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Improving the Dynamic Performance of Multitrailer Vehicles: A Study of Innovative Dollies

Volume I

Final Technical Report

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July 1986

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16. Abstract

This study of the dynamic performance of multitrailer articulated vehicles has led to the development of guidelines for the design of innovative dollies that will improve the roll stability and trailing fidelity of doubles combinations. The major effort of this project involved identification, analysis, and further development of innovative dolly and trailer hitching hardware showing potential for the reduction of rearward amplification and prevention of rollover of the second trailer. Specifically, the project (1) reviewed the current state-of-the-art in innovative coupling mechanisms, (2) performed a parametric sensitivity study, based on computer simulation techniques, on combination vehicles using existing and proposed coupling mechanisms, and incorporating various combinations of 96- and 102-in-width hardware, (3) developed a new type of dolly believed to provide superior safety performance, (4) conducted full-scale tests of combination vehicles using various dollies, including a prototype of the new dolly, and (5) examined the potential safety and economic impacts of the use of innovative dolly hardware.

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INTRODUCTION

This document constitutes the final technical report for the research project entitled "Techniques for Improving the Dynamic Ability of Multitrailer Combination Vehicles." The project was conducted by The University of Michigan Transportation Research Institute (UMTRI) and was sponsored by the Federal Highway Administration (FHWA) of the U.S. Department of Transportation under Contract No. DTFH61-84-C-00026.

The Surface Transportation Assistance Act of 1982 allows the use of double-trailer combination vehicles nationwide on the designated highway system. It also allows for the increase of commercial vehicle widths from 96 in (2.44 m) to 102 in (2.59 m). This Act is generally expected to result in a major increase in the number of multitrailer commercial vehicles in use throughout the U.S. At the same time, pressure for allowing the use of triples is increasing. In light of the fact that multitrailer vehicles are known to suffer from special dynamic characteristics that can limit their stability and emergency maneuverability characteristics, vis-a-vis the tractor-semitrailer, these developments have led to concern over the potential for degradation of the safety quality of the U.S. commercial vehicle fleet. The primary purpose of this research study was to obtain (and disseminate) information on developments in heavy-vehicle technology which might provide improvement in the dynamic performance of multitrailer vehicles, while this envisioned transition from singles to doubles was in progress. The purpose of the project was addressed through two specific goals, viz. (1) to develop safer, practical coupling mechanisms for multitrailer combinations, and (2) to determine the safety effects of various width combinations possible under the 102-in- (2.59-m-) width limitation.

For purposes of this study, the goal of "improving the dynamic performance" of multitrailer vehicles was taken to imply that the conventional A-train double combination vehicle be taken as the reference. It is well established in the literature that maneuvering quality of the tractor-semitrailer portion of an A-train doubles combination vehicle is virtually unaffected by the presence of the full trailer, but that, in emergency maneuvers, the second trailer of the doubles suffers from a "crack-the-whip" phenomenon in which the second trailer substantially exaggerates, or amplifies, the motions of the tractor.[1-15] The major safety consequence of this "rearward amplification" is the premature rollover of the second trailer. Rearward amplification and the resulting propensity toward rollover of the second trailer is generally recognized as the property of the double which distinguishes (and degrades) its dynamic performance capability from that of the tractor-semitrailer combination vehicle.

The major effort of this project, then, involved identification, analysis, and further development of innovative dolly and trailer hitching hardware showing potential for the reduction of rearward amplification and prevention of rollover of the second trailer. Specifically, the project (1) reviewed the current state-of-the-art in innovative coupling mechanisms, (2) performed a parametric sensitivity study, based on computer simulation techniques, on combination vehicles using existing and proposed coupling mechanisms, and incorporating various combinations of 96- (2.44-) and 102-in- (2.59-m-) width hardware, (3) developed a new type of dolly believed to provide superior safety performance, (4) conducted full-scale tests of combination vehicles using various dollies, including a prototype of the new dolly, and (5) examined the potential operational impact of the use of innovative dolly hardware.

The major motivation for the use of multiply articulated trains by commercial trucking interests is to obtain a vehicle with high cargo volume which retains the practical benefit of good, low-speed maneuverabilty. Within the constraints of vehicle height and width laws, more cargo volume is attained by lengthening the vehicle. Generally, as vehicle length increases, so do maneuvering problems, since the magnitude of low-speed offtracking is directly related to vehicle length. However, the offtracking of a vehicle of a given length is generally reduced by the introduction of additional yaw articulation joints. By virtue of these facts, the so-called A-train doubles combination has become a popular commercial vehicle.

An A-train consists of a tractor-semitrailer pulling one or more conventional full trailers, where a conventional full trailer consists of a semitrailer whose forward end is supported by a dolly which (1) articulates in yaw relative to the semitrailer, (2) is connected to the towing unit by a single pintle hitch, and (3) has one or more axles whose wheels are non-steering relative to the dolly frame. While the A-train meets the primary need of providing a large-volume vehicle which can be maneuvered relatively easily at low speed, it is also known to be less stable at highway speeds than is the conventional tractor-semitrailer.

The dynamic stability of the A-train suffers from the phenomenon known as rearward amplification, wherein, in steering maneuvers of relatively high frequency content, trailing units in the train will tend to experience higher lateral accelerations than their towing unit. Thus, lateral acceleration "amplifies" as one moves rearward along the train, and the rearmost trailer may experience accelerations much larger than those experienced by the driver in the tractor. The most serious safety consequence of the phenomenon is the resulting rollover of the rear trailer. Further, the addition of yaw articulation joints tends to reduce the yaw damping of the vehicle, and the reduction of low-speed offtracking tends to aggravate high-speed offtracking.

In recent years, the safety-degrading influence of additional articulation joints has become broadly recognized. Nonetheless, the economic motivations for the use of multitrailer trains is so compelling that the use of such vehicles does and will continue to grow. Efforts to improve the safety quality of these vehicles have led, however, to a number of innovative dollies and hitching mechanisms being developed. This project examines these developments and makes its own contribution towards the goal of improving the dynamic performance capability of multitrailer commercial vehicles.

The project, as structured, included eight tasks. The manner in which these tasks fit with one another in pursuing the goals of the project is shown in figure 1. As indicated, the study began with a literature and "market" review task. This activity was intended to identify all of the innovative ideas in dolly and hitching hardware which were currently under development or in use around the world. Following this review, a simulation plan was developed and a simulation study was conducted. Initially, all of the ideas identified by the review were screened to determine which of them appeared to hold significant promise of improved performance. The more promising schemes were selected and studied in more detail. As the simulation study proceeded, the knowledge and understanding gained were used to help develop a new dolly concept whose performance was also studied in the simulation task. Under task D, this new concept was reduced to operating hardware, and in task E, three other real dollies representing the other promising concepts were acquired. These four dollies were then subjected to a full-scale test program, task F, to confirm the findings of the simulation study. Task G included several activities: the prototype dolly was placed in commercial service as an additional "test" of the concept; in addition, analyses were conducted to determine the potential safety and economic gains which might be attained through the use of improved dollies. This final report was developed under task H.

The following sections report on the activities undertaken and the findings developed in the four major study areas, namely, the survey, simulation, testing, and operational impacts portions of the study. The concept of the prototype dolly is discussed in the simulation study portion of the report, and the resulting hardware prototype is described in the section on testing. Recommendations and conclusions deriving from the study are presented in the final section of the main body of the report.

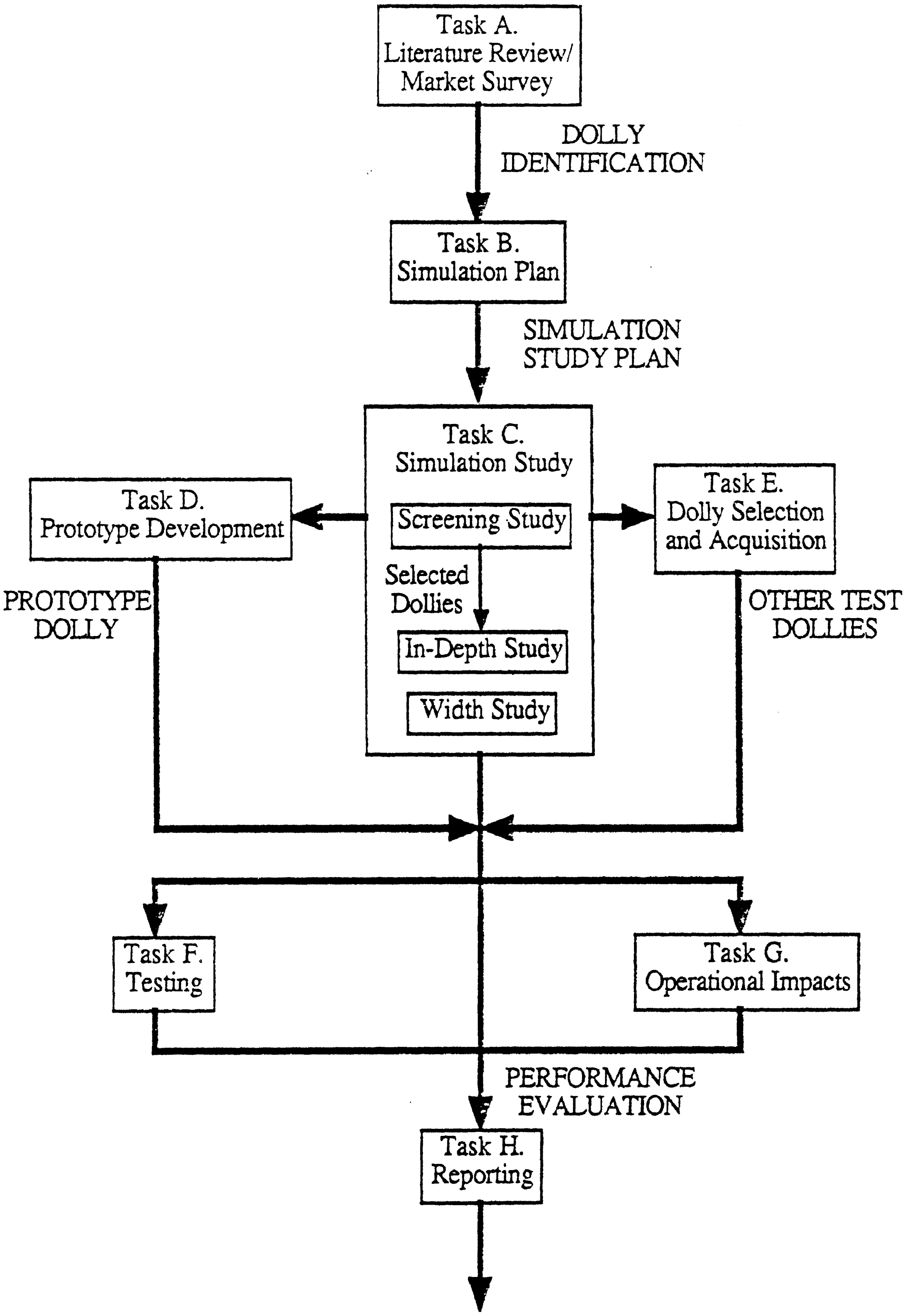


Figure 1. Project flow chart.

A REVIEW OF INNOVATIVE DOLLIES AND COUPLING MECHANISMS

Task A of this study was a "literature review" intended to identify new, innovative, trailer-to-trailer hitching mechanisms available or being developed worldwide. This project's primary interest in such devices was to identify those which are most capable of improving the dynamic performance of multitrailer vehicles, vis-a-vis the A-train, while retaining the desirable, practical properties of the A-train to the greatest extent possible.

In conducting this review, UMTRI contacted individuals or organizations in the U.S. and Canada who are involved in the development and/or manufacture of innovative dollies or hitching hardware. In addition, letters of inquiry were mailed out worldwide to individuals involved in manufacturing, trade associations, or regulatory offices, and members of the academic community involved in commercial vehicle research. In addition to the U.S. and Canadian contacts, responses were received from the following countries:

- Argentina
- Australia
- Austria
- Brazil
- Belgium
- France
- Germany
- Great Britain
- Hungary
- Italy
- Japan
- The Netherlands
- Sweden
- Switzerland

These responses made it clear that the trucking industry, worldwide, is currently experimenting with a variety of innovative dollies and hitching mechanisms. Most of this activity is taking place in Canada, where use of the so-called B-dolly is growing. Considerable activity is taking place elsewhere, however. Unconventional dollies are in use in Australia and New Zealand, where multitrailer combination vehicles are very common. In Europe, much of the development in new dolly hardware is not specifically aimed at altering dynamic performance, but rather at achieving increased trailer volume by reducing the spacing between trailers. These so-called close-coupling dollies permit unusually close trailer spacing by providing special mechanisms to prevent trailer-to-trailer interference in tight cornering. Some of these design concepts have been examined in this study, since they clearly have potential for influencing dynamic performance.

Many individual examples of innovative dollies in use or under development were identified. These dollies are listed in generic groupings in table 1. (Where applicable, this table shows the inventor or the commercial enterprises associated with a given dolly.) A discussion of the generic qualities defining the groups and some specific details of the individual dollies follows.

Table 1. Innovative Dollies and Hitching Hardware Identified in Task A.

Dolly Type	Description	Inventor, Commercial Interest, or Manufacturer.*
Modified A-Dollies	Dollies which retain yaw articulation capability between the first trailer and the dolly.	
Shifted IC Dollies	Dollies whose "pintle" hitch hardware causes a shift in the center of rotation of the dolly and first trailer away from the hitch point.	
Symmetric Trapezoid Hitch	Double drawbars, hinged at both ends, form a symmetric trapezoid about the longitudinal centerline in the plan view with the narrower end forward. The IC is forward of the physical hitch point.	Used by Michelin test fleet. No known active producer.
Asymmetric Trapezoid Hitch	Double drawbars, as above except one bar on centerline.	Norman Gallatin. Trapezoid Corp.
Converter, Trapezoid Hitch	Symmetric trapezoid, converts to rigid connection for low speed maneuvering.	Marcard Trailer Services
Double-Crossed Hitch	Double drawbars, hinged at both ends, criss-cross in plan view. The IC is at the cross point, rearward of the physical hitch point.	Arnies Welding. Hamelex Transport.
Roller Cam Hitch	Cam surface of hitch provides forward IC at small articulation angles, rearward IC at large articulation angles.	A. Pavluk, L. Segel, P. Fancher.
Forced Steer Dolly	The wheels of the dolly are forced to steer as a function of pintle articulation angle. Different types are produced with a variety of steering linkages. Used in Europe for "close-coupling."	Royce Curry, ASTL. Doll "AVL." Kogel Kassbohrer. Wackenhut. Ackermann-Fruehauf.
Linked Articulation Dolly	A-dolly with an additional linkage attached directly from trailer to trailer. A fixed relationship bewteen the pintle and fifth wheel articulation angles results.	Truck Safety Systems.

Table 1. (cont)

Dolly Type	Description	Inventor, Commercial Interest, or Manufacturer.*
Modified A-Dollies (cont.)		
Skid Steer Dolly	The yaw articulation joint at the dolly fifth wheel is eliminated. That is, the front tires of the full trailer do not steer at all.	Doetcker Industries.
K-Train	Modification of the skid steer concept. An "auto-steer," self steering axle is used for the dolly axle, so that the front tires of the full trailer steer by caster.	Knight Industries.
Roll-Stiffened Pintle Hitch	Fifth wheel-like device is used at the drawbar hitch.	Truck Safety Systems.
Extending Drawbar Dollies	The drawbar is caused to lengthen as either pintle articulation or fifth wheel articulation angle increases. For "close-coupling."	Blumhart. Pietz. Meier-Bürstadt. Eck.
Locking A-Dolly	Single point drawbar equipped with device which can "lock-out" yaw articulation. Operates as an A-dolly when "unlock" and as a B-dolly when locked.	VBG, Sweden
B-Dollies	Dollies which eliminate the yaw articulation between the first trailer and the dolly by using a rigid, double drawbar.	
Slider B-Dolly	B-dolly with fixed, non-steering axles which "slides" under the cargo area of the first trailer when the second trailer is absent.	Arquin Trailer. Monon Trailer. Fruehauf Corp. E. Tenn. Transport.
Auto-Steer B-Dolly	A B-dolly with a self-steering axle. The dolly axle is equipped with "automotive style" steering knuckles on positively castered kingpins. The steering system has a centering spring mechanism.	Royce Curry. ASTL. Knight Industries. Sterling Axle. Independent Trailer.
Turntable Steer B-Dolly	A B-dolly with a self-steering axle. Steering results from the rotation of a solid axle about a positive castered steering pivot located on the dolly centerline. The steering system has a centering spring mechanism.	ASTL. Arnies Welding. Westank-Willock. Knight Industries.

To bring order to the findings, we have identified two major categories, and several subgroups of dollies and hardware, as follows:

Modified A-Dollies

- Shifted-IC Dollies
- Forced-Steering Dollies
- Linked-Articulation Dollies
- Skid-Steer Dollies
- Roll-Stiffened Pintle Hitch
- Extending-Drawbar Dollies
- Locking A-Dolly

B-Dollies

- Non-steering B-Dollies
- Self-steering B-Dollies

The so-called A-dolly is, of course, the conventional single-drawbar dolly, which connects to the first semitrailer trailer with a single pintle hitch and to the second trailer with a conventional fifth wheel (converter dolly) or with a turntable bearing (turntable dolly). Modified A-dollies, as used herein, are dollies which retain the pintle hitch or any other form of coupling which permits yaw articulation between the dolly and the first trailer. B-dollies, on the other hand, are dollies which practically eliminate yaw motions between the first trailer and the dolly, usually by using rigid double drawbars and two pintle hitch connections. The "basic" A-dolly and B-dolly configurations are illustrated in figure 2.

1. Modified A-Dollies

a. Shifted IC Dollies.

In the parlance of the mechanical engineer, the pintle connection point between the first trailer and the dolly can be identified as the "Instant Center of Rotation" (IC) of the relative motion of these two bodies. For example, to a viewer located in the first trailer, all relative motions of the dolly and trailer would be seen as rotations of the dolly, with respect to the trailer, about the pintle connection point. To a viewer located on the dolly, these motions would be seen as rotations of the trailer. Thus the connection point is the IC of these two bodies. In the plan view,where only yaw motions are observed, it is the center of rotation "in yaw."

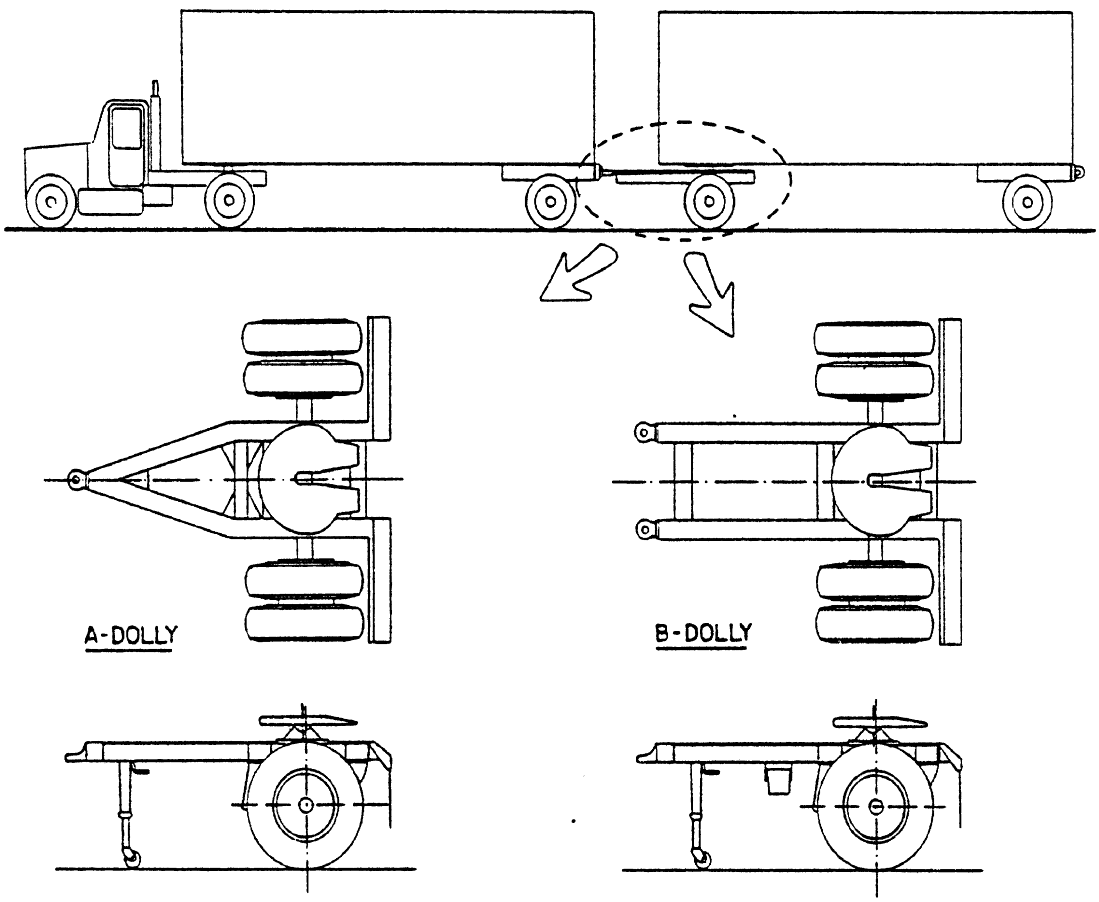


Figure 2. The A-dolly and B-dolly.

The technical literature establishes that the location of the pintle hitch can have an important influence on the dynamic behavior of doubles.(4,10,12,13,15) Further, it has been recognized that the importance of the pintle hitch location in this regard is not so much in the location of the actual physical connection, but rather in the location of the IC, in yaw, of the first trailer and the dolly.

Several dolly types, or hitching mechanisms, which effectively shift the IC away from the location of the actual physical connection point, were identified in the review. These are identified in table 1 as the "Shifted-IC Dollies."

The "trapezoidal dolly," shown in figure 3, is prominent in the Shifted-IC group. This dolly is characterized by a double-drawbar hitching arrangement wherein the drawbars are arranged in a trapezoidal pattern in the yaw plane. The drawbars are each connected to the dolly frame with a hinge joint, and to the first trailer with a pintle or ball joint. They are arranged so that the narrower end of the trapezoid is at the first trailer connection. The two drawbars and the frames of the trailer and dolly form what is known as a "four-bar" linkage. Classic linkage analysis shows that the IC of any two links (in this case, the trailer and dolly) lies at the intersection of the projections of the other two links (the drawbars). The IC of the dolly and trailer is located by this method in figure 3. The IC actually moves some as the dolly and trailer articulate (which is the reason for the use of the adjective *instant* in the term *instant center of rotation*), but for the relatively small articulations which occur at highway speeds, this motion is so small as to be negligible. Given that the dynamic performance is determined by the location of the IC (not the location of the actual hardware), it becomes clear that the trapezoidal dolly can be expected to have dynamic performance similar to that exhibited by an imaginary, conventional A-dolly with its pintle hitch located at the IC. This concept is illustrated in figure 4.

It is well established in the technical literature that forward IC locations are advantageous for reducing rearward amplification.(10,12,13,15) However, IC's which are well forward in the trailer exaggerate low-speed offtracking. An advantage of the trapezoidal drawbar design is that the IC can be moved rearward during low-speed operating conditions by providing a mechanism for bringing the forward drawbar hitching points closer together.

Several examples of trapezoidal dollies exist. For example, the Michelin Tire Corporation uses some specially built, symmetric trapezoidal dollies (figure 3) in its U.S. test fleet.

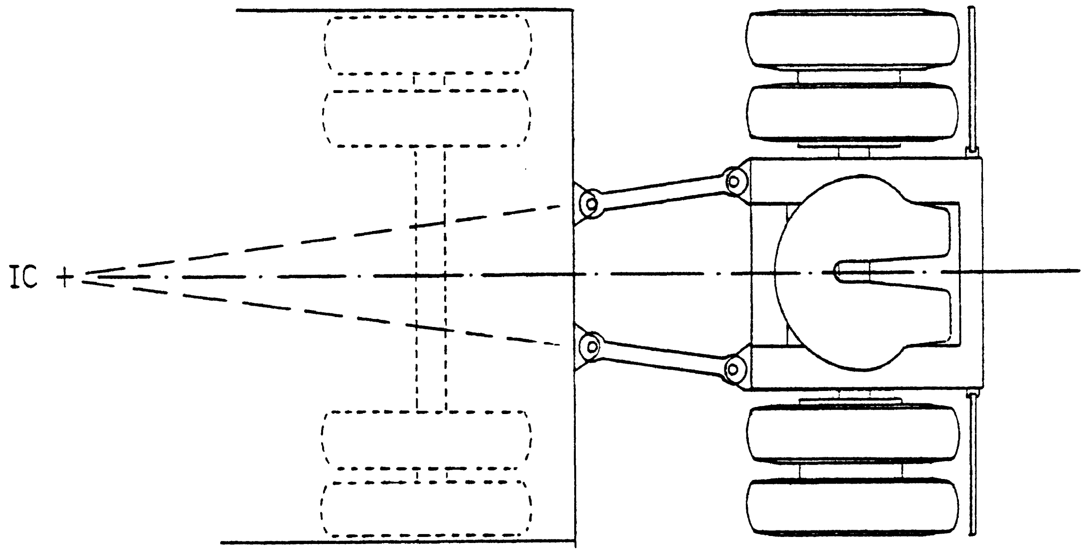
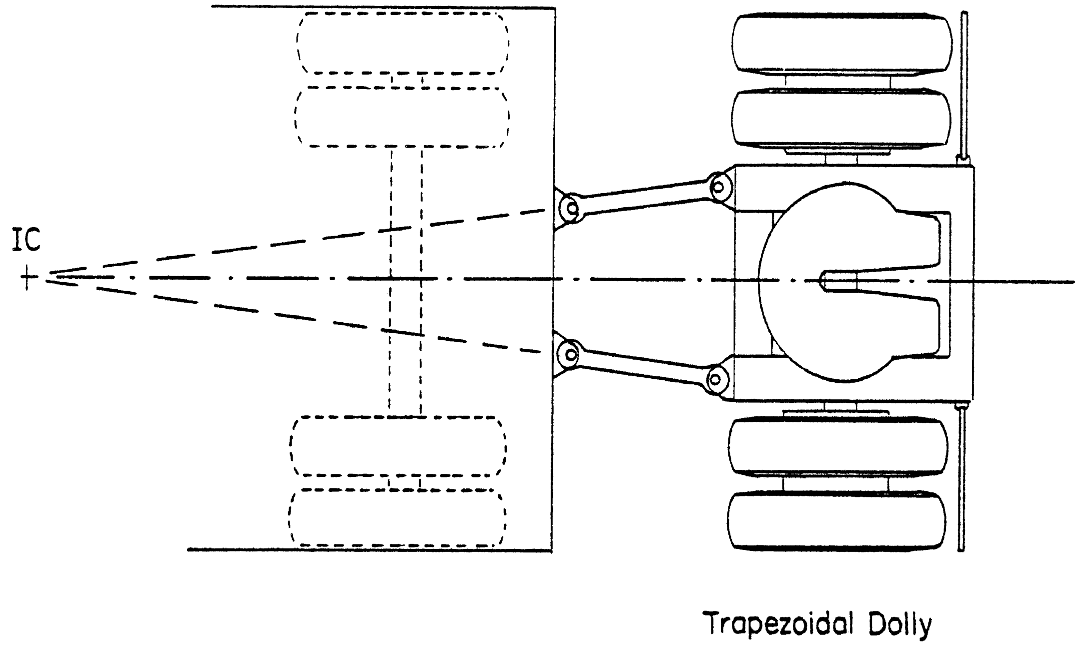


Figure 3. The symmetric, trapezoidal dolly.



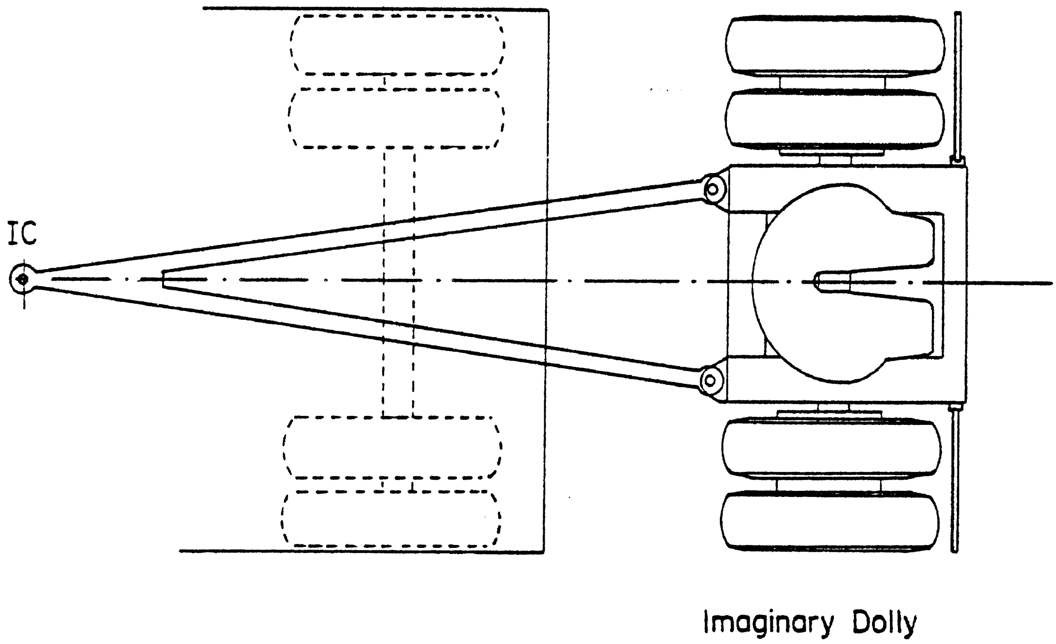


Figure 4. A trapezoidal dolly and the equivalent, imaginary A-dolly.

The Trapezoidal Corporation has produced at least one prototype of an asymmetric trapezoidal dolly design, as shown in figure 5. The asymmetric feature allows for a stout, central drawbar and a lighter-weight, second drawbar. This arrangement is thought by the inventor to yield easier hitching. It also provides a simpler, automatic relocation of the IC, since only the hitching point of the lighter arm need be moved. For the small articulations which occur at highway speeds, the lateral asymmetry is insignificant.

The Marcard Trailer Service has produced a "converter" trapezoidal dolly. Shown in figure 6, the dolly operates as a trapezoidal dolly with a forward IC at highway speeds, but "converts" to a rigid connection to dramatically reduce low-speed offtracking in tight maneuvering.

Another type of Shifted-IC dolly is the "double-crossed" dolly, shown in figure 7. The hitching arrangement of this dolly is also a "four-bar" linkage, but the drawbar links are crossed between the dolly and trailer frames. Thus the IC of the trailer and dolly is at the crossing point, unusually rearward in the trailer. Because of this, the double-cross arrangement can be expected to provide good, low-speed offtracking but high levels of rearward amplification. The double-crossed dolly is known to be manufactured in Canada (Arnie's Welding) and in Australia (Hamlex Transport).

The "roller cam" hitch is another innovative idea which falls in the Shifted-IC group. Shown in figure 8, this concept causes the IC to be located well forward when the articulation between the first trailer and dolly is relatively small. At large articulations, as experienced in tight, low-speed maneuvering, the IC shifts rearward. A patent disclosure has been filed in the U.S., but no hardware version has been developed.

While these five types of Shifted-IC Dollies are substantially different in mechanical detail, and, accordingly, may have substantially different practical, operating properties, they can be considered as similar for purposes of a simulation study. In essence, each of these devices could be simulated as an A-dolly, with a "pintle hitch" location (IC) which is a function of the specific design of the hitch. In the simulation study, the performance of these dollies, as a group, were studied by examining the sensitivity of performance to the location of the IC, over a range of locations implied by the several different specific designs.

b. The Forced-Steer Dollies.

The survey identified a number of similar, commercially available dollies which can be identified as "forced-steer" dollies. These dollies are in the A-dolly category, since they

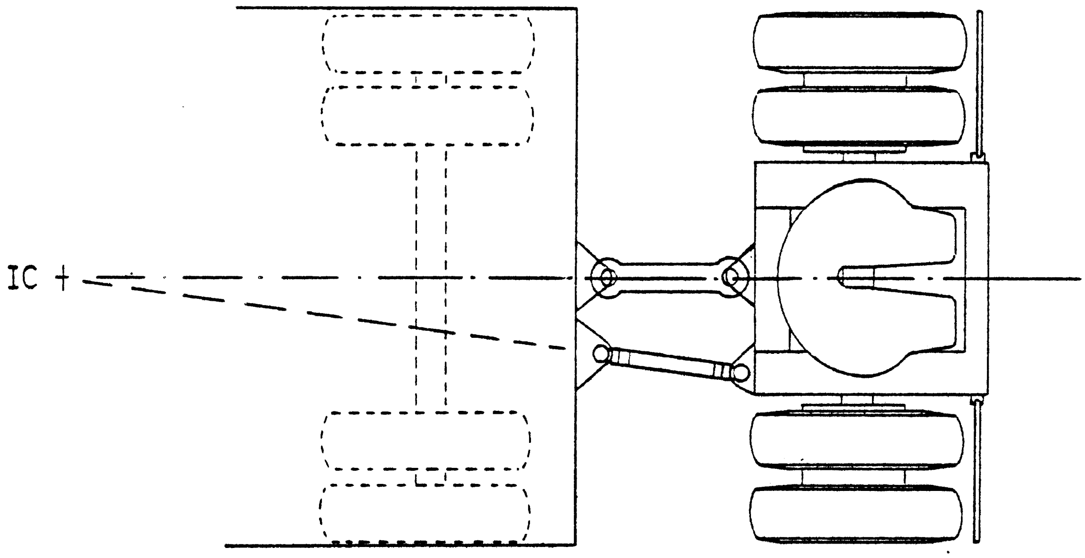


Figure 5. The asymmetric, trapezoidal dolly.

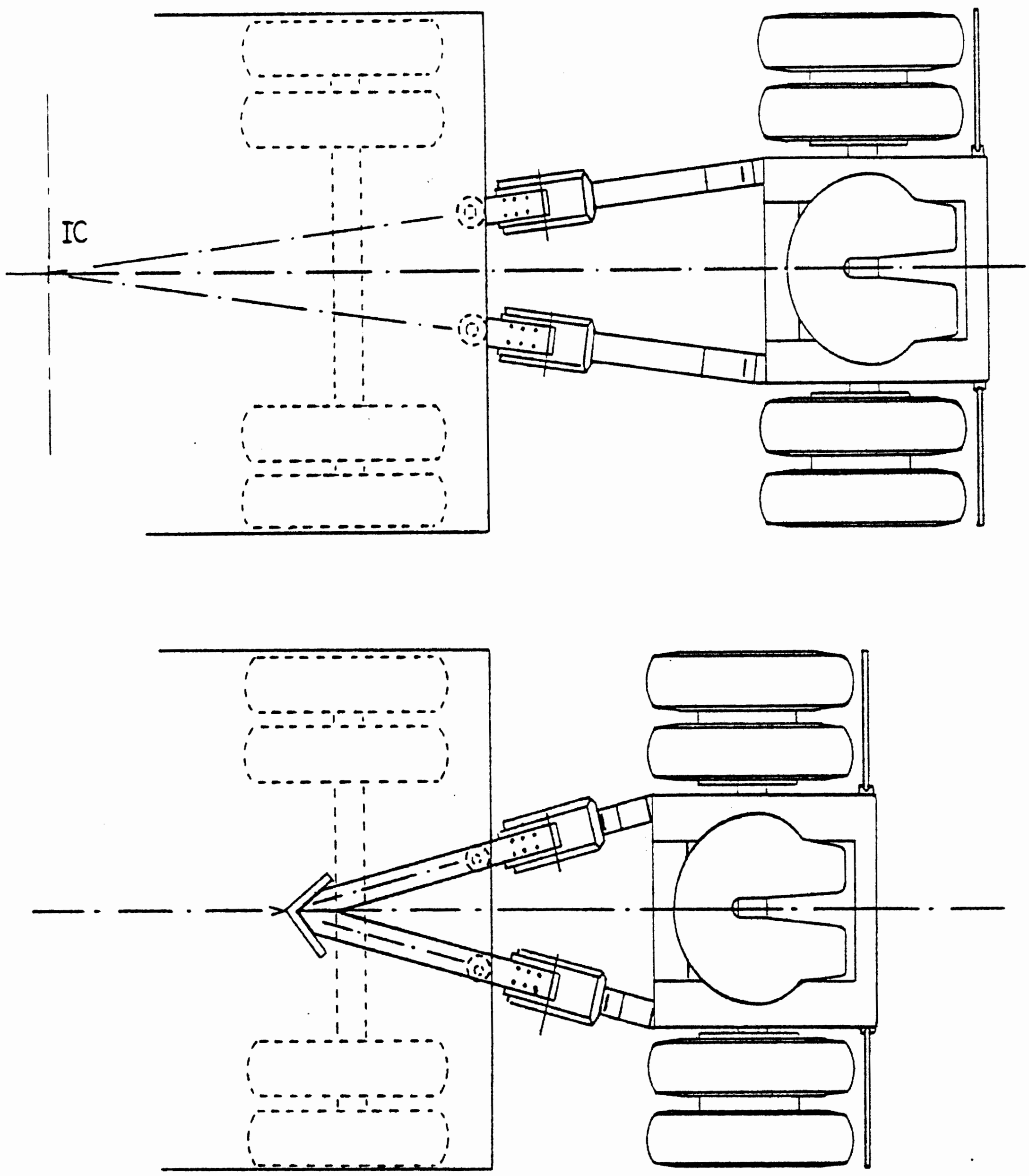


Figure 6. The converter, trapezoidal dolly.

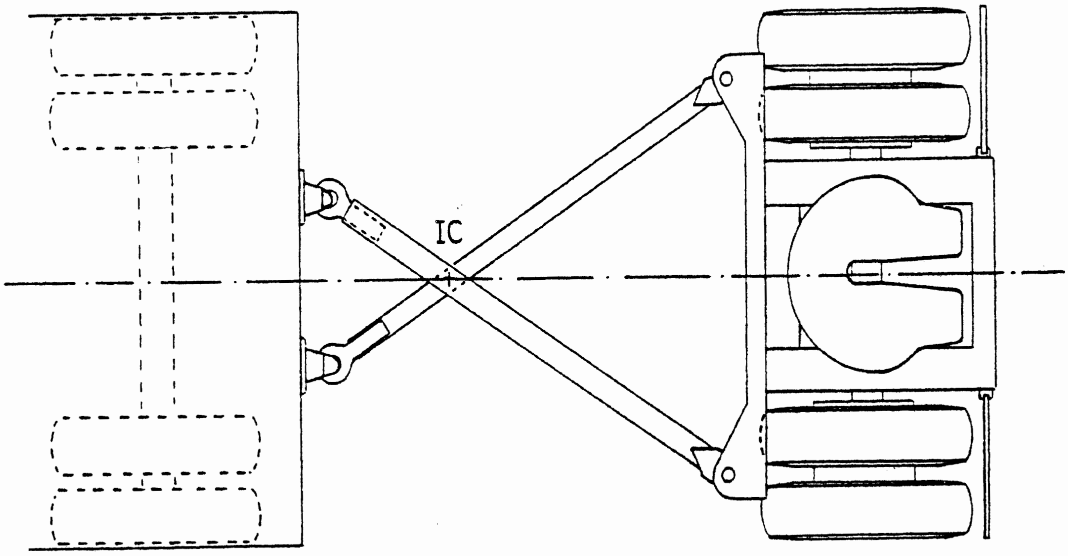


Figure 7. The double-cross drawbar.

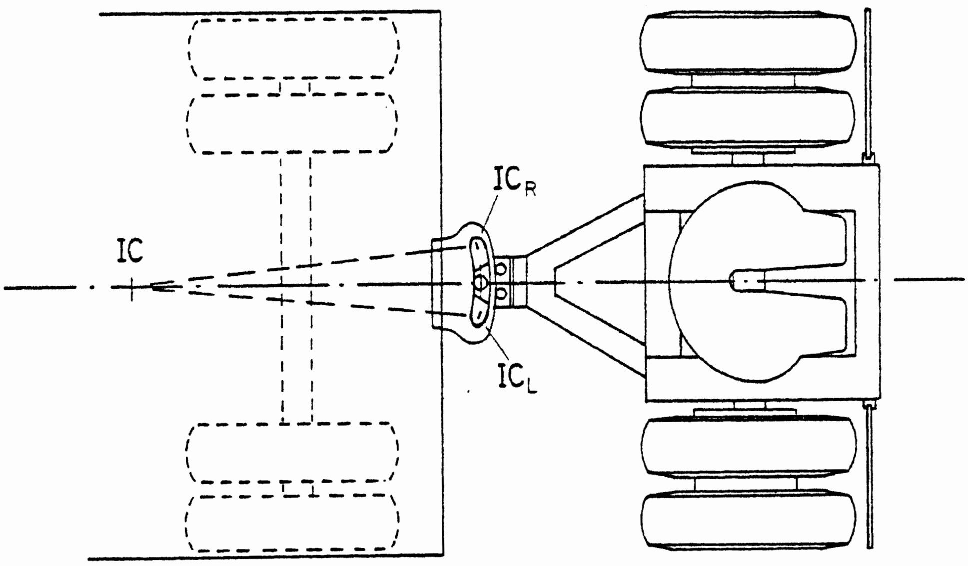


Figure 8. The A-dolly with roller cam hitch.

retain a pintle-hitch-like connection to the first trailer. However, they each possess a mechanism which provides for controlled steering of the dolly tires (relative to the dolly frame) as a function of pintle hitch articulation. Most, but not all, of the known examples are European developments where the primary interest is in providing advantageous steering geometry for close-coupling trailers. As shown in figure 9a, these designs generally cause the tires of the dolly to steer to a greater angle (relative to the first trailer) than they would by dolly articulation alone. However, another forced-steer dolly designed in Canada (by ASTL), shown in figure 9b, causes the dolly tires to steer in the other direction.

In the case of forced-steer dollies, the significant characteristic distinguishing one dolly from another is the relationship between articulation angle and tire steer angle--that is, the steering gain. In the simulation study, these dollies were evaluated as a group by examining the sensitivity of performance to steering gain, over a broad range of gains.

c. The Linked-Articulation Dolly.

In the normal operation of the conventional A-dolly, the yaw articulation that occurs between the first trailer and the dolly (about the pintle connection) and the yaw articulation which occurs between the dolly and the second trailer (about the fifth-wheel connection) are independent of one another. These articulations are not "linked" so that, within the range allowed, any pintle angle can exist with any fifth-wheel angle. In engineering terms, the existence of these two independent yaw articulation joints means that the mechanical system has "two degrees of freedom" in yaw.

At least two similar devices are known to exist which modify a conventional A-dolly arrangement in a manner which "links" the two articulation angles. The "Telescopic Steering Stabilizer" (a patented device of the Truck Safety Systems Company), shown in figure 10, is such a device. On the order of 100 such devices have been built and sold for use on double-tanker vehicles in Michigan. Another, similar device which uses an accordian (rather than telescopic) arm has been seen on the road in Michigan, but the manufacturer is not known.

Each of these devices mounts an extra piece of hitching hardware between the rear of the first trailer and the front of the second trailer (that is, not connected to the A-dolly at all). The connection to the second trailer is a ball joint, and the device causes this hitching point (on the second trailer) to always remain on the projection of the centerline of the first trailer. By doing so, a specific relationship is caused to exist between the articulation angle between the first trailer and dolly and the articulation angle between the dolly and second

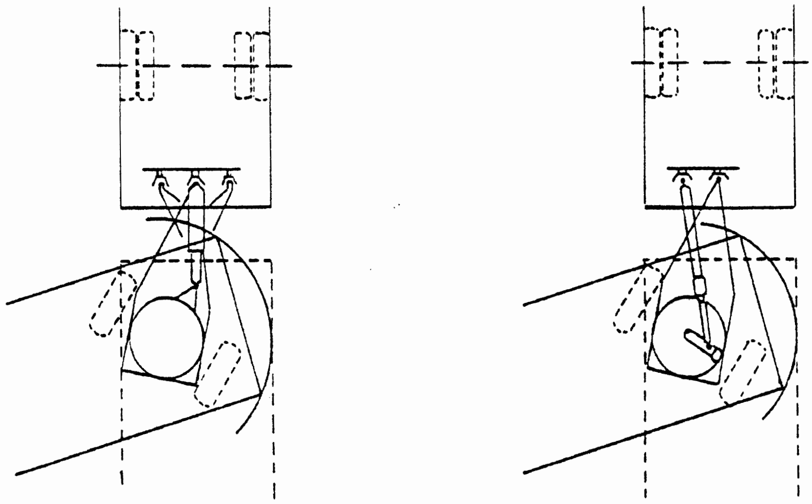


Figure 9a. Two European forced-steer dollies.

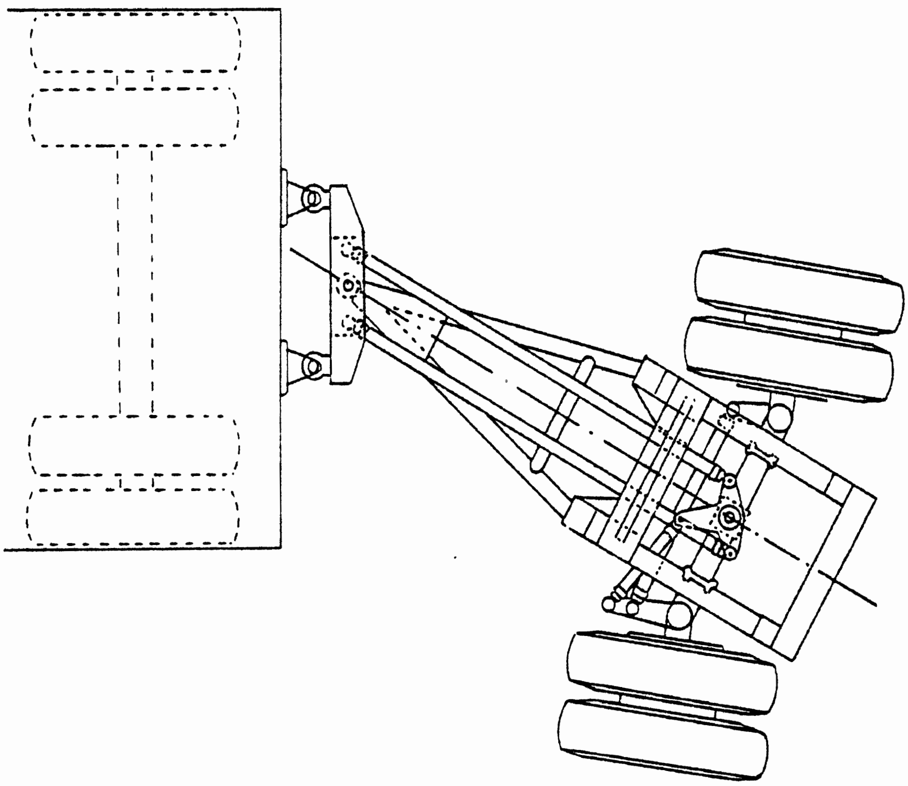


Figure 9b. A Canadian forced-steer dolly.

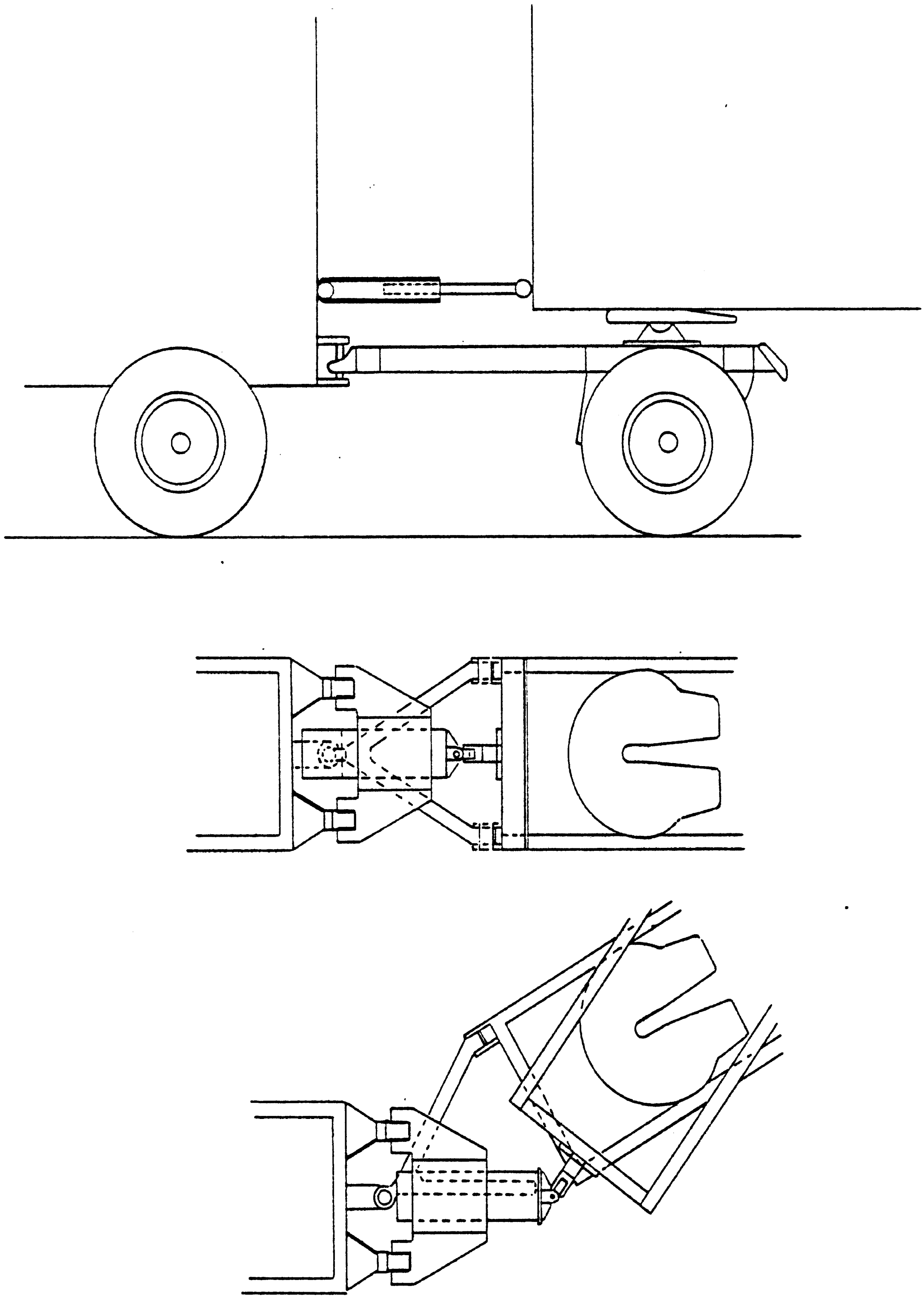


Figure 10. Linked articulation dolly with telescoping member.

trailer. The articulation angles are "linked." Even though two articulation joints exist, the mechanical system has been reduced to one degree of freedom, since there is only one "independent" (and one "dependent," or resulting) motion.

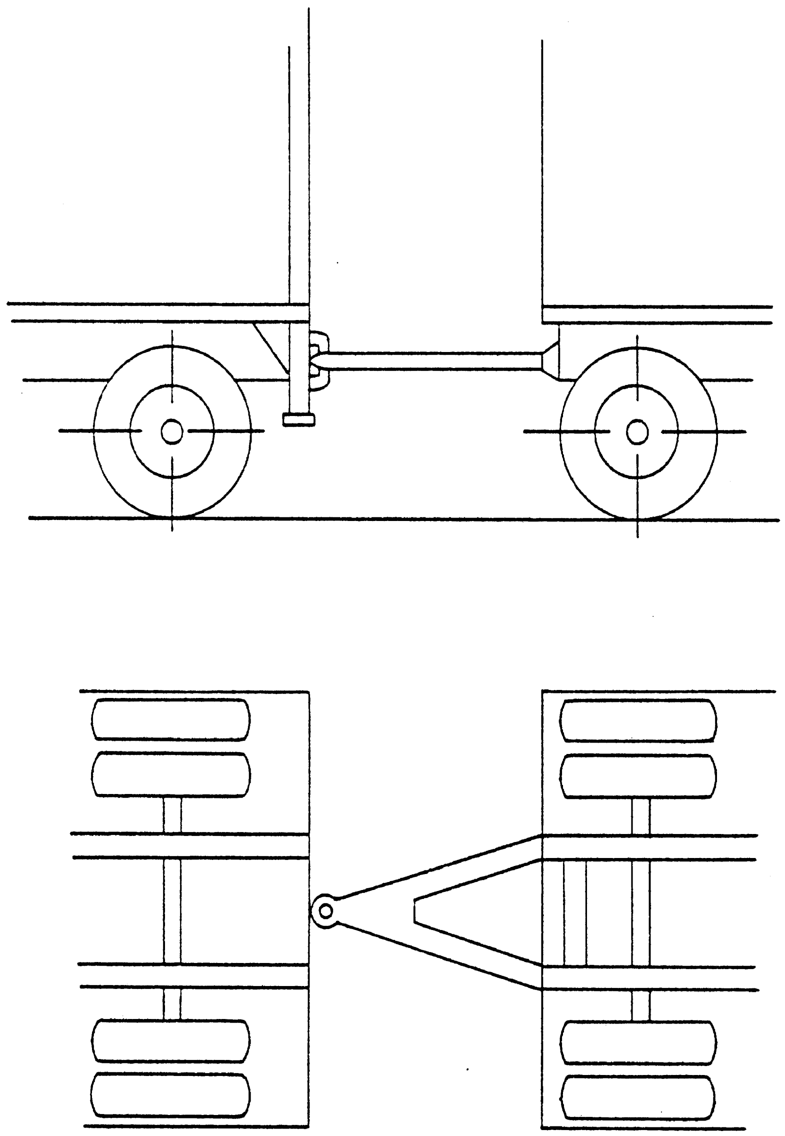
The elimination of a degree of freedom represents a fundamental change to the vehicle system, and significant changes in dynamic performance can be expected to follow. The significant system parameter to be examined in the simulation study is the "gain," which is established between the articulation angles. (If the pintle articulation angle to fifth-wheel articulation angle gain is zero, the linked articulation dolly reverts to a non-steering B-dolly. If this gain is infinity, it reverts to a skid-steer dolly.) (See below.)

d. The Skid-Steer Dollies.

The "skid-steer" dolly is shown in figure 11. This dolly is similar to an A-dolly with a normal pintle hitch, but yaw articulation about the fifth-wheel connection with the second trailer has been eliminated. That is, the skid-steer dolly converts the second semitrailer to a full trailer whose front axle does not steer. Previous analysis has shown that the simple skid-steer arrangement produces a pup which is very lightly damped in yaw. High pintle hitch forces and difficulty in maneuvering can be expected also. Unlike the usual situation, low-speed offtracking becomes a strong function of tire properties, loading conditions, and roadway friction. Skid-steer dollies are known to have been operated in the Canadian Province of Saskatchewan.

At least one working example of the so-called K-train exists and has been used on the road in Saskatchewan. The K-train is a modification of the skid-steer dolly concept where there is no fifth wheel on the second full trailer. The drawbar is fixed to the frame of the full trailer, but the front axle of the full trailer is an "auto-steer," self-steering axle. As will be discussed in more detail later, the wheels of these axles are allowed to steer by castering action by overcoming a "centering spring," or steering-resisting mechanism. If a very high level of steering resistance is used, the K-train would obviously behave as a double with a skid-steer dolly.

The skid-steer dolly and K-train are grouped together for consideration in the simulation study. The characteristic system parameter of interest is the level of steering resistance at the dolly axle.



Note:

No "fifth wheel".
Dolly is "solid"
part of trailer.

Figure 11. The skid-steer dolly.

e. The Roll-Stiffened Pintle Hitch.

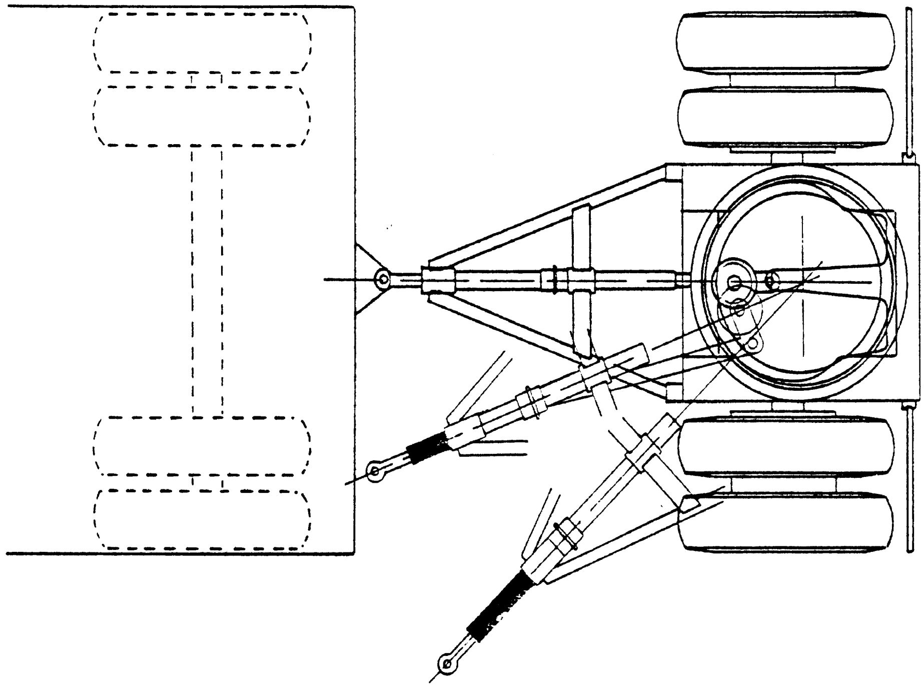
Truck Safety Systems of Michigan has marketed a dolly hitching device which provides a roll-resistant coupling between the dolly and the first trailer. This device replaces the usual pintle hitch connection with a fifth-wheel-like coupling. In dynamic turning maneuvers, when the roll motions of the two trailers of the doubles are out of phase, the roll coupling between trailers serves to improve the roll stability of each. Depending on the tires used, rearward amplification may be reduced due to changes in dynamic tire loading. In the simulation study, the parameter of interest is the effective level of roll stiffness in the trailer-to-trailer roll coupling.

f. Extending Drawbar Dollies.

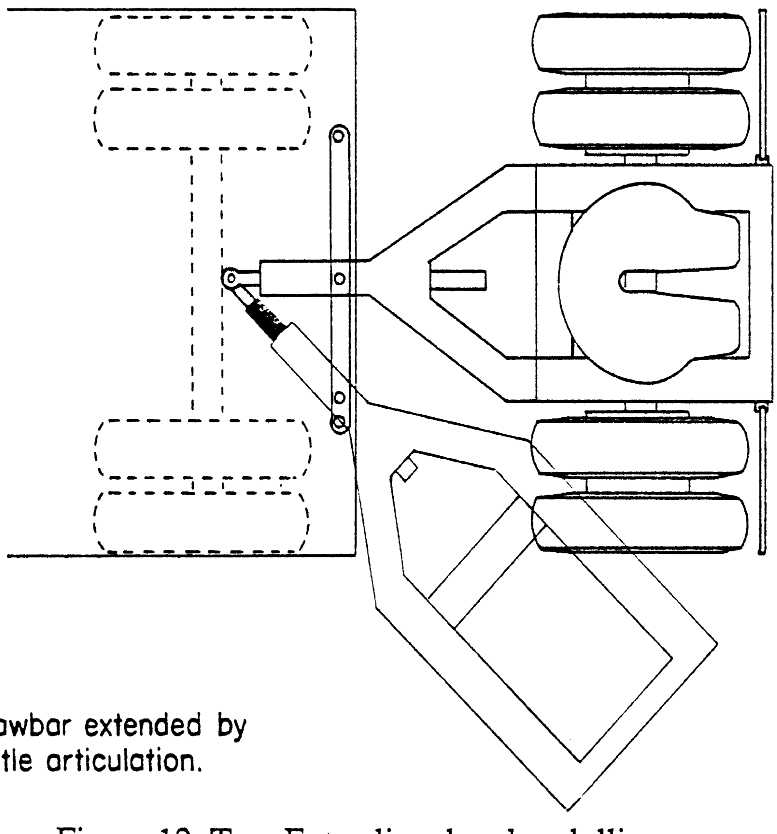
Extending drawbar dollies (figure 12) are another type of close-coupled dolly available in Europe. These dollies are, in effect, A-dollies with the exception that their drawbar length adjusts automatically as a function of pintle articulation angle. Specifically, the drawbar, in one manner or another, telescopes, such that it is shortest at zero articulation and increases in length at off-center conditions. This feature allows for close coupling the trailers in straight running while avoiding trailer-to-trailer interference in tight maneuvering. Some versions control tongue length as a function of pintle articulation (e.g., Blumhart system) and some operate on fifth-wheel articulation (e.g., Meier system). The judgment was made that this feature was not likely to have a significant influence on dynamic stability, and, accordingly, this dolly type was not examined in the simulation study.

g. The Locking A-Dolly.

Figure 13 shows a dolly being developed by VBG of Sweden. This is the only European device which was identified as a scheme specifically to enhance the dynamic performance of multiply articulated vehicles. It would seem reasonable to have classified this dolly as either a modified A-dolly or as a B-dolly. In its "locked" configuration, this dolly functions as a simple B-dolly, and in its unlocked configuration, it acts as a conventional A-dolly. The dolly is "locked" at highway speeds to provide good dynamic performance, and unlocked at low speeds to provide good maneuverability and prevent high frame stresses. This study does not treat this dolly separately, since its performance is either that of an A-dolly or that of a simple, non-steering B-dolly.



Drawbar extended by fifth wheel articulation.



Drawbar extended by pintle articulation.

Figure 12. Two Extending drawbar dollies.

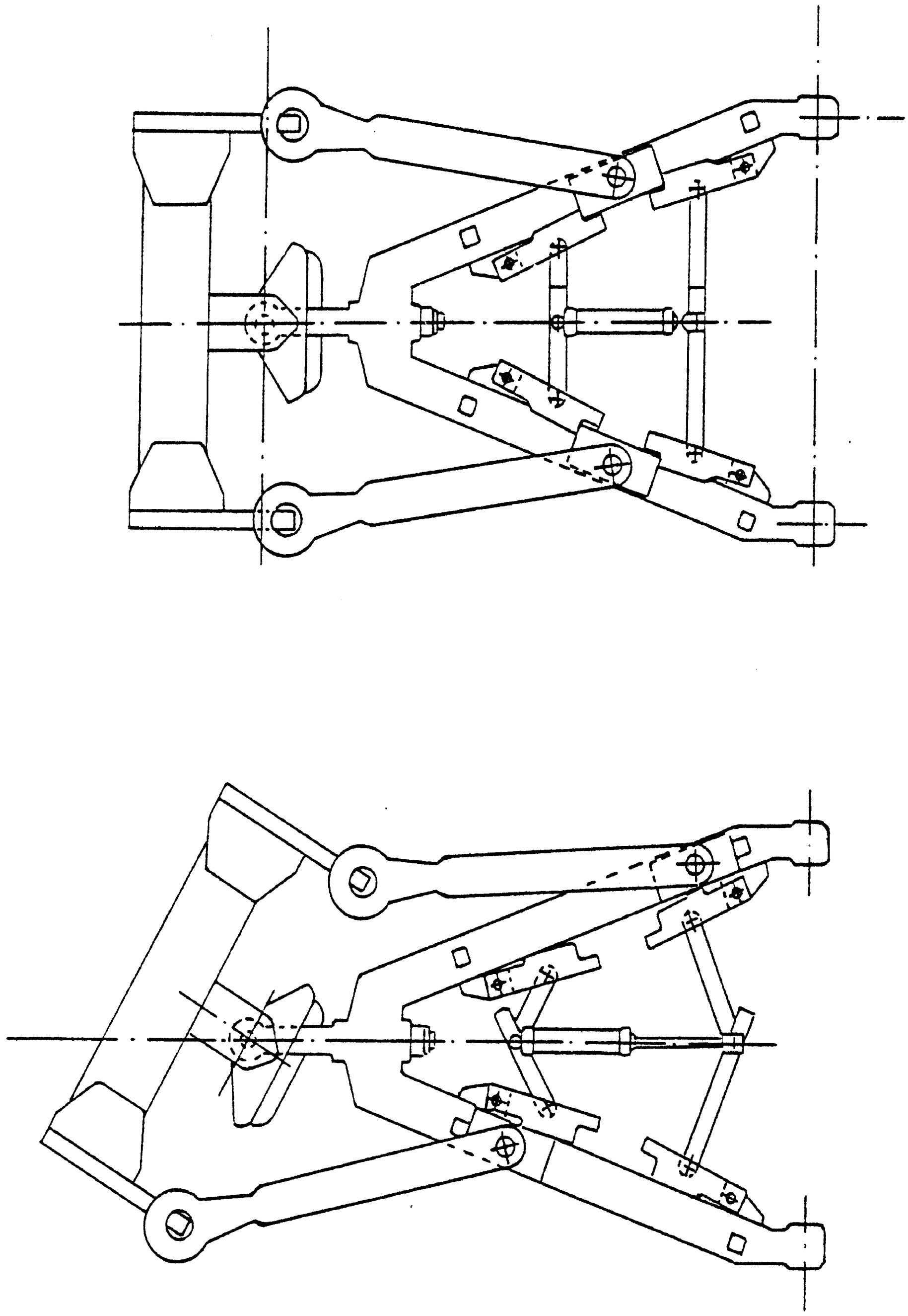


Figure 13. The locking A-dolly.

2. B-Dollies

All the preceding devices are considered to be "modified" A-dollies, in that they each retain a pintle-hitch-like articulation joint. B-trains and B-dollies, on the other hand, eliminate the yaw articulation which occurs at a pintle joint.

B-dollies are actually an extension of the B-train concept. The B-train (figure 14) is a multitrailer vehicle employing only semitrailers, i.e., no full trailers. Each towing trailer is equipped with a rigid frame extension aft of its cargo area which is fitted with a fifth wheel for coupling to the following semitrailer. The fifth wheel may be conventional, but often, and particularly on tank trailers, a so-called compensating fifth wheel may be used to reduce stresses imposed on the frame. The improved rearward amplification performance of this vehicle in comparison to the A-train is well documented.[5,6,9,14] Roll stability, per se, is also improved, particularly with the rigid fifth wheel. Offtracking performance is somewhat degraded relative to the A-train, and many practical considerations of cost, frame stressing, incompatibility of existing trailers, etc., serve to limit the applicability and acceptance of the B-train.

a. Non-Steering B-Dollies.

Non-steering B-dollies provide a "first approximation" of the B-train. Rather than having a single-point pintle hitch, B-dollies are equipped with a forward frame extension, or "double drawbar," which connects to the lead trailer at two points separated laterally at about the spacing of the frame rails. In yaw, the B-dolly is, effectively, a rigid frame extension of the lead trailer. In pitch, however, it maintains the pintle-like articulation joint and from a strictly practical view, it is a separate, detachable piece of hardware. This latter feature provides many cost, operating, and logistical advantages. B-dollies are made in both "converter" style and in turntable style. Both versions are shown in figure 15. As with A-dollies, converter dollies are a separate dolly using a conventional fifth-wheel connection to the second semitrailer. The double drawbar is formed by a rigid extension of the dolly frame. The turntable-style dolly mates with its trailer using a turntable bearing and is, more or less, permanently attached to the trailer. Since there is no pitch freedom at this joint, the double drawbar is attached to both the front trailer and the dolly by joints which have a pitch degree of freedom.

Sliding B-dollies are a modification of the simple, non-steering B-dolly. When used between trailers in multitrailer trains, these devices function equivalently to standard B-dollies. When not in use, however, sliding B-dollies remain part of the towing trailer, and, in a manner similar to the operation of sliding trailer suspensions, the dolly may be slid

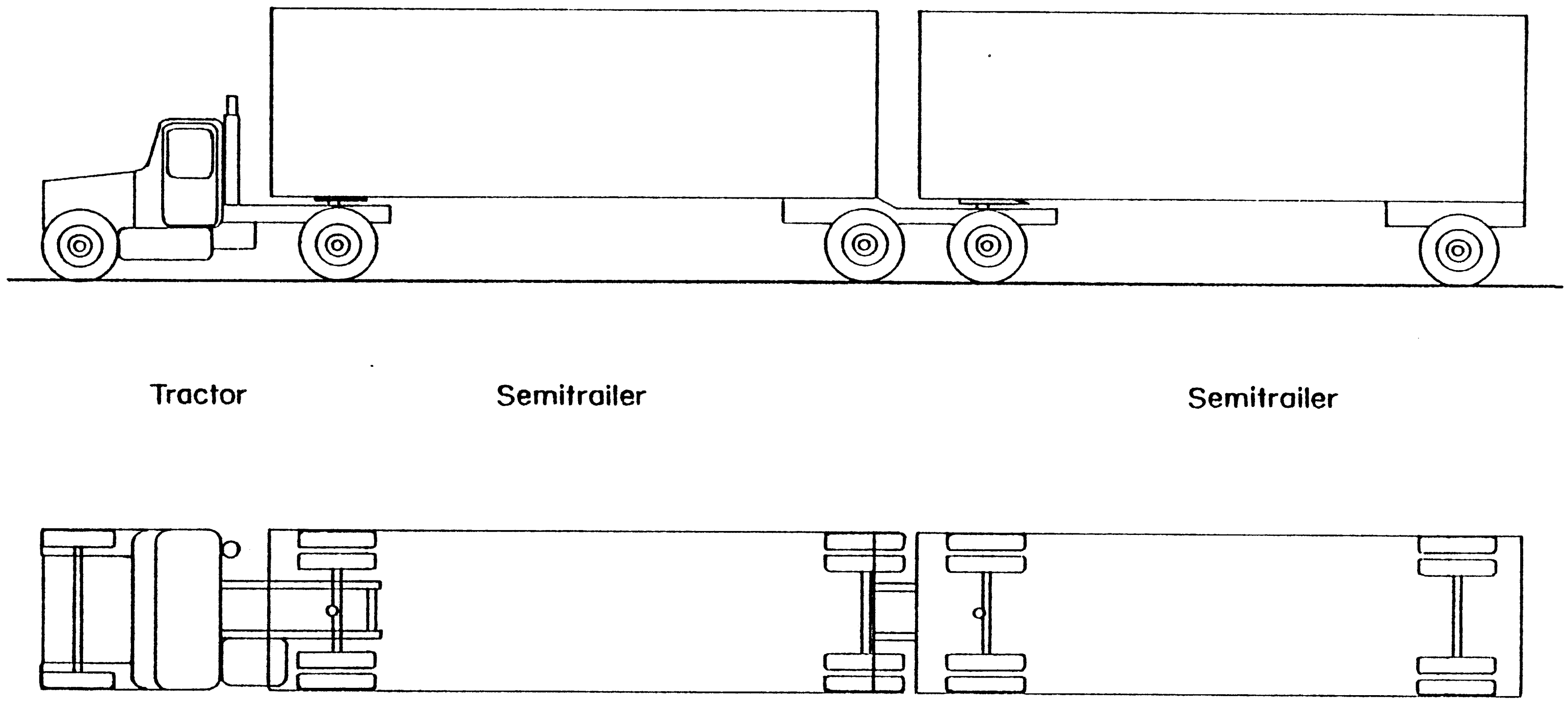


Figure 14. A B-train is composed of a tractor towing two or more semi-trailers. The towing trailers have an extended frame with 5th wheel for attaching the next trailer made of a B-dolly and semitrailer.

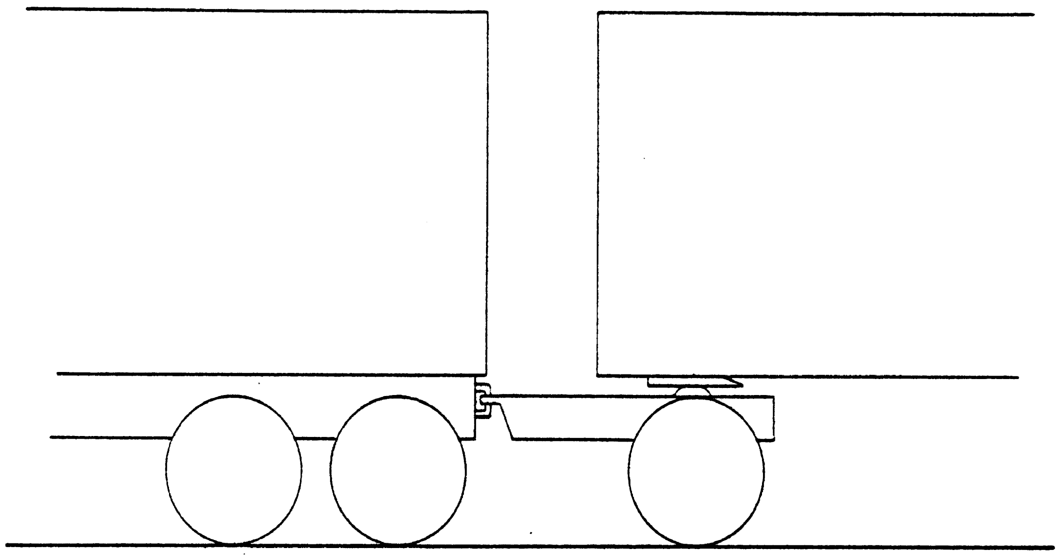


Figure 15a. Converter style B-dolly.

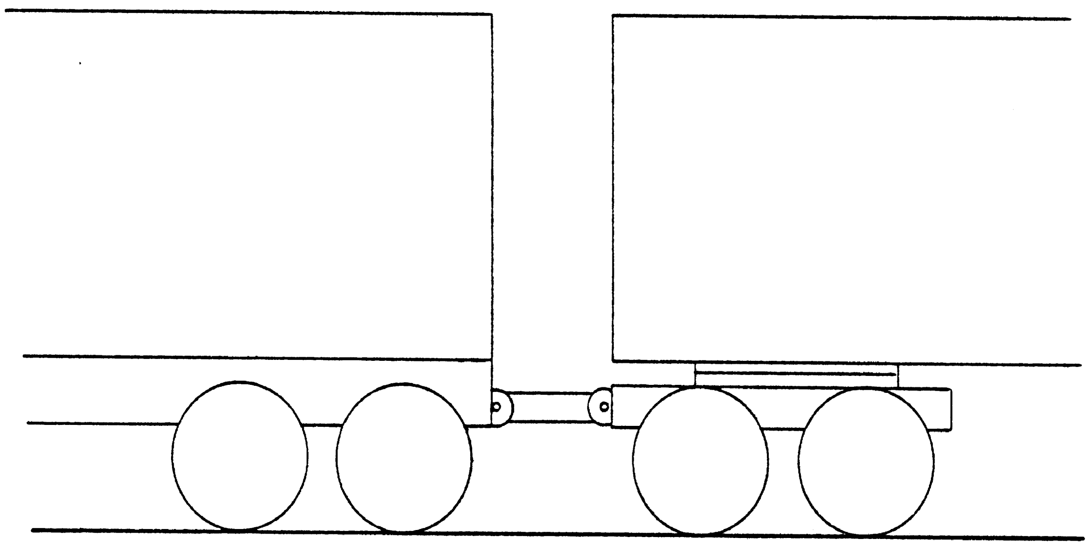


Figure 15b. Turntable style B-dolly.

forward, under the cargo area of the towing trailer. The practical advantages of sliding dollies have to do with times when the doubles vehicle is "broken down." The dolly is no longer a "loose" piece of equipment but conveniently remains with the first trailer. It can serve to "convert" the first trailer from a single- to a tandem-axle trailer. When slid underneath the cargo area, the dolly does not interfere when the trailer is backed into loading docks.

With respect to their dynamic yaw and roll performance, B-trains, fixed B-dollies, and sliding B-dollies are virtually indistinguishable. Parametrically, there may be some differences. For example, the geometry of the train layout, particularly the position of the axle on the towing trailer (or trailer and dolly), varies between B-trains and different B-dolly applications. Among B-dollies, the rigidity of the frame and coupling affects the rigidity of the roll connection between trailers. In the simulation study, axle locations appropriate to a B-train and B-dolly configurations are considered. The influence of variations in the rigidity of the trailer-to-trailer roll coupling is also examined.

b. Steerable B-Dollies.

"Steerable" B-dollies are a major variation of the B-dolly which are rapidly gaining popularity in Canada. Their structure and coupling mechanism are identical to the non-steering B-dolly, but the axle or axles on the dolly are equipped with a caster steering mechanism. With the so-called "auto-steering" (i.e., "automotive") style (figure 16), the steering freedom is provided by a kingpin and steering knuckle arrangement similar to that found on truck steering axles. "Turntable"-style steering (figure 17), on the other hand, involves a rigid axle pivoting, relative to the dolly frame, about a central, castered kingpin. In practice, both types of steering mechanisms are generally equipped with a "centering" spring device of some sort. (Three examples of centering devices used on auto-steering axles are shown in figure 18.) These mechanisms, along with the varying levels of Coulomb friction developed in the kingpin joints, provide a moment resisting steering which must be overcome by tire forces acting about the caster pivot. In general, resistance to steering is set sufficiently high such that, at highway speeds, little or no steering takes place, and dynamic performance is effectively that of a B-train. However, steering does occur in large amplitude maneuvers so as to significantly mitigate low-speed offtracking, tire scuffing, and frame stress problems that otherwise arise in the operation of B-trains and non-steering B-dollies. In practice, the performance of the steerable B-dolly depends on the compromise implied by these conflicting requirements. The system parameter of interest in the simulation study is the level of steering resistance.

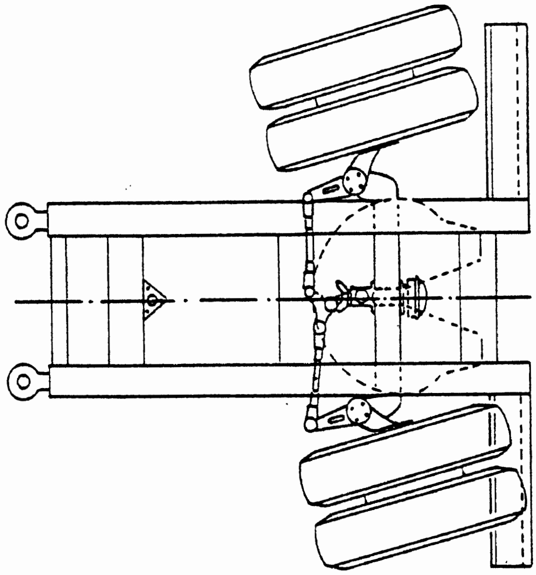


Figure 16. Auto-steering B-dolly.

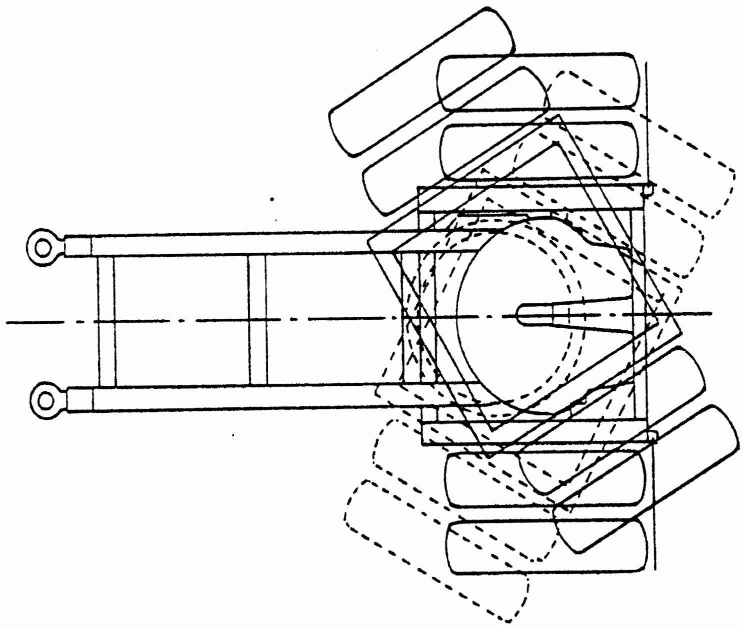


Figure 17. Turntable-steering B-dolly.

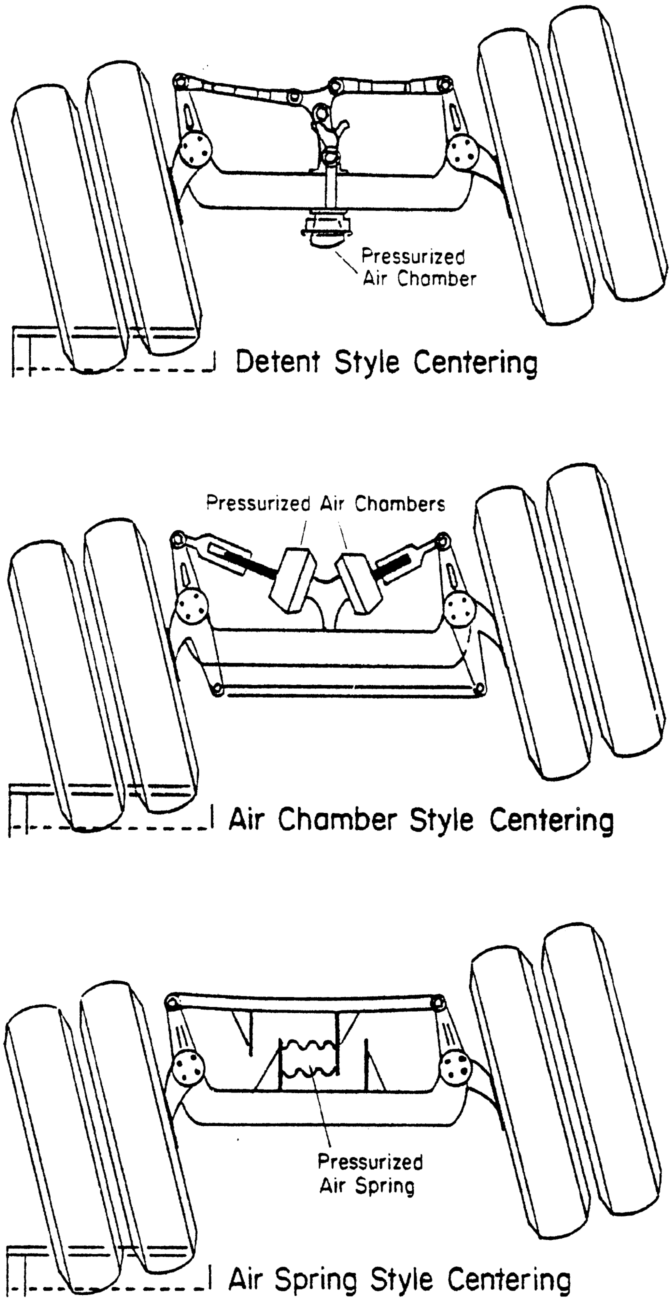


Figure 18. Three types of centering devices for the auto-steering axle.

The steerable B-dollies are susceptible to a unique performance problem related to braking. The B-dolly axles steer in response to torque about the steering pivot, which result from tire forces acting at some distance from the pivot. Normally, the force of interest is tire side force acting at the caster length, and generally the steering moments produced by left- and right-side tire forces will be additive. But steering moment may also be generated by braking force acting at the kingpin offset dimension. Normally, left- and right-side torques deriving from braking forces tend to cancel, but if brake force is unbalanced side-to-side, a net steering torque will result. A brake force imbalance of 20 percent is not uncommon on heavy-duty vehicles due to brake property variations, and much greater imbalances may result from differences in tire/road friction side to side. Given a certain level of brake imbalance, the sensitivity of the system response will depend, in large part, on the kingpin offset dimension. Accordingly, dollies employing the turntable steering mechanism are far more sensitive to unbalanced brake forces, since its kingpin offset dimension is equal to half of the track width of the axle.

Table 1 summarizes the innovative hitching mechanisms which were selected for study in the simulation activity. The candidates fall into two major groupings, viz., modified A-dollies and B-dollies. Modified A-dollies retain the yaw articulation degree of freedom at the first trailer-to-dolly connection, while B-dollies eliminate this articulation. Subgroups, defined by generic operating concepts, have been identified for each of the major categories. In many cases, a number of specific mechanical designs are known to exist within each subgroup. However, the interest herein is focused on the performance potential of the concept, rather than on the performance of any specific design example.

THE SIMULATION STUDY

In this project, simulation methods were used to address two separable, if not independent, technical subjects, viz.:

1) the evaluation of the dynamic qualities of multitrailer combination vehicles utilizing the innovative coupling mechanisms identified in task A. (The influence of such mechanisms on rearward amplification, and ultimately on dynamic roll stability, was of primary interest. The influence of the use of new coupling devices on the structural loading patterns to which dollies, trailers, and the couplings themselves might be subjected was also examined.)

2) evaluation of the influence of mixing hardware of different widths (96 and 102 in (2.44 and 2.59 m)) in otherwise conventional A-train combination vehicles. (Issues of static and dynamic roll stability and lightly damped yaw oscillations are of concern.)

Most of the simulation runs conducted on behalf of the "innovative couplings" subject employed a simulation program known as the UMTRI "Yaw/Roll" model.(16) In addition, UMTRI's simplified offtracking model was utilized.(17) Further, a limited number of braking runs were made using the UMTRI "Phase IV" model.(18) UMTRI's "Static Roll,"(19) "Yaw/Roll," and "Phase IV" simulation programs were utilized to examine the second subject, namely, the "width study."

The vehicle configuration used as the "test vehicle" in the simulation study is described below. The conduct and results of the simulation study, addressing the two subjects indicated above, are discussed later in this section.

1. The "Test" Vehicle

It was recognized that the primary motivation for this investigation of the performance characteristics of new coupling devices was the enactment of the Surface Transportation Act of 1982, most specifically, that portion of the act which permits the nationwide use of the twin, 28-foot- (8.53-m-) trailer combination vehicle (the Western double). It is assumed that this configuration of commercial vehicle will expand in use greatly in the coming years. Accordingly, now is an opportune time to introduce new coupling hardware which can improve the dynamic quality of this vehicle. Thus, the Western double, using a conventional A-dolly, was seen as the appropriate reference, or baseline vehicle, against whose performance new-configuration vehicles should be judged. Further, this same vehicle should serve as the "test buck" on which the subject hitching mechanism would be "installed" for evaluation in the simulation study.

Most of the calculations were done using the "Western Double," shown in figure 19, as the test vehicle. The geometry shown in the figure is typical. Most runs were conducted with the vehicle "fully loaded." Other runs were made with the first or second, or both, trailers empty. The data shown in the figure, correspond to a full load of medium-density freight. These weights and dimensions were used for all dolly types. (In current practice, most of the innovative dolly types weigh several hundred to a thousand pounds more than a typical A-dolly. Such differences were not taken into account, since our interest was in evaluating the generic qualities of the innovative dollies.)

Figure 20 shows two modifications of the test vehicle which were also considered in the study. The vehicle in figure 20.a is a hypothetical B-train similar to a Western double. In addition to the absence of a dolly, the primary difference between the B-train and the baseline configuration is the location of the third and fourth axles. Figure 20.b shows the "long-tongue" variation of the baseline vehicle, which was used in the in-depth study to demonstrate the influence of towbar length in certain configurations.

The same tires were assumed to be used on all vehicles. They were described on the basis of test data obtained on the Michelin 10.00 R 20 steel-belted, radial-ply tire.

Appendix A contains several "data echo" listings showing examples of complete parametric descriptions of the vehicle used in the simulations performed with the "Yaw/Roll" program.

2. Findings with Respect to the Use of Innovative Coupling Mechanisms.

In order to produce findings of maximum utility, a decision was made to evaluate the performance potential of each of the *generic types* of dollies rather than to examine the performance of only the specific examples that had been identified. The simulation programs were modified to allow simulation of the generic quality of the dollies, rather than simulating the specific mechanical layout of the examples found in the survey. For each dolly type, the characteristic property of the dolly was varied over a broad range, so that the performance potential of the concept could be evaluated. At a minimum, the range of variation of this basic parameter would cover all of the individual examples.

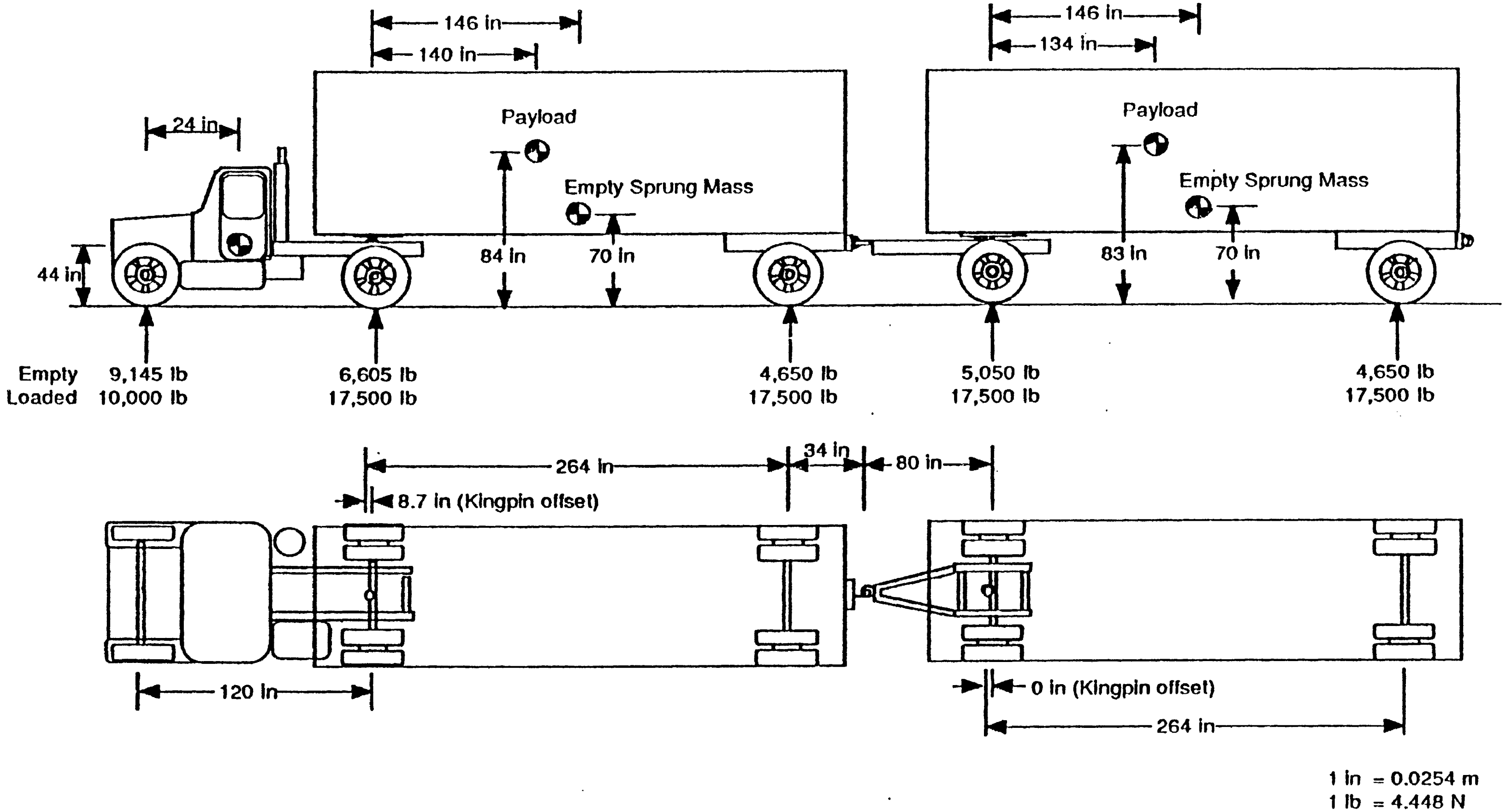
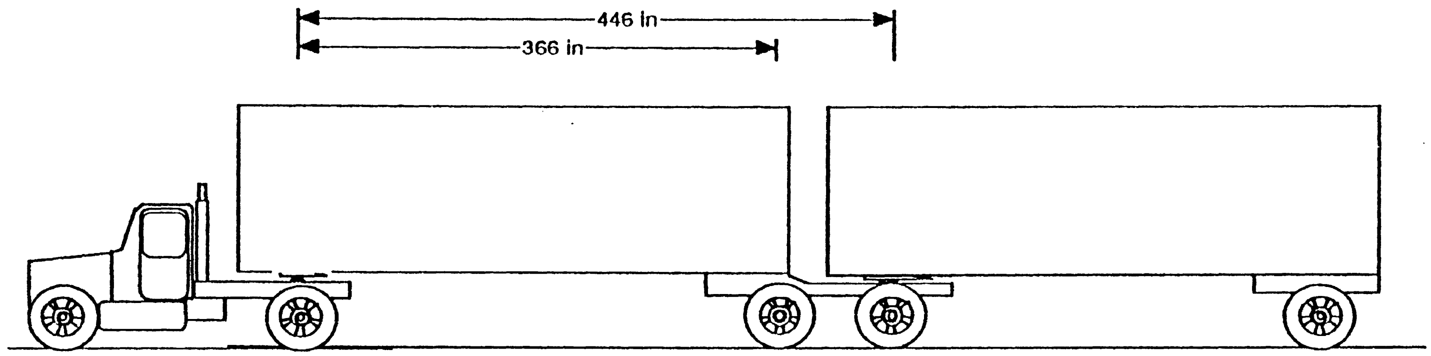
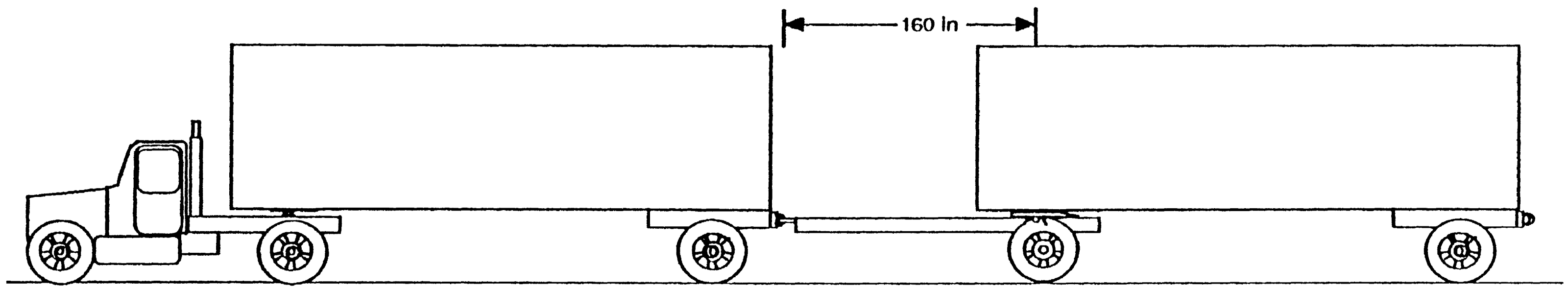


Figure 19. The baseline simulation test vehicle : the western double.



a. A B-train western double.



1 in = 0.0254 m

b. A western double with a long drawbar.

Figure 20. Two modifications to the baseline vehicle.

Accordingly, UMTRI's Yaw/Roll and Phase IV simulation models were altered to include generalized features which would allow representation of each of the dolly types. The "shifted instant center of rotation" (IC) group of dollies could already be accommodated using the basic A-dolly simulations with artificially shifted hitching points.

A forced-steering feature was added to the simulations to accommodate the various forced-steer dollies. This feature causes the tires of the dolly axle to be steered as a linear function of yaw articulation angle at the pintle. That is:

$$\delta_4 = G_{\delta_4\Gamma_2} \times \Gamma_2 \qquad (1)$$

where:

$G_{\delta_4\Gamma_2}$	is the system gain, a simulation input parameter.
δ_4	is the steer angle of the dolly tires.
Γ_2	is the yaw articulation angle at the pintle hitch.

By entering the appropriate system gain, as determined by a specific steering linkage, any of a range of forced-steer dollies could be simulated.

A similar option, which caused the yaw articulation angle at the pintle to be a linear function of the yaw articulation angle at the dolly fifth wheel, was added to both models. This model introduced the necessary yaw moments across the pintle and dolly fifth wheel joints so that the following geometric constraint resulted:

$$\Gamma_2 = G_{\Gamma_2\Gamma_3} \times \Gamma_3 \qquad (2)$$

where:

$G_{\Gamma_2\Gamma_3}$	is the system gain, a simulation input parameter
Γ_2	is the yaw articulation angle at the pintle hitch
Γ_3	is the yaw articulation angle at the dolly fifth wheel.

The introduction of this constraint enables a full range of linked articulation dollies to be simulated. Note that, as a special case, $G_{\Gamma_2\Gamma_3}$ may be set equal to zero and yaw articulation at the pintle is effectively eliminated. Thus a B-dolly is simulated. Further, if $G_{\Gamma_2\Gamma_3}$ is made very large, yaw articulation at the dolly fifth wheel is virtually eliminated, and a skid-steer dolly results

A self-steering axle option had previously been installed in the Yaw/Roll model,(14) and a similar option was added to Phase IV. In the Yaw/Roll model, this option permits the steering resistance functions to be entered as a table of centering torque versus steering displacement. Coulomb friction is also represented and the mechanical caster, in inches, must be entered. In Phase IV, the tabular input is replaced with a simpler, linear spring function. The lateral location of the kingpin is included so that the influence of brake force imbalance can be introduced. These self-steering axle options could be applied in simulating a self-steering B-dolly and in simulating the K-train.

Finally, a linear, spring function was added at the pintle hitch so that the influence of roll-stiffened pintle hitches, and the influence of the roll stiffness of B-dolly frames, could be examined.

The innovative-coupling study was divided into two efforts, viz., (1) a "baseline" or screening study and (2) an in-depth study. All of the innovative coupling techniques reviewed earlier, with the exception of extended-towbar dollies, were examined in the screening study. (The effort required to account for the extended-towbar feature would have been substantial, and, based on previous work, it was concluded that this feature would not substantially influence rearward amplification behavior or the rollover limit.) The screening study examined the rearward amplification and dynamic roll stability of the fully loaded vehicle, along with low-speed offtracking performance. Three of the more promising coupling mechanisms were selected for further study in the in-depth effort. The in-depth effort (1) expanded the range of study using the same performance measures, and (2) examined additional performance measures and properties, including stability during braking and issues of structural loading. The in-depth activities were aimed at the final objectives of (1) providing a broader performance analysis of the more promising types of dollies, and (2) identifying an "optimum" parametric description of a dolly, which could be translated into hardware in the prototype development portion of the project. Accordingly, the in-depth study included the study of a prototype dolly concept, as well as the three selected "commercial" dolly concepts.

a. The Screening Study: Study Vehicles.

Table 2 identifies the vehicles studied. The table names the vehicle, or dolly, and gives a description of the generic quality and the associated simulation parameter of interest. The "shorthand" code used to identify the specific simulation "test vehicles" is given, as is the parameter value which distinguishes that vehicle. Figure 21 aids in defining the characteristic parameters of IC-group, forced-steer, and linked-articulation dollies.

Figure 22 shows the steering resistance of self-steering axles adopted as the reference characterization. The steering resistance properties shown, combined with 5 in (0.13 m) mechanical trail (caster) of the steering system and, approximately, 2 in (.05 m) of

Table 2. The Screening Study Vehicles

Dolly Type	Generic Description and Simulation Parameter Of Interest	Simulation Study Code	Parameter Value
A-trains			
A-dolly	Baseline vehicle, twin 18 ft (5.49 m) van trailer, 5 axle A-train	AT	Baseline parameter set x_{IC} = 160 inches
Shifted IC dollies	Pintle hitch mechanism causes the instant center of yaw rotation of the dolly and the first trailer to shift away from the hitch point. The parameter variation of interest is the longitudinal position of this instant center of rotation (IC) in the first trailer coordinate system.	IC1 IC2 IC3 IC4	$x_{IC} = 0$ $x_{IC} = 62$ x_{IC} = 124 inches x_{IC} = 200 inches
Forced-steer dollies	Steering linkage provided which causes the tires of the dolly to steer as a direct result of yaw articulation at the pintle. The parameter variation of interest is the steering system gain.	FS1 FS2 FS3 FS4	$G_{\delta 4\Gamma 2} = 0.75$ $G_{\delta 4\Gamma 2} = 1.50$ $G_{\delta 4\Gamma 2} = 2.25$ $G_{\delta 4\Gamma 2} = 3.00$
Linked-articulation dollies	A linkage is added to the basic A-dolly which causes a fixed relationship to be established between the yaw articulation angle at the pintle and the yaw articulation angle at the dolly fifth wheel. The parameter variation of interest is the gain of the articulation angle relationship.	LA1 LA2 LA3 LA4	$G_{\Gamma 2\Gamma 3} = 0.6$ $G_{\Gamma 2\Gamma 3} = 1.3$ $G_{\Gamma 2\Gamma 3} = 2.0$ $G_{\Gamma 2\Gamma 3} = 2.6$
Skid steer dollies	The yaw articulation freedom of the dolly fifth wheel is eliminated. The tires of the basic skid steer dolly do not steer. K-train versions provide for self-steering (castering) of the dolly axle. The parameter variation of interest is the level of steering resistance of the dolly axle.	SS K1 K2	Non-steering axle* Self-steering axle with reference steering resistance. Self-steering axle with no steering resistance.

1 in = 0.0254 m

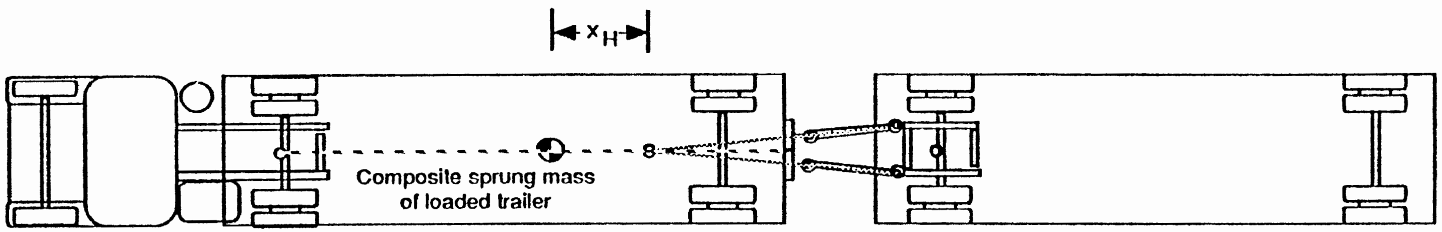
*Equivalent to LA dolly with $G_{\Gamma 2\Gamma 3}=\infty$

Table 2. The Screening Study Vehicles (continued)

Dolly Type	Generic Description and Simulation Parameter Of Interest	Simulation Study Code	Parameter Value
Roll stiffened pintle hitches	The standard A-dolly is equipped with a pintle hitch providing roll coupling between the dolly and first trailer. The parameter variation of interest is the level of roll stiffness across the hitch.	RR	$K_{2xx} = 10^6$ in-lb/deg (Roll rigid)
		RC1	$K_{2xx} = 60{,}000$ in-lb/deg
		RC2	$K_{2xx} = 30{,}000$ in-lb/deg
		RC3	$K_{2xx} = 15{,}000$ in-lb/deg
B-trains			
B-train	A tractor-semitrailer-semitrailer doubles with no separate dolly and with conventional fifth wheel couplings between units. Virtually equivalent to a non-steering B-dolly with rigid roll coupling.	BT	$K_{2xx} = \infty$
Roll compliant (non-steering) B-dolly	Dolly which eliminates the yaw articulation and provides roll coupling at the pintle hitch. The dolly axle has no steering capability at its axle. The parameter variation of interest is the stiffness of the roll coupling.	B1	$K_{2xx} = 30{,}000$ in-lb/deg
		B2*	$K_{2xx} = 0$ in-lb/deg
Steerable axle B-dolly	As above, but the tires of the dolly axle are allowed to steer by caster. A steer centering-spring mechanism is included. The steering resistance of the centering mechanism must be overcome for steering to occur.	SA1	Reference steering resistance.
		SA2	1/2 of reference steering resistance.
		SA3	No steering resistance.

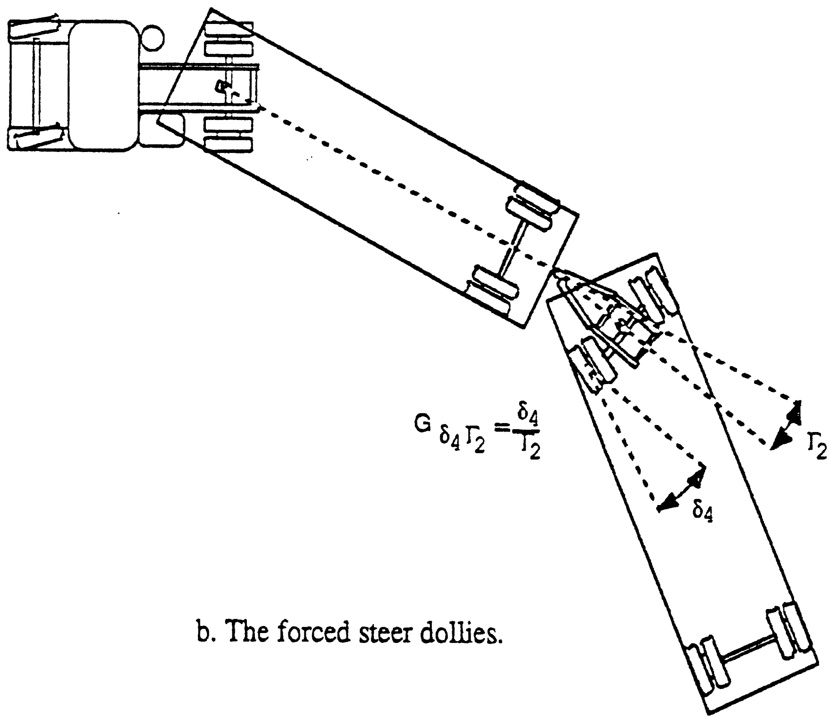
1 in-lb/deg = 0.113 N-m/deg

*Equivalent to LA dolly with $G_{\Gamma_2\Gamma_3}=0$

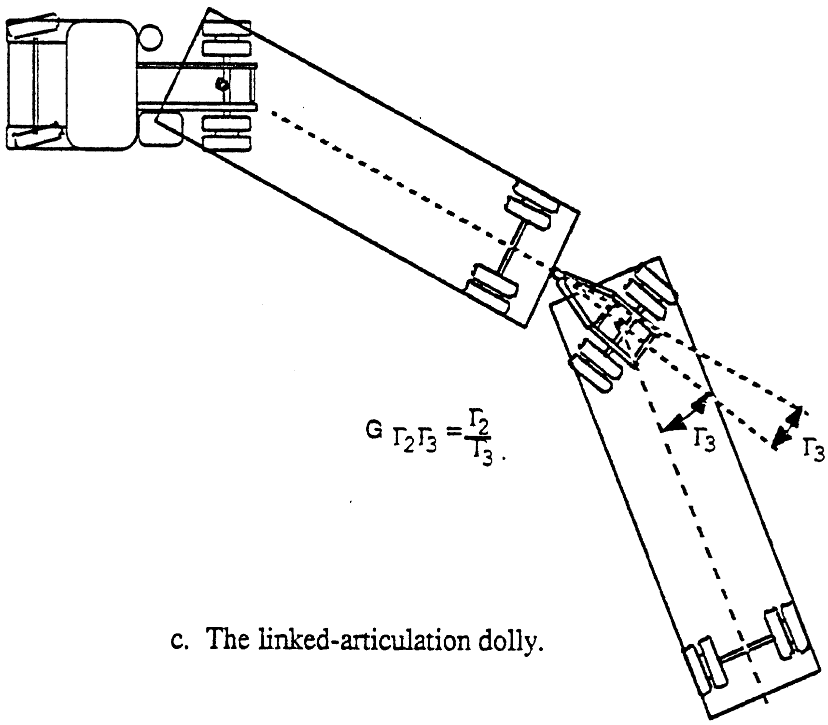


a. The shifted IC dollies.

Figure 21. Characteristic parameters of three types of screening study dollies.



b. The forced steer dollies.



c. The linked-articulation dolly.

Figure 21.(contd). Characteristic parameters of three types of screening study dollies.

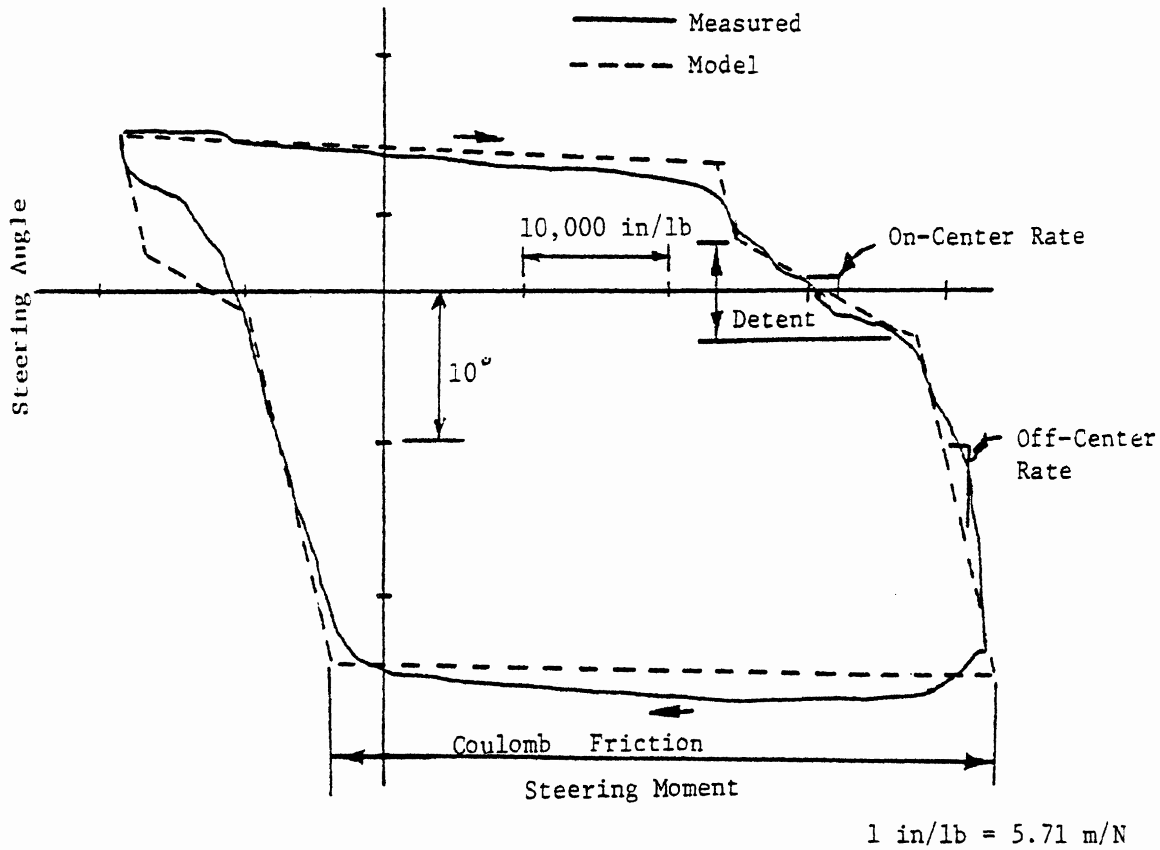


Figure 22. Steering resistance performance of the reference self-steering axle.

pneumatic trail, provide a steering resistance function which effectively resists steering of the B-dolly tires until lateral tire forces on the B-dolly total about 5,200 lbs (23,130 N), representing a lateral friction utilization coefficient of about 0.3, given an axle load of 17,500 lb (77,840 N).

The first vehicle listed is the baseline Western double with a conventional A-dolly. According to the philosophy of the study, the performance of this vehicle is the reference against which the performance exhibited by the remaining vehicle types is compared.

The second dolly category represents modified A-dollieswhose effective center of rotations (pintle hitch) have been shifted. This category includes all of the four-bar linkage hitches, including the double-cross hitch, plus the roller-cam hitch. Given that these hitches are conceptually equivalent, notwithstanding their use of different mechanical elements, the study simply examined the influence of the location of the instant center on dynamic performance.

The remainder of the modified A-dolly vehicle types are also listed in table 2 as subjects for the screening study. These are the roll-rigid A-dolly, dolly types with no fifth wheel articulation, forced-steer A-dollies, and dollies with linked pintle and fifth wheel articulations. For each of these types of dollies, the numerical range of the identifying characteristic whose influence was examined in the screening study is identified.

Table 2 includes the B-train, the non-steering B-dolly, and the steerable B-dolly as the final entries. The functional difference between the B-train and the non-steering B-dolly, at the level of investigation pursued in the screening study, is minimal, since the additional pitch articulation added by the B-dolly hitch is of little import in the issues examined in the screening effort. However, the level of roll compliance across the B-dolly hitch (assumed to be considerably greater than that of the equivalent section of trailer frame in a B-train) may be of some importance to dynamic roll stability. The influence of this parameter was briefly examined.

The primary parameter of interest of the steerable B-dolly is the level of resistance to steering. At the time of the screening study, little data were available to describe the actual performance of the steering-resistant properties of self-steering axles. Since this study was not funded to develop such data, the one set of such data, previously developed by UMTRI (see figure 22), was used as reference.(14) Although a casual examination of self-steering axles shows that the design intent is for steering resistance to derive from an "on-center preload" of a centering spring mechanism, the measurements indicate that Coulomb friction (generated apparently at the kingpin bearings consisting of brass bushings) provides the larger amount of steering resistance at higher axle loads. In the screening study, vehicle performance was examined at three levels of simulated steering resistance: zero resistance, 1/2 of the reference level, and the reference level.

b. The Screening Study: Results.

The dynamic portion of the screening study consisted of a set of simulation runs planned to produce measures of both rearward amplification and dynamic roll stability for each of the subject vehicles. The matrix consisted of (1) a "frequency sweep" of sine-steer, or lane-change-like, maneuvers, conducted at 55 mi/h (88.5 km/h) and at low levels of lateral acceleration, for characterizing rearward amplification, and (2) an excursion into higher levels of lateral acceleration, using the same maneuver, to examine dynamic rollover stability limits.

The sine-steer maneuver was conducted by using a "path-follower" option in the program. The lateral displacement and length of the predefined "lane-change" path were chosen such that, at 55 mi/h (88.5 km/h), the time history of the lateral acceleration of a vehicle exactly following the path would be a sine wave of the appropriate magnitude and frequency. Example data showing the paths and the acceleration time histories of the tractor and second trailer during such a maneuver are shown in figure 23. The definition of rearward amplification is illustrated in the figure. The limitations of the path follower are evident, in that the tractor acceleration time history is not a perfect sine wave.

Rearward Amplification. The results of the screening study simulation runs examining rearward amplification are shown in figures 24 through 30. Each of these figures is a plot of the rearward amplification properties of the subject vehicles as a function of frequency. Each individual plot shows the performance of a specific group of test vehicles (table 2).

The performance of the A-train is given in each plot as a reference. Indeed, the rearward amplification performance of the A-train, as shown in these figures, is indicative of "the doubles problem." At very low frequencies, both trailers follow the path of the tractor virtually exactly, so that rearward amplification is (the very desirable) unity. The rearward amplification of the A-train reaches 2.37 at both 3 and 4 rad/sec (and surely somewhat higher in between). Thus, at these frequencies, the second trailer experiences maneuver levels which are more than twice as severe as those experienced by the tractor.

Regarding the study vehicles, the data in these figures indicate the following:

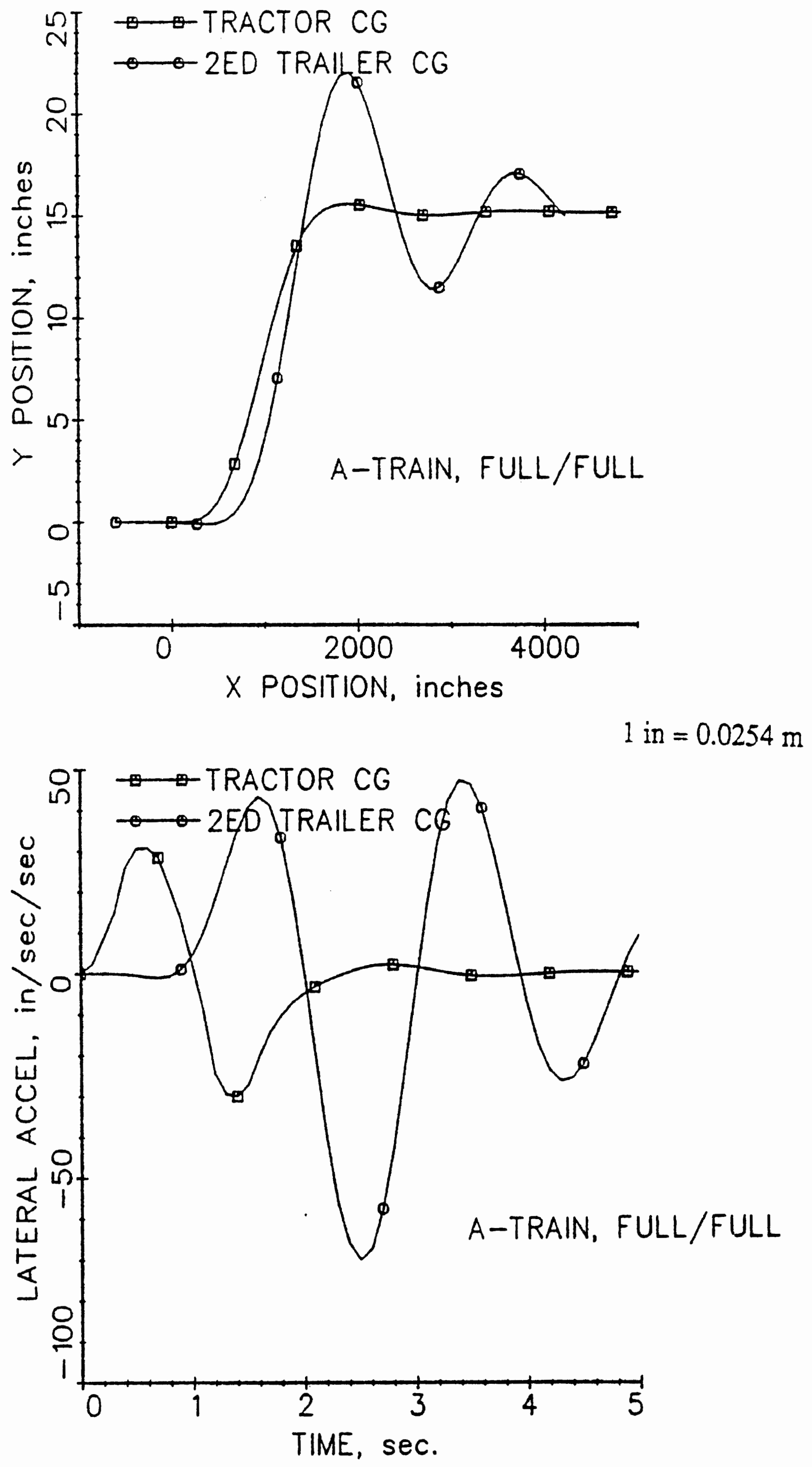


Figure 23. Example path and acceleration data from a lane-change maneuver performed with the Yaw/Roll model.

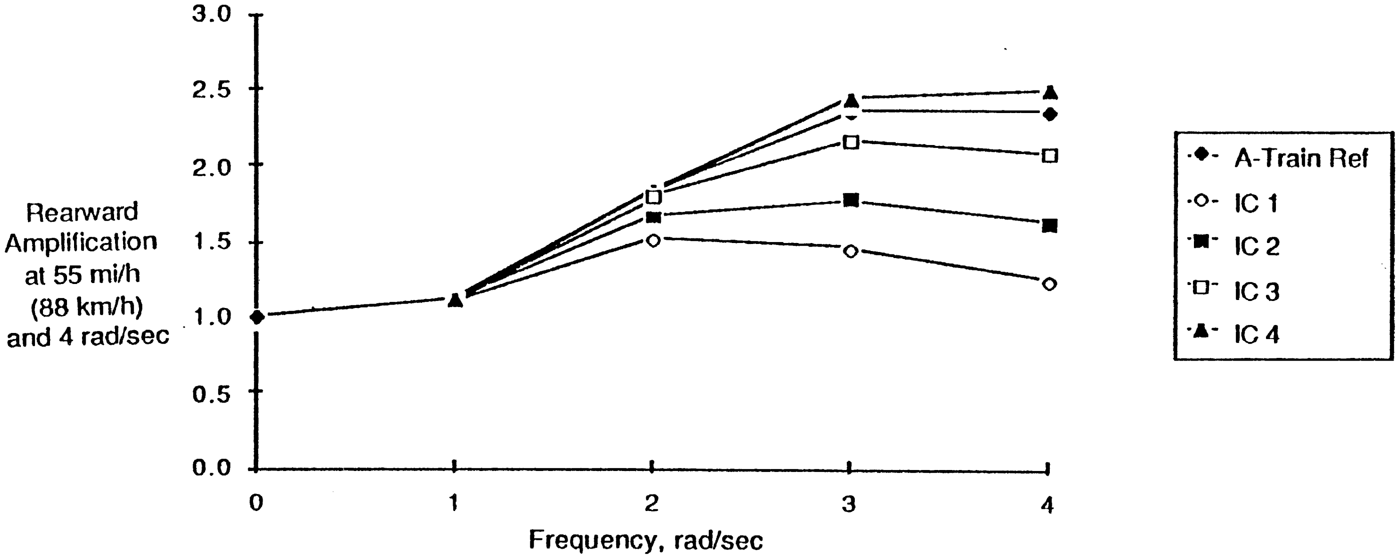


Figure 24. Rearward amplification in the frequency domain: the shifted IC dollies.

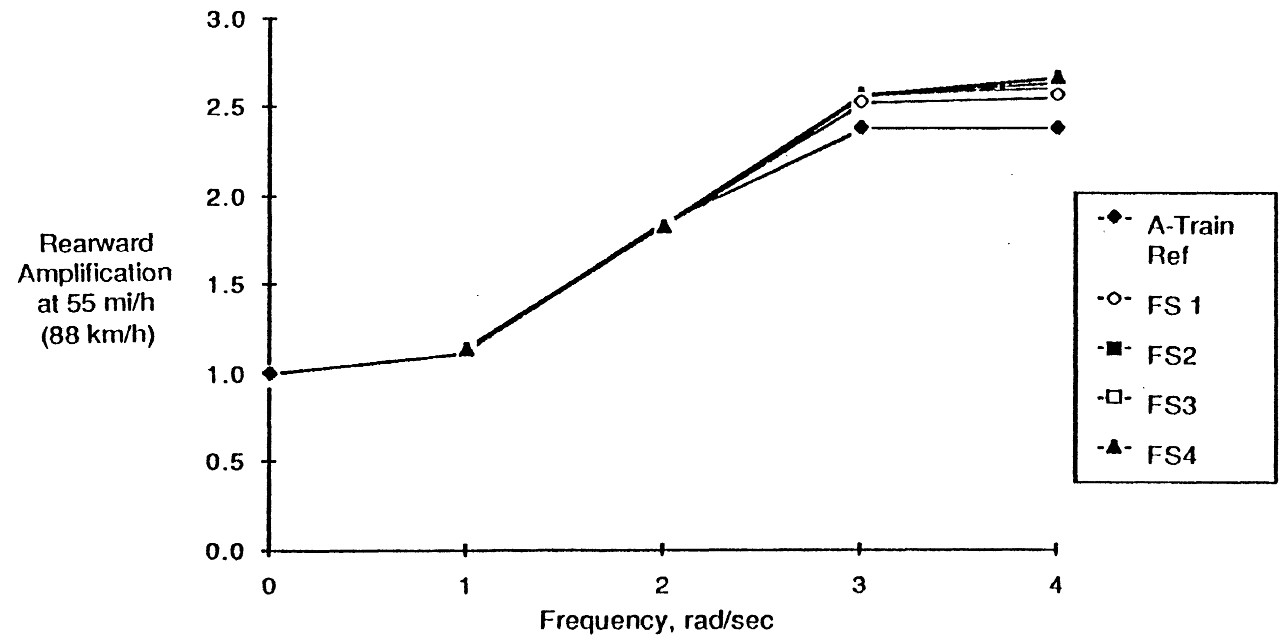


Figure 25. Rearward amplification in the frequency domain: forced-steer dollies.

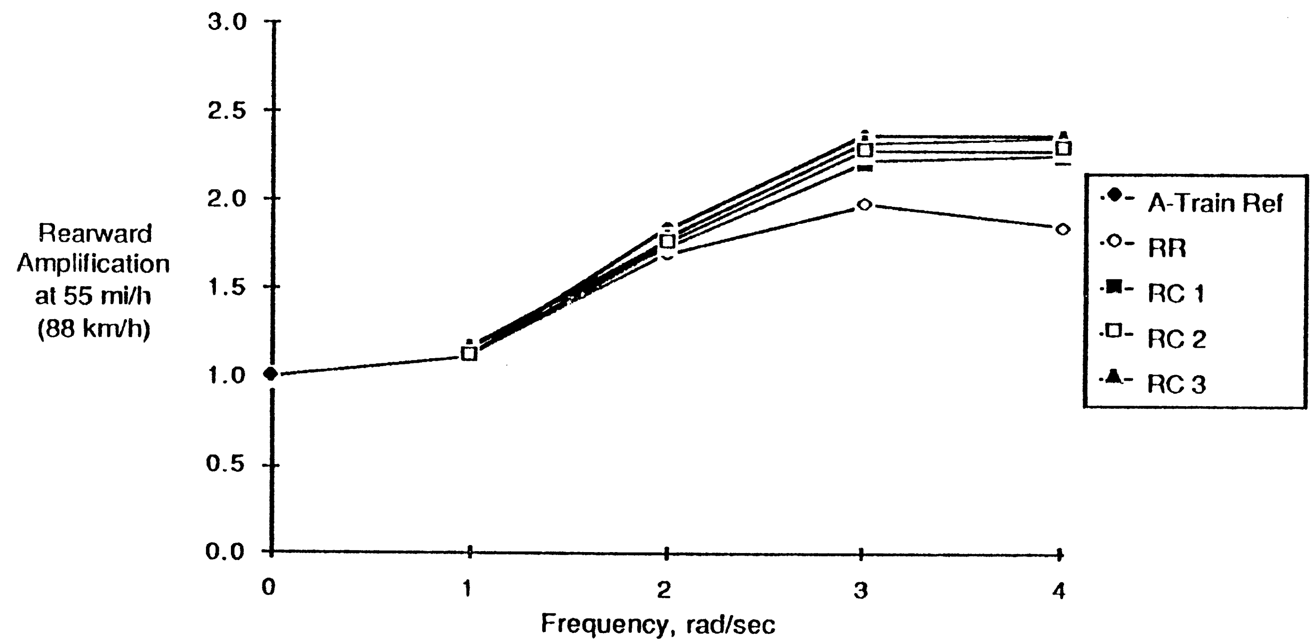


Figure 26. Rearward amplification in the frequency domain: the roll-stiffened pintle hitches.

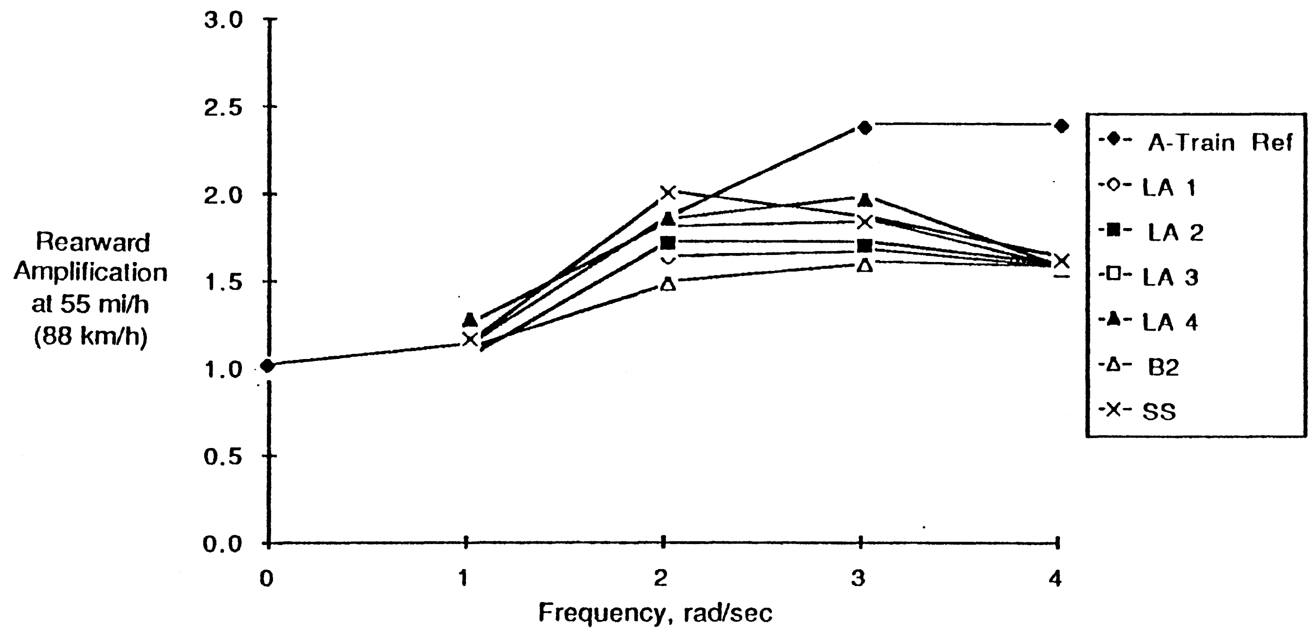


Figure 27. Rearward amplification in the frequency domain: the linked-articulation dollies.

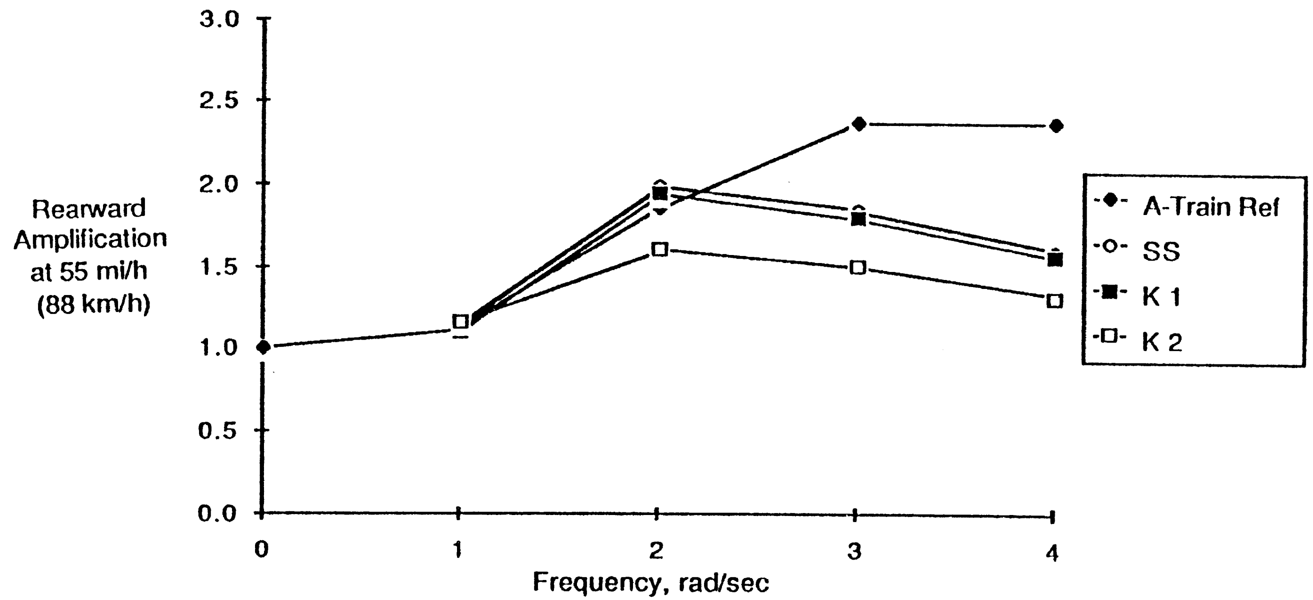


Figure 28. Rearward amplification in the frequency domain: the skid-steer dollies.

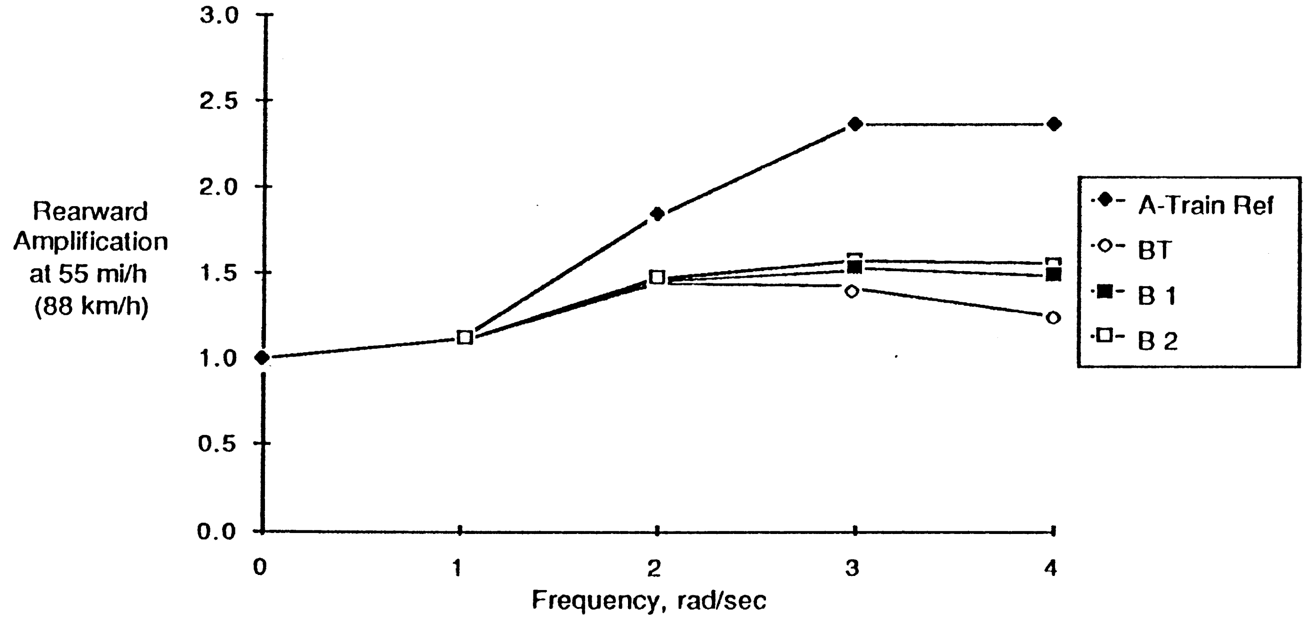


Figure 29. Rearward amplification in the frequency domain: the roll-compliant B-dollies.

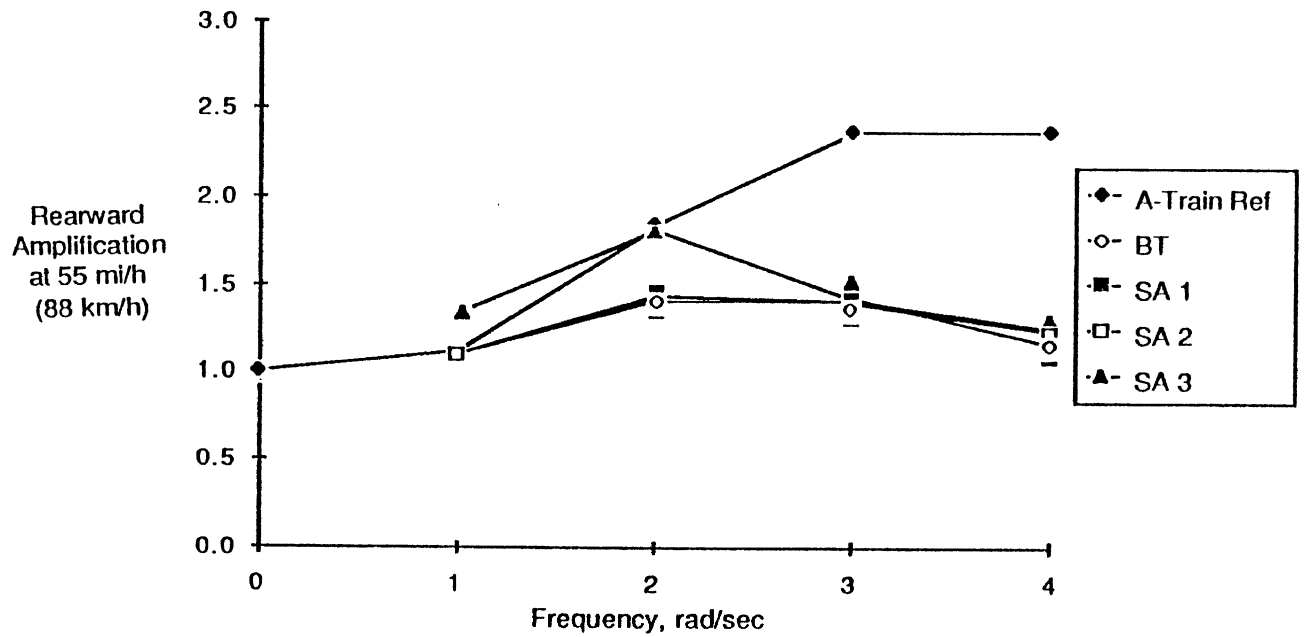


Figure 30. Rearward amplification in the frequency domain: the self-steering B-dollies.

- The IC Dollies (figure 24): The results shown in this figure confirm the previously known fact that as the instant center of rotation (the "effective hitch point") of the dolly moves forward in the frame of the first trailer, rearward amplification of the train is reduced. Fancher has clearly shown that the most important factor in this regard is the location of the IC in the first trailer, and not the effective length of the towbar.[12,13] The figure shows that the strongest influence is at the higher frequencies, so that peak rearward amplification is lowered, but it occurs at a lower (more often used) maneuvering frequency. Nonetheless,moving the IC forward reduces rearward amplification at all frequencies.

- The Forced-Steer Dollies (figure 25): The rearward amplification of each of the forced-steer dollies examined is slightly larger than the rearward amplification of the A-train. Since the steering gain of the dollies examined were all of the polarity in which the dolly tires steer toward the outside of a steady turn (and their slip angle and level of side force generation thereby tend to be reduced), the polarity of this finding is as would be expected. Nevertheless, the relatively low sensitivity of performance to steering gain was somewhat surprising. The influence of the parameter appears to be "saturating" within the range of study. This observation, and further analysis, led to one of the more interesting findings of the study, viz., the development of the "steer-point" concept (to be discussed later).

- The Roll-Stiffened Pintle Group (figure 26): These results indicate that providing realistic levels of roll coupling (RC1, RC2, and RC3) at the pintle hitch tends to decrease rearward amplification , very slightly. If a hitch (and frame) which were very rigid in roll (RR) could be applied, rearward amplification could be reduced appreciably. The explanation lies in nonlinear tire properties. It is well known that the nonlinear sensitivity of cornering stiffness of truck tires to vertical load results in a reduction of the total cornering stiffness of all the tires on a given axle as load is transferred from side to side due to rolling during cornering.[20] Since, in dynamic maneuvers, the roll response of the tractor-semitrailer and of the full trailer of the doubles tend to be substantially out of phase, coupling the two units together in roll tends to reduce the maximum roll of each. As a result, the extent of cornering stiffness reduction is also reduced. Fancher [10,12,13,15] and Ervin [21] have both shown that the reduction of cornering stiffness of the tires of a double generally tends to increase rearward amplification.

- The Linked-Articulation Dolly (figure 27): In addition to the four linked-articulation (LA) dollies of varying linkage gain, and the reference A-train, the performance of both the B2 dolly (a non-steering B-dolly with no roll coupling) and the skid-steer dolly (SS) are included in this figure. This is done since it has been observed that the B2 dolly is conceptually the equal of a linked-articulation dolly with an articulation gain of zero ($G_{\Gamma_2\Gamma_3} = 0$), and the skid-steer dolly is conceptually the equal of the linked-articulation dolly with an infinite gain ($G_{\Gamma_2\Gamma_3} = \infty$). (Note that all the vehicles represented in this figure have no roll coupling at the pintle hitch joint, so that only the influence of yaw mechanisms is being considered.) The results shown in the figure clearly affirm these conceptual observations. In general, the data show that removing a yaw degree of freedom at the dolly, either at the drawbar (B2), at the fifth wheel (SS), or "in between" (LA), aids in reducing rearward amplification. Judged by rearward amplification alone, the reduction of yaw articulation at the drawbar is preferable (and, as will be seen, other performance measures strongly support this choice). Further, linked articulation gains in the vicinity of unity and less seem to achieve nearly the level of improved performance as can be attained by elimination of pintle yaw articulation.

- The Skid-Steer Dollies (figure 28): Figure 28 shows that rearward amplification (as measured at the low amplitude of 0.1 g's) is reduced by both the basic skid-steer dolly and the K-train variations. When the K-train is equipped with a self-steering axle with the reference level of steering resistance, virtually no steering activity is experienced at the relatively low maneuvering level of these runs. Accordingly, the K1 vehicle performance is virtually identical to the SS vehicle. When the K-train dolly axle has no steering resistance (K2), rearward amplification is actually reduced further, but as will be discussed later, other dynamic performance characteristics of this vehicle are seriously degraded. Unlike other dollies being examined, the performance of the K-train is highly dependent on the nonlinear steering-resistance mechanism. Thus, performance measures such as rearward amplification are more readily susceptible to change as a function of the amplitude of the maneuver.

- The Roll-Compliant B-Dollies (figure 29): By comparing the performance of the reference A-train with several non-steering B-dolly configurations, figure 29 illustrates clearly the reduction of rearward amplification which results from the elimination of the yaw articulation degree of freedom at the pintle. The figure also illustrates that the rearward amplification advantage previously observed when roll coupling stiffness was added between the units of the A-train is also realized when roll coupling stiffness is added to the B-train.

- The Steerable-Axle B-Dollies (figure 30): The rearward amplification of the three steerable-axle B-dollies (SA) is shown in figure 30. For reference, the figure also includes the performance of the B-train as well as the A-train. As was the case with the skid-steer dollies, the highly nonlinear quality of the steering-resistance feature of the SA1 and SA2 dollies means that the results shown are, in part, dependent on the amplitude of the maneuver, and may vary for more severe maneuvers. At the low levels experienced here, virtually no self-steering action occurred on either the SA1 or SA2 vehicles, so that their performance is virtually identical to the B-train. When the B-dolly axle has no resistance to steer (SA2), rearward amplification in the 2 rad/sec range degrades to that of the A-train, but remains low at higher frequencies. Like the K2 vehicle, however, it will be seen that other performance measures of the B-dolly vehicle can be seriously degraded when steering resistance of the self-steering axle is very low.

The rearward amplification performance of all of the dolly types studied is summarized in figure 31. This figure is a bar chart presenting the *peak* rearward amplification displayed by each vehicle. The maneuvering frequency at which that peak occurred is indicated. The data presentation is arranged in "rank order" with the best performers (by this measure) at the left and the worst at the right. The rearward amplification performance of the reference A-train and of the B-train are emphasized.

The Steer-Point Concept. Before proceeding with other specific findings of the screening study, an explanation of a finding of a more general nature, viz., the concept of the "steer-point" of the full trailer, will be presented. The impetus for developing this concept derived from the somewhat surprising rearward amplification performance of the forced-steer dollies, as noted above.

Consider the following:

1) Fancher has observed that, in the analysis of the directional behavior of the trackless train, the so-called full trailer may be mathematically decoupled from the other elements of the train at the drawbar hitch point.[10,12,13,16] In physical terms, this mathematical decoupling is equivalent to the fact that lateral forces at the drawbar hitch point are so small as to be insignificant with regard to motivating trailer lateral or yaw motions. Rather, hitch forces provide only the power necessary to steer the tires of the front axle of the trailer; the

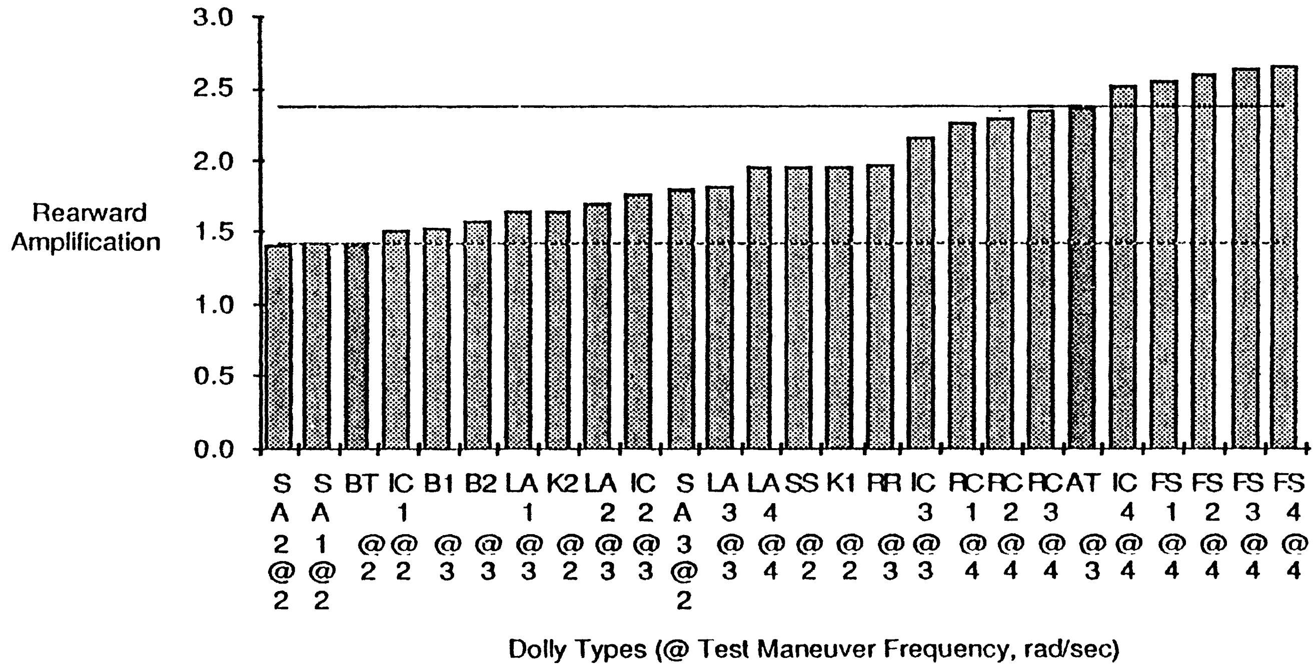


Figure 31. Peak rearward amplification of all the screening study vehicles.

trailer tires (front and rear), in turn, provide the lateral forces that actually motivate trailer lateral and yaw motions.

2) Fancher has also shown the significance of the full trailer hitch point as regards rearward amplification. He has shown that, while drawbar length is relatively unimportant, the location of the hitch point in the *towing unit* is very powerful as regards rearward amplification. As the towing point moves aft relative to the c.g. of the towing vehicle, rearward amplification of the train increases. The conventional arrangement with the hitch point several feet aft of the towing trailer's rear axle generally promotes large rearward amplification.

3) We add the simple observation that, for a conventional A-train and trailer, the "steering geometry" is such that the front axle of the trailer is always steered to point toward the hitch point in the lead trailer.

Combining these three facts suggests that the significance of hitch point geometry in the lead unit is not actually associated with the location of the "hitch" point, but with the location of the "steering" point.

If, indeed, the significance of the hitch point is its "steering" function rather than its "hitching" function, it follows that a similar effect on rearward amplification should be obtainable by other mechanisms which steer the full trailer front axle such that it points toward a "steer point" located forward in the towing trailer.

The model of figure 32 can be used to illustrate that a mechanism which steers the tires of the dolly axle as a linear function of the yaw articulation angle at the drawbar hitch point can provide either a forward or rearward shift of the dolly axle "steer point" away from the drawbar hitch point. That is:

H marks the towbar hitch point
S marks the dolly "steer point"

$$G_{\delta_4 \Gamma_2} \equiv \delta_4 / \Gamma_2 \qquad (3)$$

where:

δ_4 = dolly tire steer angle
Γ_2 = towbar articulation angle
$G_{\delta_4 \Gamma_2}$ = dolly axle steering system gain (shown positive in figure 31)

For small angles:

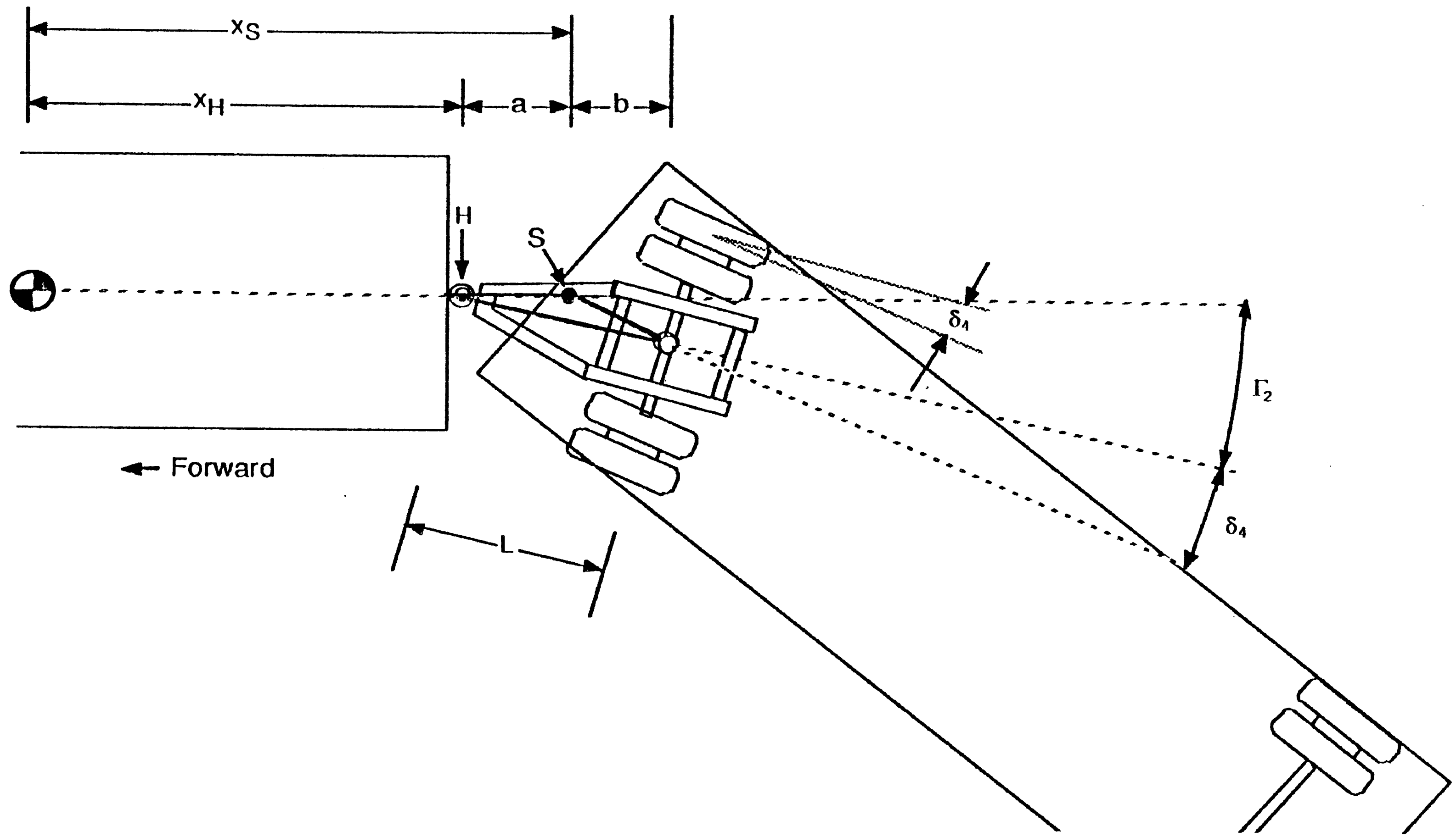


Figure 32. Schematic diagram illustrating the location of the steer point for forced-steer dollies.

$$a\Gamma_2 = b\delta_4 \quad (4)$$

$$\delta_4/\Gamma_2 = a/b = G_{\delta_4\Gamma_2} \quad (5)$$

$$a + b = L \quad (6)$$

Then

$$a = G_{\delta_4\Gamma_2} / (1 + G_{\delta_4\Gamma_2})\, L \quad (7)$$

$$x_S = x_H + G_{\delta_4\Gamma_2} / (1 + G_{\delta_4\Gamma_2})\, L \quad (8)$$

If	$G_{\delta_4\Gamma_2} < -1$	then	$x_S > x_H + L$	(9a)
If	$G_{\delta_4\Gamma_2} = -1$	then	$x_S = \pm\infty$	(9b)
If	$-1 < G_{\delta_4\Gamma_2} < 0$	then	$x_S < x_H$	(9c)
If	$G_{\delta_4\Gamma_2} = 0$	then	$x_S = x_H$	(9d)
If	$G_{\delta_4\Gamma_2} > 0$	then	$x_S > x_H$	(9e)
If	$G_{\delta_4\Gamma_2} \rightarrow \infty$	then	$x_S \rightarrow x_H + L$	(9f)

Figures 33 and 34 present simulation results that support the premise that it is the steer point, rather than the hitch point, or even the instant center of rotation of the dolly in the towing vehicle, that is the truly significant factor of A-dolly design influencing vehicle performance. The data presented in these figures derive from the performance of twelve test vehicles, viz., the reference A-train (AT), the four shifted-IC and the four forced-steer (FS) vehicles, and three additional special (SP) vehicles. The SP1 vehicle has both shifted-IC and forced-steer properties combined in one dolly. The SP2 and SP3 vehicles have IC's in the normal position, but use a negative steering gain to produce a forward steer-point position. Table 3 identifies the significant parameters of each dolly, including the steer-point position, according to the notation of figure 32.

Figure 33 shows the rearward amplification of these twelve vehicles as a function of steer-point location. The maneuver used to obtain these measures of rearward amplification was a lane change producing a tractor lateral acceleration time history of sinusoidal wave form with a magnitude of 0.1 g and a frequency of 4 rad/sec. The figure shows clearly that rearward amplification at a fixed velocity and frequency is approximately a linear function of the steer point, regardless of whether the steer point is established by special hitch-point or forced-steer properties. Figure 34 presents data which indicate a similar linear relationship between steer-point position and low-speed offtracking.

Given the validity of the steer point concept, equations (9e) and (9f) provide the explanation for the performance of the forced-steer dollies as presented in figure 25. Each of the forced-steer dollies shows rearward amplifications greater than the reference A-train, because each has a positive steering gain producing a steer point aft of the hitch point (9e), but the increase in rearward amplification is limited as steering gain increases because, as

Table 3. Significant Dimensions of the Simulated Vehicle of Figures 31 and 32.

Vehicle ID	X_H, in	L, in	G	X_S, in
AT	160	80	0.0	160
IC1	0	240	0.0	0
IC2	62	178	0.0	62
IC3	124	56	0.0	124
IC4	200	40	0.0	200
FS1	160	80	0.75	194
FS2	160	80	1.5	208
FS3	160	80	2.25	215
FS4	160	80	3.0	220
SP1	0	240	2.0	160
SP2	160	80	-0.5	80
SP3	160	80	-0.67	0

1 in = 0.0254 m

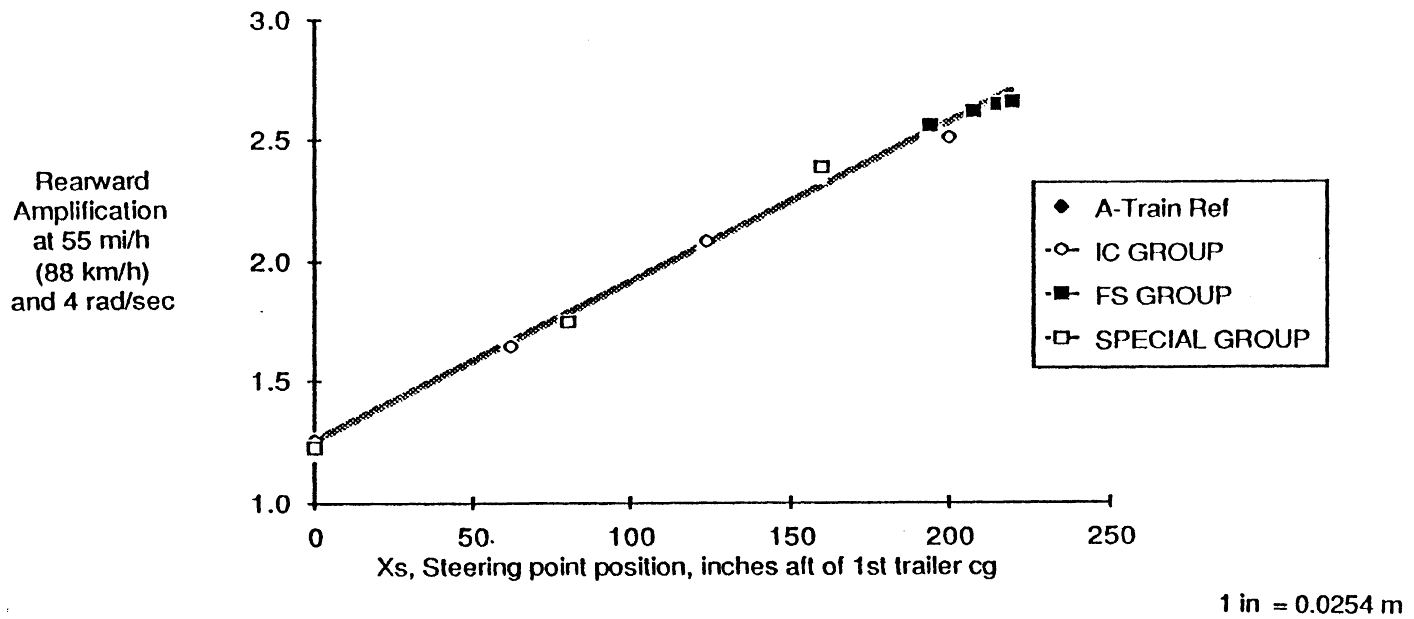


Figure 33. Rearward amplification as a function of steer-point position.

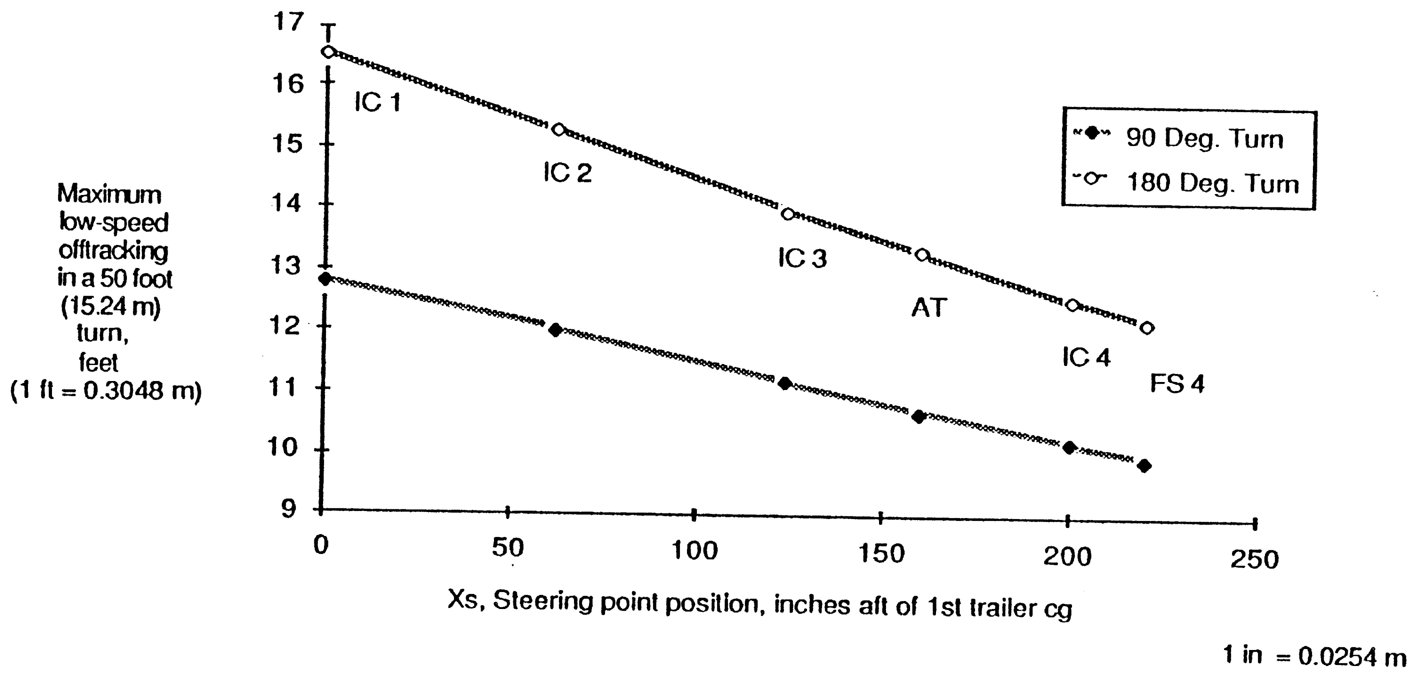


Figure 34. Low-speed offtracking in a 50-foot turn as a function of steer-point position.

equation (9f) points out, the limit position of the steer point, as steering gain increases positively, is the position of the dolly axle centerline.

Given the findings of this analysis, and the recognition that the study of the IC dollies and the FS dollies was a redundant exercise, three of the FS dollies were discarded. Only the FS4 dolly was retained (since it provides the most rearward steer point of the original set) for study along with the IC dollies.

Dynamic Roll Stability Limit. The second portion of the screening study examined the dynamic roll stability limit of the study vehicles in the emergency lane-change maneuver. The study plan called for an exercise in which each of the study vehicles would be subject to an iterative series of severe lane-change maneuvers, until the rollover limit of the vehicle was determined to within 0.015 g of peak tractor lateral acceleration. (Dynamic rollover limit is characterized by the maximum lateral acceleration of the tractor, since this is the significant measure of maneuvering capability of the train in general. Of course, it is generally expected, and was always the case in this study, that the second trailer would be the unit which rolls over, and that it would experience substantially higher lateral accelerations than the tractor.)

The frequency range covered in examining rearward amplification was 1 to 4 rad/sec at 1 rad/sec intervals. The rollover testing maneuvers were to be conducted at only one selected frequency for each vehicle. Since rearward amplification is frequency dependent, and different vehicles showed different frequency sensitivities in this regard, it was decided that each vehicle would be tested for rollover at the frequency at which it had displayed the greatest rearward amplification. For example, from the data of figure 24, it would be determined that vehicle IC4 would be tested for roll stability using 4 rad/sec lane changes, since that is the frequency at which it displayed its greatest rearward amplification. For the same reason, vehicles IC2 and IC3 would be tested at 3 rad/sec and vehicle IC1 would be tested at 2 rad/sec.

In retrospect, this plan was perhaps not the best. Results of the later, in-depth study make it clear that frequency "tuning" properties in the roll response of the vehicle, per se, are important in determinimg roll stability limit. Later it became clear that the roll response of the second trailer is particularly sensitive to excitations in the 2 rad/sec frequency range. Thus, vehicles tested for roll limit at lower frequencies were "penalized" more severely for this roll resonance than were vehicles tested at higher frequencies. (Later work suggests a "penalty" of about 0.05 g's, in dropping from 3 to 2 rad/sec, for dollies without roll coupling and more for dollies with roll coupling.) In interpreting the following results, vehicles tested at 3 and 4 rad/sec should be "derated" relative to those tested at 2 rad/sec.

Further, it should be remembered that the fidelity of the measure for each vehicle is in the vicinity of 0.015 g.

Figure 35 displays the results for the shifted-steer-point dollies. Accounting for the differences in test frequency, the relative roll stability of these vehicles is as would be expected from our understanding of steer point and its influence on rearward amplification. That is, the vehicle with the most forward steer point has the highest dynamic rollover threshold, and rollover threshold declines as the steer point moves aft. An additional point of interest is that, for the IC1 vehicle, rollover of the tractor-semitrailer unit occurred in a lane change only slightly more severe than that required to roll over the second trailer.

Figure 36 clearly shows the advantage of roll coupling between trailers in dynamic maneuvering. Although roll coupling had only a modest influence on rearward amplification, it can significantly raise the rollover limit by directly improving roll stability. The A-train with a roll-rigid hitch was actually the most stable vehicle in roll which was simulated. Unfortunately, an A-dolly hitch/frame which is even as rigid as that of RC1 is probably not practically attainable.

The dynamic rollover thresholds of the linked articulation dollies are shown in figure 37. As was done in discussing the rearward amplification results, the roll-free B-dolly (B2) and the skid-steer dolly (SS) are also shown, since they represent the "limit" cases of linked articulation. Except for the SS dolly, the results again are right in line with expectations resulting from rearward amplification results. The B2 and the LA1 dollies show the highest rollover threshold of all dollies without trailer-to-trailer roll coupling. The skid-steer dolly shows a very poor rollover threshold, however. This dolly suffers somewhat from the 2 rad/sec test frequency, but also from the fact that yaw motions of the second trailer of theis type of vehicle are very lightly damped. This low damping characteristic is well established in the literature.[5,6]

Figure 38 shows the results for the skid-steer group of dollies, including the K-trains. While the K-train with the reference level of steering resistance (K1) appears to be virtually the same as the skid-steer dolly, the K-train with no steering resistance (K2) has a surprisingly high rollover threshold.

Figure 39 shows the high level of rollover threshold attained with non-steering B-dollies of varying levels of roll coupling. The B2 vehicle, with no roll coupling, does well due only to the good yaw plane performance of the B-train configuration. With 30,000 in-lb/deg (3,390 N-m/deg) of roll coupling, the B vehicle does better. The B-train, which is

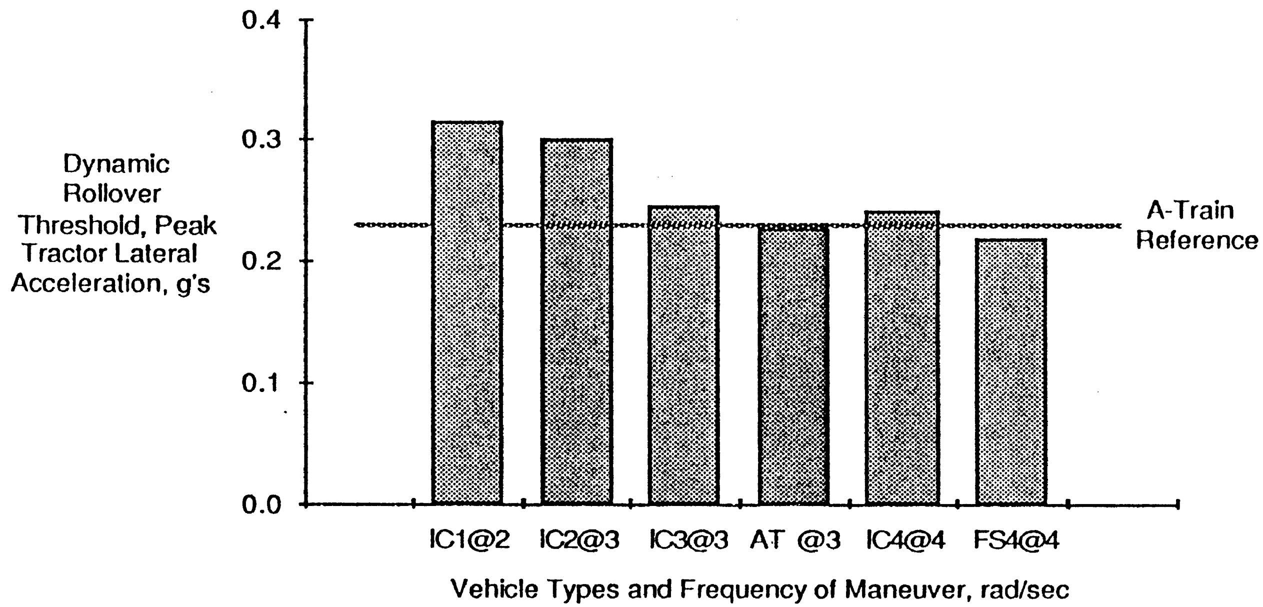


Figure 35. Dynamic rollover threshold in an emergency lane change: the shifted-steer-point dollies.

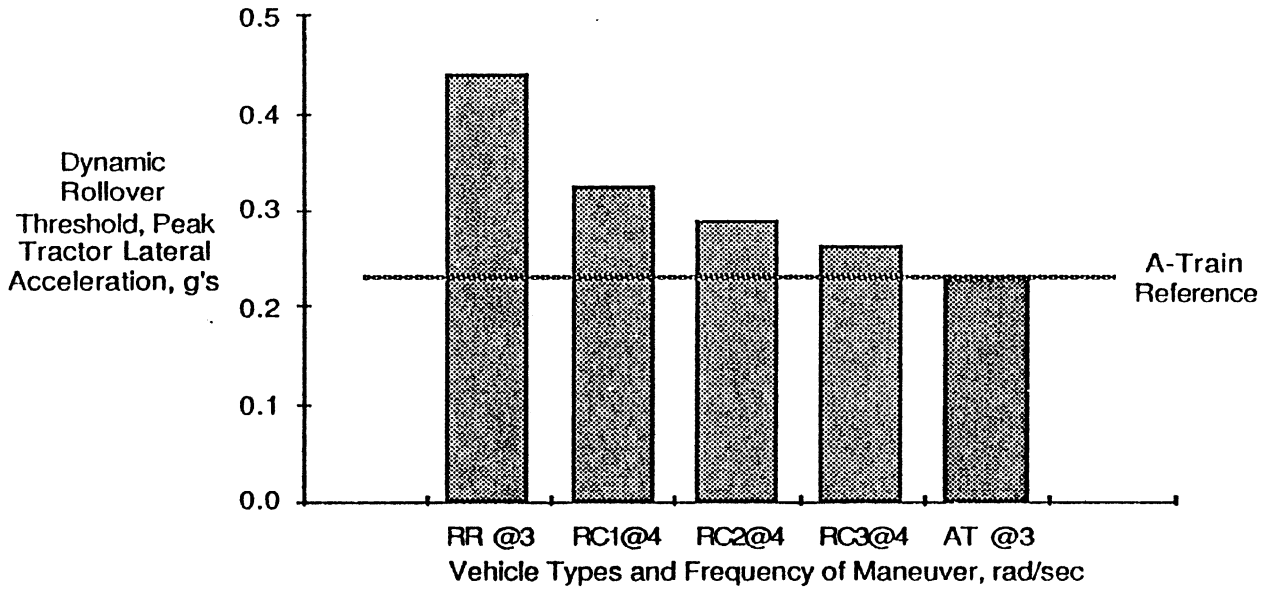


Figure 36. Dynamic rollover threshold in an emergency lane change: the roll-stiffened pintle hitches.

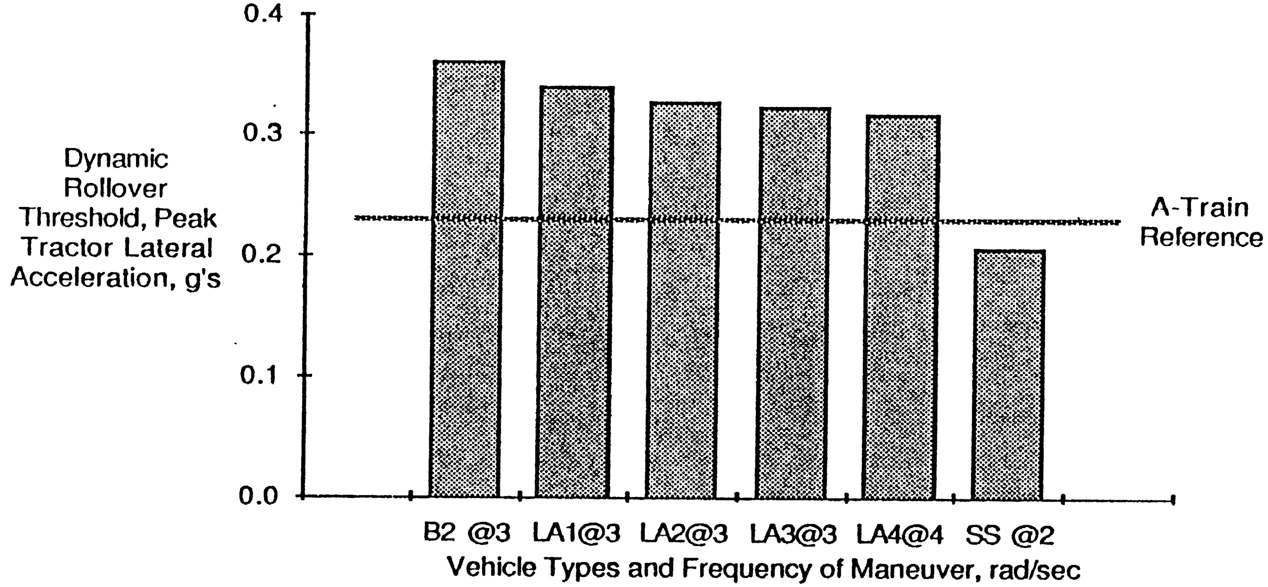


Figure 37. Dynamic rollover threshold in an emergency lane change: the linked-articulation dollies.

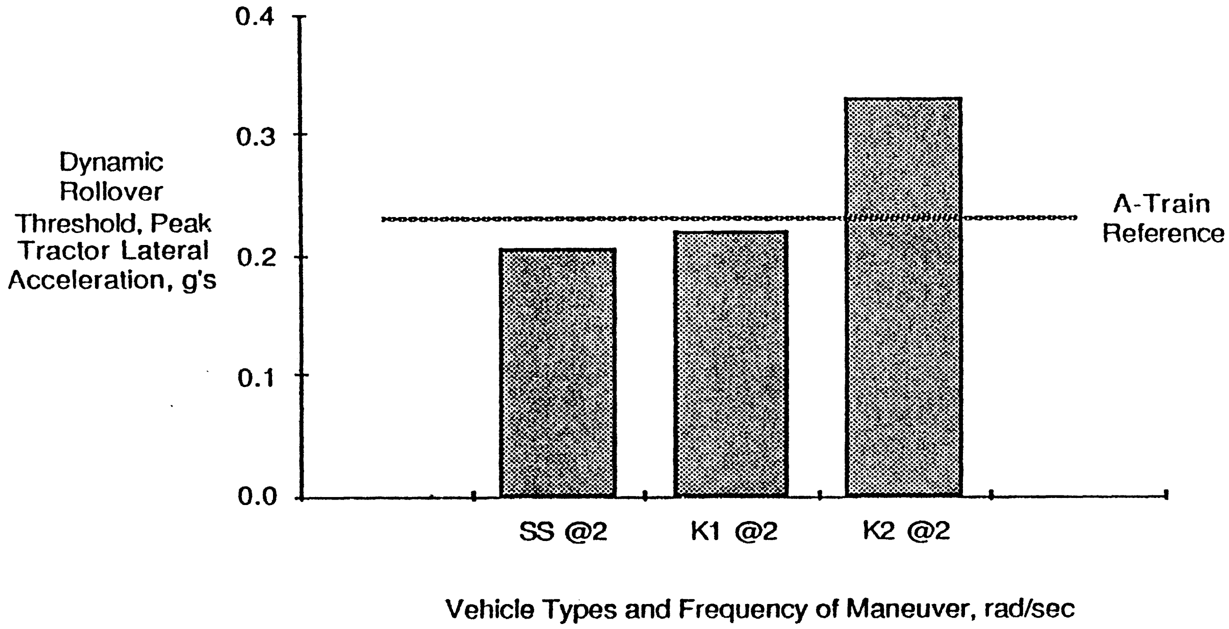


Figure 38. Dynamic rollover threshold in an emergency lane change: the skid-steer dollies.

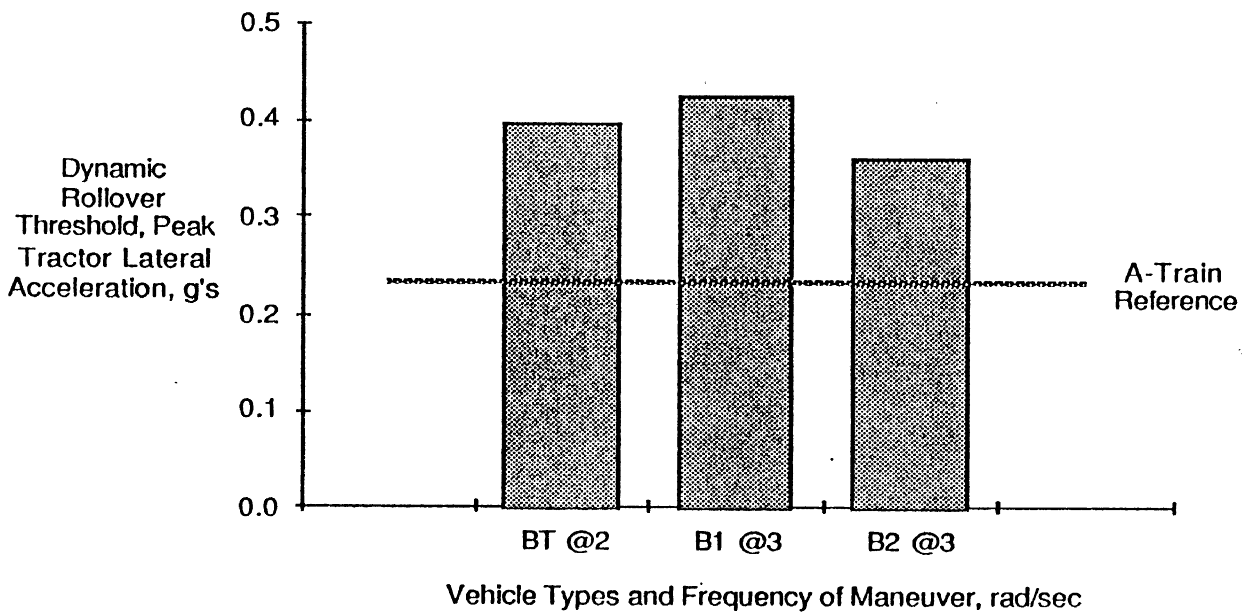


Figure 39. Dynamic rollover threshold in an emergency lane change: the roll-compliant B-dollies.

rigid in roll, would be expected to perform even better, except that it is penalized by being tested at 2 rad/sec.

All the B-dollies of figure 40 are essentially roll rigid, but vary in the steering-resistance properties of their axles. For the SA dollies, rollover threshold improves as steering resistance increases. The slightly poorer performance of the B-train (with non-steering axle) is somewhat surprising. (Note that the thresholds of the BT, SA1, and SA2 vehicles are all virtually within the fidelity of the measure.) Seemingly, the only explanation is the slightly different axle layout of the B-train. (See figures 19 and 20.)

The rollover thresholds of all the test vehicles of the screening study are shown in rank order in the bar chart of figure 41. Better-performing vehicles are to the left. The figure clearly shows the advantage of the yaw and roll coupling of the B-dolly configuration. The roll-rigid A-train (RR) performs as well as the B-trains in this measure, but the A-trains with realistic levels of roll couplings show moderate to small levels of improvement over the reference A-train. All versions of the linked-articulation dollies, the dollies with a strong forward shift of the steer point, and the K-train with completely free steering of the dolly axle, fall into the "middle" group of dollies by this measure. Dollies with a rearward steer point (including the reference A-train) and the remaining two dollies of the skid-steer group are the poorer performers.

Low-Speed Offtracking. The screening study also examined the low-speed offtracking performance of the study vehicles. The obvious advantage of the doubles configuration (over a single-trailer vehicle providing equal cargo-carrying capacity) is the improved low-speed offtracking performance which allows such a long vehicle to be practical. Although the main goal of this study is to determine methods of improving dynamic performance of the double, attaining that goal should not be allowed to seriously degrade this performance advantage of the double.

The low-speed offtracking performance of the subject vehicles was examined by subjecting each to maneuvers involving 90- and 180-degree turns of 50-foot 15.2-m) radii. Results describing the path of the angles were scanned, and the maximum offtracking occurring between the first and last axle of the train was taken as the performance measure of interest.

UMTRI's simplified Tractrix offtracking model and the Yaw/Roll model were both used. The Tractrix model is limited to conventional vehicles with no more than one non-steering axle installed on each yaw-independent unit of the vehicle. To be cost effective, this model was used when possible, but the Yaw/Roll model was used whenever the study

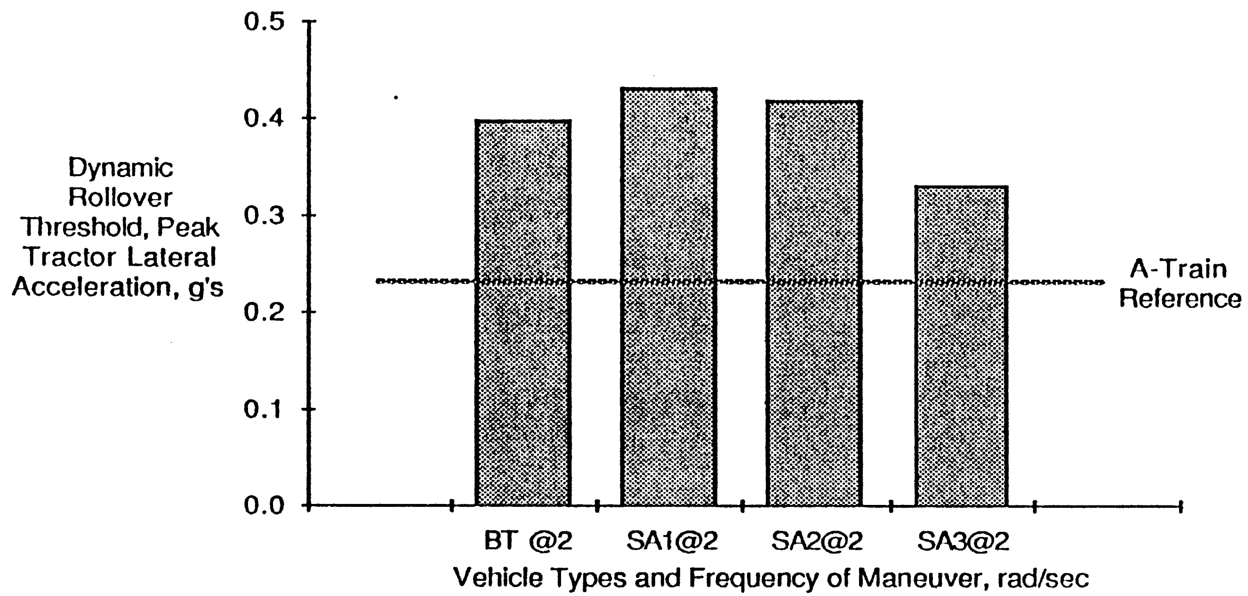


Figure 40. Dynamic rollover threshold in an emergency lane change: the self-steering B-dollies.

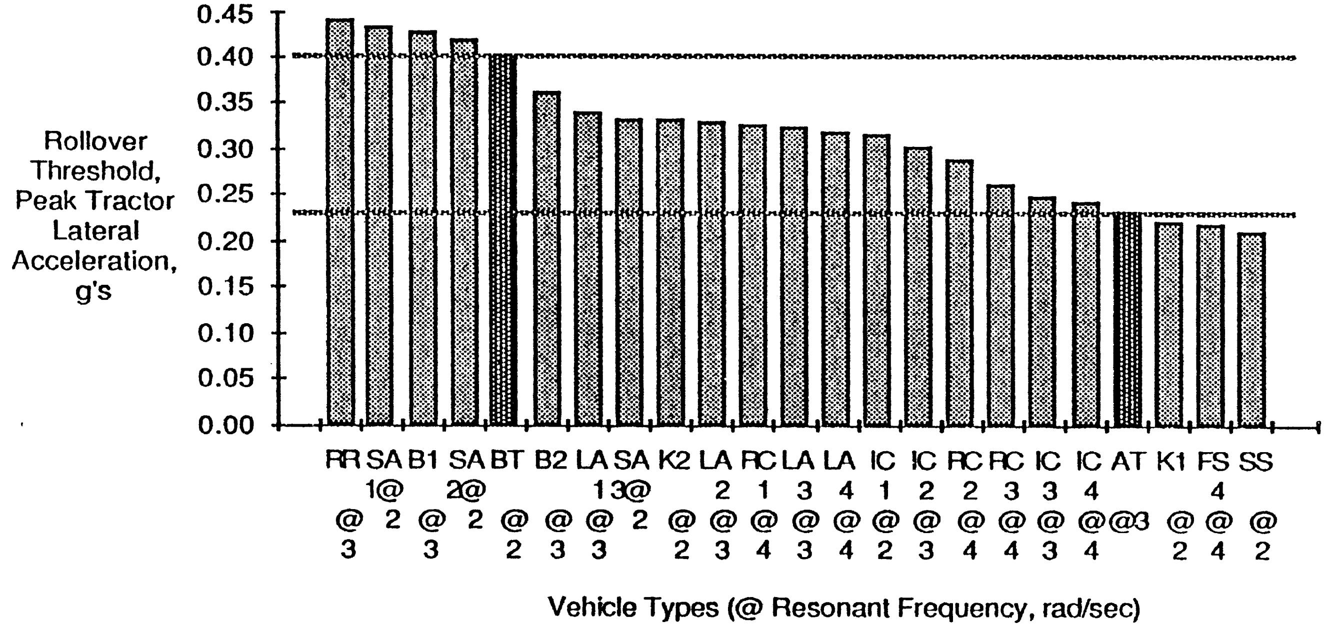


Figure 41. Dynamic rollover threshold in an emergency lane change of all the screening study dollies.

vehicle could not be accurately portrayed within the constraints of the simple model, i.e., whenever the vehicle had a definable unit with one yaw degree of freedom that included more than one axle with fixed or restricted steering properties.

The Tractrix model predicts the offtracking performance that would result at very low speed; i.e., it assumes zero lateral acceleration on all units of the vehicle. The Yaw/Roll model, however, does not make this simplifying assumption, and low-speed offtracking runs must proceed at some real speed that produces some small level of lateral acceleration. At slower speeds, the unwanted lateral acceleration is lower, but the run requires more time and is more expensive. Low-speed offtracking runs using Yaw/Roll were conducted at 10 ft/sec (3 m/sec) forward velocity (0.06 g lateral acceleration in a 50-foot (15.2-m) radius turn). The influence of this speed, and more specifically the resulting lateral acceleration, produces discrepancies between the predictions of the simple model and the Yaw/Roll model. Runs allowing comparison between the results of the two models were made with two study vehicles.

Figure 42 summarizes the results of the offtracking runs. The figure includes results from 180- and 90-degree turns as run on the Tractrix and Yaw/Roll simulation models, respectively. The reference A-train (AT) vehicle and the SA3 vehicle were simulated with both models to provide the needed comparison. The performance measures of these vehicles are shown at the left and right ends of the graph, respectively.

The shifted-steer-point vehicles (IC's and FS4) show, as expected, that offtracking degrades as the steer point moves forward and improves as it moves rearward.

The linked-articulation vehicles show some improvement in offtracking relative to the A-train, except for LA4 in the 180-degree turn. We note here that the 180-degree turn is a "better" measure of steady-state offtracking, while the 90-degree turn is influenced more by the spatial lag characteristic of transient offtracking performance.[13,15,22] The suggestion is that the steady-state performance of LA4 is poor, but that the linked articulation character "stretches out" the transient performance so that the vehicle is not penalized in the shorter turn.

The skid-steer and K-train vehicles show wide-ranging performance. The skid-steer dolly and the K-train equipped with the steerable axle with the reference steer resistance perform nearly the same. (The reference steer resistance is sufficient to minimize self steering.) The simple performance measure of figure 42 indicates very good performance for these dollies. However, examining the full data sets (for example, see the axle paths of the skid-steer vehicle in a 180-degree turn, shown in figure 43) indicates that these vehicles

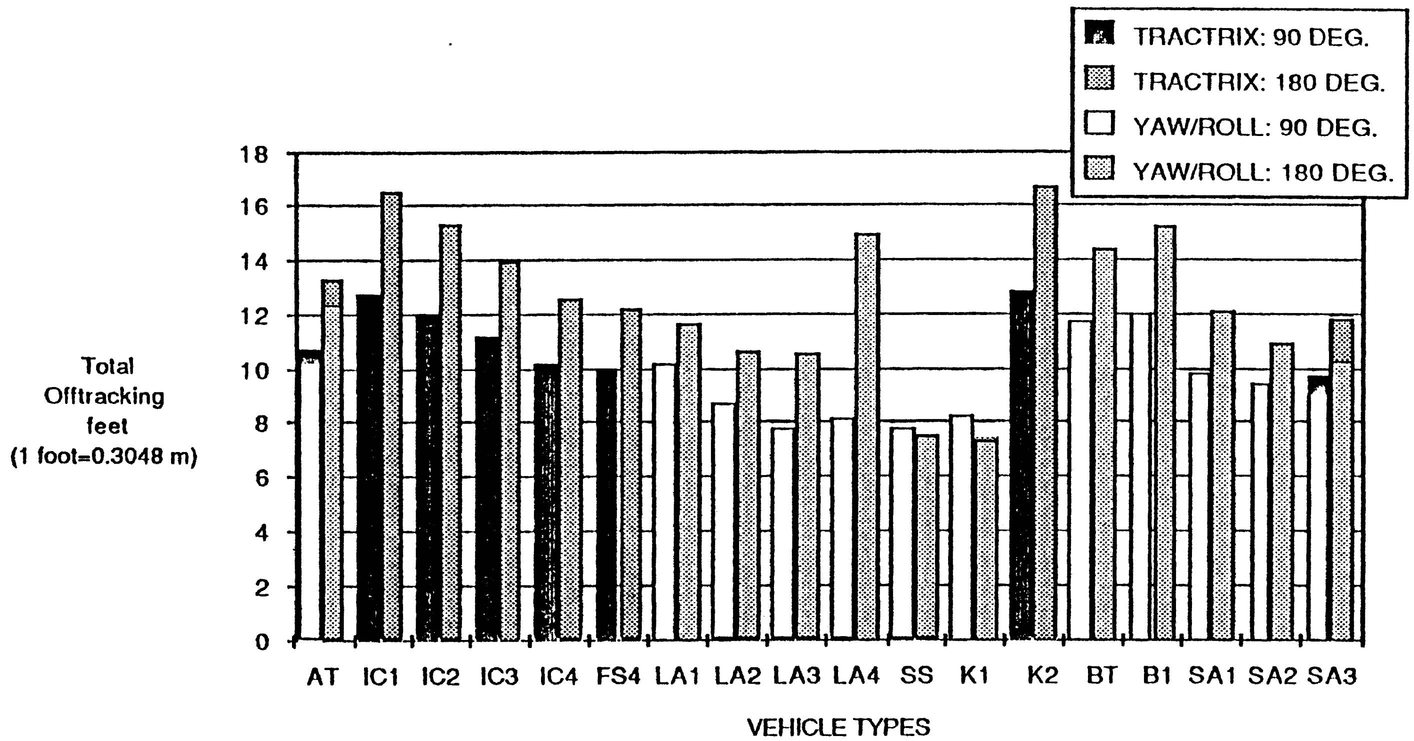


Figure.42 Maximum offtracking performance of the screening study vehicles in a 50-foot radius turn.

1 in = 0.0254 m

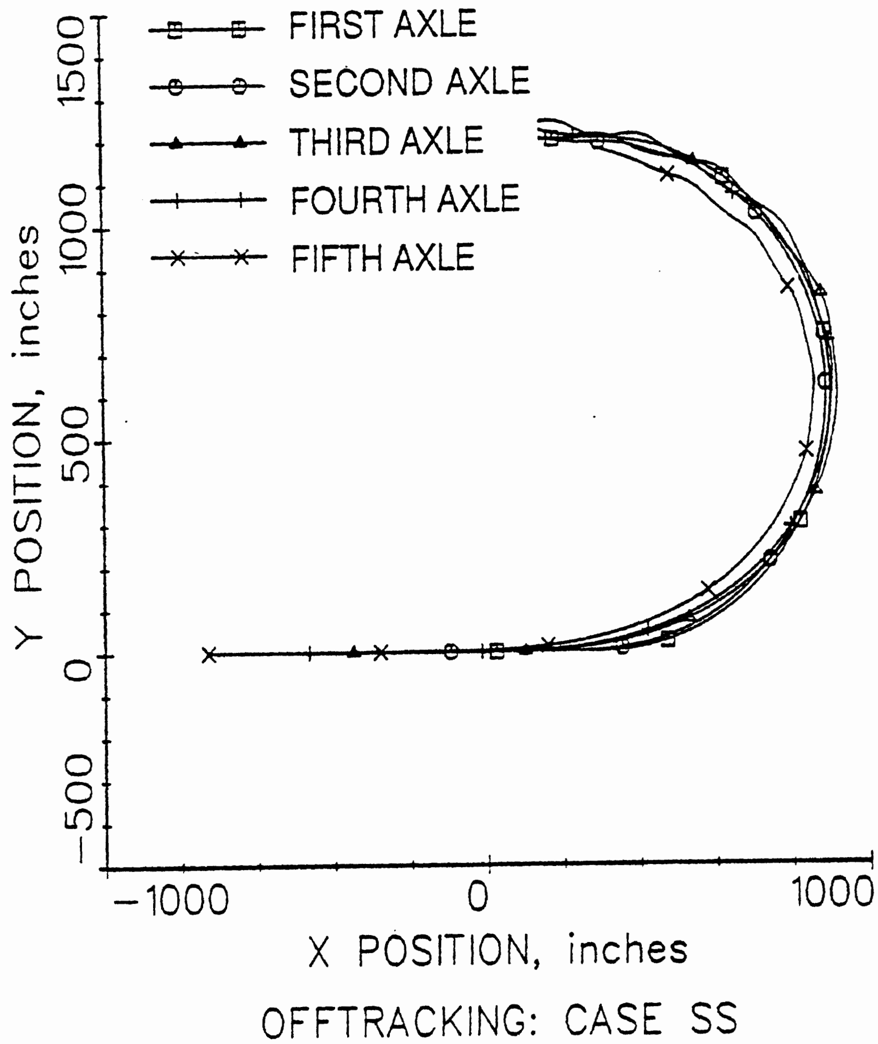


Figure 43. Offtracking behavior with the skid-steer dolly.

track erratically. The skid-steer trailer must, indeed, be steered by skidding the dolly tires sideways as it proceeds through tight turns. The strong tendency of the trailer to proceed straight ahead actually produces outboard offtracking at low speed. (This strongly suggests that high-speed offtracking, which is normally outboard, of these vehicles would be rather poor.[15]) On the other hand, the K-train equipped with a freely steering dolly axle displays very high (inboard) offtracking. In the yaw plane, the second trailer of this vehicle behaves essentially as a long-wheelbase semitrailer with the "fifth wheel" at the pintle. This long unit provides a large contribution to inboard offtracking.[22]

As expected, the B-train (BT) and the vehicle equipped with a B-dolly with a non-steering axle (B1) (the difference being small adjustments in axle positions, see figures 19 and 20) offtrack slightly more than the A-train. The vehicles with the self-steering B-dollies (SA's) show slightly improved offtracking, with that improvement increasing as the steer resistance properties of the self-steering axle decreases.

c. The In-Depth Study.

The objectives of this portion of the simulation study were (1) to provide a more complete performance analysis of the several specific dollies brought forward from the screening study, and (2) to provide an "optimum" parametric description of a dolly which could serve as a guide in the design of the prototype dolly to be constructed in this project. Actual samples of the dollies selected for the in-depth study would later be subjected to performance testing on the test track. These objectives would be addressed by expanding, in both breadth and depth, the study begun in the screening activity, and initiating new investigations into the yaw-damping properties of the vehicles, stability during braking, and hitch-loading issues.

In the in-depth study, the investigation of rearward amplification was expanded to include the examination of influences of various vehicle loading conditions and changes in velocity. For each of the dolly types, rearward amplification was examined under four loading conditions, viz., (1) both trailers fully loaded (F/F), (2) first trailer fully loaded, second trailer empty (F/E), (3) first trailer empty, second trailer fully loaded (E/F), and (4) both trailers empty (E/E). The influence of velocity on the rearward amplification behavior of each vehicle was also examined. Rearward amplification was determined at 55, 40, and in some cases, 25 mi/h (88, 64, 40 km/h). This was done at selected loading conditions and frequencies, depending on the dolly type.

As in the screening study, the dynamic rollover threshold of the test vehicle in the F/F loading condition was determined. This measure was taken for both 2 and 3 rad/sec lane-change maneuvers. Hitch loading levels were also examined during the most severe maneuvers.

The yaw damping performance of the vehicles equipped with the selected dollies was also examined. It is revealed in the literature that some configurations of multitrailer vehicles display very lightly, or even negatively, damped (unstable) dynamic modes of motion.[5,6] The eigenvector of these modes is usually dominated by motions of the last trailer so that, in practice, these modes are revealed as seemingly unprovoked "wagging" of the last trailer during normal running. One fatal accident of a doubles using a steerable-axle B-dolly, which occurred in Saskatchewan, is strongly suspected of being caused by a similar phenomenon.

To investigate the yaw damping quality of the subject dollies, they were each subjected to simulated "pulse steer maneuvers." At the outset of the maneuver, a sharp steering pulse is introduced and, thereafter, the steering is held fixed at zero. Damping of the vehicle is judged by observing the manner in which the oscillating lateral accelerations of the second trailer die out. Loading is known to be influential to multitrailer vehicle damping, so this investigation included the four loading conditions.[5,6] Further, a general understanding of vehicle dynamics suggests that, for self-steering B-dollies, the level of steering resistance and drawbar length of the dolly (and more precisely, longitudinal spacing of the first trailer and dolly axles) should also be very influential. The influence of drawbar length was briefly examined for the B-dolly and prototype dolly.

Finally, the emergency-level braking performance of vehicles using the subject dollies was examined. The "self-steering" behavior of B-dolly axles is intended to result from castering action, that is, as a result of the development of tire side forces. However, along with caster, the steering geometry of these axles also includes lateral kingpin offset, so that longitudinal tire forces (braking forces) also produce steering moments. Nominally, these moments are balanced when braking forces develop evenly, side-to-side. However, commercial vehicle brakes are highly variable, so that large, side-to-side brake force differences are common. Further, if the tires of the axle are operating on surfaces of different friction (one side on ice, for example) during braking, or if one tire drops off a shoulder into soft soils, very large longitudinal force imbalance can result. These effects are very powerful for the "turntable steer" axles, since the effective kingpin offset is large, and are one reason why auto-steer-style axles are favored. Simulated braking performance runs were conducted on the test vehicles. These included straight-line braking on a split-m surface and braking in a turn on a low-friction surface.

All of these simulation activities were conducted with three dolly types brought forward from the screening study, plus a "prototype" concept dolly developed within the study.

d. The In-Depth Study: Commercial Dollies Selected from the Screening Study.

Three "commercial" dollies were selected from the screening study vehicles for further study. Selections were made on the basis of (1) predicted performance quality as indicated by the screening study, (2) a reasonable expectation for obtaining or fabricating a working example, and (3) background theoretical and practical knowledge of the field.

The dollies selected were (1) the (ASTL) auto-steer-style, self-steering B-dolly, (2) the (Truck Safety Systems) linked-articulation dolly, and (3) the (Trapezoid Corp.) asymmetric trapezoidal-drawbar dolly.

The results of the screening study clearly indicated that the selection of a B-dolly for further study was in order. Existing understanding of the braking performance issues discussed above led to selection of the auto-steering type over the turntable steering styles. Again, since only one set of steering resistance data was available, it was used as the reference. (This dolly continued to be designated as SA1.) Some runs were also conducted with low steering resistance (SA3). In the screening study, the steerable-axle B-dollies had been simulated with infinite roll coupling stiffness. In the in-depth study, they were simulated with a more realistic value of 30,000 in-lb/deg (3,390 N-m/deg).

Of all of the modified A-dollies in the screening study, the linked-articulation style appeared to be among the more promising. The rearward amplification performance of this modified A-dolly approached that of the B-dollies. In contrast to the shifted-steer-point dollies, the low-speed offtracking performance of the linked-articulation dolly is not degraded as rearward amplification improves. Though hitch loads may be high in severe maneuvers, it should be possible to provide a design wherein the repetitive loadings of regular use could be minimized. There was the further attraction that no known analyses of this vehicle had previously been conducted. The apparent drawbacks of this dolly were the lack of trailer-to-trailer roll coupling, and the practical problem of the "extra hardware" which could make coupling and uncoupling difficult, and restrict rear access to the first trailer cargo area.

The only examples of linked-articulation dollies known to exist in practice are in use in the Michigan petroleum tanker fleet, and it was determined that the linked-articulation hardware was no longer in production. It was felt that for the testing program to come later, hardware could be fabricated and adapted to the "Western doubles" test vehicle. The articulation angle gain to be used in the in-depth study could be established by choice.

The articulation gain chosen was that which establishes, within the small angle approximation, "Ackerman" geometry between the axles of the dolly and the two trailers. The so-called Ackerman steering relationship is established when the projection, in the plan view, of all of the wheels in question intersect at a common point, which is the turn center. Ackerman steering assures that, during low-speed turning, all tires track with no slip and no resulting side force. Tire scuffing and wear are minimized, as are structural loading on the steering system and frame. It is worth emphasizing that at low speed, the A-dolly maintains Ackerman geometry through its natural tracking behavior.

Ackerman geometry of the dolly and two trailers is illustrated in figure 44. Using the notation of the figure, and assuming small angles, it can be shown that:

$$\frac{\Gamma_2}{\Gamma_3} = \frac{OH + TL}{WB} \qquad (10)$$

And since

$$G_{\Gamma_2\Gamma_3} \equiv \frac{\Gamma_2}{\Gamma_3} \qquad (11)$$

then

$$G_{\Gamma_2\Gamma_3} = \frac{OH + TL}{WB} \qquad (12)$$

A steering gain value of 0.44 results in Ackerman steering for the "Western double" used in this simulation study. This gain was used in the in-depth study, and the linked articulation vehicle was designated as "LA.44." Given the results of the screening study, good dynamic performance could be expected from this vehicle.

The asymmetric trapezoidal dolly was selected as the third commercial dolly. This design was seen as one which could take advantage of the good dynamic performance which results from dollies whose steer point is forward, and the good offtracking performance resulting from a rearward location of the steer point. The major shortcoming of this design is that it lacks trailer-to-trailer roll coupling. It has the advantage of simplicity, and it is probably inherently the lightest of all the dollies included in the in-depth study (a property whose importance will be seen in a later section.) The Trapezoid Corporation design allows for adjusting the geometry of the secondary towbar (see the previous section) so that the IC may be forward for travel at highway speeds and rearward

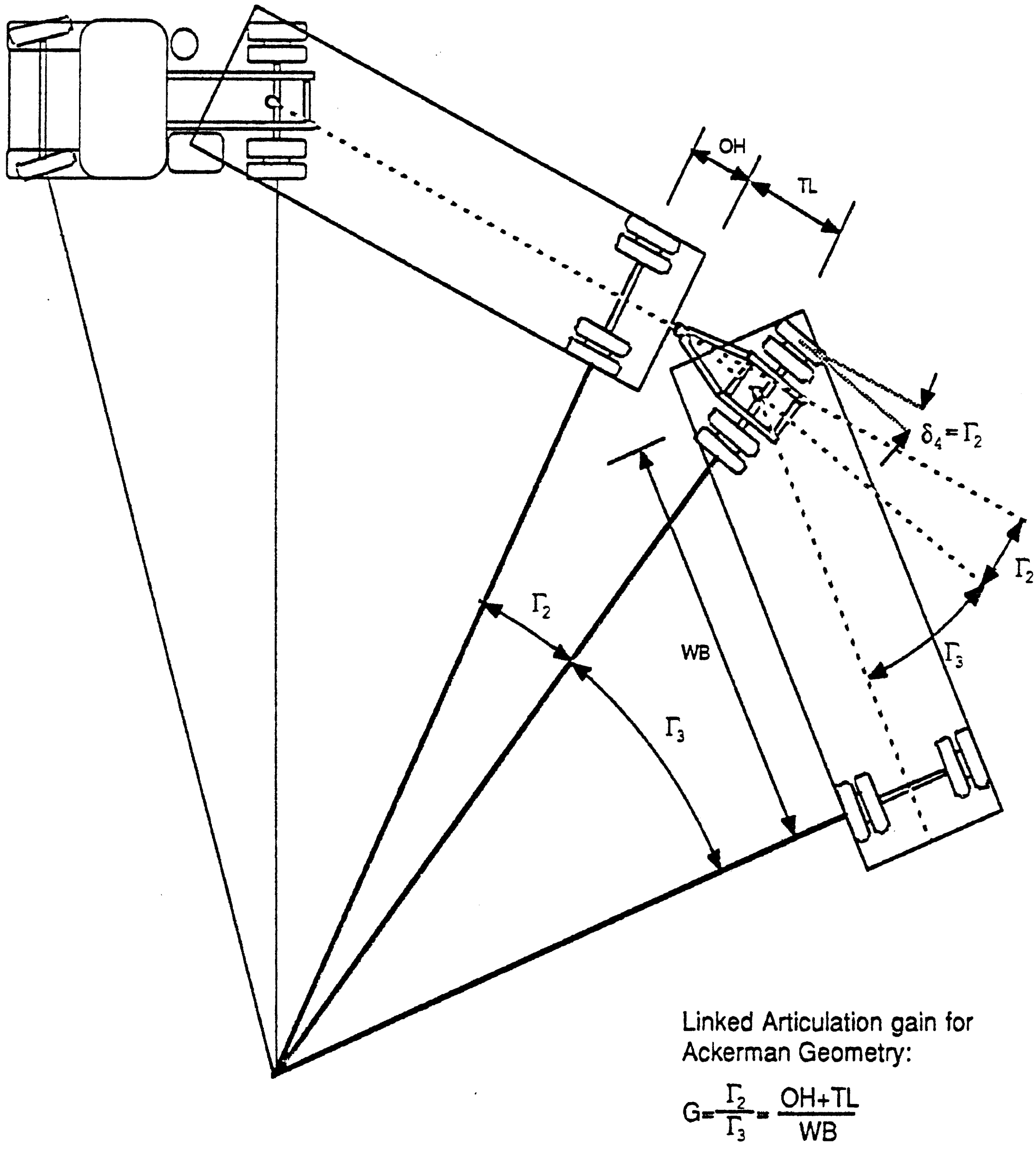


Figure 44. Ackerman steer geometry applied to the linked-articulation dolly.

for low-speed maneuvering. Using the design parameters provided by Trapezoid, these conditions corresponded to x_H (= x_S) values of 41 in (1.04 m) and 168 in (4.27 m), respectively (reference figure 32).

e. The Prototype Dolly, a Controlled-Steering B-Dolly.

The concept of the prototype dolly actually evolved during the progress of the in-depth study, but this "controlled-steering B-dolly" (CSB) was subjected to the same series of simulation runs as the other subject dollies.

Initially, the "protoype development" task was envisioned as an activity in which one of the commercial dolly configurations would be identified as "the best" and the parameters of that configuration would then be "tuned" for optimum performance. In fact, a different approach was taken in which attractive performance concepts from more than one dolly type were merged in one dolly.

The rigid double-drawbar concept of the B-dolly is seen as extremely attractive, in that it (1) eliminates the yaw degree of freedom at the drawbar hitch point, and (2) provides strong trailer-to-trailer roll coupling. The first is the best-known method for improving rearward amplification, and the second is known to be very powerful in providing dynamic roll stability. The biggest drawback of the double drawbar is the introduction of new, potentially large, hitch loads which result from the new yaw and roll constraints, and the related tire scuffing and wear problems that result from the yaw constraint.

To relieve the yaw-related problems somewhat, the self-steering axle has been applied to the B-dolly. In general terms, the theory of operation of the steerable-axle B-dolly is that resistance to steering should be sufficiently high that, at highway speeds, little or no steering takes place, thus assuring good dynamic performance (preventing unacceptably low yaw damping), but, at the same time, steering is also sufficiently free as to significantly mitigate tire scuffing and frame stress problems that would otherwise occur, particularly in low-speed, tight-turning maneuvers. This compromise is fundamentally difficult, since the levels of tire forces desirable for good emergency-level (i.e., unusual) dynamic performance are, indeed, large if envisioned as frequently experienced at low speed. This is so even without considering the high levels of steering resistance required to resist side-to-side brake force imbalance. Although, conceptually, steering resistance could be altered as a function of speed, there is major resistance in the trucking industry to the use of the type of "electronic gadgets" which would likely accompany such a system. Further, "proper" level of steering resistance will always be a function of axle load, requiring an assumption of load at the design stage, which may always be violated in use, or adjustment of steering resistance in the field, which might be automatic (probably electronic) or manual (subject to maladjustment). In short, "tuning" of the self-steering properties of the B-dolly did not appear to be a promising route.

On the other hand, looking at fundamental vehicle properties, it was observed that the linked-articulation dolly concept (1) eliminated a yaw degree of freedom at the dolly which generally resulted in improved rearward amplification, but (2) retained positive control of the yaw orientation ("steer") of the dolly tires. As explained above (and to be substantiated by the presentation of results to follow), the establishment of Ackerman "steer" geometry of the linked-articulation dolly (axle) results in dynamic performance in the yaw plane which is comparable with that of fixed-axle (or high-steering-resistance) B-dollies, while minimizing hitch loads, frame stressing, and tire scuffing during low-speed maneuvering.

The goal in developing the prototype was to combine the attractive elements of these two approaches into one concept. The result is the controlled-steering B-dolly (CSB-dolly). In this concept, the dolly is a double, rigid drawbar style which eliminates the yaw degree of freedom at the drawbar hitch and provides trailer-to-trailer coupling in roll. The tires of the dolly steer relative to the dolly frame in a controlled manner as a function of the yaw articulation angle between the dolly and the following trailer, i.e., the dolly fifth wheel articulation angle. As defined in figure 45, the characteristic parameter of this dolly would be the steering system gain ($G_{\delta 4 \Gamma 3}$).

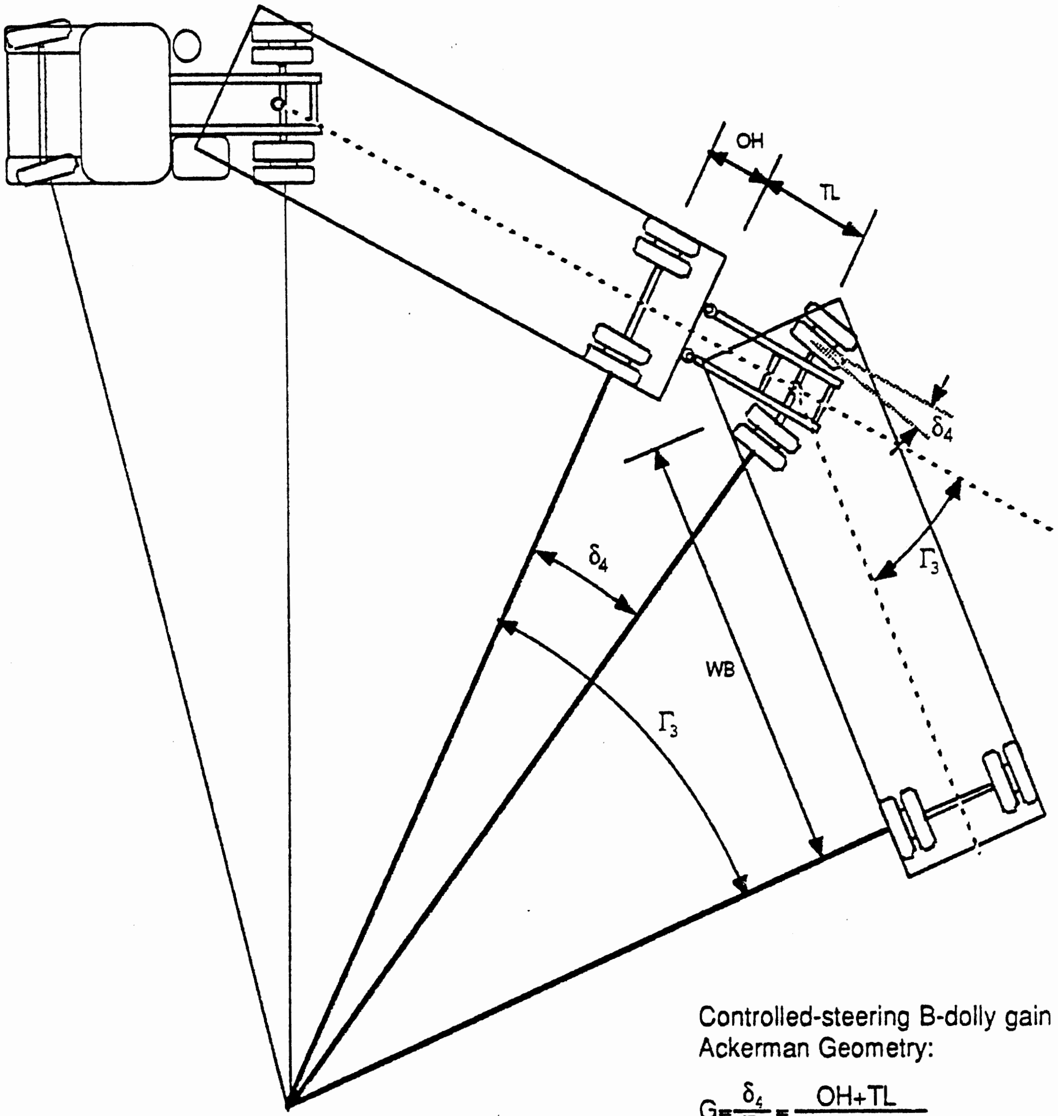
Ackerman steering geometry can also be applied to the CSB-dolly concept. Assuming small angles and using the notation of figure 45:

For Ackerman steering
$$\frac{\delta_4}{\Gamma_3} = \frac{OH + TL}{WB + OH + LT} \quad (13)$$

And since
$$G_{\delta 4 \Gamma 3} \equiv \frac{\delta_4}{\Gamma_3} \quad (14)$$

then
$$G_{\delta 4 \Gamma 3} = \frac{OH + TL}{WB + OH + LT} \quad (15)$$

For the geometry of the Western doubles simulation test vehicle, $G_{\delta 4 \Gamma 3} = 0.3$ for Ackerman steering. This is the steering gain used for the CSB-dolly in the in-depth study. This dolly is designated as CSB.30.



Controlled-steering B-dolly gain for Ackerman Geometry:

$$G = \frac{\delta_4}{\Gamma_3} = \frac{OH+TL}{WB+OH+TL}$$

Figure 45. Ackerman steer geometry applied to the controlled-steering dolly.

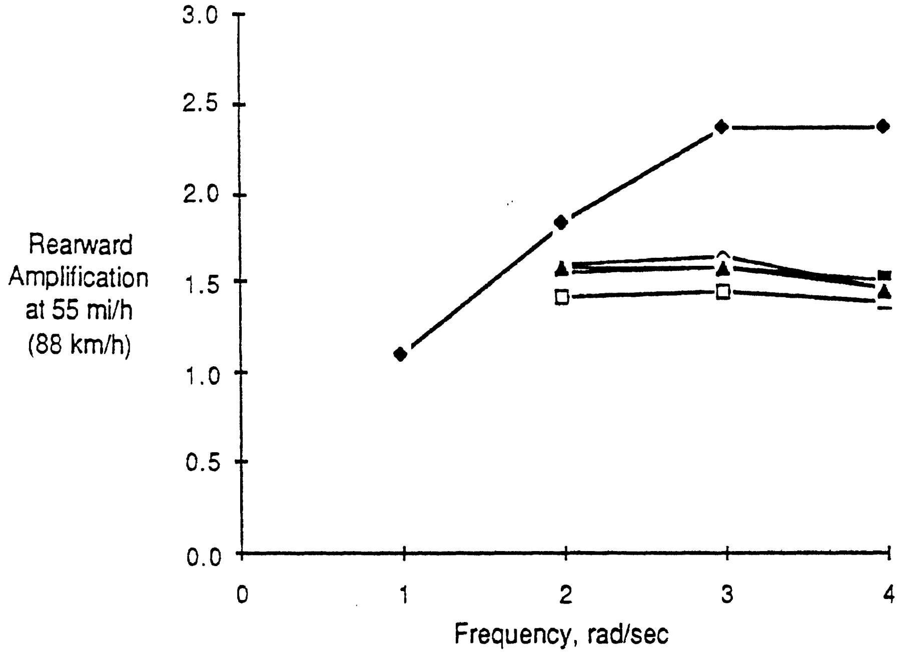
f. The In-Depth Study: Results.

Rearward Amplification. Figures 46 and 47 display the rearward amplification performance of the in-depth study vehicles in four loading conditions. Figure 46 presents the data in a manner which allows comparison of the various dollies in the several loading conditions. Figure 47 shows the influence of loading on performance with each of the dolly types, respectively. Figure 46 shows that the four "improved" dolly types have remarkably similar performance, particularly in the critical full/full loading condition. In the fully loaded condition, three of the dollies are virtually indistinguishable, with the full-resistance self-steering B-dolly being slightly better in this simulated condition. The similarity between the linked-articulation dolly and CSB-dolly is expected because of their similar "steering" concepts. The screening study findings indicated that the self-steering B-dolly (with sufficient steering resistance to prevent any steering in these simulation runs) should be expected to show slightly less rearward amplification. The fact that the performance of the trapezoidal dolly is so similar may be "coincidence" or may be the result of some still-unknown, fundamental similarity of the mechanisms.

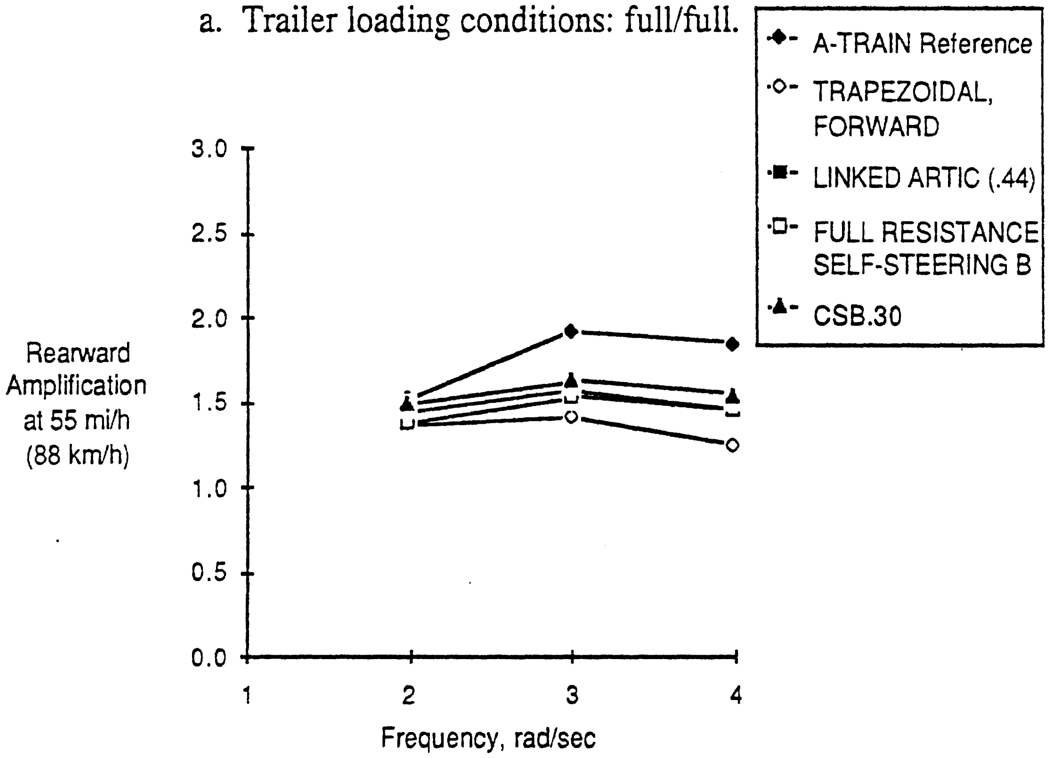
Figure 47 shows that the full/full loading condition produces the highest, or nearly highest, rearward amplification for each dolly. With the A-dolly and trapezoidal dolly, the loading condition of the two trailers appears to have nearly equal influence. For the other three dollies (with the common property of just one yaw articulation degree of freedom at the dolly), the loading condition of the first trailer appears more influential.

Figure 48 shows the rearward amplification performance of the trapezoidal dolly and the self-steering B-dolly in their less favorable (for dynamic performance) states. The trapezoidal dolly shows the expected high levels of rearward amplification when the hitching linkage is arranged for the rearward IC position. With very low steering resistance, the B-dolly shows a wide range of response, depending on loading. In the empty/full condition, rearward amplification is very low. These levels of rearward amplification of less than unity indicate that the second trailer is "under-responding" and not following the path of the tractor. Without the cornering power of the dolly tires, the lightly loaded tires of the first trailer are insufficient to guide both the rear of the first trailer and the front of the second trailer.

Figure 49 shows the influence of forward velocity on the rearward amplification of the test vehicle equipped with the several dollies. The data presented cover many combinations of loading condition and maneuvering frequency. The data indicate what is well established in the literature, viz., that rearward amplification is a strong function of speed and that it increases as speed increases. None of the "improved" dolly types violate this

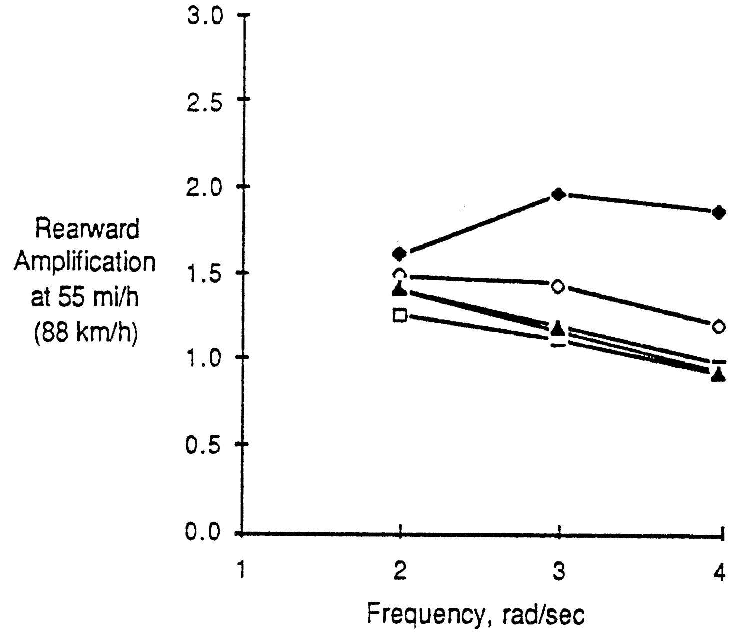


a. Trailer loading conditions: full/full.

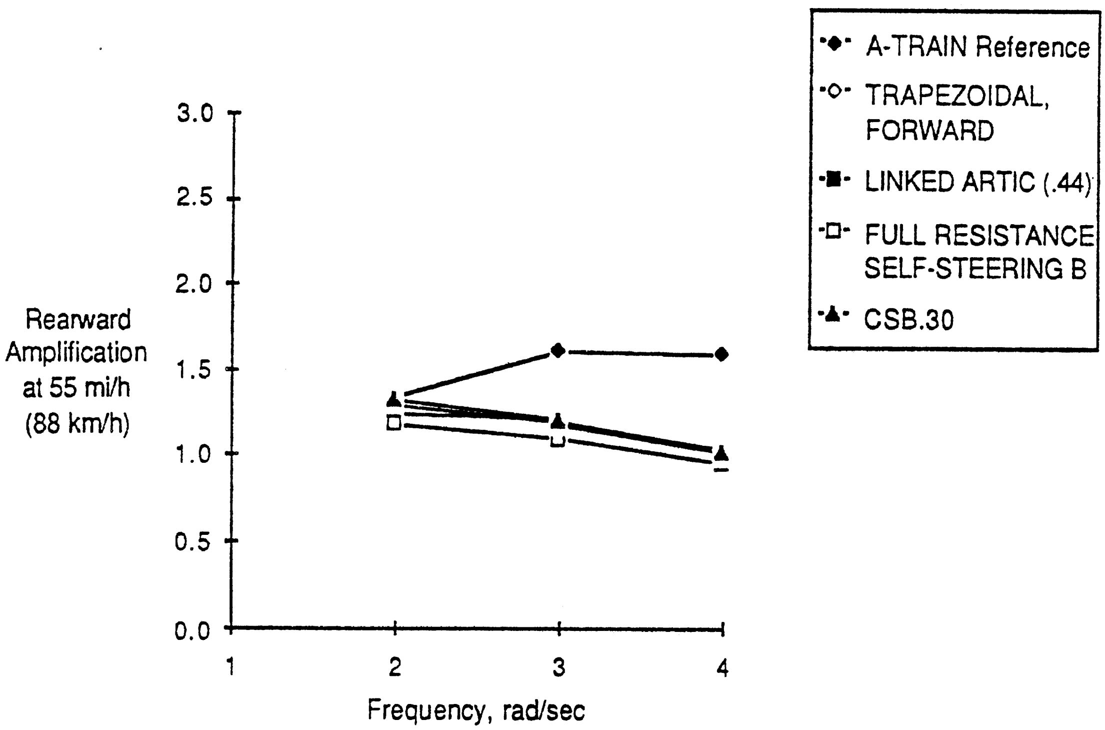


b. Trailer loading conditions: full/empty.

Figure 46. Comparison of the rearward amplification of the improved dollies under four loading conditions.

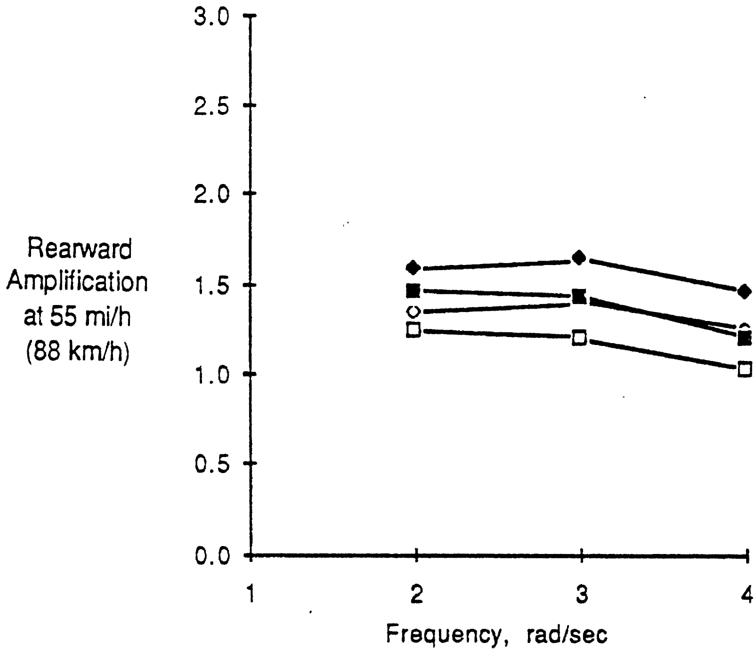


c. Trailer loading conditions: empty/full.

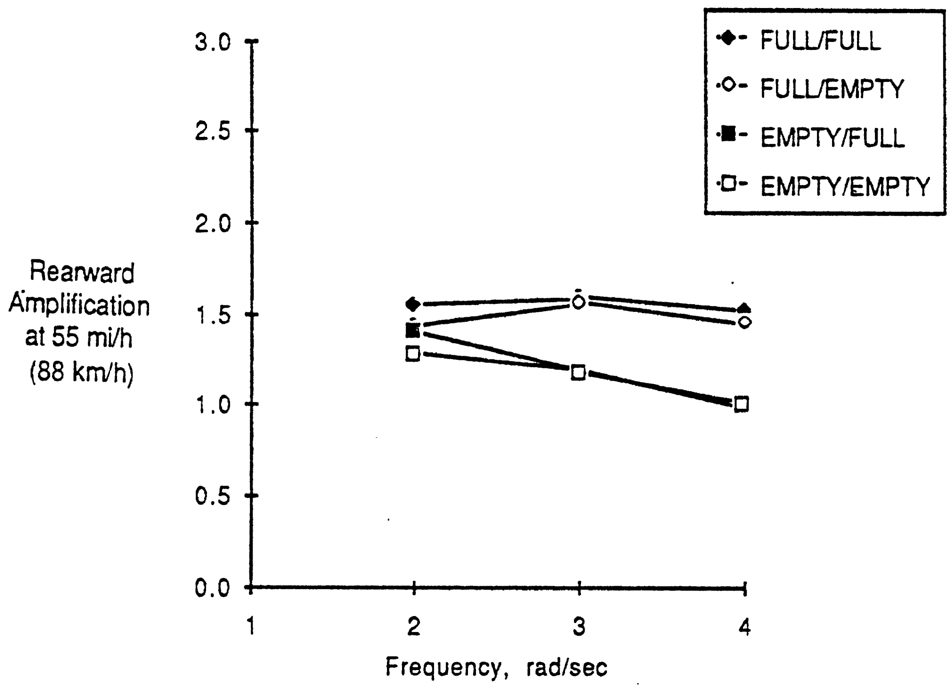


d. Trailer loading conditions: empty/empty.

Figure 46. Comparison of the rearward amplification of the improved dollies under four loading conditions.

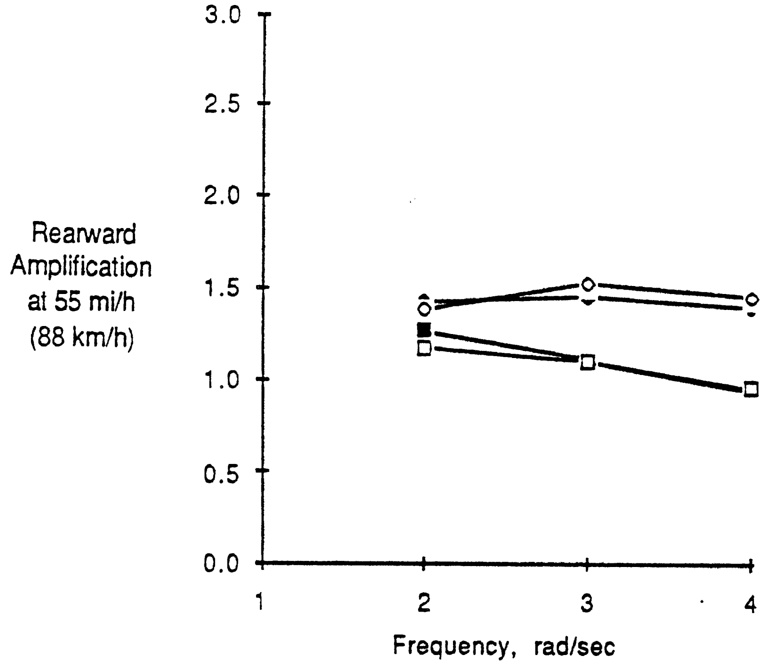


a. The trapezoidal dolly, forward IC position.

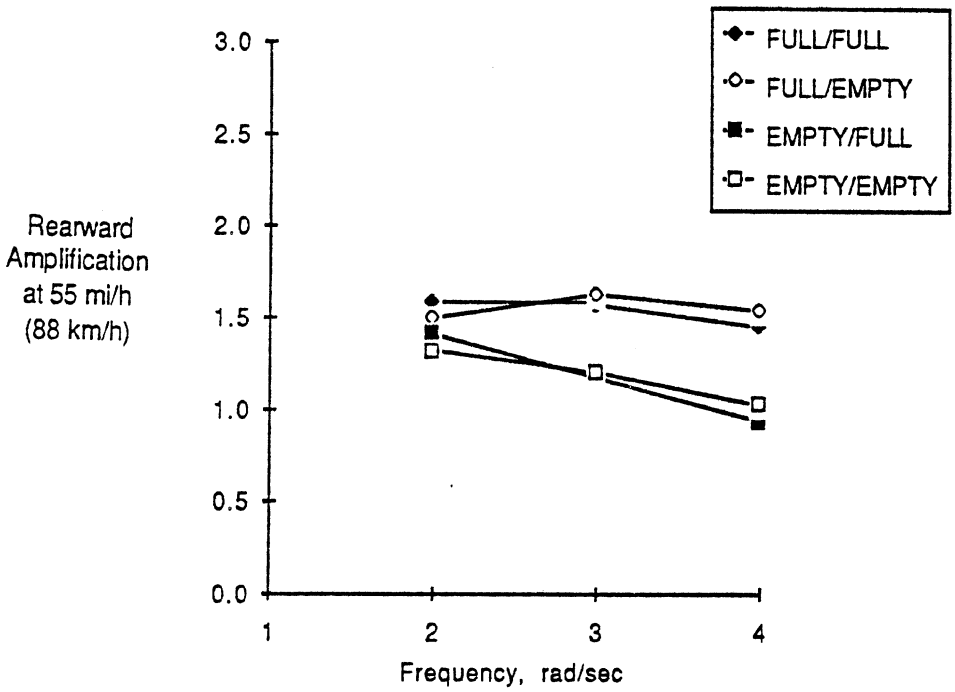


b. The linked-articulation dolly, 0.44 system gain.

Figure 47. The influence of loading condition on the rearward amplification of the improved dollies.

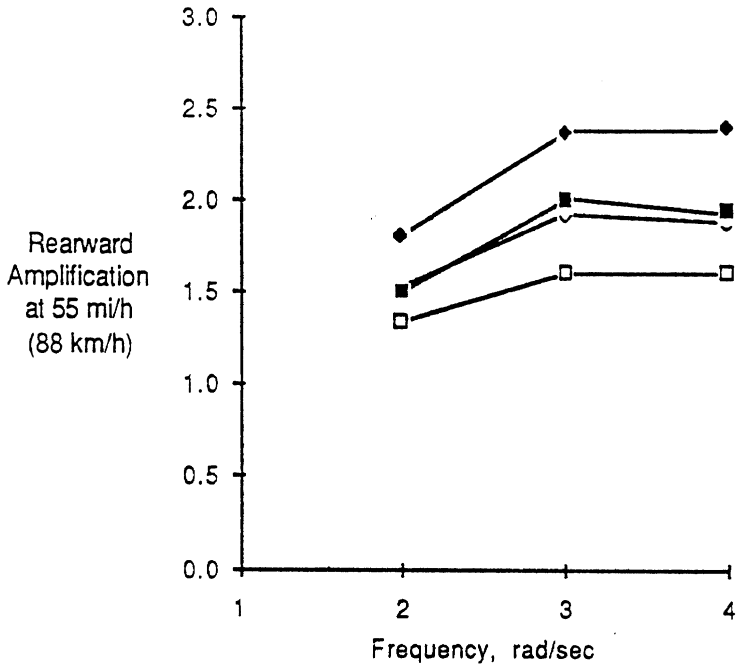


c. The self-steering B-dolly, full steering resistance.

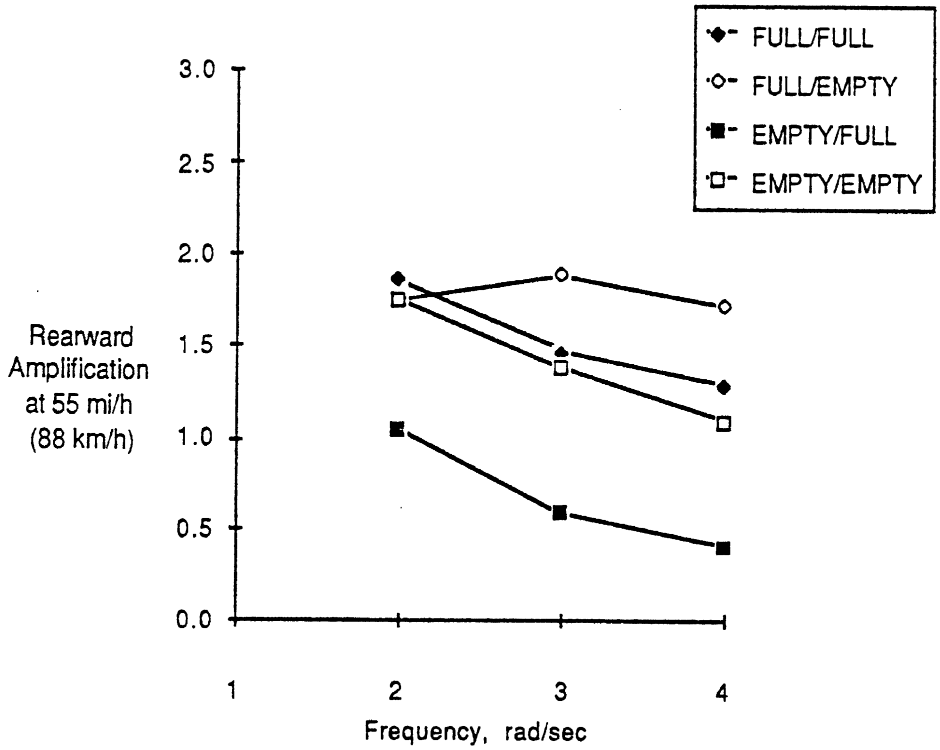


d. The CSB-Dolly, 0.30 steering gain.

Figure 47. The influence of loading condition on the rearward amplification of the improved dollies.

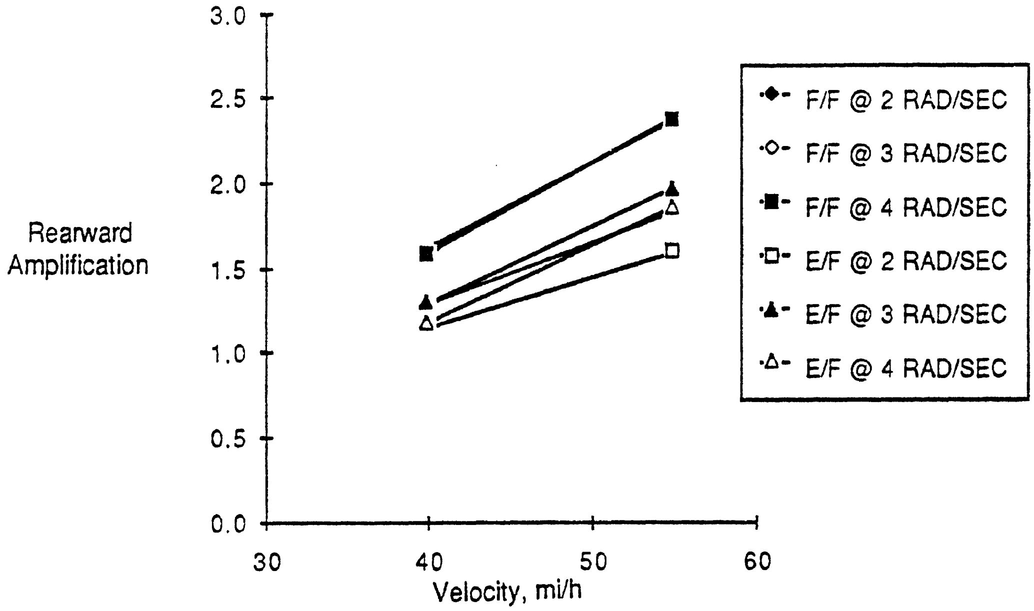


a. The trapezoidal dolly, rearward IC position.



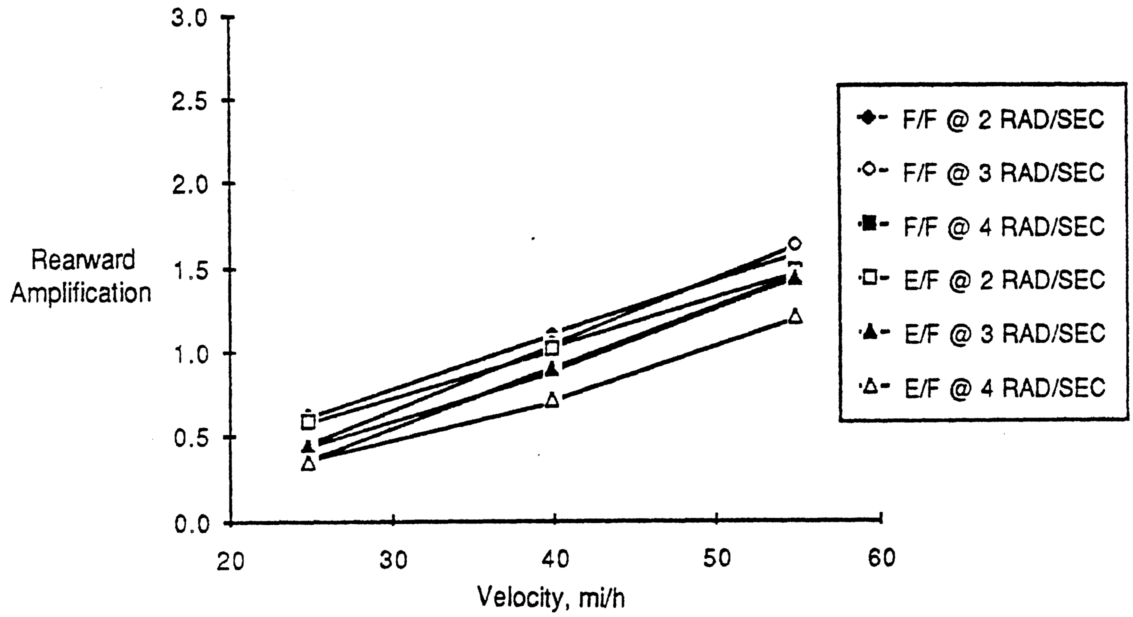
b. The self-steering B-dolly, low steering resistance.

Figure 48. Less favorable rearward amplification performance.



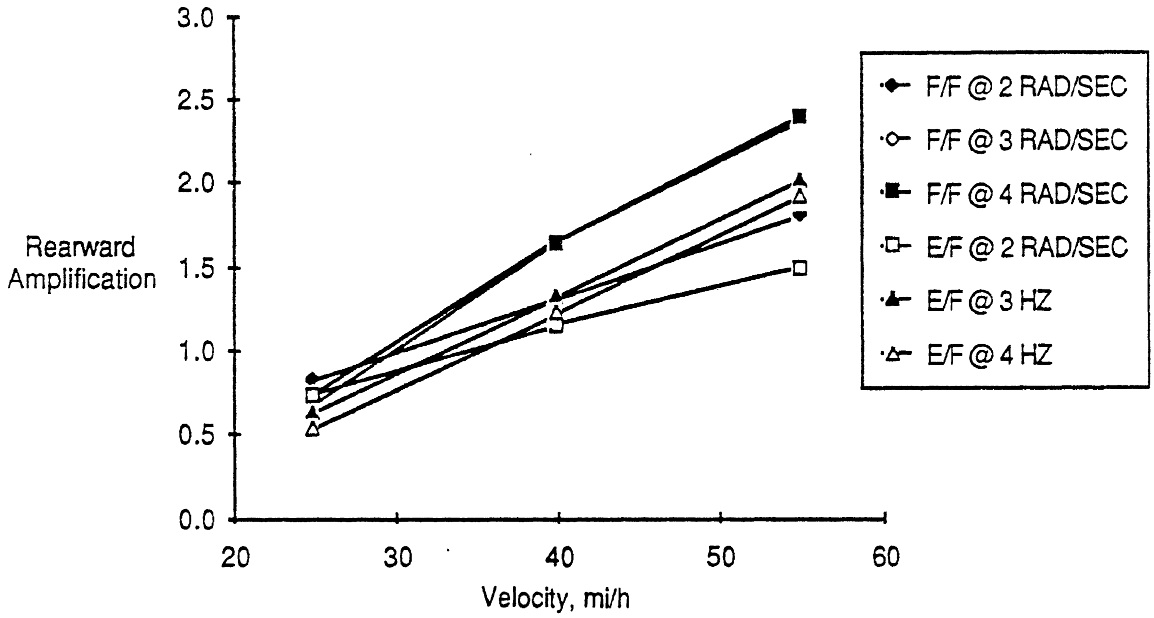
a. The A-train.

1 mi/h = 1.609 km/h

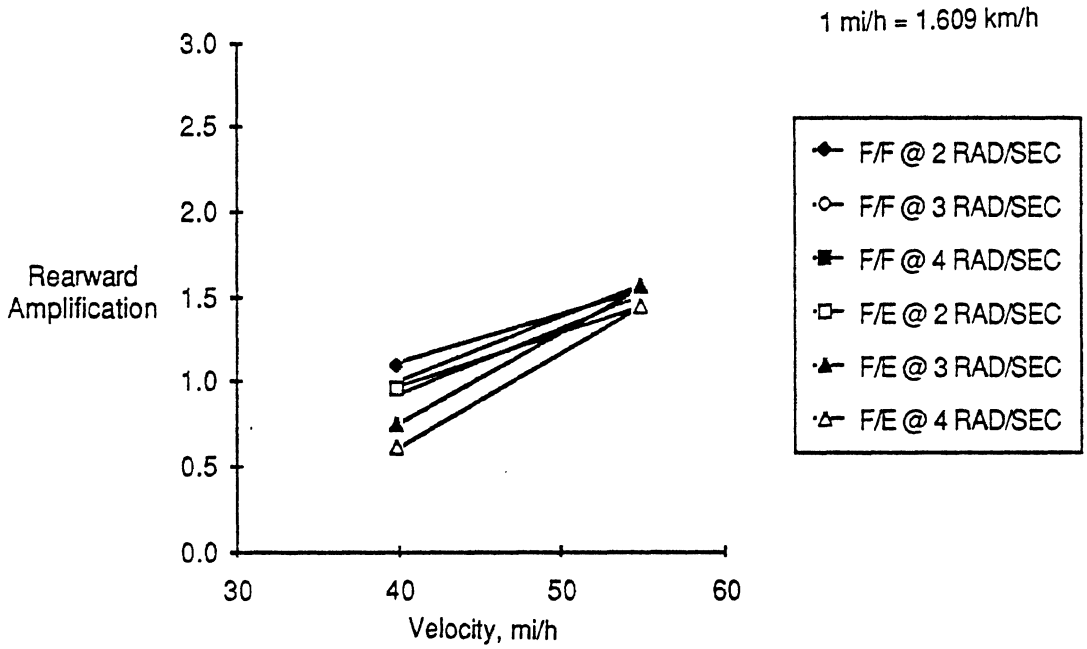


b. The trapezoidal dolly, forward IC position.

Figure 49. The influence of forward velocity on rearward amplification.

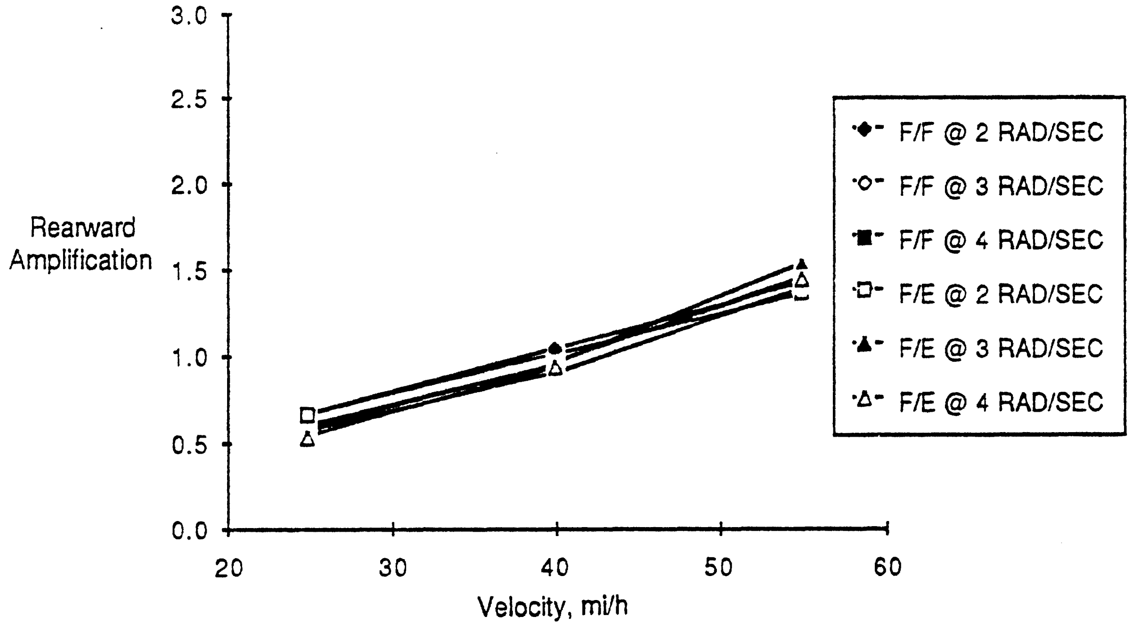


c. The trapezoidal dolly, rearward IC position.

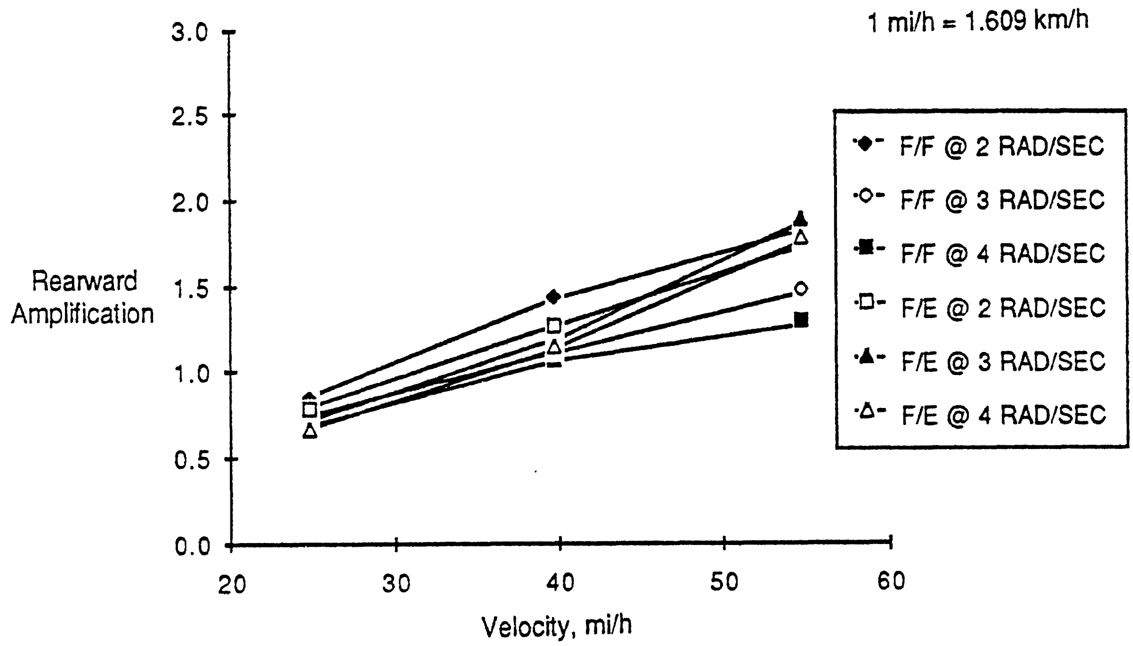


d. The linked-articulation dolly, 0.44 system gain.

Figure 49. The influence of forward velocity on rearward amplification.



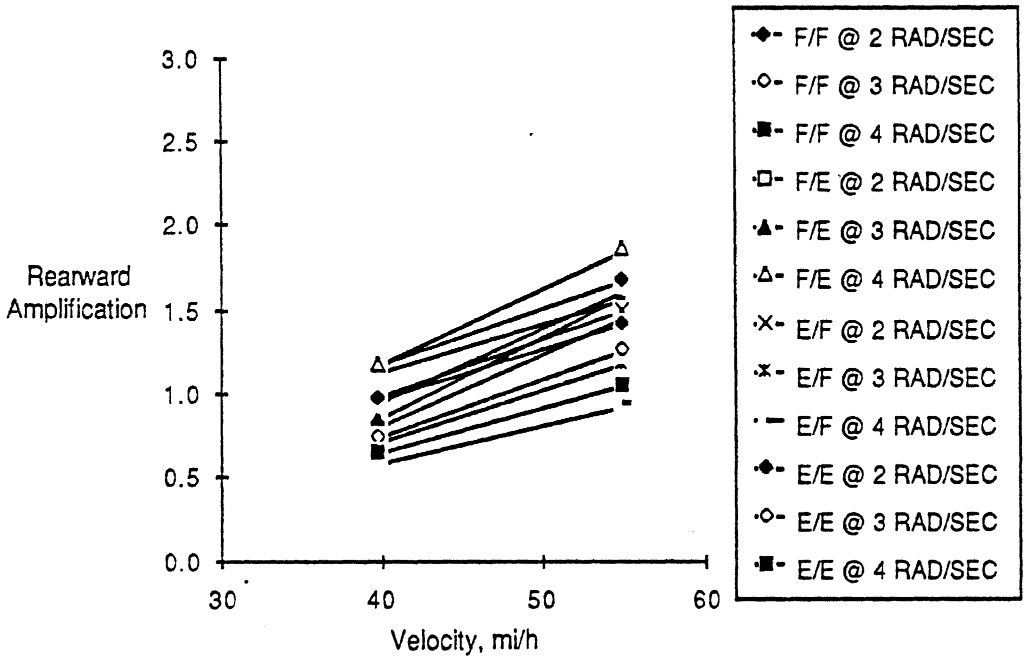
e. The self-steering B-dolly, full resistance steering.



f. The self-steering B-dolly, low resistance steering.

Figure 49. The influence of forward velocity on rearward amplification .

1 mi/h = 1.609 km/h



g. The CSB-dolly, 0.30 steering gain.

Figure 49. The influence of forward velocity on rearward amplification .

tenant. The sensitivities of rearward amplification to speed appear linear and range from about 0.025 to 0.055 h/mi (0.015 to 0.034 h/km).

Dynamic Rollover Threshold. The dynamic rollover threshold in the emergency lane-change maneuver of the four improved dolly types is shown in figure 50 in comparison to that of the A-train. In this portion of the study, this measure was taken at maneuvering frequencies of both 2 and 3 rad/sec. The measure was taken only in the full/full loading condition. The figure indicates that each of the improved dolly styles provides significant improvement in rollover threshold relative to the A-train. The two B-dollies clearly benefit from trailer-to-trailer roll coupling, and the full-resistance B-dolly is the best performer in this regard. Each of the vehicles is more resistant to rollover at the higher frequency. At 3 rad/sec, both of the B-dolly-equipped vehicles were still successfully resisting rollover at tractor maneuvering levels of 0.45 g. This was judged to be a more than reasonable maximum for this maneuver, and the exercise was stopped without obtaining rollover of these vehicles at this frequency.

Low-Speed Offtracking. The low-speed offtracking performance of the test vehicle equipped with the selected dollies is shown in figure 51. The performance of the A-train is shown as the usual reference, and the performance of the typical tractor-semitrailer with a 45-ft (14-m) trailer is also shown. The test vehicle performance is comparable or slightly better with each of the selected dollies than it is with the A-dolly, except for the trapezoid dolly in its forward IC position state. All of the doubles exhibit better performance than the single-trailer vehicle, pointing out the low-speed offtracking advantage of the double configuration in this area.

Yaw Damping Behavior. Simulation runs of the so-called "pulse-steer" maneuver were used to evaluate the influence of dolly type on yaw damping quality. The steering pulse consisted of 2 degrees of (roadwheel) steer for 0.2 sec duration. Figure 52 shows the lateral acceleration response of the tractor and second trailer of the A-train under the full/full and empty/full loading conditions. The tractor shows a sharp response to the pulse which generally excites the system. The oscillatory response of the second trailer then decays quickly, showing that the system is fairly well damped. The literature reveals that damping of the critical mode of motion of the A-train is decreased by larger loads and rearward location of loads; thus the choice of loading conditions shown.[5,6]

To examine all of the vehicles of the study in this manner, effective damping of the second-trailer lateral acceleration response was determined using the logarithmic decrement technique. The damping ratio of interest is that of the least lightly damped mode of motion. Accordingly, the logarithmic decrement should be taken late in the run, so that the other

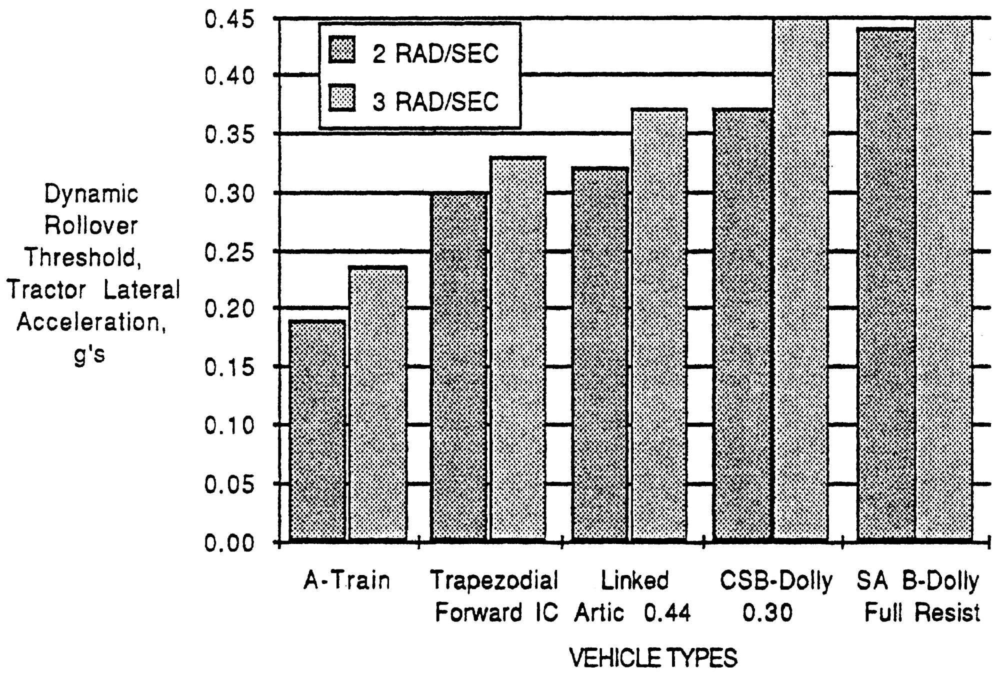


Figure 50. Dynamic rollover threshold of the improved dollies.

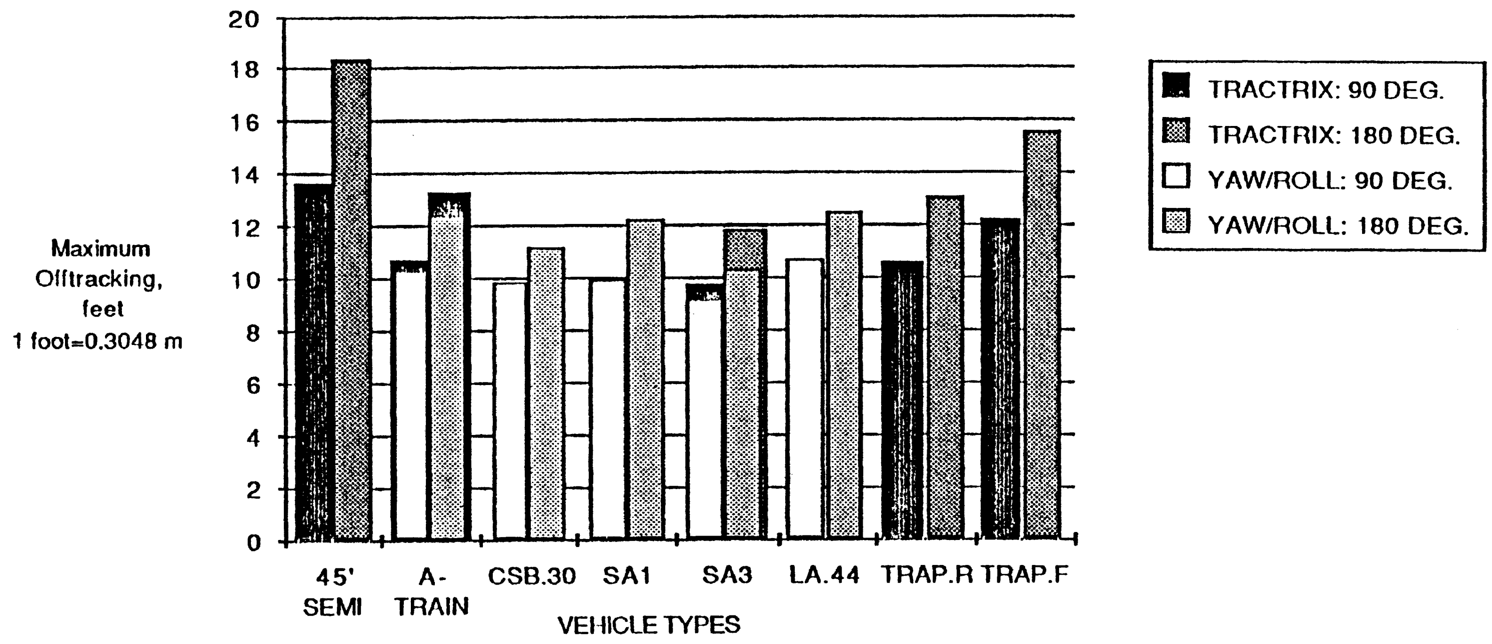
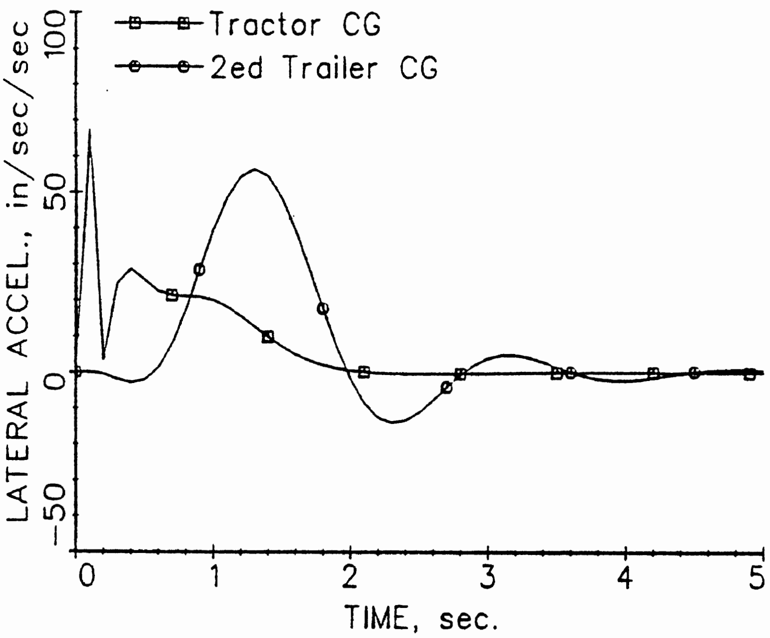
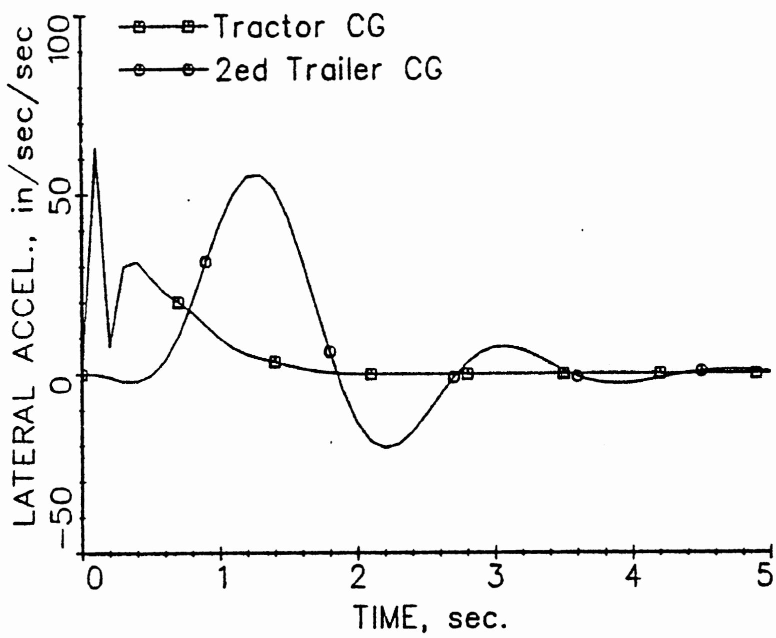


Figure 51. Low-speed offtracking performance of the selected dollies.



a) PULSE STEER: AT.F/F

1 in/sec/sec = 0.0254 m/sec/sec



b) PULSE STEER: AT.E/F

Figure 52. Lateral acceleration response to a steering pulse: the reference A-train.

modal responses have had time to die out. Referring to figure 52, in general, the first large negative peak (at about 2.3 sec.) and the following positive peak (at about 3.2 sec.) were used to calculate damping. (Some vehicles were so well damped that the second positive peak virtually did not exist, and the first positive peak had to be used.) Table 4 shows the calculated damping ratios for all of the runs conducted on the A-train and improved dollies. The test vehicles all displayed good damping properties with all of these dolly types in all the loading conditions tested. (As a point of reference in interpreting the values of tables 4 and 5, Klein and Szostak have recommended minimum damping ratios of 0.15 for passenger cars towing trailers.[23]) The trapezoidal dolly showed performance very near to the baseline A-train in both the forward and rearward IC conditions. Damping with the linked-articulation dolly and CSB-dolly was improved over the baseline (figures 53 and 54). When equipped with the self-steering B-dolly with full steering resistance, the vehicle was very well damped, as can be readily seen in figure 55.

As noted previously, it was to be expected that the level of steering resistance and the tongue length would have considerable influence on yaw damping performance of B-dollies. To demonstrate this influence, pulse-steer runs were conducted using the self-steering B-dolly with very low steering resistance and with long-drawbar B-dollies (figure 20.b). The long drawbar was applied to the self-steering B-dolly with both full and low levels of steering resistance and to the CSB-dolly. The damping ratios calculated for these vehicles appear in table 5, and example time histories are given in figures 56 through 59.

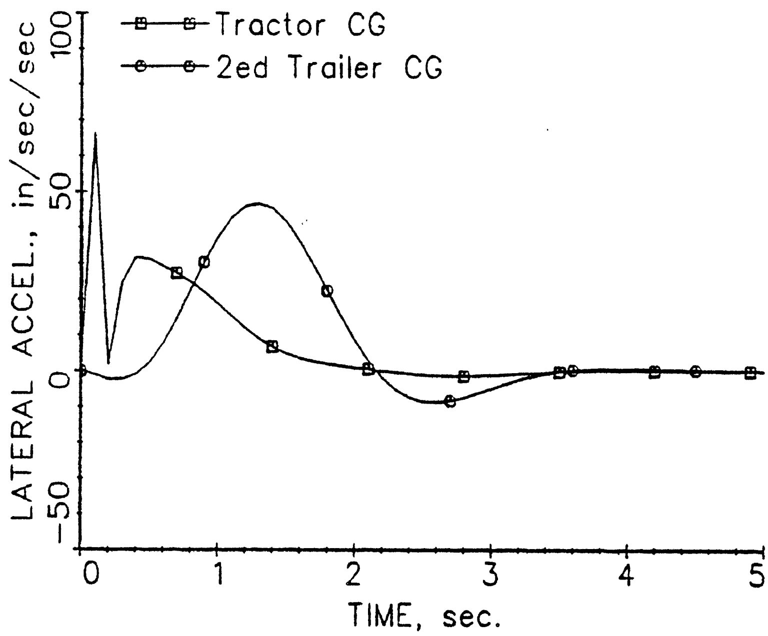
Figure 56 shows that, with the low-steering-resistance B-dolly, the fully loaded test vehicle is very lightly damped, and with load in the rear trailer only, this vehicle is unstable. Figure 57 simply shows that the long-drawbar configuration makes the performance of this same vehicle still worse, so that the vehicle also becomes unstable in the full/full loading condition.

The influence of long-drawbar geometry on the CSB-dolly was also examined. Figure 58 shows the time histories of lateral acceleration which result from applying the long drawbar to the CSB-dolly with a steering system gain of 0.30, i.e., the gain which was shown to be appropriate for the baseline test vehicle and has been used throughout. The long-drawbar configuration has clearly reduced the system damping in the CSB-dolly application.

Figure 58 reflects the influence of increased drawbar length only. To apply the CSB-dolly concept completely to the longer drawbar configuration requires a change in the steering gain to accommodate the change in longitudinal axle geometry. The appropriate steering gain to maintain Ackerman steering for the long-drawbar condition is 0.43. Figure

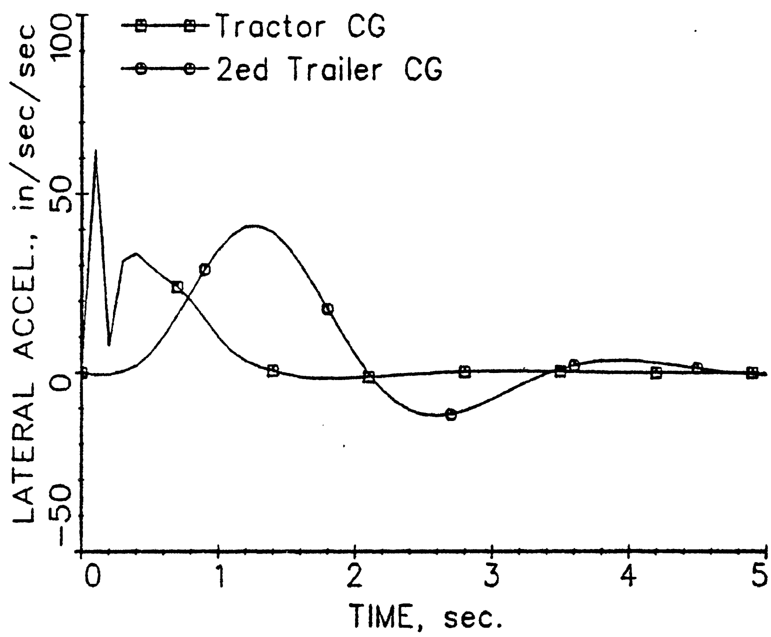
Table 4. Effective Damping Ratio of the Test Vehicle in a 55 mi/h (88.5 km/h) Pulse-Steer Maneuver, Equipped with the Improved Dollies and Under Differing Load Conditions.

Dolly Type	Load Condition	Damping Ratio
A-Train Reference	F/F	0.32
	E/F	0.31
Trapezoidal Dolly, Forward IC Position	F/F	0.37
	E/F	0.35
Trapezoidal Dolly, Rearward IC Position	F/F	0.32
	E/F	0.31
Linked Articulation Dolly, 0.44 System Gain	F/F	0.59
	F/E	0.72
	E/F	0.37
	E/E	0.50
Self-Steering B-Dolly, Full Steering Resistance	F/F	0.68
	E/F	0.51
CSB-Dolly, 0.30 Steering Gain	F/F	0.55
	F/E	0.74
	E/F	0.34
	E/E	0.45



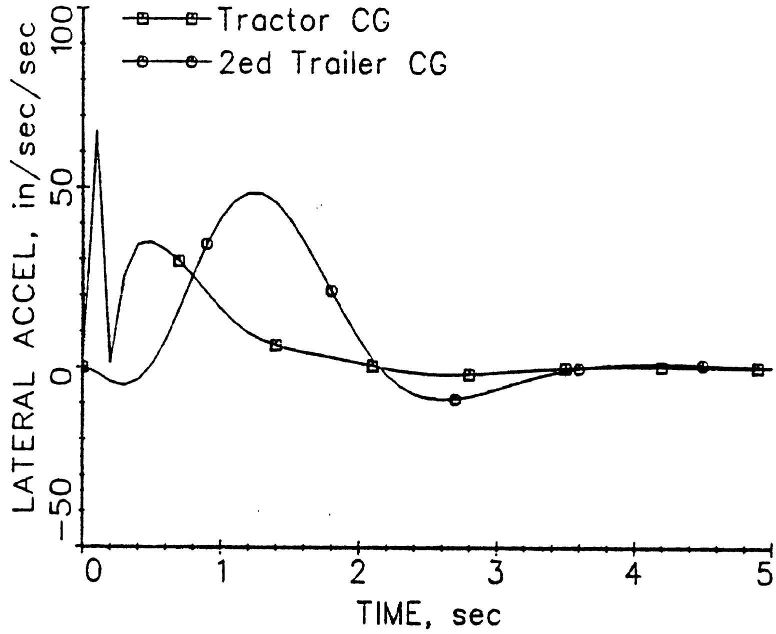
a) PULSE STEER: LA.44.F/F

1 in/sec/sec = 0.0254 m/sec/sec



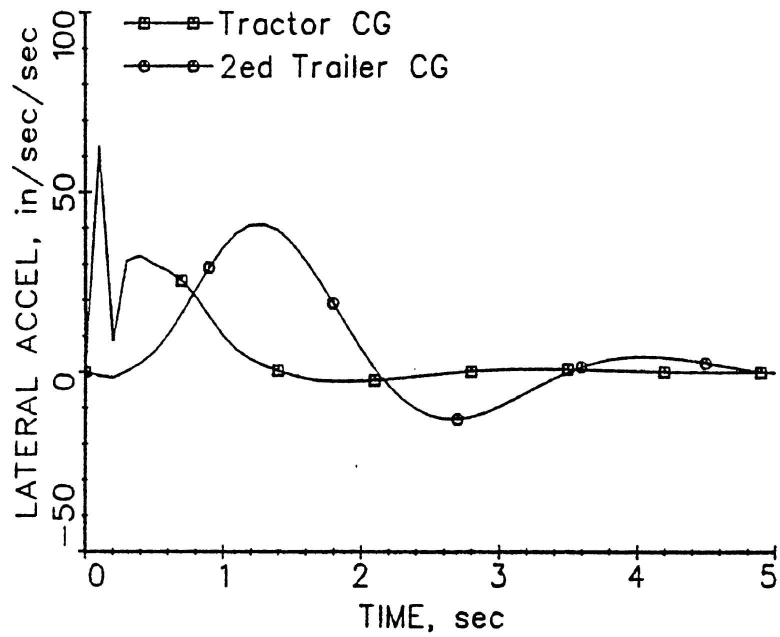
b) PULSE STEER: LA.44.E/F

Figure 53. Lateral acceleration response to a steering pulse: the linked-articulation dolly, 0.44 system gain.



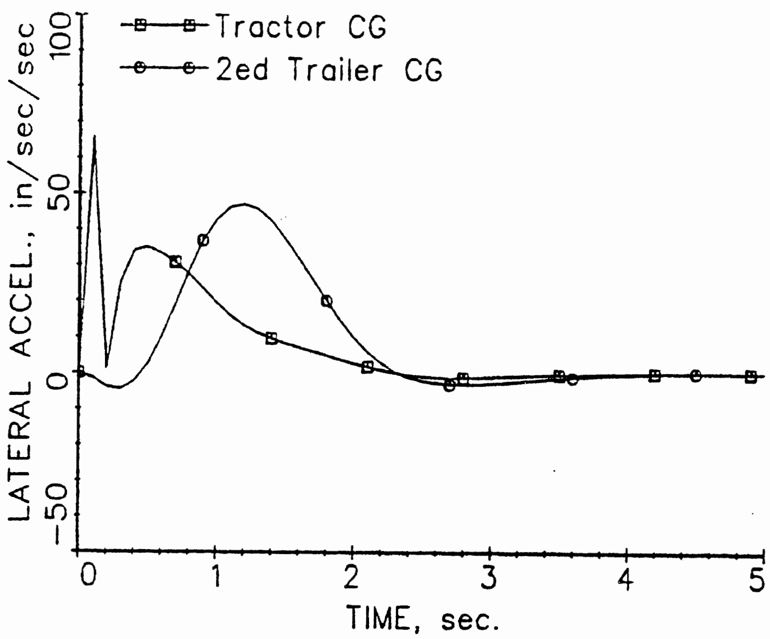
a) PULSE STEER: PRO.30.F/F

1 in/sec/sec = 0.0254 m/sec/sec



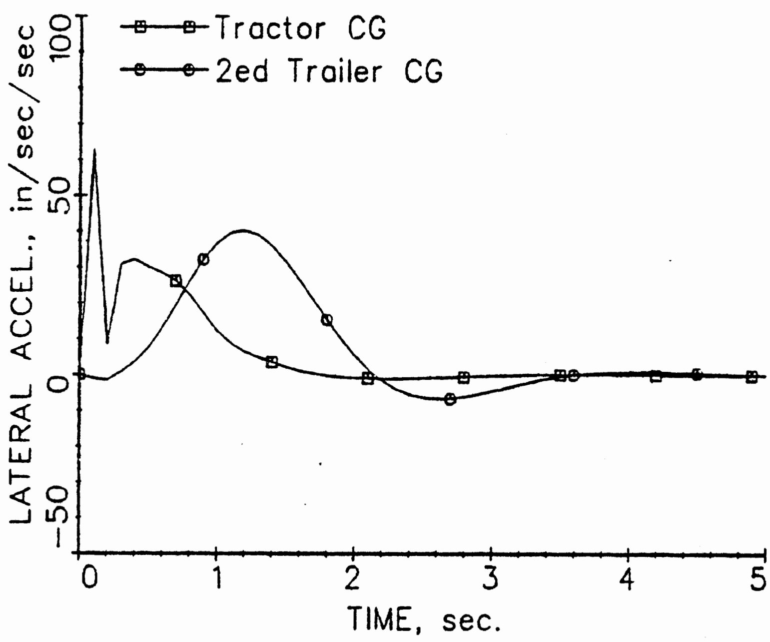
b) PULSE STEER: PRO.30.E/F

Figure 54. Lateral acceleration response to a steering pulse: the CSB-dolly, 0.30 steering gain.



a) PULSE STEER: SA1.F/F

1 in/sec/sec = 0.0254 m/sec/sec



b) PULSE STEER: SA1.E/F

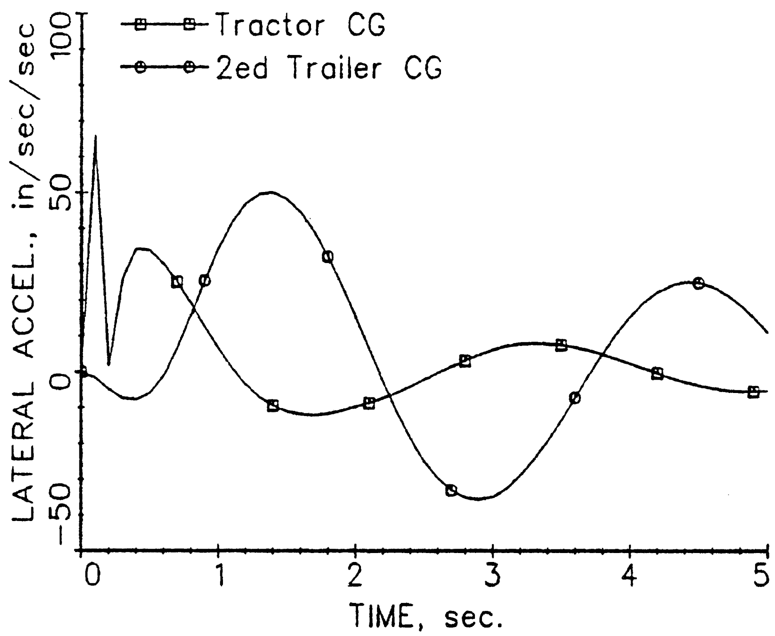
Figure 55. Lateral acceleration response to a steering pulse: the self-steering B-dolly, full steering resistance.

Table 5. The Influence of Dolly Drawbar Length and Steering Properties on the Damping Ratio of Test Vehicle Equipped with the B-Dollies.

Dolly Type	Load Condition	Steering Property	Drawbar Length, in.	Damping Ratio
Self-Steering B-Dolly	F/F	Full Resistance	80	0.68
		Full Resistance	160	0.65
		Low Resistance	80	0.11
		Low Resistance	160	-0.10*
	F/E	Low Resistance	80	0.51
	E/F	Low Resistance	80	-0.16*
	E/E	Low Resistance	80	0.16
CSB-Dolly	F/F	$G\delta_4\Gamma_3 = 0.30$	80	0.55
	F/F	$G\delta_4\Gamma_3 = 0.30$	160	0.48
	F/F	$G\delta_4\Gamma_3 = 0.43$	160	0.32

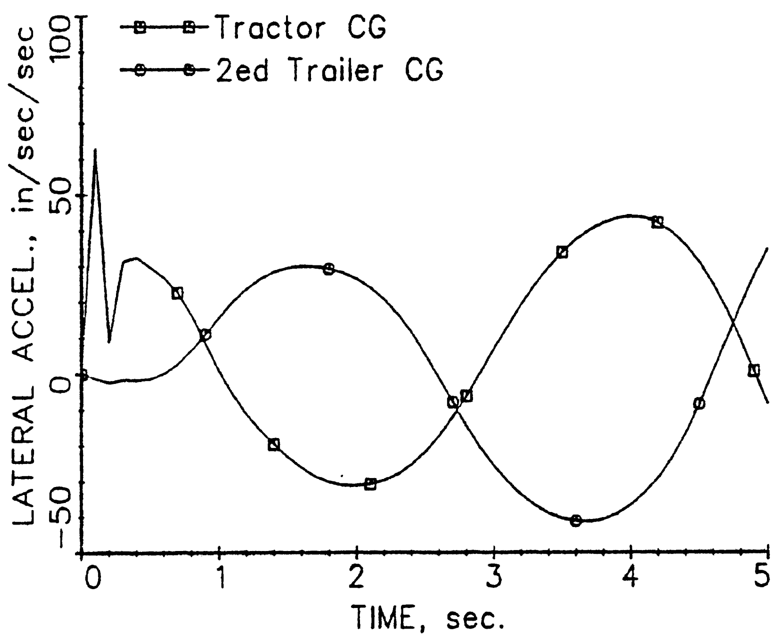
1 in = 0.0254 m

*Negative damping indicates an unstable system.



a) PULSE STEER: SA3.F/F

1 in/sec/sec = 0.0254 m/sec/sec



b) PULSE STEER: SA3.E/F

Figure 56. Lateral acceleration response to a steering pulse: the self-steering B-dolly, low steering resistance.

1 in/sec/sec = 0.0254 m/sec/sec

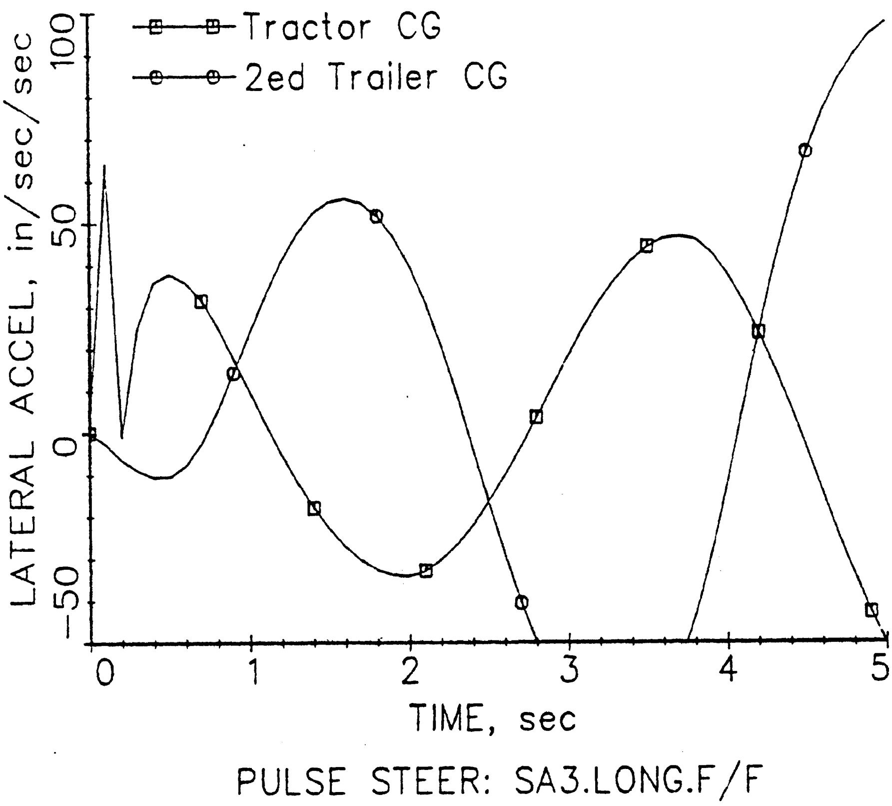


Figure 57. Lateral acceleration response to a steering pulse: the self-steering B-dolly, long drawbar, low steering resistance.

1 in/sec/sec = 0.0254 m/sec/sec

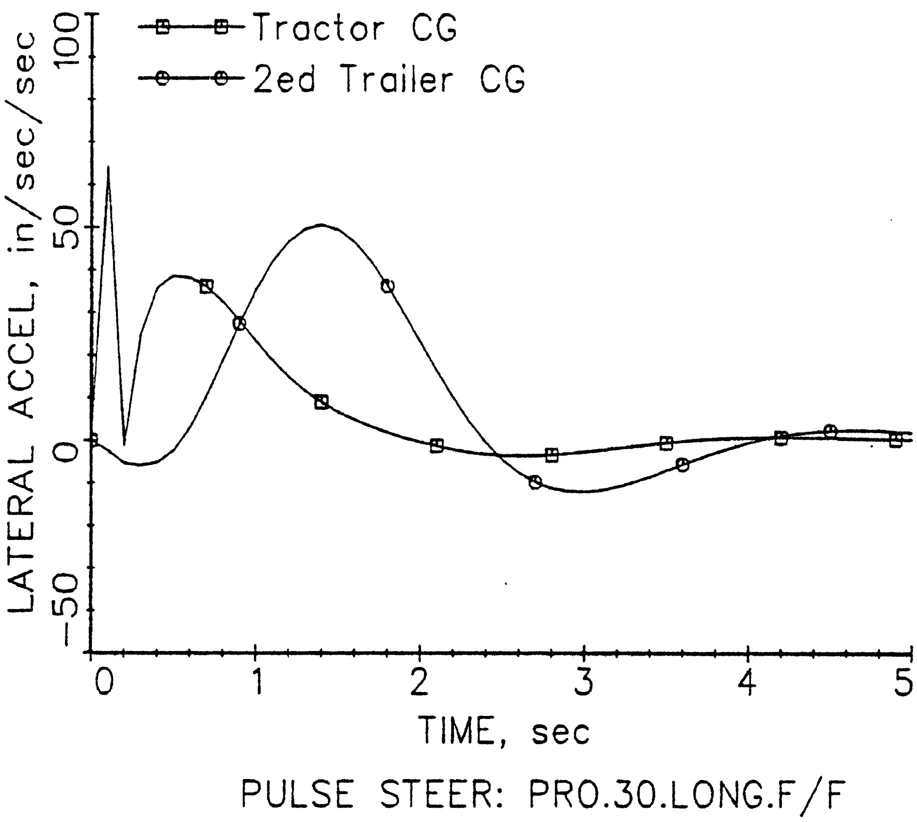


Figure 58. Lateral acceleration response to a steering pulse: the CSB-dolly, long drawbar, 0.30 steering gain.

1 in/sec/sec = 0.0254 m/sec/sec

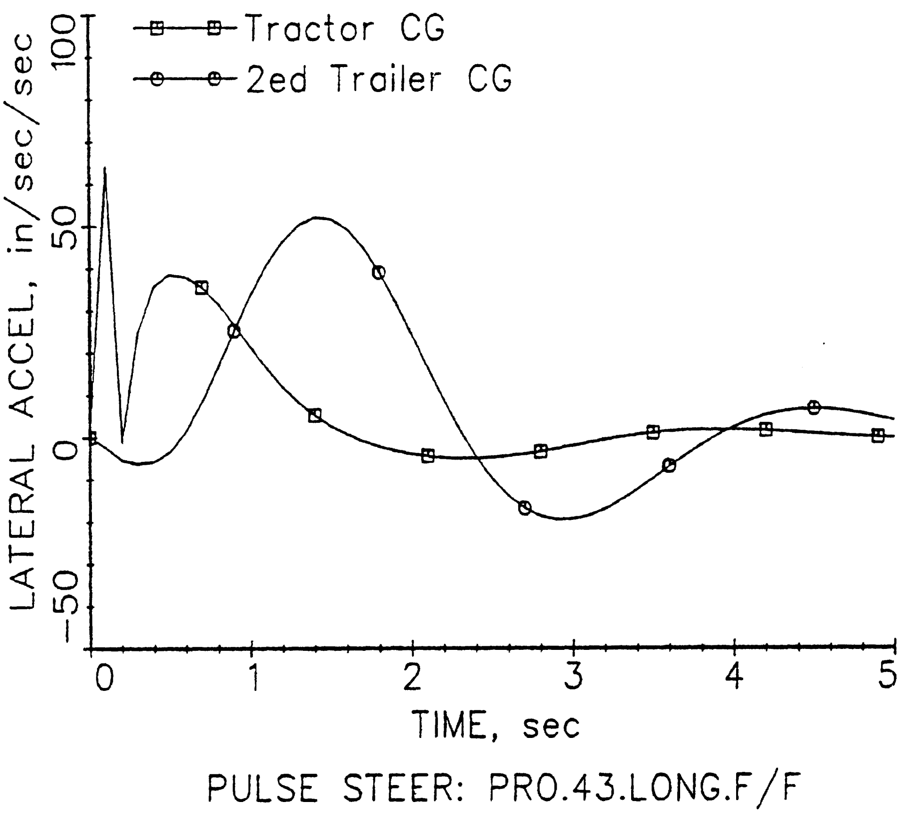


Figure 59. Lateral acceleration response to a steering pulse: the CSB-dolly, long drawbar, 0.43 steering gain.

59 indicates that, while the system is still reasonably well damped, damping has clearly been significantly reduced in this long-drawbar configuration of the CSB-dolly vehicle.

Drawbar Hitch Loadings. One advantageous feature of the traditional A-dolly configurations is that hitch loads resulting from steering and rolling actions of the vehicle are small. This is not true of either of the B-dolly or the linked-articulation configurations. (Forward acceleration, and deceleration under braking, as well as longitudinal "chugging," produce substantial longitudinal and vertical pintle loads in the A-train configuration, but none of the new configurations substantially alter these loads.)

The maximum loadings of interest, achieved in the simulation study, came from the runs used to determine the dynamic rollover threshold of the fully loaded test vehicle. Table 6 lists the maximum hitch loads achieved in the most severe simulation runs in which rollover did not actually occur. The "resultant" loads are the resultant forces and moments which exist across the coupling of the first trailer and dolly, defined by an axis system at the "hitch center," i.e., the "equivalent" pintle hitch point, as shown in figure 60. The "component" loads are forces which would develop in "typical" hardware members and joints, also as shown in figure 60.

Table 6 shows that (1) the trapezoidal dolly, as expected, produces no significant loading problems over and above the conventional A-dolly; (2) for both the linked-articulation dolly and the two B-dollies, the yaw coupling moment is substantial (as, after all, it must be, to so substantially alter the yaw response of these vehicles) and results in a couple composed of large longitudinal forces at either the B-dolly pintles or the "steering-stabilizer" hinge joint; (3) the roll coupling moment exceeds 600,000 in-lb (67,300 N-m) for both B-dollies, producing another couple at the pintles composed of large vertical forces. This third result, however, is highly dependent on the torsional stiffness of the two trailers and of the dolly about their respective longitudinal, elastic axes. While the simulation treats these bodies as rigid in this regard, they of course are not. The choice of 30,000 in-lb/deg (3,390 N-m/deg) of roll coupling compliance at the pintle is, essentially, an educated guess at attempting to "lump" the influence of these three compliances. Since in the simulation process, many of the approximations made are of the type in which (slightly) compliant bodies are assumed to be rigid, we can expect that the simulation programs may be predicting excessive roll coupling moments (and overly effective roll coupling.)

Braking. The purpose of these braking calculations is to investigate the possibility that some unforeseen stability problem could arise for one of the innovative dollies when the brakes are applied. Five vehicle configurations considered are referred to as the "A-train,"

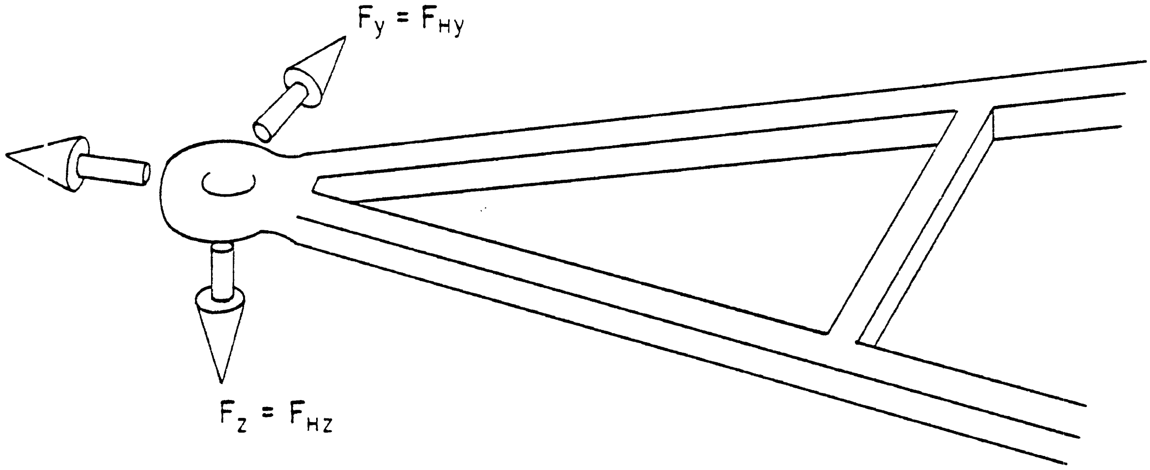


Figure 60a. The A-dolly hitch loads.

1 in = 0.0254 m

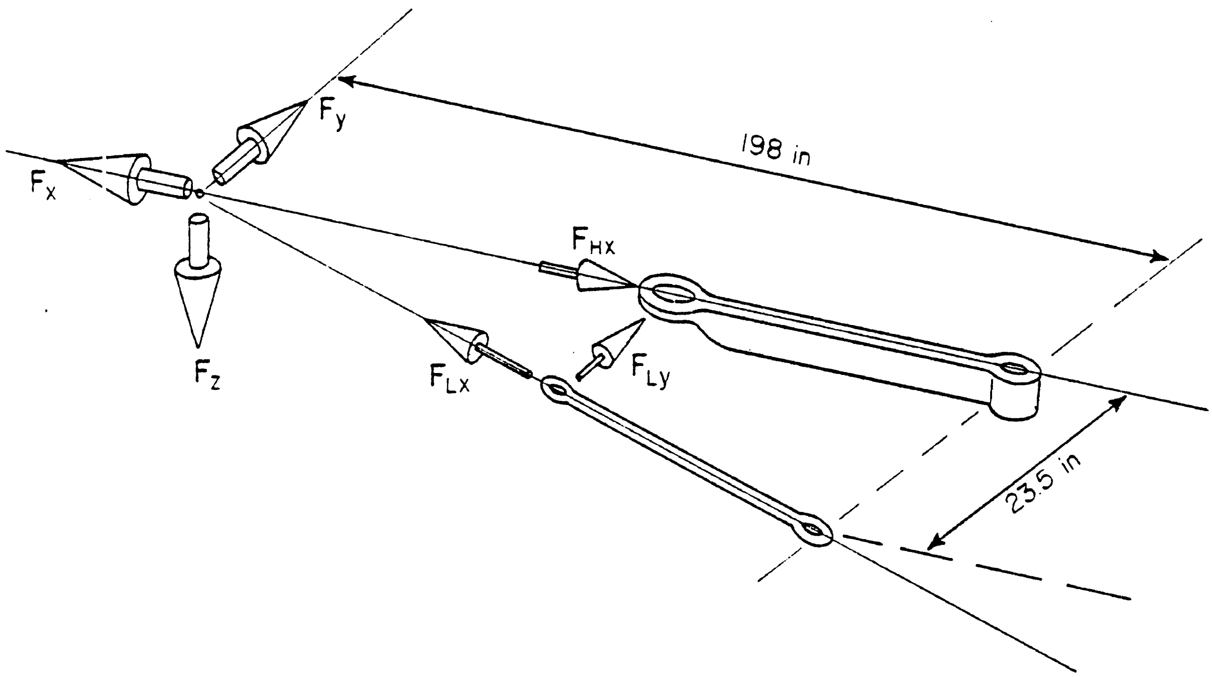


Figure 60b. The trapezoidal dolly hitch loads.

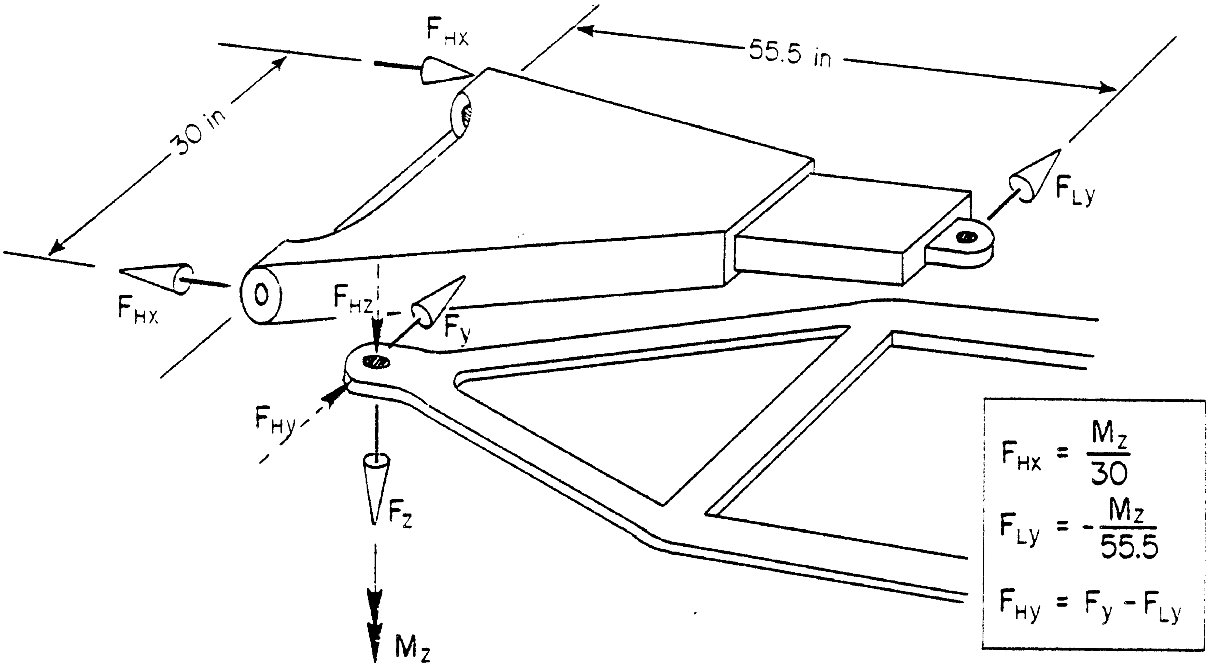


Figure 60c. Linked articulation dolly hitch loads.

1 in = 0.0254 m

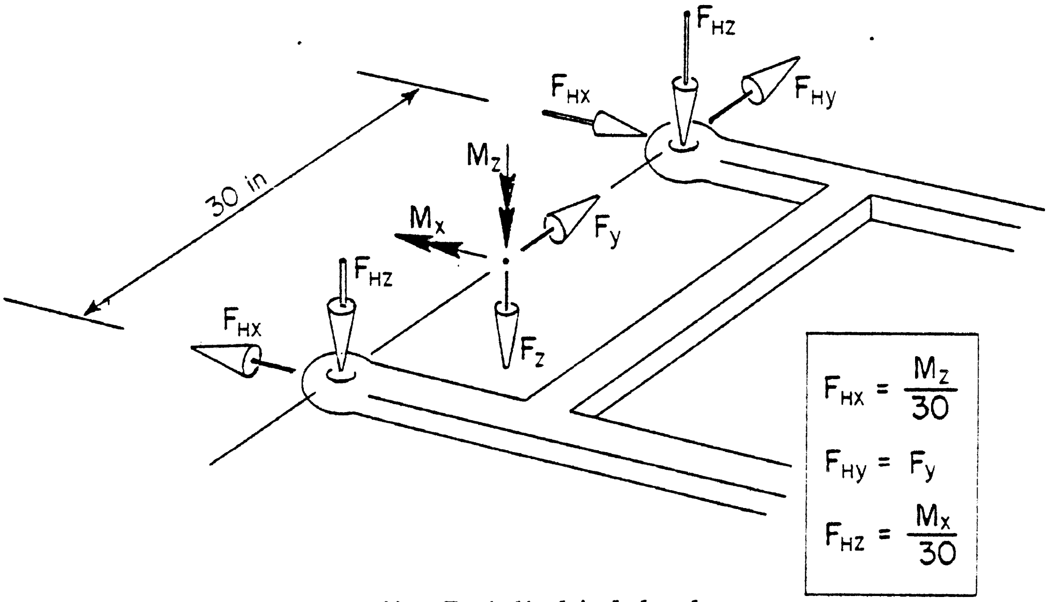


Figure 60d. B-dolly hitch loads.

Table 6. Maximum Absolute Drawbar Hitch Loads in Emergency Lane-Change Maneuvers at the Rollover Threshold

Dolly Type	Lane Change Frequency	Resultant Forces and Moments At the Drawbar Center			"Component" Forces and Moments				
	rad/sec	F_y, lb	M_x, in-lb	M_z, in-lb	F_{Hx}, lb	F_{Hy}, lb	F_{Hz}, lb	F_{Lx}, lb	F_{Ly}, lb
A-Train, Reference	3	147				147	85		
Trapezoidal Dolly, Forward IC Position	2	55				460		460	55
	3	56				470		470	56
Linked Articulation Dolly, 0.44 System Gain	2	4,014		207,100	6,900	300			3,700
	3	4,854		253,900	8,500	250			4,600
Self-Steering Dolly, Full Steering Resistance	2	8,660	672,400	659,800	22,000	8,660	22,413		
	3	5,894	312,600	473,200	15,800	5,894	10,400		
CSB-Dolly, 0.30 Steering Gain	2	4,591	640,100	346,200	11,500		21,300		
	3	5,270	535,300	418,600	14,000	17,800			

1 lb = 4.448 N

1 in-lb = 0.113 N-m

"B-train," "prototype," "linked-articulation," or "forward IC," depending upon the type of dolly installed in the doubles combination. In this case, the B-train employed a steerable B-dolly in which the dolly wheels could steer if the preload torque was exceeded. The so-called "Phase 4" (comprehensive braking and steering model) was used to simulate three braking maneuvers: (1) straight-line braking on a "poor, wet road" with the vehicle empty, (2) braking-in-a-turn for both the empty vehicle when the road was wet and the loaded vehicle when the road was dry, and (3) braking the loaded vehicle on a split-friction road for which the right side corresponded to a poor, wet surface and the left side corresponded to a good, dry surface. The results from these calculations served as a screening for directional control problems during severe braking.

Although these types of maneuvers are often considered for use in vehicle test programs, they pose difficulties for evaluating the effects of dolly properties through computer simulation. First, the typical heavy truck (with brakes as proportioned in the United States) will have a tendency to jackknife if too much braking is applied for the available tire-road friction. This undesirable instability is brought about by properties of the braking system and, although dolly properties may have an influence on when jackknifing occurs, the dolly is not the basic problem. The problem is that the front brakes on the tractor are not effective enough and the rear brakes on the tractor are too effective for the load that they carry.

Another difficulty in simulating directional control during braking is that a "driver" representation is needed to attempt to steer the vehicle along a desired path. The properties of the driver interact with those of the vehicle to determine the overall result. To the extent that the driver model cannot simulate the emergency performance of a person trying to avoid a jackknife, information on the limit performance of the driver-vehicle system needs to be evaluated on the proving grounds through vehicle testing.

Nevertheless, simulation can be used to show whether high levels of deceleration (compared to the maximum available tire-road friction) can be attained with only modest steering activity required to maintain directional control. The results that follow show the levels of braking performance that have been predicted for situations in which dollies have been interchanged, but otherwise the vehicles and "drivers" have been kept constant.

As shown in figure 61, the straight-line braking performances of all of the vehicle/dolly combinations exceed 8 feet per second squared (0.25 g) in a situation in which the vehicle is empty and the road is wet. Above this level of deceleration, wheels on various axles will lock up and directional control problems will ensue equally for the A-train as well as for any of the combinations employing innovative dollies.

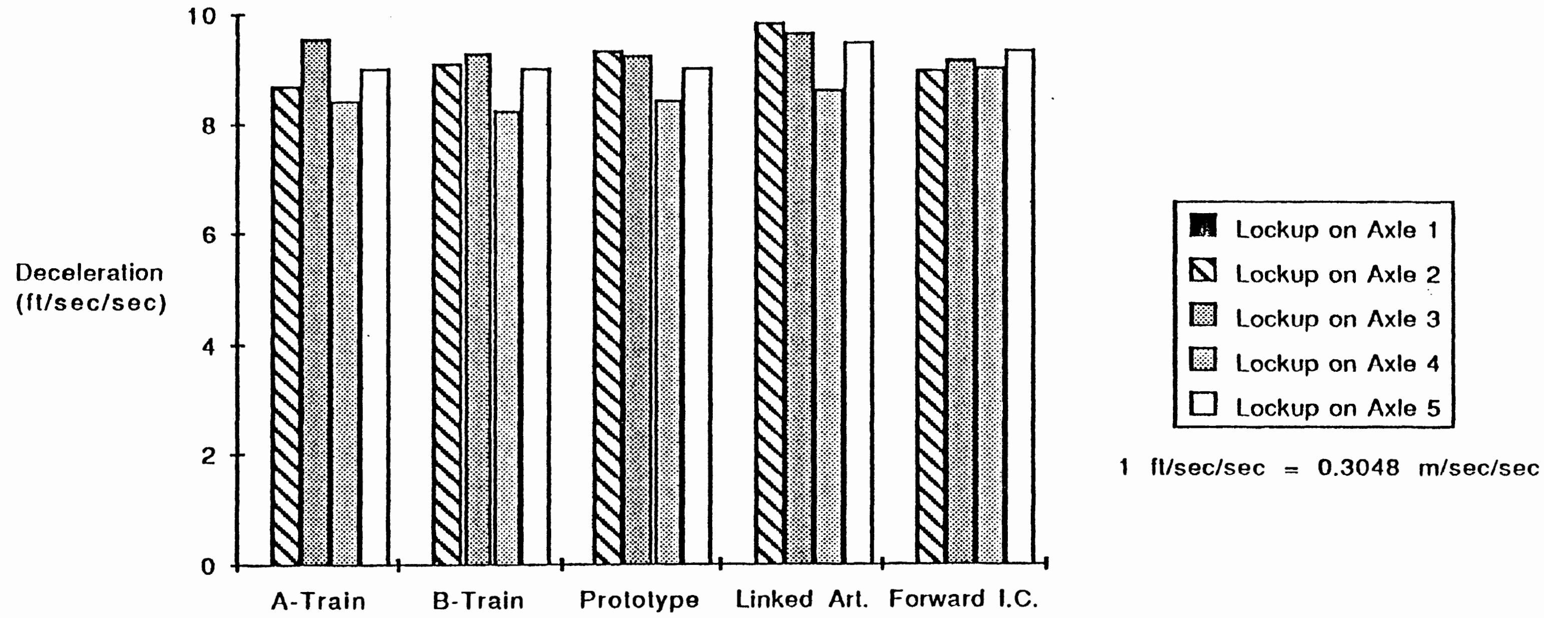


Figure 61. Straight-line braking (wet surface), deceleration at the occurrence of wheel lock, empty vehicles.

The levels of deceleration at which excessive articulation angles develop are presented in figure 62. These results do not indicate a problem with any one of the dollies. The A-train gets into directional response problems at approximately the same level of maneuver as that of any of the other combinations.

The efficiency with which doubles can utilize tire-road friction to obtain deceleration is known to be low (on the order of 0.5 to 0.6) when the vehicle is empty. The results presented here are typical regardless of the dolly involved.

The second set of results to be presented here pertain to braking in a turn. This maneuver is selected to provide a greater challenge to directional control than straight-line braking. In the first instance, the maneuver is simulated for the empty vehicle performing a turn at 40 mi/h (64 km/h) with a lateral acceleration of 0.15 g . The surface is again the poor, wet road.

The results show that all of the doubles combinations can achieve decelerations exceeding 0.25 g without encountering lockup of all the wheels on any axle (see figure 63). In this case, the patterns of axle lockup vary, depending upon the dolly type. Even though further study would be required to explain why the linked-articulation dolly is predicted to increase the tendency for the tractor's axles to lock, this property is worth mentioning because a situation in which the tractor's rear wheels lock first will lead to a violent jackknife. As shown in figure 64, the simulation predicts a large articulation angle between the tractor and the first semitrailer at a low level of lateral acceleration when the linked-articulation dolly is employed. However, the subsequent vehicle tests did not indicate that the predicted tendency was so strong that the test driver could not control the test vehicle. Furthermore, as already observed, the jackknifing tendency is fundamentally a brake system problem. In this case, the properties of the linked-articulation dolly appear to exacerbate the existing problem.

In the next evaluation using the braking-in-a-turn maneuver, the vehicles were loaded and the road surface was dry with a high coefficient of friction. All of the vehicles were able to perform well under these conditions, as indicated in figures 65 and 66. Again, the vehicle with the linked-articulation dolly had the poorest performance because the tractor's rear axle locked first (at the lowest level of lateral acceleration). However, in this case, the vehicle achieved a high level of deceleration before a large articulation angle developed (see figure 66).

Figure 67 shows that all of the vehicles could achieve good braking performance when one side is on wet pavement and the other side is on dry pavement. One of the reasons for

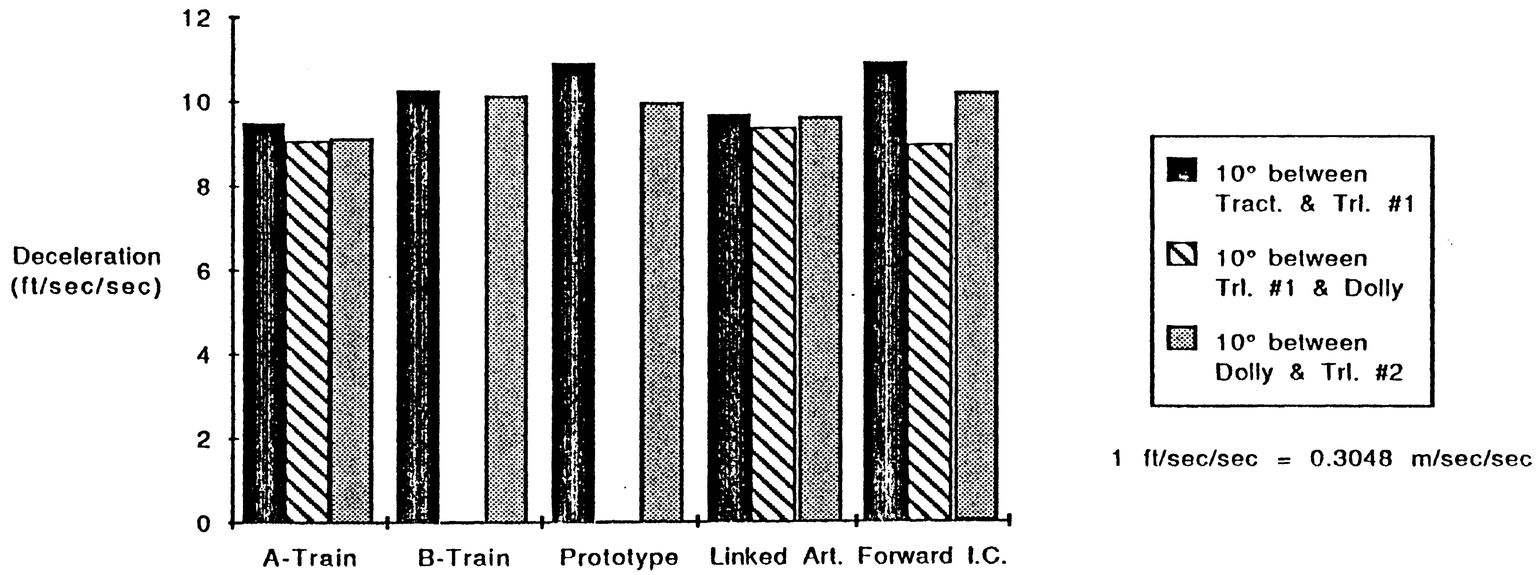


Figure 62. Straight-line braking (wet surface), deceleration at the occurrence of 10° articulation angles, empty vehicles.

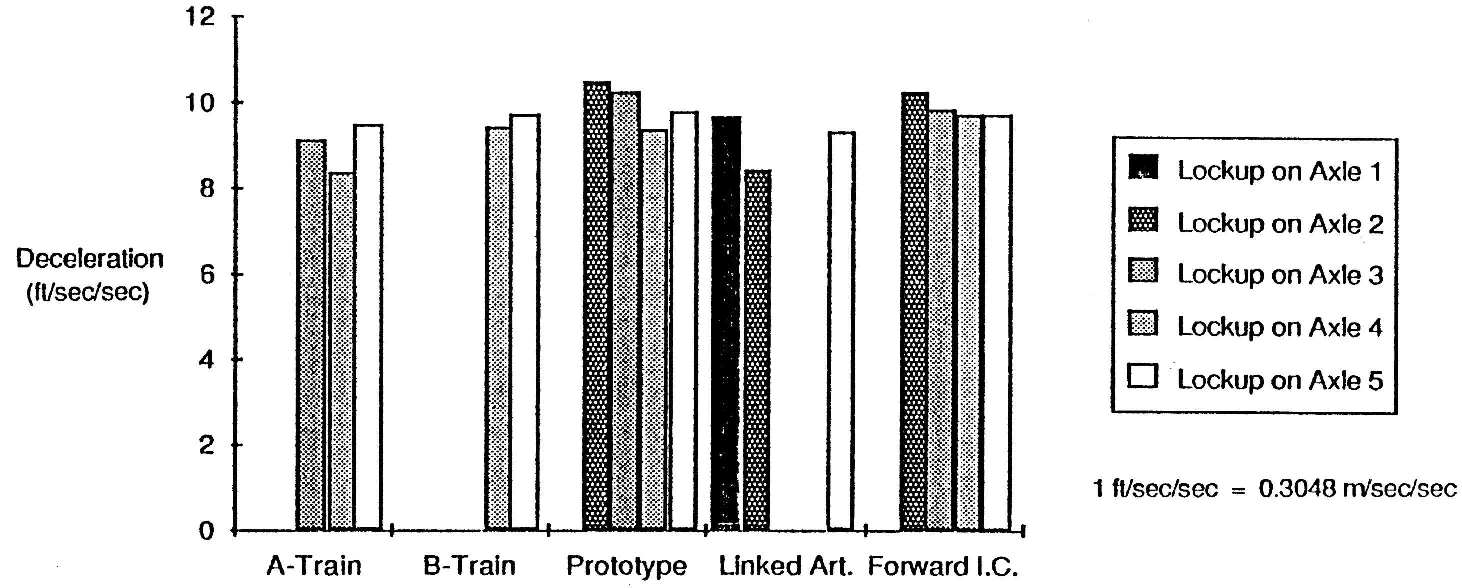


Figure 63. Braking in a turn (wet surface), deceleration at the occurrence of wheel lock, empty vehicles.

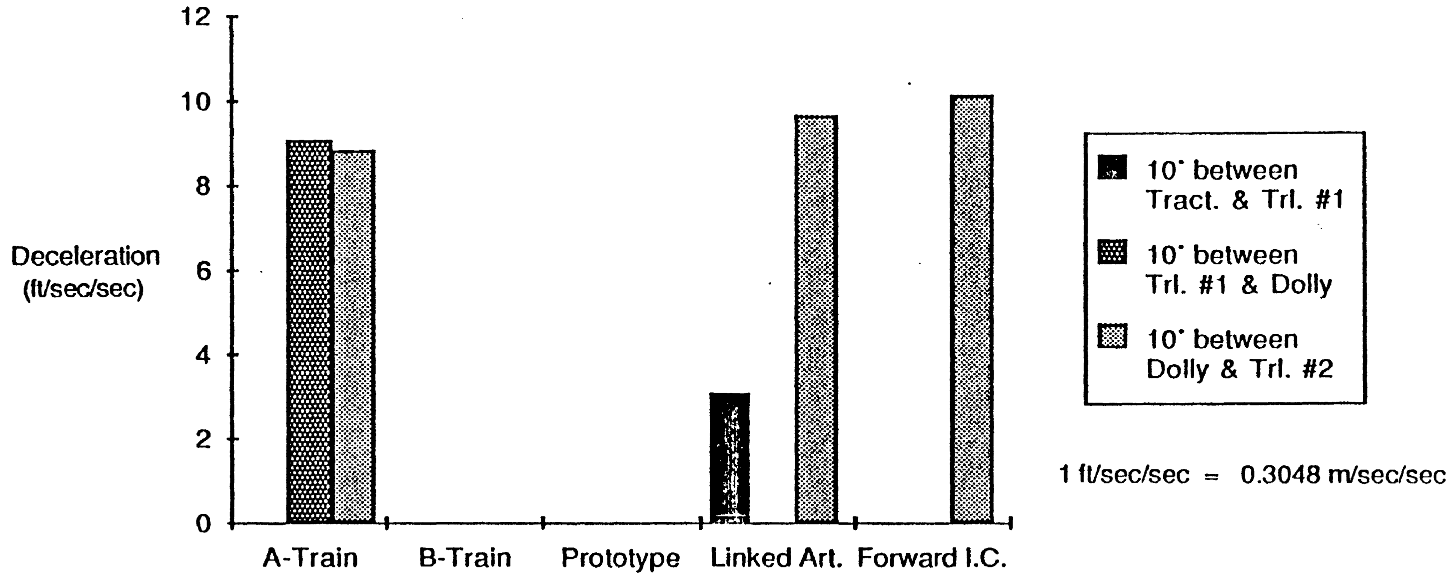


Figure 64. Braking in a turn (wet surface), deceleration at the occurrence of 10° articulation angles, empty vehicles.

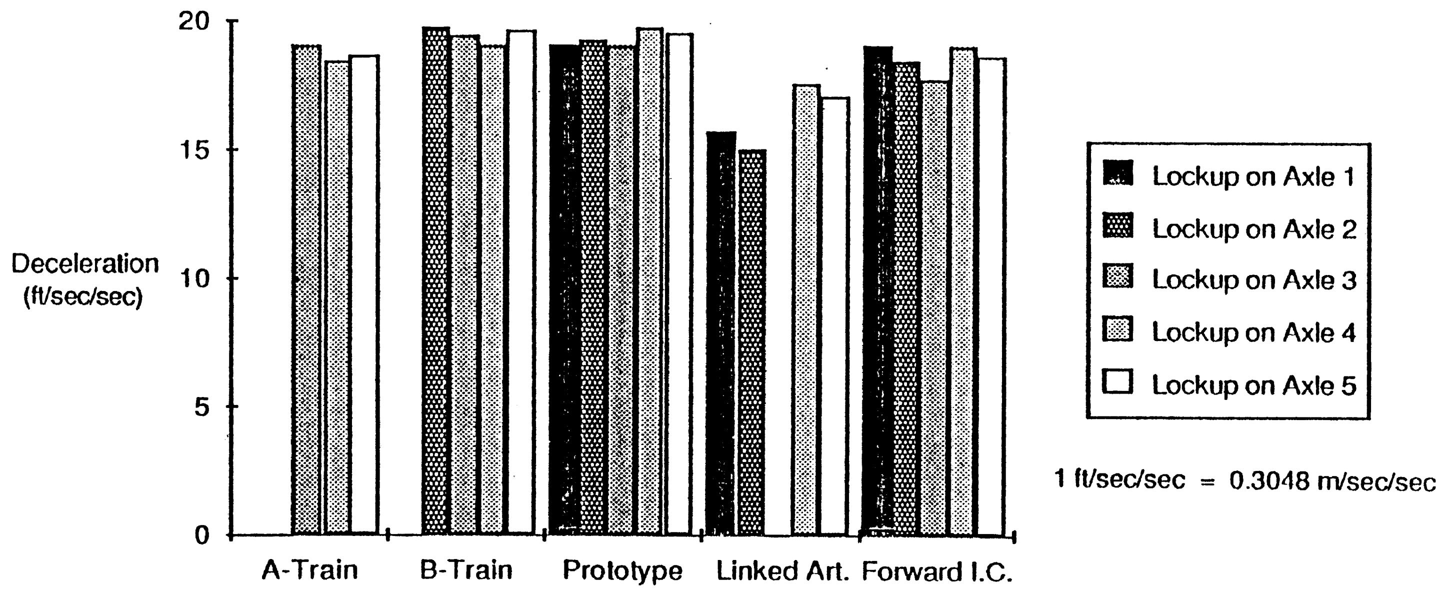


Figure 65. Braking in a turn (dry surface), deceleration at the occurrence of axle lock, loaded vehicles.

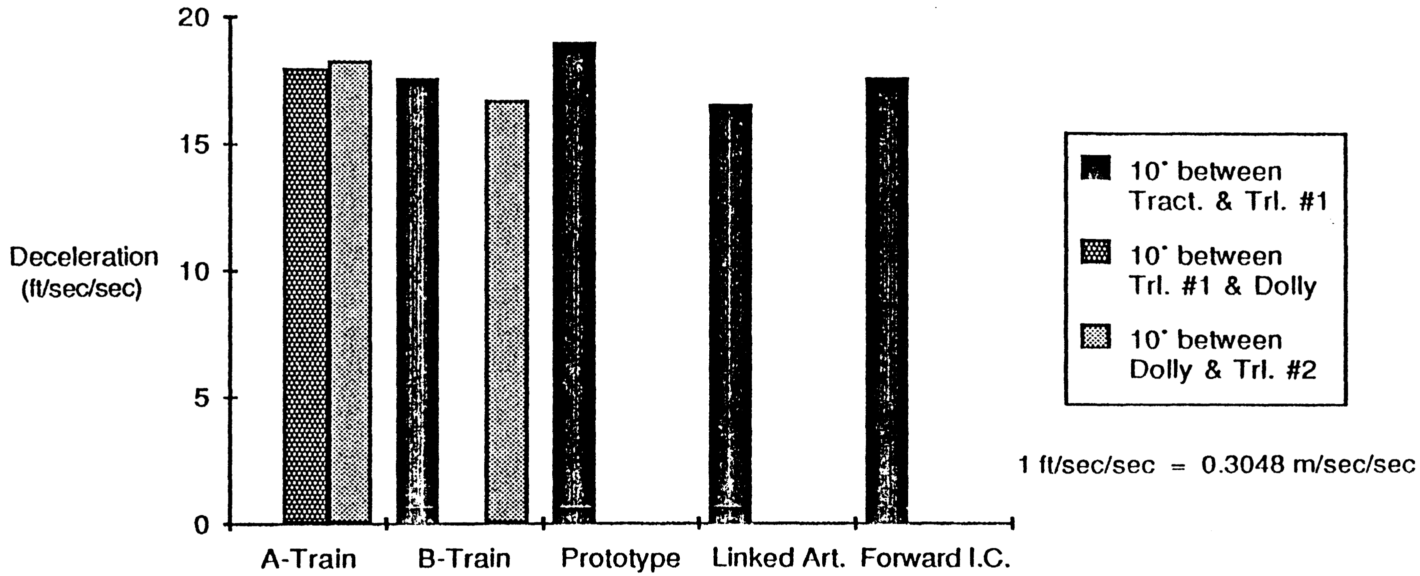


Figure 66. Braking in a turn (dry surface), deceleration at the occurrence of 10° articulation angles, loaded vehicles.

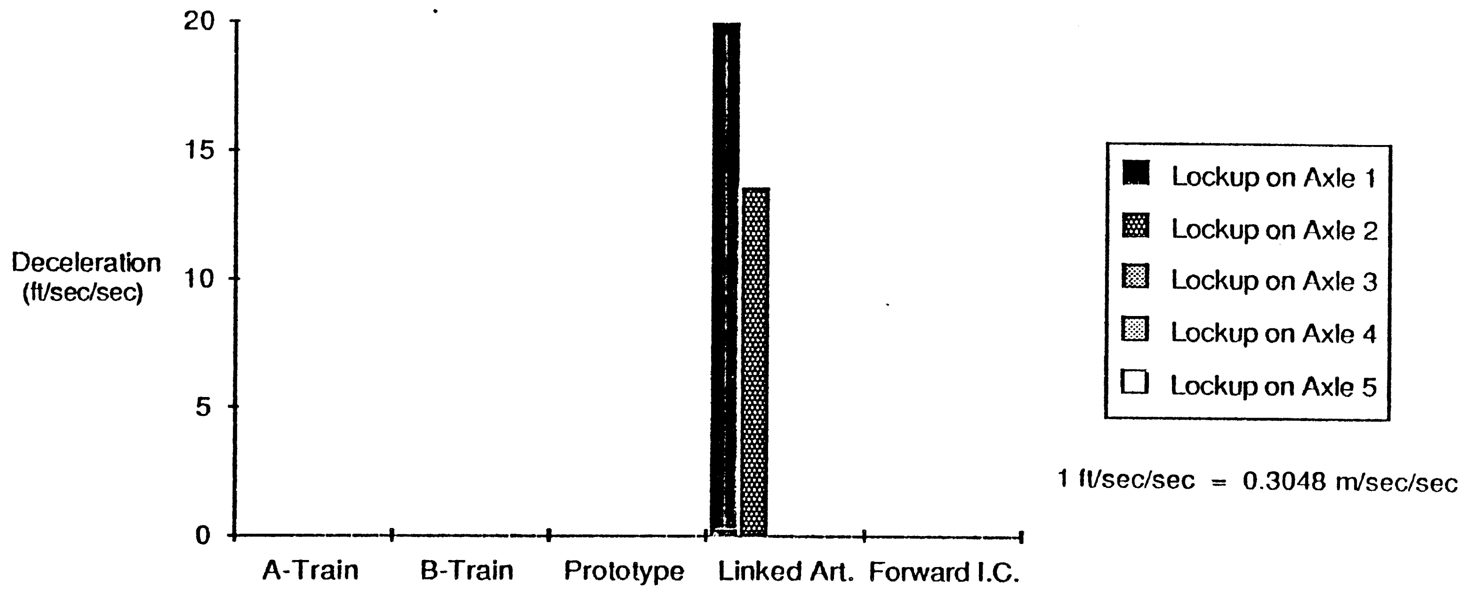


Figure 67. Straight-line braking (split surface), deceleration at the occurrence of wheel lock, loaded vehicles.

pursuing this maneuver was to see if the side-to-side difference in braking force would steer the steerable wheels mounted on the B-dolly. It did. However, even though these wheels steered towards the dry side of the road, the influence of the side forces created did not disrupt the ability of the simulated driver to maintain directional control. This same result was also observed during vehicle testing and in the simulated braking-in-a-turn maneuver for the loaded vehicle on the dry surface. Apparently this unplanned steering does not disrupt directional behavior to the extent that the driver loses control of the path of the vehicle.

In figure 67, only the performance of the linked-articulation vehicle reaches the point where both wheels on an axle lock. It is unfortunate that an appropriate explanation for this phenomenon is not known. Since the prototype dolly is based on achieving steering similar to that of the linked-articulation dolly, one might wonder why the prototype configuration does not show similar directional control difficulties. This has led us to scrutinize test results. The test results indicate that drivers can handle these vehicles with no more trouble than braking the A-train. Nevertheless, the findings of this simulation study warrant further investigation.

3. Findings with Respect to the Use of Mixed-Width Axles

As indicated earlier, the second technical subject addressed in the simulation study was concerned with the influence of mixing hardware having different widths. This information is out of the mainstream of the study of innovative dollies, but it was convenient to perform the desired analyses as a part of the overall simulation study.

Additional simulations were run for the conventional, A-type, five-axle double equipped with axles having both 96- and 102-in (2.44- and 2.59-m) overall width. Also, calculations were made for a case in which the A-train was converted to a double-drawbar B-train. The problem addressed in these calculations concerns the roll stability implications of using dollies and trailers, in combination, which are constructed with differing-width axles. At present, it is known that some carriers are operating with mixed hardware in their fleets. Some newer units have been purchased with 102-in- (2.59-m-) width running gear while older trailers and dollies having 96-in (2.44-m) widths are also in use in the same fleets. Also, it is known that certain large carriers have opted to keep a uniform 96-in (2.44-m) dimension on all dollies while otherwise purchasing 102-in (2.59-m) trailers. Accordingly, there is a need for information upon which to evaluate the stability implications of the mixed-width combinations.

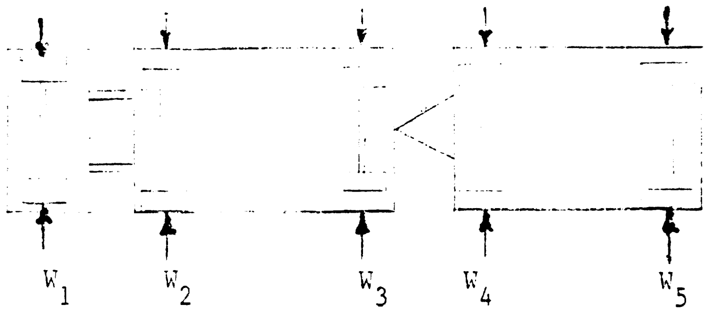
Although all road tractors currently being sold in the U.S. incorporate a 96-in (2.44-m) outside width, there is an expectation that 102-in- (2.59-m-) wide tractors will become available in the future. Thus, some of the calculations include the wider tractor dimension, as well as the current 96-in (2.44-m) value.

The matrix of width dimensions covered in this set of calculations is presented in table 7. A total of six cases are listed, covering the various combinations of widths at the trailer and dolly axles, plus an additional case in which the tractor axle widths are increased by six inches. Note that, while the matrix includes all possible combinations of axle widths on the respective "front" and "rear" units, it does not provide for the complete mixing of widths on the overall vehicle combination as a whole. This matrix of values is selected, recognizing that the pintle hitch coupling between the front and rear units does not transmit significant levels of either lateral force or roll moment--such that each unit essentially rolls independently of the other. Thus, each simulation tending to produce rollover with a double can be looked upon as providing a separate rollover evaluation for the front and rear units, independently.

Three operating conditions have been simulated in order to address what are seen as the key static and dynamic stimuli which tend to roll over vehicles in service. Firstly, UMTRI's static roll model has been employed to examine the classic static roll stability which applies to operation in a steady turn. Next, the so-called "Yaw/Roll" model has been employed to examine the dynamic rollover conditions which prevail during rapid steering maneuvers producing large levels of rearward amplification. In this maneuvering condition, high levels of lateral acceleration are experienced at the second trailer, and a highly transient rollover impetus is developed. This condition is simulated with each of the different cases of width variation on the rear trailer. The severity level of the steering input is increased in subsequent simulation runs until rollover is achieved. A third simulation, employing UMTRI's "Phase IV" comprehensive simulation, has been used to simulate rollover at the roadside. In this case, the vehicle travels onto an inclined plane representing the side slope and simultaneously encounters an abrupt reduction in wheel elevation at the right side, representing tire sinkage in soft soil.

The results of the width study are summarized in tables 8 through 11. For the steady-turning situation (table 8) and the obstacle-avoidance maneuvers (tables 9 and 10), the performance measure is the lateral acceleration of the tractor at rollover. For running on a side slope, the vertical loads on the trailer axles are used to assess the influence of width changes. The study shows the small, but important, improvements that can be made by going from 96-in to 102-in (2.44-m to 2.59-m) widths. However, the most striking comparison is between the rollover thresholds of the A-trains and the B-trains in the

Table 7. Combinations of Axle Widths Simulated.



(Width across the outside of tires at each axle.)

Cases	Dimensions, inches W_1	W_2	W_3	W_4	W_5
1) Conventional 96 inch combination	93	96	96	96	96
2) 102 inch trailer and dolly	93	96	102	102	102
3) 102 inch 1st trailer	93	96	102	96	96
4) 102 inch 1st trailer and dolly	93	96	102	102	96
5) 102 inch 1st and 2nd trailers and dolly	93	96	102	96	102
6) 102 inch Tractor, 1st and 2nd trailer and dolly	99	102	102	102	102

1 in = 0.0254 m

Table 8. Rollover Thresholds of A-Trains
Static Roll Model

Number	Case No. as per Table 7	Wide Axle Positions	Specific Units in Roll	Rollover Thresholds (in g's)
1	Case 1	None	Tractor – Semi	0.39
2			Full Trailer	0.38
3	Case 2	3,4,5	Tractor – Semi	0.41
4			Full Trailer	0.43
5	Case 3	3	Tractor – Semi	0.41
6			Full Trailer	0.4
7	Case 4	3,4	Tractor – Semi	0.41
8			Full Trailer	0.4
9	Case 5	3,5	Tractor – Semi	0.41
10			Full Trailer	0.4
11	Case 6	1,2,3,4,5	Tractor – Semi	0.44
12			Full Trailer	0.43

Table 9. Maximum Lateral Acceleration (in g's) Developed at the Tractor During a 2 radian/second Lane-Change Maneuver

A-Train: Yaw/Roll Model

Number	Case No. as per Table 7	Wide Axle Positions	Maximum Lateral Acceleration (g's)
1	Case 1	None	0.19
2	Case 2	3,4,5	0.2
3	Case 3	3	0.19
4	Case 4	3,4	0.21
5	Case 5	3,5	0.21
6	Case 6	1,2,3,4,5	0.22

Table 10. Maximum Lateral Acceleration (in g's) Developed at the Tractor During a 2 radian/second Lane-Change Maneuver

B-Train: Yaw/Roll Model

Number	Case No. as per Table 7	Wide Axle Positions	Maximum Lateral Acceleration (g's)
1	Case 1	None	0.4
2	Case 2	3,4,5	0.43
3	Case 3	3	0.42
4	Case 4	3,4	0.43
5	Case 5	3,5	0.43
6	Case 6	1,2,3,4,5	0.47

Table 11. Side Slope and Road Sinkage (25 ft (7.62 m) Distance Constant): Sinkage Constrained to the Right Side of the Vehicle

A-Trains with a Forward Velocity of 27.72 mi/h (44.60 km/h)

Number	Case No. as per Table 7	Wide Axle Positions	Smoothed Left Side Vertical Load Axle 4 (@ 9 sec)	Smoothed Right Side Vertical Load Axle 4 (@ 9 sec)	Smoothed Left Side Vertical Load Axle 5 (@ 9 sec)	Smoothed Right Side Vertical Load Axle 5 (@ 9 sec)
1	Case 1	None	235 lb	17672 lb	304 lb	17297 lb
2		1,2,3	841 lb	16885 lb	840 lb	16635 lb
3		1,2,3,4	1175 lb	15800 lb	2100 lb	15200 lb
4		1,2,3,5	1903 lb	15751 lb	1503 lb	16016 lb
5	Case 6	1,2,3,4,5	2436 lb	15073 lb	2234 lb	15134 lb

1 lb = 4.448 N

obstacle-avoidance maneuver. The B-train performs approximately two times as well as the A-train.

As shown in table 8, the 96-in (2.44-m-)-wide vehicle has a rollover threshold of 0.30 g, which corresponds to the level of steady-turning maneuver at which the rear trailer will roll over. By widening all axles to 102 inches (2.59-m), the rollover threshold can be increased to 0.43 g, that is, approximately 13% higher. Analyses of accident data indicate that this would lead to a significant change in the percentage of rollover accidents.(11)

In general, all of the results in tables 8 through 11 indicate that nothing is ever lost by making some combination of axles wider. Widening an axle has two benefits: (1) it improves (reduces) the effective c.g.-height-to-width factor, and (2) it increases the roll stiffness of the axle/suspension set involved. The improvement in c.g. height to effective width is always beneficial. The increased roll stiffness is a benefit if the axle involved is not associated with a "stiff" suspension such that wheels on the inside of that suspension lift off at lateral acceleration levels below the rollover threshold.(12) Nonetheless, widening any axle will improve roll stability.

The results for the A-train (presented in table 9) are much less than the steady-turning results because of rearward amplification between the tractor and the rear trailer. The tractor need only perform a maneuver with a lateral acceleration of approximately 0.2 g in order to obtain enough lateral acceleration at the rear trailer to roll over the rear trailer.

On the other hand, the results for the B-train (table 10) indicate that the maximum lateral accelerations are twice as large as those for the A-train. This is because (1) the B-train has much less rearward amplification than the A-train, and (2) all the units are coupled together in roll such that they can help hold each other up in an obstacle-evasion maneuver. The results show that the increased width of any axle is always a benefit to the roll stability of the B-train.

The "side slope and sinkage" routine was intended to see if some dynamic factor might be of importance. The maneuver did not excite any special transient involvement, however. The results are presented in terms of wheel loads as a convenient way to illustrate the safety margin that could be obtained by widening axles. As shown in table 11, these gains are sizeable for the lightly loaded wheels of the rear trailer of the A-train. Clearly, widening all axles produces the greatest margin of safety.

THE VEHICLE TEST PROGRAM

As noted in a previous section, this project was structured such that the simulation task would constitute the primary source of research findings. The purpose of the test program was to provide a check, or confirmation, of the findings of the simulation study. As indicated, the simulation study identified three commercially available dollies which yield double-trailer vehicles with improved dynamic performance as compared to the performance of such vehicles equipped with a conventional A-dolly. Further, an additional dolly concept was developed which also provides improved performance. Each of these four dolly types, plus a typical A-dolly, were the "subjects" of a testing program involving the so-called "Western double."

The specific dynamic performance measures of greatest interest are "rearward amplification" of the doubles vehicle and the "dynamic roll stability limit" of the last trailer. Rearward amplification, and the resulting low effective roll stability limit of the second trailer, is seen as ***the problem*** of the multitrailer vehicle. Accordingly, the main thrust of this research project was to improve performance in this area. Performance measures of secondary interest are low- and high-speed offtracking, directional stability in braking maneuvers, and structural loading and mobility quality. With some exceptions, the test program pursued the investigation of these performance qualities using the same "tests" as were used in the simulation study.

To create a "Western double," UMTRI used its own two-axle, COE Ford tractor and two short-wheelbase trailers, loaned by the Fruehauf Corporation. The descriptive geometric parameters of interest are given in figure 68. Note that the trailers are each 26 ft (7.9 m) in length, i.e., slightly shorter than the lengths assumed in the simulation study.

Figure 69 is a photograph of the test vehicle. Note that each of the trailers was equipped with outriggers to prevent rollovers during testing. Further, each of the yaw articulation joints was equipped with chains to limit yaw articulation angle and prevent damage due to jackknifing.

Most of the testing was conducted with the trailers in the fully loaded condition. Test "payloads" consisted of the elevated iron blocks, shown in place in figure 70. Loading was such that:

(1) GVW $\approx$ 80,000 lbs (36,281 kg)

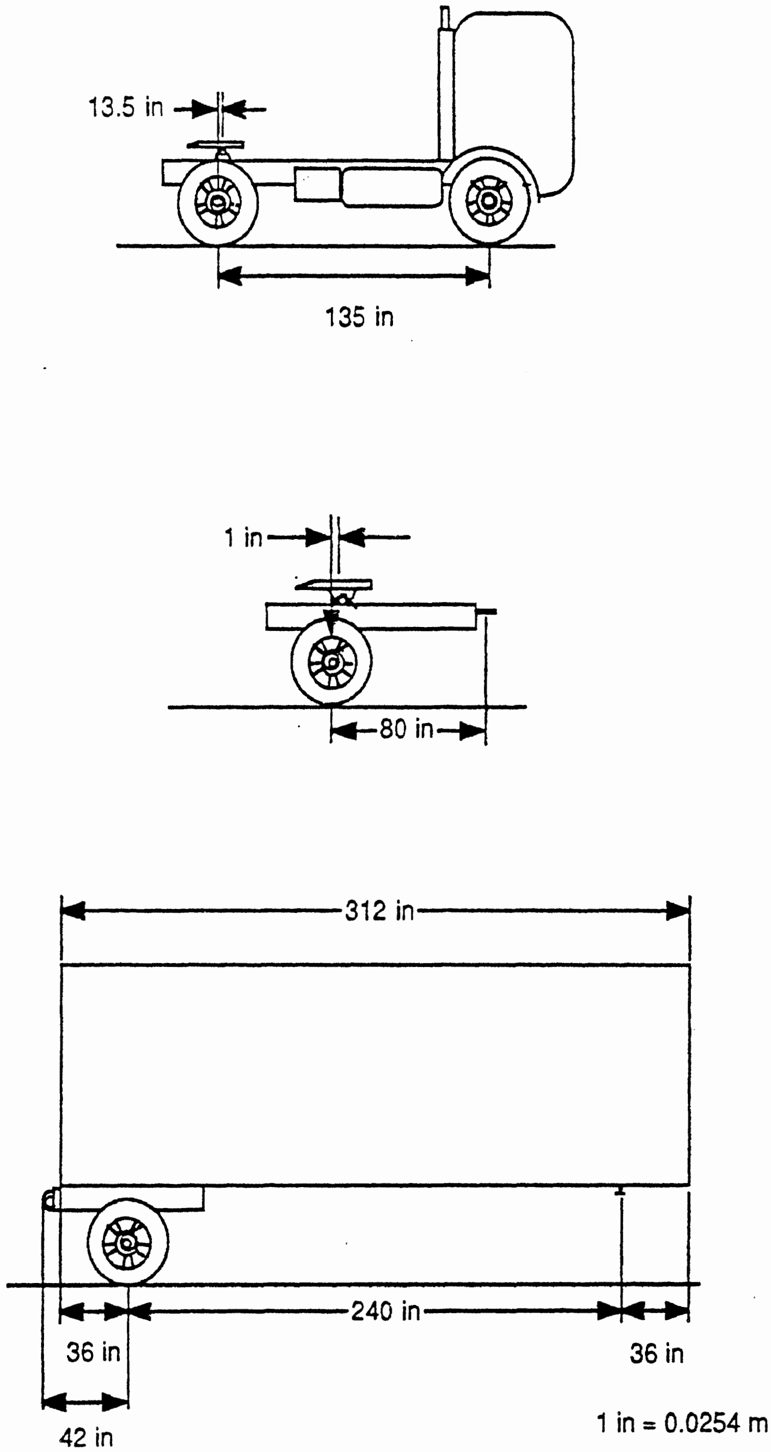


Figure 68. The geometry of the test vehicle.

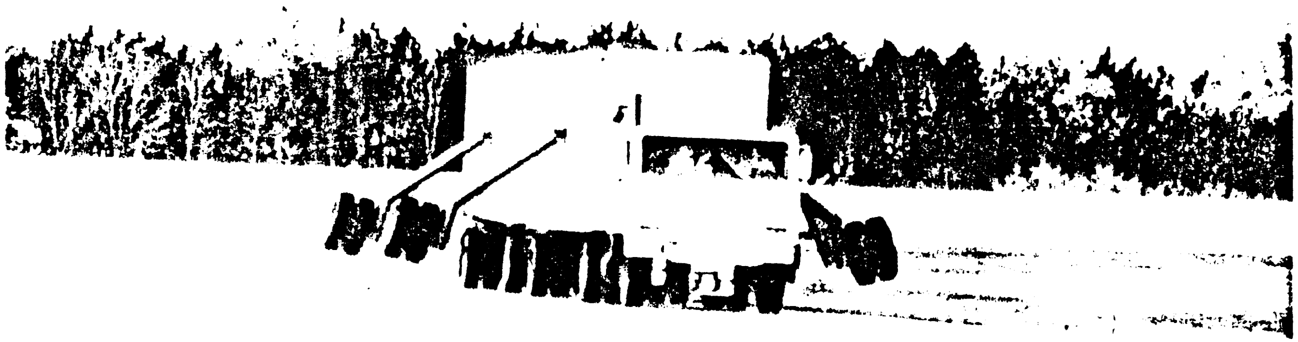


Figure 69. The test vehicle.

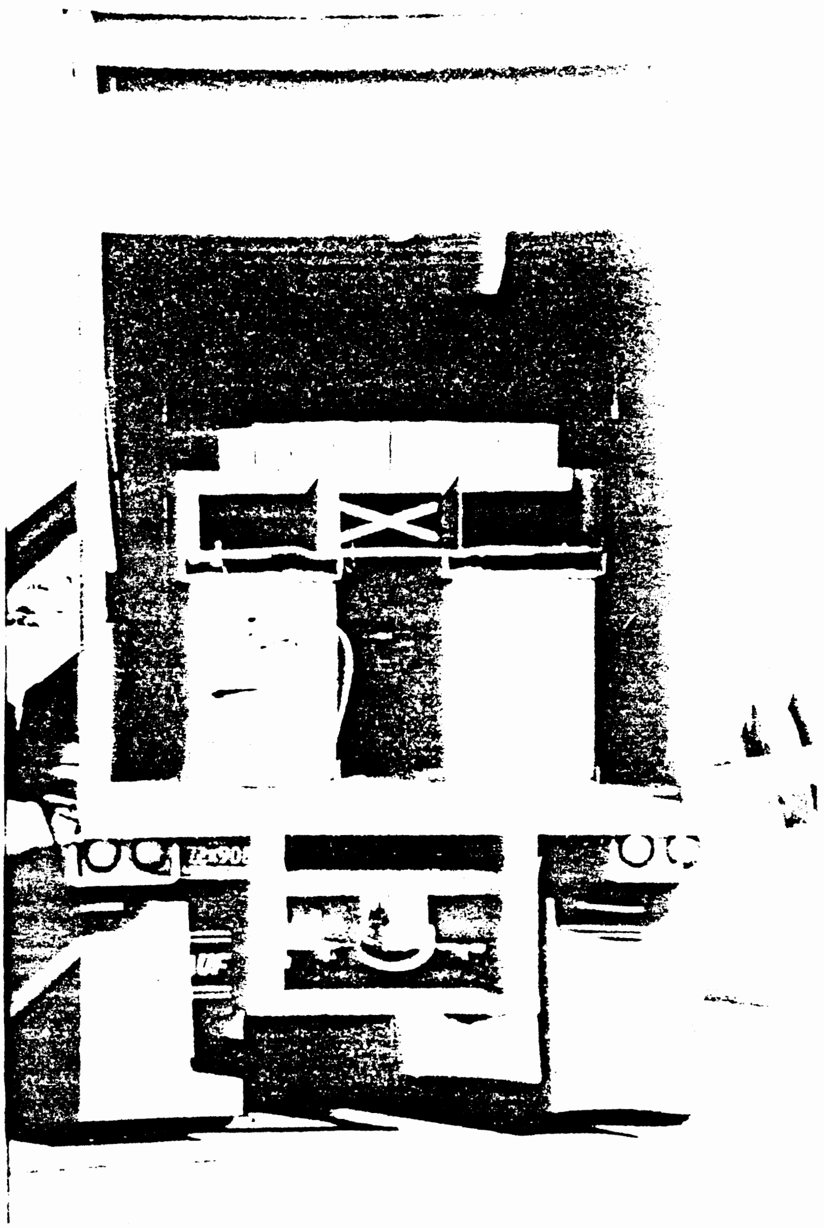


Figure 70. Test vehicle loading.

(2) Tractor front axle load was approximately 10,000 lbs (44,480 N) and all other axle loads were approximately 17,500 lbs (77,840 N)

(3) The composite sprung mass c.g. height of each trailer was approximately 80 in (2 m)

Table 12 gives more specific axle load information.

A number of braking performance tests were conducted with the trailers empty. It should be noted that Michelin 10.00R20 G, steel-belted radial tires were installed on all axles, including the dolly.

UMTRI's 16-channel digital data acquisition system served as the central element of the instrumentation system. Stabilized platform instrumentation packages were used in the tractor and the second trailer to sense vehicle motion. Other instrumentation elements were a fifth-wheel speedometer, steering-wheel displacement transducer, brake pressure transducer, and potentiometers for measuring articulation angles and steer angle of the dolly axle. A new transducer, developed under this project, was used to measure drawbar hitch loads. Pavement markers (water spraying nozzles), mounted on the first and last axles, were used to mark axle paths for measuring offtracking. Figure 71 illustrates the drawbar hitch load transducer. Table 13 summarizes the variables transduced and recorded by the instrumentation system.

The following subsections identify the test dollies and discuss the results of the testing program in the various performance areas of interest.

1. Test Dollies

Tests were conducted with five different types of dolly/hitch hardware, viz.:

(1) The conventional A-dolly (AT)

(2) An asymmetric "4-bar" linkage dolly (TRAP.F and TRAP.R)

(3) A "linked articulation" dolly (LA.8)

(4) A steerable axle B-dolly (SA.60 and SA.0)

(5) The prototype, Controlled Steering B-dolly (CSB.30)

Figures 72 through 76, respectively, are photographs of the test dollies. The symbols enclosed in parentheses will be used to refer to these dollies.

Table 12. Test Vehicle Wheel Loads

Axle No.	Wheel Loads, lb Left Side	Right Side	Total
1	5,140	4,950	10,090
2	8,930	8,740	17,670
3	9,620	9,240	18,860
4	8,640	8,680	17,320
5	9,090	9,280	18,350
Total	41,420	40,890	82,310

*With the trapezoidal dolly. 1 lb = 4.448 N

Table 13. Variables Transduced and Recorded During Testing

TRANSDUCED VARIABLE	DATA CHANNELS RECORDED NON-BRAKING RUNS	BRAKING RUNS
Tractor		
Longitudinal acceleration		1
Lateral acceleration	1	1
Yaw rate	1	1
Forward velocity	1	1
Steering wheel angle	1	1
Brake pressure		1
5th wheel articulation angle	1	1
Dolly		
Pintle articulation or axle steer angle	1	1
Pintle hitch loads (F1 through F6)	6	6
Second Trailer		
Lateral acceleration	1	
Yaw rate	1	1
Roll angle	1	
5th wheel articulation angle	1	1
	16	16

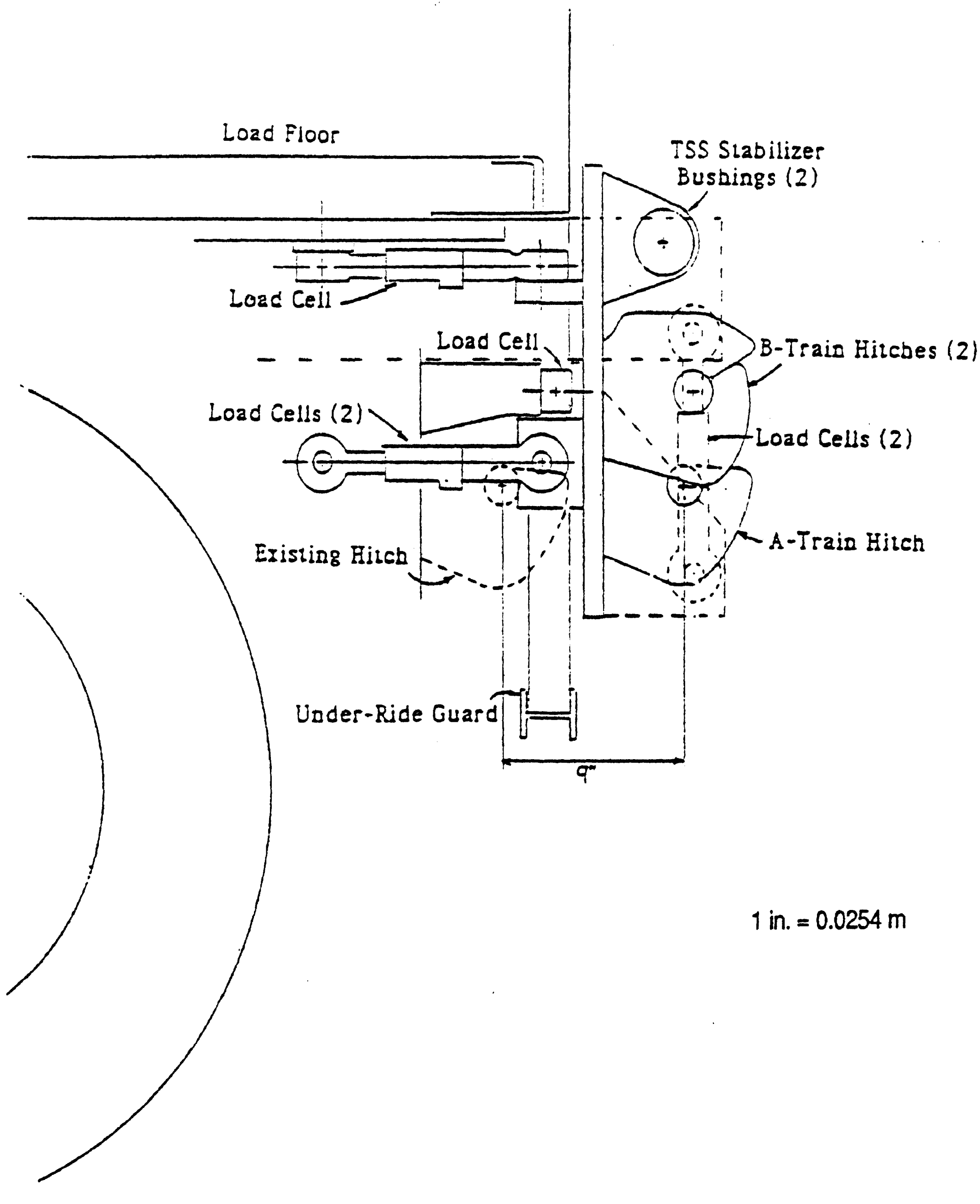


Figure 71. The drawbar hitch load transducer.

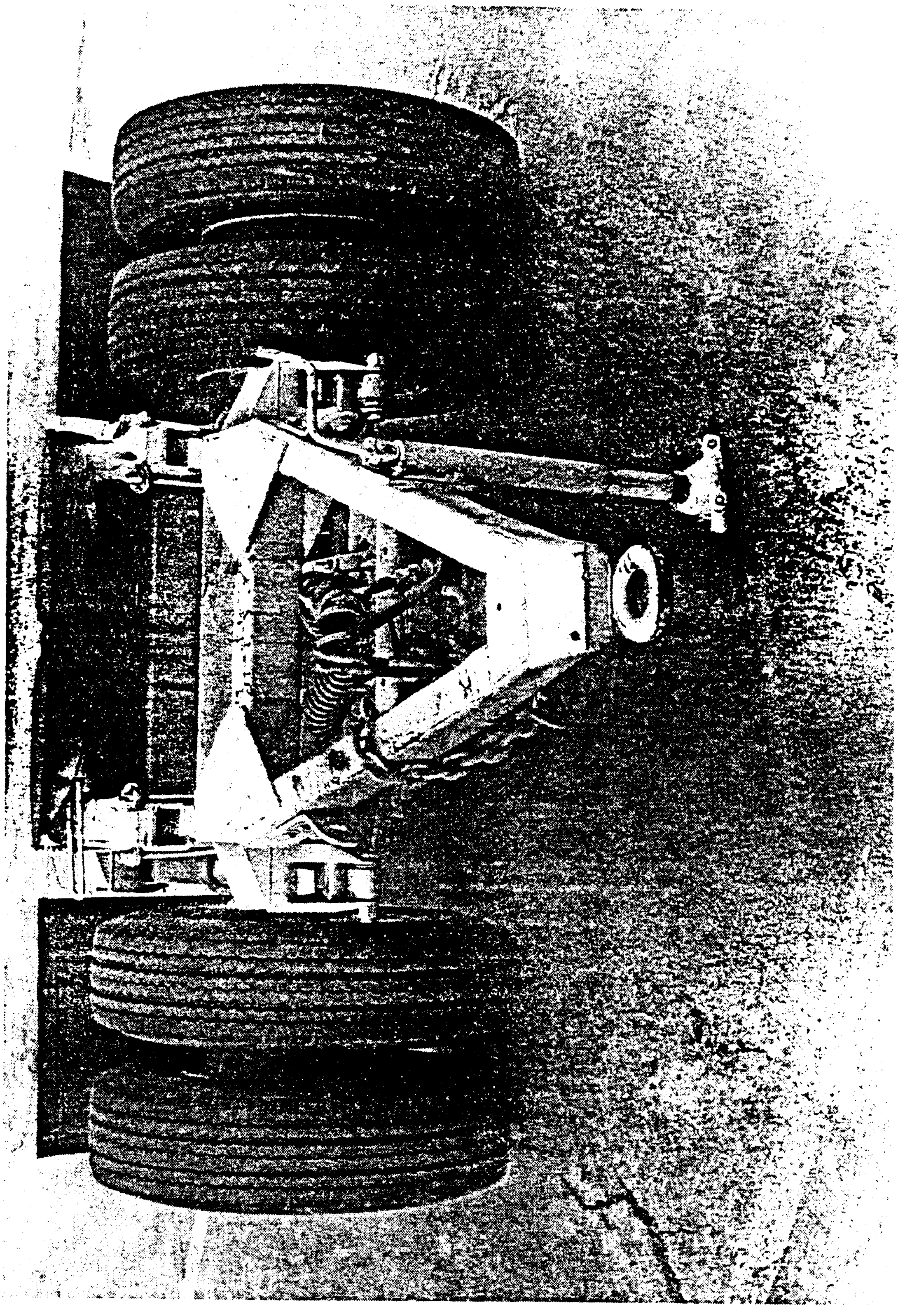


Figure 72. The A-dolly.

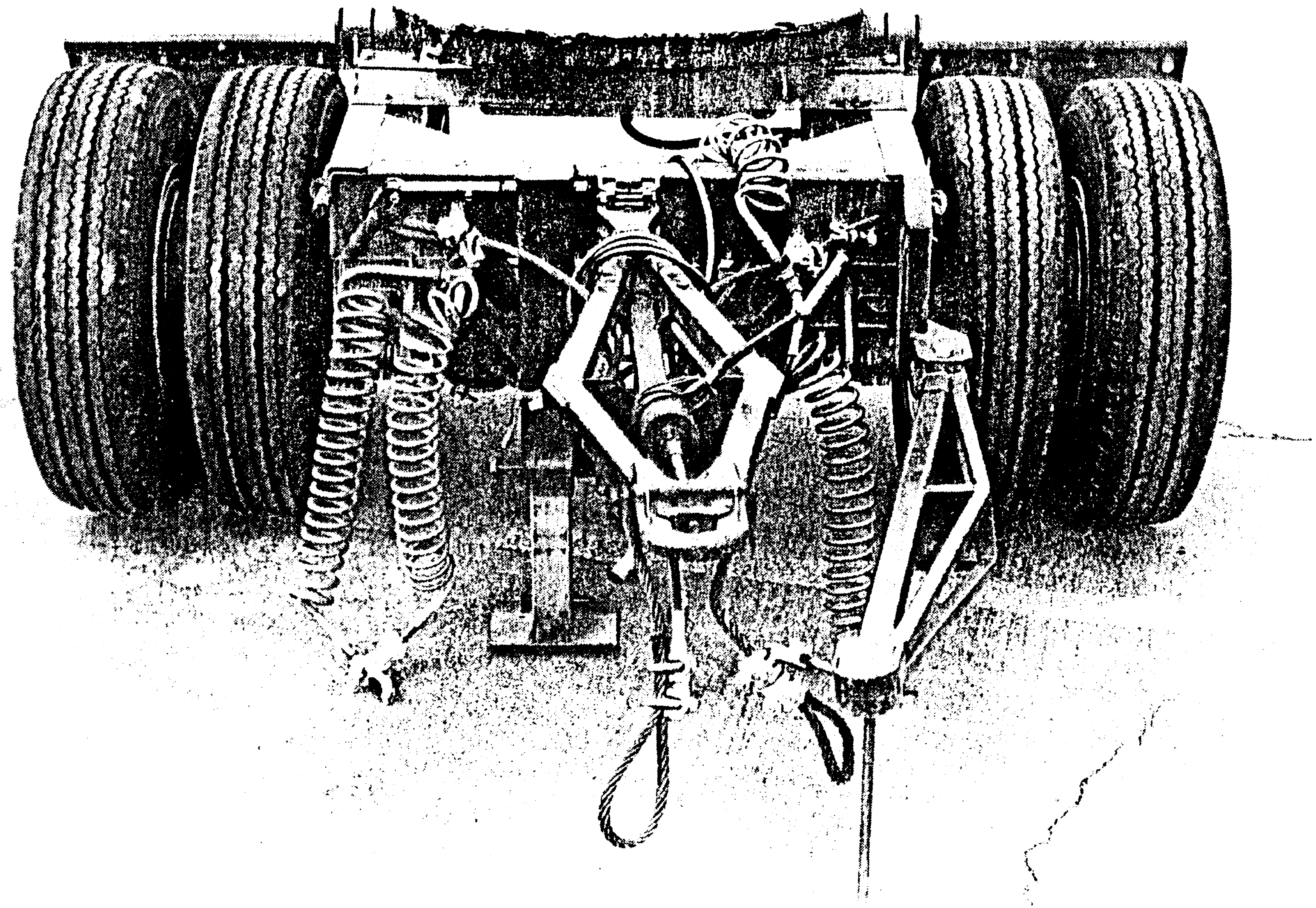


Figure 73. The asymmetric trapezoidal dolly.

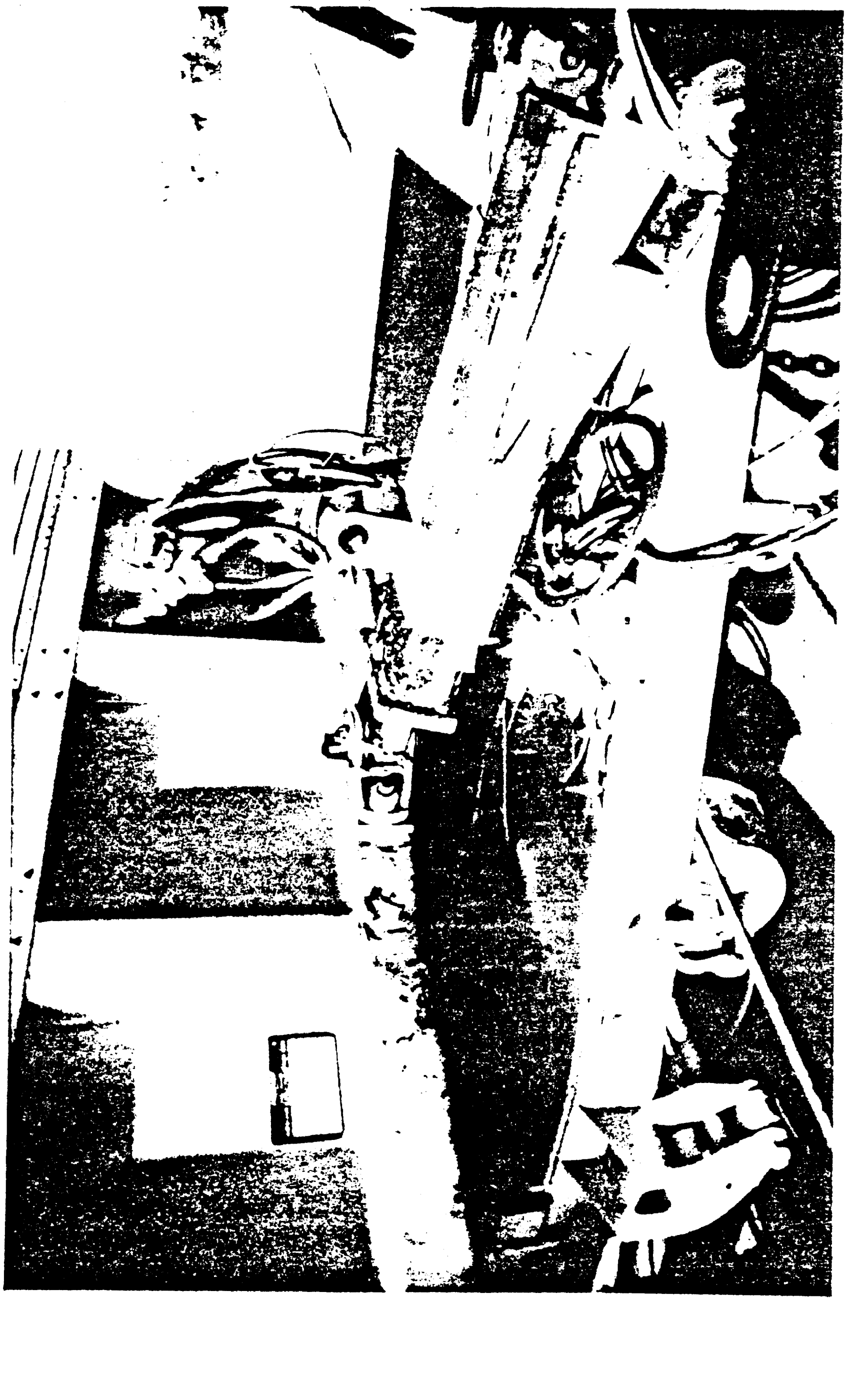


Figure 74. The linked-articulation dolly.

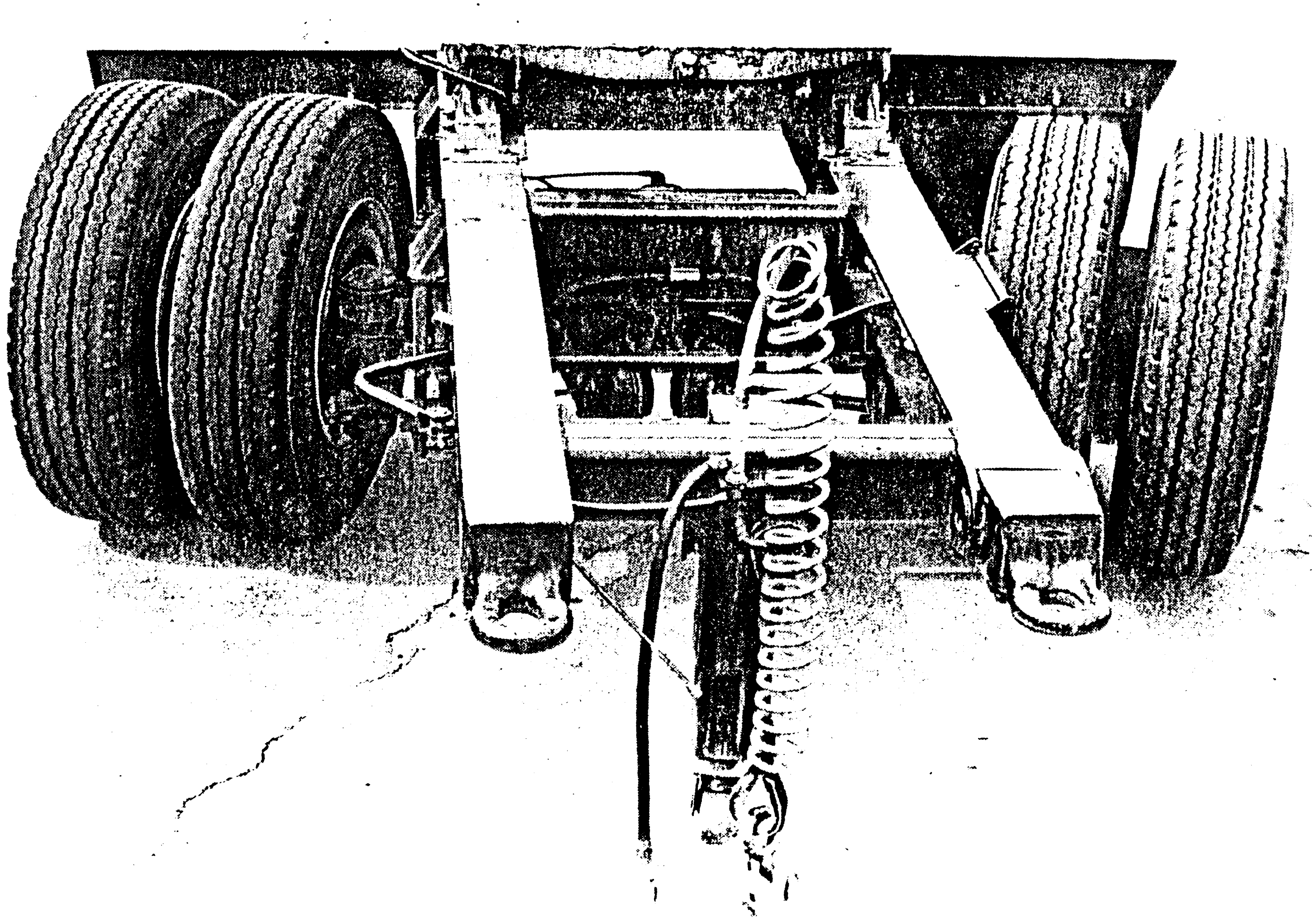


Figure 75. The auto-steering B-dolly.

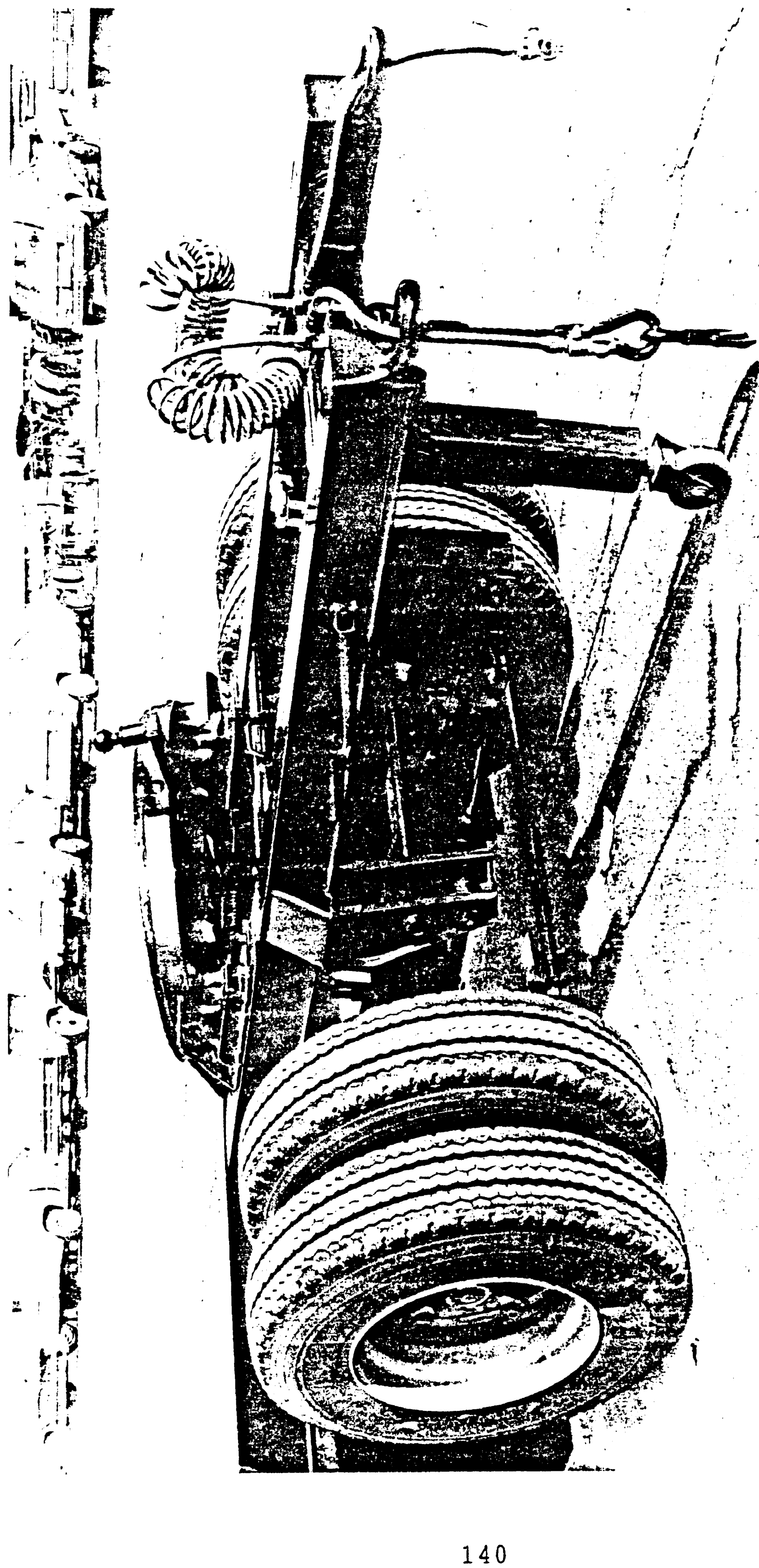


Figure 76. The controlled-steering B-dolly.

The conventional A-dolly needs no introduction.

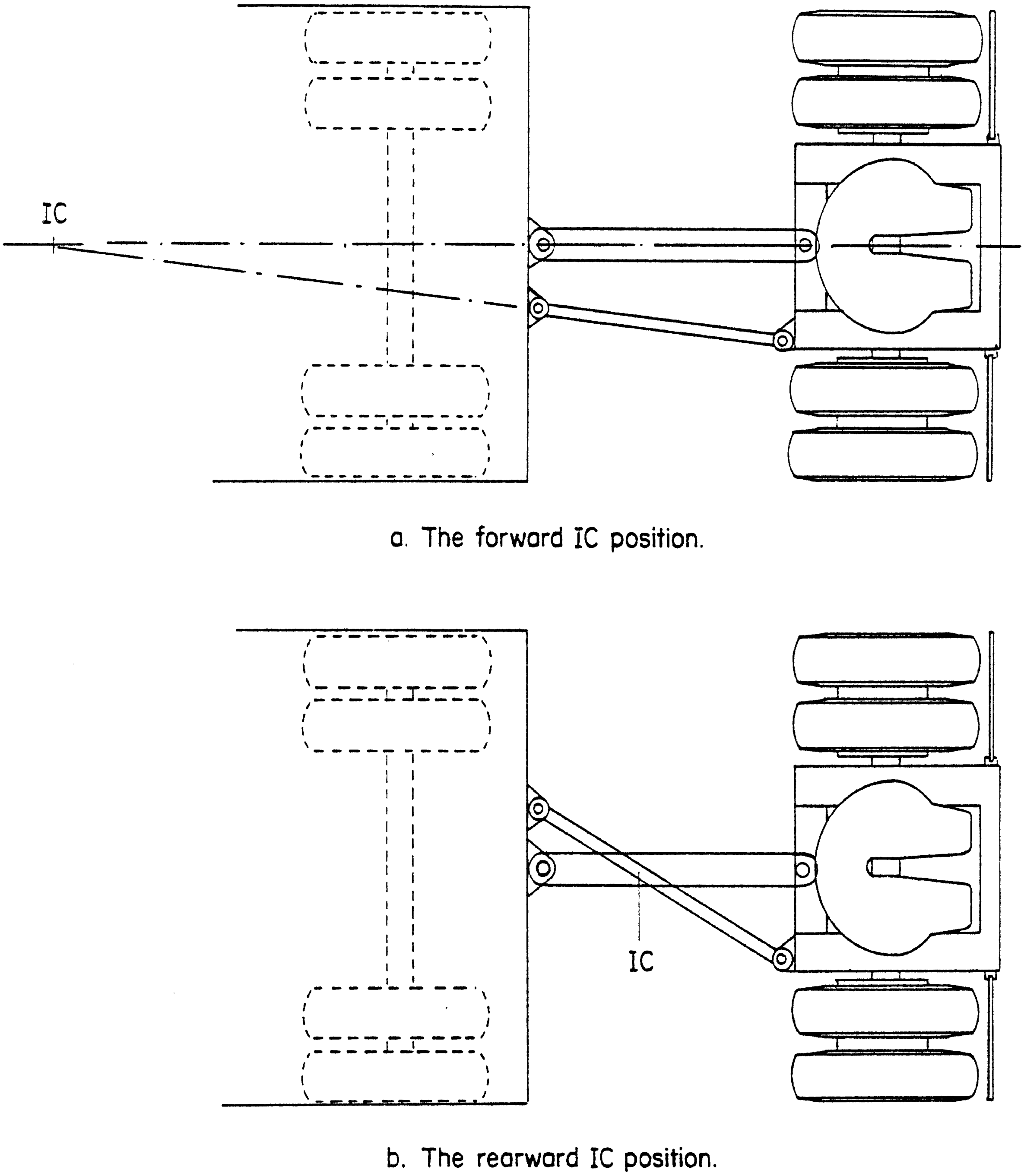
The TRAP-dolly is a prototype dolly provided by the Trapezoid Corporation of Cedar Rapids, Iowa, and is the invention of Mr. N. Gallatin. Although the 4-bar hitch concept is the best known of the non-conventional concepts to be tested, this was the only version identified which was currently intended for the marketplace. The double-drawbar, trapezoidal hitch design is of the asymmetric style. Two hitching configurations were tested, viz., the "forward IC" (TRAP.F) and "rearward IC" (TRAP.R) positions, shown schematically in figure 77. An automatic device for switching the hitch configuration based on speed of the vehicle was not available for the test program.

The LA-dolly hardware tested was an adaptation of commercially available hardware, fabricated by UMTRI. This hardware is patented and has been marketed for use on "Michigan double" tankers by Truck Safety Systems (TSS) of Tecumseh, Michigan. Adaptation to the 80-in (2-m) A-dolly and the van trailers used in this test program provided some difficulty. Although a system articulation gain of about 0.5 was desirable for "Ackerman steering," a gain of about 0.8 (LA.8) was actually used.

The steerable axle B-dolly tested was a product of Auto Steering Trailers, Inc. (ASTL) of Oakville, Ontario. It was of the "auto-steer" type using a "brake chamber" style centering mechanism (figures 18.b, 78 and 79). It is one of several functionally similar products manufactured in Canada. It was tested using air chamber pressures of 60 psi (413,700 Pa), as recommended by ASTL for "loaded" operation (SA.60), and 0 psi. (SA.0). Coulomb friction in the steering kingpin assemblies provided significant steering resistance in this latter condition.

The CSB-dolly was fabricated by UMTRI, based on an ASTL B-dolly frame and a BPW steerable axle. This prototype is a B-dolly, since yaw and roll articulation are eliminated at the drawbar hitch point. Steering of the dolly tires is controlled, however, as a function of dolly fifth wheel articulation angle. The steering linkages which provide this function are illustrated in the drawings and photographs of figures 80 through 83. For this test program, the linkages were arranged to produce an "on center" steering ratio of 0.30 (CSB.30), producing approximate "Ackerman" steering on the test vehicle.

Each of the two B-dollies tested (CSB and SA) were equipped with a special feature to reduce repetitive frame stressing during normal operation. As can be seen in figures 75 and 76, the left pintle eye is attached to the left towbar using a lateral hinge joint which allows vertical motion of the pintle eye. The motion of the hinge allows for free (uncoupled) roll motion between the dolly and the towing trailer. The motion of the hinge is limited so that



a. The forward IC position.

b. The rearward IC position.

Figure 77. The forward and rearward IC positions of the trapezoidal dolly hitch .

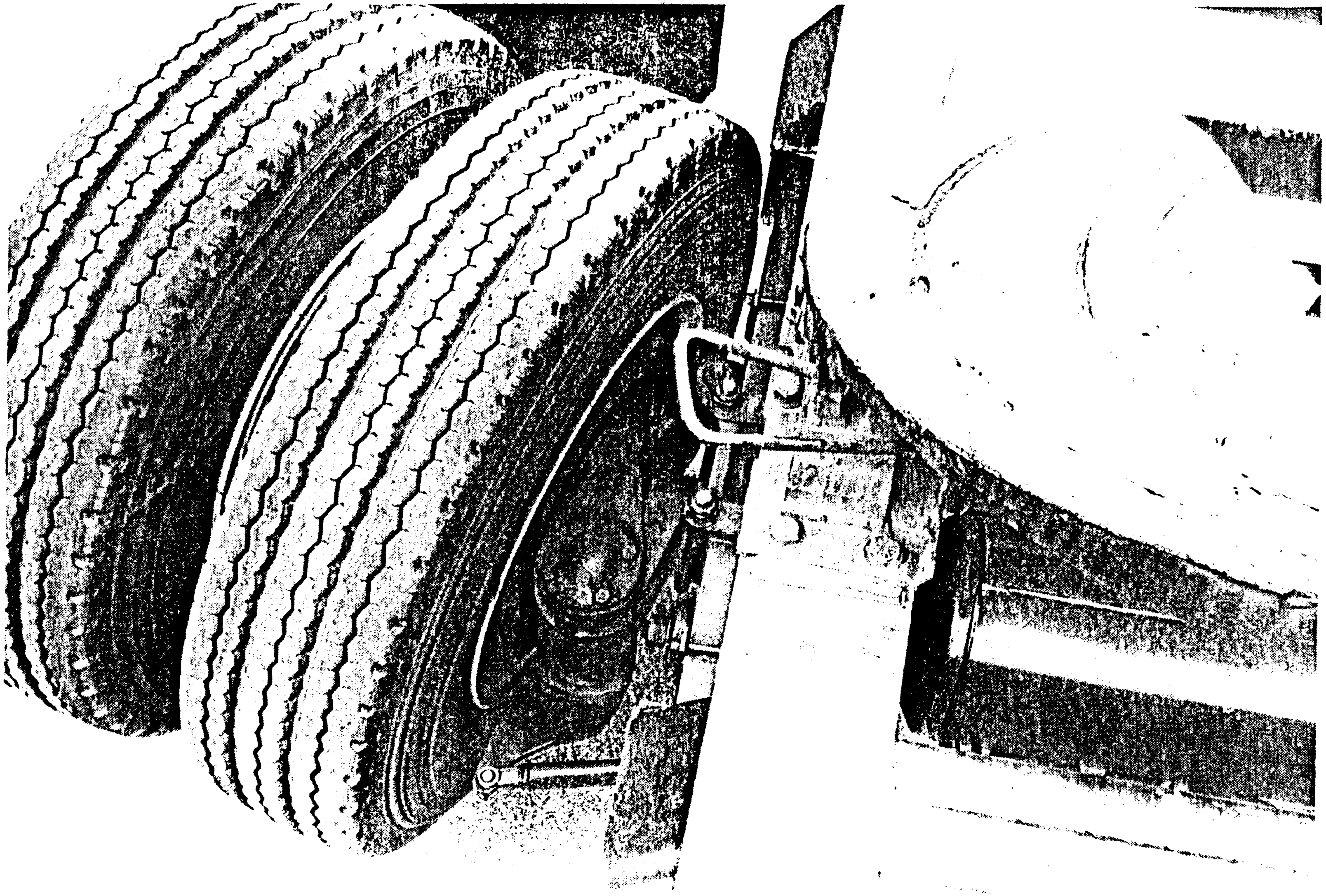


Figure 78. Castered steering system kingpin of the auto-steer-style B-dolly.

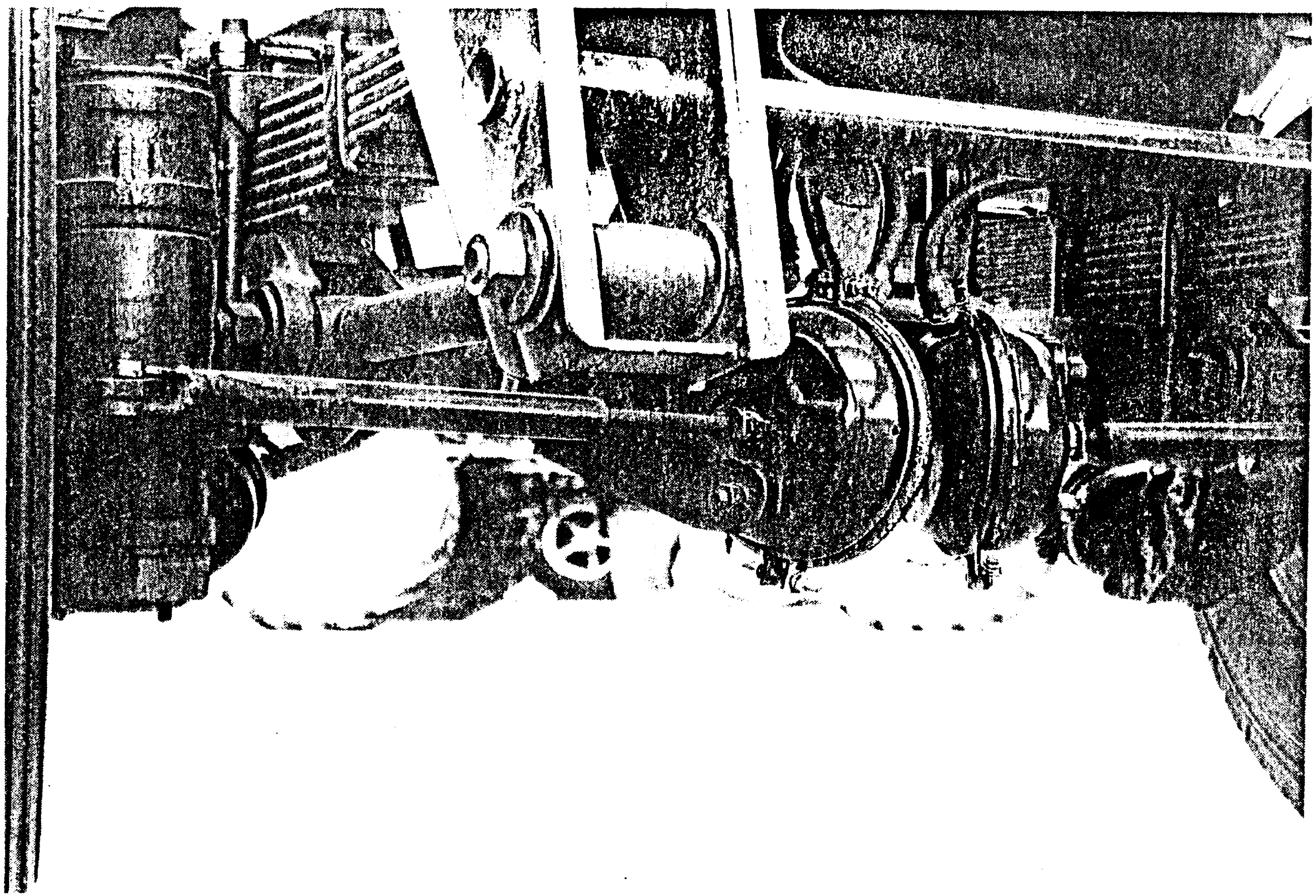


Figure 79. The centering mechanism of the self-steering B-dolly.

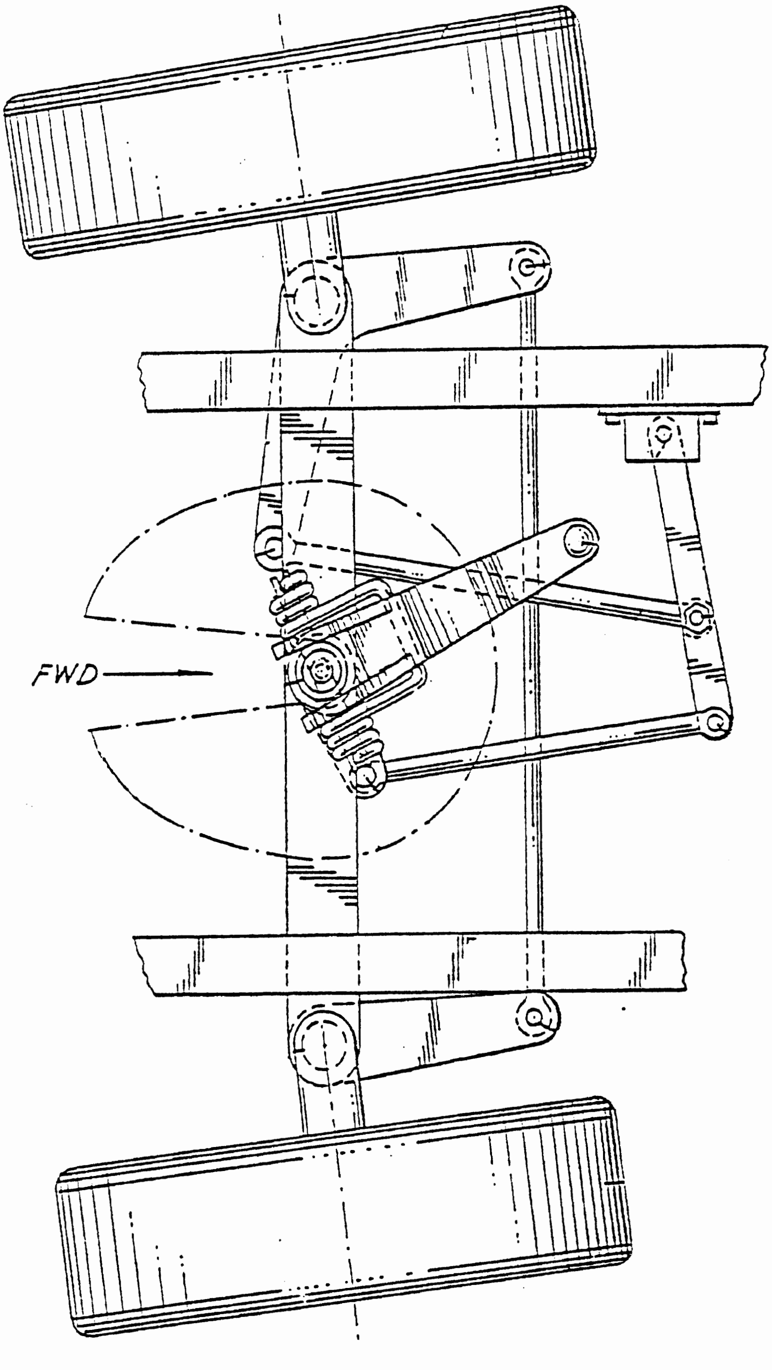


Figure 80. Schematic diagram of the CSB-dolly steering linkage.

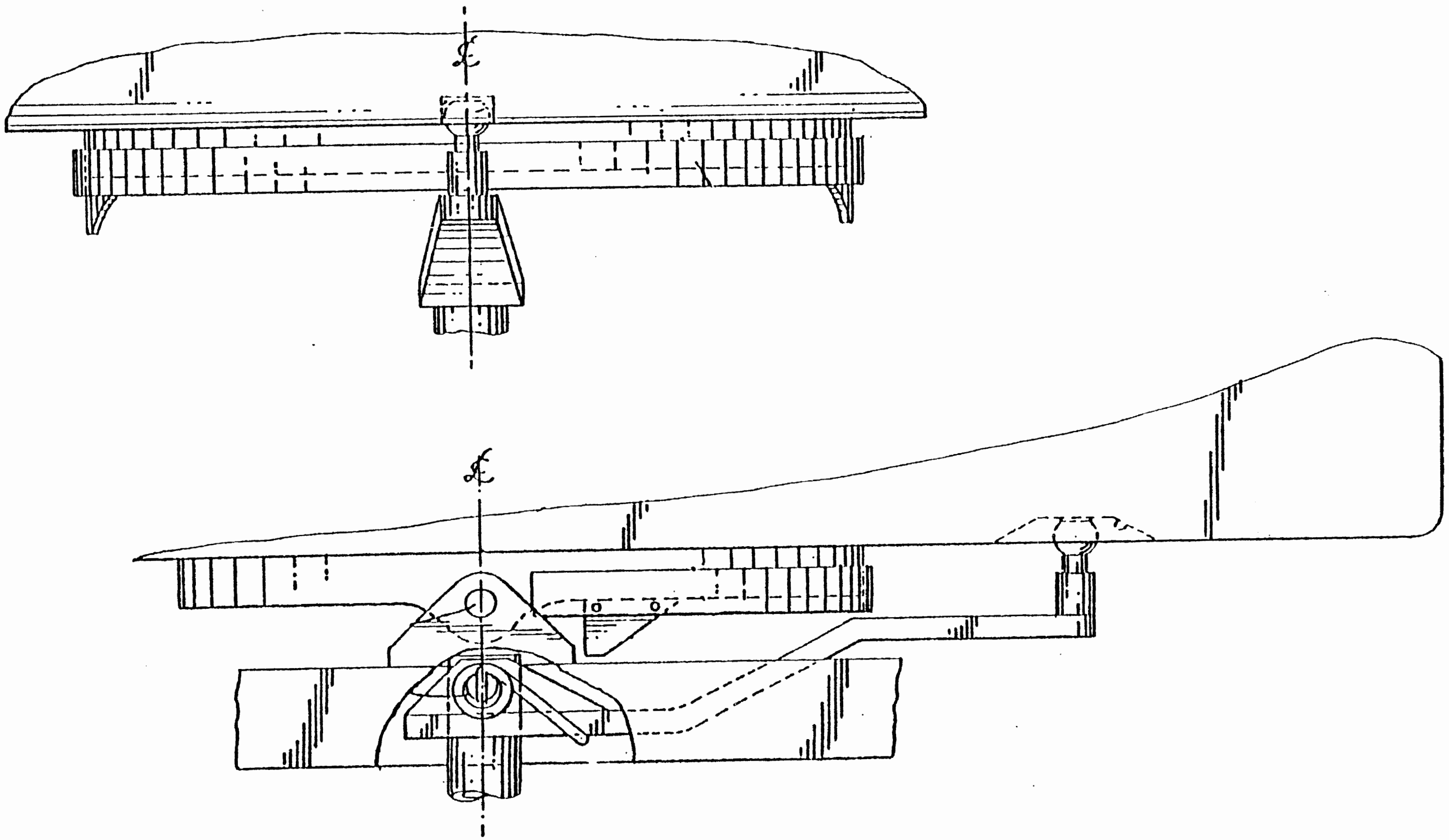


Figure 81. Schematic diagram of the CSB-dolly steering connection with its trailer.

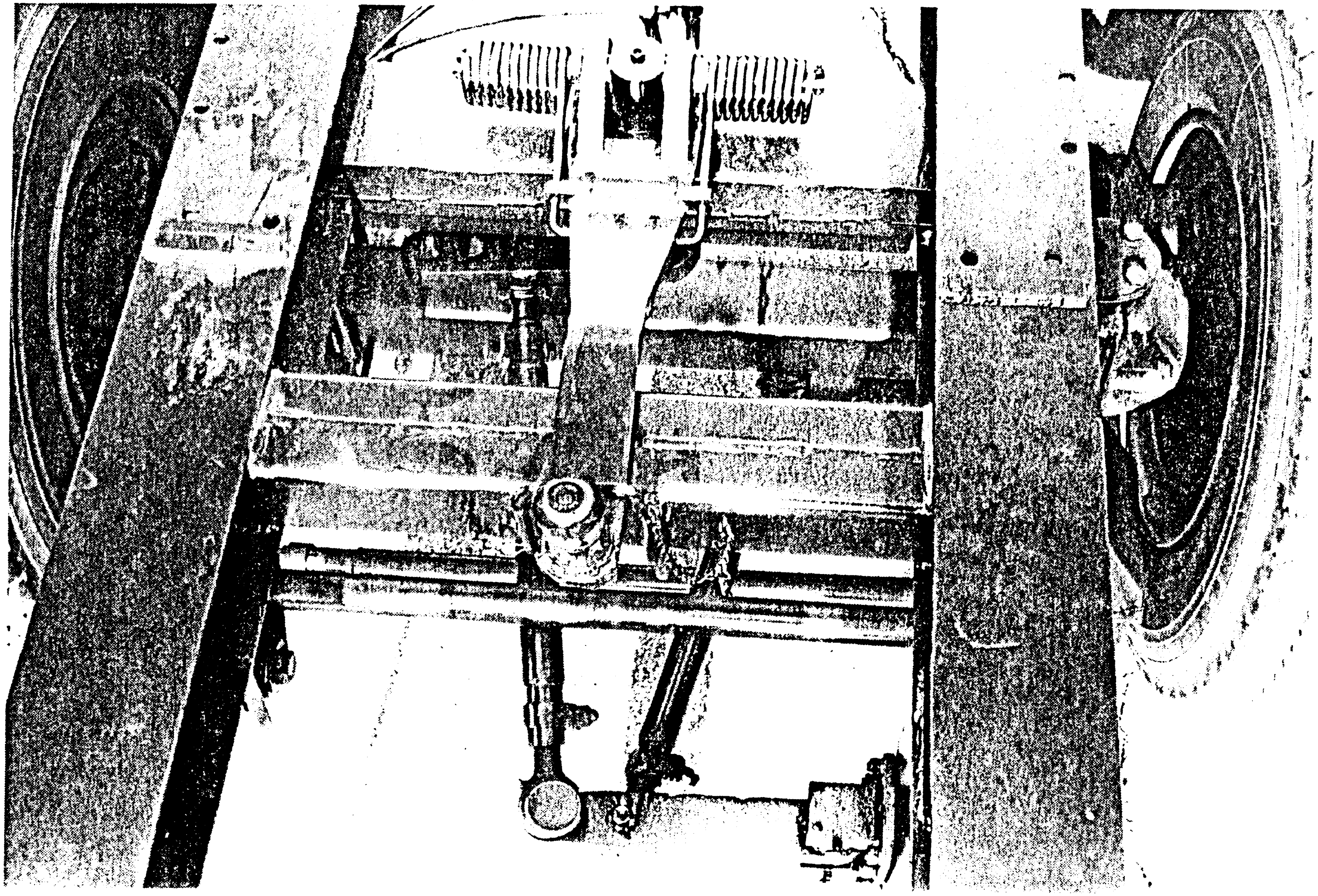


Figure 82. A top view of the CSB-dolly upper steering arm.

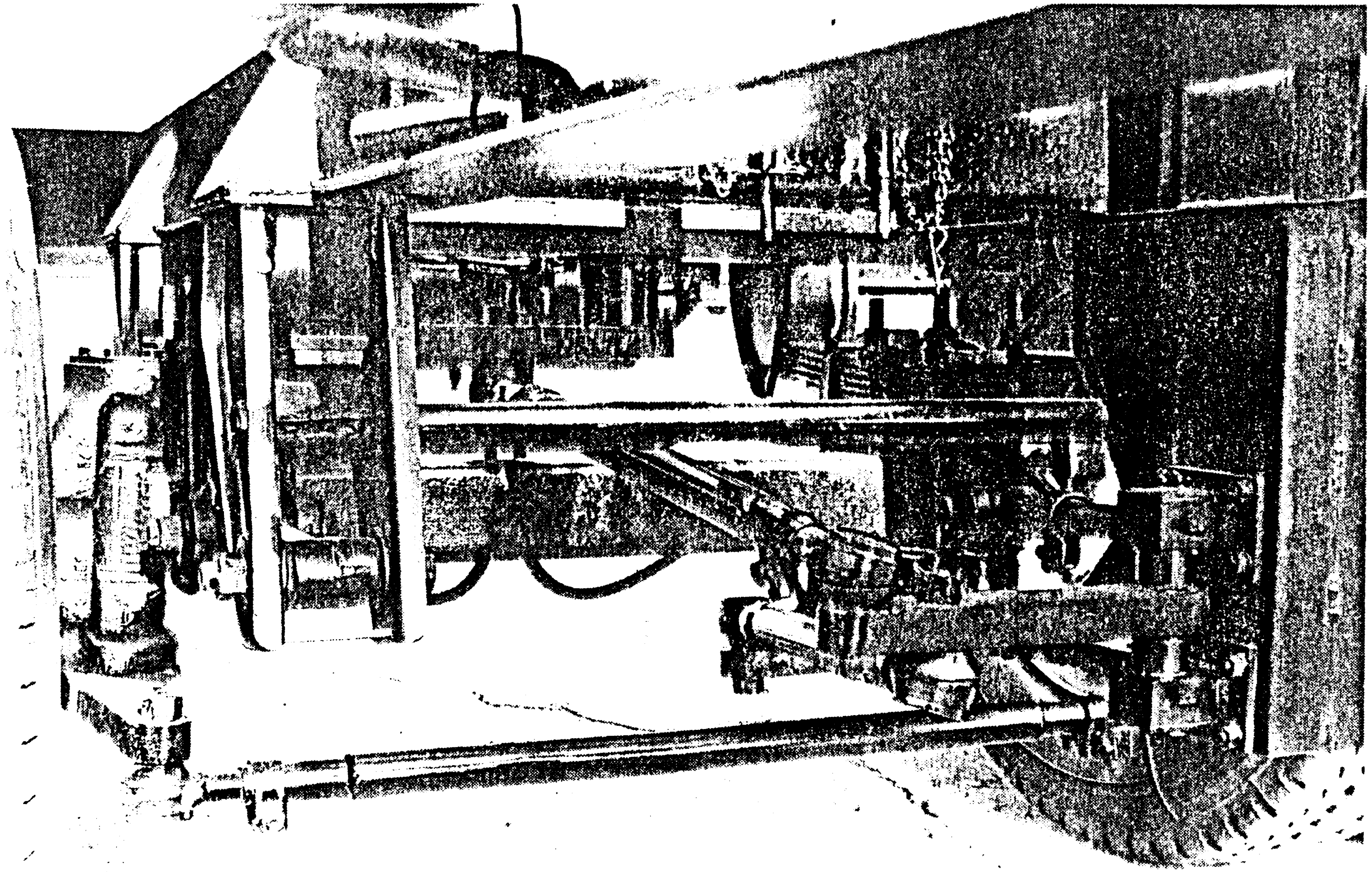


Figure 83. The CSB-dolly steering linkage.

the effective range of this "lash" in roll coupling is approximately ± 2.75 degrees. This roll lash about center reduces the amount of repetitive frame stressing which otherwise occurs during normal running, but retains roll coupling during severe maneuvers. The effectiveness of the roll coupling in promoting roll stability will, of course, be reduced as compared to the simulation results.

All of the test dollies had a drawbar length (pintle to axle) of 80 in (2 m), and a forward fifth wheel offset from the axle of one in. Dolly weights are given in table 14.

2. Test Program Findings

a. Rearward Amplification.

As in the simulation study, low-level (peak tractor) lateral accelerations were used to evaluate the rearward amplification of the test vehicle equipped with the various test dollies. All these tests were run in the fully loaded condition. The test maneuvers used were the well known open-loop, sine-steer maneuvers rather than closed-loop lane changes used in simulation. In these maneuvers, the driver steers the vehicle with a sine-wave-like steering displacement time history in which he deliberately controls frequency (with the aid of a metronome) and steering magnitude (with the aid of steering-wheel stops.) The resulting vehicle path is a lane-change-like maneuver.

Limitations of the vehicle and the test site prevented testing at 55 mi/h (88 km/h). Runs were conducted in the 45 to 48 mi/h (72 to 77 km/h) range. All results have been "corrected" to 55 mi/h (88 km/h) using the expression:

$$RA_{55} = RA_V + (C_1 + C_2 \times Fr) \times (55 - V) \qquad (16)$$

where

RA_{55}	is the rearward amplification at 55 mi/h (88 km/h)
RA_V	is the rearward amplification measure at V mi/h
V	is the test speed in mi/h
Fr	is the frequency of the maneuver in rad/sec
C_1 and $_2C$	are constants, derived from the velocity sensitivities displayed in figure 49. These constants appear in table 15.

Figure 84 shows the results for all the dollies and configurations tested. Figure 85 presents a more comprehensible display of the test data, comparing the results with

Table 14. Test Dolly Weights

Dolly	Weight, lb
A-Dolly	2,520
Linked-Articulation Dolly (A-Dolly Plus Steering Stabilizer)	2,910
Trapezoidal Dolly	2,826
Self-Steering B-Dolly	3,656
CSB-Dolly	3,972

1 lb = 0.454 kg

Table 15. Constants Used in Correcting Rearward Amplification Measures for Variations of Test Velocity from 55 mi/h (88.5 km/h).

$$RA_{55} = RA_V + (C_1 + C_2 \times Fr) * (55 - V)$$

Test Dolly	Simulation Ref Dolly	C_1, h/mi	C_2, sec-h/mi
AT	AT	0.023	0.008
LA.8	LA.44	0.024	0.004
TRAP.F	TRAP.F	0.024	0.004
TRAP.R	TRAP.R	0.023	0.008
SA.60	SA1	0.019	0.0035
SA.0	SA3	0.052	-0.0095
CSB.30	CSB.30	0.028	0.004

1 h/mi = 0.622 h/km
1 sec-h/mi = 0.622 sec-h/km

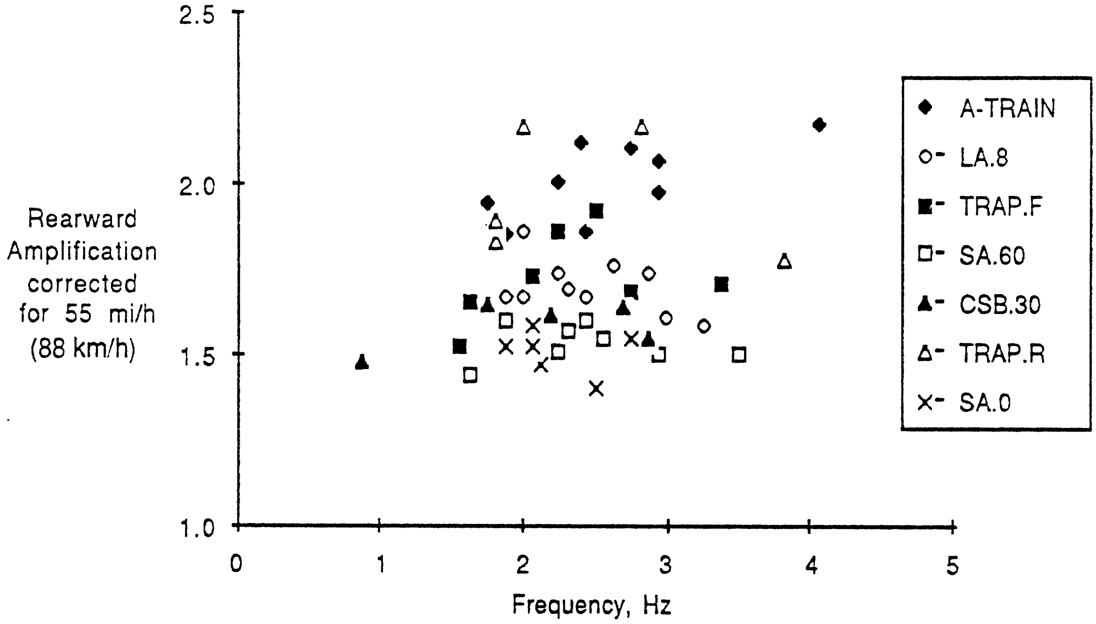
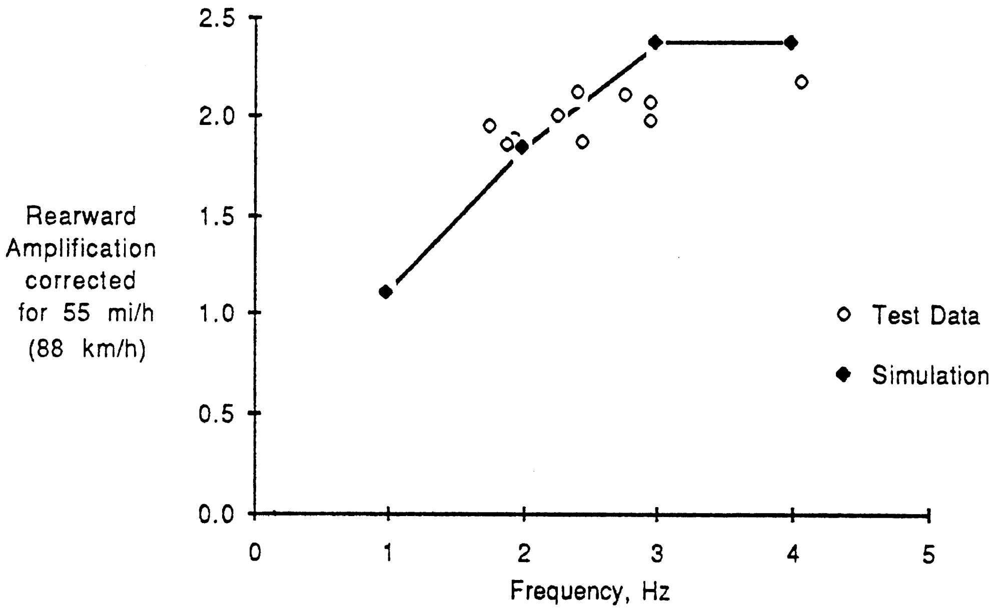
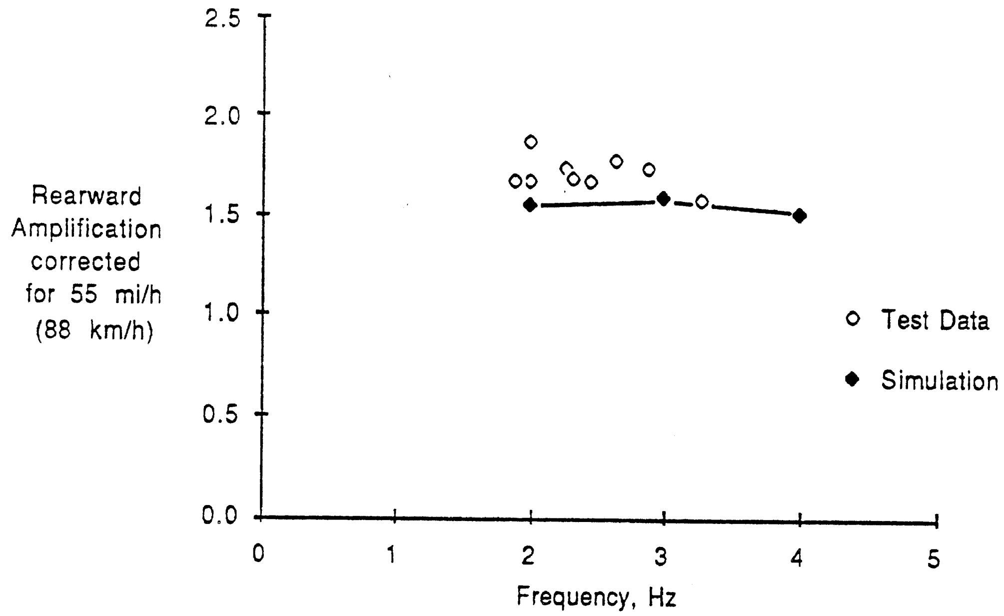


Figure 84. Rearward amplification performance results for all of the test dollies.

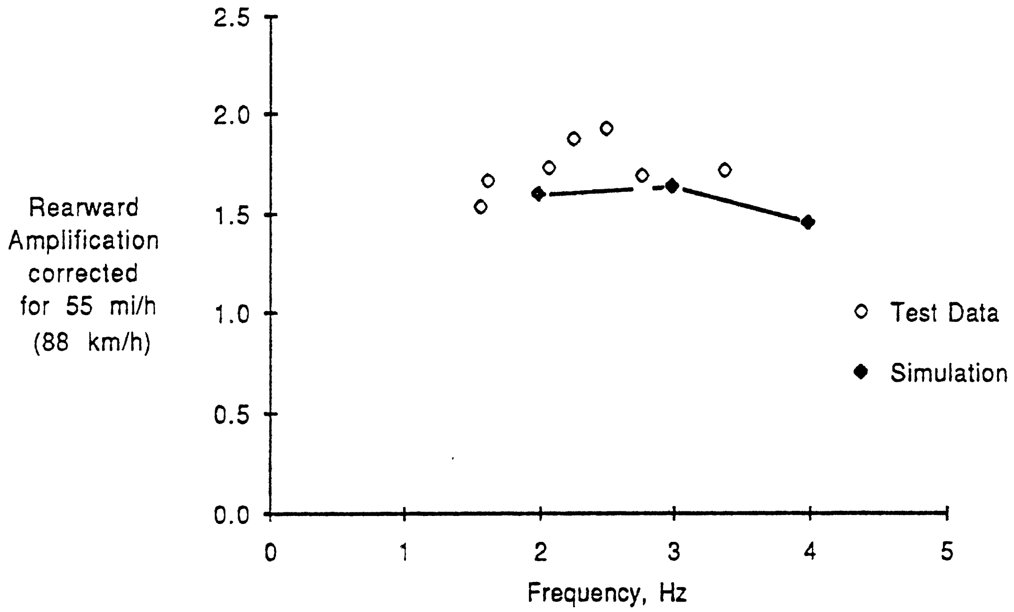


a. A-train.

Figure 85. Comparison of test and simulation rearward amplification performance measures.

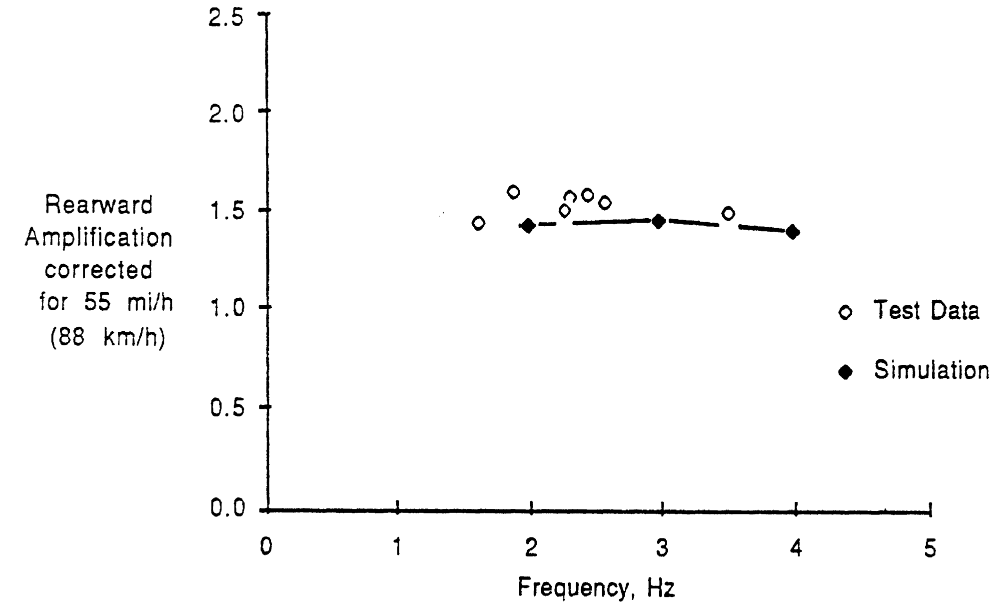


b. Linked articulation dolly.

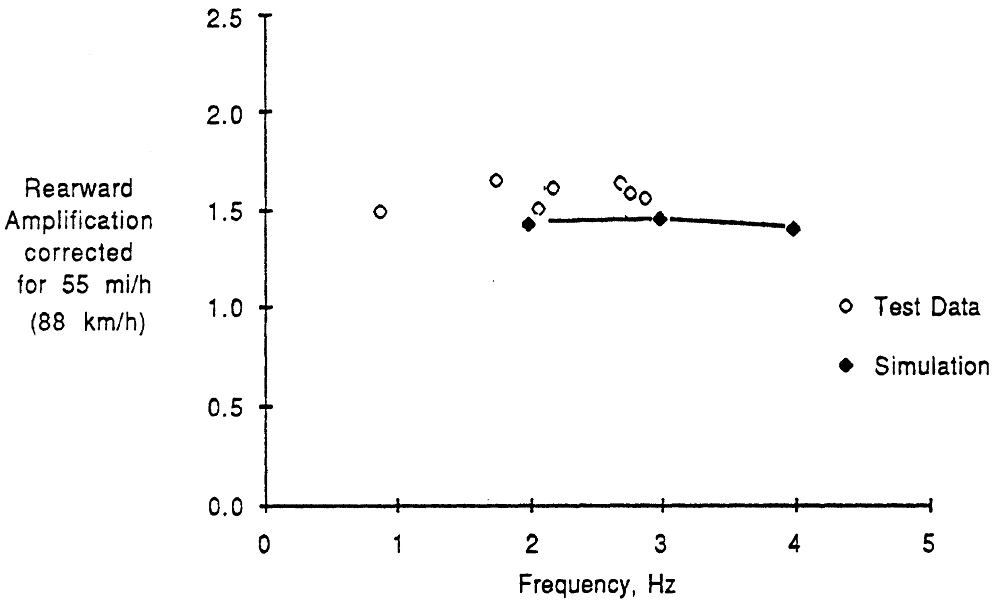


c. Trapezoidal dolly, forward position.

Figure 85. Comparison of test and simulation rearward amplification performance measures.



d. Self-steering B-dolly, full resistance steering.



e. Controlled-steering B-dolly, 0.30 steering system gain.

Figure 85. Comparison of test and simulation rearward amplification performance measures.

simulation results. In general, the testing confirms the simulation findings (recall that many specific parameters are not matched exactly between test and simulation activities). The A-train performs somewhat better in test than simulation, and the inverse is true of the Trap.F and LA dollies. (The 0.8 gain of the test LA dolly accounts for that difference, but no similar explanation is readily available of the Trap.F dolly discrepancy.) The performances of the SA- and CSB-dollies match well with simulation.

b. Dynamic Rollover Threshold.

Sinusoidal steer tests were conducted at higher levels in order to determine the dynamic rollover threshold of the test vehicle with the various dollies. Again all maneuvers were done in the loaded condition. Since the outriggers arrest rollover response, an arbitrary indication of rollover is required. The measures reported, then, are the peak lateral accelerations experienced at the tractor in maneuvers in which the roll angle of the second trailer did not exceed 10 degrees (virtual outrigger touchdown).

Figure 86 shows the measured rollover threshold, as a function of the test dolly, in comparison to the simulation results presented previously. These findings clearly confirm the relative findings of the simulation study. Not surprisingly, the simulations predict higher rollover thresholds for all vehicles than actually occurs. (This result is to be expected since virtually all simplifying assumptions made in simulating the roll response result in several small, destabilizing, compliance effects being ignored.)

c. Yaw Damping.

The effective yaw damping of the test vehicle equipped with the various test dollies (and fully loaded) was examined using a pulse steer maneuver. The maneuver and data reduction methods are similar to those used in the simulation study. Figure 87 shows an example of the time histories of steering-wheel input and lateral acceleration responses. Damping ratios were calculated using the first two large acceleration peaks of the trailer response. Damping ratio results are given in table 16 and example response time histories for each vehicle appear in figure 88.

The low level of repeatability, reflected in table 16, indicates that the fidelity of this measure is not high. In this light, the data can be seen to indicate only the clear increase in damping which generally results from the elimination of a yaw degree of freedom at the dolly. That is, the B-dollies and the LA.8 dolly as a group show higher damping than the A-dolly and trapezoidal dolly. The data do show that this self-steering B-dolly has

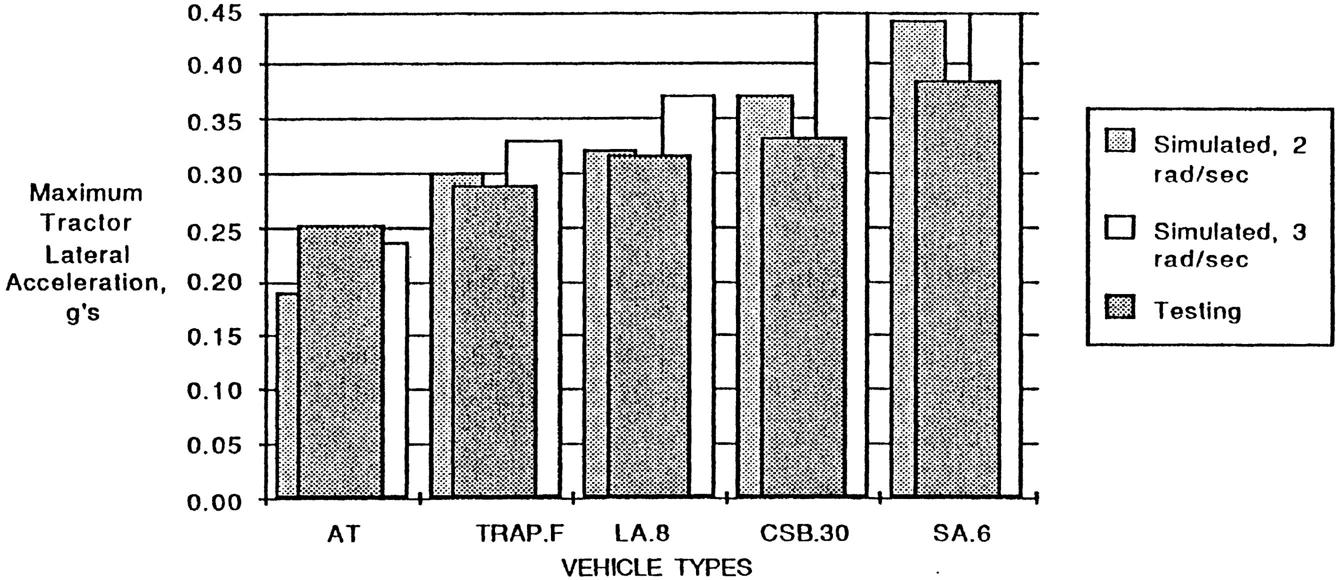


Figure 86. Rollover threshold of the test vehicle equipped with the various test dollies.

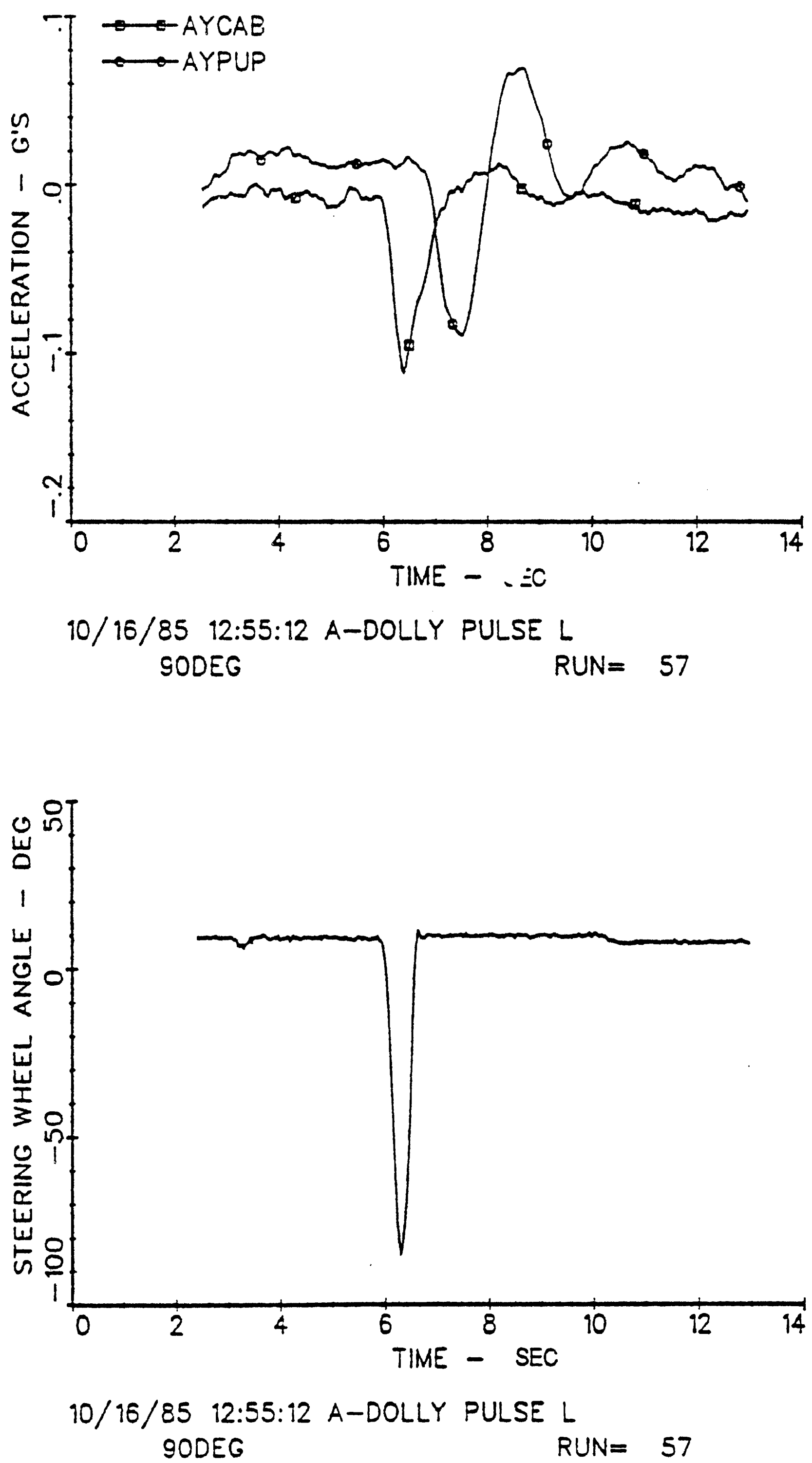
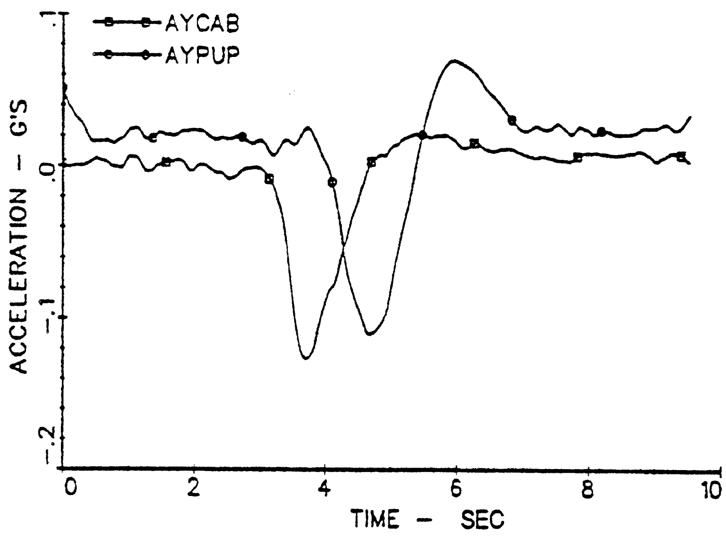


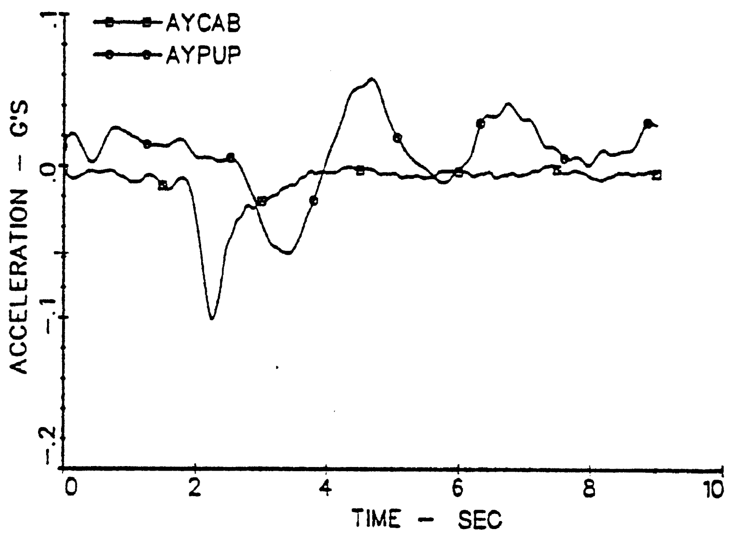
Figure 87. Example pulse-steer time histories of steering and lateral acceleration response.

Table 16. Damping Ratios Measured in the Vehicle Testing Program

Test Dolly Type	Run Number	Damping Ratio
A-Dolly	56	0.206
A-Dolly	57	0.173
A-Dolly	58	0.334
LA.80	92	0.343
LA.80	93	0.256
LA.80	94	0.267
TRAP.R	169	0.134
TRAP.R	170	0.066
TRAP.F	198	0.150
TRAP.F	199	0.230
TRAP.F	200	0.258
SA.60	250	0.665
SA.60	251	0.464
SA.0	264	0.619
SA.0	265	0.392
CSB.30	323	0.536
CSB.30	324	0.435

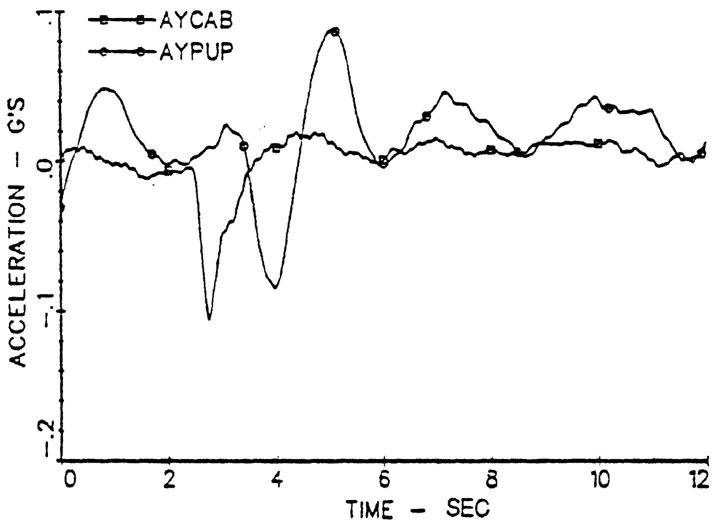


a) The LA.8 dolly

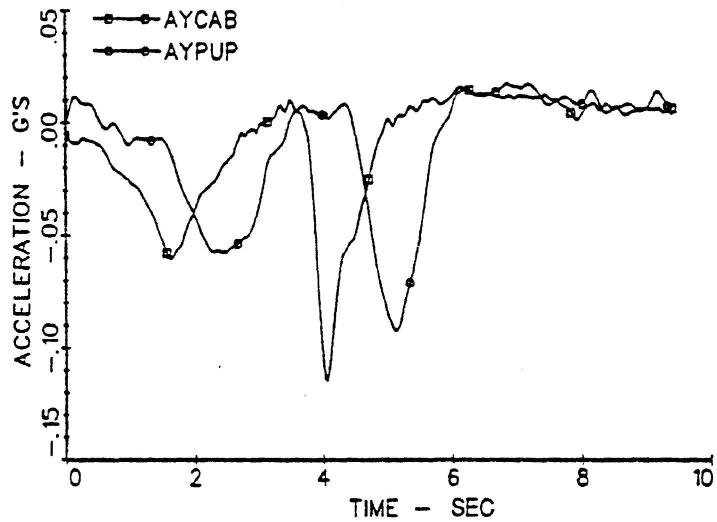


b) The TRAP.F dolly

Figure 88. Example pulse-steer response for each test dolly.

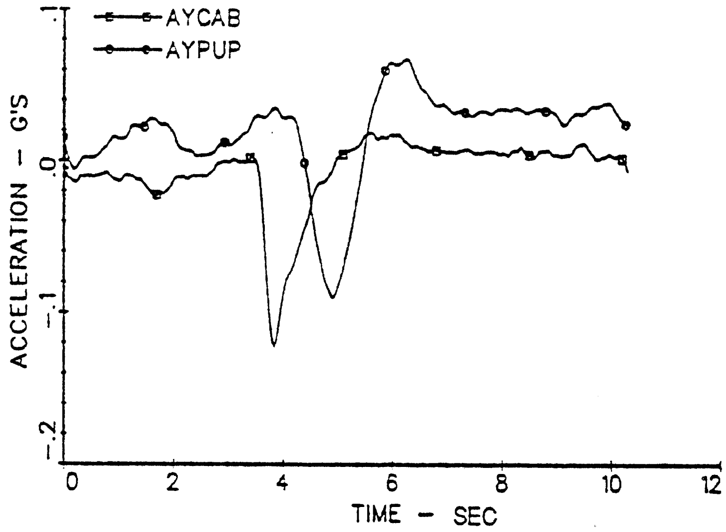


c) The TRAP.R dolly

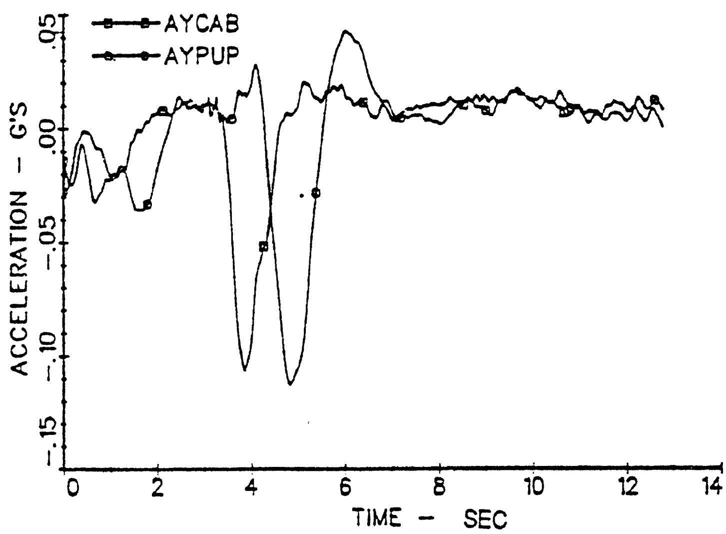


d) The SA.60 dolly

Figure 88. Example pulse-steer response for each test dolly (cont'd.).



e) The SA.0 dolly



f) The CSB.30 dolly

Figure 88. Example pulse-steer response for each test dolly (cont'd.).

sufficient steering system friction to retain well damped responses in this low-level maneuver even with no air pressure supplied to the centering device.

d. Offtracking.

Both low-speed and high-speed offtracking experiments were conducted in the testing program. Low-speed experiments mimicked the 50-ft- (15-m-) radius turning maneuvers of the simulation study. The high-speed experiments measured offtracking in a turn of 1,000-ft (305-m) radius at 45 mi/h (72 km/h) (0.15 g). Both tests used pavement markings (made by water spraying nozzles mounted on the centerline of the first and last axles of the test vehicle) to measure offtracking performance.

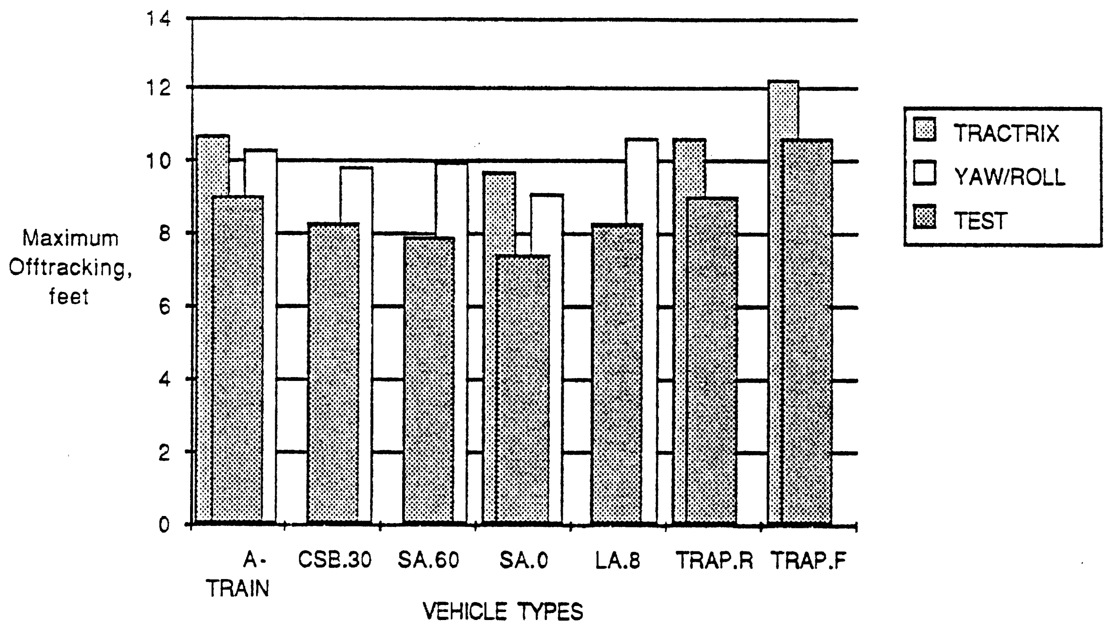
The low-speed offtracking performance measures obtained in the 90-degree and 180-degree turning experiments are shown in figure 89. These data are superimposed on the corresponding simulation study results. The absolute differences between simulation and experiment result largely from the shorter trailers used in the experiment. The relative performance qualities generally hold.

The high-speed experiments have no parallel in the simulation study results. They were undertaken with particular interest in determining whether the unusual LA.8 and CSB.30 configurations would influence these performance properties. In the experiment, offtracking on the 1,000-ft- (304-m-) radius arc was measured at low (5 mi/h (8 km/h)) and high speeds (45 mi/h (72 km/h)). In this way, the high-speed component (the difference between low- and high-speed offtracking) as well as the absolute high-speed offtracking could be determined.

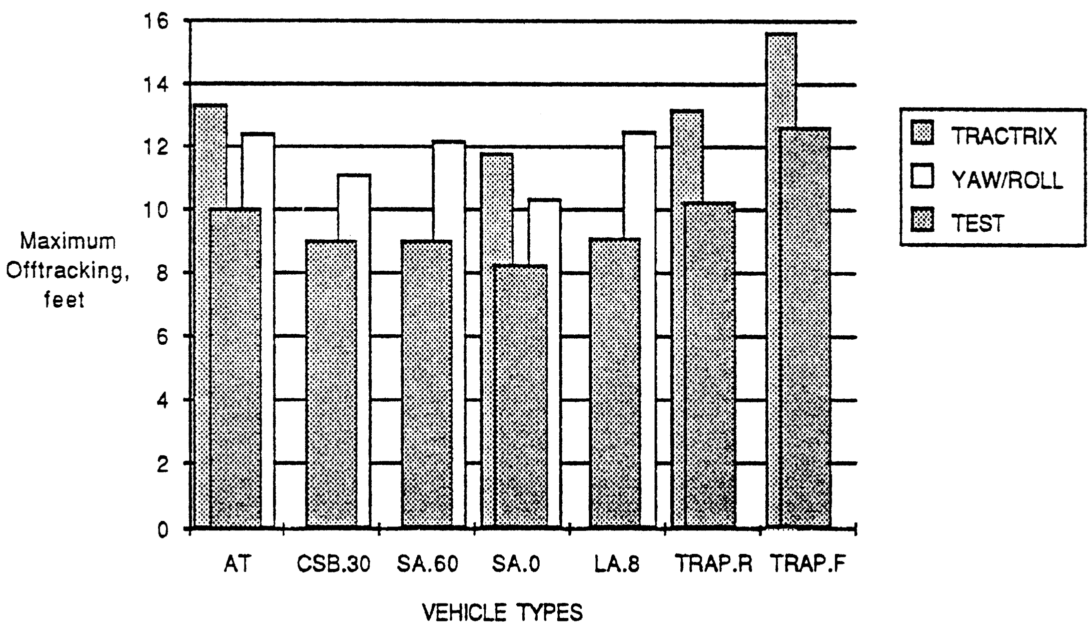
The results of these experiments appear in figure 90. Whereas all of the low-speed offtracking measures presented have been "inboard offtracking," the measures of these two figures are "outboard." The "range" indicated is indicative of the fact that steady tracking of this arc was hard to achieve, and the trailer axle marking, especially, "wanders" somewhat. Given the fidelity of the measure, only the performance of the SA.0 dolly can be distinguished. The low cornering power of the tires on the dolly axle, which results from the relatively free steering of the axle, yields larger outboard offtracking at speed.

e. Stability in Braking.

Braking-in-a-turn tests on a dry surface and straight-line braking tests on a split-friction wet surface were performed to investigate directional stability during braking. Both tests were conducted in both empty and loaded conditions.



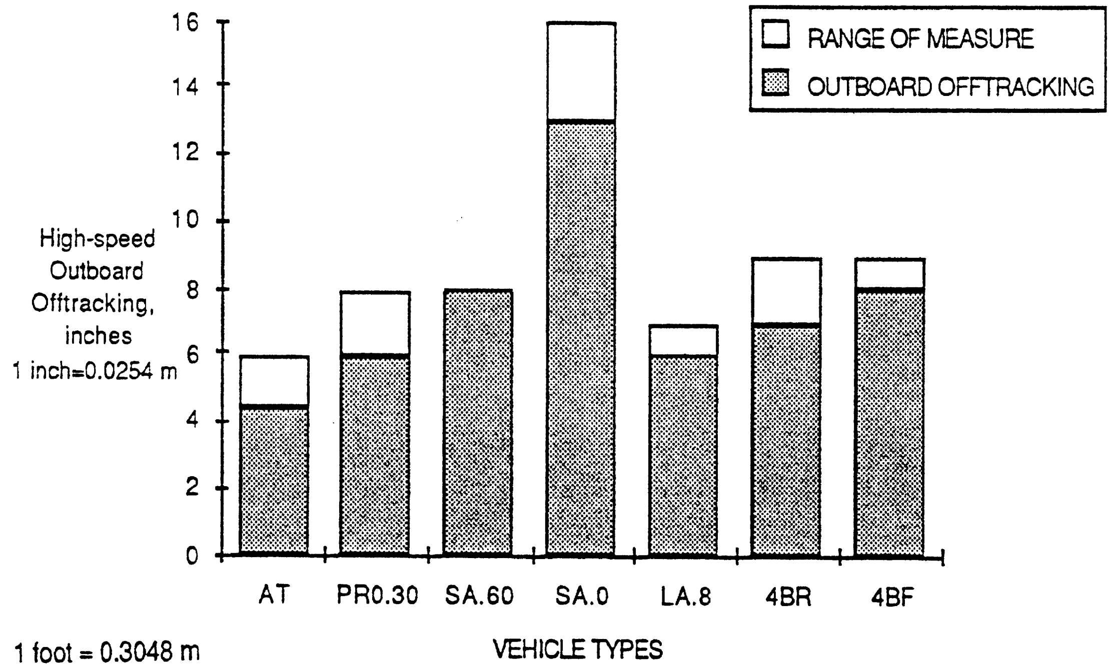
a. 50-foot-radius, 90 degree arc.



b. 50-foot-radius, 180 degree arc.

1 foot = 0.3048 m

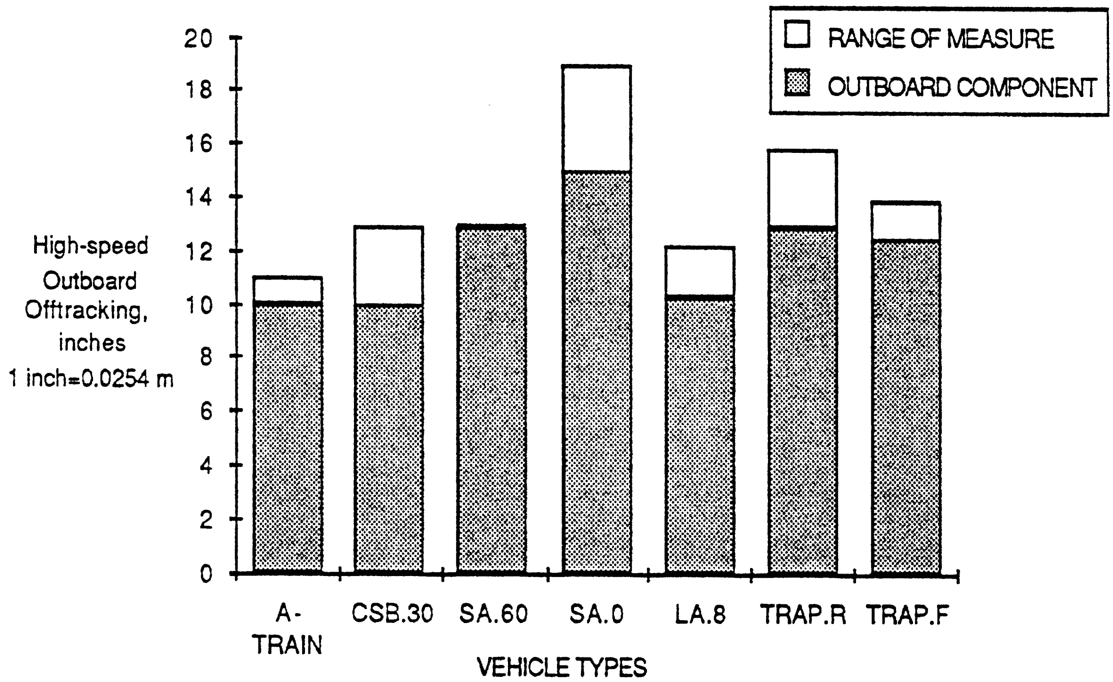
Figure 89. Low-speed offtracking in a 50-foot-radius turn.



1 foot = 0.3048 m

a. Absolute outboard offtracking

1 mi/h = 1.609 km/h



b. High-speed outboard offtracking component.

Figure 90. High-speed offtracking in a 1000-foot radius turn at 45 mi/h.

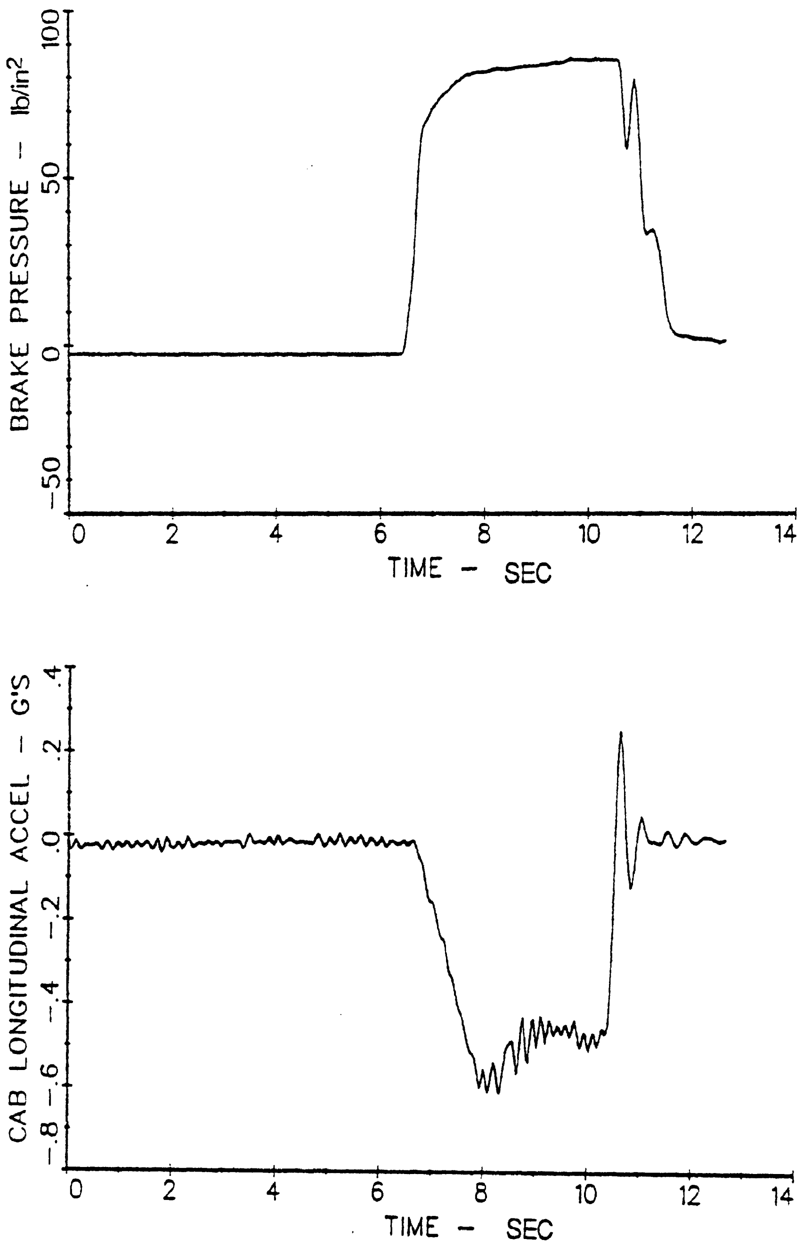
Braking-in-a-turn tests . Braking-in-a-turn tests were conducted on a 1000-ft (304-m) radius arc laid out on a high friction vehicle dynamics test pad. This maneuver was initiated at a velocity of 45 mi/h (72 km/h), corresponding to a 0.135 g left turn. Test runs were performed at increasing levels of brake line pressure. The driver provided a "best effort" at steering to maintain the prescribed path.

As long as none of the heavily loaded wheels lock up, the driver has no difficulty controlling the vehicle in any of these tests. However, control difficulties occur as more wheels lock up. Figure 91 shows the brake pressure and longitudinal deceleration (negative acceleration) obtained in a test in which all wheels except those on the front axle locked. In this case the vehicle was empty. The longitudinal acceleration trace shows that the longitudinal forces on the vehicle build up in an approximately linear manner until a maximum deceleration is obtained. As wheels lock up, the maximum force from the tires decreases and consequently the deceleration decreases to a level that is noticeably less than the peak deceleration. When the vehicle comes to a stop, the deceleration drops to zero immediately. Since the deceleration and pressure waveforms are similar for all tests, only the times when deceleration started, the maximum occurred, and the vehicle stopped are indicated on subsequent graphs of steering angles and articulation angles.

The left-most vertical dashed line on figure 92 indicates when braking deceleration began. Before this time, the driver has applied slightly less than 100 degrees of steering wheel angle to get into the turn. The vehicle has taken on the articulation angles shown at 6.5 seconds in figure 92. These angles are denoted by AA1 for the articulation angle at the tractor fifth wheel, AA3 for the articulation angle at the pintle hitch, and AA2 for the articulation angle at the dolly fifth wheel. These articulation angles are all small before braking is applied.

By the time the maximum braking deceleration occurs (as indicated by the middle dashed vertical line at 8 seconds), the tractor has begun to jackknife, as shown by the divergent nature of the curve labelled AA1. The driver responds quickly and steers drastically in an attempt to avoid the impending jackknife. However, his efforts are of little avail. The steering trace and AA1 saturate at the maximum levels allowed in the recording system. The vehicle is completely out of control.

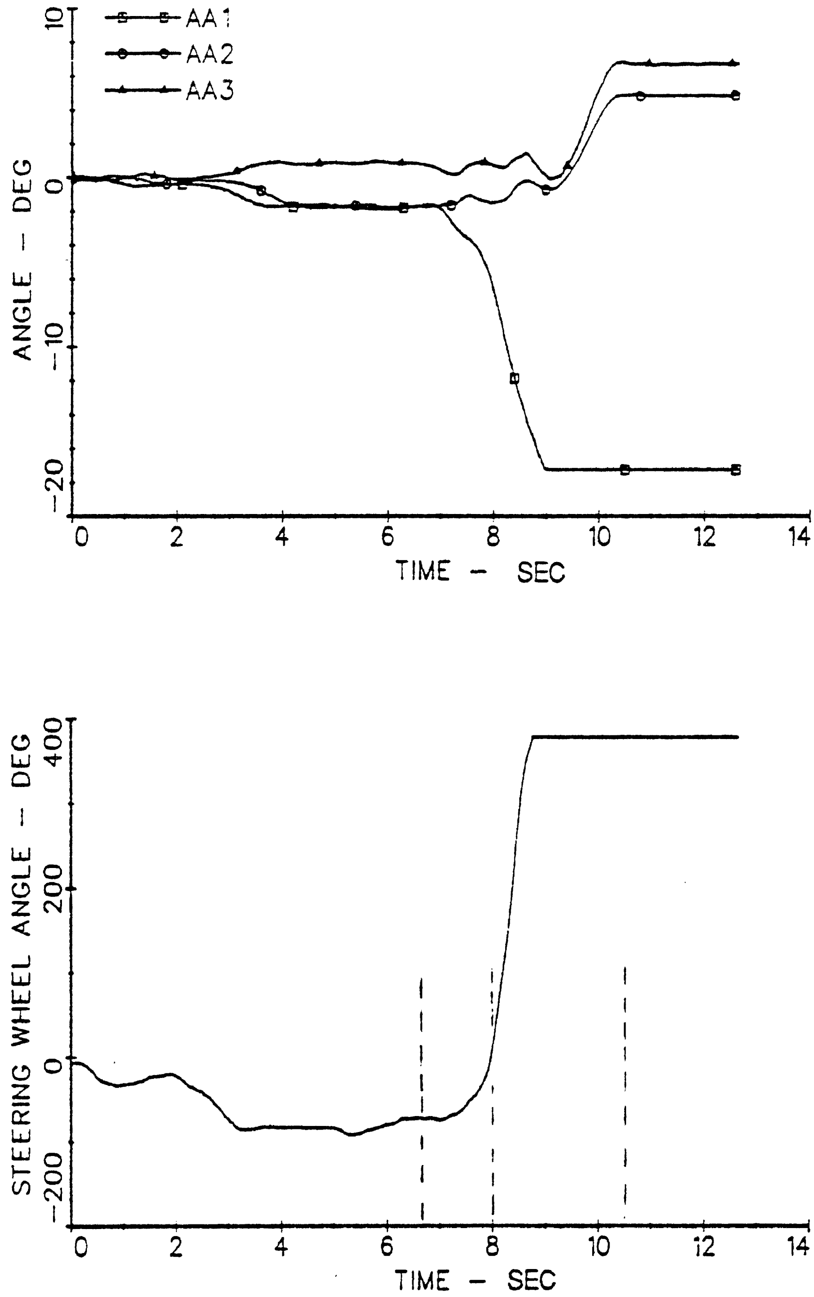
Figure 93 shows the situation in which the driver is able to control the A-train. In this case the braking level is not as severe, and even though all wheels except those on the front axle and on the right side of axle 3 have locked, the driver's efforts are enough to correct the jackknifing tendency that was begun during braking.



1 lb/in^2 = 6.895 kPa

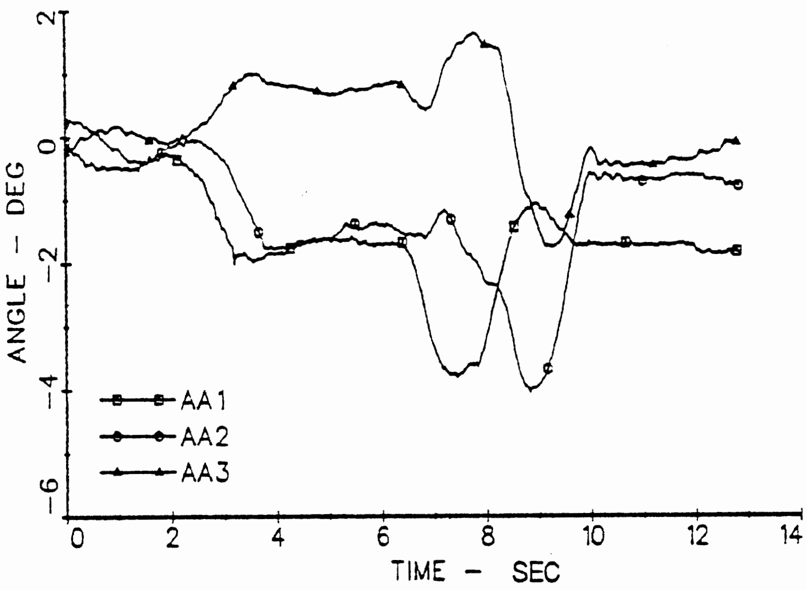
Dolly Type: A-Dolly
Loading Condition: Unloaded
Brake Pressure: Full
Wheels Locked: 2-5R, 2-5L

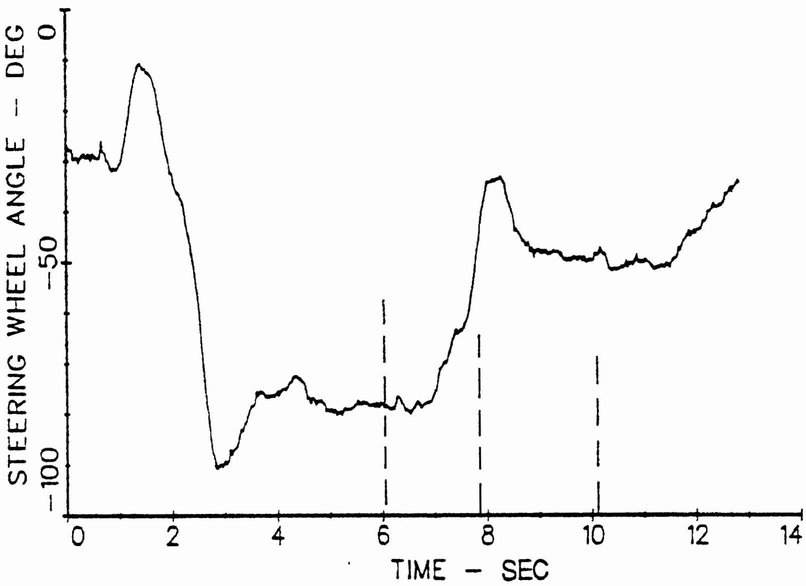
Figure 91. Brake pressure and longitudinal deceleration time histories during braking-in-a-turn: run number 443.



Dolly Type: A-Dolly
Loading Condition: Unloaded
Brake Pressure: Full
Wheels Locked: 2-5R, 2-5L

Figure 92. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 443.





Dolly Type: A-Dolly
Loading Condition: Unloaded
Brake Pressure: 65 lb/in^2 (448 kPa)
Wheels Locked: 2R, 4R, 5R, 2-5L

Figure 93. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 442.

Figures 94 through 99 present similar results for each of the innovative dollies that were tested. In general, all of the combinations did very well up to the point that the tractor jackknifed at a rate that was beyond the driver's ability to correct. Examination of these figures indicates some differences but nothing that indicates a major shortcoming of any particular type of dolly.

In figure 94 the notation "AXLST" refers to the steer angle of the wheels on the steerable B-dolly. In this case the centering force was zero and the wheels steered some because of the imbalanced braking force, but this did not affect the driver's ability to control the path of the vehicle. Even at full braking pressure the driver was able to keep the vehicle from a complete jackknife, as indicated in figure 94.

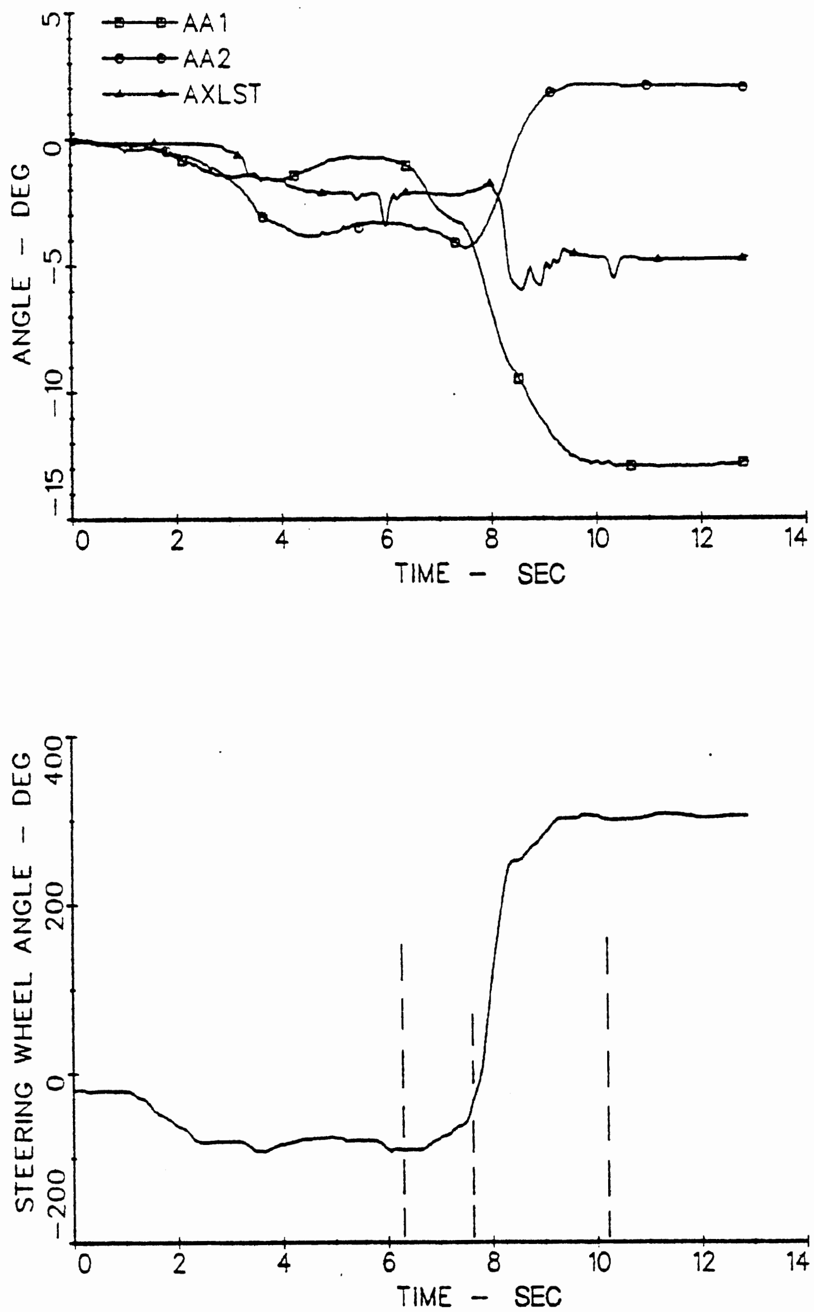
As shown in figure 95, the driver was also barely able to keep the vehicle with the prototype dolly from jackknifing at full braking pressure. At a pressure of 70 psi (482,650 Pa) the driver could hold the articulation angle AA1 to less than 10 degrees (see figure 96).

The simulation study indicated that the LA.3 arrangement might have a slightly greater tendency to jackknife than the other arrangements. The test results shown in figure 97 do not provide evidence supporting the trend noted in the calculations. The performance of the LA.8 is seen to be as good as that of the prototype arrangement.

The four-bar or trapezoidal arrangement was tested with both a short and long distance to the instant center of the linkage. In this case the test results, shown in figures 98 and 99, indicate that uncontrollable jackknifing did occur at slightly lower pressures than those obtained previously. However, this difference is not large and could well be due more to the brakes of the dolly than to the steering arrangement. (Each dolly has its own set of brakes, and some dollies are equipped with less powerful brakes than those typically used on semitrailer axles.) Since the 4BF performed slightly better than the 4BR with the same set of brakes, the results seem to indicate a slight advantage for the arrangement with the longer distance to the instant center.

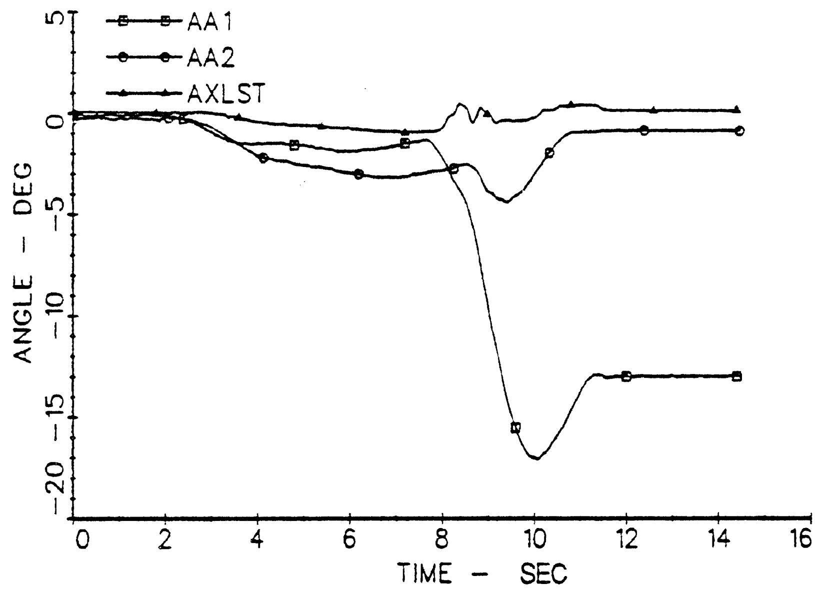
The braking-in-a-turn tests were also run for each of the dolly arrangements with the vehicle *loaded.* The brakes on the double were not effective enough to lock up more than one set of wheels in any of these tests. Under these conditions the driver had no difficulty controlling the vehicle. Braking performance was satisfactory for all of the dolly arrangements.

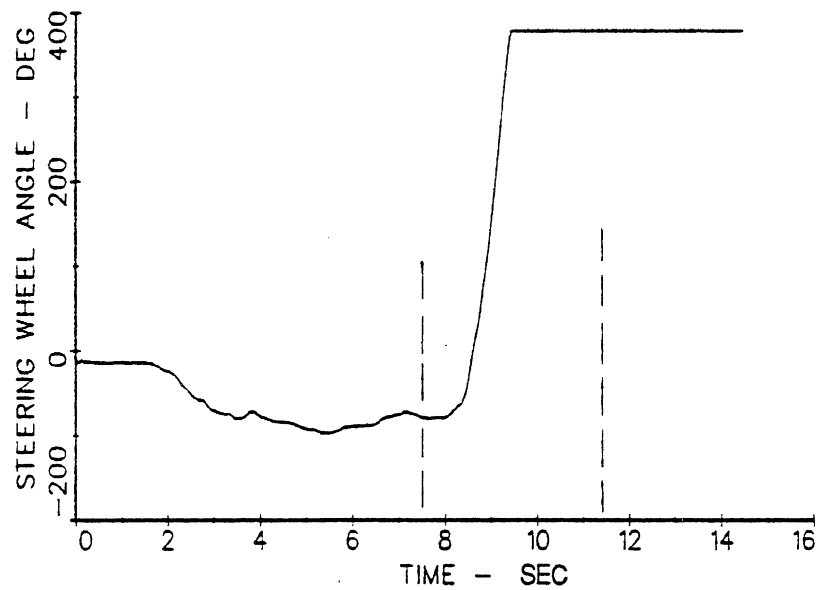
Straight-line braking tests . Straight-line braking tests were conducted on a wet split-friction surface with the vehicle having an initial velocity of 30 mi/h (48 km/h). The slippery side was on the right. This side was slippery enough that all wheels on the right



Dolly Type: SA.0
Loading Condition: Unloaded
Brake Pressure: Full
Wheels Locked: 2-5R, 2-5L

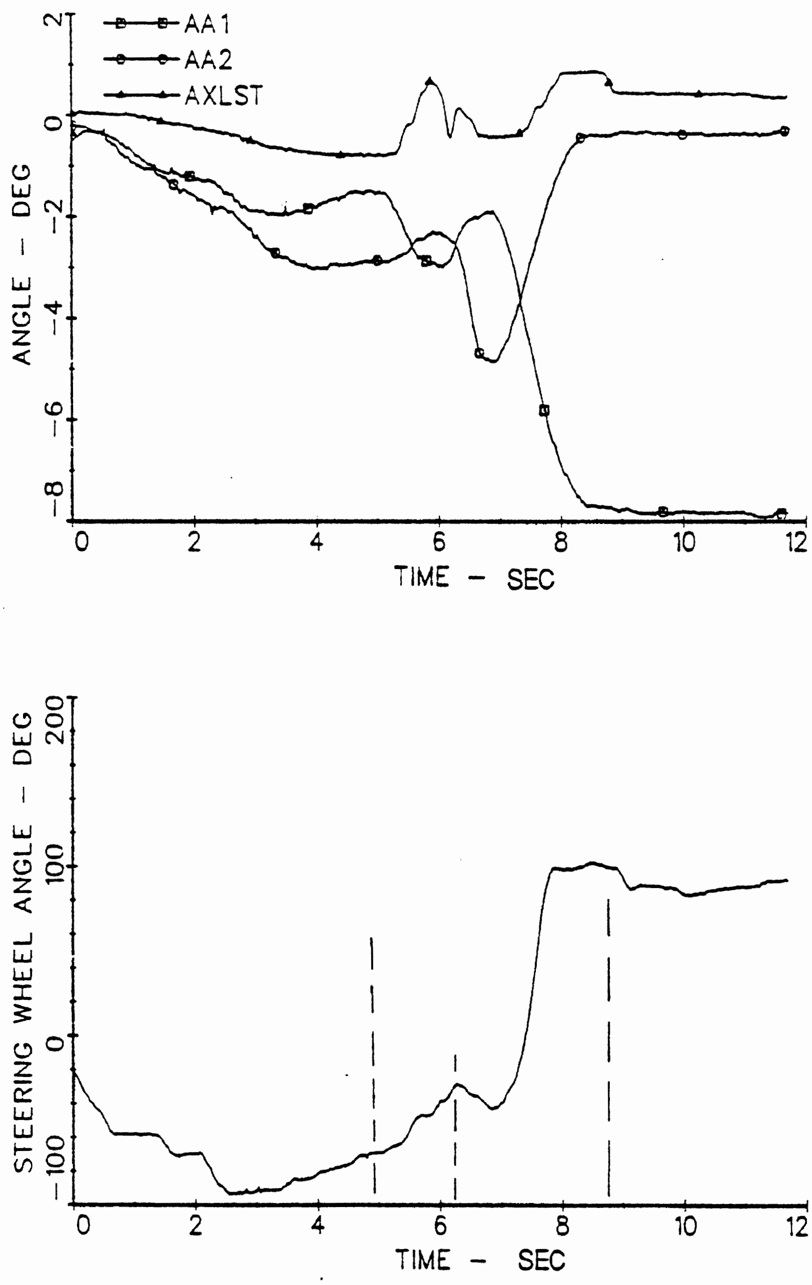
Figure 94. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 460.





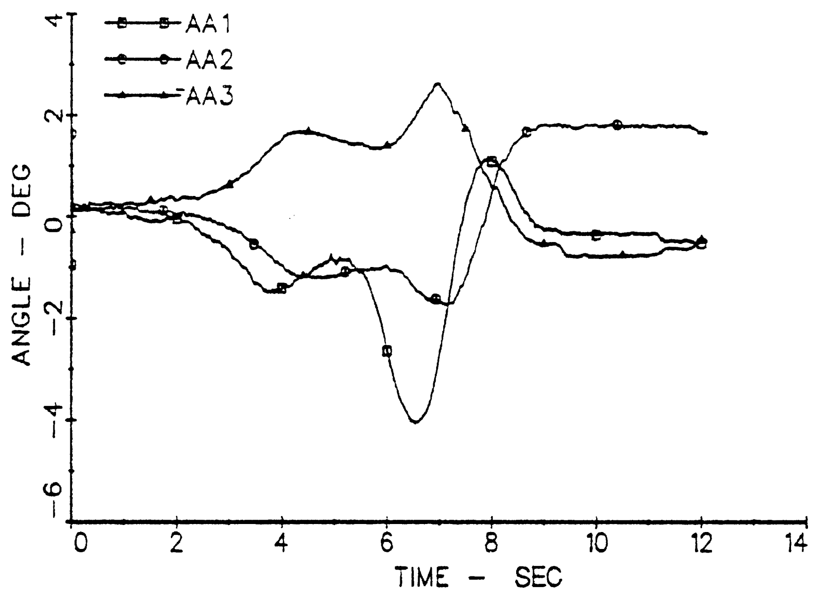
Dolly Type: CSB.30
Loading Condition: Unloaded
Brake Pressure: Full
Wheels Locked: 2R, 4R, 5R, 2-5L

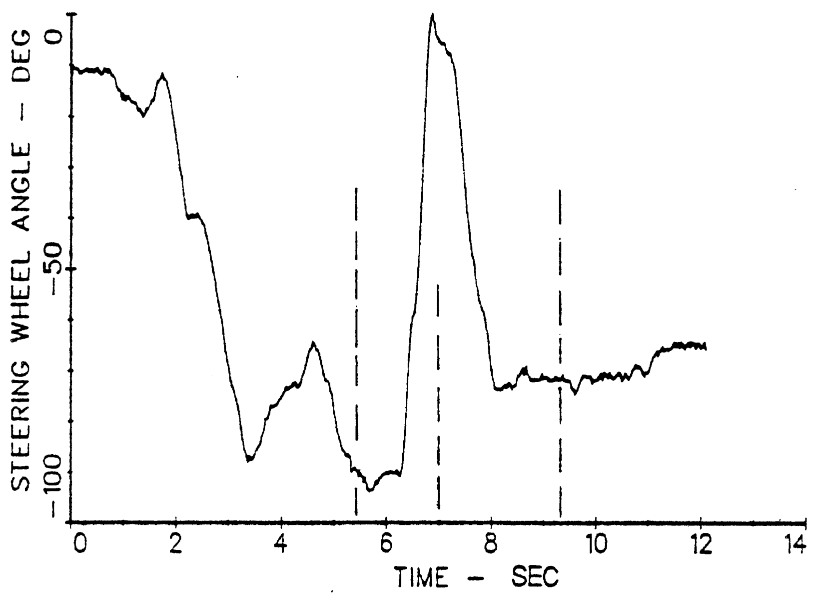
Figure 95. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 466.



Dolly Type: CSB.30
Loading Condition: Unloaded
Brake Pressure: 70 lb/in^2 (483 kPa)
Wheels Locked: 2R, 4R, 5R, 2-5L

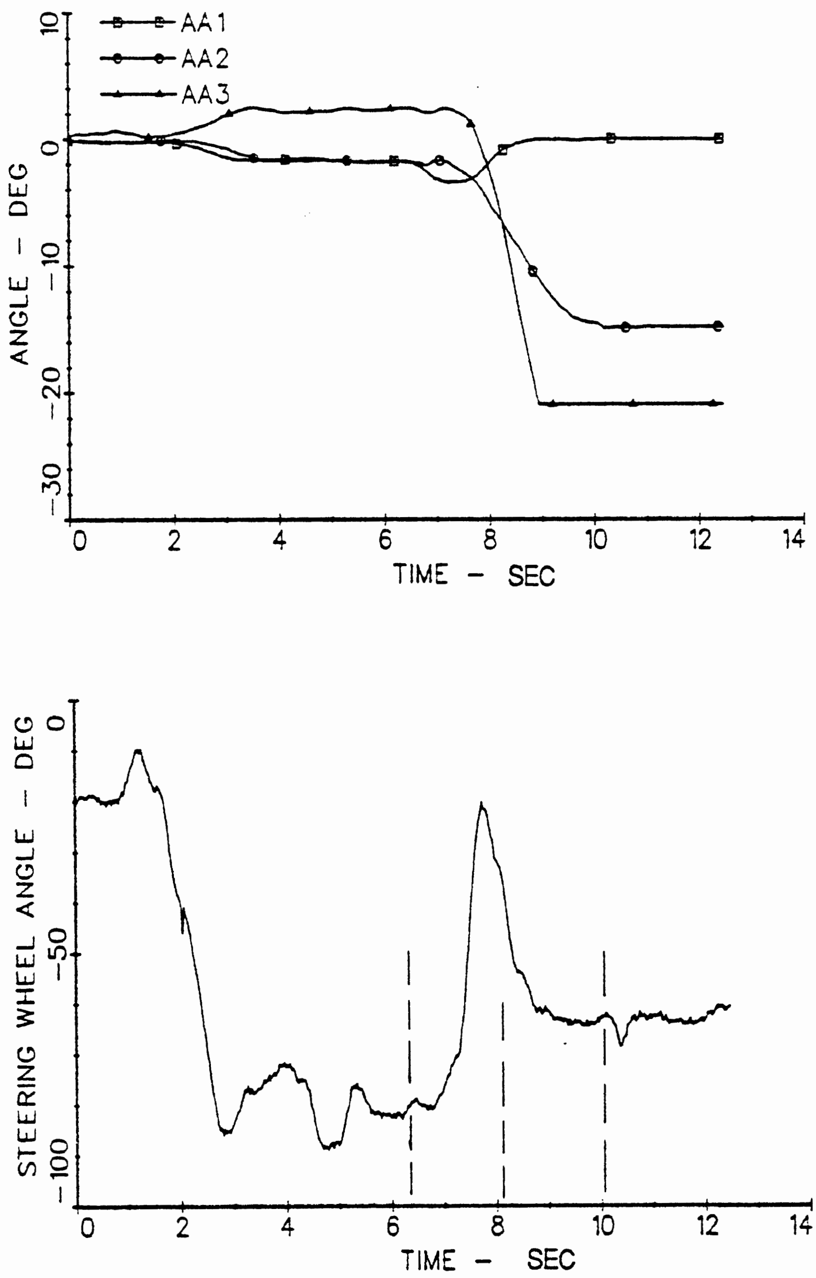
Figure 96. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 465.





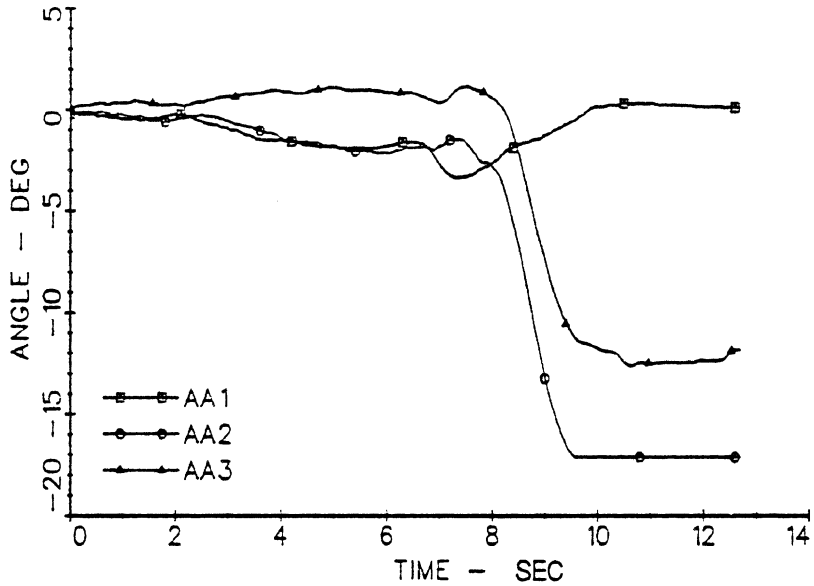
Dolly Type: LA.80
Loading Condition: Unloaded
Brake Pressure: 70 lb/in^2 (483 kPa)
Wheels Locked: 2-5R, 2-5L

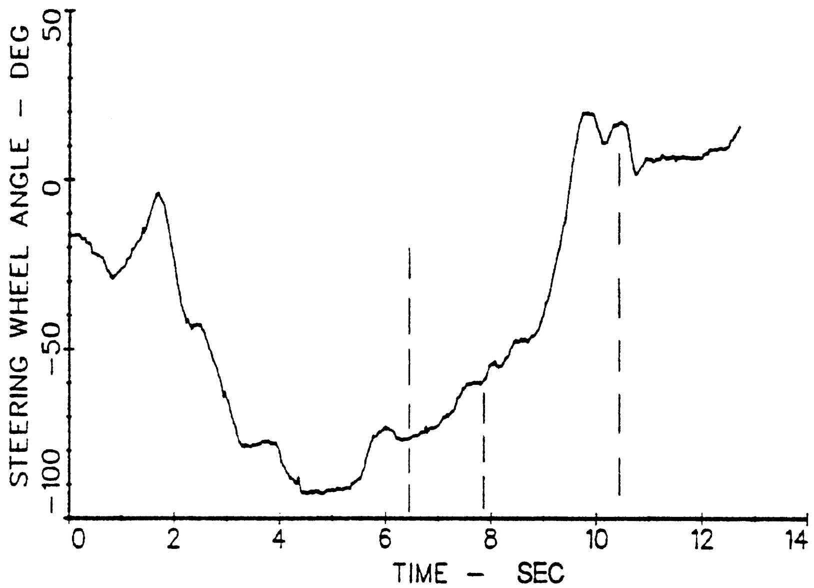
Figure 97. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 437.



Dolly Type: TRAP.F
Loading Condition: Unloaded
Brake Pressure: 65 lb/in^2 (448 kPa)
Wheels Locked: 2R, 4R, 5R, 2L, 4L, 5L

Figure 98. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 447.





Dolly Type: TRAP.R
Loading Condition: Unloaded
Brake Pressure: 55 lb/in^2 (379 kPa)
Wheels Locked: 4R, 5R, 2L, 4L, 5L

Figure 99. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 450.

side of the vehicle locked at very low braking pressures and well before any wheels locked on the left side of the vehicle. The left side was a rough concrete that had a high friction level even when wet. During braking the vehicle veered to the left and the driver countersteered to the right to maintain a straight path.

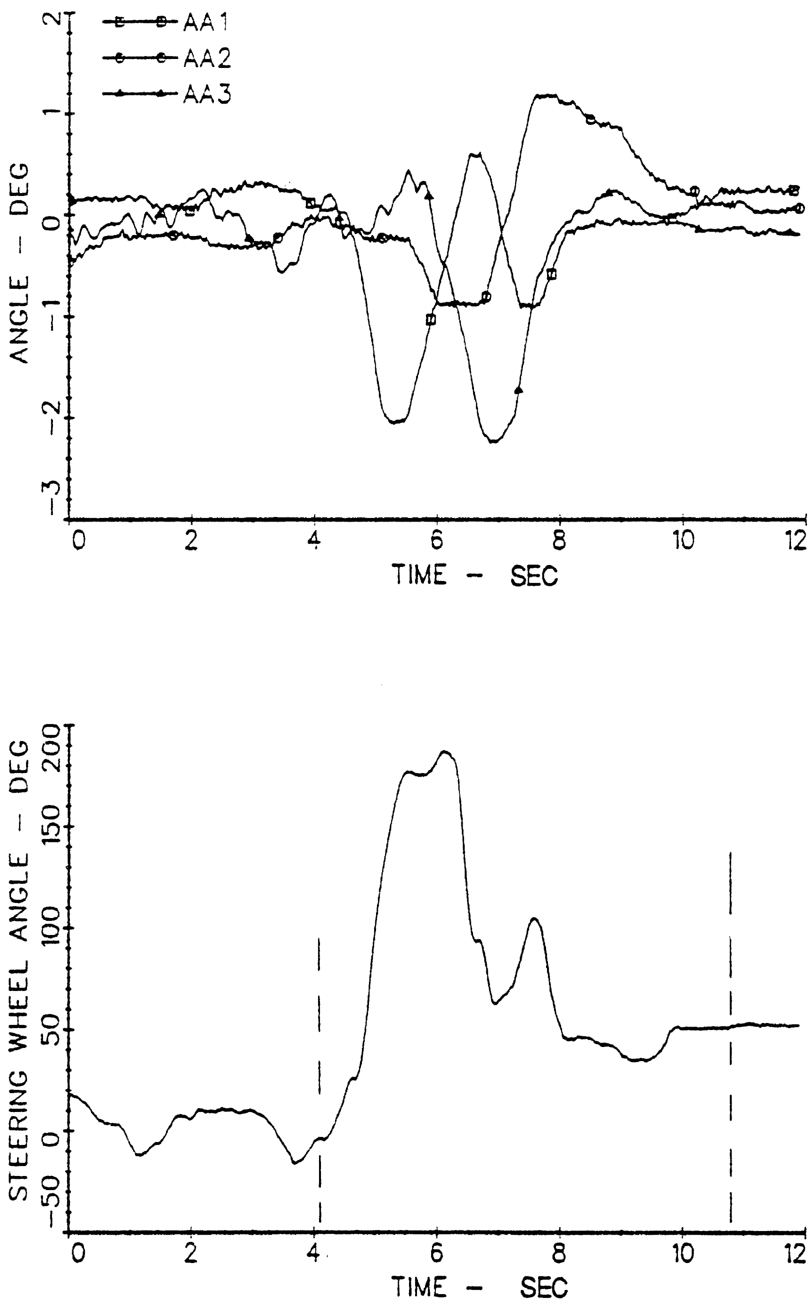
Tests were performed with the vehicle empty and fully loaded. When the vehicle was empty, the driver could control braking on the split-friction surface up to the point where the wheels on all axles except the left side of the front axle locked. This control capability was exhibited regardless of the dolly arrangement.

With the vehicle loaded, very dramatic jackknifing occurred when all the wheels on the rear axle of the tractor locked. As shown in figure 100, the driver can steer to control the vehicle when only the wheels on the right side are locked. However, as soon as the left side of axle 2 locks, the articulation angle at the tractor's fifth wheel diverges at a rate that is too rapid for the driver to correct (see figure 101). (In the case of the empty vehicle the driver could control this situation.) As before, this result is independent of the dolly arrangement; that is, the same type of jackknifing occurred as soon as all wheels on axle 2 locked. These results provide evidence showing the danger of locking all wheels on the rear axle of the tractor, but they do not indicate that any dolly arrangement has an advantage over the others.

f. Structural Loads and Mobility.

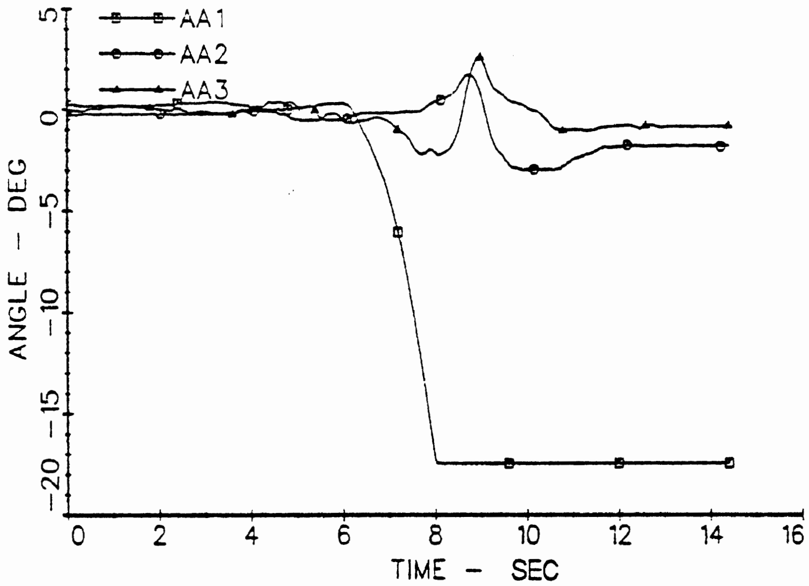
Structural Loads. Loading patterns at the connection joint between the leading semitrailer and the dolly are of interest in designing dollies with adequate structural strength. The dollies (which provide significant additional constraints beyond those of the A-dolly) are the LA, SA, and PRO dollies. The simulation study indicated that vehicles equipped with these dollies can survive *such* aggressive lane-change maneuvers that those dynamic maneuvers (and possibly braking maneuvers involving loss of control) may define the maximum hitch loading conditions. Nonetheless, the test vehicle, equipped with each of these three dolly types, was subjected to additional tests that impose exceptionally large loads at the pintle joint.

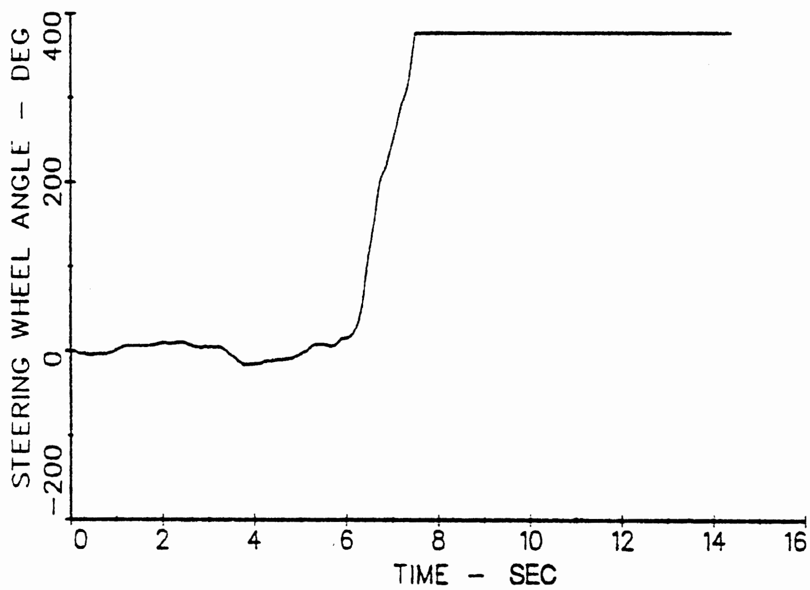
Two tests were used, viz. (1) curb climbing, and (2) severe steer. In the curb-climbing test, the vehicle traverses an 8-in (.20-m) raised "curb," at low speed, in the manner illustrated in figure 102. The 8-in (.20-m) height was chosen as a maximum realistic height for this test, since that number is given as the upper range of "barrier curb" heights in A Policy on Geometric Design of Highways and Streets, AASHTO, 1984. This test was



Dolly Type: A-Dolly
Loading Condition: Loaded
Brake Pressure: 50 lb/in^2 (345 kPa)
Wheels Locked: 1-5R

Figure 100. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 131.





Dolly Type: A-Dolly
Loading Condition: Loaded
Brake Pressure: 60 lb/in^2 (414 kPa)
Wheels Locked: 1-5R, 2L

Figure 101. Articulation angle and steering wheel angle time histories during braking-in-a-turn: run number 132.

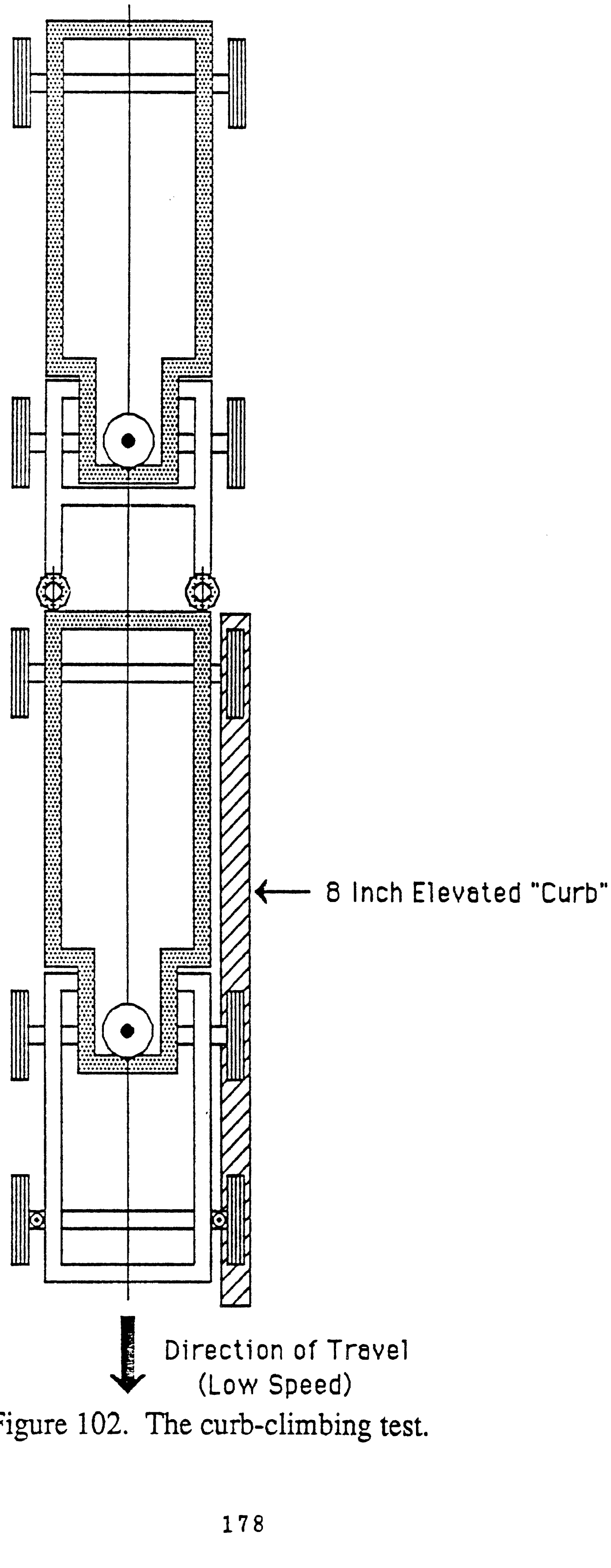


Figure 102. The curb-climbing test.

expected to subject the pintle joint to particularly high levels of roll moment as might possibly be experienced in low-speed maneuvering on uneven surfaces.

Figure 103 illustrates the geometry of the severe-steer maneuver. As the figure shows, the test begins with the trailers and dolly in a straight line and the tractor turned 90 degrees. (Such an arrangement can occur during hitching, and/or tight maneuvering of a train.) From this initial condition, the tractor moves straight ahead at low speed. The initial motion is expected to produce large tire sideslip at the first trailer and dolly tires, so that relatively high yaw moment and lateral forces are expected at the pintle.

Both of these tests were performed with fully loaded trailers.

Table 17 gives the maximum lateral force (F_y), yaw moment (M_z), and roll moment (M_x) obtained in the curb-climbing, severe-steer, and sine-steer maneuvers. The need for the B-dollies and the LA-dolly to withstand higher forces and moments is evident from the results presented in table 17. Note also that the maximum loads measured in the sine steer maneuvers generally confirm the simulation results, except that the measured roll moments (M_x) are smaller than those found in simulation. This discrepancy, however, is expected due to the special feature of the left pintles of the test dollies which was noted in the earlier description of the test dollies.

Mobility. Some applications of B-dollies in real service have generated mobility problems. Specifically, in fuel tanker service in Michigan, operators have experienced loss of driving traction at the tractor during tight maneuvering on uneven surfaces. Consider a scenario in which a fuel delivery unit is exiting a service station in which the apron on the service station slopes sharply downward (or upward) toward the public road. The vehicle must descend (ascend) the ramp and make a sharp exit turn onto the road. The difficulty occurs during the later stages of the turn, when the tractor semi-trailer unit has straightened out but (a) the second trailer remains sharply articulated with respect to the semi-trailer, and (b) the rear of the second trailer remains elevated. In this situation, the severely pitched second trailer may apply a large roll moment to the B-dolly through the fifth wheel. The B-dolly pintle coupling passes this moment through to the semi-trailer and finally to the tractor. In some cases, the conditions were found to be severe enough to lighten one side of the tractor drive axles sufficiently to cause wheel-slip and loss of mobility.

To examine this problem, static experiments were conducted with the test vehicle equipped with the self-steering B-dolly. (The special hinge joint at the left side pintle, which was discussed earlier in this chapter, was made rigid for this test.) The test vehicle was parked with weigh-scales under each of the four tractor wheels, and with the second

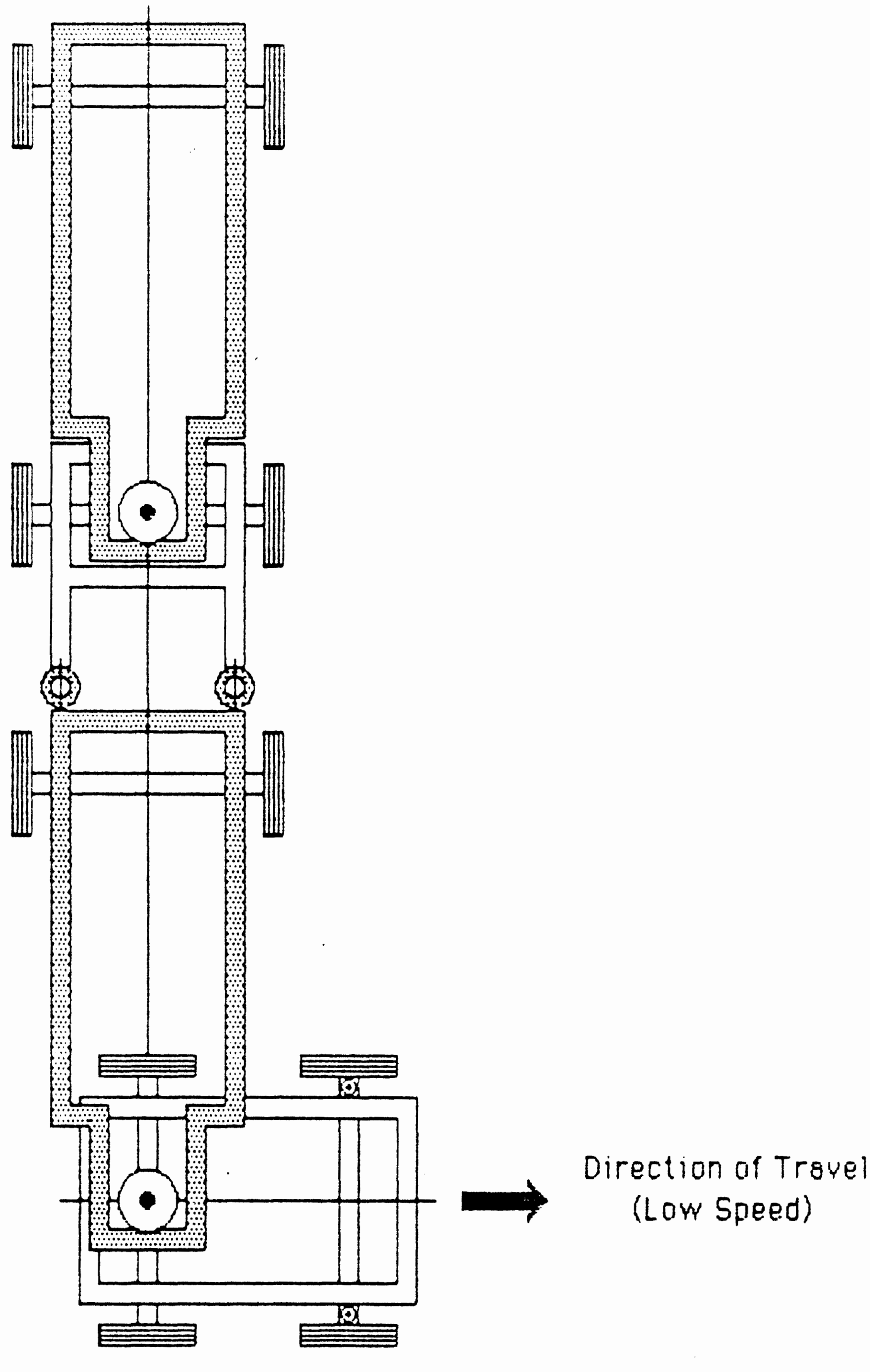


Figure 103. The severe steer test.

Table 17. Maximum Loadings

Vehicle Configuration	Fy, lbs		Mz, in-lbs		Mx, in-lbs	
	Sine Steer	Severe Steer	Sine Steer	Severe Steer	Sine Steer	Curb Climbing
AT	1,013	NA	6,400	NA	NA	NA
4BF	1,406	694	31,900	39,658	NA	NA
4BR	1,372	815	12,768	10,670	NA	NA
LA	3,087	5,863	136,353	305,580	NA	NA
SA-60	6,256	6,409	563,564	641,418	219,438	206,706
PRO	3,561	6,220	385,468	688,858	226,262	243,266

1 lb = 4.448 N
1 in-lb = 0.113 N-m

trailer articulated to 90 degrees with respect to the dolly (and first trailer). The first trailer was fully loaded, but half of the load was removed from the rear of the second trailer. The rear of the seconded trailer was then elevated to produce a severe pitch angle, and the change in vertical load at the tractor wheels was observed.

The results of this experiment clearly indicate that Western doubles using typical van trailers should not experience mobility problems similar to those observed in fuel tanker service. Unlike tank trailers, the van trailer was found to be sufficiently flexible in torsion along its length that the side-to-side load transfer at the tractor drive axle was low. In the most extreme test condition, the rear of the second trailer was elevated 44 inches (1.1 meters) producing a 9.2 degree pitch angle. At this extreme condition, the wheel loads at the tractor rear axle were found to be 6860 lb (3112 kg) at the left wheel and 9720 lb (4409 kg) at the right wheel, i.e. a 41 to 59 percent side-to-side distribution. Throughout the experiment, load transfer was found to be proportional to pitch angle.

OPERATIONAL IMPACTS

This discussion of operational impacts is divided into two parts--one addressing accident studies and the other presenting an economic analysis. The accident studies provide (1) an assessment of those aspects of the accident experience that might be improved by the use of innovative dollies and (2) a prediction of the safety benefits to be gained from the use of innovative dollies. The economic analysis considers the sensitivity of operating costs to the introduction of innovative dollies. Accident costs derived from the accident studies are used in the economic analysis.

1. Accident Analysis

The goal of this accident analysis was to determine the potential safety benefit of innovative dollies that would cause twin-trailer combinations to have handling characteristics similar to those of tractor-semitrailers (singles). The ideal way to do this would be to isolate all the accidents to which the performance of the existing dollies contributed and to assess the impact of the reduction or elimination of these accidents. There is, however, no way to identify such a group of accidents in existing data. An alternative strategy was therefore employed. This was to compare the accident experience of conventional doubles and singles, treating the latter as a surrogate for doubles combinations using innovative dollies. In this strategy particular emphasis is placed on those kinds of accidents, such as rollovers, that might benefit from a new dolly.

A number of studies comparing the accident experience of singles and doubles already exist, and most, if not all, of these are being evaluated in the current Double Trailer Truck Monitoring Study by the Transportation Research Board. However, these prior studies do not have the particular focus required here, and some of them have deficiencies,(24) while others depended on data that did not have complete coverage or did not clearly distinguish singles from doubles.(25) In any case, more recent data than that used previously are now available both for accident and for exposure information. It is worthwhile, therefore, to conduct a new analysis, not with the purpose of systematically analyzing all the differences in accident experience between singles and doubles, but rather with the purpose of predicting the potential benefit of replacing the current fleet of A-trains with a fleet of modified B-trains whose handling would approximate that of current tractor-semitrailers.

Treating the performance of singles as representative of the predicted performance of the improved doubles is in some ways making a "best case" argument for the effectiveness of the new dolly. It assumes that all current doubles deficiencies, as compared to singles, are related to steering performance and to rearward amplification. However, in some respects singles may currently have worse performance than doubles. Vehicle offtracking is one example. If these singles "problems" are significant, we might actually be able to improve doubles performance beyond that of singles and the potential benefit of innovative dollies may be underestimated. The strategy also assumes that doubles are currently operated in the same environment as singles, an assumption that might lead to an underestimation of the safety problem for the traditional doubles to the extent that they are actually operated under more favorable conditions. Here, once again, the potential benefit might be underestimated.

The accident analysis is organized as follows. First, there is a discussion of the sources of accident and exposure information employed by this study. This is followed by an attempt to corroborate and reconcile the accident databases. The recently developed UMTRI file of Trucks Involved in Fatal Accidents (TIFA) has been used as a yardstick here. Only those sources or reporting levels that appeared to match TIFA have been regarded as appropriate for use in calculating *numbers* of accidents. Other sources or reporting levels have been used for *descriptive* information where this was unlikely to be affected by bias from underreporting. Based on this assessment, the overall accident experience of the A-train doubles is compared with that of the singles representing the modified B-trains. Particular attention is then paid to those accidents where vehicle handling appears to be a factor and to those differences whose elimination might produce an economic benefit. Finally, the accident assessment is translated into economic terms in order to quantify the dollar value of the improvement in safety expected from the new dolly.

a. Sources of Information.

Before presenting the results of the safety analysis, a brief review of the relative merits of the available data files is in order. Three different national files of accidents were used. The first was UMTRI's Trucks Involved in Fatal Accidents file, which documents all fatal accident involvements by trucks, other than pickups, with a gross vehicle weight rating of over 10,000 lb (4,536 kg). The second was the file of carrier-reported accidents maintained by the Bureau of Motor Carrier Safety (BMCS), which, in theory, covers all injury accidents and all property-damage accidents resulting in $2,000 or more damage that occur to interstate motor carriers. The third source was the National Accident Sampling System (NASS) of the National Highway Traffic Safety Administration (NHTSA), under which a sample of all the police-reported accidents in the United States are investigated by trained staff. For exposure information, a single source was used: the 1982 Truck Inventory and Use Survey conducted by the Bureau of the Census.

Trucks Involved In Fatal Accidents (TIFA). In 1981 a survey of all large trucks involved in fatal accidents in the United States was initiated by UMTRI, with 1980 being the first accident year covered. This survey combines information from four sources: the Fatal Accident Reporting System (FARS) of the U.S. Department of Transportation's National Highway Traffic Safety Administration, the accident reports filed with the Federal Highway Administration's Bureau of Motor Carrier Safety, the hard copies of police accident reports, and telephone surveys conducted by UMTRI.

When UMTRI began its follow-up on fatal accidents, no *national* database of large truck accident involvements existed that had both comprehensive coverage and a detailed description of the truck. For example, the BMCS accident reports cover both property-damage and injury accidents and contain a detailed description of the truck, including number of trailers, cargo, empty weight, and cargo weight. The BMCS data do not, however, cover intrastate-only carriers and suffer from underreporting by private interstate carriers. The FARS data nominally provide complete coverage of all involvements in fatal accidents, but pay short shrift to the description of the truck. UMTRI decided to provide a dataset that combined the coverage of FARS with the descriptive detail of BMCS. The FARS file would be used as the starting point in selecting cases for follow-up and it would be assumed that FARS provided a complete census of fatal accidents. Limiting the information to involvements in fatal accidents was not ideal. It did, however, offer the advantage of convenience in that no other national sampling frame of truck involvements existed. It was also the case that involvements in fatal accidents constituted a large proportion of large-truck involvements in serious injury accidents. Using information from the NASS for 1981 through 1984, it is possible to calculate the proportion of all large-truck involvements in accidents that result in a serious injury (defined as AIS-3 or greater) that also result in a fatality. For all large trucks, the fatal involvements constitute 29 percent of the serious-injury involvements; for tractors, they constitute 32 percent. Thus the involvements reported by FARS represent almost a third of the national total of large-truck involvements in serious-injury accidents.

The TIFA database provides detailed descriptions of all medium and heavy trucks (greater than 10,000 lb (4,536 kg) gross vehicle weight rating) that were involved in a fatal accident in the continental United States, excluding Alaska. To produce the file, BMCS reports are first matched to FARS cases; for cases that cannot be matched, telephone interviews are conducted to obtain company and vehicle descriptions. Extensive editing and consistency checking is performed on all information obtained by interview. For example, VINs are decoded to confirm that the make and model information and the power unit description conform to published model specifications. UMTRI has acquired an extensive database on cargo weights and densities so that a cargo weight can, if necessary, be imputed from information on cargo type and volume. For cases matched with BMCS reports, the BMCS descriptions of vehicle type are checked for reasonableness and consistency with FARS.

The database documents roughly 5,000 involvements per year. Each of these involvements was originally obtained from FARS, but some cases were subsequently dropped because they were identified as not being medium or large trucks or because they were not involved as traffic units (i.e., they were legally parked). Overall, only some 87 percent of FARS identifications of large truck combination type are determined to be correct. Each year, some cases described by FARS as having a gross vehicle weight rating over 10,000 lb (4,536 kg) are discovered to be, in fact, light trucks, while other cases described by FARS as being light trucks are found to be medium or heavy trucks.

TIFA datasets have been prepared for each accident year from 1980 through 1983. In addition, UMTRI has combined the individual years into a single four-year file containing information on 19,962 large-truck fatal involvements. However, at the time most of the analysis reported here was performed, the 1983 file was not yet completed. Therefore, the three-year file covering 1980-82 and containing information on 15,018 involvements was used.

Overall, the TIFA files have a very low missing-data rate for the variables which document the truck configuration. In the 1980-82 file, the vehicle combination type is unknown for only one percent of the cases. Given this low rate of missing data, combined with the complete coverage of fatal involvements and the extensive checking performed for accuracy, there is every reason to believe that the TIFA data provide an accurate description of the relevant vehicles and accidents.

Bureau of Motor Carrier Safety (BMCS) Accident File. All interstate motor carriers are required by law to report all injury accidents and all property-damage accidents producing $2,000 or more damage to the Bureau of Motor Carrier Safety. (The threshold for property-damage accidents was recently raised to $4,200.) These reports, filed on form MCS 50-T, are keypunched by the agency. The version of the files used here was that built and maintained at UMTRI.

Since the BMCS reports are filed by the vehicle owner, the descriptions of the vehicle are, in the main, accurate. The problem with the BMCS files lies in underreporting. The 1980-82 TIFA file documents 9,811 involvements in fatal accidents by interstate carriers. Of these, only 5,201 cases could be matched with a BMCS accident report. This indicates that only 53 percent of the fatal involvements that are required to be reported to BMCS are actually reported. The underreporting is presumably greater for accidents that result only in injury or property damage. The picture is better for the ICC-authorized carriers. The 1980-82 TIFA file reports 5,432 fatal involvements by this group. Of these, 3,754, or 69 percent, were reported to BMCS. This underreporting has been taken into consideration when using the BMCS data for information on the distribution of involvements between singles and doubles combinations.

National Accident Sampling System (NASS). Under the NASS system, a small fraction of all the police-reported accidents in the United States are investigated by trained personnel. A complex, multistage sampling procedure is used to select cases in a jurisdiction with a NASS team, with sampling probabilities depending on the severity of the accident and the type of vehicle(s) involved. Accidents involving large trucks are sampled at somewhat higher probabilities than most other accidents of a given severity. In 1984, NASS teams investigated 11,598 accidents involving a total of 18,486 vehicles. But of these vehicles, even with the oversampling of large-truck accidents, only 531 were tractors pulling at least one trailer. Thus the main difficulty with using NASS cases as a representative sample of all large trucks involved in police-reported accidents is the small number of cases (and in consequence large sampling errors). There is an additional problem: in spite of the large amount of effort spent in case investigation, the description of large trucks in NASS is inadequate. Tractors are distinguished from straight trucks, and tractors pulling trailers are distinguished from bobtails. But there is no single variable indicating the number and type of trailers in a combination. Instead, there are variables which indicate the number of axles on each trailer or the absence of the appropriate trailer (first, second, or third). This method of reporting results in a large number of cases with unknown trailer arrangement; about 18 percent (unweighted) of the tractors indicated having at least one trailer are pulling an unknown number of trailers.

To alleviate the problem of the small number of relevant cases in NASS, a combined 1981-84 file was built for this study. This file contained all NASS involvements by tractors pulling one or more trailers other than those cases where a third trailer was definitely indicated. The number of cases in this four-year combined file was 2,700. Of these, 93 were involvements in fatal accidents, 1,032 were involvements in injury accidents, and 1,575 were involvements in property-damage-only accidents. This distribution points to another shortcoming of the NASS system: an inordinate number of the accidents investigated resulted in no injury.

Truck Inventory and Use Survey (TIUS). This survey is conducted by the Bureau of the Census every five years as part of the Census of Transportation. For the 1982 survey, a sample was drawn of all trucks in the national registration files maintained by R.L. Polk.

The sampling frame was stratified by state into two strata of light trucks and three strata of medium and heavy trucks. These three strata consisted of straight trucks with gross vehicle weight of 26,000 lb (11,790 kg) or less, straight trucks of over 26,000 lb (11,790 kg), and truck tractors. Surveys were performed by mail and recipients were informed that response was required by law. The overall response rate was 90 percent. The file contains information on 84,334 vehicles of which 19,663 are tractors. The estimated national population of tractors is 906,537.

For TIUS, respondents to the medium and heavy truck survey form are asked to indicate whether their vehicle pulled trailers and if so what the typical configuration was. By combining these responses with the reported annual miles traveled, it is possible to obtain national estimates of mileage for single- and twin-trailer tractor units. The 1982 TIUS does provide proper identification of single-trailer and double-trailer tractor combinations where the earlier, 1977 survey, had not done so.

It should be emphasized, however, that the results obtained are based on reported *typical* use; minority use of the tractor is not counted. In addition, because the mileage is derived from reported annual totals, it is not possible to estimate travel by road class, time of day, cargo weight, etc.

b. Corroboration of the Data Sources.

With the potential to incorporate in the analysis several different accident databases (TIFA, NASS, and BMCS), some procedure was needed to validate the various sources and assess their quality and utility. A good means to do so seemed to be to examine the reasonableness of the results obtained and to attempt to reconcile results from different sources. Where the results appeared absurd or the sources could not be reconciled, one source or another would be rejected. Since the TIFA data were believed to be complete and accurate, they provided a convenient yardstick for the assessment of NASS and BMCS.

Tables 18 and 19 show the comparison between NASS and BMCS on the one hand and TIFA on the other. They also show the number of involvements reported at different accident severities. Because of the small number of large-truck involvement cases in any single year of NASS, a four-year file of all the tractor-trailer involvements was created. The counts obtained are shown in Table 18 and the good correspondence on the fatals between NASS and TIFA should be noted. This shows that, in spite of small sample size, the NASS estimates for tractor involvements at the fatal level, and by inference at the injury level, can be trusted.

Table 18. Tractor-Trailer Accident Involvements by Data Source and Number of Trailers

Data Source	Number of Trailers	
	Single	Double
NASS 1981-84[a]		
Property damage only	465,521	5,996
Injury (excl. fatal) . . .	210,486	10,898
Fatal	12,806	673
TIFA[b]	13,103	627

[a]The cases in NASS where the vehicle had a trailer but the number of trailers was unknown were distributed proportionately to the cases with a known number of trailers within each accident severity level.

[b]The numbers in the TIFA file for 1981 through 1983 were inflated to four-year estimates.

Table 19. ICC-Authorized Tractor-Trailer Accident Involvements by Data Source and Number of Trailers

Data Source	Number of Trailers	
	Single	Double
BMCS 1980–83		
Property damage only	39,673	1,968
Injury (excl. fatal) . . .	41,071	1,786
Fatal	5,106	241
TIFA 1980–83	6,475	296

If the TIFA numbers for fatal involvements are combined with the NASS estimates of injury and property-damage involvement, one can calculate a ratio of property-damage to injury to fatal involvements for each class of vehicle. This works out to 36:16:1 for the singles and 10:17:1 for the doubles. If these numbers are to be believed, then for every fatal involvement of a single there are 16 injury involvements and 36 property-damage involvements. For each doubles fatal involvement, there are 17 injury involvements and 10 property-damage involvements. The very large difference between the two classes of vehicle in the ratio of property-damage to fatal involvements does not appear credible. This difference is apparently an artifact of the data and can be attributed to doubles units not being identified in NASS property-damage accidents. NASS has difficulty in identifying *any* kind of large truck in an accident because the vehicle has frequently left the area before the investigation begins. It seems reasonable that this problem would be more acute in the less severe accidents and that doubles, which are more likely to be long-haul, would have a greater tendency than singles to have left the area. Therefore, the NASS estimates of property-damage accident involvement will be excluded as unreliable.

In Table 19, BMCS counts of involvements for tractors with trailers are shown by accident severity and number of trailers. They can be compared with numbers obtained from the TIFA database. Accidents in Alaska and Hawaii were excluded from the BMCS data because they are not covered by TIFA. In addition, a recode was performed on the BMCS combination type field in the UMTRI file. Examination of the BMCS cases in the TIFA file indicated that all but a handful of the vehicles reported as tractors with full trailers and tractors with other trailers are in, fact tractors, with semitrailers. Similarly, almost all of the tractors reported as pulling a semitrailer and some other, non-full trailer were in fact pulling a semi- and a full trailer. Therefore, the appropriate recode was performed and this table reflects the result. Because of known underreporting of accidents to BMCS by nonauthorized carriers, the counts have been restricted to the ICC-authorized carriers. The TIFA numbers have been similarly restricted.

Comparing the BMCS counts of fatal accident involvements with the numbers from TIFA, it is clear that even for fatal accidents there is a certain amount of underreporting. However, this underreporting is almost identical for the singles and the doubles: for the former it is 21.1 percent and for the latter 18.6 percent. Thus any estimates of *injury* accident involvement rates derived from BMCS are not likely to suffer from differential reporting. There does, however, appear to be very substantial underreporting of property-damage accidents to BMCS. Even given the reporting threshold of $2,000 of damage, the roughly equal numbers of injury and property-damage accidents do not appear credible. If the counts of BMCS-reported property-damage accidents are to be disregarded, this does not mean that all the information on them provided by the file has no value. The descriptive information would only be questionable if one could hypothesize a bias effect from missing data, i.e., a situation in which the *unreported* cases might change one's conclusions about, for example, the proportion of rollover accidents by number of trailers or the amount of property damage from rollover accidents as compared to nonrollover accidents. In many situations the effect of such bias is unlikely to be great and the data from the BMCS property-damage accidents can be used for the description of accidents and their consequences.

c. Results of the Safety Analysis: Overall Comparison of the Singles and Doubles.

Using data from TIFA, NASS, and BMCS an overall comparison can be made between the safety experience of the current twin-trailer vehicles and the single-trailer vehicles representing the modified doubles. Such a comparison will not, given existing usage data, be able to take into account the operating environment in which the two classes of vehicles are used, but it will enable us to observe if there are any differences in safety that are of sufficient magnitude to affect the overall picture.

Table 20 combines counts of tractor-trailer fatal accident involvements from TIFA with exposure estimates from the 1982 TIUS to provide fatal accident involvement rates. Using 1982 TIFA alone, the doubles units appear to have a slightly *lower* rate of fatal accident involvements, both overall and for the vehicles operated by the ICC-authorized carriers. However, if we instead use accident data from three years because of the relatively small number of doubles units involved in fatal accidents in a single year (130 in 1982), the doubles have a slightly *higher* rate overall, but a somewhat *lower* rate for the ICC-authorized carriers. A reasonable conclusion would be one of no difference in fatal accident involvement rate between the singles and doubles.

Table 21 provides rates of involvement in accidents that resulted in at least one injury. The two sources of the involvement counts here are the 1981-84 combined NASS file and the 1982 BMCS file limited to ICC-authorized carriers only. Here the doubles have a slightly lower rate, but the difference is small enough and the data quality is uncertain enough to lead to a conclusion of no difference in injury accident involvement rates. Thus the overall assessment is one of no difference in either fatal or injury accident involvement rates between singles and doubles. It is possible that, particularly for the ICC-authorized carriers, the doubles may have a somewhat lower rate. The rates for the ICC-authorized carriers derived from the 1982 TIFA and BMCS files suggest that the ICC doubles may have a 21 percent lower fatal accident involvement rate and a 14 percent lower injury accident involvement rate than the ICC singles. However, these numbers do not take into account the operating environment in which the vehicles are used.

Table 20. Tractor-Trailer Fatal Accident Involvement Rates
by Data Source and Number of Trailers

Data Source for Involvement Counts	Number of Trailers	
	Single	Double
TIFA 1982		
All	6.9	6.7
ICC only ...	9.8	7.7
TIFA 1980–82		
All	7.2	7.6
ICC only ...	9.7	8.6

NOTE: Rates are per 100 million miles.

Table 21. Tractor-Trailer Injury (incl. Fatal) Accident Involvement Rates
by Data Source and Number of Trailers

Data Source for Involvement Counts	Number of Trailers	
	Single	Double
NASS 1981–84 (All) ..	123.2	115.5
BMCS 1982 (ICC only)	72.0	61.7

NOTE: Rates are per 100 million miles.

In order to gain at least some insight into the operating environment in which the two classes of vehicle being compared are used, the distribution of involvements by road class was ascertained. In the absence of real estimates of exposure by road class, the accident data can serve as a surrogate. Table 22 shows the proportions of fatal accident involvements by road class for the singles and doubles. Forty-eight percent of the doubles fatal involvements occur on divided roads as opposed to 41 percent for the singles. Table 23 shows the same comparison using all BMCS-reported involvements by ICC-authorized carriers. Here a remarkable 70 percent of the doubles involvements are on divided roads as compared to 52 percent of the singles involvements. Given the relative safety of the divided roads, the data may indicate that well over three-quarters of the ICC-authorized doubles' travel is on divided roads.

The distributions of involvements by road class point out the need for more detailed exposure data. The problem here is that the *overall* involvement rates for singles and doubles may indicate little or no difference in safety between the two classes of vehicle. However, the distributions in tables 22 and 23 indicate that the doubles have more than half of their fatal involvements and almost three-quarters of their overall involvements (at least for the ICC group) on divided highways. One possible explanation for the very large concentration in table 23 of doubles involvements on divided highways might be that rearward amplification is more of a problem on high-speed roads. However, if the ICC doubles involvements are broken out by accident severity, the divided roads account for 60.5 percent of the fatal involvements, 72.5 percent of the injury involvements, and 68.8 percent of the property damage involvements. Rearward amplification, which may be a major causal factor in a few fatal accidents and perhaps in some injury accidents, cannot be expected to account for all of the observed distribution of accidents by road class. This distribution appears to be a reflection of usage. It perhaps indicates that the overall fatal and injury accident picture is not quite as favorable to the current doubles as appears at first glance, since the doubles put on a large proportion of their mileage on relatively safe divided highways.

d. Results of the Safety Analysis: The Potential for Improving the Performance of Doubles.

While the analysis of the accident involvement rates of single- and twin-trailer vehicles cannot be carried any further, pending the availability of more detailed exposure data, the accident data alone can be examined for indications of areas in which the safety performance of the current doubles fleet might be improved through the use of innovative dollies. The focus here will be on handling-related factors, since they are the relevant ones

Table 22. TIFA 1980–82: Tractor-Trailer Fatal Accident Involvements by Road Class and Number of Trailers

Road Class	Number of Trailers			
	Single		Double	
	N	%	N	%
Divided .	4.057	40.9	215	48.0
Undivided	5,783	58.3	231	51.6
Unknown	74	0.7	2	0.4
Total . . .	9,914	100.0	448	100.0

Table 23. BMCS 1984: All ICC-Authorized Tractor-Trailer Accident Involvements by Road Class and Number of Trailers

Road Class	Number of Trailers			
	Single		Double	
	N	%	N	%
Divided .	13,029	51.6	959	70.0
Undivided	10,383	41.2	364	26.6
Unknown	1,819	7.2	47	3.4
Total . . .	25,231	100.0	1,370	100.0

in considering the safety benefit of a new dolly design. The experience of the singles, representing the modified doubles, will serve as a benchmark.

Tables 24 and 25 show the proportions of single- and multi-vehicle accident involvements for the two classes of vehicle. Here the hypothesis is that, if the current doubles fleet have greater handling problems than the singles fleet, the doubles should be overrepresented in the single-vehicle accidents. This indeed appears to be the case. Both the fatal accidents in table 24 and the ICC-authorized overall accidents in table 25 show an excess of doubles involvement in single-vehicle accidents.

The next two tables examine the distribution of first harmful event and most harmful event for the fatal involvements. In table 26, on first harmful event, the doubles are underrepresented in collisions with motor vehicles in transport, which follows from their overrepresentation in single-vehicle accidents. They have an excess of collisions with pedestrians and bicyclists, which may hint at some urban-related problems for doubles. As regards handling issues, the doubles are overrepresented in collisions with fixed objects which might result from loss of control, but they have proportionately fewer first-event rollovers than singles. The picture is not all that different in table 27, which gives the distribution of most harmful event. Once again the doubles have an excess of fatal collisions with pedestrians and bicyclists, and once again they have an excess of collisions with fixed objects. Now, however, the doubles slightly exceed the singles in the proportion of overturns. This suggests that the unmodified doubles have a tendency to rollover once an accident has begun, and that these rollovers are associated with severe injury.

This conclusion is reinforced by the distribution of rollover for the fatal involvements, shown in table 28. The doubles have a somewhat lower probability of a first-event rollover, but a considerably higher probability of a subsequent-event rollover. Table 29 examines another handling-related factor, jackknifes (which, as coded in FARS, include trailer swings). Here the doubles have a large excess of first-event jackknifes, but a slightly lower probability of a subsequent-event jackknife. Thus, from the fatal accidents at least, there is clear substantiation of handling-related problems for the doubles.

Tables 30 and 31 examine whether the indication of handling problems for doubles in the fatal data is borne out by information on all involvements reported by ICC-authorized carriers. Table 30 shows the distribution of noncollision type for the involvements reported to BMCS in 1984. The doubles have a smaller proportion of involvements in collision accidents. They are overrepresented in every major type of noncollision accident, particularly overturns. The probability of a rollover for a double is two-and-a-half times

Table 24. TIFA 1980-82: Tractor-Trailer Involvements by Number of Vehicles Involved and Number of Trailers

Number of Vehicles Involved	Number of Trailers			
	Single		Double	
	N	%	N	%
One vehicle	2,159	21.8	119	26.6
More than one vehicle	7,753	78.2	329	73.4
Unknown	2	0.0	0	0.0
Total	9,914	100.0	448	100.0

Table 25. BMCS 1984: All ICC-Authorized Tractor-Trailer Accident Involvements by Number of Vehicles Involved and Number of Trailers

Number of Vehicles Involved	Number of Trailers			
	Single		Double	
	N	%	N	%
One vehicle	12,203	48.4	786	57.4
More than one vehicle	13,028	51.6	584	42.6
Total	25,231	100.0	1,370	100.0

Table 26. TIFA 1980–82: Tractor-Trailer Involvements by First Harmful Event and Number of Trailers

First Harmful Event	Number of Trailers			
	Single		Double	
	N	%	N	%
Collision with:				
motor veh. in transport	7,245	73.1	301	67.2
pedestrian	682	6.9	44	9.8
pedalcycle	92	0.9	9	2.0
parked motor veh.	131	1.3	9	2.0
other non-fixed object .	254	2.6	14	3.1
fixed object	845	8.5	43	9.6
Overturn	603	6.1	25	5.6
Other non-collision	62	0.6	3	0.7
Total	9,914	100.0	448	100.0

Table 27. TIFA 1980–82: Tractor-Trailer Involvements by Most Harmful Event and Number of Trailers

Most Harmful Event	Number of Trailers			
	Single		Double	
	N	%	N	%
Collision with:				
motor veh. in transport	6,775	68.3	295	65.8
pedestrian	722	7.3	47	10.5
pedalcycle	90	0.9	9	2.0
parked motor veh.	79	0.8	3	0.7
other non-fixed object .	180	1.8	10	2.2
fixed object	423	4.3	24	5.4
Overturn	964	9.7	46	10.3
Other non-collision	287	2.9	14	3.1
Unknown	394	4.0	0	0.0
Total	9,914	100.0	448	100.0

Table 28. TIFA 1980–82: Tractor-Trailer Involvements
by Rollover and Number of Trailers

Rollover	Number of Trailers			
	Single		Double	
	N	%	N	%
None	8.251	83.2	357	79.7
First event	618	6.2	23	5.1
Subsequent event	1.045	10.5	68	15.2
Total	9.914	100.0	448	100.0

Table 29. TIFA 1980–82: Tractor-Trailer Involvements
by Jackknife and Number of Trailers

Jacknife	Number of Trailers			
	Single		Double	
	N	%	N	%
None	8,966	90.4	384	85.7
First event	719	7.3	56	12.5
Subsequent event	229	2.3	8	1.8
Total	9,914	100.0	448	100.0

Table 30. BMCS 1984: All ICC-Authorized Tractor-Trailer Accident Involvements by Non-Collision Type and Number of Trailers

Non-Collision Type	Number of Trailers			
	Single		Double	
	N	%	N	%
Ran off road	1.616	6.4	117	8.5
Jackknife	1,749	6.9	138	10.1
Overturn	1.942	7.7	262	19.1
Separation of units .	130	0.5	16	1.2
Fire	172	0.7	5	0.4
Cargo loss or spillage	132	0.5	2	0.1
Cargo shift	97	0.4	2	0.1
Other non-collision . .	47	0.2	1	0.1
Collision	19,346	76.7	827	60.4
Total	25,321	100.0	1,370	100.0

Table 31. BMCS 1984: ICC-Authorized Tractor-Trailer Property-Damage Accident Involvements by Non-Collision Type and Number of Trailers

Non-Collision Type	Number of Trailers			
	Single		Double	
	N	%	N	%
Ran off road	724	6.1	45	6.7
Jackknife	1,177	9.9	79	11.8
Overturn	813	6.8	191	28.5
Separation of units .	111	0.9	14	2.1
Fire	159	1.3	3	0.4
Cargo loss or spillage	102	0.9	1	0.1
Cargo shift	62	0.5	2	0.3
Other non-collision . .	31	0.3	1	0.1
Collision	8,700	73.2	334	49.9
Total	11,879	100.0	670	100.0

greater than the probability for a single. Table 31 makes the same comparison for property-damage-only accidents reported to BMCS by the ICC-authorized carriers. Here less than half the doubles involvements are in collision accidents, compared to almost three-quarters of the singles involvements. For these accidents, the doubles have a probability of rollover that is more than four times greater than that for singles.

In the BMCS injury-level (not including fatal) involvements reported by the ICC-authorized carriers, the doubles have about a 25 percent higher probability of rollover. Since there is evidence (from table 27) of doubles rollovers being correlated with injury, one might expect doubles accidents to result in somewhat more serious injuries than singles accidents. An examination of the NASS data, shown in table 32, tends to confirm this. Here the distribution of maximum AIS, or the highest level coded on the Abbreviated Injury Scale for any injury incurred in the accident, is shown. Injuries of unknown severity (AIS-7) have been added to the AIS-2 group.* According to the table, the doubles are involved in a lower proportion of MAIS-2 accidents but a higher proportion of MAIS-3 accidents. Thus the most severe injury incurred is likely to be more severe in an injury accident involving a twin-trailer combination than in an injury accident involving a single-trailer combination. Whether this difference is entirely attributable to handling-related accidents, or whether it is a by-product of road class, cannot be concluded from the NASS data. Unfortunately, there are insufficient cases to examine any accident factors. For the economic analysis, it will be assumed that this difference is related to handling and is therefore susceptible to elimination through a new dolly design.

Thus, although doubles have approximately the same overall accident involvement rate as singles, there are clear indications in the accident data that, in certain areas of performance, conventional twin-trailer vehicles do not perform as well as singles. Rollovers, in particular, are more common for these vehicles than for the tractor-semitrailer combinations. These rollovers tend to be costly. According to the 1984 BMCS data, the ICC-authorized carriers reported cargo spillage for 31 percent of their rollover involvements, but only for 4 percent of their non-rollover involvements.

e. The Expected Value of the Safety Benefit of a New Dolly.

* Of the singles involvements, 13.5 percent had MAIS-7; of the doubles involvements none. Adding the 13.5 percent to the MAIS-1 proportion for the singles would have resulted in concluding, purely on the basis of reallocating the MAIS-7 involvements, that there was a difference in the distribution of MAIS-1 and MAIS-2 involvements between the singles and the doubles and that this difference was unfavorable to the doubles. It was felt that it was more conservative here to allocate the MAIS-7 involvements to the MAIS-2 category, since it is improbable that any of the AIS-7 injuries are really AIS-3 or greater.

Table 32. NASS 1981–84: Maximum AIS (MAIS) for
Injury-Level Tractor-Trailer Involvements by Number of Trailers

MAIS	Number of Trailers			
	Single		Double	
	Weighted N	%	Weighted N	%
MAIS-1	105,565	64.7	5,248	62.1
MAIS-2	45,921	28.2	1,579	18.7
MAIS-3	9,628	5.9	1,566	18.5
MAIS-4	1,401	0.9	55	0.6
MAIS-5	614	0.4	0	0.0
Total ..	163,129	100.0	8,446	100.0
N	850		26	

There is no definite evidence of an overall difference in fatal and injury accident involvement rates between single- and twin-trailer tractor combinations. Therefore, the presumed safety benefit of a new dolly, which would give doubles combinations the handling characteristics of singles combinations, lies mainly in reducing the rate of single-vehicle rollover accidents at the property damage level and in improving the ability of a doubles combination to remain upright once an accident has begun. The *economic* benefit from a safer dolly lies in the elimination of those costs associated with the safety debits of the current doubles design.

The economic penalty from the current doubles safety experience can be defined as resulting from three elements. The first is the greater probability for a doubles of a rollover in an injury-level or a fatal accident. Since most of these rollovers occur as subsequent events, it will be presumed not that the injury or fatal accident would have been avoided entirely by a better-handling doubles combination, but merely that the rollover probability would have been reduced to that for singles combinations. The second current penalty results from the greater injury severity for the doubles injury accident involvements observed from the NASS data. The third penalty lies in the greater propensity of doubles to be involved in rollover accidents at the property-damage level. Since these are generally single-vehicle accidents in which the primary event is the rollover, the prediction here will be that better-handling twin-trailer combinations would be involved in fewer such rollovers and therefore in fewer property-damage accidents.

For the purposes of this analysis, it is necessary to estimate the overall number of doubles involvements at the various levels of accident severity. Because the current exposure data, TIUS, was collected in 1982, the same year will be used for the estimated accident data. According to the TIFA file for 1982, there were 131 involvements in fatal accidents by tractors pulling one semi- and one full trailer. The TIFA data does not include Alaska and Hawaii, but an examination of the FARS data for 1982 indicates only one potential doubles involvement in those two States. The number of injury-accident involvements can be estimated by applying the ratio of NASS injury to TIFA fatal involvements for doubles from table 18. Using this method, the number of doubles injury-accident involvements in 1982 is estimated at 2,294. Earlier, the apparent NASS underestimation of the number of doubles involvements in property-damage-only accidents was discussed. It seems more appropriate to use the NASS ratios for tractor-semitrailer combinations in estimating the number of doubles involvements in property-damage accidents. The resulting estimate is 5,074 involvements.

Using these estimated numbers of involvements, we can now calculate the three elements in the current safety penalty. The first of these resulted from the added potential for doubles to roll in injury and fatal accidents. From the TIFA data presented in table 28, it can be seen that the doubles have approximately a 50 percent greater probability of rollover than the singles. It can be presumed that, if the doubles behaved like singles, one-third of the subsequent-event rollovers in injury and fatal accidents would be eliminated. Instead of such rollovers occurring in 15 percent of the injury and fatal accidents, they would occur in 10 percent of such accidents. This would eliminate 121 rollovers. The cost of a doubles rollover can be estimated from data on property damage supplied to BMCS by the ICC-authorized carriers. For the period 1980 through 1984, the mean property damage in a doubles injury or fatal involvement in which a rollover occurred was $22,805. For nonrollover involvements, the mean property damage was $15,905. The cost penalty of the rollover, as reported by the carriers, was therefore $6,900. The additional cost of the 121 rollovers per year is therefore $834,000.

The next element in the safety penalty for the doubles is the higher injury severity in injury-level involvements. Currently, according to table 32, 18.5 percent of the doubles accidents result in at least one AIS-3 level injury, compared to 5.9 percent of the singles accidents. We can assume that, if doubles combinations behaved like singles combinations, 12.6 percent of the involvements would shift from MAIS-3 to MAIS-2. We can also assume that this shift would only affect one injury per accident. Then the shift would result a reduction in severity for injuries. The Multi-Injury Priority Rating model developed by Carsten estimates the societal cost of motor-vehicle injuries.[26] Applying this model to vehicle occupants in a combined 1980-83 NASS file, a mean societal cost of an injury at a given AIS can be calculated. For an AIS-3 injury, the cost is $11,577 and for an AIS-2 injury it is $5,026. The difference in the costs is $6,551. Presumably these costs are borne by carriers in the form of insurance premiums. Applying the cost difference to the 289 injuries, a total current penalty of $1,893,239 is obtained.

The final and most important element in the current cost penalty for the doubles is the effect of the additional rollovers at the property-damage level. According to table 31, the doubles rollovers constitute 28.5 percent of their property-damage involvements, as compared to 6.8 percent for the singles. The doubles rollovers can therefore be estimated at 1,446 per year. However, here the rollover is not a consequence of the accident, but constitutes the accident. Thus, eliminating a rollover would eliminate the accident entirely. Applying this logic, if the proportion of rollovers for the doubles were reduced to that of the singles, 1,181 rollover accidents would be eliminated. The mean cost reported by the ICC-authorized carriers for rollover accidents at the property-damage-only level was $11,634 for years 1980 through 1984. Thus, the cost penalty to the doubles of the additional rollover involvements is $13,739,754.

The total cost penalty per year from all three elements of the doubles safety deficit is \$16,466,993. Applying the 1982 TIUS estimated total annual mileage for the doubles of 1,967,620,000 mi (3,165,900,000 km) the estimated cost penalty is 0.84 cents per mi (0.52 cents per km). This 0.84 cents per mi (0.52 cents per km) is the expected value of the safety benefit of a new dolly that gives twin-trailer combinations the handling characteristics of single-trailer combinations. This analysis indicates that over 80 percent of this dollar safety benefit will result from a reduction in property-damage rollover accidents.

2. Economic Analysis

a. Introduction.

Objective. This economic analysis is designed to determine the costs and/or benefits produced by introducing B-dollies (or other innovative dollies) into a fleet that currently uses conventional A-dollies.

Method of Approach. In this study, certain issues surfaced as affecting the cost difference. Most of these issues were raised during a fact-finding trip to Canada to meet with operators of B-dollies. (Canadian truckers using B-dollies are offered the added incentive of being able to carry higher axle loads, and they may operate on secondary roads where doubles with conventional dollies are not allowed to operate.) This analysis addresses issues that were brought up most often and issues that trucking fleets indicated to be important. Those issues included loss of revenue from less weight hauled due to the additional weight of the B-dolly, the increased initial capital cost, tire wear, maintenance, scheduling and training, increased safety, and the ability to back up the vehicle.

Method of Analysis. To analyze the data gathered, a financial model was designed that represented the situation observed. A sensitivity analysis was performed to determine the significance of each of the issues involved. A "benchmark" situation was entered into the model, and variations relating to each issue were studied. In analyzing this data, each scenario was reduced to the change in the operating cost of a vehicle per mile per dolly to determine the effect that adding a B-dolly to a fleet would have on the fleet's operating costs. (See appendix B for a detailed listing of the calculations performed.)

b. Data Gathering.

Methods. Information about the costs involved in operating conventional A-dollies and B-dollies was obtained through discussions with both Canadian and U.S. trucking fleets. A fact-finding trip to Canada allowed us to meet with representatives of trucking fleets in Alberta and Saskatchewan. Those fleets operate approximately 140 B-dolly doubles with an annual mileage of approximately 20 million mi (32 million km). These discussions provided information on the changes in operational costs involved in converting to and operating B-dollies. Contacts with U.S. trucking fleets provided basic information about the costs involved in operating conventional dollies in the U.S. trucking environment, thereby providing information needed to compute the difference in costs between A- and B-dollies.

Further information about the costs involved in operating conventional A-dollies and B-dollies was obtained through written correspondence with Canadian and U.S. trucking fleets. The Canadians provided information about the operational impacts and costs of introducing B-dollies into a fleet that currently uses conventional A-dollies, since they had already experienced the effects of the change-over. U.S. trucking fleets provided information about the costs of operating an A-dolly, and estimates of the costs that the introduction of another type of dolly into their fleet would produce.

U.S. trucking fleets and equipment manufacturers were contacted to obtain additional information about the costs involved with operating a conventional A-dolly (e.g., the cost of a replacement set of tires).

c. The Financial Model.

Type of Analysis. The objective of the model is to determine the financial effects of using innovative dollies--for example, the double drawbar B–dolly as an alternative to the conventional A–dolly. The costs/benefits (cash flows) resulting from the investment are defined as an increase/decrease in the cost of operating the two dollies. In other words, the model analyzes the future *incremental* cash flows resulting from an *additional* investment made today.

Life of the Project. The life of the project--that is, the period over which the two dollies would be compared--is determined by the life of an A–dolly. The B–dolly has been in operation in Canada since 1979, and has not shown any structural problems unique to its design. With the help of information gathered from fleet operators, and from the reasoning that normal operation of double-trailer combinations results in relatively minor wear on the dolly, the useful life of a conventional dolly is assumed to be ten years.

Assumptions Concerning Economic Issues. The following items are incorporated in the financial analysis:

• Initial cost of the dolly. The B–dolly is assumed to cost $3,000 more than the A–dolly. This assumption is based on the fact that a Fruehauf single axle A–dolly (with tires) costs $4,500 and an ASTL B–dolly (with tires) costs $7,500. Differences in scrap value were taken to be negligible.

• Converting existing equipment. At least one semitrailer must be modified for every B–dolly purchased. The cost of installing the additional hitching hardware is estimated at $500.

In the event that double trailers are backed up at loading areas, there can be a cost savings associated with the elimination of yard tractors. Fleet operators believe that a yard tractor is justified for 60 trailer moves per day. Assuming that an inbound tractor spots its lead trailer, a yard tractor is justified for 30 double trailer combinations being assembled and disassembled (two trailer moves for each double) every day. The annual operating cost of a yard tractor is assumed to be $15,000. The assumed saving is the possibility of eliminating yard tractors.

• Major overhauls. Canadian operators of both dollies believe that B–dollies must undergo a major overhaul twice as often as A–dollies. In a few situations, B–dollies are operated under extreme conditions and need an overhaul as often as every year. The industry standard is, however, to overhaul an A–dolly every 500,000 mi (800,000 km) and a B–dolly every 250,000 mi (400,000 km). As an overhaul includes, among other things, fifth wheels, drawbar eyelets, steering systems, brakes, and springs, the cost of a major overhaul is kept as a variable and is defined as a percentage of the initial cost of the dolly. This cost is assumed to include factors related to both (a) time and materials for maintenance and (b) service time lost during maintenance.

• Preventive maintenance. The cost of regular maintenance such as inspection and lubrication depends upon the size of the fleet and the frequency at which maintenance is done. There is, however, a general view in the Canadian trucking industry that maintenance costs of the B–dolly are twice that of the A–dolly. The increase in maintenance cost is attributable to the maintenance of the steering and air systems of the steerable axle or wheels.

• Tire wear. During normal operation, the tires on conventional dollies last for 100,000–120,000 mi (160,000-193,000 km). Canadian fleet operators have determined that tires on B–dollies tend to wear out 10-15 percent faster than tires installed on A–dollies. The model assumes that radial tires ($900 for a set of four) are used, and considers replacement as an alternative to retreading.

• Scheduling costs. Scheduling varies across truck fleets, and practices are dependent on the size of the operation. Some large operations have delegated most of the scheduling exercise to computer programs which route tractors, semitrailers, and dollies according to variables such as trip length and freight being hauled. On the other hand, fleets with fewer units are more comfortable maintaining scheduling as part of the day–to–day administration of the trucking operation. Because the B–dolly introduces another variable to the scheduling problem, where dollies and semitrailers stop being completely interchangeable, there is bound to be an increase in scheduling costs. It is assumed, however, that there is a learning curve associated with the scheduling process, and the increase in cost will disappear over time.

The number of B–dollies, considered as a percentage of the total number of dollies, plays a fairly significant role in the scheduling exercise. A complete changeover, or a dolly fleet that is 100 percent B–dollies, would not affect the process of scheduling. If, however, half of the total number of dollies are B–dollies, then the increase in scheduling costs is assumed to be at its highest level. To account for this trend, the model assumes a triangular distribution in which scheduling cost varies as a percentage of the B–dollies in the fleet. The model assumes a single expense to update computer programs and any scheduling-related data bases.

• Training/loss of productivity. To address the fact that drivers and yard personnel must deal with a new piece of equipment, the model accounts for training and a cost associated with a temporary loss of productivity. The increase in time required to hitch the B–dolly is a specific example of a loss of productivity. Operators of B–dollies believe that, with some exceptions (such as hitching on uneven yard surfaces), hitching a B–dolly becomes as routine as hitching an A–dolly. The model uses a learning curve to account for the temporary nature of this cost.

• Backing up. Assembling and disassembling double-trailer combinations is a time-consuming task. The driver of an inbound vehicle leaves the rear trailer and the A–dolly in a drive–through staging area and backs the lead trailer into its loading dock. The driver then picks up the rear trailer from the staging area and backs it into its loading dock. Assembling a doubles combination would be a reverse of the process described above. Depending upon the distance from the loading area to the yard, the entire process of assembling and disassembling a set of double trailers could take up to an hour of the driver's time. Through elimination of one of the articulation points, the B–dolly gives the driver the ability to back up both trailers to their loading docks without using the staging area. The model assumes that the driver saves twenty minutes by not having to make two trips <u>to and from</u> the staging area. Assuming an internal labor rate of $21 (including benefits) the fleet operator saves $7 for each double-trailer combination that is assembled and disassembled.

• Loss of revenue from hauling less weight. Due to the steerable axle and a sturdier frame, the B–dolly weighs 1,000-1,500 lb (454-680 kg) more than the conventional dolly. Under conditions where vehicles are operated at maximum gross combination weight, the extra weight of the dolly displaces an equivalent amount of freight. For example, the loss of revenue depends upon a number of factors--type of freight (freight class), trip length, etc. The revenue from shipping 10,000 lb (4,535 kg) of freight from Ann Arbor, Michigan to San Diego, California (a distance of 2,373 mil (3,818 km)) is $2,125. If a vehicle is forced to forego carrying 1,000 lb (454 kg) of freight, then the loss of revenue for the trip is $212.50.

• Savings from fewer accidents. See "Accident Studies," part 1.e of this section.

• Ability to operate on secondary roads. A number of States regulate the operation of double-trailer combinations on their State and supplemental highways. Considering a situation where both trailers in a doubles combination are headed for the same destination off the federal highway system, the combination must be disassembled and each trailer must be transported to the site independently. If such regulation is removed because of the improved dynamic performance of doubles equipped with B–dollies, there could be a cost savings associated with the elimination of two trips to and from the local drop-off site.

• Permit to increase axle loads. As the loss of revenue from operating overweight dollies is so great, some Provinces in Canada have allowed truck fleets to increase their gross vehicle weights on a permit basis. This assumption, very similar to the one discussed above, addresses current highway regulation and has been included to describe a possible situation.

The Investment Rule. The Net Present Value (NPV) rule is used as a basis for analyzing the investment decision. The NPV rule reduces all forecasted cash flows to current dollars (based on a given discount rate) and is reliable in ranking projects which offer different patterns of cash flow. Other investment rules such as payback and average return on book are inadequate when analyzing incremental cash flows.

d. Application of the Financial Model.

The Independent Variables. The various types of variables used in the model are:

1. Influences of the excess weight of the B–dolly.

a. Percent of trips at gross vehicle weight (GVW). Though it is desirable to operate vehicles cube–full and at maximum axle loads, the actual loading situation is determined by the density of the freight being shipped. The reference condition assumes a hypothetical fleet operating its vehicles at gross vehicle weight 60 percent of the time.

b. Excess weight of the B–dolly. Most of the B–dollies being operated in Canada weigh 1,000–1,500 lb (454-680 kg) more than A–dollies. Some B–dollies manufactured in the United States are designed to operate at lower axle loads and are 500 lb (227 kg) lighter than their Canadian counterparts.

c. Miles per year per dolly. In addition to determining the maintenance costs of the dolly, this variable helps estimate the loss of revenue from having to operate a heavier dolly. The industry average for annual dolly-miles is 100,000 mi (161,000 km).

d. Freight charges. The freight charge has a direct bearing on the loss of revenue due to displaced cargo. Among other factors, the charge is dependent upon the distance the freight is to be shipped. For the reference condition, it is assumed that the charges are $21.21 per 100 lb (45 kg) of freight shipped from Ann Arbor, Michigan to San Diego, California. (However, the charges from Ann Arbor to Toledo, Ohio are $4.00 per 100 lbs (45 kg). On a per mile basis, the San Diego rate is $0.00894 per 100 lb (45 kg) per mi (1.6 km), and the Toledo rate is $0.08 per 100 lb (45 kg) per mi (1.6 km).)

2. Size of the fleet. The size of the operation and the proportion of B–dollies being added to the fleet determines the scheduling and training costs a company might incur. The pertinent variables are:

a. Number of B–dollies added to the fleet.

b. Total number of dollies owned by the fleet.

3. Maintenance.

a. Increase in tire wear. The reference truck fleet experiences 15 percent more tire wear on its B–dollies than on its A–dollies.

b. Cost of a major overhaul. The cost of a major overhaul is defined as a percentage of the original cost of the dolly. The model assumes that a B–dolly undergoes a major overhaul every two years while an A–dolly has a major overhaul once every four years. The cost of a major overhaul for the hypothetical fleet is assumed to be 20 percent of the cost of the dolly--that is, $1,500 for a B–dolly and $900 for an A–dolly.

c. Cost of preventive maintenance. The difference in the annual cost of preventive maintenance is assumed to be $500.

4. Number of backups per day. If a fleet operates over short distances where double trailer combinations must be assembled and disassembled more than once every day, then the ability to back up could have an impact on the profitability of the operation. The reference fleet, however, does not consider backing up to be a cost-saving alternative.

5. Accident savings. As the B–dolly's improved dynamic ability reduces the possibility of accidents, the B–dolly is assumed to save the fleet operator $0.008 per dolly per mi (1.6 km).

6. Discount rate. The discount rate is used to reduce future cash flows to current amounts and is assumed to be 10 percent after taxes for the shipping and transportation industry.

7. Scheduling and training.

a. Scheduling programs and data bases. This variable tries to address the single expense incurred by large fleets when scheduling-related computer programs and data bases are updated. A large fleet is assumed to operate at least 30 dollies.

b. Administrative training. The training of managers and administrative personnel is associated with a learning curve and is defined as the training cost per B–dolly during the first year of its introduction.

c. Driver/yard personnel training. The training of drivers and yard personnel is defined in a similar fashion.

8. Local deliveries, that is, the ability to operate on secondary roads. Assuming a change in regulation, the model assumes that a double trailer vehicle saves the fleet operator $30 for every local (off the federal highway system) trip it makes.

9. Permit to increase gross vehicle weight. Assuming a change in regulation, an increase in gross vehicle weight is used to offset the additional weight of the B–dolly.

Refer to table 33 for a list summarizing the variables and their reference values.

Exercising the Model for a Selected Reference Condition. If the financial model is used to analyze the decision by a fleet operator to purchase six B–dollies, then the incremental cash flows projected over ten years are as shown in table 34. (The variables used in the model are displayed in table 33.)

Table 33. The Variables Used in the Financial Model

Variable Names	Reference Values
Percentage of trips at max GVW	60 %
Additional dolly weight	1000 lbs
Miles per year per dolly	100,000 miles
Charge/lb/mile for freight hauled	$0.0000894
B-dollies added to the fleet	6 B-dollies
Total number of dollies owned	15 Dollies
Percent of tire wear increase over A-dolly	15 %
Overhaul cost (percentage of initial dolly cost)	20 %
Preventive maintenance - per year	$500.00
Double assembly & disassembly (B-dolly backup)	0 per day
Accident savings per mile per B-dolly	$0.008
Annual discount rate	10 %
Upgrading scheduling programs	$0.00
Administrative Expenses (first year)	$800.00
Driver/yard personnel training (first year per dolly)	$1,000.00
Local deliveries	0 per year
Overweight hauling allowance	0 lbs

1 lb = 0.454 kg

1 mi = 1.609 km

Table 34. The Reference Condition, Results Correspond to the Purchase of Six B-Dollies.

Δ costs/benefits between A and B dollies	Year 0	Year 1	Year 2	Year 3	Year 4
Initial cost of dollies	($18,000.00)	$0.00	$0.00	$0.00	$0.00
Converting existing equipment	($3,000.00)	$0.00	$0.00	$0.00	$0.00
Major overhauls	$0.00	($8,670.00)	$0.00	($4,110.00)	$0.00
Tire wear	($810.00)	($810.00)	($810.00)	($810.00)	($810.00)
Preventive maintenance	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)
Scheduling	($800.00)	($294.30)	($108.27)	($39.83)	($14.65)
Training	($6,000.00)	($2,207.28)	($812.01)	($298.72)	($109.89)
Ability to back up	$0.00	$0.00	$0.00	$0.00	$0.00
Less weight hauled	($32,184.00)	($32,184.00)	($32,184.00)	($32,184.00)	($32,184.00)
Fewer accidents	$4,800.00	$4,800.00	$4,800.00	$4,800.00	$4,800.00
Ability to operate on secondary roads	$0.00	$0.00	$0.00	$0.00	$0.00
Allow higher GVW	$0.00	$0.00	$0.00	$0.00	$0.00
Total	($58,994.00)	($42,365.58)	($32,114.28)	($35,642.55)	($31,318.55)
Net Present Value	($276,223.58)				
Cost increase to cover loss /100lb / mile	$1.15E-04				
Change in operating cost / dolly / mile	$0.0435				

The table entries are the incremental benefits or (costs) associated with using B-dollies instead of A-dollies.

1 lb = 0.454 kg

1 mi = 1.609 km

Table 34. The Reference Condition, Results Correspond to the Purchase of Six B-Dollies. (Continued.)

Δ costs/benefits between A and B dollies	Year 5	Year 6	Year 7	Year 8	Year 9
Initial cost of dollies	$0.00	$0.00	$0.00	$0.00	$0.00
Converting existing equipment	$0.00	$0.00	$0.00	$0.00	$0.00
Major overhauls	($8,670.00)	$0.00	($4,110.00)	$0.00	($8,670.00)
Tire wear	($810.00)	($810.00)	($810.00)	($810.00)	($810.00)
Preventive maintenance	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)
Scheduling	($5.39)	($1.98)	($0.73)	($0.27)	($0.10)
Training	($40.43)	($14.87)	($5.47)	($2.01)	($0.74)
Ability to back up	$0.00	$0.00	$0.00	$0.00	$0.00
Less weight hauled	($32,184.00)	($32,184.00)	($32,184.00)	($32,184.00)	($32,184.00)
Fewer accidents	$4,800.00	$4,800.00	$4,800.00	$4,800.00	$4,800.00
Ability to operate on secondary roads	$0.00	$0.00	$0.00	$0.00	$0.00
Allow higher GVW	$0.00	$0.00	$0.00	$0.00	$0.00
Total	($39,909.82)	($31,210.86)	($35,310.20)	($31,196.28)	($39,864.84)
Net Present Value					
Cost increase to cover loss /100lb / mile					
Change in operating cost / dolly / mile					

The table entries are the incremental benefits or (costs) associated with using B-dollies instead of A-dollies.

1 lb = 0.454 kg

1 mi = 1.609 km

The first column in table 34 refers to the economic issues outlined previously in part 2.c. of this section. The following columns, titled Year 0 (the current year) through Year 9, contain the annual cash flows resulting from each of the items mentioned in the first column. Negative cash flows, or expenses, are shown in parentheses.

In the model, cash flows occuring in Year 0 result from operational costs and one–time investments such as purchasing, scheduling, and equipment conversions. Cash flows in the following years result from changes in operational costs only. In the reference case, a fleet adding six B–dollies versus one adding six A–dollies would have to spend an additional $18,000 to cover the initial cost of the dollies. This cost, plus other initial investments and operational costs, results in a loss of $58,994 in the first year of the project. During the second year, the fleet operator would lose $42,439.16 due to increases in operational costs alone.

The Net Present Value (NPV) of the sum of the incremental cash flows over the life of the project results in a total negative cash flow of $276,524.07. It is important to emphasize that this loss is an *incremental loss* due to a decision to buy B–dollies instead of A–dollies.

Assuming that the reference fleet could raise its shipping charges to cover its incremental loss, the freight charges would have to be increased by $27.29 ($0.000115 per 100 lb (45 kg) per mi (1.6 km) as indicated in table 34) for 10,000 lb (4,536 kg) of cargo to be shipped from Ann Arbor to San Diego. This is a rate increase of approximately 1.3 percent.

The increased operating cost of a B–dolly--that is, the NPV of the investment less the one–time costs of scheduling, purchasing, and converting equipment--is computed (per dolly per mile (1.6 km)) in the last row of the column of Year 0. It is this value (0.0435 dollars per dolly per mi (1.6 km)) that is used as the reference value in the following sensitivity analysis.

Sensitivity Analysis. It is often helpful to see how a project fares under various scenarios. Sensitivity analysis is helpful in determining the key variables that determine whether a project fails or succeeds. Table 35 contains a list of the reference values and variations used in the analysis.The influences of the variations listed in table 35 are displayed in figures 104 and 105. Figure 105 shows that reasonable increases or decreases in some of the independent variables have little influence on the operating cost. (The reference or baseline conditions are enclosed in square brackets for easy identification in the figures.) Examination of figure 104 indicates that increases in (1) freight charges, (2)

Table 35. Variations Used in Sensitivity Analysis

Variables	Reference Values	Sensitivity Variations
Percentage of trips at max GVW	60%	0%; 100 %
Additional dolly weight	1,000 lbs	500; 1500 lbs
Miles per year per dolly	100,000 miles	60,000; 140,000 miles
Charge/lb/mile for freight hauled	$0.0000894	$0.0000447; 0.0001656
Percent of tire wear increase over A-dolly	15 %	0%; 30%
Overhaul cost (percentage of initial dolly cost)	20 %	0%; 40%
Preventive maintenance - per year	$500	$0; $1,000
Double assembly & disassembly (B-dolly backup)	0 per day	0.5; 2
Accident savings per mile per B-dolly	$0.008	$0.00; $0.16
Annual discount rate	10 %	8.5%; 11.5%
Upgrading scheduling programs etc.	$5,000	$0.00; $10,000
Administrative expenses (first year)	$1,000	$0.00; $2,000
Driver/yard personnel training (first year)	$1,000	$0.00; $2,000
Local deliveries	0 per year	130; 260
Overweight hauling allowance	0 lbs	1,000 lbs; 1,500 lbs

1 lb = 0.454 kg

1 mi = 1.609 km

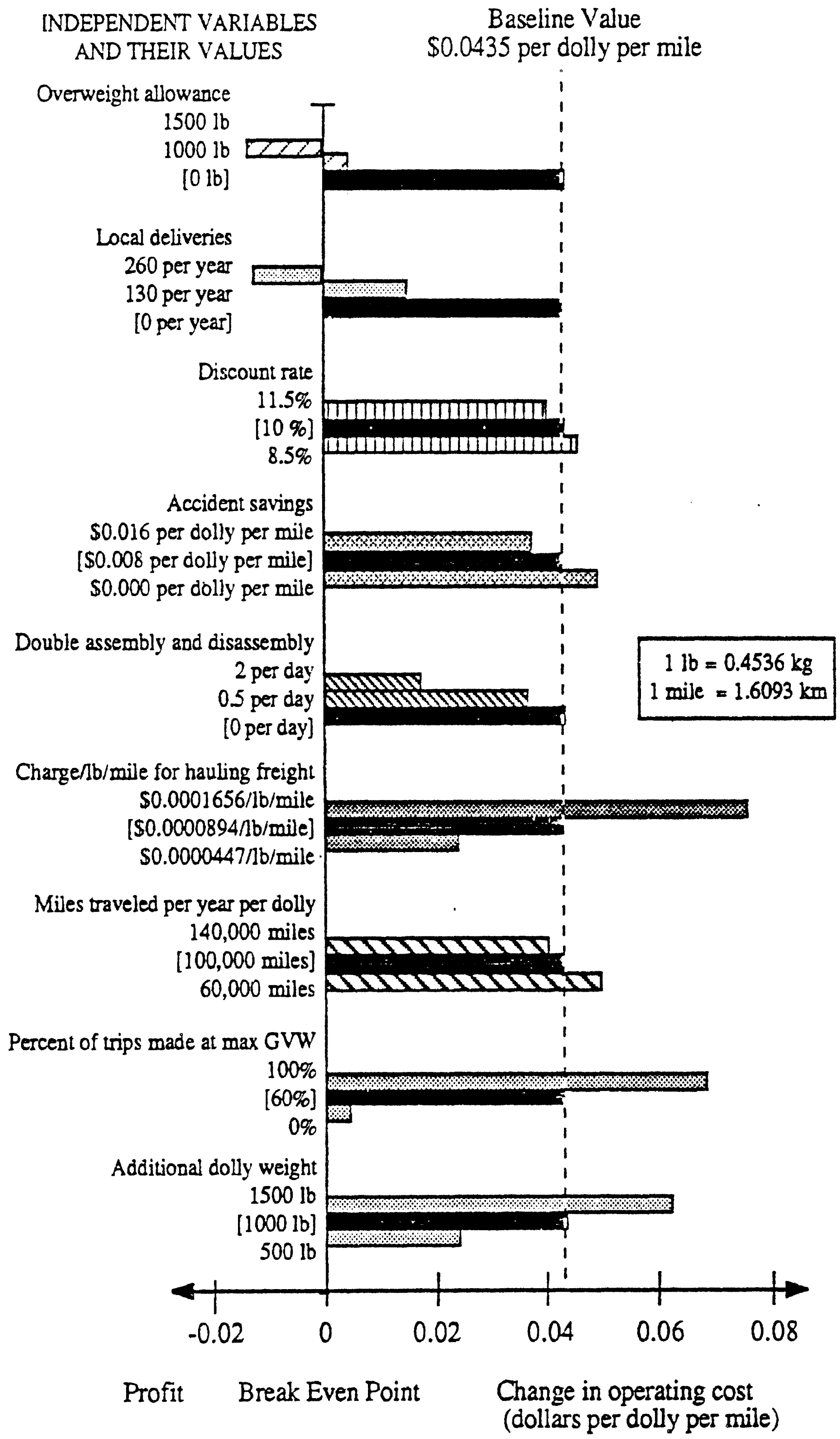


Figure 104. Operating cost sensitivities for a current small fleet (the more important variables).

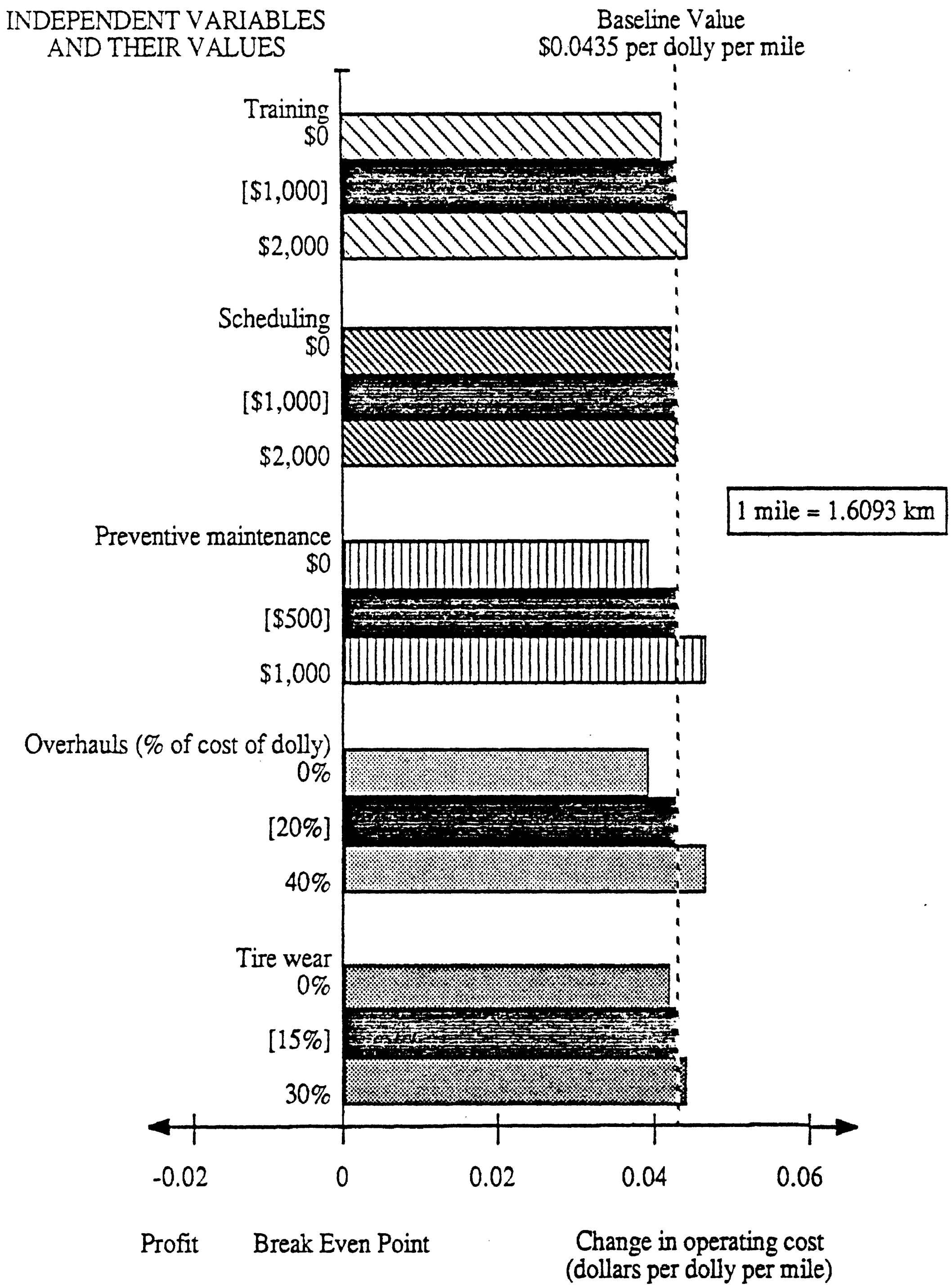


Figure 105. Operating cost sensitivities for a current small fleet (the less important variables) .

percentage of trips made at GVW, and (3) dolly weight have significant influences on the changes in operating cost associated with acquiring B-dollies. The "Break Even Point" in figure 104 is where the costs associated with purchasing and operating an A-dolly are equal to the costs associated with purchasing and operating a B-dolly. Clearly, decreases in the values of these variables are desired in order to approach a profit-making situation. The profit side of the bar chart is reached if (a) the owners of B-dollies are given a 1,500-lb (680-kg) weight allowance to compensate for the additional weight of the B-dolly, or (b) each B-dolly is allowed in local service such that it is involved in approximately 220 local deliveries a year. This ability to make local deliveries is associated with the ability of the double to be backed up if it employs a B-dolly. The ability to back up also makes it convenient to avoid some of the assembly and disassembly operations associated with A-dollies. With regard to accident costs, the results presented in figure 104 show that accident costs have only a moderate influence on the financial picture.

The economic analysis presented in the preceding discussion has painted a picture which indicates that the introduction of B–dollies into the United States may not be a profitable investment decision. The reference condition presented earlier was designed to reflect the current regulatory environment. Introducing, however, a hypothetical situation where heavier gross vehicle weights and the use of secondary roads were allowed, the cash flows shown in table 36 demonstrate that the decision to invest in B-dollies could be quite profitable. (The values of the variables used in this case are displayed in table 37.) The present use of B–dollies in the United States is limited and, from a financial point of view, may be expected to stay that way unless highway regulations are eased in recognition of the improved dynamic ability of the B–dolly.

With regard to the engineering of B-dollies, this economic analysis indicates that the weight of the dolly is a crucial issue. Small changes in productivity have a major influence on operating costs. It appears that reductions in dolly weight might pay for the increases in dolly purchase prices that would accompany the introduction of lighter and stronger materials. (Of course, reduced dolly weight would lead to more productive vehicles whether they employ A- or B-dollies.)

If a market for B-dollies were to develop, improved designs would probably be created. For example, assume that the weight penalty between A- and B-dollies was reduced to approximately 400 lb (181 kg). Under this assumption, a new reference situation, entitled "a lighter B-dolly," has been developed (see tables 38 and 39). The baseline value of the change in operating cost for this reference condition is $0.0075/dolly/mi(1.6 km). Table 40 lists the variations examined with respect to this reference condition. The results, displayed in figure 106, are dominated by the influence of

Table 36. The Hypothetical Situation

Δ costs/benefits between A and B dollies	Year 0	Year 1	Year 2	Year 3	Year 4
Initial cost of dollies	($18,000.00)	$0.00	$0.00	$0.00	$0.00
Converting existing equipment	($3,000.00)	$0.00	$0.00	$0.00	$0.00
Major overhauls	$0.00	($8,670.00)	$0.00	($4,110.00)	$0.00
Tire wear	($810.00)	($810.00)	($810.00)	($810.00)	($810.00)
Preventive maintenance	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)
Scheduling	($800.00)	($294.30)	($108.27)	($39.83)	($14.65)
Training	($6,000.00)	($2,207.28)	($812.01)	($298.72)	($109.89)
Ability to back up	$4,199.83	$4,199.83	$4,199.83	$4,199.83	$4,199.83
Less weight hauled	$0.00	$0.00	$0.00	$0.00	$0.00
Fewer accidents	$4,800.00	$4,800.00	$4,800.00	$4,800.00	$4,800.00
Ability to operate on secondary roads	$18,000.00	$18,000.00	$18,000.00	$18,000.00	$18,000.00
Allow higher GVW	$0.00	$0.00	$0.00	$0.00	$0.00
Total	($4,610.17)	$12,018.25	$22,269.55	$18,741.28	$23,065.29
Net Present Value	$112,325.36				
Cost increase to cover loss /100lb / mile	$0.00E+00				
Change in operating cost / dolly / mile	($0.0212)				

1 lb = 0.454 kg

1 mi = 1.609 km

Table 36. The Hypothetical Situation (Cont.)

Δ costs/benefits between A and B dollies	Year 5	Year 6	Year 7	Year 8	Year 9
Initial cost of dollies	$0.00	$0.00	$0.00	$0.00	$0.00
Converting existing equipment	$0.00	$0.00	$0.00	$0.00	$0.00
Major overhauls	($8,670.00)	$0.00	($4,110.00)	$0.00	($8,670.00)
Tire wear	($810.00)	($810.00)	($810.00)	($810.00)	($810.00)
Preventive maintenance	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)
Scheduling	($5.39)	($1.98)	($0.73)	($0.27)	($0.10)
Training	($40.43)	($14.87)	($5.47)	($2.01)	($0.74)
Ability to back up	$4,199.83	$4,199.83	$4,199.83	$4,199.83	$4,199.83
Less weight hauled	$0.00	$0.00	$0.00	$0.00	$0.00
Fewer accidents	$4,800.00	$4,800.00	$4,800.00	$4,800.00	$4,800.00
Ability to operate on secondary roads	$18,000.00	$18,000.00	$18,000.00	$18,000.00	$18,000.00
Allow higher GVW	$0.00	$0.00	$0.00	$0.00	$0.00
Total	$14,474.01	$23,172.98	$19,073.63	$23,187.55	$14,518.99
Net Present Value					
Cost increase to cover loss /100lb / mile					
Change in operating cost / dolly / mile					

1 lb = 0.454 kg

1 mi = 1.609 km

Table 37. Values of the Variables Used in the Hypothetical Situation

Variable Names	Values
Percentage of trips at max GVW	60 %
Additional dolly weight	1000 lbs
Miles per year per dolly	100,000 miles
Charge/lb/mile for freight hauled	$0.0000894
B-dollies added to the fleet	6 B-dollies
Total number of dollies owned	15 Dollies
Percent of tire wear increase over A-dolly	15 %
Overhaul cost (percentage of initial dolly cost)	20 %
Preventive maintenance - per year	$500.00
Double assembly & disassembly (B-dolly backup)	0.3846 per day
Accident savings per mile per B-dolly	$0.008
Annual discount rate	10 %
Upgrading scheduling programs	$0.00
Administrative Expenses (first year)	$800.00
Driver/yard personnel training (first year per dolly)	$1,000.00
Local deliveries	100 per year
Overweight hauling allowance	1000 lbs

1 lb = 0.454 kg

1 mi = 1.609 km

Table 38. Another Reference Condition: Lighter B-Dolly

Variable Names	Values
Percentage of trips at max GVW	60 %
Additional dolly weight	400 lbs
Miles per year per dolly	100,000 miles
Charge/lb/mile for freight hauled	$0.0000894
B-dollies added to the fleet	6 B-dollies
Total number of dollies owned	15 Dollies
Percent of tire wear increase over A-dolly	15 %
Overhaul cost (percentage of initial dolly cost)	20 %
Preventive maintenance - per year	$500.00
Double assembly & disassembly (B-dolly backup)	1 per day
Accident savings per mile per B-dolly	$0.008
Annual discount rate	10 %
Upgrading scheduling programs	$0.00
Administrative Expenses (first year)	$800.00
Driver/yard personnel training (first year per dolly)	$1,000.00
Local deliveries	0 per year
Overweight hauling allowance	0 lbs

1 lb = 0.454 kg

1 mi = 1.609 km

Table 39. Lighter B-Dolly Results

Δ costs/benefits between A and B dollies	Year 0	Year 1	Year 2	Year 3	Year 4
Initial cost of dollies	($18,000.00)	$0.00	$0.00	$0.00	$0.00
Converting existing equipment	($3,000.00)	$0.00	$0.00	$0.00	$0.00
Major overhauls	$0.00	($8,670.00)	$0.00	($4,110.00)	$0.00
Tire wear	($810.00)	($810.00)	($810.00)	($810.00)	($810.00)
Preventive maintenance	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)
Scheduling	($800.00)	($294.30)	($108.27)	($39.83)	($14.65)
Training	($6,000.00)	($2,207.28)	($812.01)	($298.72)	($109.89)
Ability to back up	$10,920.00	$10,920.00	$10,920.00	$10,920.00	$10,920.00
Less weight hauled	($12,873.60)	($12,873.60)	($12,873.60)	($12,873.60)	($12,873.60)
Fewer accidents	$4,800.00	$4,800.00	$4,800.00	$4,800.00	$4,800.00
Ability to operate on secondary roads	$0.00	$0.00	$0.00	$0.00	$0.00
Allow higher GVW	$0.00	$0.00	$0.00	$0.00	$0.00
Total	($28,763.60)	($12,135.18)	($1,883.88)	($5,412.15)	($1,088.15)
Net Present Value	($60,240.46)				
Cost increase to cover loss /100lb / mile	$2.51E-05				
Change in operating cost / dolly / mile	$0.0075				

1 lb = 0.454 kg

1 mi = 1.609 km

Table 39. Lighter B-Dolly Results (Cont.)

Δ costs/benefits between A and B dollies	Year 5	Year 6	Year 7	Year 8	Year 9
Initial cost of dollies	$0.00	$0.00	$0.00	$0.00	$0.00
Converting existing equipment	$0.00	$0.00	$0.00	$0.00	$0.00
Major overhauls	($8,670.00)	$0.00	($4,110.00)	$0.00	($8,670.00)
Tire wear	($810.00)	($810.00)	($810.00)	($810.00)	($810.00)
Preventive maintenance	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)	($3,000.00)
Scheduling	($5.39)	($1.98)	($0.73)	($0.27)	($0.10)
Training	($40.43)	($14.87)	($5.47)	($2.01)	($0.74)
Ability to back up	$10,920.00	$10,920.00	$10,920.00	$10,920.00	$10,920.00
Less weight hauled	($12,873.60)	($12,873.60)	($12,873.60)	($12,873.60)	($12,873.60)
Fewer accidents	$4,800.00	$4,800.00	$4,800.00	$4,800.00	$4,800.00
Ability to operate on secondary roads	$0.00	$0.00	$0.00	$0.00	$0.00
Allow higher GVW	$0.00	$0.00	$0.00	$0.00	$0.00
Total	($9,679.42)	($980.46)	($5,079.80)	($965.88)	($9,634.44)
Net Present Value					
Cost increase to cover loss /100lb / mile					
Change in operating cost / dolly / mile					

1 lb = 0.454 kg

1 mi = 1.609 km

Table 40. Variations Used in Sensitivity Analysis: Lighter B-Dolly

Variables	Reference Values	Sensitivity Variations
Percentage of trips at max GVW	60%	0%; 100 %
Additional dolly weight	400 lbs	0; 800 lbs
Charge/lb/mile for freight hauled	$0.0000894	$0.0000447; 0.0001341
Percent of tire wear increase over A-dolly	15 %	0%; 30%
Overhaul cost (percentage of initial dolly cost)	20 %	10%; 30%
Preventive maintenance - per year	$500	$0; $1,000
Double assembly & disassembly (B-dolly backup)	1 per day	0; 2
Accident savings per mile per B-dolly	$0.008	$0.00; $0.016
Local deliveries	0 per year	500 per year

1 lb = 0.454 kg

1 mi = 1.609 km

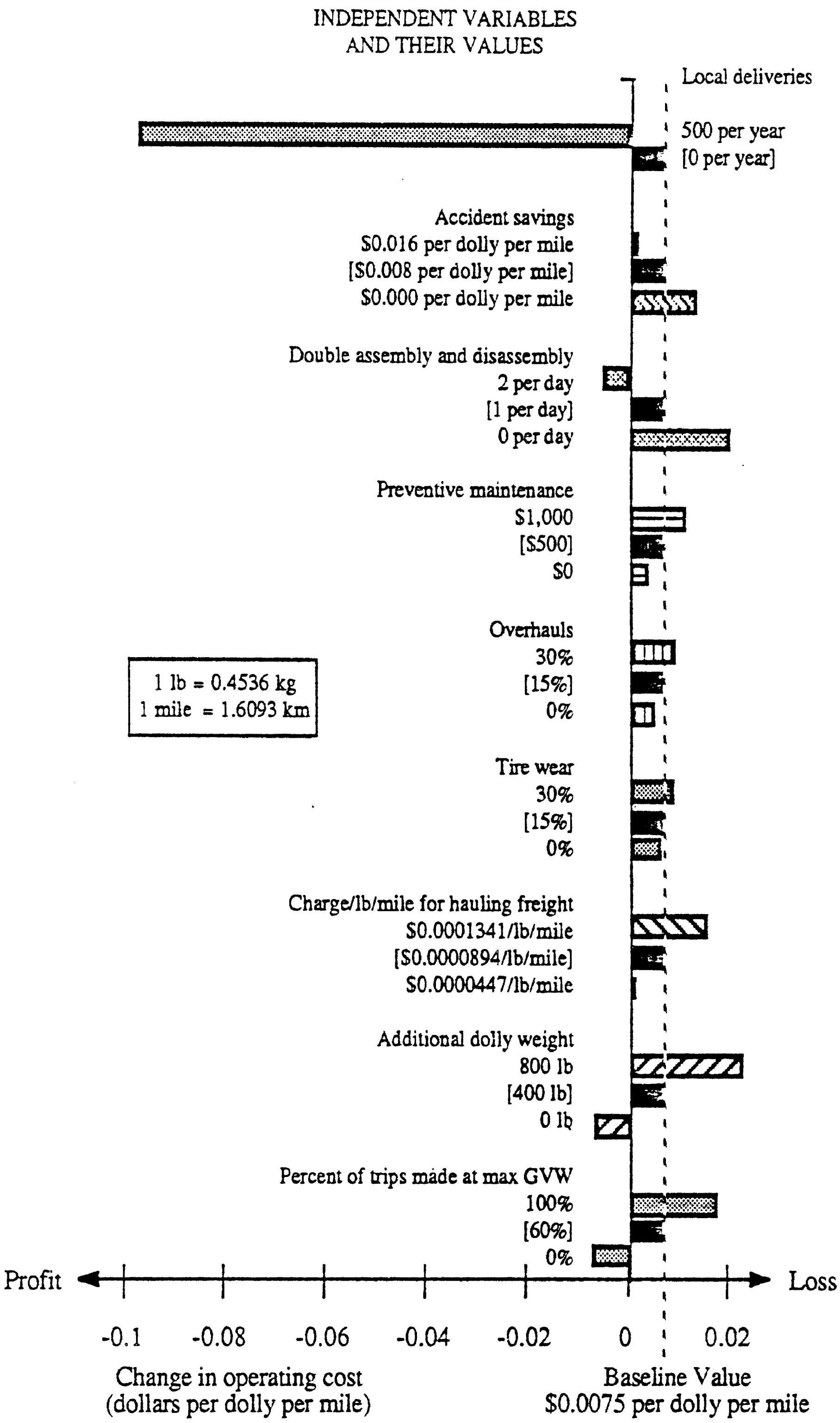


Figure 106. Operating cost sensitivities for a lighter B-dolly.

a large number of local deliveries. Nevertheless, the ability to assemble and disassemble (backup) will result in a profit situation, and fleets that only operate at GVW approximately 30 percent of the time are predicted to show a profit from purchasing B-dollies. This example provides an estimate of the financial situation that might evolve after several years of B-dolly development.

Additional savings would also be realized by fleet operators if they were able to take full advantage of time savings allowed by features characteristic of the B-dolly. The prior financial analyses used conservative estimates of the benefits associated with operations such as assembling and disassembling a set of doubles. In those analyses, it was assumed that the driver saves an average of 20 minutes on this operation by being able to backup a set of doubles equipped with an innovative dolly. With an internal labor rate of $21, the 20 minutes saved benefits the fleet operator by $7. This assumes that both the vehicle and the driver are idle for the period. However, if the driver and the vehicle were put to productive uses for the 20 minutes, such as hauling freight, then the benefits would tend to overshadow the increased costs of operating a B-dolly.

For example, the additional benefits produced from 20 minutes of extra hauling time can be calculated in the following manner. Assuming an average transportation speed of 20 mi/h (including stops, delays, etc.) and a freight hauling charge of $0.0000894 per lb (0.45 kg) per mi (1.6 km), then a fully loaded vehicle would earn an additional $30 per 20-minute period. When this additional productivity is introduced into the reference condition described in table 38 and evaluated in table 39, a significant change in the results is obtained. A "lighter B-dolly" that is assembled and disassembled once a day results in a net profit of $125,000 over the life of the project, as opposed to a loss of $60,240. Referring to figure 106, this change would shift the "Baseline Value" from a loss of $0.0075 to a profit of $0.0231.

As shown by the analyses presented here, the difference between profit and loss in the trucking industry depends primarily on productivity. An increase in productivity can offset increases in the costs of operating a B-dolly double. Time savings and the amount of weight hauled are both key factors affecting the productivity of a trucking fleet. Although the B-dolly causes a decrease in the amount of weight that can be carried, that loss in productivity might be compensated for if fleet operators can find ways to use features of innovative dollies (such as the ability to back up) to save time.

CONCLUSIONS AND RECOMMENDATIONS

1. Dolly Performance and Design Guidelines

The results of the simulation study, and generally supported by the vehicle test program, suggest that it is both reasonable and practical to develop commercial vehicle dollies which can significantly improve the dynamic performance of the multitrailer combination vehicle. Accordingly, a set of reasonable performance and design "guidelines" can be enumerated which define goals for the development of innovative commercial vehicle dollies. The guidelines set forth below apply specifically to the vehicle configuration commonly known as the Western double, in the fully loaded, 80,000-lb (36,320-kg) gvw condition with both trailers having sprung mass c.g. heights of 80 in (2 m) (typical of "medium density" freight). The reference vehicle is shown in figure 19 of this report. Performance expectations would be different for other configurations. In that regard, current understanding suggests that caution should be exercised in applying B-dollies in long-drawbar configurations.

Guidelines for Vehicle Dynamics Performance Properties. Results of the simulation study indicate that innovative dolly designs can achieve substantial improvements in rearward amplification and dynamic rollover threshold without degrading other performance qualities of multitrailer vehicles. The following represent reasonable and practical vehicle performance goals which have been shown to be attainable for the Western double with several innovative dolly types:

- A maximum rearward amplification of less than 1.75 over the usable range of maneuvering frequencies (0 to 4 rad/sec)

- A dynamic rollover threshold (measured by peak lateral acceleration of the tractor in sine-steer maneuvers) of 0.3 g. or greater, over the usable range of maneuvering frequencies

- A minimum effective damping ratio of 0.25 or greater (as determined from the lateral acceleration response of the second trailer in a pulse-steer maneuver)

- A low-speed offtracking performance equal to or improved over that of the Western double equipped with an A-dolly.

Guidelines for Dolly Mechanical Properties. The simulation study indicates that there are several design approaches which can achieve some or all of the above performance goals. These properties are associated with the yaw and roll articulations of the dolly with respect to the first trailer. The general mechanical qualities of merit are as follows:

1) elimination or alteration of the yaw articulation behavior of the dolly relative to the first trailer (for improving rearward amplification performance) by means of one of the following methods:

   - shifting of the dolly steer point substantially forward (at least 100 in (3 m) forward of the typical pintle position) during travel at highway speeds (above approximately 30 mi/h (48 km/h)) through the application of special hitching hardware or appropriately steering the dolly axle as a function of drawbar/articulation angle. To prevent degradation of low-speed offtracking, the steer point may be shifted rearward to the vicinity of the pintle during low-speed maneuvering.

   - providing a mechanism to link the yaw articulation behaviors of the dolly relative to the first trailer and of the dolly relative to the second trailer, thus eliminating one yaw degree of freedom from the vehicle. A mechanism which provides approximately the articulation angle linkage gain of:

     $$G_{\Gamma_2\Gamma_3} = \frac{OH + TL}{WB} \tag{17}$$

     (see figure 44) is desirable since it provides "Ackerman steering" at low speed and good dynamic performance at high speed.

   - eliminating the yaw degree of freedom between the dolly and first trailer, typically through the use of a rigid, double drawbar on the dolly of the B-dolly configuration. To prevent unacceptably high levels of tire scuffing and structural stress in low-speed maneuvering, this may be accompanied by the introduction of "controlled steering" or "self-steering" of the dolly tires. A controlled steering mechanism providing for steering of the dolly tires as a function of the articulation angle between the dolly and the second trailer is desirable. A mechanism which provides approximately the following steering gain (see figure 45) is appropriate since this gain provides for "Ackerman steering" at low speed and good dynamic performance at high speed:

     $$G_{\delta_4\Gamma_3} = \frac{OH + TL}{WB + OH + LT} \tag{18}$$

Self-steering mechanisms require "centering spring" devices (or steering lock) which effectively prevent steering of the dolly wheels in dynamic highway maneuvers. This project has not specifically identified the level of steering resistance required to establish good performance, but a device which prevented steering at lateral tire friction utilization levels of approximately 0.3 was found to provide very good dynamic performance.

2) Connection of the first and second trailers in roll, typically through the use of rigid, double drawbars of the B-dolly configuration. This action proves to be very powerful in improving dynamic rollover threshold directly. It may also be moderately effective in reducing rearward amplification, depending on the properties of the tires installed on the vehicle. The level of rigidity attained in this coupling is critical in determining its effectiveness. A minimum stiffness of 30,000 in-lb (3,390 N-m) per degree of relative roll is desirable.

Guidelines for Worst-Case Static Loading. These guidelines are significant only to dollies of the linked-articulation and B-dolly configurations. These dollies provide new constraints in yaw and/or roll between the dolly and first trailer, thereby introducing significant new loads at the coupling between the dolly and first trailer. (Dollies which effectively relocate or alter the conventional single-point pintle, do not substantially alter hitch loadings.) These loadings, particularly in response to roll, are highly dependent on the rigidity of the dolly structures and the trailer structures, and this study examined a very limited sample in this regard. The simulation study assumed roll stiffness of the B-dolly drawbar coupling of 30,000 in-lb/deg (3,390 N-m/deg), and otherwise effectively rigid dolly and trailer structures. Accordingly, the results from the simulation are expected to be conservative. All the simulation results are from extreme lane-change maneuvers at the rollover threshold of the second trailer. The physical testing employed conventional van trailers, a linked-articulation dolly mechanism with unknown, but certainly significant, compliance, and B-dollies with unknown yaw compliance, but with approximately ±2.5 degrees of roll-coupling lash. Vehicles of differing structural quality could be expected to yield different results. Maximum structural loads in the physical testing all came during low-speed maneuvers specifically designed to stress the couplings.

The worst-case loadings derived from the results of this study are listed in table 41. These values of forces and moments are intended to serve as first-order estimates for guiding the design of innovative dollies.

(The large difference between test measurements and simulation results for M_x is due to a special stress relieving feature of the test dolly. See the discussion presented in

Table 41. Worst-Case Loading Values

	Fy, lb Lateral Force		Mx, in-lb Roll Moment		Mz, in-lb Yaw Moment	
Dolly Type	Simulation	Test	Simulation	Test	Simulation	Test
Linked-Articulation Dollies	4,014	5,800	N.A.	N.A.	253,900	305,600
B-Dollies (SA-60 or PRO)	8,660	6,256	672,400	243,300	659,800	688,900

1 lb = 4.448 N

1 in-lb = 0.113 N-m

connection with table 17. If this stress relieving feature is not included in the design of the dolly, the simulation results provide a first-order estimate of the maximum roll moment.)

Low-speed maneuvering tests that can be performed in a large parking area can be used to introduce these forces and moments into prototype versions of new designs. Repeated applications of these forces and moments could be used to investigate structural fatigue. However, this would not rule out the possibility of other types of fatigue failure.

2. Further Development of Innovative Dollies

The findings of this study indicate that B-dollies are dynamically superior to A-dollies and other types of innovative dollies because of (a) the roll coupling between the leading semitrailer and the dolly, and (b) the possibility of steering the dolly wheels to achieve good trailing fidelity of the last trailer. However, estimates from accident analyses indicate that the safety benefits obtained by employing B-dollies result in a reduction of operating costs of only 0.84 cents per dolly per mile (1.6 km) (due primarily to an anticipated reduction in the number of rollover accidents). This reduction in operating costs due to accident prevention is not predicted to offset other increased costs associated with B-dollies. **As long as productivity is the ruling force, there is not much *economic* incentive to use heavier B-dollies in place of lighter, simpler A-dollies.**

The permit system that exists in the Western Provinces of Canada encourages the use of B-dollies there. Weight allowances and the right to operate on secondary roads are strong economic incentives that promote the use of B-dollies in Canada. The ability to back up and make local deliveries means that the operation of doubles with B-dollies can be very attractive and profitable in certain types of service. In order for B-dollies to become popular in the United States, economic incentives may need to be developed. These incentives might come from (a) reduction, through design or special permission, of the weight penalty associated with B-dollies, (b) extraordinarily unfavorable changes in insurance rates and/or increases in settlements from law suits, thereby increasing the economic importance of safety, or (c) allowance to travel off of the interstate and primary highway system to make deliveries and pick ups.

This study has demonstrated that innovative dollies can improve the dynamic performance of multi-trailer combination vehicles. In addition, fleet owners in Canada indicate that their drivers have greater confidence in the dynamic performance of doubles equipped with B-dollies. Nevertheless, several technical matters have not been studied enough to provide a comprehensive engineering understanding of the phenomena involved.

With regard to the mechanics of combination vehicles employing innovative dollies, the following subjects warrant further investigation:

- the sensitivity of dynamic rollover to the frequency response properties of the roll motions of doubles
- the influences of "dolly steering" rules on trailing fidelity (rearward amplification versus offtracking)
- the fatigue of dolly and trailer structures due to long-term use in normal service.

In the area of accident studies, more information is needed on the operation of doubles--specifically, data on accident costs, accident types (rollovers of the rear trailer only, for example), and exposure (what types of multi-trailer combinations are operated by what types of drivers on what types of roads).

The findings of this study are positive enough with respect to B-dollies to support a recommendation that combinations with new types of dollies be tested and evaluated in practical service in the United States. This evaluation effort would be in addition to a field trial of the prototype dolly that is currently underway in Canada. The evaluation efforts should include an investigation of the time savings, and thereby cost savings, that can be achieved through the ability to back up doubles equipped with B-dollies.

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