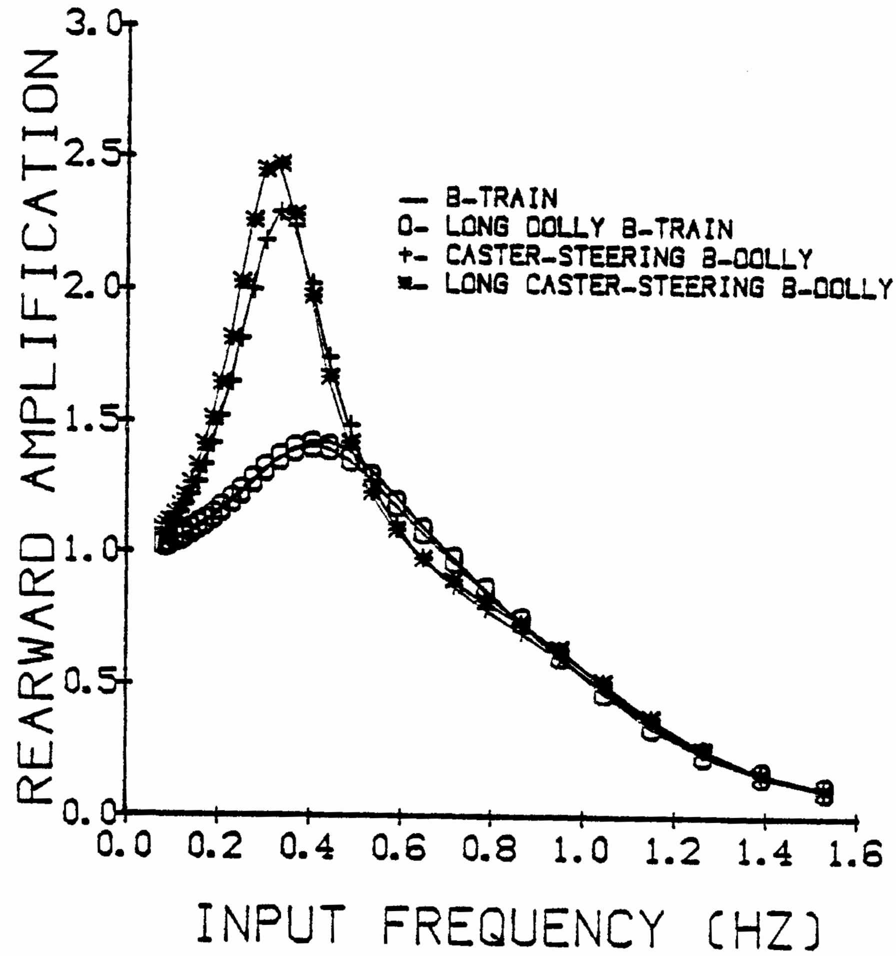


Figure 4.16. Influence of Drawbar Length on Rearward Amplification.

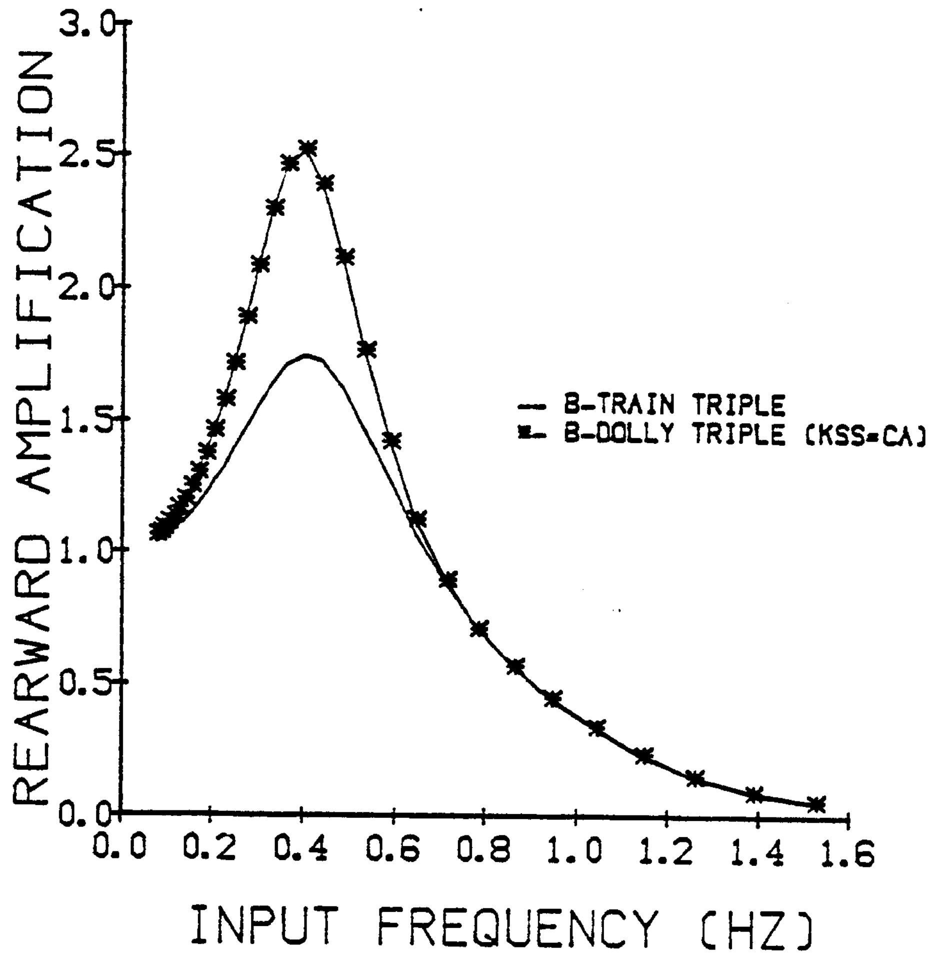


Figure 4.17. Rearward Amplification Frequency Response of Triple-Trailer Combinations - Linear Model.

the cornering stiffness of the dolly axle. The caster-steered case is not illustrated due to the very large level of amplification (approximately equal to 9.5!) observed with this combination. This level of amplification is so high that the equipped vehicle would be expected to suffer rollover in response to even the minor steering perturbations occurring in normal driving. By way of contrast, an A-train triple (not shown in the figure) produces a rearward amplification level of approximately 3.6 at 100 Km/hr, while negotiating a two-second lane-change maneuver. A similar triple coupled with B-dollies having the "locked" type of steering behavior produces an amplification level of 1.75—less than half that of the A-train triple. Halving the "apparent cornering stiffness" of the dolly axle by introducing steering compliance at the B-dollies results in an increase in rearward amplification level to 2.5.

Moreover, we can conclude from these results that the application of B-type converter dollies, particularly those with an effectively rigid steering function, offers a significant improvement in the amplification performance of triple-trailer trains.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

The results of this study have led to four basic conclusions concerning the B-type converter dolly:

1) High-speed performance can be equivalent to that of a B-train

2) High-speed performance characteristics are strongly influenced by the friction and stiffness properties of the B-dolly's self-steering system

3) Low-speed offtracking performance can exceed that of an A-train

4) The loading of the pintle hitches of the dolly can be very high compared to that normally accruing in conventional dolly applications

The use of a B-type dolly in place of a conventional A-type dolly has the immediate effect, regardless of self-steering system properties, of coupling the rearmost trailer to the rest of the train in roll. This benefits the rollover immunity of the train, especially during transient maneuvers, by allowing the units of the vehicle to share the reaction of roll moment between their respective suspensions. This tends to reduce roll angles accrued during maneuvering and therefore the potential for rollover of an individual trailing unit.

If the self-steering system is held fixed by a locking mechanism or by friction, the resulting vehicle is further stabilized by a reduction in rearward amplification relative to an A-coupled train. This configuration is essentially a B-train with the attendant low rearward amplification. Reduction in the effective stiffness of the self-steering system, by reducing either friction or the stiffness of the centering spring, may cause a degradation in the vehicle's performance at speed. With a decrease in the stiffness of the steering system comes an increase in high-speed offtracking and rearward amplification. The increased amplification was also seen to be accompanied by a reduction in the frequency at which peak amplification occurs. In a limit case, involving a B-dolly steering system with zero stiffness and zero friction, the peak amplification can occur at low enough frequencies to approach the realm of normal maneuvering.

The most desirable set of self-steering system parameters are those that prevent the tires on the dolly axle from steering during the large-radius maneuvers conducted at highway speeds, while otherwise allowing steering during short-radius maneuvers around intersections and the like. Such parameters will assure that low levels of high-speed offtracking and rearward amplification are obtained in double- and triple-trailer combinations.

For low-speed maneuvering, the self-steered B-dolly can actually outperform the offtracking behavior of the A-train if sufficiently low values of friction and centering stiffness are incorporated within the B-dolly's steering system. Such low stiffness and friction levels may directly oppose the needs of high-speed maneuvering, however, such that a compromised design may be warranted. It should be recognized, however, that to sacrifice high-speed stability for a gain in low-speed maneuverability is to compromise the interests of traffic safety in favor of economics (vis-a-vis tire wear) and operational convenience. One "uncompromised" solution to this conflict in dolly requirements is to engage some form of locking mechanism to prevent steering at high speed, while unlocking the device at low speed so as to keep tire scrubbing to a minimum and to improve low-speed maneuverability.

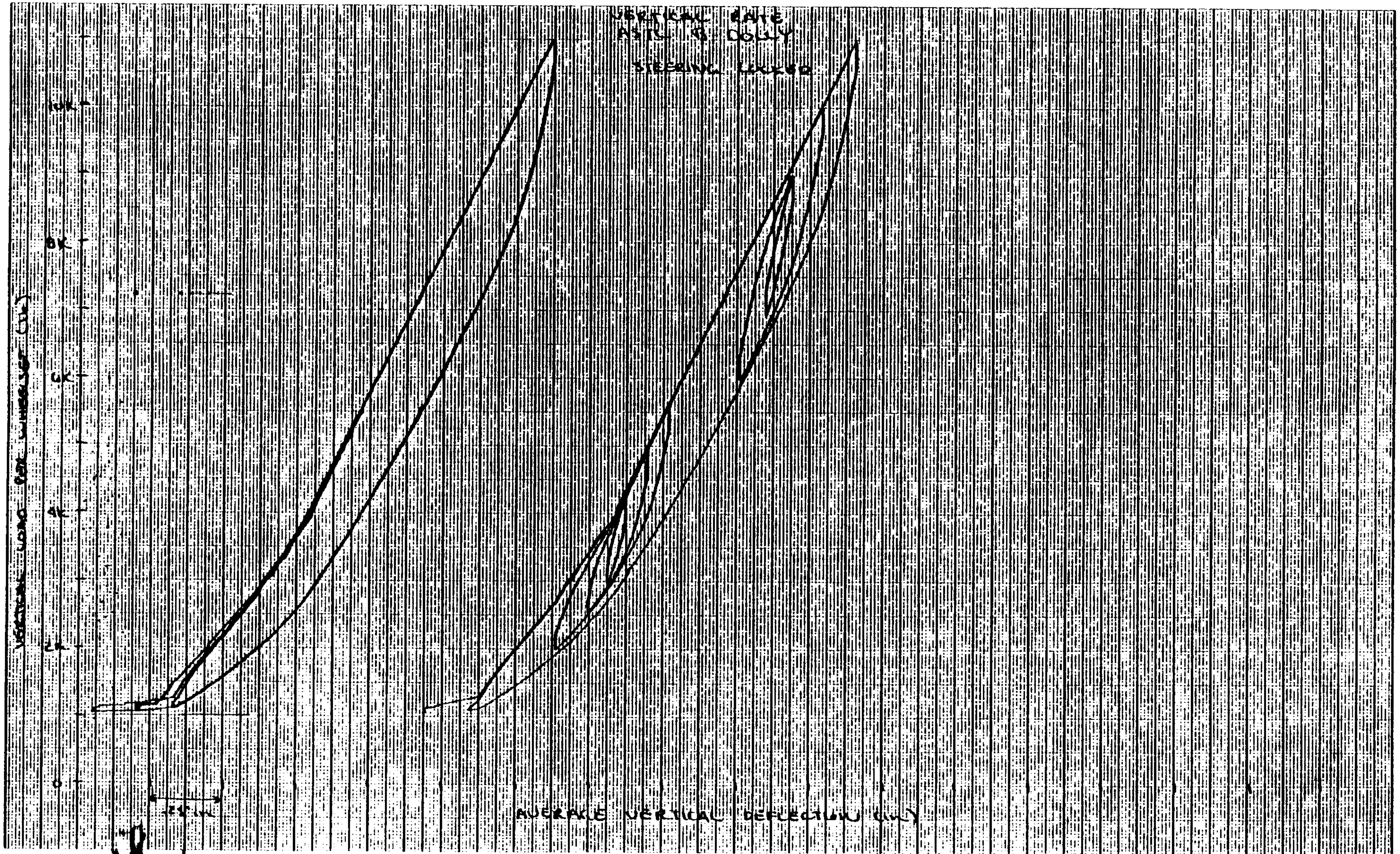
A specific B-dolly whose properties were measured in this study provided a reasonable compromise in low- versus high-speed properties by means of a large, but not excessive, amount of coulomb friction in the self-steer system. While the resulting stability and control properties of this dolly were virtually as good as those of a B-train, it was noted that the retention of these properties over time depends entirely upon the stationarity of the friction feature.

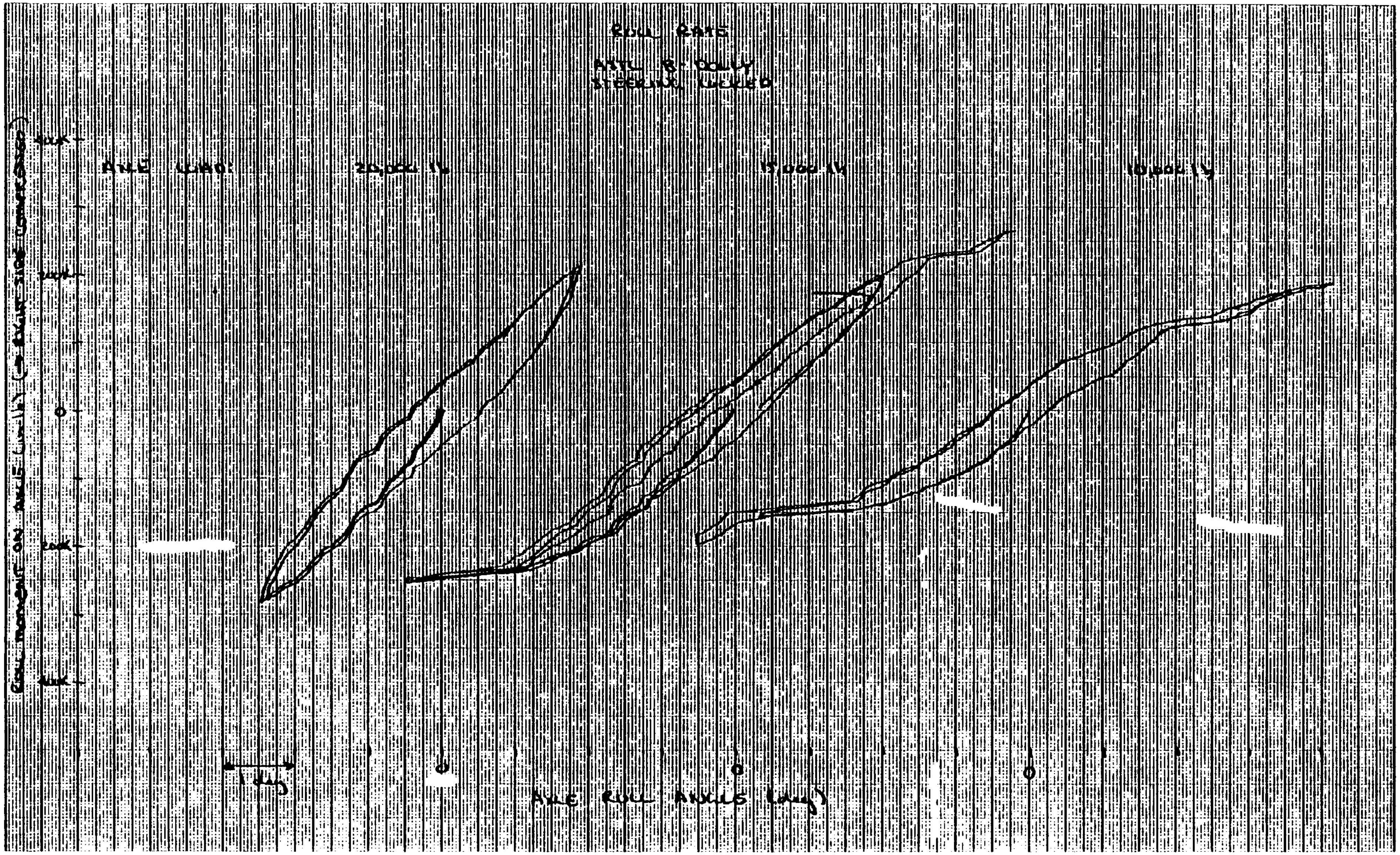
The hitch loading arising from B-dolly installations is an area of concern. It was found that the pintle hitches attaching the dolly to the preceding trailer can be subjected to very high loads in the direction tending to "unlatch" the hitching mechanism. Existing and future hitching mechanisms intended for B-dolly applications should be examined to determine whether they can reliably withstand this kind of loading. The threat of one lunette eye becoming unhitched from the leading trailer is obvious and is presumed to pose a dangerous condition.

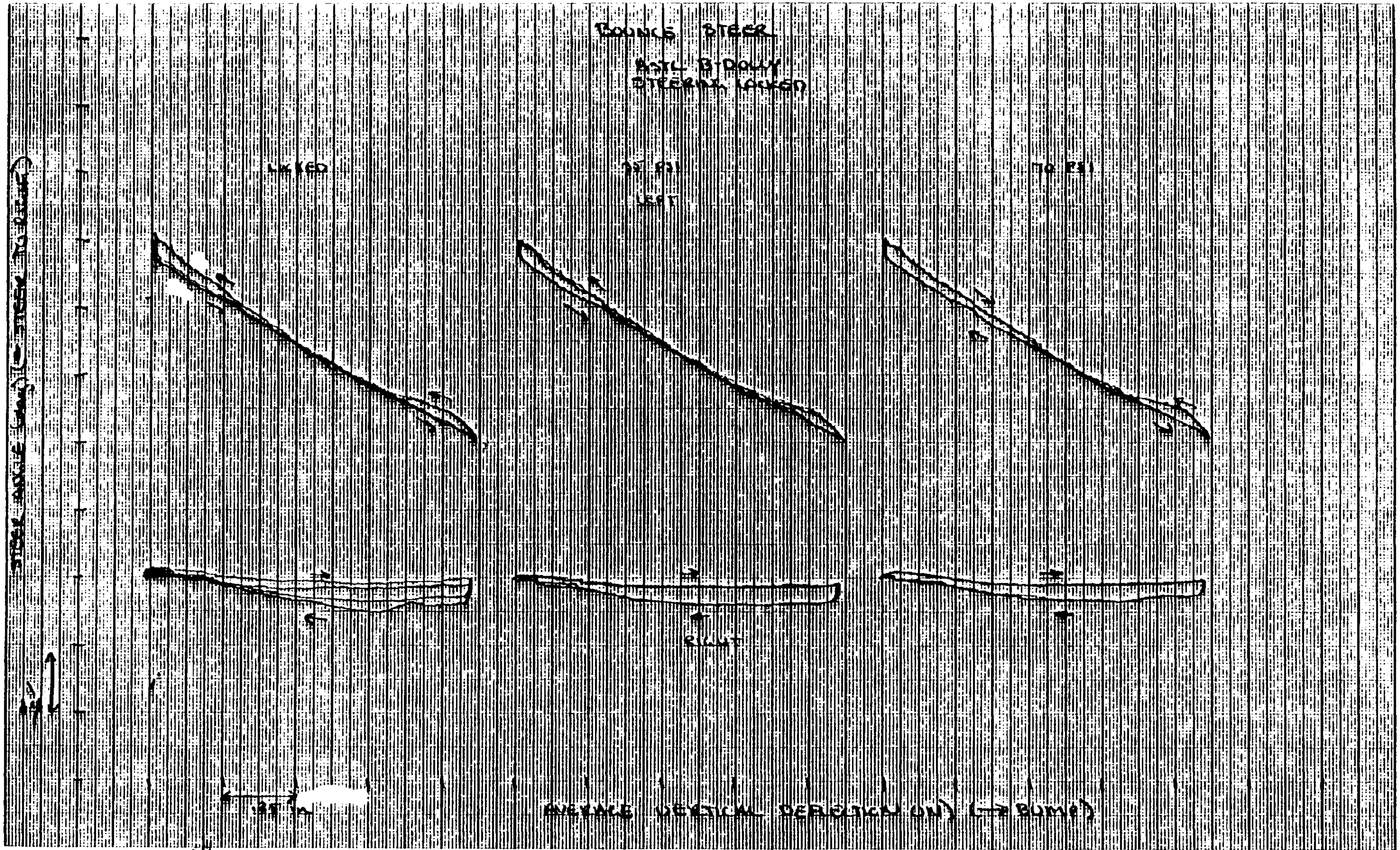
It is recommended that future B-dolly designs be scrutinized carefully to avoid the placing of an unsuitable product on the market. The major criteria of design suitability, from a safety point of view, are hitching strength and the minimization of the self-steering function during high-speed operation. With these two criteria satisfied, the B-dolly presents a very appealing alternative to the A-dolly in terms of dynamic performance, combining the active safety of the B-train with the coupling convenience and fleet versatility of a conventional A-train.

APPENDIX A

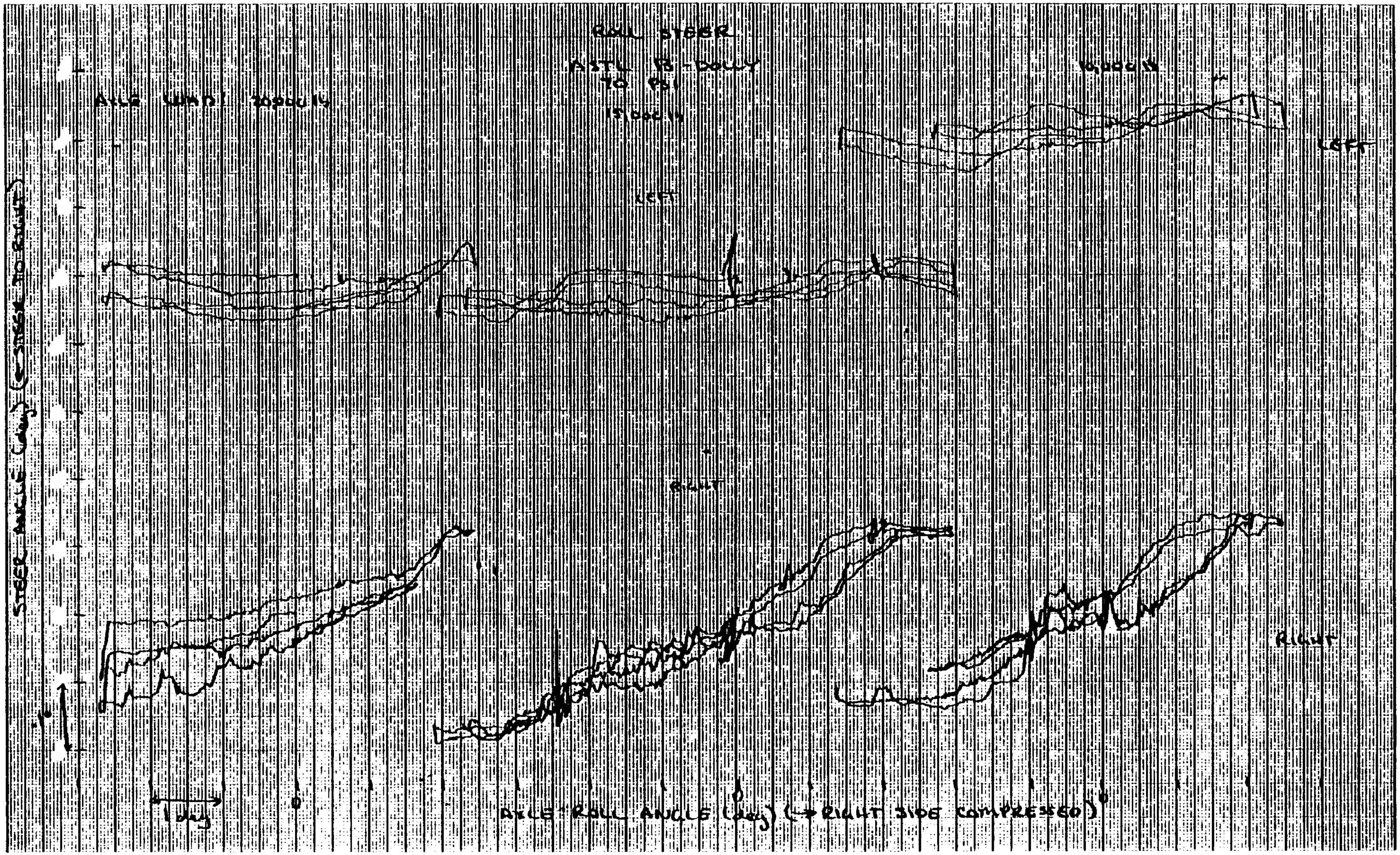
DETAILED SUSPENSION PARAMETER MEASUREMENT RESULTS

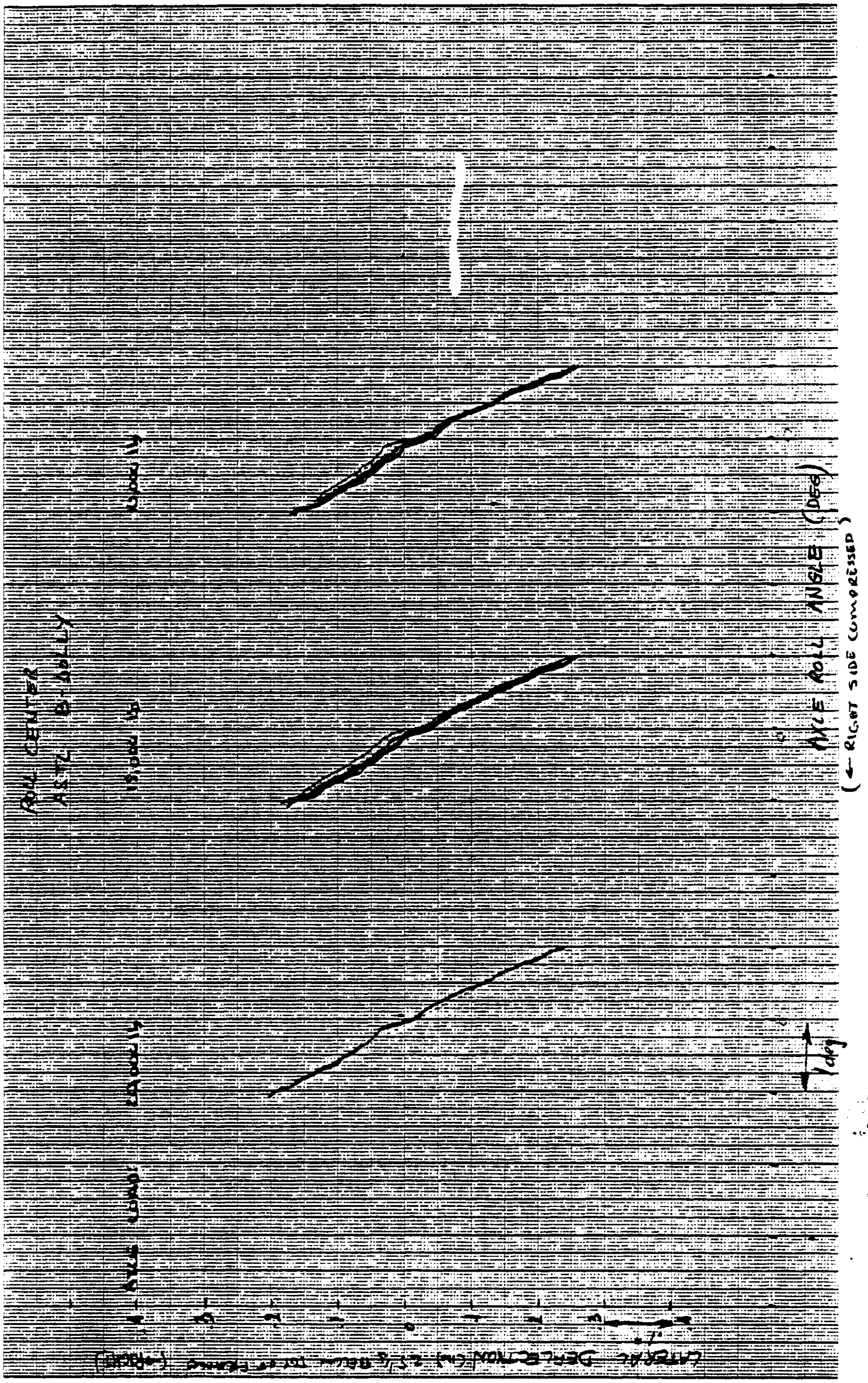

AVERAGE VERTICAL DEFLECTION (in.)
10K
8K
6K
4K
2K
0


ROLL RATE
AXLE 8: DOLLY
STEERING LOCKED
AXLE LOAD:
20,000 lb
15,000 lb
10,000 lb
0
0
0
0
1 deg
AXLE ROLL ANGLE (deg)


BOUNCE STEER
B-DOLLY
STEERING LOCKED
LOCKED
35 PSI
LEFT
70 PSI
RIGHT
STEER ANGLE (DEG) (← STEER TO RIGHT)
.25 in
AVERAGE VERTICAL DEFLECTION (IN) (→ BUMP)

47 1512


ROLL STEER
AXLE 13 - DOLLY
70 PSI
AXLE LOAD: 20,000 lb
15,000 lb
10,000 lb
LEFT
LEFT
RIGHT
RIGHT
STEER ANGLE (deg) (→ STEER TO RIGHT)
1 deg
0
AXLE ROLL ANGLE (deg) (→ RIGHT SIDE COMPRESSED)



ROLL CENTER
ASTL B-DOLLY
AXLE LOAD:
20,000 lb
15,000 lb
10,000 lb
1 deg
AXLE ROLL ANGLE (DEG)
(← RIGHT SIDE COMPRESSED)
LATERAL DEFLECTION (in)

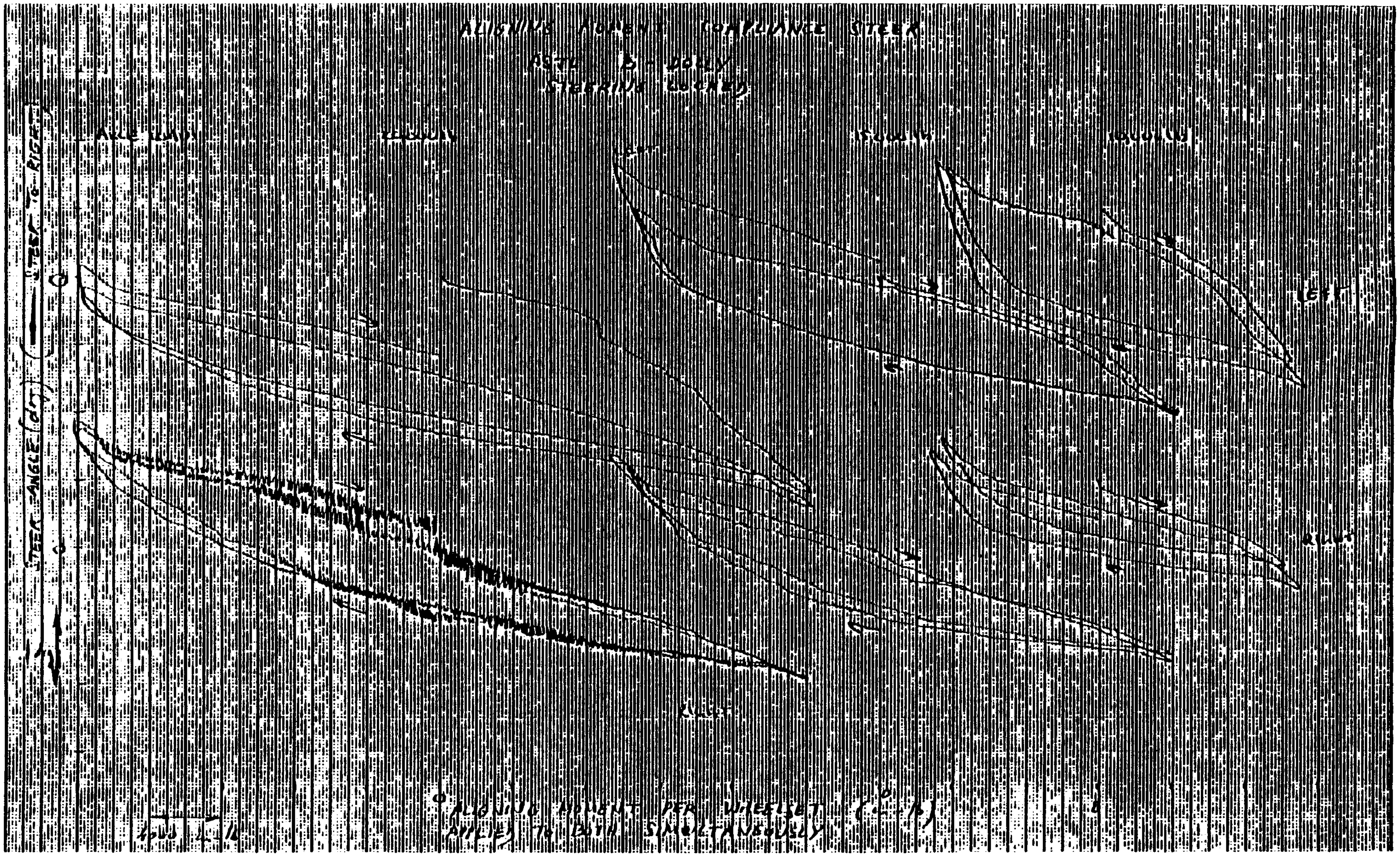
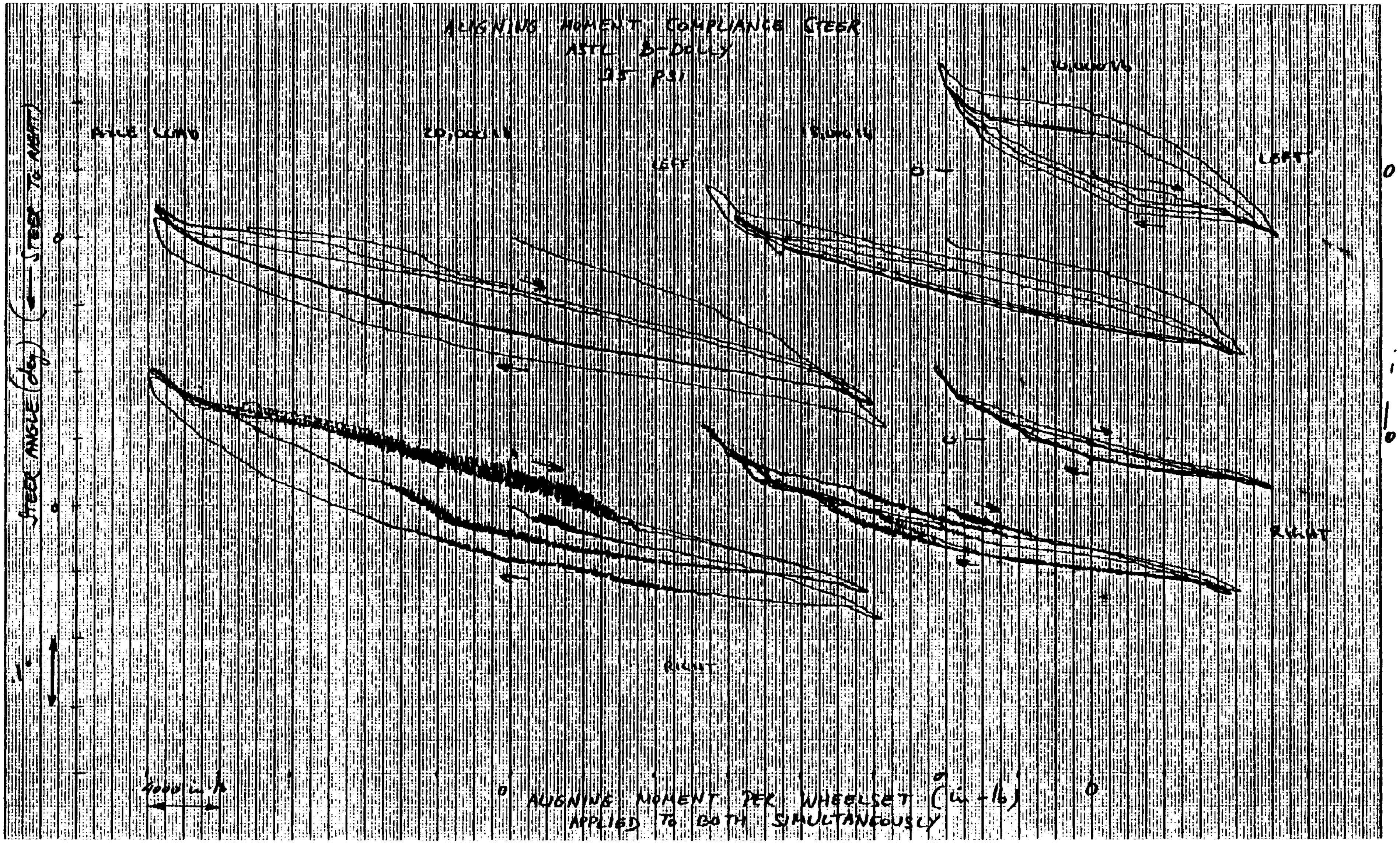
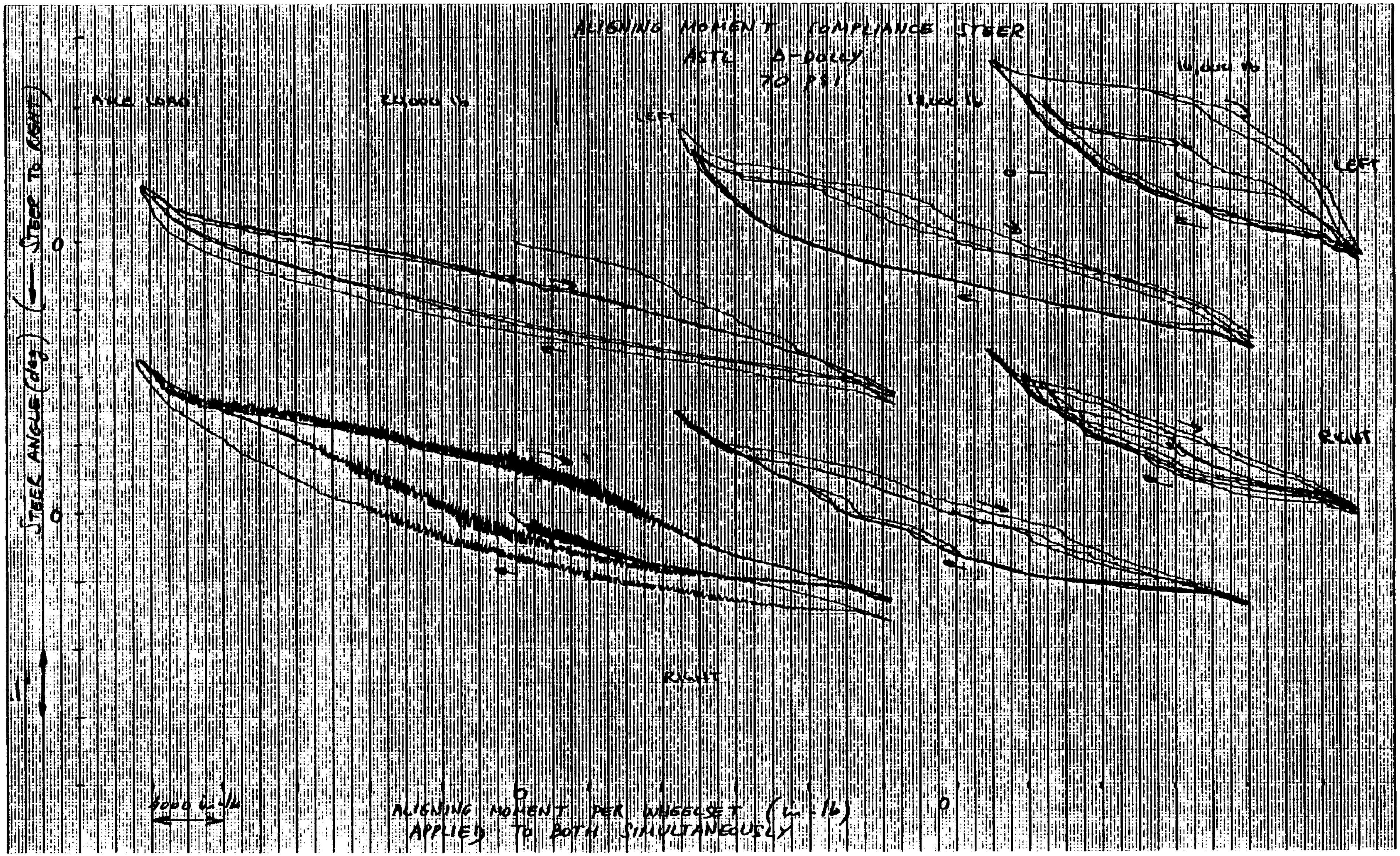
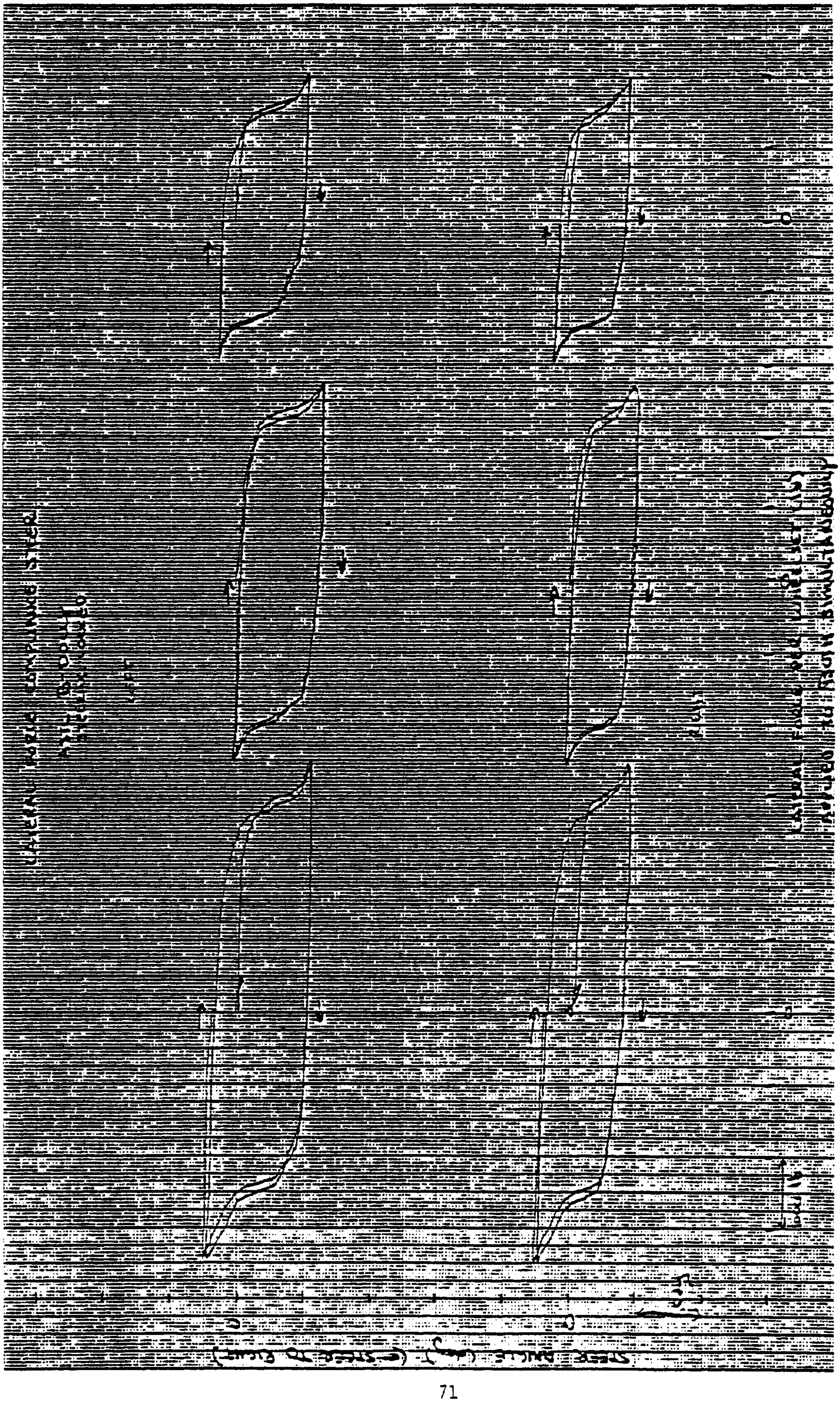
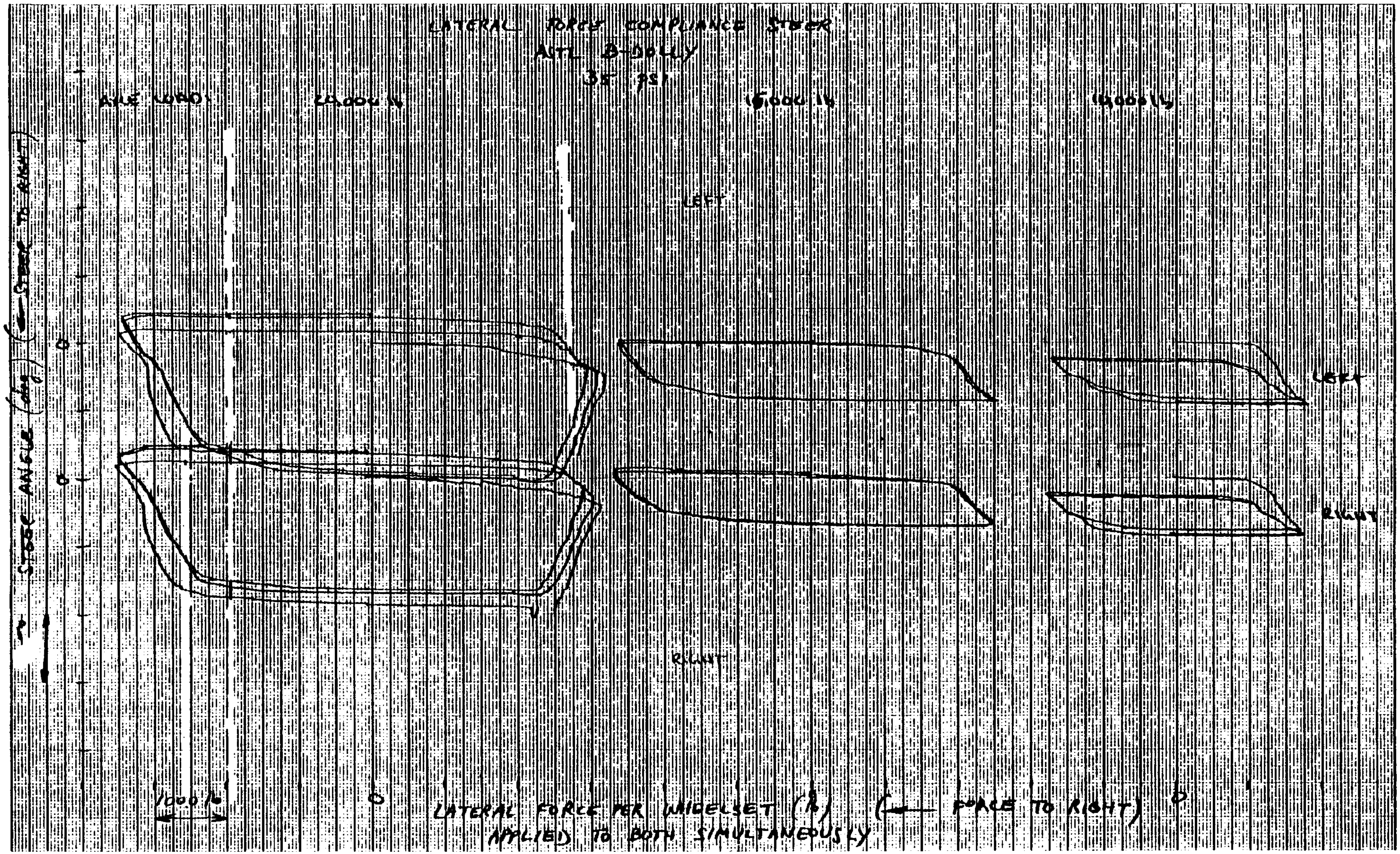

ALIGNING MOMENT COMPLIANCE STEER
B-BODY
STEERING LOCKED
LEFT
RIGHT
ALIGNING MOMENT PER WHEELSET
APPLIED TO BOTH SIMULTANEOUSLY
STEER ANGLE
STEER TO RIGHT

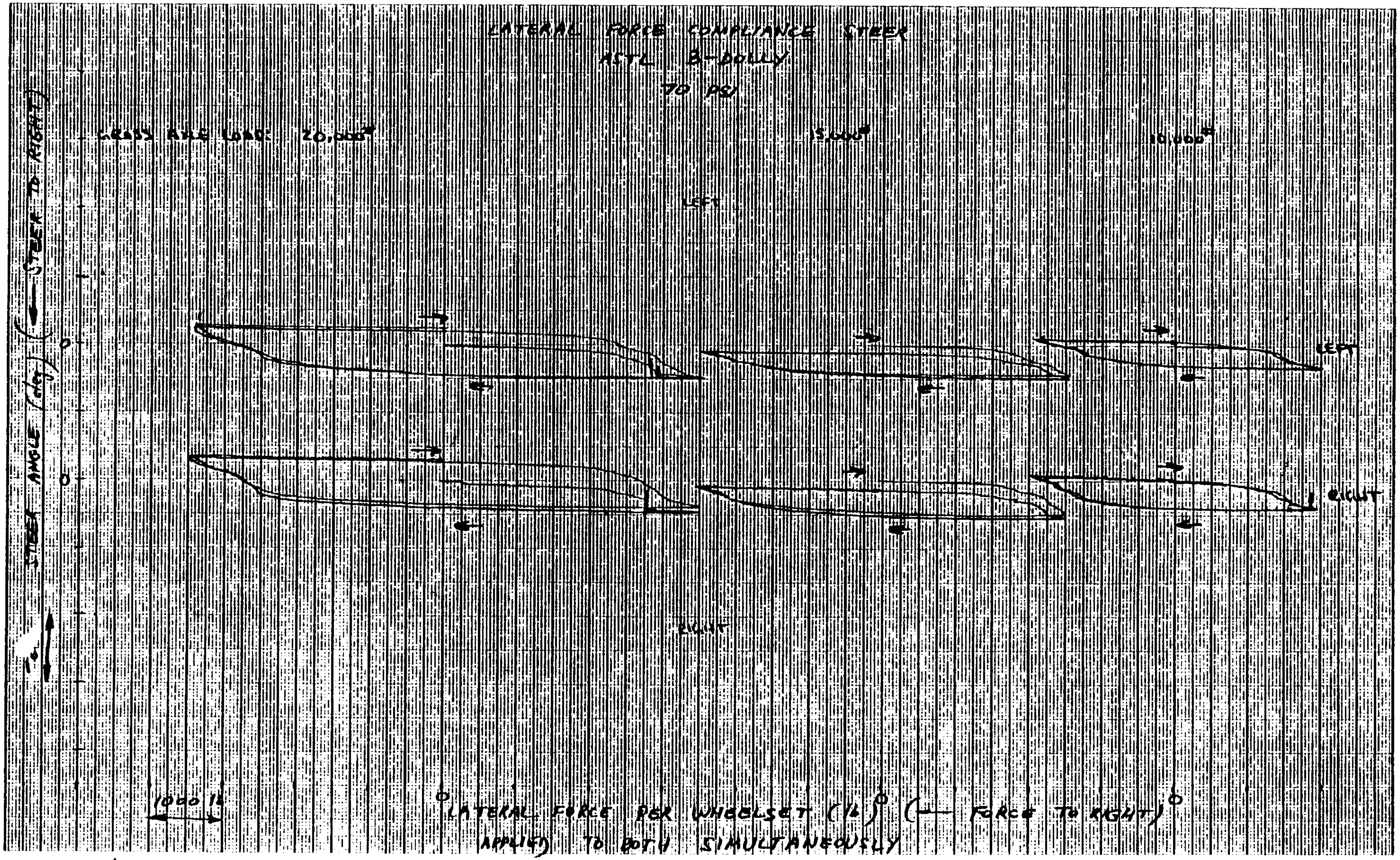
Fig 2.5

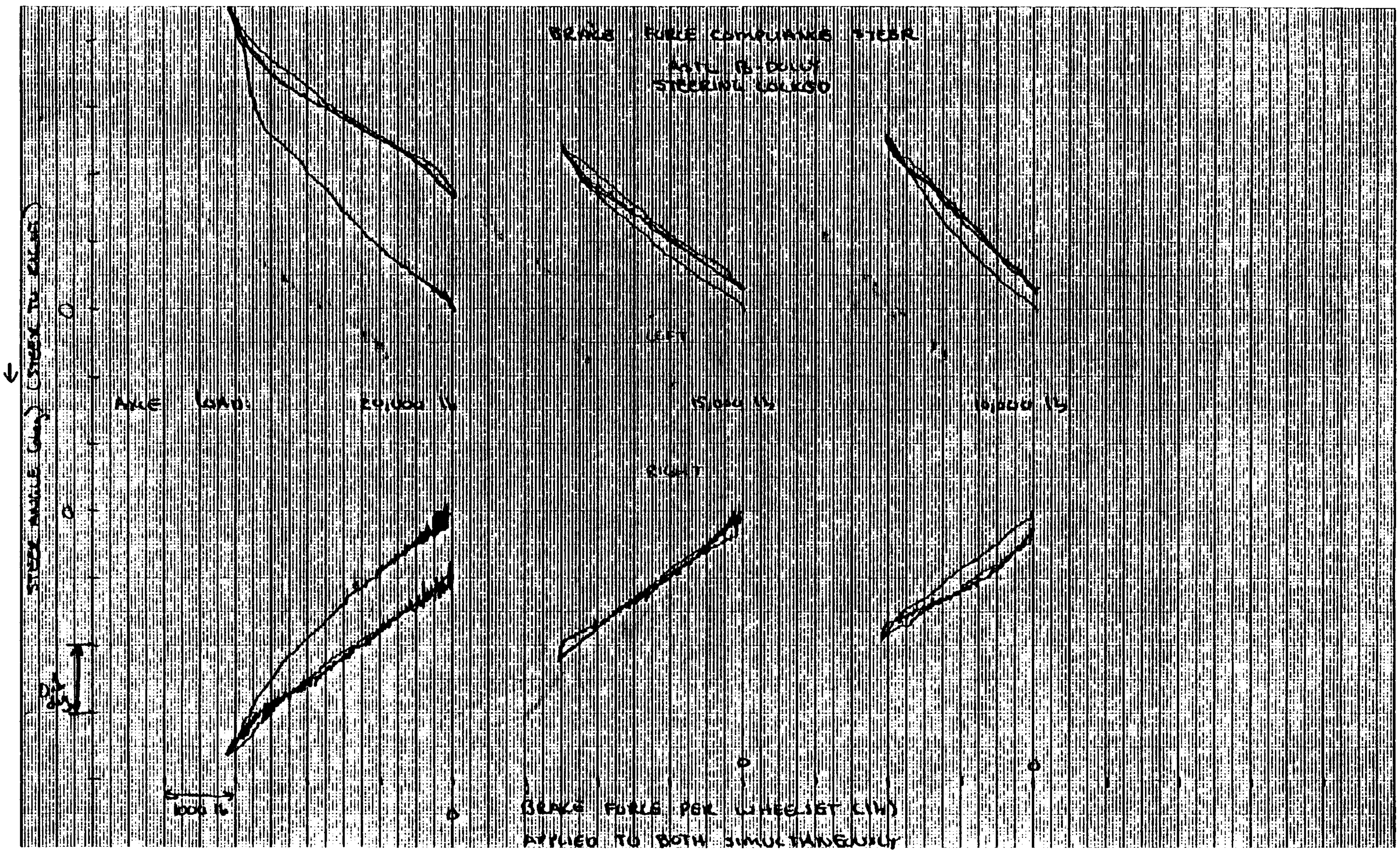

ALIGNING MOMENT COMPLIANCE STEER
ASTL B-DOLLY
35 PSI
AXLE LOAD
20,000 lb
15,000 lb
LEFT
RIGHT
LEFT
RIGHT
STEER ANGLE (deg) (← STEER TO RIGHT)
4000 in-lb
ALIGNING MOMENT PER WHEELSET (in-lb)
APPLIED TO BOTH SIMULTANEOUSLY

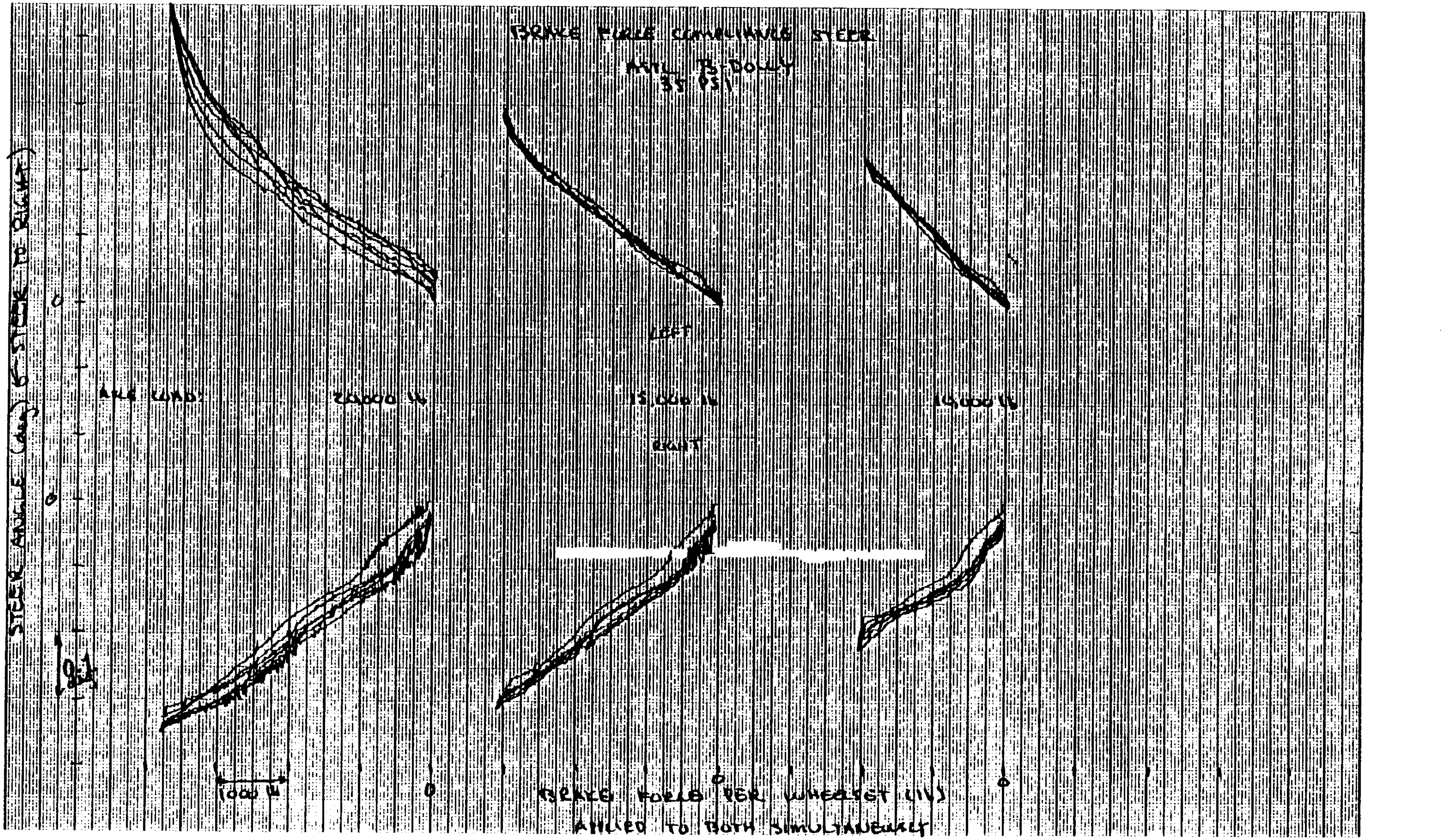

ALIGNING MOMENT COMPLIANCE STEER
ASTL A-DOLLY
70 PSI
AXLE LOAD
LEFT
RIGHT
LEFT
RIGHT
STEER ANGLE (deg) (← STEER TO RIGHT)
0
0
4000 in-lb
0
0
ALIGNING MOMENT PER WHEELSET (in-lb)
APPLIED TO BOTH SIMULTANEOUSLY

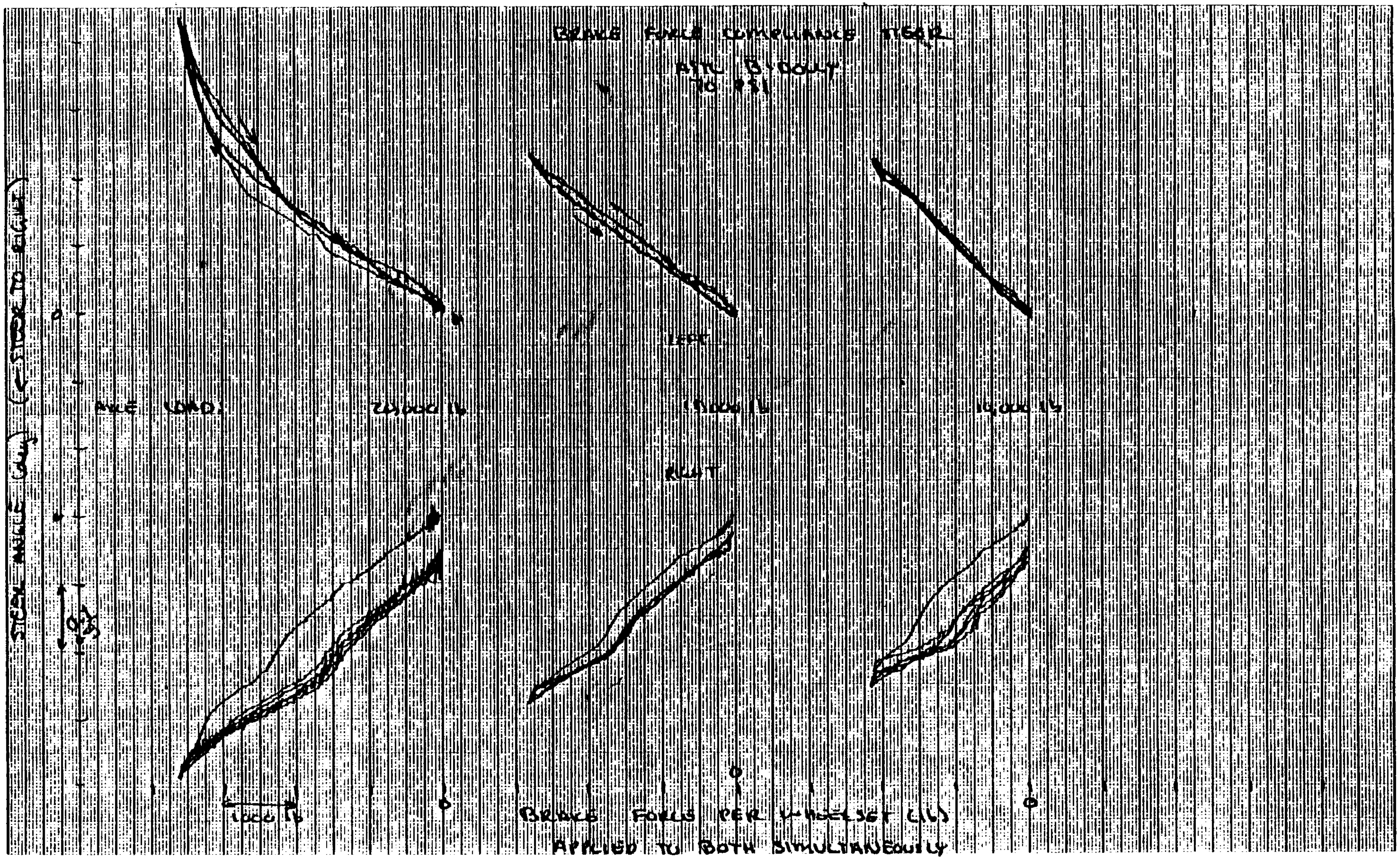

LATERAL FORCE COMPLIANCE STEER
LEFT
RIGHT
LATERAL FORCE PER WHEEL SET (lbs)
APPLIED TO BOTH SIMULTANEOUSLY
STEER ANGLE (deg) (+ STEER TO RIGHT)

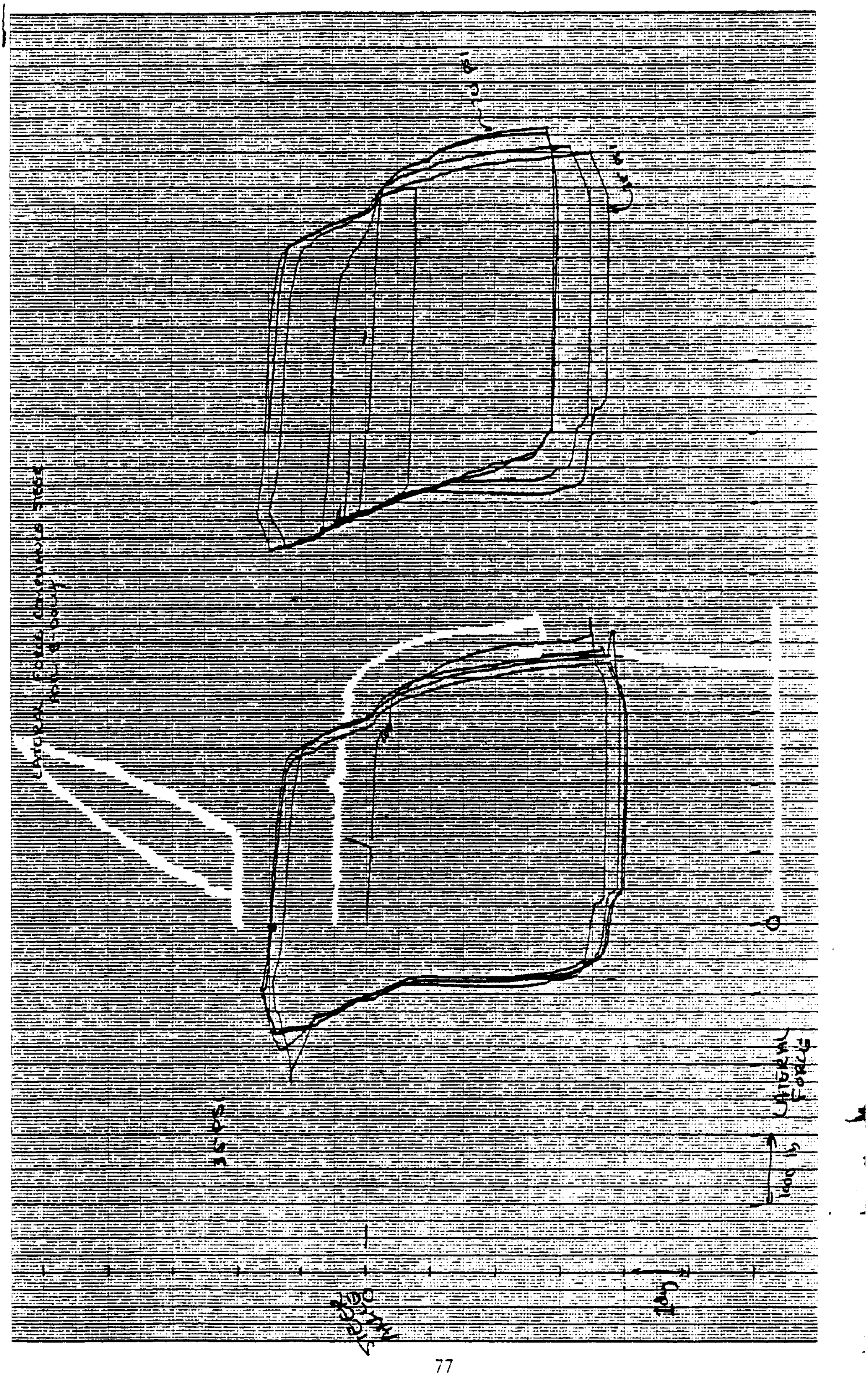

LATERAL FORCE COMPLIANCE STEER
ATL B-DOLLY
35 PSI
AXLE LOAD
LEFT
RIGHT
LEFT
RIGHT
1000 lb
0
0
0
LATERAL FORCE PER WHEELSET (lb) (← FORCE TO RIGHT)
APPLIED TO BOTH SIMULTANEOUSLY
STEER ANGLE (deg) (← STEER TO RIGHT)


LATERAL FORCE COMPLIANCE STEER
ASTL B-DOLLY
70 PSI
GROSS AXLE LOAD: 20,000#
15,000#
10,000#
LEFT
RIGHT
STEER ANGLE (deg) (← STEER TO RIGHT)
1°
1000 lb
LATERAL FORCE PER WHEELSET (lb) (← FORCE TO RIGHT)
APPLIED TO BOTH SIMULTANEOUSLY


BRAKE FORCE COMPLIANCE STEER
AXLE B-DOLLY
STEERING LOCKED
AXLE LOAD: 20,000 lb
15,000 lb
10,000 lb
LEFT
RIGHT
STEER ANGLE (deg) (STEER TO RIGHT)
0.1 deg
1000 lb
BRAKE FORCE PER WHEELSET (lb)
APPLIED TO BOTH SIMULTANEOUSLY


BRAKE FORCE COMPLIANCE STEER
AXLE 13-DOLLY
35 PSI
LEFT
AXLE LOAD:
20,000 lb
15,000 lb
10,000 lb
RIGHT
STEER ANGLE (deg) (+ STEER TO RIGHT)
0
0
1000 lb
0
0
0
BRAKE FORCE PER WHEELSET (lb)
APPLIED TO BOTH SIMULTANEOUSLY


BRAKE FORCE COMPLIANCE TEST
LEFT
AXLE LOAD:
24,000 lb
19,000 lb
14,000 lb
RIGHT
STEER ANGLE (deg) (← STEER TO RIGHT)
0.1°
0
1000 lb
BRAKE FORCE PER WHEELSET (lb)
APPLIED TO BOTH SIMULTANEOUSLY


36 PSI
LATERAL FORCE

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1. Gillespie, T.D. and MacAdam, C.C. "Constant Velocity Yaw/Roll Program - User's Manual." Final Report to Ford Motor Company under P.O. NP-47-A-885054-2, Transportation Research Institute, The University of Michigan, Report No. UMTRI-82-39, October 1982.

2. Ervin, R.D. and MacAdam, C.C. "The Dynamic Response of Multiply-Articulated Truck Combinations to Steering Input." SAE Paper No. 820973, August 1982.