Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.

4. Title and Subtitle	
Evaluation Of Innovative Converter Dollies - Volume I Final Technical Report	5. Report Date December, 1993 6. Performing Organization Code
7. Author(s) Winkler, C.B.; Bogard, S.E.; Ervin, R.D.; Horsman, A.; Blower, D.; Mink C.; Karamihas S.	8. Performing Organization Report No. UMTRI-93-47-1
9. Performing Organization Name and Address The University of Michigan Transportation Research Institute 2901 Baxter Road, Ann Arbor, Michigan 48109	10. Work Unit No. (TRAIS) 11. Contract or Grant No. DTFH61-89-C-00081
12. Sponsoring Agency Name and Address Federal Highway Administration 400 Seventh St., S.W., Washington, DC 20590	13. Type of Report and Period Covered Final 14. Sponsoring Agency Code

15. Supplementary Notes

Volume I - Final Technical Report — COTR: Mr. Dennis Miller
Volume II - Appendices
Volume III - Technical Summary

16. Abstract

An extensive study of the dynamic performance of multitrailer vehicles, and the influence of double-drawbar dollies (C-dollies) on that performance is reported. Six vehicle configurations (five double-trailer combinations and one triple) are considered. The performance of the six vehicles is examined using a matrix of seven different converter dollies (an A-dolly and 6 C-dollies) and 15 different vehicle parametric variations (e.g., center-of-gravity height, tire-cornering stiffness, roll stiffness, etc.). The performance quality of the vehicles is judged using measures such as rearward amplification, yaw-damping ratio, static rollover stability, offtracking, and dynamic-load-transfer ratio.

The results from over 2800 computer simulation runs are used in a statistical regression analysis to produce simple methods for predicting performance numerics for A-trains based on vehicle parameters easily obtained in the field. Performance improvement factors for C-dollies are also developed. Recommendations for minimum performance standards and for C-dolly specifications are also reported.

An economic analysis comparing A-dollies and C-dollies is presented. This analysis is based on data from a field survey and the literature and includes purchase, start-up, operational, and accident cost considerations.

The report also includes the ancillary performance issue of backing ability.

Extensive appendices are included in Vol II. Vol III is a Technical Summary.

17. Key Words	18. Distribution Statement
A-Dolly, C-Dolly, Self-Steering, Controlled-Steering, stability, control,	No restrictions. Available through the National Technical Information Service, Springfield, VA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
		108	

Evaluation of Innovative Converter Dollies:

Volume I

Final Technical Report

Contract No. DTFH61-89-C-00081

Submitted to:

U.S. Department of Transportation

Federal Highway Administration

By:

The University of Michigan
Transportation Research Institute

2901 Baxter Road
Ann Arbor, Michigan 48109-2150

December 1993

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INTRODUCTION

This document constitutes the final report to the Federal Highway Administration under contract number DTFH61-89-C-00081, *Evaluation of Innovative Converter Dollies*. This work follows from prior research and field experience that addresses the stability and control of multitrailer truck combinations. In particular, this report addresses the manner in which the mechanism for coupling rear-placed trailers influences the dynamic behavior of the overall combination vehicle. Commercially, innovative coupling mechanisms (i.e., converter dollies) are an important issue in truck transportation since they may allow longer vehicle combinations. However, greater vehicle lengths also impact vehicle handling and stability and hence pose concerns regarding public safety.

Growth of the number of multitrailer combinations took a decided upward turn when the Surface Transportation Assistance Act of 1982 was signed into law, preempting state prohibitions against double-trailer combinations and opening a nationwide road network for their usage. The newly granted access to a nationwide grid of designated highways, combined with the inherent attractiveness of such vehicles—especially in the so-called less-than-truckload (LTL) type of freight-hauling operation—has led to a large increase in the number of double-trailer vehicles in use throughout the U.S. At the same time, there is increasing pressure to allow the use of triple-trailer versions of the same equipment. Most of this demand derives from the productivity, labor, and fuel advantages of employing a third trailer. The addition of a third trailer effects an approximate 50 percent reduction in shipping costs below those incurred with a doubles combination having the same basic trailer units. Triple-trailer vehicles, in one form or another, may now operate legally in many states.

Since conventional doubles combinations, and especially triples, tend to suffer from special dynamic characteristics that reduce their stability and emergency maneuvering ability below that achieved by tractor-semitrailers, there is concern over the safety effects if such vehicles proliferate across the U.S. As countermeasures to the dynamic deficiencies noted in multitrailer combinations, certain types of innovative dollies and hitching techniques have been consistently highlighted in research aimed at uncovering potential safety improvements. In a prior FHWA-sponsored study, entitled *Improving the Dynamic Performance of Multi-trailer Vehicles: A Study of Innovative Dollies*, the University of Michigan Transportation Research Institute (UMTRI) identified a surprisingly large variety of innovative dolly designs that tended to improve the dynamic performance (i.e., the roll stability and control characteristics) of such vehicles.[1][1] This work, reported in references

[1] Numbers in brackets refer to references listed at the end of this report.

1, 2 and 3, clearly indicates that improved dynamic performance of multitrailer vehicles is feasible within reasonably practical scenarios of hardware design and in-field usage.

Noting that innovative dollies are attractive for attaining high levels of dynamic performance, *this study has sought to identify a useful set of functional specifications for such dollies, based upon a broad quantification of performance.* The challenge of this dolly-specification task lies in the fact that the commercial vehicle fleet includes such a large variety of multitrailer vehicles, each subject to a similarly large variety of maneuvering conditions, loadings, and operating requirements. As an outcome, dolly specifications that are both necessary and sufficient for one vehicle combination and operating condition may differ substantially from those needed for another vehicle.

Recognizing this situation at the outset, this study was structured to establish a *method* for choosing an appropriate dolly for use with a given vehicle, rather than establishing a single, completely general dolly specification. Further, the method was to be practical in that it should be usable by people in the trucking community who are not familiar with vehicle dynamics analysis methods. The approach taken to accomplish this can be broken down into the following tasks:

- Establish a set of relevant (i.e., influenced by dolly properties) vehicle performance measures and related minimum vehicle performance goals.
- Establish a *simple* means for predicting these performance measures for specific multitrailer vehicles when equipped with conventional dollies.
- Establish a simple means for predicting the improvement in the performance measures attainable with innovative dollies based on relevant specifications of the dolly.

The existence of these three elements would allow people in trucking to both establish warrants for the use of innovative dollies and specify dollies appropriate to their vehicles and performance needs.

In this study, performance was assessed by means of computer simulations, using simulation models that had been previously validated against full scale tests with instrumented truck combinations. The goal was to condense the findings from a large simulation study into very simple formulations, which could be used by people in trucking. Similar to the observation made above regarding the basic task of the study, the challenge of the simulation study also lies in the fact that a substantial variety of vehicle configurations must be considered, each with a variety of component design parameter variations, and each under a variety of maneuvering and loading conditions. When a broad set of dolly parameters is also considered, an enormous matrix of cases emerges. Thus, this report is primarily composed of information associated with the computerized analysis. Although an extensive set of appended material serves to document the computed results, the report has been designed to give the general reader a complete review of the methods and findings deriving from each stage of the analysis.

The bulk of the report that follows is contained in the next section, *Presentation of the Study Method and Results*. In that section, we review the technical details of the simulation study and discuss specific findings related to the dynamic performance of multitrailer vehicles using both conventional or innovative dollies. This section concludes with the analysis and discussion of the economic burdens and potential safety benefits of employing innovative dollies in practice. The final section of the text of this report presents the *Summary of the Research Findings* and *Conclusions Pertaining to Dolly Specifications*. A separate volume contains appendices A through G. These present plotted and tabular results, plus background discussions that provide the rational basis for various aspects of the computerized analyses as well as condensed forms of the numerical results.

Finally, throughout this report, contrast is drawn between A-dollies and C-dollies and between A-trains and C-trains. These dollies, shown in figure 1, are pieces of equipment that serve to couple one semitrailer to the next in the multitrailer combination. An A-train is a multitrailer vehicle made up using A-dollies. Similarly, a C-train uses C-dollies. The A-dolly represents conventional practice in the U.S. and is a mechanically simpler device than the innovative C-dolly.

The defining difference between the two styles of dolly is the configuration of the drawbar and leading hitch. The A-dolly employs a single-point hitch (typically, a pintle hitch) that allows free yaw, pitch, and roll articulation between the dolly and the unit that tows it. In contrast, the C-dolly connects to its towing trailer with two rigid drawbars and a pair of pintle hitches. This arrangement eliminates the yaw and roll freedom at the hitch point between the lead trailer and dolly. More discussion of the distinctive features of dollies is presented later in the text.

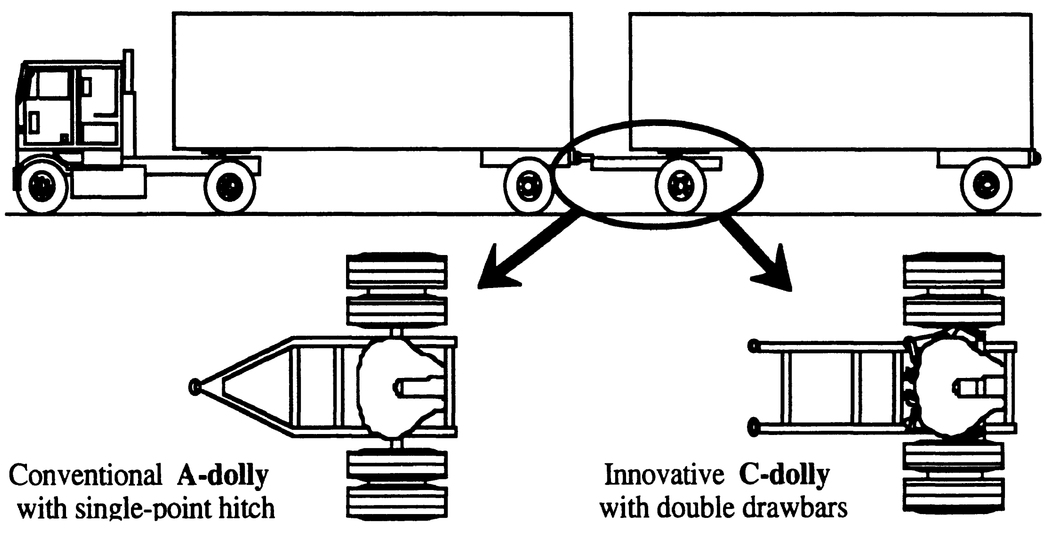


Figure 1. The two styles of converter dollies

PRESENTATION OF THE STUDY METHOD AND RESULTS

In this section, the research effort is presented in capsule form, as a condensed account of the material appearing in greater detail in the appendices. Each of five main subjects is discussed from the viewpoint of the methodology employed and the results obtained. The subjects are as follows:

- *Vehicle Performance Measures.* This section presents a series of vehicle performance measures that are seen to be both (1) significant with respect to the safety qualities of the complete multitrailer combination vehicle and (2) relevant to this study in that dolly properties can reasonably be expected to influence these measures. Along with presenting the rationale for the choice of measures, the section presents arguments for specific (numerical) performance goals.

- *Dolly Properties and Characteristics.* This sections identifies the generic dolly properties that are expected to have a substantial influence on the vehicle performance measures. This serves to both (1) establish the potential elements that a dolly specification might include and (2) provide a partial basis for defining the dimensions of the simulation study matrix.

- *Elements of the Simulation Study.* This section presents all the elements of the simulation study. The discussion touches on the baseline vehicles and their parameter variations, the dollies, the test maneuvers, the final structure of the complete simulation matrix, and the presentation of the parameter sensitivity results.

- *Generalized Assessments of Vehicle Performance.* The large mass of vehicle performance data gathered in the simulation study is condensed into (1) simplified predictors of the performance of A-trains and (2) generalized characterizations of the performance difference of A-trains and C-trains. This section also presents certain ancillary technical issues, addressing a few miscellaneous aspects of performance that are important in their own right, but tend to fall outside of the conventional assessment of stability and control quality.

- *Economic Analysis.* The economics of using innovative dollies in the business of trucking is examined. Potential cost reductions associated with reduced accidents and other factors, as well as the costs increases arising from the purchase, maintenance, and operation of the innovative equipment, are considered.

In the presentation that follows, reference is made to appendices whose contents form the basis for the discussion. In each case where appended material exists, a sample of the material is discussed so that the reader can assimilate the nature and relative level of significance of data that underlies the findings and conclusions from this research.

VEHICLE PERFORMANCE MEASURES

The performance measures used to evaluate the dynamic characteristics of multiple trailer vehicles are listed in table 1. The measures were selected based on the premise that they are of primary or secondary importance when measuring the dynamic performance characteristics of A- or C-dollies. Several of the measures and the maneuvers used to generate them were developed in earlier studies performed for the Roads and Transportation Association of Canada (RTAC).[5] In the sections that follow, each of these measures is discussed in the context of its significance as related to dolly performance. The simulation details and specifications used in their derivation can be found in appendix A.

Table 1. Vehicle performance measures for multiple trailer vehicles

Performance Measures
Static Rollover Threshold
Rearward Amplification
Dynamic Roll Stability
High-Speed Transient Offtracking
Yaw-Damping Ratio
Low-Speed Offtracking
High-Speed Steady-State Offtracking

Static Roll Stability

A common measure of static roll stability is the static rollover threshold. It is defined as the maximum level of lateral acceleration that the vehicle can sustain in a steady turn without rolling over. The static rollover threshold is an RTAC performance measure and is an important safety criterion in the design of commercial vehicles.[5]

Static roll stability and the vehicle properties that influence it are well understood. Center-of-gravity height and track width are the most important vehicle properties involved in determining rollover threshold. Compliances in tires, suspensions, and other components are also important as are the kinematic properties of suspensions. Because the dolly possesses some of these elements, it too plays a significant role in establishing the rollover threshold of the vehicle. The role of the dolly implied here, however, is similar for A-dollies and C-dollies.

The defining property of the C-dolly—the double drawbar hitch—can also play a role in establishing rollover threshold. The single pintle hitch of an A-dolly acts to decouple the A-train in roll. That is, with an A-dolly, the portion of the vehicle that is forward of the pintle, the tractor semitrailer, rolls independently of the portion that is aft of the pintle, the full trailer. Each independent roll unit has its own static rollover threshold. If these thresholds are different, we generally characterize the stability of the whole vehicle by the least stable unit.

When a C-dolly is used, the two roll units are no longer independent. The double drawbar coupling requires that they roll more or less in unison, depending on the roll stiffness of the dolly frame. If, with an A-dolly, the two roll units would have different static rollover thresholds, the threshold of the entire vehicle with a C-dolly will fall somewhere between these limits. Thus, the roll-coupling property of the C-dolly is seen to raise the static rollover threshold of such a vehicle.

But this improvement in stability only comes about if the two units have different thresholds in the A-train configuration. In many doubles operations, loading and other properties of the two roll units are so similar that the rollover thresholds do not vary significantly. In this case, the roll coupling provided by the C-dolly has no influence on static stability. This is generally the case for the vehicle configurations examined in this study. (It should be noted, however, that the roll coupling provided by C-dollies does play a significant role in the *dynamic* roll stability measure used in this study.)

Static rollover threshold is measured by means of simulating a steady turn condition using the RTAC-A maneuver. This maneuver includes a slowly, but steadily increasing spiral turn. The increasing lateral acceleration eventually causes the vehicle to roll over. In this study, capturing the instant of rollover was done by calculating and comparing the stabilizing and destabilizing moments experienced by each roll unit of the vehicle. The details of this maneuver and calculation can be found in appendix A.

Rearward Amplification

The rearward amplification phenomenon, and the specific manner in which it has come to be measured, is illustrated in figure 2. The upper portion of the figure shows the paths of the tractor and second trailer of a double as they may develop during a rapid evasive maneuver. The lower section illustrates the resulting time history of the lateral acceleration of the tractor and second trailer. The amplified nature of the trailer response is evident. The amount of rearward amplification is measured by the ratio of peak values of trailer and tractor lateral acceleration. For an ideal vehicle, rearward amplification ratio would be unity, implying that the second trailer would experience the same severity of motion as the

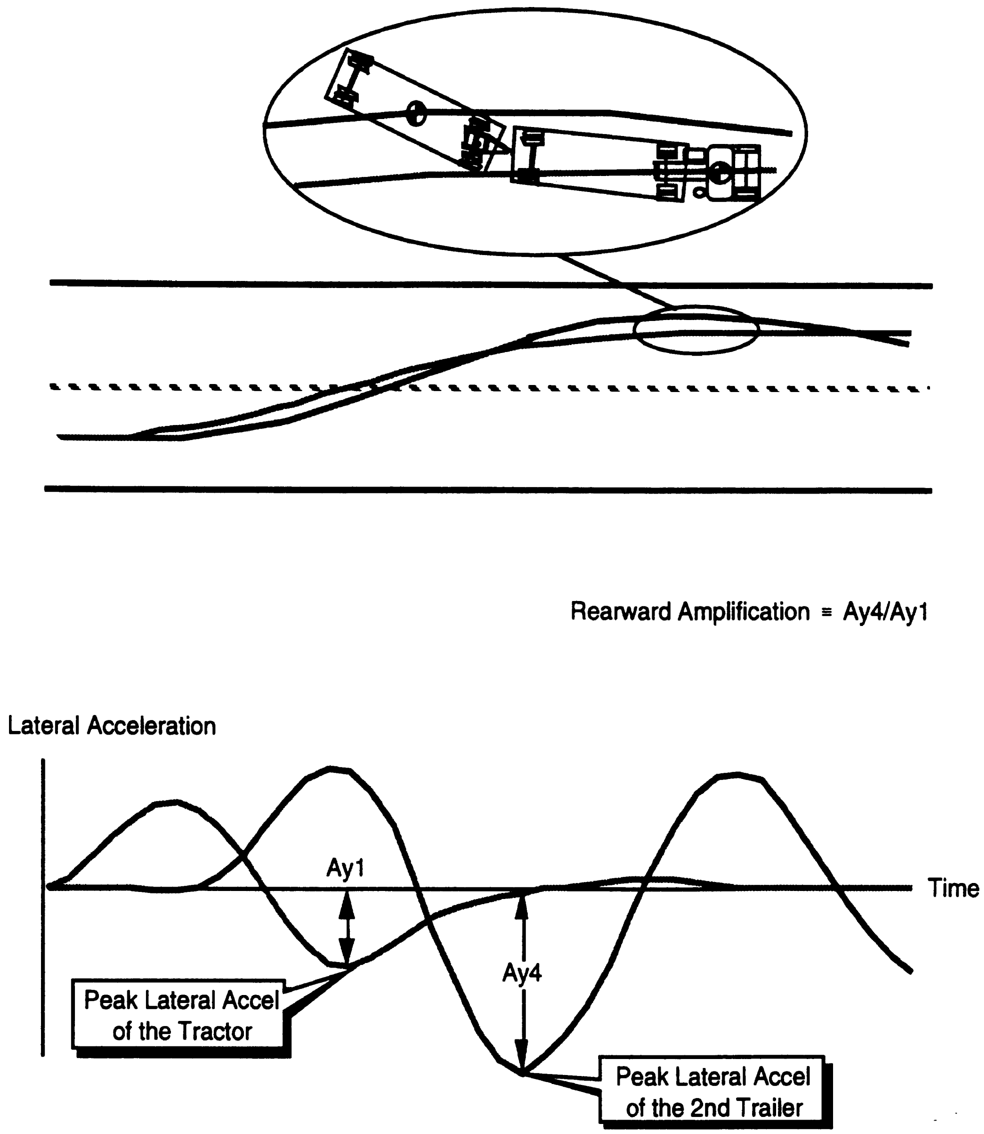


Figure 2. Illustration of the rearward amplification phenomenon

tractor and, given proper phasing, would travel along the same path as the tractor. However, in practice the rearward amplification resulting from a rapid evasive maneuver is greater than unity because the trailers of the vehicle combination typically exhibit more severe lateral motions than the tractor.

The properties of the dolly and its hitches play a major roll in determining the rearward amplification performance of multitrailer vehicles. Fancher's linear analyses [4] indicate that, for vehicles using A-dollies, tire-cornering stiffness, tire loads, trailer wheelbases, and location of the steer point are the key parameters that determine the severity of rearward amplification. The single one of these four important parameters that can be most directly influenced by A-dolly design is the location of the steer point. Innovative adaptations of the A-dolly that result in a more forward location of the steer point can reduce rearward amplification.

While effective A-dolly and pintle hitch design can mitigate the effects of rearward amplification, the most powerful mechanism discovered to date for reducing rearward amplification is the elimination of yaw articulation between the dolly and the lead trailer—the B-train and C-dolly concept. Here the term "B-train" is used to classify a group of combination vehicles in which the rear of the lead trailer is rigidly extended and outfitted with a fifth-wheel for the pup trailer (there is no converter dolly in a B-train combination). In the U.S., B-trains typically employ flatbed type trailers and are used in heavy load applications and situations where trailer interchangability is not important.

The conventional means of measuring rearward amplification is the so-called RTAC-B maneuver. The specifications of this evasive maneuver were established by Ervin [5] and are described in appendix A. Because rearward amplification is known to be a frequency dependent phenomena, it is measured in each of three versions of the maneuver covering the significant range of steering frequencies (i.e., periods of 2.0, 2.5, and 3.0 seconds). The results of each maneuver are compared and the largest (worst) rearward amplification ratio is reported for that vehicle.

Dynamic Roll Stability

Dolly design can have a very powerful influence on the dynamic roll stability of multitrailer vehicles. Dynamic roll stability refers to the resistance to rollover in dynamic maneuvering situations. It is directly related to the combined properties of rearward amplification and basic static roll stability. That is, rearward amplification generates exaggerated levels of lateral acceleration at the last trailer during dynamic maneuvering, and those higher levels of lateral acceleration challenge the basic roll stability of the unit.

The tendency toward dynamic rollover of the rear trailer can, therefore, be reduced by reducing rearward amplification. But the tendency toward dynamic rollover of the trailer can also be reduced by enhancing the rear trailer's ability to survive high levels of lateral acceleration without rolling over. The roll coupling that results from the C-dolly is particularly effective at enhancing the resistance to dynamic rollover. [1,5]

The RTAC performance measure known as the dynamic-load-transfer ratio (DLTR) [5] is a measure of dynamic roll instability that occurs during an aggressive dynamic maneuver. Specifically, DLTR is a measure of the peak side-to-side load transfer that occurs during a dynamic maneuver. The measure ranges from 0.0, when the unit is at rest, to 1.0 when a complete load transfer has occurred (i.e., all the tires on one side of a roll unit have lifted off the ground). Like the static-rollover-stability calculation, DLTR is computed for each roll unit of a combination vehicle. The measure is calculated using the results from the three obstacle-avoidance maneuvers used to measure rearward amplification (i.e., the RTAC-B maneuvers). The maximum value for all roll units that occurred during the three maneuvers is the DLTR measure.

The following function is used to calculate the DLTR for each roll unit and at each instant (each time step of a simulation) during the maneuver:

$$\left|\sum_{i=m}^{n} (F_{Li} - F_{Ri})\right| \Bigg/ \left(\sum_{i=m}^{n} (F_{Li} + F_{Ri})\right) \tag{1}$$

Where:

F_{Li}	is the vertical load on the left-side tires of axle i
F_{Ri}	is the vertical load on the right-side tires of axle i
m	is the first axle of the roll unit
n	is the last axle of the roll unit

(Note: The steering axle of the tractor is omitted from the calculation due to its comparatively soft suspension. For details of the underlying rationale, see [5].)

High-Speed Transient Offtracking

High-speed transient offtracking is a safety concern during transient maneuvers of multitrailer vehicles. The upper portion of figure 2 illustrates the nature of this event. The exaggerated motions of the second trailer result in significant overshoot of the rear units of a multitrailer vehicle relative to the path followed by the tractor. This can result in an *intrusion* of the rear trailer into adjacent traffic lanes.

Particularly for A-train doubles, large values of high-speed, transient offtracking can and do result directly from large values of rearward amplification. However, large values of high-speed, transient offtracking can also be exhibited by vehicles with low levels of rearward amplification. This behavior is characteristic of C-dolly designs that provide insufficient steer-centering force and, therefore, insufficient tire side forces. The condition is also intensified if the dolly tongue is long.

During a rapid evasive maneuver the insufficient side force at the dolly tires results in a sluggish response of the trailing trailer. Although this may seem beneficial in the early stages of an evasive maneuver (i.e., the trailer may not experience the excessive motions of the dolly) it develops into a problem as the maneuver continues. Because the trailer response is slow, its recovery is also slow. This leads to the possibility of the trailer significantly overshooting the lane boundaries. In some situations, the trailers may overshoot repeatedly at the end of the maneuver, indicating a low level of yaw damping.

The RTAC performance measure for high-speed transient offtracking is the largest axle overshoot relative to the path of the tractor front axle as determined from the three RTAC-B lane-change maneuvers used for measuring dynamic roll stability and rearward amplification. The largest offtracking value from the three maneuvers characterizes this performance measure for a given vehicle.

Yaw Damping

Damping characteristics are a fundamental measure of stability in virtually all dynamic systems. High levels of positive damping result in stable systems. Low levels of damping produce marginally stable systems that may exhibit highly oscillatory behavior. Negative damping, of course, results in an unstable system whose response *goes to infinity* at the slightest provocation.

A-trains normally have adequate levels of yaw damping. However, A-trains whose trailers have very short wheelbases in relation to their yaw moment of inertia, and/or with unusual rearward load bias may exhibit negative damping at high speed. The *Michigan petroleum double* was an example of a vehicle with this characteristic. [6,7]

Low levels of yaw damping can be a problem with poorly designed C-dollies. Inadequate levels of steering resistance and/or excessive tongue length can lead to a condition in which the dolly tires are unable to generate significant lateral forces. This can result in poor damping and/or sluggish response in dynamic conditions and in excessive high-speed, transient and steady-state offtracking.

Low or negative yaw damping is undesirable in a multitrailer vehicle system due to the basic fact that such a vehicle may exhibit excessive or sustained oscillatory motions of the rear trailer even with little or no excitation at the tractor. These motions can result in lane intrusion or vehicle rollover. Yaw damping can be measured by observing the rate at which trailer lateral motion dies out (or grows) after a brief, minor disturbance input. This study uses two maneuvers to evaluate yaw damping. By extending the simulation time of the RTAC-B maneuver, sufficient data are produced to allow the damping qualities of the vehicle to be measured. Yaw damping is also measured using a second maneuver, called a pulse steer. This maneuver, conducted at 62 mph (100 kph) constant forward velocity, consisted of a 2 degree (road-wheel) steering pulse maintained for 0.2 second duration followed by 5 seconds of zero steer. [1]

Low-Speed Offtracking

Low-speed offtracking refers to the tendency of the rear axle of the vehicle to track inboard of front axles during low speed maneuvering. Low-speed offtracking is certainly not a *dynamic* performance quality, but it is a characteristic of commercial vehicle behavior that is of some importance to operational and safety performance. The more common dolly designs generally have little influence on low-speed offtracking behavior. However, this general observation can be, and is, violated by some exceptional designs.

As reported in the literature on many occasions (for example, [8]) the amount of low-speed offtracking is strongly influenced by the vehicle length and the number of articulation joints. A good approximation of low-speed offtracking is:

$$OT_{V=0} = R_1 - \sqrt{R_1^2 - \sum_{i=1}^{j} WB_i^2 + \sum_{i=1}^{k} OH_i^2} \qquad (2)$$

Where:

$OT_{V=0}$	is the low-speed offtracking
R_1	is the radius of the turn at the front axle
WB_i	is the i^{th} wheelbase of the vehicle (including tractor, each semitrailer, and each dolly)
OH_i	is the i^{th} hitch overhang of the vehicle (distance from a rear axle of a unit rearward to the rear articulation joint of that unit)
j	is the number of units (including dollies)
k	is the number of articulation joints

Since wheelbases are much longer than hitch-point overhangs (fifth wheel overhangs are generally 0 or slightly negative, and pintle overhangs are only a few feet), the dominant term in equation (2) involving a vehicle property is the sum of the squares of the wheelbases. Equation (2) illustrates that increasing the length of a vehicle increases low-speed offtracking by tending to increase the sum of the squares of the wheelbases. By adding articulation joints to a vehicle of fixed length (e.g., changing a long single to a double) the offtracking is decreased by reducing the sum of the squares of the wheelbases.

Low-speed offtracking is an RTAC performance measure. It is evaluated using the RTAC-C maneuver, which consists of a low-speed turn of 90 degrees with a 32.15 foot (9.8 m) radius as measured from the tractor front axle. Typically, in a turn of this type, steady-state offtracking is not fully achieved. Offtracking response develops in a transient manner as the vehicle leaves a straight-line path and enters a turn. In a sharp, 90 degree turn, the tractor will get back onto a straight-line path before the steady-state offtracking pattern is established. Thus, as a practical matter, low-speed offtracking is typically less than predicted by equation (2). The RTAC measure takes the maximum offtracking achieved in this transient situation as the low-speed offtracking measure.

High-Speed Steady-State Offtracking

The fact that the trailers of combination vehicles track inboard of the tractor during low speed maneuvering is broadly recognized. Fewer individuals realize that, during high speed cornering, the trailers of combination vehicles may track outboard of the tractor. When high-speed offtracking is large, it represents a potential safety problem through intrusion into adjacent lanes.

The actual amount of outboard offtracking in high-speed cornering depends on the level of tire slip angle (and, therefore, on tire properties, tire loads, and the level of lateral acceleration) and on the significant length parameters of the vehicle. In fact, it has been shown [11] that if tire loading and cornering stiffness are reasonably uniform from axle to axle then a good approximation of high-speed offtracking is given by:

$$OT = OT_{V=0} - a_y \bullet {}^{F_Z}\!/_{C_\alpha} \bullet \sum_i L_i \tag{3}$$

Where:

OT	is offtracking (inboard direction is positive)
$OT_{V=0}$	is low-speed offtracking
a_y	is lateral acceleration
F_Z	is tire vertical load
C_α	is tire-cornering stiffness
L_i	is the wheelbases of the vehicle

Equation (3) indicates that high-speed offtracking will increase (OT becomes more negative) with decreasing low-speed offtracking, decreasing tire stiffnesses, and increasing vehicle length.

High-speed steady-state offtracking can be a significant, safety-related quality of multitrailer vehicle systems. In the case of the normal A-train, it is dominated by trailer length properties. The design properties of the dolly generally cannot cause a major improvement in this performance property. However, poorly designed C-dollies can result in excessive high-speed steady-state offtracking. This is a performance property of interest for classifying multitrailer vehicles but is of primary importance in specifying dolly types. The RTAC measure for high-speed steady-state offtracking is the measured offtracking during a steady 0.2 g turn at 62 mph (100 kph) constant forward velocity.

DOLLY PROPERTIES AND CHARACTERISTICS

In this section, a select set of important dolly properties and operating characteristics are discussed. This presentation is complemented by an extensive discussion in appendix B, where dolly properties are given numerical values and compared to a baseline condition for all of the parameter variations conducted in this project.

Since it is our ultimate goal to determine the necessary and sufficient dolly properties to ensure acceptable dynamic performance in the multitrailer combination, each dolly property that *deserves special scrutiny* must be linked to some hypothesis that declares the apparent pertinence of the property. We also note that dolly properties warranting attention here fall into two categories, viz., those that might be seen as *mandatory* from a dynamic performance point of view and those that are appraised as *desirable* for the sake of economic (cost and productivity) value rather than for the sake of vehicle performance. In each subsection that follows, individual parameters describing the dolly geometry, construction, and mechanical properties are discussed in turn.

Tongue Length

The tongue length of the dolly is defined as the longitudinal distance from the pintle hitch to the dolly axle. It is also often referred to as the dolly wheelbase. In the case of the A-dolly, this parameter has long been recognized as the leading cause of low-speed offtracking performance. It does not, however, have a major influence on rearward amplification.

For C-dollies, the tongue length effectively adds to the *rear overhang* of the body structure of a lead trailer, thus determining the lever arm at which dolly tire side forces act. When the C-dolly's tongue length is considered, together with the dolly-steering properties that determine the magnitude of dolly tire side force that is generated in a given maneuver, greater values of tongue length tend to degrade high-speed offtracking and damping ratio measures. The significance of the degradation is then determined by the relative size of the loads carried by the fixed axles of the lead trailer and the dolly axles.

Overall Track Width

Overall track width is a basic parameter in determining the roll stability potential of any vehicle. While a greater track width is better from the viewpoint of stability, a value of 102 inches (2.6 meters) is the widest track allowed under current U.S. law.

Hitch Position—Height

As regards the A-dolly, unpublished work has shown that rearward amplification can be reduced by lowering vertical height of the pintle hitch. Presumably the mechanism involved is that roll motions of the lead trailer add to the lateral motion of the pintle hitch. By setting the height of the pintle hitch at the roll center of the lead trailer's suspension the magnitude of lateral motion transferred from the trailer to the pintle hitch will be reduced, leading to less amplification.

For C-dollies, hitch heights are not seen as particularly significant to performance, but consistent hitch height is obviously necessary to assure the interconnectability of trailers and dollies.

Hitch Position—Lateral Spacing

Again, consistent lateral spacing for C-dolly hitches is important as a logistical matter for dolly/trailer interchangeability. For strength and stiffness, it is also important to position the hitches close to the lateral positions of the frame rails of a typical trailer. Once set, the lateral spacing of C-dolly pintle hitches is instrumental for establishing the strength and stiffness properties of the trailer rear, hitches, and dolly frame.

Effective Roll Compliance

This property refers to the combined influence on roll stability of the dolly suspension roll compliance and roll center height, and any compliance of the dolly structure between the fifth wheel and suspension. The effective roll compliance of the dolly, applying to the forward end of a full trailer, must be balanced with the rear suspension compliance on the same trailer so roll moments will be well-distributed front to rear. Research has shown that roll stability is maximum when roll compliances are uniformly proportioned to axle loads carried front and rear.

Hitch and Frame Strength

Obviously, the hitch and frame members of a dolly must be sufficiently strong so that they ultimately withstand the maximum loads that will arise under the most severe maneuvering conditions. Parameters that serve to define hitch and frame structural strength involve minimum values of longitudinal, vertical, and lateral loading that must be sustained at the hitch positions without yielding. With A-dollies, the loading component that is typically of concern is the longitudinal load during braking. Vertical and lateral loads are typically much smaller than longitudinal loads and are of little interest once the longitudinal requirement is satisfied.

The C-dolly, on the other hand, experiences significant hitch loads in all three directions. Longitudinal loads are produced from both towing forces and the yaw moment across the hitch points generated by the side force of the tires and lateral reactions through the dolly's fifth wheel. Vertical loads derive both from dolly pitch moments arising during braking and from a roll coupling (twisting). Roll coupling occurs when the lead and trail trailers assume different roll angles. Lateral loads are borne at the C-dolly hitches according to the summation of the dolly tire side forces plus any lateral force transferred to the dolly through the fifth wheel. Most of these load components can appear simultaneously, thus the hardware must be strong enough to withstand any combination of these loads acting at one time.

Trailer-to-Trailer Roll Stiffness

The roll (or torsional) stiffness of a dolly frame will determine the extent of *windup* between two successive trailers in roll, as one tends to *lean on* the other during a strong transient steering maneuver of the combination. One can imagine defining the property of interest by a *test* in which a dolly, connected to a pair of pintle hitches mounted on a suitable loading frame, is then subjected to a roll moment applied through another loading frame coupled to the dolly's fifth wheel. For a given applied roll moment, the torsional stiffness value would determine the relative angular deflection appearing between the two frames (or simulated trailers).

Tire-Cornering Compliance

The cornering compliance of tires installed at the dolly axle, and all other axles of the vehicle combination, represents a property that is important in every dimension of vehicle handling performance. The property indicates the magnitude of the slip angle at which the tire must operate to generate a value of side force equal to the axle load that is being carried. Generally, lower values of cornering compliance (i.e., a higher value of cornering stiffness) are better for stability and control.

Suspension Roll Steer Coefficient

Suspension roll steer refers to a property that defines the extent of induced steer in the dolly axle by a given value of roll in the dolly frame. It is known that roll steer of the proper polarity can have the same beneficial effect as a corresponding decrease in tire cornering compliance. Although this parameter can influence rearward amplification in A-trains, the impact of roll steer coefficient at the dolly axle, alone—or at any one of the other individual axle—is relatively small. Modification of roll-steer coefficient at all trailer and dolly axles, on the other hand, could be a significant method of reducing rearward amplification.

Dolly-Steering System Characteristic

The axle of a C-dolly may be either *self steered* in the sense of a constrained castering response, or *controlled steered* in the sense of kinematically linked steering as a function of the inter-trailer articulation angle. Although the two mechanisms are inherently different from one another, the design of each will strongly influence the yaw behavior of the combination vehicle, especially for yaw-damping properties and high-speed offtracking and, to a lesser degree, rearward amplification. Each mechanism will be introduced separately below.

The Self-Steering C-Dolly

The tires of the C-dolly axle may be self-steering by means of a caster geometry that is constrained by a centering mechanism. Figure 3 is an illustration of a self-steering C-dolly. Typically, side forces at the dolly tires must exceed a certain fraction of the rated axle load of the dolly before a significant steer response is generated. At higher levels of tire side force, the dolly wheels steer freely to a greater steer angle, but return to nearly zero steer before the side force reduces to zero, thus assuring *centering* for normal operation. The self-steering mechanism must also resist steering in response to imbalanced braking forces, right and left, up to some defined value of brake imbalance. It is also common that the self-steering mechanism employ a locking device that can be engaged by the driver from inside the cab of the tractor. The requirement for a center lock is to (1) allow for backing, and (2)

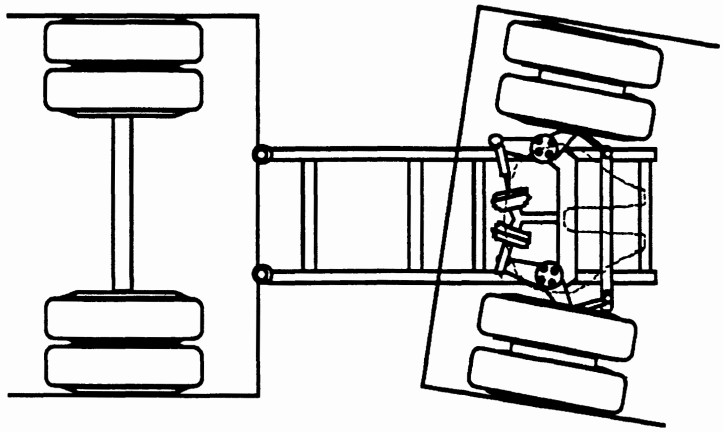


Figure 3. Illustration of a self-steering C-dolly

provide an *emergency* and/or *poor road conditions* operating mode in which the self-steering axle is essentially converted to a nonsteering axle.

The Controlled-Steering C-Dolly

The controlled-steering C-dolly incorporates a mechanism that provides positive steer displacement at the dolly tires as a function of the yaw articulation between the dolly and its towed trailer. Figure 4 is an illustration of a controlled-steering C-dolly. Properties of the steering system are such that the rate of dolly steering, per degree of intertrailer articulation, is determined by geometric parameters defining the overhang placement of the dolly fifth wheel, aft of the lead trailer axle, and the effective wheelbase of the trailing trailer. At articulation angles of magnitude greater than, say, 30 degrees, the steering system may *disengage* and allow free castering of the dolly tires but will reengage when the magnitude of the articulation angle drops below 30 degrees. Dolly-steering systems of this type may require special modification of the towed trailer to accommodate the steering linkage, but such modifications must leave the trailer compatible with conventional dollies.

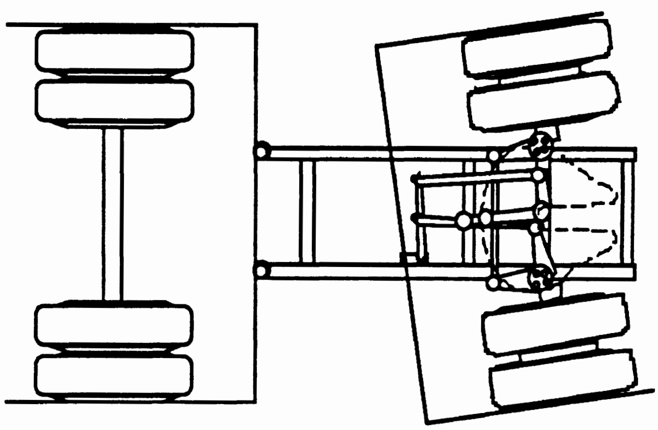


Figure 4. Illustration of a controlled-steering C-dolly

Weight

Dolly weight is obviously a property that falls outside of the realm that influences dynamic performance of the vehicle. That is, minimization of dolly weight is simply an important goal in dolly design. Clearly the desire for low dolly weight tends to conflict with the desire for both stiff and strong dolly structures. Since dolly purchasers and manufacturers are motivated by economic factors to minimize dolly weight, this property is seen as the object of continuing design refinement for the sake of competition.

ELEMENTS OF THE SIMULATION STUDY

A very extensive simulation task was undertaken to measure the performance of changing parametric sensitivities of A- and C-train combinations, as a function of dolly related characteristics. The simulation study was structured to address several themes, as follows:

- Mapping the relationships between the performance measures and the properties of baseline, A-trains.
- Investigating the influence of the selected dolly properties on the performance measures and mapping the ability of the specified dollies to improve primary performance.
- Checking the influence of the specified dollies on the secondary performance measures.
- Conducting ancillary studies to examine stability in backing and to measure pintle-hitch forces for the various maneuvers.

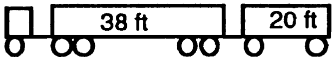
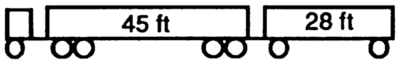
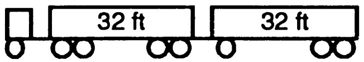
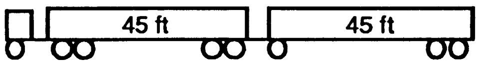
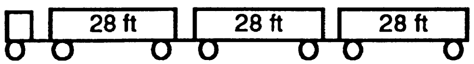
The following discussion describes the set of baseline vehicle configurations, the parametric changes from the baseline, and the simulation *testing* undertaken for each task.

Baseline Configurations

Six baseline configurations of multitrailer vehicles were selected as study vehicles. These are presented in table 2. These particular configurations were selected as reasonably representative of the full range of the common multitrailer vehicles used in the U.S., and cover a broad range of important physical properties that are embodied in the configuration dimensions.

The first configuration, the five-axle 28x28-foot double is surely the most common U.S. double. (The 28-foot doubles, with up to nine axles, are the only form of multitrailer combination that is specified as legal at the national level.) The Rocky Mountain and turnpike doubles are two other configurations in common, but regional, use. The Rocky Mountain double is more common in the western mountain states, and the turnpike double is more common in the Northeast. (The turnpike double often uses a two-axle dolly with a

Table 2. Vehicle configurations

Configuration	*Description*	*Application*	*GAWR's (kilo lbs)*	*Tractor-Semi Wt. (kilo lbs)*	*Full Trailer Wt. (kilo lbs)*	*GCW (kilo lbs)*
	5-axle doubles	Interstate transport	10/17.5/17.5/ 17.5/17.5	45	35	80
	7-axle doubles	Intrastate transport	12/34/34/ 20/20	80	40	120
	7-axle doubles	Rocky Mtn. doubles	12/34/34/ 20/20	80	40	120
	8-axle doubles	Northwestern states	10/26/26/ 17/26	62	43	105
	8-axle doubles	Turnpike doubles	12/34/34/ 20/34	80	54	134
	7-axle triples	Western states	10/17.5/17.5/ 17.5/17.5/ 17.5/17.5	45	35 x 2	115

34,000-pound load rating. This study considered only single-axle dollies. Further, the turnpike configuration being considered here is common.) The eight-axle double, composed of two 32-foot trailers, is representative of configurations seen in the Northwest and the northern tier of states east to Minnesota. The specific configuration shown here is quite common in Washington State. The seven-axle double, composed of a 38-foot and a 20-foot trailer, is meant to represent a class of vehicle found in many western or mountain states. Specifics vary from state to state, but they are typically characterized by a moderate length lead trailer and short pup. The general configuration is often used in bulk commodity freight applications. Finally, until recently, the 28x28x28-foot triple was a configuration gaining acceptance in the Northwest and mountain states. Major segments of the trucking industry are and will be lobbying for its further acceptance, perhaps at the Federal level. As of this writing, however, the 1991 Intermodal Surface Transportation Efficiency Act has put a freeze on LCV implementation.

The configurations chosen provide a good spread of trailer lengths (wheelbases), unit weights, and axle loads. These are:

Lead trailer lengths:	28, 32, 38, and 45 feet
Pup trailer lengths:	20, 28, 32, and 45 feet
Tractor-semitrailer weights:	45, 62, and 80 thousand pounds
Full trailer weights:	35, 40, 43, 54 thousand pounds
Weight per axle (not front):	13, 17, 17.5, and 20 thousand pounds

Payload for the baseline vehicles represents the fully loaded condition with a median-level cargo center-of-gravity height. The baseline vehicles are equipped with tires that are representative of typical tire design (i.e., radials with average lateral and vertical stiffness properties). Similarly, baseline suspension parameters are representative of leaf-spring suspensions found commonly at the three generic positions—the tractor front, tractor rear, and trailer suspension positions. Table 3 details the various characteristic values for defining the baseline condition. Some of the parameters are described as normal. In these areas, the best judgment of the researchers was used to select values typical of the existing commercial fleet. Also given is a three letter classification to identify a baseline vehicle. Similar identification letters are described later and are used to identify the different vehicle and parametric variations.

Parameter Variations

Loading Conditions

Simulations were conducted with vehicles fully loaded in their baseline condition. In addition, loading was varied to alter height of the payload center of gravity over ranges of (1) low—70 inches, (2) medium—85 inches, and (3) high—100-inch positions. These

Table 3. Characteristics for the baseline vehicles

Area of Variation	*Variable*	*Value*	*File ID*
Dolly	Tongue length	80 inches	BAS
Tire	Tire Stiffness	11R22.5 Steel Radial, New	
Hitch Location	Hitch Long. Position	Normal	
Payload	C.G. Height	85 inches	
	Inertia	Normal	
Suspension	Width	Trailer 102 inches	
	Roll Stiffness	Multi-leaf	

numbers are representative of conditions in which the vehicle is loaded with (1) uniform freight of median density, (2) less than truckload (LTL) freight that is nearly full by volume but has two-thirds of the cargo weight in the bottom half of the load, and (3) a uniform cargo yielding a cube-full, maximum gross weight load that results in the height of the center of gravity midway between floor and ceiling.

Another variation in loading condition was moment of inertia—represented in these simulations using the simple formula for rectangular prismatic shapes. Variations on moment on inertia were 200 percent and 50 percent of the baseline.

Hitch Location

Although the selection of the six baseline vehicles provides a substantial range of both wheelbases and pintle hitch locations (i.e., longitudinal distance from the center of gravity), these geometric parameters are interrelated by the body length so that they are not independent variables. Thus, additional variations of plus and minus 12 inches in hitch location were used to introduce hitch placement as an independent variable.

Tires

The baseline vehicles employed a median design radial tire. Two other types of tires (worn, and, therefore, stiffer radials and relatively low-stiffness bias tires) were also represented in the investigation. For specific values of tire-cornering stiffnesses, see appendix D.

Tongue Length

Typical A-dolly tongue lengths are in the range of 72 to 80 inches. Variations covering values of 80 inches, 100 inches, and 120 inches were employed as a method to investigate this elementary dolly parameter.

Suspensions

The suspensions of all tractor front axles were represented with median level parameters and were not varied (since we note little or no influence on the performance measures of interest here). As for tractor rear suspensions, two cases were represented: the baseline suspension representative of a leaf spring suspension and one variation representative of air suspensions. Three variations in trailer suspensions were considered: the baseline suspension, a relatively low-stiffness, leaf-spring suspension, and an air suspension having a high value of roll stiffness. The nominal roll stiffness values and other details of the suspensions can be found in appendix D.

The tractor and trailer suspension changes were combined to produce (1) a low-stability variation composed of the low-stiffness versions of tractor air suspension and trailer leaf-spring suspension and (2) a high-stability combination composed of the tractor leaf-spring suspension and the trailer air suspension. The baseline condition combines the tractor leaf spring suspension with the stiffer trailer leaf-spring suspension. As for axle width, the baseline vehicles incorporated 96-inch axles, while a 102-inch variation for trailer axles was also included.

Table 4 summarizes the parameter changes for the variations from the baseline condition. Also shown is the file identification. For filenaming purposes, a three letter classification was selected that would uniquely designate each off-baseline vehicle file.

The Dollies

The simulation study included three classes of dollies, which are identified as:

- Class 1: A-dolly
- Class 2: Light C-dolly
- Class 3: Heavy C-dolly.

The A-dolly was a conventional A-dolly. The C-dollies were divided into two classes: light and heavy. The differences in performance-related properties of these two classes of dolly are in the area of frame torsion and stiffness.

A characteristic property of C-dollies is that, usually, the dolly tires steer relative to the dolly frame. Three options were used in this regard: self-steering axles with two different tire types, and controlled-steering axles. The steering ratio of the controlled-steer, dollies was determined by a formula from [1].

Table 5 defines the seven dolly variations. The category of *File ID* serves to identify the dolly used for a particular simulation.

Table 4. Values for the vehicle and dolly parameter variations

Area of Variation	*Variable*	*Value*	*File ID*
Dolly	Tongue length	100 inches	DO1
		120 inches	DO2
Tire	Tire file	11R22.5 Steel Radial, 2/3 Worn	TI1
		10x20 Bias Ply	TI2
Hitch Position	Hitch Longitudinal	Forward 12 inches	SE1
	Position	Rearward 12 inches	SE2
Payload	C.G. Height	100 inches	PL1
		70 inches	PL2
	Inertia	Larger value (twice)	PL3
		Smaller value (half)	PL4
Suspension	Width	Trailer 96 inches	SU1
	Roll Stiffness	soft	SP2
		Stiff 1 (Multi-leaf)	SP3
		Stiff 2 (Air Susp.)	SS4

Test Maneuvers

The performance measures of vehicles were determined using six different simulation maneuvers. Each maneuver was simulated using the UMTRI *Yaw/Roll* model. These were:

- A modified J-turn maneuver, specifically the RTAC-A maneuver [5], was used to determine high-speed steady-state offtracking and static rollover threshold.
- Three rapid lane-change maneuvers, specifically the RTAC-B type maneuvers [5], were used to determine rearward amplification, dynamic-load-transfer ratio, high-speed transient offtracking, and yaw damping.
- A pulse steer maneuver, consisting of a 2 degree (road-wheel) steering pulse maintained for 0.2 seconds duration followed by 5 seconds of zero steer [1]. This maneuver was also used to determine yaw damping.
- Low-speed offtracking and friction demand in a tight turn were evaluated using the RTAC-C maneuver.

A complete description of these six maneuvers can be found in appendix A.

Table 5. Dolly variations

Dolly	*Class*	*Steering*	*Description*	*File ID*
A-dolly	1	N/A	Single pintle hitch with baseline tires.	A
C-dolly	2	All	Light, dual draw-bar dolly with roll stiffness of 20000.0 in-lb/deg	
		Self-steer	Baseline tires modified to saturate Fy at 0.3Fz	2C1
		Self-steer	Baseline tires modified to saturate Fy at .25Fz	2C2
		Controlled -steer	See controlled-steer formula in [1]	2C3
	3	All	Heavy, dual draw-bar dolly with roll-stiffness of 40000.0 in-lb/deg	
		Self-steer	Baseline tires modified to saturate Fy at 0.3Fz	3C1
		Self-steer	Baseline tires modified to saturate Fy at .25Fz	3C2
		Controlled -steer	See controlled-steer formula in [1]	3C3

Computer Simulation Matrix

In total, 2880 simulation runs were conducted, which provided values for the eight primary and secondary performance measures of the various vehicle combinations for each run. (Yaw damping was measured using two different maneuvers.) The overall dimensions of the total simulation matrix consisted of the following;

- Six vehicle configurations, shown in table 2.
- 15 baseline and off-baseline parameter variations, shown in tables 3 and 4.
- Seven dolly variations, shown in table 5.
- Six maneuvers (RTAC-A and C, three RTAC-B, and pulse-steer maneuvers).

Given this large range (3780 total combinations) of possible simulations, not every combination in the matrix was simulated. In many cases it could be determined that duplicate answers would result if certain simulations were run. For example, varying the inertia property of a vehicle simulating the RTAC-C maneuver would have negligible results on the low-speed offtracking characteristic of the vehicle as compared with the baseline. In other cases, it was desired to find the worst- or best-performing vehicles among the various parameter changes, allowing intermediate or benign vehicle combinations to be excluded. A breakdown of which simulations were run is given below.

Mapping the Primary and Secondary Performance Measures of A-trains

With the test vehicles configured as A-trains, the matrix of simulation runs shown in table 6 was conducted to map the primary and secondary performance measures. All six vehicle combinations were run for each cell in table 6. The RTAC-A column is a straightforward full matrix for each of the test vehicle variations. The RTAC-B column is similar, except that it indicates three repeats of each vehicle set. This implies one run at each of the three steer-input frequencies specified by the RTAC procedure. The Pulse Steer and RTAC-C columns are a full matrix of runs for each of the variations.

Table 6. Number of simulation runs in the matrix for mapping the primary and secondary performance measures of A-trains

Parameter Variations	*J-turn RTAC-A*	*Rapid Evas. RTAC-B*	*Slow Turn RTAC-C*	*Pulse Steer*
None (Baseline)	6	18	6	6
2 C.G. Height	12	36	12	12
2 Inertia	12	36	12	12
2 Tire Compliance	12	36	12	12
3 Susp. Stiffness	18	54	18	18
1 Suspension Width	6	18	6	6
2 Hitch Position	12	36	12	12
2 Tongue Length	12	36	12	12
Totals (540)	90	270	90	90

Mapping the Primary and Secondary Performance Measures of C-trains

In this portion of the study, the parameter variations of interest are no longer just those of vehicle properties, but also those of dolly properties. In the above mapping of A-train properties, parameter variations were undertaken largely to reveal the level of noise that a highly simplified, vehicle-classification scheme must deal with. The second phase of simulations seeks to define the power of each of the specified dollies for controlling the response of various configurations of multitrailer vehicles.

With the test vehicles configured as C-trains, the matrix of simulation runs shown in table 7 was conducted to map the primary and secondary performance measures. In the high-speed simulations only five vehicle combinations were run with the six different C-dollies shown in table 5. The eight-axle turnpike double (45x45) was so benign as an A-train that using the C-dolly with this vehicle was not warranted. The RTAC-A and pulse steer columns are a straightforward full matrix for each of the test vehicle variations. The

Table 7. Number of simulation runs in the matrix for mapping the primary and secondary performance measures of C-trains

Parameter Variations	*J-turn RTAC-A*	*Rapid Evas. RTAC-B*	*Slow Turn RTAC-C*	*Pulse Steer*
None (Baseline)	30	90	18	30
2 C.G. Height	60	180	-	60
2 Inertia	60	180	-	60
2 Tire Compliance	60	180	-	60
3 Susp. Stiffness	90	270	-	90
1 Suspension Width	30	90	-	30
2 Hitch Position	60	180	36	60
2 Tongue Length	60	180	36	60
Totals (2,340)	450	1,350	90	450

RTAC-B column is similar, except that it indicates three repeats of each vehicle set. This implies one run at each of the three steer-input frequencies specified by the RTAC procedure. Only a subset of the possible combinations was run to determine the low-speed offtracking with the RTAC-C maneuver. The RTAC-C simulations were performed on all six vehicle configurations (including 45x45 configurations), but only on the class 2 C-dolly variations, and on five of the 15 different parameter variations.

Filenames

Files were generated (with some exceptions) for every vehicle configuration with every dolly, using all parameter variations (including baseline). The construction of a file name consisted of (vehicle)x(variation)x(dolly). For example, a 28x28-foot, five-axle double with a dolly tongue length of 100 inches (DO1) using a class 2 C-dolly that has controlled-steer (2C3) would result in a file named: 28x28DO12C3. Table 8 lists the files generated using the class 1 A-dolly. The same vehicle configurations and variations were used for all of the C-dollies. The files generated for each type of C-dolly are the same as table 8 with file names varying only in the dolly extension, i.e., replace the A suffix with 2C1, 2C2, 2C3, 3C1, 3C2, or 3C3 for the tables of the corresponding dollies.

Special Task—Stability in Backing

The newly developed AUTOSIM [10] has been used to create a simplified yaw plane model for multitrailer vehicles that includes the ability to back. This model will be used to evaluate the influence of dolly design on the stability in backing. The six baseline

Table 8. Filenames for all vehicle configurations and variations using the A-dolly

A	*28x28*	*38x20*	*45x28*	*32x32*	*45x45*	*3x28*
BAS	28x28BASA	38x20BASA	45x28BASA	32x32BASA	45x45BASA	3x28BASA
DO1	28x28DO1A	38x20DO1A	45x28DO1A	32x32DO1A	45x45DO1A	3x28DO1A
DO2	28x28DO2A	38x20DO2A	45x28DO2A	32x32DO2A	45x45DO2A	3x28DO2A
TI1	28x28TI1A	38x20TI1A	45x28TI1A	32x32TI1A	45x45TI1A	3x28TI1A
TI2	28x28TI2A	38x20TI2A	45x28TI2A	32x32TI2A	45x45TI2A	3x28TI2A
SE1	28x28SE1A	38x20SE1A	45x28SE1A	32x32SE1A	45x45SE1A	3x28SE1A
SE2	28x28SE2A	38x20SE2A	45x28SE2A	32x32SE2A	45x45SE2A	3x28SE2A
PL1	28x28PL1A	38x20PL1A	45x28PL1A	32x32PL1A	45x45PL1A	3x28PL1A
PL2	28x28PL2A	38x20PL2A	45x28PL2A	32x32PL2A	45x45PL2A	3x28PL2A
PL3	28x28PL3A	38x20PL3A	45x28PL3A	32x32PL3A	45x45PL3A	3x28PL3A
PL4	28x28PL4A	38x20PL4A	45x28PL4A	32x32PL4A	45x45PL4A	3x28PL4A
SU1	28x28SU1A	38x20SU1A	45x28SU1A	32x32SU1A	45x45SU1A	3x28SU1A
SP2	28x28SP2A	38x20SP2A	45x28SP2A	32x32SP2A	45x45SP2A	3x28SP2A
SP3	28x28SP3A	38x20SP3A	45x28SP3A	32x32SP3A	45x45SP3A	3x28SP3A
SS4	28x28SS4A	38x20SS4A	45x28SS4A	32x32SS4A	45x45SS4A	3x28SS4A

configurations will be evaluated with A-dollies, A-dollies locked on center, and controlled- and self-steer C-dollies. The maneuver will consist of backing the vehicle from an initial condition in which the articulation angles are set to a very small, but nonzero value. The measure of interest will be distance traveled before the occurrence of (1) a specified articulation angle and/or (2) a specified lateral offset (from the projected straight path of the tractor). The measure is crude, but the differences between the A-dolly and the others can be expected to be so dramatic as to indicate generally the presence or absence of the ability to back the vehicle.

Parametric Sensitivities of Combination Vehicles

Appendix C presents a complete set of summary plots of the response metrics for the selected configurations of A- and C-trains. In this section, the form of these results will be discussed, example plots will be presented, and prominent results will be highlighted.

The six types of multitrailer combinations are each represented with seven versions of dolly coupling. The performance of each of these 42 configurations is, in turn, characterized by six plots—one for each of the measures of performance of concern. Each plot covers the variations in response resulting from changes in each of seven parameters.

Shown in figure 5 is a sample for the case of the rearward amplification response of the 3C2 version of 28x28-foot double, covering the following;

- The doubles combination having twin 28-foot trailers.
- The class-3, heavy C-dolly (having the higher value of dolly roll stiffness).
- Version 2 of dolly-steering (caster-steered, with breakaway at 0.25 g's).

Table 9 has been included as a guide for interpreting the symbols and values in figures 5, 6, and 7. We note that the seven selected truck and dolly parameters, defined in the second column, are varied over a set of numerical values that are distinguished by (-1), (0), (1), or (2) values of a variation code. (See the tables in appendix C for a full explanation of these codes.) Looking at figure 5 and noting the symbols designated for each parameter, the changes in response associated with each parametric variation are registered at coordinates of the variation code on the x axis and the computed performance level on the y axis. Following the four variations in *Suspension roll stiffness*, for example, we see that the filled-triangle symbol appears at coordinates of (-1,1.75), (0,1.72), (1,1.67), and (2,1.73).

Table 9. Guide for interpreting sensitivity plots

		Parameter Variations			
Symbol	*Parameter*	*-1*	*0*	*1*	*2*
—□—	Payload cg height, inches	70	85	100	None
—△—	Yaw moment of inertia, in-lb-sec^2	1/2 of Baseline	Baseline[1]	2 times Baseline	None
—◇—	Tire-cornering stiffness, lb/deg	New Bias 564	New Radial 881	Worn Radial 1124	None
—▲—	Suspension roll stiffness, in-lb/deg	117,800[1] (nominal)	137,600[1] (nominal)	175,000[1] (nominal)	203,700[1] (nominal)
—✱—	Overall axle width, inches	96	102	None	None
—■—	Pintle hitch overhang, inches	Baseline-12	Baseline[1]	Baseline+12	None
—◆—	Dolly tongue length (wheelbase), inches	None	80	100	120

[1] Vehicle Dependent

The figure illustrates that the rearward amplification performance of this C-dolly-equipped twin trailer combination can vary from 1.63 to 1.88 due to common changes in system properties. The baseline vehicle (i.e., the 0 variation code) registers a performance level of 1.72. Among the more important parameters, tire-cornering stiffness and height of the payload center of gravity are prominent but have the opposite trend in their effects—with rearward amplification falling when tire-cornering stiffness increases, but rising when the height of the center of gravity increases.

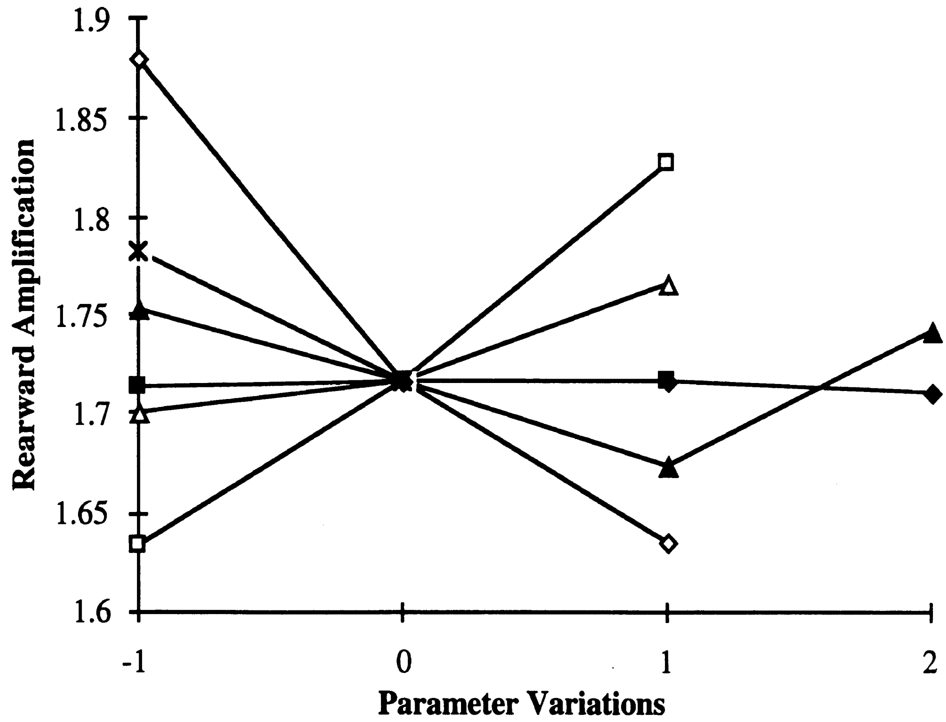


Figure 5. Parametric sensitivity in the rearward amplification performance of the 28x28-foot five-axle 3C2 double

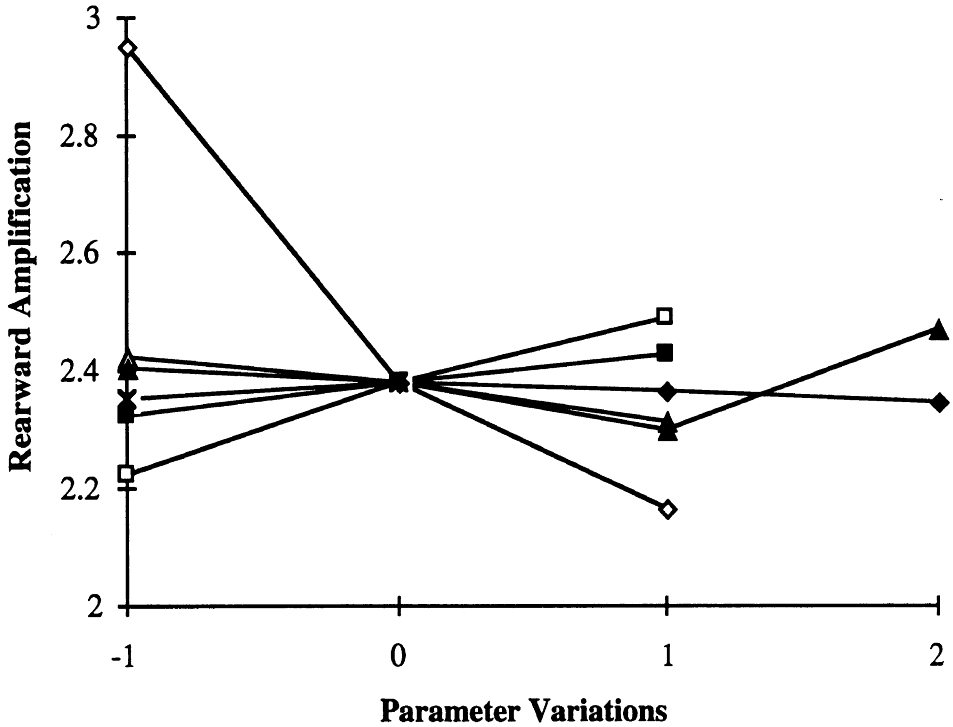


Figure 6. Parametric sensitivity in the rearward amplification performance of the 28x28-foot, five-axle A-train double

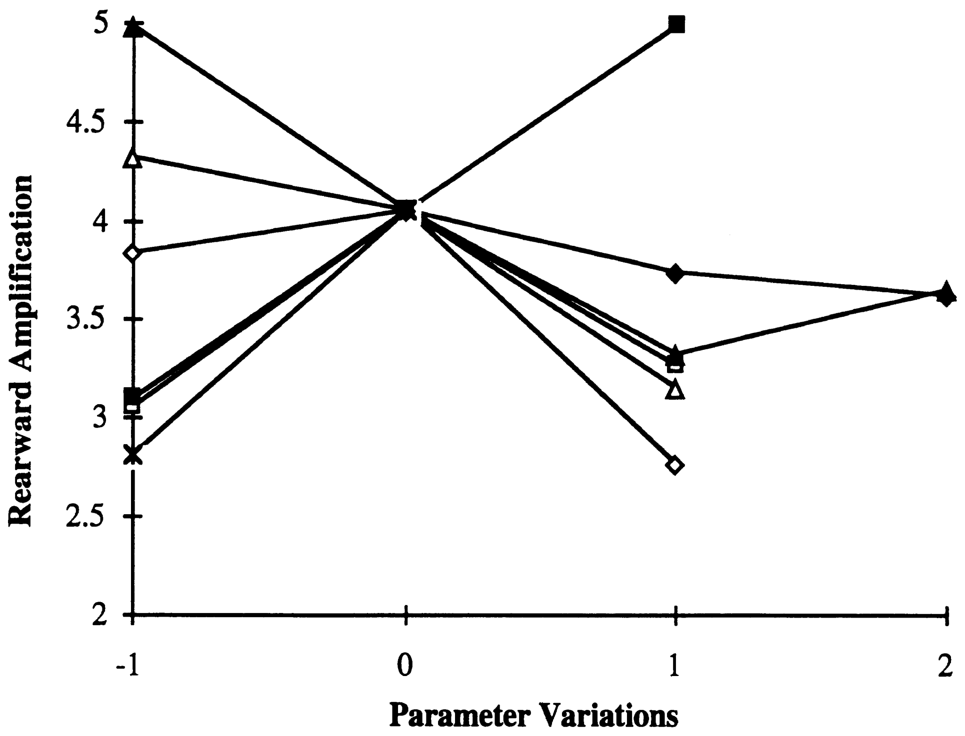


Figure 7. Parametric sensitivity in the rearward amplification performance of the 28x28x28-foot seven-axle A-train triple

Figure 6 is presented as the corresponding plot of rearward amplification for the A-train version of the same 28x28-foot, twin-trailer layout. In this plot, the same matrix of parametric variations yields the plotted set of values shown at the bottom. Now, in contrast with the results presented above, the A-train double shows a baseline (0^{th}) rearward amplification level of 2.4. When this vehicle is equipped with bias-ply tires, the rearward amplification level rises to almost 3.0 due to the lower cornering stiffness of bias-ply tires as compared to the baseline radial tires.

As a third illustration of plots appearing en masse in appendix C, figure 7 shows the rearward amplification response levels for the A-train version of the triples combination. Here we note that the baseline, 0^{th}, performance level is 4.0 and that two parameters have the power to increase rearward amplification up to approximately 5.0. Namely, the cases involving either bias-ply tires or the larger (+12 inch) value of pintle-hitch overhang both result in rearward amplification levels near 5.0. In general, all of the parametric variations appearing here one at a time will superpose upon one another if introduced in combination.

Moreover, the computer simulation exercise has produced a broad characterization of performance for each of the A- and C-train configurations of interest. Examination of the multiple plots covering all of the cases supports the following observations.

- C-trains are virtually indistinguishable from their corresponding version of A-train in terms of static roll stability and high-speed offtracking performance levels.

- C-trains are always superior to the corresponding A-train in their rearward amplification, dynamic load transfer coefficient, and high-speed transient offtracking performances.
- The distinctions between A- and C-train performance, as measured by the yaw-damping characteristic, are mixed. While some versions of C-dolly effect an improvement in some cases, the improvement is not large, nor does it accrue when other parametric variations are present.
- No compelling differences in performance are seen between the light and heavy classes of C-dolly in essentially any vehicle configuration or set of parametric variations. A small, but probably inconsequential, increase in dynamic load transfer coefficient is seen to appear when the stiffer dolly is employed.
- Modest differences are seen between the two versions of C-dolly steering systems. Namely, in the following measures, the caster-steered versions are seen to be somewhat higher (better) in performance than the controlled-steer variety:
  - — rearward amplification
  - — high-speed, transient offtracking
  - — yaw-damping ratios.

GENERALIZED ASSESSMENTS OF VEHICLE PERFORMANCE

The general premise of this study was to develop a method for specifying dollies for multiple-trailer vehicles by a two-step process. The first step of this process would be to characterize the vehicles' performance quality in their baseline state, that is, when equipped with conventional A-dollies. Assuming that the performance of the A-train was not adequate, an innovative dolly providing sufficient incremental performance improvement to meet a minimum performance requirement would be specified in the second step. The hope was that both steps could be accomplished through a highly simplified method, which would require only *very simple* calculations and *simple* vehicle-descriptive parameters that could be easily obtained in the field.

The first two subsections that follow deal with these two basic tasks. In the first, simplified predictors of the performance measures of A-trains are developed from the results of the simulation study. In the second, a simple means for predicting the incremental improvement in performance through the use of C-dollies is addressed. These predictors are restricted to the primary dynamic performance qualities of interest for multiple-trailer vehicles—the measures derived from the RTAC-A, B, and C maneuvers plus a pulse-steer maneuver. The third subsection deals with two additional performance issues, namely stability during backing and potential pintle hitch loads.

Simplified Predictors of the Performance of A-trains.

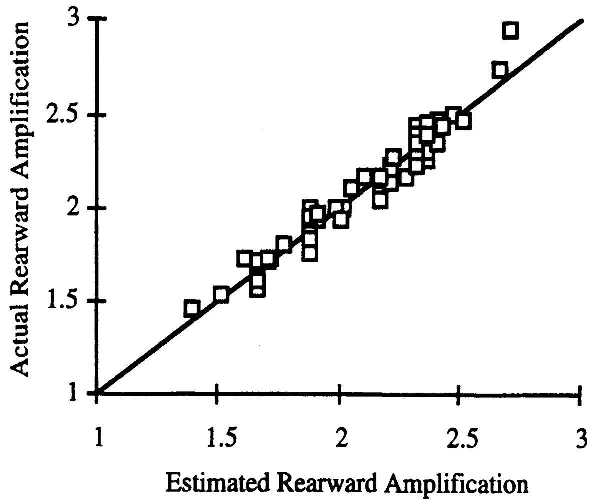
The effort to obtain simplified formulations for predicting the performance measures of A-train vehicles was surprisingly successful. In a general sense, the approach was simply to apply linear regression techniques to determine the relationships between the dependent variables—the performance measures of interest (table 1)—and the independent variables—the parameter variations implied by the six vehicle types equipped with A-dollies in their 15 variations (tables 2, 3, and 4). In detail, the task was rather more complex and required a great deal of trial and error searching for the most useful set of independent variables.

The independent variables that were included in the statistical-analysis process extended well beyond the individual parameters varied in the simulation matrix. An extensive set of independent variables constructed of nonlinear combinations of the basic parameters were added to the list. These terms were created out of a mechanistic understanding of vehicle performance and in the expectation that they might have a more direct relationship to the performance measures. Perhaps the best means of explaining the rationale behind these constructed variables is an example. Track width (T) and center-of-gravity height (H) are two vehicle parameters that were varied independently in the simulation matrix. Both can be expected to have a substantial influence on performance measures that are influenced by roll behavior, for example, static rollover threshold. But physical analysis has long since established that the influence of these two parameters is *not* generally of the linear form, i.e., aT + bH (where a and b are constants), which would be revealed by including T and H separately in a multiple linear-regression analysis. Rather, mechanistic analysis of vehicle roll stability has lead to the understanding that the influence of these two variables on roll-related behavior is often (linearly) proportional to their nonlinear combination, T/2H. Thus, while T and H might be included individually as independent variables in a regression analysis, it is likely to be more effective to include T/2H as an independent variable.

Many such nonlinear combinations of basic parameters were included in the investigation. It will be seen that the most successful were T/2H and certain nonlinear combinations of the trailer wheelbases.

Figures 8 through 15 show the results of the regression analyses relating to the eight performance measures studied, respectively. Appendix D presents a listing of the data, including parameter definitions, on which these results are based. Many other basic and constructed independent variables were examined and discarded in the analysis process.

First, to explain the form of the figures, consider figure 8. This figure shows the results of three separate regression analyses of the relationship between the rearward amplification and several independent variables. Each analysis is represented by a graph and by the tabular data immediately to the right of the graph. Proceeding from the top to the bottom, the three analyses are based on progressively *simpler input data sets*, but

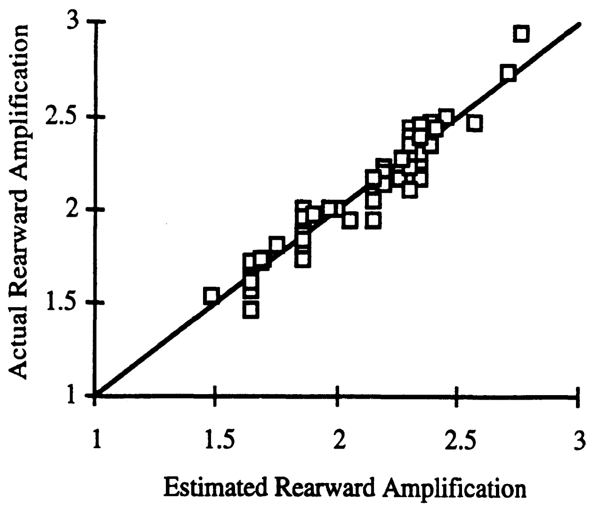


$r^2 = .96$

Residual RMS = 0.066

F = 373

Variable	Coefficient	Std. Err.	Partial F
C_α	-.001089	.000074	214
$(WB2*WB3)^{1/2}$	-.01813	.0023	61
T/2H	1.183	.18	43
(WB2*WB3)	-.0000208	.0000035	34
Constant	7.373		

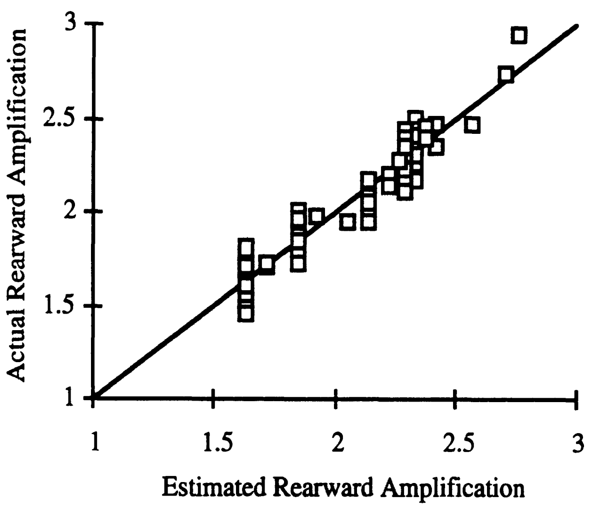


$r^2 = .93$

Residual RMS = 0.081

F = 244

Variable	Coefficient	Std. Err.	Partial F
Radial (1) or Bias (0)	-.412	.037	121
$(WB2*WB3)^{1/2}$	-.01813	.00283	41
T/2H	-1.209	.2202	30
(WB2*WB3)	.0000202	.0000043	23
Constant	6.820		



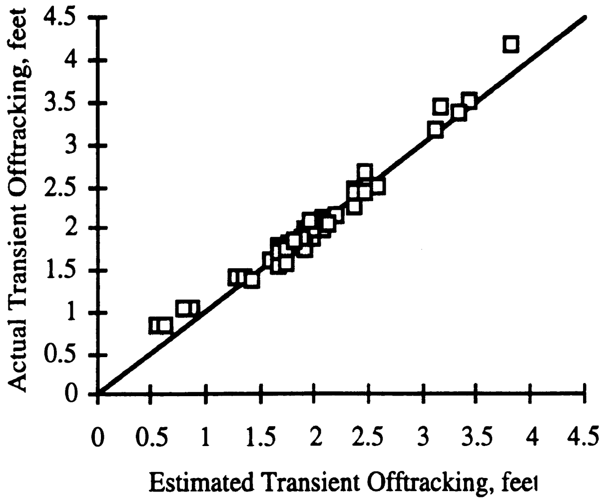
$r^2 = .91$

Residual RMS = 0.093

F = 184

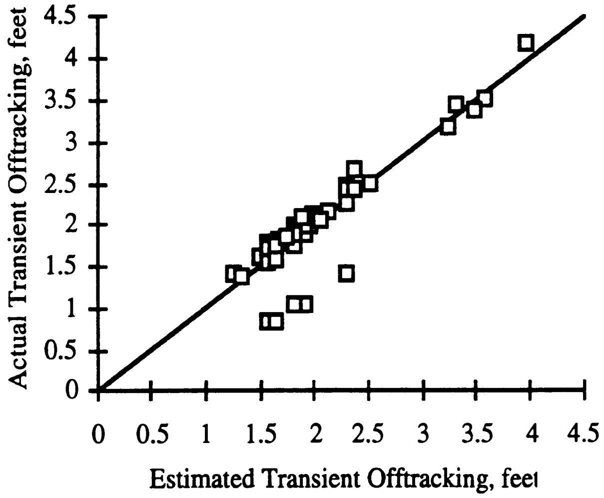
Variable	Coefficient	Std. Err.	Partial F
Radial (1) or Bias (0)	-.4208	.0430	96
$(WB2*WB3)^{1/2}$	-.01813	.00323	31
Track Width	-.0138	.0053	7
(WB2*WB3)	-.0000202	.0000048	17
Constant	7.504		

Figure 8. Simple predictors for estimating rearward amplification of A-train doubles



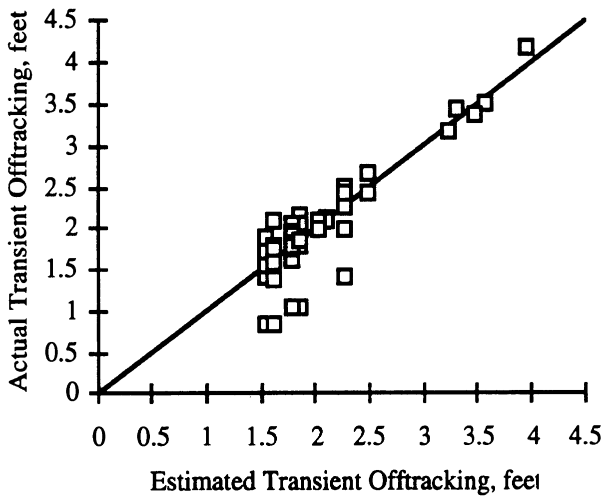
$r^2 = .97$
Residual RMS = 0.107
F = 381

Variable	*Coefficient*	*Std. Err.*	*Partial F*
C_α	-.00454	.000120	1431
T/2H	-2.423	.291	69
WB2	-.1787	.0215	69
WB3	-.2404	.0294	67
$(WB2*WB3)^{1/2}$	.4046	.049	67
Constant	11.57		



$r^2 = .81$
Residual RMS = 0.258
F = 60

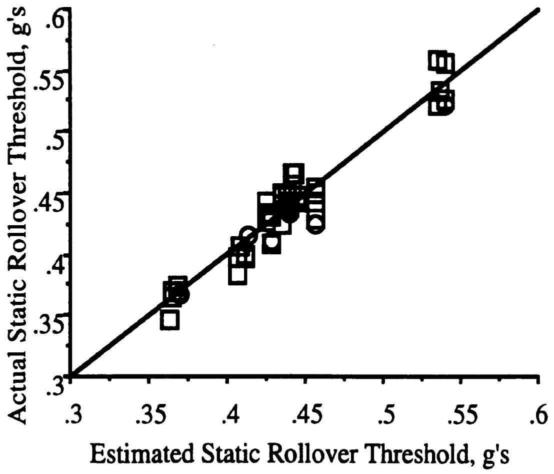
Variable	*Coefficient*	*Std. Err.*	*Partial F*
Radial (1) or Bias (0)	-1.669	.114	213
T/2H	-2.529	.672	14
WB2	-.1787	.0495	13
WB3	-.2404	.0678	13
$(WB2*WB3)^{1/2}$	.4046	.114	13
Constant	9.21		



$r^2 = .80$
Residual RMS = 0.258
F = 54

Variable	*Coefficient*	*Std. Err.*	*Partial F*
Radial (1) or Bias (0)	-1.697	.124	199
Track Width	-.0394	.0147	7
WB2	-.1787	.052	12
WB3	-.2404	.071	12
$(WB2*WB3)^{1/2}$	.4046	.119	12
Constant	11.71		

Figure 9. Simple predictors for estimating high-speed transient offtracking of A-train doubles



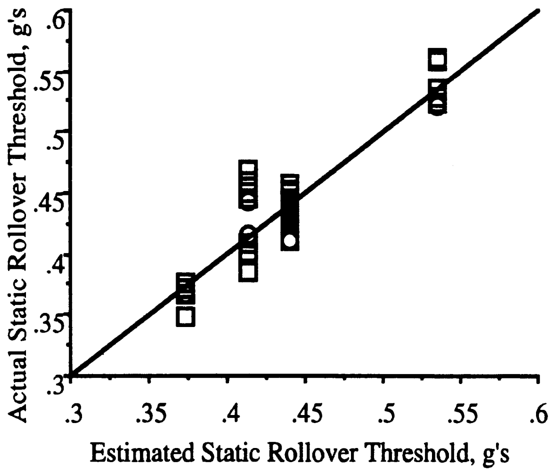
Doubles — □ — and Triples — ○

$r^2 = .90$

Residual RMS = 0.011

F = 377

Variable	*Coefficient*	*Std. Err.*	*Partial F*
T/2H	.7809	.0287	739
Roll Stiffness	.515E-6	.061E-6	71
Constant	-.102		



$r^2 = .81$

Residual RMS = 0.015

F = 380

Variable	*Coefficient*	*Std. Err.*	*Partial F*
T/2H	.7398	.0379	380
Constant	-.0039		--

Doubles — □ — Alone:

$r^2 = .91$

Residual RMS = 0.011

F = 347

Variable	*Coefficient*	*Std. Err.*	*Partial F*
T/2H	.8009	.0308	677
Roll Stiffness	.554E-6	.065E-6	73
Constant	-.118		

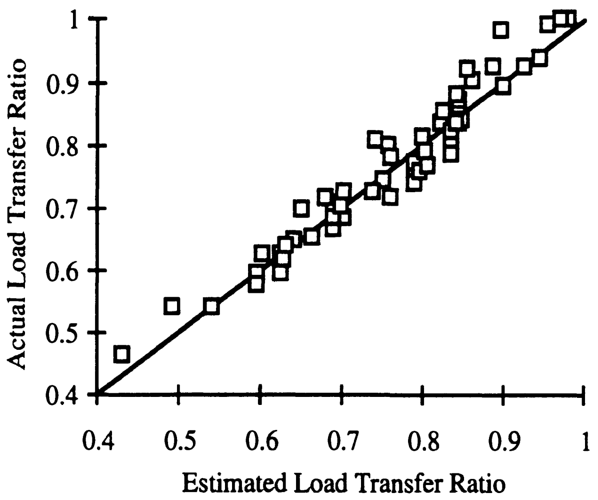
$r^2 = .81$

Residual RMS = 0.016

F = 314

Variable	*Coefficient*	*Std. Err.*	*Partial F*
T/2H	.756	.043	314
Constant	-.013		

Figure 10. Simple predictors for estimating static rollover threshold of A-train doubles and triples

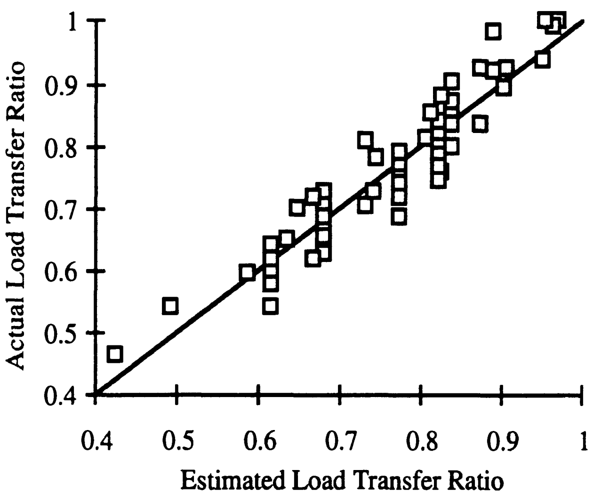


$r^2 = .93$

Residual RMS = 0.030

F = 210

Variable	*Coefficient*	*Std. Err.*	*Partial F*
T/2H	-1.509	.083	332
C_α	.349E-3	.034E-3	107
$(WB2*WB3)^{1/2}$	-.00571	.00105	30
(WB2*WB3)	.632E-5	.158E-5	16
Roll Stiffness	-.683E-6	.175E-6	15
Constant	3.206		

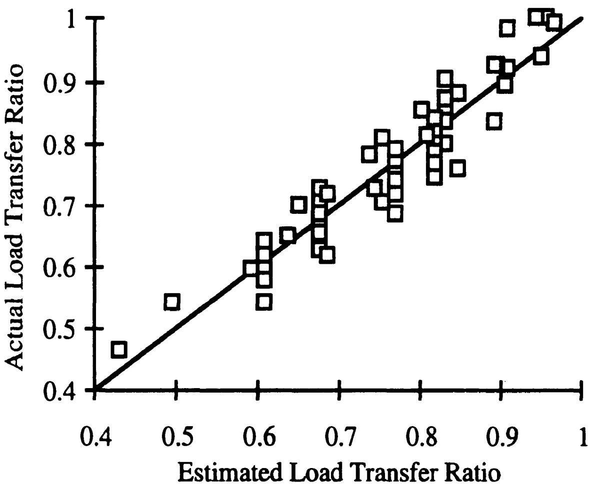


$r^2 = .90$

Residual RMS = 0.037

F = 162

Variable	*Coefficient*	*Std. Err.*	*Partial F*
T/2H	-1.462	.102	206
Radial (1) or Bias (0)	-.1269	.0174	53
$(WB2*WB3)^{1/2}$	.657E-5	.197E-5	20
(WB2*WB3)	-.00586	.00131	11
Constant	2.917		



$r^2 = .91$

Residual RMS = 0.036

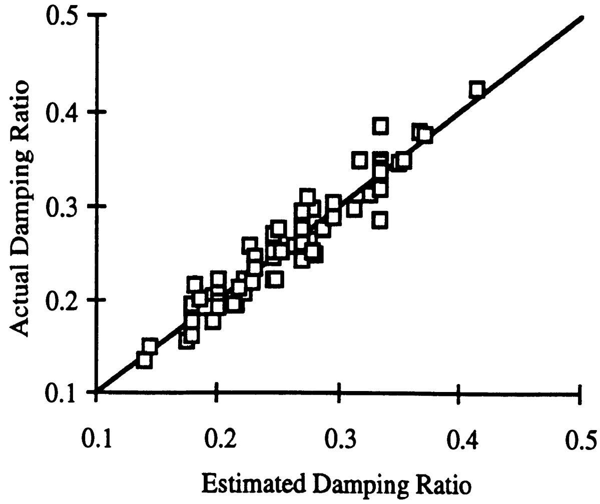
F = 344

Variable	*Coefficient*	*Std. Err.*	*Partial F*
Est. Rearward Amplitude*	.3161	.0145	474
Est. Static Roll†	-1.850	.1309	200
Constant	.910		

* This is the estimated rearward amplitude derived from the third analysis of figure 8.

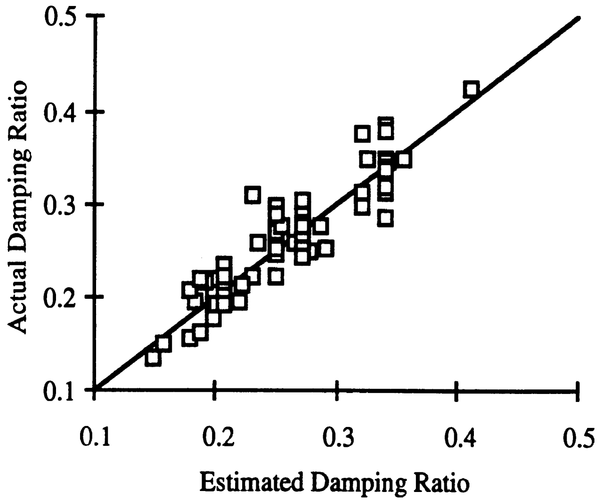
† This is the estimated static rollover threshold derived from the second analysis of figure 10.

Figure 11. Simple predictors for estimating dynamic load transfer ratio of A-train doubles



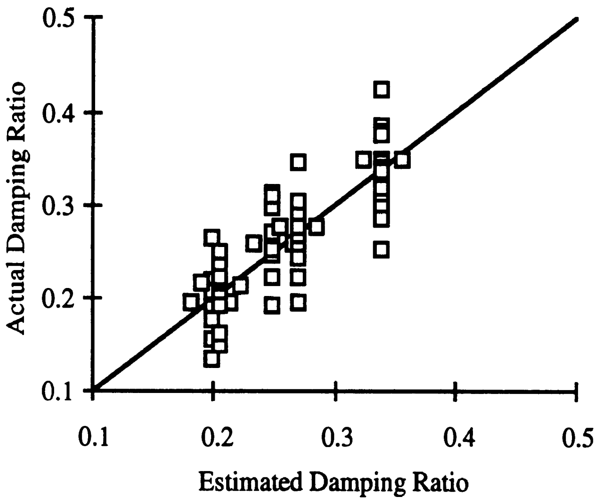
$r^2 = .92$
Residual RMS = 0.018
F = 159

Variable	*Coefficient*	*Std. Err.*	*Partial F*
WB3	.611E-3	.037E-3	270
T/2H	.619	.049	157
Roll Stiffness	.252E-3	.031E-3	68
WB2	.761E-6	.105E-6	52
OH2	.00152	.00027	31
Constant	-.558		



$r^2 = .86$
Residual RMS = 0.024
F = 107

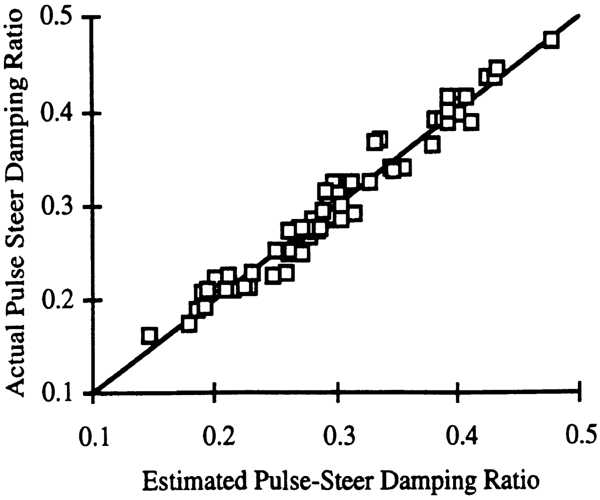
Variable	*Coefficient*	*Std. Err.*	*Partial F*
WB3	.597E-3	.049E-3	149
T/2H	.557	.064	75
WB2	.262E-3	.040E-3	42
OH2	.00132	.00036	13
Constant	-.403		



$r^2 = .71$
Residual RMS = 0.034
F = 58

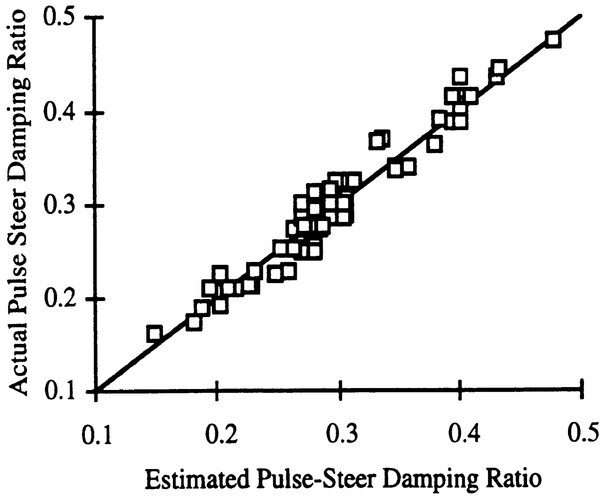
Variable	*Coefficient*	*Std. Err.*	*Partial F*
WB3	.597E-3	.070E-3	73
WB2	.262E-3	.058E-3	21
OH2	.00132	.00051	7
Constant	-.227		

Figure 12. Simple predictors for estimating lane-change damping ratio of A-train doubles



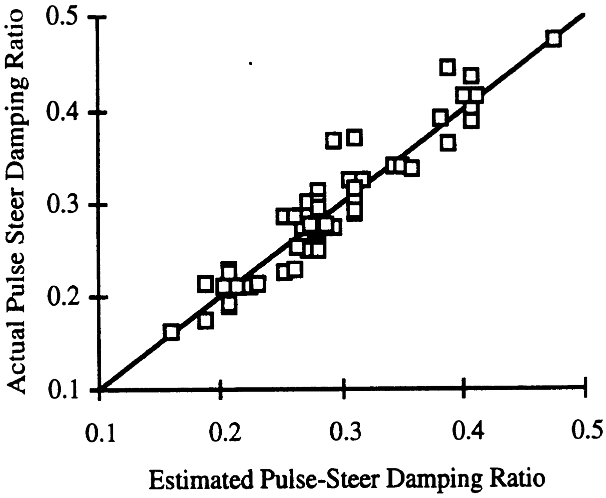
$r^2 = .971$
Residual RMS = 0.014
F = 320

Variable	*Coefficient*	*Std. Err.*	*Partial F*
WB3	.001092	.000028	1547
T/2H	.5973	.0386	240
Roll Stiffness	.724E-6	.082E-6	78
Yaw Inertia Ratio	.0237	.0056	18
OH2	.624E-3	.195E-3	10
Cα	-.333E-4	-.157E-4	5
Constant	-.4895		



$r^2 = .92$
Residual RMS = 0.021
F = 261

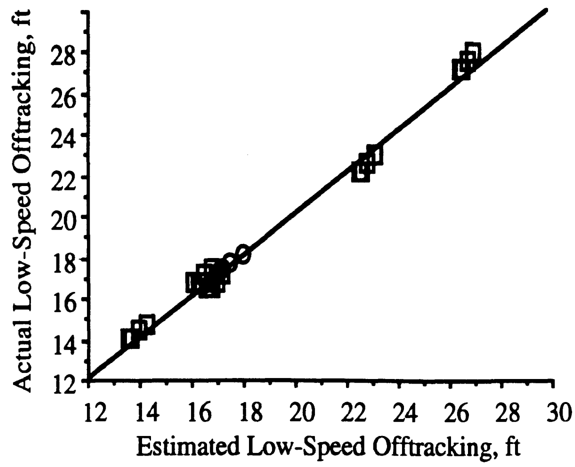
Variable	*Coefficient*	*Std. Err.*	*Partial F*
WB3	.00108	.000042	652
T/2H	.540	.058	87
OH2	.459	-.296	2
Constant	-.346		



$r^2 = .80$
Residual RMS = 0.258
F = 54

Variable	*Coefficient*	*Std. Err.*	*Partial F*
WB3	.00108	.0000626	298
OH2	.459E-3	.437E-3	1
Constant	-.023		

Figure 13. Simple predictors for estimating pulse-steer damping ratio of A-train doubles



Doubles — □

$r^2 = 1.00$ Resd. RMS = 0.402 F = 10403

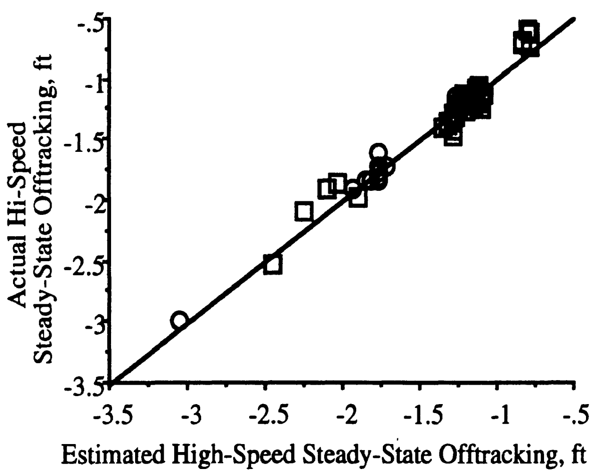
Variable	*Coefficient*	*Std. Err.*	*Partial F*
WHI Len	.7346	.0072	10403
Constant	-10.50		

Triples — o

$r^2 = 1.00$ Resd. RMS = 0.004 F = 66652

Variable	*Coefficient*	*Std. Err.*	*Partial F*
WHI Len	.6707	.0026	66652
Constant	-9.49		

Figure 14. Simple predictors for estimating low-speed offtracking of A-train doubles and triples



Doubles — □

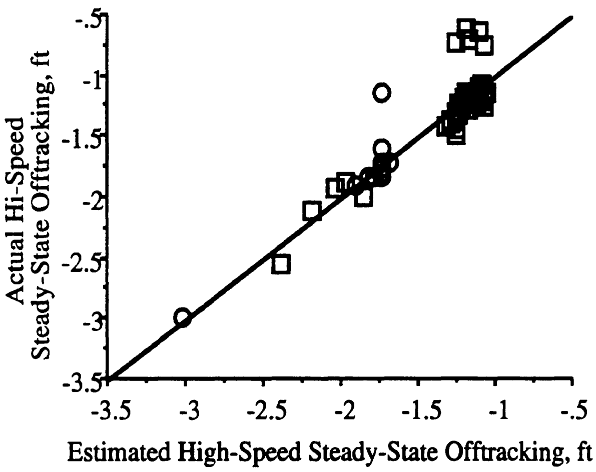
$r^2 = .93$ Resd. RMS = 0.077 F = 484

Variable	*Coefficient*	*Std. Err.*	*Partial F*
Overall Len/Cα	-20.05	0.66	910
WHI Len	.02595	.00183	202
Constant	-.511		

Triples — o

$r^2 = .97$ Resd. RMS = 0.065 F = 435

Variable	*Coefficient*	*Std. Err.*	*Partial F*
Overall Len/Cα	-21.43	1.03	435
Constant	.535		



Doubles — □

$r^2 = .74$ Resd. RMS = 0.149 F = 100

Variable	*Coefficient*	*Std. Err.*	*Partial F*
Overall Len/ Rep Cα	-19.06	1.40	185
WHI Len	.02467	.00372	44
Constant	-.521		

Triples — o

$r^2 = .79$ Resd. RMS = 0.175 F = 50

Variable	*Coefficient*	*Std. Err.*	*Partial F*
Overall Len/ Rep Cα	-20.93	2.97	50
Constant	.516		

Figure 15. Simple predictors for estimating high-speed steady-state offtracking of A-train doubles and triples

provide progressively less accurate predictions. The topmost model represents the best model found using a reasonably limited number of independent variables, that is, not just throwing in everything, but using only variables with both reasonable statistical significance and substantive relative power in determining the result (see below). Unfortunately, this *best* model requires input values that could be difficult to obtain in the field (for example, tire-cornering stiffness, C_α, or center-of-gravity height, H). The other two models progressively cull these variables in favor of surrogates that are easier to obtain. In most cases, the final model can be satisfied with data that can be obtained with little more than a tape measure.

For each individual analysis, the graph shows the *actual* values of performance measure versus the *estimated* value of the measure. The so-called *actual* value is the value obtained by the complex computer simulation analysis. The *estimated* value is value predicted by the far simpler regression models which have been developed.

The table presented with each graph contains a variety of information about the regression model. The four columns of the table show (1) a listing of the independent variables used in the regression and the (2) coefficients, (3) standard errors, and (4) partial F-values related to those independent variables. Above the table, the r^2 value, the root-mean-square (RMS) value of the residuals, and the F-test value are given.

The variables and coefficients in the top table in figure 8 describe the linear equation for predicting the performance measure. For example, the first table prescribes the following formula for predicting rearward amplification (RA):

$$\begin{aligned} RA = & - 0.001089\, C_\alpha \\ & - 0.01813\, (WB2*WB3)^{1/2} \\ & + 1.183\, T/2H \\ & - 0.0000208\, (WB2*WB3) \\ & + 7.373 \end{aligned} \tag{4}$$

The statistical measures above the table indicate how well this regression model explains the observed variation in the rearward amplification values obtained from the simulation study. The r^2 value is the percent of this variation explained by the model. (A value of unity implies a perfect model.) That is, the regression model at the top of figure 8 explains 96 percent of the variation observed in rearward amplification. The residual RMS is the root-mean-square value of that portion of the variation not explained by the model. (A value of zero implies a perfect model.) That is, in this example, the remaining "noise" (scatter about the 45-degree line in the plot) has an RMS value of 0.066. The F value is the ratio of distributions, which serves to compare the portion of the variation explained by the model and that portion not explained. (A large F implies a good model. For the number data points in this analysis, F values in the range of 3 or 4 would generally imply high statistical significance.)

While the measures above the table apply to the whole model, the standard error and partial F values in the table relate to the statistical qualities of the individual variable in the model. The standard error, in relation to the coefficients, indicates the statistical significance of the variable. That is, if the standard error is proportionately much smaller than the coefficient, than the variable is highly significant. A ratio of 10 to 1 of the coefficient to the standard error is desirable. The partial F values roughly indicate the relative importance of the particular independent variable in determining the predicted value of the dependent variable. They are calculated in a manner similar to the F value above the table but relate to the contribution of the individual variable. Thus, if the partial F value is a large fraction of the F value, the variable is very important in the model. Variables whose partial F is a smaller fraction of F have less power in the model.

Some of the models shown in the eight figures are for A-train doubles only, and some are for both doubles and triples. Triples are excluded from some models for two reasons. First, the simple difference in the number of trailers in the double and triples often precludes a common solution for predicting their performance numerics. This holds especially for the measures derived from the lane-change (RTAC-B) maneuver and also for pulse-steer damping ratio. Another way to identify the models where this point is important is by the presence of trailer wheelbase parameters. The second reason largely superimposes on the first. The B maneuver typically generates a very severe response in the last trailer of a triple. The response of this unit becomes highly nonlinear, and the resulting performance measures appear to become rather chaotic (in a mathematical sense) with respect to vehicle parameter changes. (That is, small changes in parameters may result in large and disorderly changes in the response.) In those cases where inclusion of the triple in the regression analysis was not appropriate, prediction of the performance measure is provided for simply by giving the mean and standard deviation of the measure for the 15 variations of the triple examined.[2] These results are summarized in table 10.

Limitations of the Predictive Models

Before discussing the results for the individual measures, we note that all of these results are dependent on the specific matrix of vehicle parameters chosen for this study. While the matrix of vehicle configurations and the various parameter variations was rather large, it certainly was not all-inclusive, nor was it a *weighted* representation of the U.S. fleet. For example, it will be seen that tire-cornering stiffness is often the most important factor in predicting a performance parameter. But only three different tire variations were included in this matrix (although they did represent a rather broad range of tire properties). Also, if tire properties are important, it follows from the physics that axle loading should be important. But axle load was not varied substantially in this matrix. All vehicles were fully

2 Regression analyses performed on the triples results alone were uniformly unsuccessful in producing a regression model of substantive quality.

Table 10. A-train triple performance measures not included in the regression models

Performance Measure	*Mean Value*	*Standard Deviation*
Rearward amplification	3.647	0.679
High-speed transient offtracking, ft	3.573	1.127
Dynamic-load-transfer ratio	0.994	0.015
Damping Ratio in the RTAC-B maneuver	0.211	0.029
Damping Ratio in the pulse-steer maneuver	0.293	0.034

loaded in the recognition that this is generally the worst case. And, of course, these results are also dependent on the limitations of the simulation program used. As are all programs of this type, UMTRI's Yaw/Roll program is a simplified representation of the real thing. To the extent that effects not in the program influence real vehicle performance, these results are clearly lacking.

Estimating Rearward Amplification

Figure 8 shows three regression models for predicting rearward amplification. Starting at the top of the figure, we see that a model using just four variables—tire cornering stiffness, the product and square root of the product of the two trailer wheelbases, and the ratio of the half-track to cg height (T/2H)—a model with an r^2 value of 0.96 is obtained.

Of the four independent variables used, cornering stiffness is the most powerful (largest partial-F), but could be the hardest to obtain. In the second model, therefore, C_α is replaced by a surrogate, the binary true/false indicator for radial tires. For this particular population, that means the model can no longer distinguish between the more compliant new radial tires with full tread, and the relatively stiff worn radial tires. Naturally, the predictions suffer some, but this variable, remarkably, remains the most important.

Finally, in the last model, T/2H is replaced by track width alone. Since the variation in cg height in this population is far more significant than the variation in track width, this variable loses most of its significance, but it is left in the model since it does have some worth and is so easy to obtain.

Rearward amplification is one of the measures from the RTAC-B maneuver for which triples can not be lumped with doubles. The rearward amplification values for the 15 versions of the 28-foot triple studied had a mean value of 3.65 and a standard deviation of 0.68.

Estimating High-Speed Transient Offtracking

The presentation of figure 9 indicates that high-speed, transient offtracking is dependent on the same basic vehicle parameters as is rearward amplification. Tire-cornering stiffness is even more dominant, and the wheelbases of the trailers are important individually, not just combined as a product. The model yields a very good r^2 value of 0.97.

When the radial-or-bias binary variable (1 or 0) is substituted for cornering stiffness, the quality of the model suffers (r^2= 0.81). The graph shows that most of the loss is related to five specific conditions, which are the five doubles configurations equipped with the stiffer (worn) radial tires. That simply reemphasizes what a fundamental influence cornering stiffness has on this measure.

This result also points up a challenge to the general usefulness of the radial/bias surrogate. Only three types of tires have been used here. Any binary measure could fully represent two choices. If many tires had been used, the range of cornering stiffness among different radial tires may have made the binary measure appear less useful than it does here.

Finally, little is lost in the last model by replacing T/2H with track width. This is clearly expected since T/2H (as with four of the five variables) did not possess much authority in the model to start.

Again, since this measure comes from the RTAC-B maneuver, triples have not been included in the models. Transient offtracking of the 15 versions of the 28-foot triple studied had a mean value of 3.5 feet and a standard deviation of 1.13 feet.

Estimating Static Rollover Threshold

Figure 10 shows the regression models derived for predicting static rollover threshold. This is the first measure discussed for which it is appropriate to mix the results for doubles and triples. Note that the first two models, shown in the usual format with graphs, do pool the results of the triples with the those of the doubles. To make the point that follows, models derived from doubles data only are shown below solely in tabular format. Comparing results for the similar pooled and doubles-only models reveals that the coefficients vary less than plus or minus one standard error. (For example, the difference between the coefficient for T/2H determined with doubles alone and with doubles and triples combined is 0.8009 - 0.7809 or 0.0200. This is less than the standard error for T/2H from either of the tables.) Thus, it can be judged that there is no significant difference between the models, and the pooled models are appropriate for both doubles and triples.

The variables contained in the first model are certainly no surprise. They are T/2H, the well-known *rigid body* estimate of the rollover threshold, and suspension roll stiffness, the most important compliance property of the computer simulation model used. The model yields a respectable r^2 value of 0.90.

The second model drops the roll stiffness variable since it would not be generally available in the field, but no convenient surrogate is available to replace this variable. Nevertheless, the results show that T/2H alone is a useful predictor of the rollover threshold.

The user of this model is left with the need to determine H. In keeping with our method to this point, we should show a model based on track width alone. However, cg height is so basic to the mechanics of rollover that the model with track width alone is basically useless ($r^2 = 0.02$).

As mentioned previously, however, rollover threshold is the one performance measure examined for which the generic difference between doubles and triples configurations is not particularly significant. Thus, at the bottom of figure 10 we have excluded tabular results for regression analyses, which include the triple-trailer data. These results are very similar to those for the doubles and triples.

Estimating Dynamic-Load-Transfer Ratio (DLTR)

The best model for estimating dynamic-load-transfer ratio depends on the same vehicle parameters found to be important in predicting rearward amplification and static rollover threshold. (See figure 11.) Clearly, this is as expected since dynamic load transfer in the lane-change maneuver should be nearly a direct result of rearward amplification response and roll stability properties.

When cornering stiffness is replaced with the radial/bias binary, the r^2 value drops from 0.93 to 0.90. By our declared procedure, the next step would be to replace T/2H with T. (Although not shown, this yields $r^2 = 0.81$.) But this would essentially remove all the roll stability qualities (see the roll threshold discussion) leaving this prediction a virtual repetition of the rearward amplification prediction. (In fact, using only rearward amplification performance to predict DLTR results in $r^2 = 0.77$.) This notion is further illustrated in the third model of the figure. Here the previously derived, lower quality estimates of rearward amplification and rollover threshold are used to predict DLTR. The resulting r^2 is 0.91, which is nearly as good as the first model. Although not shown, if a similar model is generated using the *actual* rearward amplification and rollover threshold values, the results yield $r^2 = 0.95$.

Here again, the triples must be considered separately. For all but two of the triples studied, the third trailer (including dolly) response was so severe as to simultaneously lift all tires on one side from the pavement, i.e., DLTR = 1. (Three of these rolled over; the rest recovered.) The other two variations had DLTRs of 0.95 and 0.97.

Estimating Damping Ratios

Figures 12 and 13 display the regression models for predicting yaw-damping ratio for doubles, as derived from the lane change (RTAC-B) and pulse-steer maneuvers, respectively. In both cases, the wheelbase of the last trailer dominates, followed by the roll related properties of T/2H and roll stiffness. (The importance of these latter two parameters is almost surely embodied in the last trailer also. But that is not demonstrable here since, in this matrix, all trailer properties were always similar.) Yaw inertia, pintle hitch overhang, and tire-cornering stiffness are also shown to have small effects (the apparent lack of significant influence of cornering stiffness being quite surprising).

It is also noted that the regression models generally predict damping measured in the pulse maneuver better than they do damping measured in the lane-change maneuver. The lateral accelerations in the pulse maneuver are very low; the motions of all units of the train are relatively small and remain in the linear regime. This maneuver can, therefore, be expected to yield more orderly results, which are more predictable by this highly simplified method.

Again, the results for the triples are isolated. B-maneuver damping ratio averaged 0.2 with a standard deviation of 0.029. The mean for the p-maneuver ratio was 0.293 with a standard deviation of 0.034.

Estimating Low-Speed Offtracking

A simple means of predicting *steady-state*, low-speed offtracking has existed for some time in the form of the so-called Western Highway Institute (WHI) formula [8]. The low-speed offtracking measure used here, however, is not steady-state, but the transient maximum value occurring in a tight (approximately 32-foot radius) ninety-degree turn. Nevertheless, it could be presupposed that the WHI formulation could serve as a good basis for prediction. Thus, the constructed variable, WHI length, was used. From the form of the WHI method, this variable is defined as:

$$\text{WHI Length} = [\Sigma\, WB_i^2 - \Sigma\, OS_i^2]^{1/2} \tag{5}$$

where:

WB_i is wheelbases of the several units (tractor, semitrailer, and dollies), and

OS_i is several hitch point offsets (fifth wheel and pintle hitches).

Figure 14 shows that this single variable is an excellent predictor of the RTAC transient, low-speed offtracking measure used in this study. Note, there are separate models given for doubles and triples, however. Their coefficients values are similar, but significant improvement is obtained with separate models as compared to a single pooled model. Since the one independent variable is exclusively geometric, no simplification is warranted.

Estimating High-Speed Steady-State Offtracking

Earlier work [11] has shown that high-speed steady-state off-tracking is a function of low-speed offtracking and a term that, in this context, can be characterized as overall length divided by cornering stiffness.[3] Figure 15 shows that a regression model using this term and the WHI length provides a good estimate of high-speed offtracking for the doubles. A similar model, but without the WHI length (which proved to be statistically insignificant, presumably since it varies little within the set of triples) works well for the triples.

The second model shown substitutes representative values of 880 lb/deg and 560 lb/deg for the cornering stiffnesses of radial and bias tires (as loaded in this study), respectively. The vehicles using the stiffer (worn) radials become apparent in the graphical display. The quality of the model degrades significantly but still appears to yield a useful prediction.

Performance Contrasts, C-trains versus A-trains

In order to characterize the performance improvements that can be obtained by replacing A-dollies with C-dollies, the major portion of the large matrix of computer runs conducted on the A-train vehicles was repeated six times using the six variations of C-dollies. (The six types of C-dollies were identified in the *File ID* column of table 5.) The performance measures of the individual A-trains and C-trains were then compared to obtain performance improvement factors for the various C-dolly designs.

The matrix of C-train runs included all 15 variations (14 plus the baseline) of five of the six vehicle configurations. (Tables 3 and 4 identify the baseline condition and 14 variations.) The one configuration not included in this series was the turnpike double. The dynamic performance of this vehicle is so benign, even in the A-train configuration, that converting it to a C-train does not appear warranted. All of these 75 vehicles (15 variations of five configurations) were subjected to the RTAC-A, RTAC-B, and the pulse-steer maneuvers. All smaller set of vehicles was run through the RTAC-C maneuver. In this series of runs, only the five variations effecting longitudinal geometry (baseline plus the two variations of tongue length and hitch position) were included although the turnpike double was retained for this maneuver.

The three RTAC maneuvers plus the pulse-steer maneuver generate eight individual performance measures. These are static rollover threshold and high-speed steady-state offtracking from the A maneuver, rearward amplification, high-speed, transient offtracking, DLTR and damping ratio from the RTAC-B maneuver, the additional damping ratio from the pulse-steer maneuver, and low-speed offtracking from the RTAC-C

3 Here again, the limits of the study matrix shows. In the general application, this term also includes tire load in the numerator but is not included here since that parameter did not vary significantly in this matrix.

maneuver. In the end, only seven of these were processed. The C-trains uniformly exhibited high levels of yaw damping in the low-severity (i.e., linear-range) pulse-steer maneuver. The C-trains were so strongly damped in this maneuver that the post-processing algorithms had difficulty identifying the acceleration peaks needed to calculate damping ratio, and the measure was abandoned.

For the remaining seven performance measures, the relative performance of the A-trains and C-trains was examined by calculating both the ratio of A-train performance to C-train performance, and the difference between A-train performance and C-train performance for every individual vehicle and measure in the matrix. These two types of comparative measures, referred to herein as *A-C* and *A/C*, were then summarized by taking the means and standard deviations for the 15 vehicle variations within each performance measure/vehicle configuration/C-dolly type group. (The purpose here was, of course, to establish the hoped-for C-dolly *improvement factors* as the mean of one of these measures and to qualify the consistency of that factor by the standard deviation.) A complete listing of these summary comparisons is presented in table E-1 in appendix E. In most cases, the results presented in table E-1 show the ratio of performance to be a somewhat more consistent measure than the difference. The ratio measure will be favored here for all but the low-speed offtracking results.

Figure 16 presents example results from table E-1 in a graphical format. (Similar figures for all the performance measures also appear in appendix E.) The upper graph shows the results for rearward amplification; the lower graph presents results for damping ratio in the B maneuver. Both graphs show the ratio of A-train to C-train results (*A/C*) by vehicle and dolly type. As shown below each graph, the results for doubles are grouped to the left and the results for the triples are grouped to the right. Within each of these, the results from individual C-dolly types are also grouped. Of the six dolly types, the four to the left (under both doubles and triples) are the self-steering types (2C1, 2C2, 3C1, and 3C2); the two to the right are the controlled-steering types (2C3 and 3C3). The key at the top of the page indicates that the shaded bars present the range of the mean ± one standard deviation (for the 15 vehicle variations), and that four doubles configurations can be identified within each dolly grouping.

The rearward amplification plot is the strongest example of one type of result from the comparison analysis, namely, that changing from A-dollies to C-dollies produces a fairly orderly and predictable performance improvement. The lower plot is the best example of the second type of result, wherein the influence of C-dollies is small (i.e. the ratio *A/C* is near to one) and/or has a relatively large scatter.

The rearward amplification plot of figure 16 highlights the following qualities of the effects of C-dollies on this performance measure:

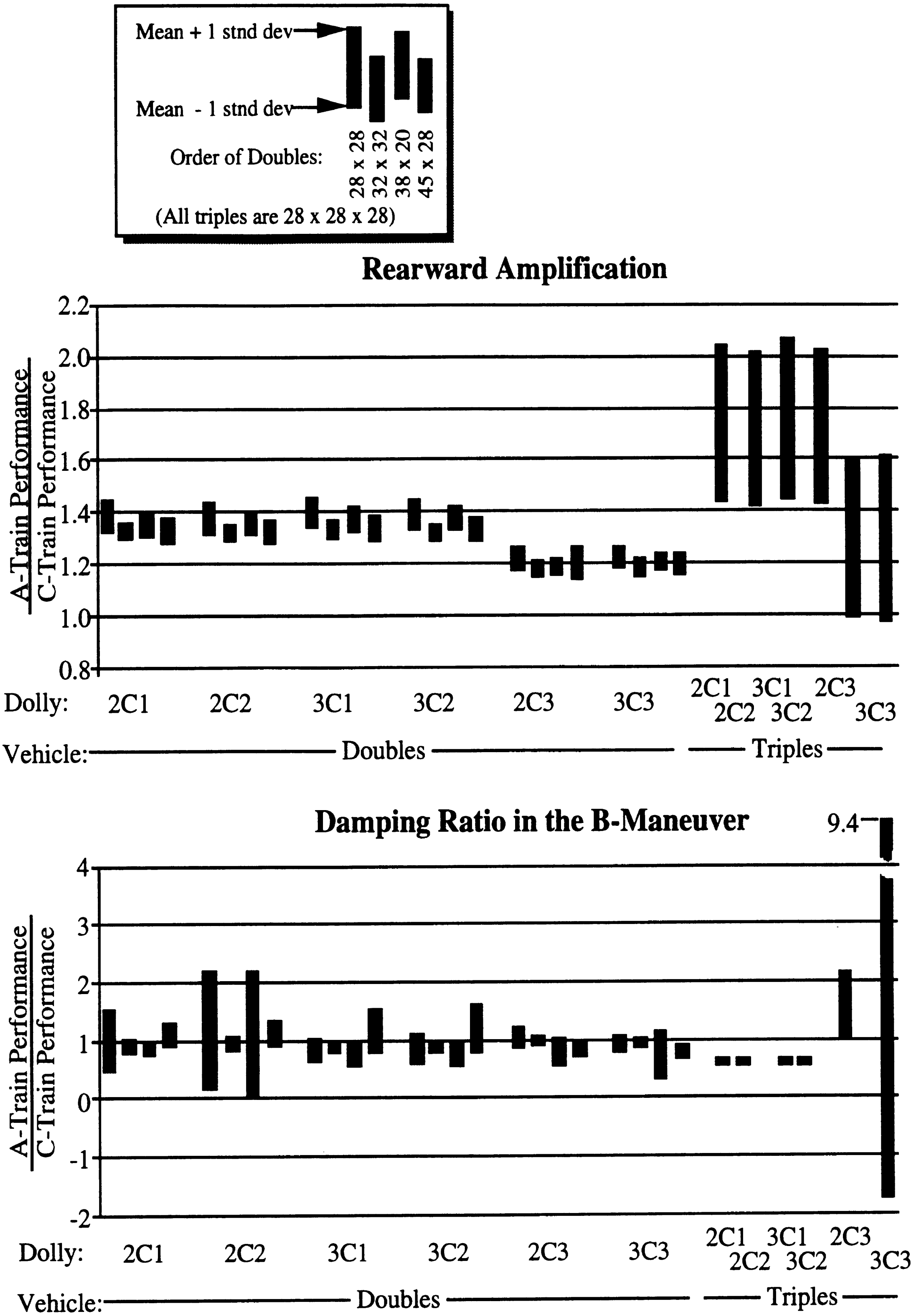


Figure 16. Two examples of the ratios of A-train and C-train performance measures

- When applied to doubles, the *self-steering* C-dollies improve (reduce) rearward amplification by a factor of approximately 1.35, with relatively little scatter resulting from either the different doubles configurations or the 15 parameter variations within configurations.[4]
- The same is generally true for the *controlled-steering* C-dollies, but the mean improvement factor is significantly less—approximately 1.20.
- For each of these two main C-dolly types, the mean improvement factor for triples is higher[5], but there *appears* to be much more scatter as a result of parameter variations for triples than for doubles.

In the last point, *appears* is emphasized because, in fact, the behavior of triples equipped with C-dollies is generally quite consistent. Rather it is the scatter in the performance of the *A-train* triples that produces the scatter in the ratio *A/C*. To explain, consider figure 17. This figure shows the mean plus and minus one standard deviation ranges (of the 15 parameter variations) for the rearward amplification of the triples, grouped by dolly type.[6] With this presentation, it becomes quite clear that the self-steering C-dollies produce very consistent, and relatively low, rearward amplification performance in the 28-foot triple. However, with either A-dollies or controlled-steering C-dollies, the results are more scattered, and rearward amplification is more severe. The higher scatter is largely a direct result of the higher mean. For example, the mean rearward amplification for the A-train triples is 3.65, implying a peak lateral acceleration of the third trailer of 0.55 g (based on a tractor peak acceleration of 0.15 in the RTAC-B maneuver). This is well into the nonlinear regime and represents very severe trailer motion. (In fact, 3 of the 15 variations of A-train triples rolled over in the RTAB-B maneuver.) The violent, nonlinear behavior of the third trailer results in a somewhat chaotic (in the mathematical sense) relationship between parameter variations and the rearward amplification measure, which is not present when the response is less severe.

The points presented above for rearward amplification can be restated nearly identically (using different numerical values, of course) for dynamic-load-transfer ratio and for high-speed, transient offtracking—the two RTAC performance measures closely related to rearward amplification. (See the relevant graphs in appendix E.) Other than the specific

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4 Regression analyses similar to those carried out for the A-train results could probably be undertaken to achieve a more precise description of the performance improvement achievable with C-dollies for this and the other measures to be discussed. Unfortunately, such analyses were beyond the resources of this project.

5 An extension of Fancher's linear analysis would suggest that the rearward amplification improvement factor for triples should be the square of that for doubles.[4,11,14] That is not quite achieved here, presumably due to the fact that the last trailer of the triples is well into the nonlinear range in the RTAC-B maneuver. Thus, the side force capability of the last trailer's tires is saturating, limiting lateral acceleration by a mechanism not as strongly in play at the second trailer.

6 Table E-2 of appendix E presents tabulated data of the type presented in figure 17. Data are presented for all seven performance measures and for the doubles configurations as well as for the triples.

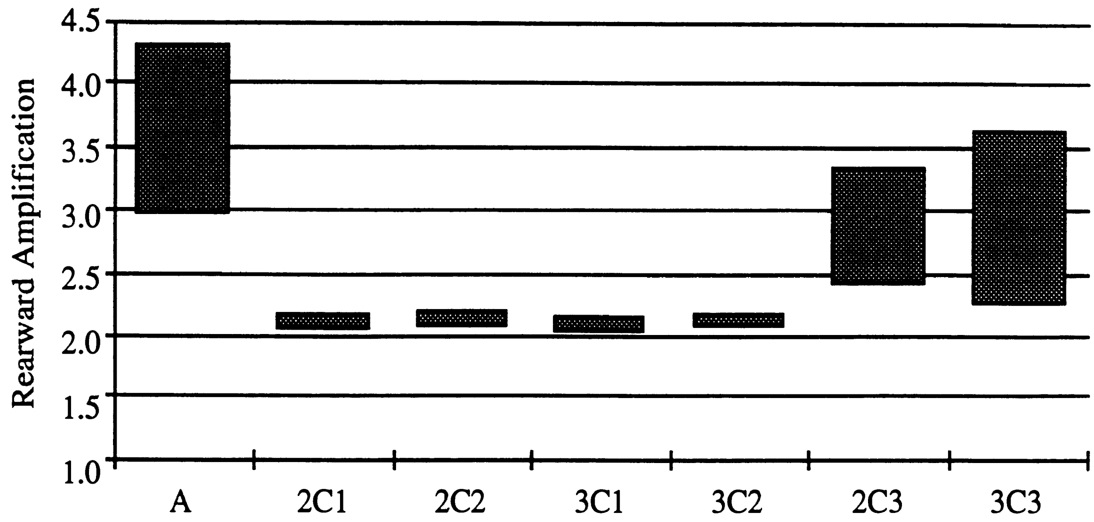

Figure 17. Mean ± one standard deviation ranges of rearward amplification of the 15 variations of the triple as a function of dolly type

numbers, the primary difference is that the mean improvement in offtracking for triples is less than for doubles and there is actually a small degradation when the controlled-steering dolly is applied to the triple.

The comparison plots (see appendix E) for the performance measures, which come from the RTAC-A maneuver—steady-state rollover threshold and high-speed steady-state offtracking—both show similar structure to those for the B-maneuver measures, but with considerably weaker C-dolly influence. Change in the steady-state rollover threshold using C-dollies instead of A-dollies is generally less than 5 percent, and less than a 10 percent change in high-speed steady-state offtracking is generally observed with C-dollies.

The influence of C-dollies on the damping ratio observed in the RTAC-B maneuver is shown in the lower portion of figure 16. (In this measure, C-dollies result in improved performance if the *A/C* ratio is less than unity.) This graph shows that, for *doubles*, C-dollies have a clear tendency to improve damping ratio (most of the bars are centered below a value of unity), but the tendency is not terribly strong or consistent. Conversely, for the *triple*, the *self-steering* C-dollies substantially and consistently improve the damping ratio. In this case, the average *A/C* ratio is 0.62. (Inverting this value implies an improvement factor of 1.61.) But on average, the use of controlled-steering C-dollies reduces damping in the B maneuver. In the case of the 3C3 dolly, three of the 15 variations of triples had virtually zero or slightly negative damping (and eventually rolled over). Since this value is in the denominator, the *A/C* ratio *blows up*, producing a very large standard deviation.

The overall quality of the results in figure 16 clearly favors the self-steering C-dollies over the controlled-steering C-dollies. The consistently better rearward amplification results for the self-steering design are important, of course, but the apparent ability for the self-steering approach to drastically reduce yaw damping in some applications is most significant. Low or negative damping is a very undesirable quality; having observed it in any vehicle configuration using this dolly argues for general caution in any application of the design approach. Further, the results of figures 16 and 17, and the similar presentations of appendix E, clearly indicate little distinction between the several variations of self-steering C-dollies examined, at least for the range of vehicle configurations and parameter variations examined.

One performance measure, low-speed offtracking, remains to be considered. Low-speed offtracking for the vehicles studied falls in the general range of 14 to 28 feet, but the change resulting from switching from A-dollies to C-dollies is only a foot or so. Thus, the *A/C* ratio is not a particularly sensitive means of examining C-dolly influence on this measure. Instead, the *A-C* measure is more effective. Figure 18 shows the *A-C* measure for low-speed offtracking of each individual vehicle examined. The results show a modest improvement in offtracking in most cases and a few cases with a slight degradation.[7] A tabulation of the data used to generate figure 16 appears in table E-3 of appendix E.

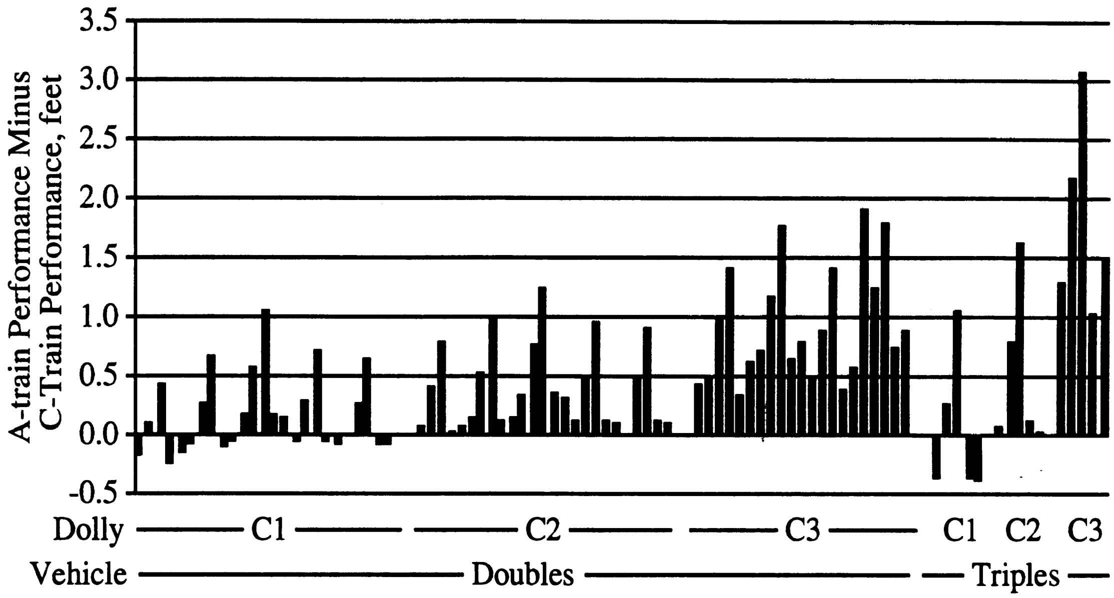

Figure 18. Individual A-C improvement factors for low-speed offtracking

Summary

The results discussed above lead to the general observation that it is appropriate to group the individual results according to dolly type and vehicle type. That is:

- For the measures studied, there appears to be little difference between the four versions of *self-steering* C-dollies considered or among the two versions of

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[7] These results depend, in part, on the fidelity with which the *driver-model* used in the *Yaw/Roll* simulation program follows the prescribed path. In fact, the *driver* is not a perfect controller and does wander slightly. This may account for the few negative results seen in figure 18.

controlled-steering C-dollies. Thus, it is reasonable to pool results into these two groupings.

- Similarly, it appears reasonable to pool the results into two groupings by vehicle type, one group for all doubles and one for triples.

Table E-4 of appendix E presents all of the *A/C* and *A-C* improvement factors pooled in this manner. The *A/C* portion of that table is presented here as table 11.

The following observations are based on the results shown in these tables (recognizing that *significant* performance differences are implied by average values differing from unity, and *consistent* influence is implied by small standard deviations).

- Predicting C-train performance by applying an improvement factor to baseline A-train performance is most appropriate for *doubles*. The relatively high scatter in the performance of *A-train* triples and the comparative orderliness of *C-train* performance, combined with the fact that there is only one basic triples configuration in common use, suggest a straightforward statement of triples' performance instead of the improvement factor approach.
- Self-steering C-dollies have a significant and relatively consistent advantageous influence on the three lane-change-related performance measures of rearward amplification, dynamic-load-transfer ratio, and high-speed, transient offtracking, when applied to both doubles and triples. Controlled-steering C-dollies also have

Table 11. *A/C* comparisons of A-train and C-train performance from pooled results

		Self-steer dollies (2C1, 2C2, 3C1, 3C2)		*Controlled-steer dollies (2C3, 3C3)*	
Performance Measure	*Vehicles*	*Average*	*Stnd Dev*	*Average*	*Stnd Dev*
Steady-State Rollover Threshold, g	All doubles	1.0119	0.0207	0.9823	0.0183
	All triples	1.0226	0.0244	0.9595	0.0166
Steady-State Hi-Speed Offtracking, feet	All doubles	1.0400	0.0492	0.9408	0.0226
	All triples	1.0686	0.0588	0.9643	0.0177
Rearward Amplification	All doubles	1.3486	0.0557	1.1973	0.0460
	All triples	1.7300	0.3064	1.2926	0.3139
Dynamic-load-transfer ratio	All doubles	1.9693	0.1213	1.8397	0.1152
	All triples	2.7023	0.1860	2.2907	0.2612
Transient High-Speed Offtracking, feet	All doubles	1.4898	0.0975	1.1660	0.0739
	All triples	1.2026	0.2784	0.8425	0.2875
B-Maneuver Damping Ratio	All doubles	0.9583	0.4472	0.8815	0.2398
	All triples	0.6199	0.0791	2.2089	3.4824
Low-Speed Offtracking, feet	All doubles	1.0150	0.0204	1.0481	0.0251
	All triples	1.0161	0.0394	1.1160	0.0570

an advantageous influence on these three measures, but it is weaker and less consistent.

- C-dollies do not have a consistent influence on steady-state rollover threshold, high-speed steady-state offtracking, or damping ratio in severe maneuvers (i.e., the B-maneuver damping measure), except for the case of damping for triples using *self-steering* C-dollies. However, use of C-dollies does produce high damping in the response to small disturbances (i.e., the pulse-steer damping measure).
- C-dollies produce a modest improvement in low-speed offtracking for doubles and triples relative to A-dollies.

The specific improvement factors and performance figures (and their standard deviations) which, following from the first and second of these four points, are particularly useful are summarized in tables 12 and 13.

Table 12. Useful *A/C* improvement factors for doubles

	Self-steering dollies		*Controlled-steering dollies*	
Performance Measure	*Average*	*Stnd Dev*	*Average*	*Stnd Dev*
Rearward amplification	1.3486	0.0557	1.1973	0.0460
Dynamic-load-transfer ratio	1.9693	0.1213	1.8397	0.1152
High-speed, transient offtracking	1.4898	0.0975	1.1660	0.0739

Table 13. Performance levels for C-train triples

	Self-steering dollies		*Controlled-steering dollies*	
Performance Measure	*Average*	*Stnd Dev*	*Average*	*Stnd Dev*
Rearward amplification	2.1057	0.0681	2.9086	0.6019
Dynamic-load-transfer ratio	0.3700	0.0286	0.4430	0.0800
High-speed, transient offtracking	2.9872	0.6161	4.4976	2.4210
B-maneuver damping ratio	0.3385	0.0359		

Ancillary Performance Issues

Two additional performance issues are addressed in the subsections below. These two issues are (1) the stability of the combination unit during backing and (2) the loads place on C-dollies and couplings during various maneuvers.

The Stability of A- and C-Trains While Backing

The issue of stability while backing constitutes one of the domains in which C-dollies offer an advantage over A-dollies. This advantage makes it feasible to back up the assembled C-train, for example at a loading dock. The advantage has been quantified during this study by means of computer simulations to be described in this section.

The basic approach was to simulate each vehicle backing through the same maneuver so as to compare performance. The simulations were carried out using a yaw/roll model for multitrailer vehicles that includes the ability to travel in reverse. The dolly designs included A-dollies, self-steering C-dollies, and controlled-steering C-dollies. The steering mechanism of the self-steering C-dollies was assumed to be locked on center so that the dolly wheels could not steer. The wheels of the controlled-steer dollies steered according to the same function used in forward travel. The following combination types were used to study the performance of each dolly design: doubles combinations having successive trailer lengths (in feet) of 28 and 28, 32 and 32, 38 and 20, and 45 and 28, and a triples combination with three 28-foot trailers. In keeping with the observation in the previous section, that the 45x45 A-train is so benign that it is not a candidate for C-dollies, this vehicle was simulated only with an A-dolly.

The simulated maneuver consisted of backing each vehicle a short distance at low speed with a small value of right, then left, steer angle at the tractor, followed by a sustained portion of straight, zero-steer movement. The initial steer input used for all the vehicles is shown in figure 19.

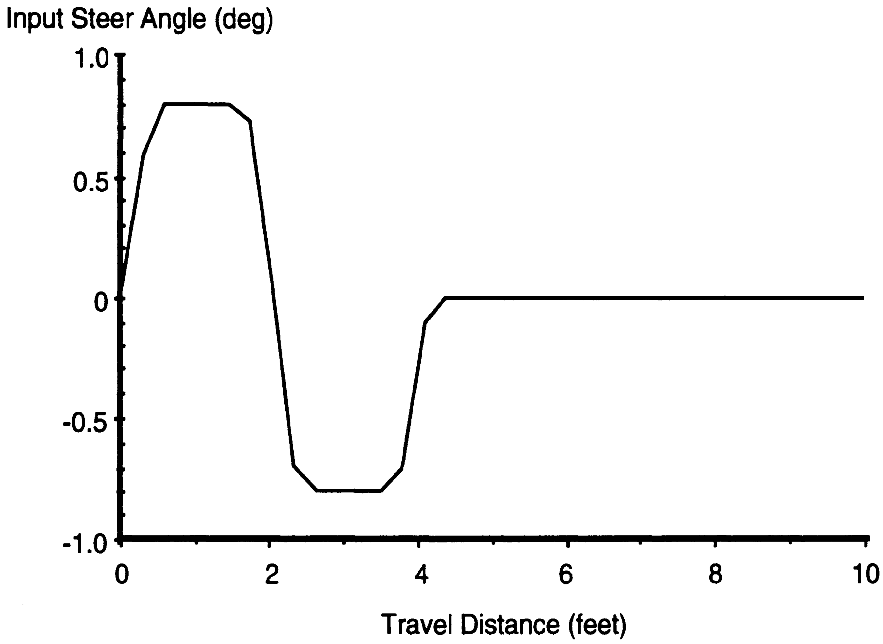

Figure 19. Steer input for backing maneuver

The small steering inputs at the start of the maneuver introduce small yaw articulations at all hitches. As the maneuver proceeds with no further steering, these angles diverge since the backing vehicle with no driver control is, of course, an unstable open-loop system. In a general sense, the rate at which yaw articulation diverges is a measure of the level of instability.

Two specific measures of divergence were used to compare vehicle performance during backing. The first was defined by the distance of travel beyond which the rearmost axle of the vehicle combination reached a value of lateral offset (relative to the projected straight path of the tractor) equal to a specified amount. The rationale for selecting this measure was that at a certain lateral offset, the driver will notice that the vehicle has reached an undesirable condition and will typically stop backing further. The second measure was defined by the travel distance needed to double the lateral offset once it reached a defined minimum value. This measure served to compare how quickly each backing vehicle would reach an unacceptable condition once it was already diverging from a straight line. The measure is independent of the initial disturbance. The travel distance used to double lateral offset is analogous to the *doubling time* measure commonly used in control system theory for characterizing monotonically divergent systems. Both of these chosen measures constitute representations of the stability (or instability) quality of the *open-loop* vehicle. In other words, they do included the actions of a driver who could potentially *close* the loop and stabilize the system. These measures do provide, however, an assessment of the relative difficulty a driver would have in keeping the different vehicles stable while backing.

Figure 20 shows the travel distance to reach a lateral offset of 2 feet for all of the

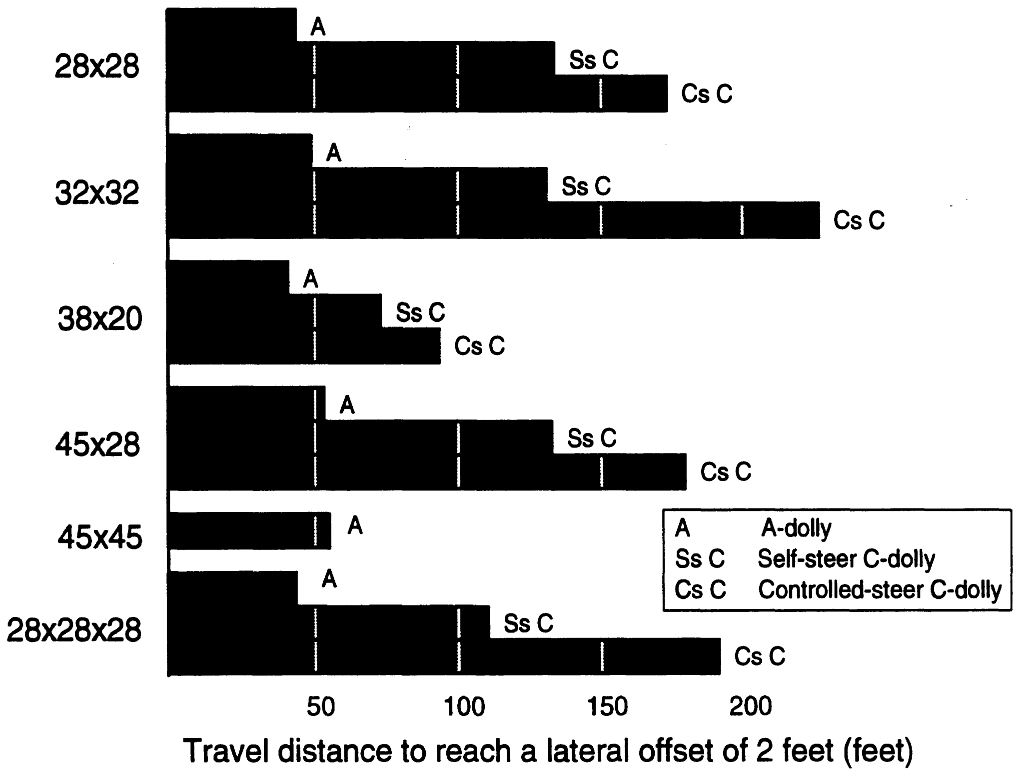

Figure 20. Stability in backing for various vehicle types: two-foot lateral offset measure

vehicle combinations and dolly types. The figure demonstrates that, as a group, the combinations with C-dollies are much more stable in backing than those with A-dollies. That is, they back much further without an undesirable lateral offset. We also note that the controlled-steer C-dollies are superior to self-steer C-dollies.

Data representing the doubling distance measure of the same cases are demonstrated in figure 21. Again, we see that both innovative dolly types are superior to A-dollies in facilitating the backing process. The reasons for the measured differences in stability of the various configurations will be discussed below.

Although our primary interest, here, is in the influence of dolly type, it is interesting to note the influence of trailer length on stability during backing. In particular, the data show that doubles combinations with shorter pup trailers diverge from their projected straight path more rapidly than those with longer pup trailers. Further, the tendency to diverge rapidly is exacerbated with mixed-length trailers, as in the case of the 38x20 combination, where the pup trailer is significantly shorter than the lead trailer.

A-dollies

Figures 20 and 21 demonstrate that vehicles with A-dollies are less stable in backing than vehicles equipped with self-steer or controlled-steer C-dollies. The property that distinguishes the behavior of an A-dolly in backing from that of the C-dollies is that it

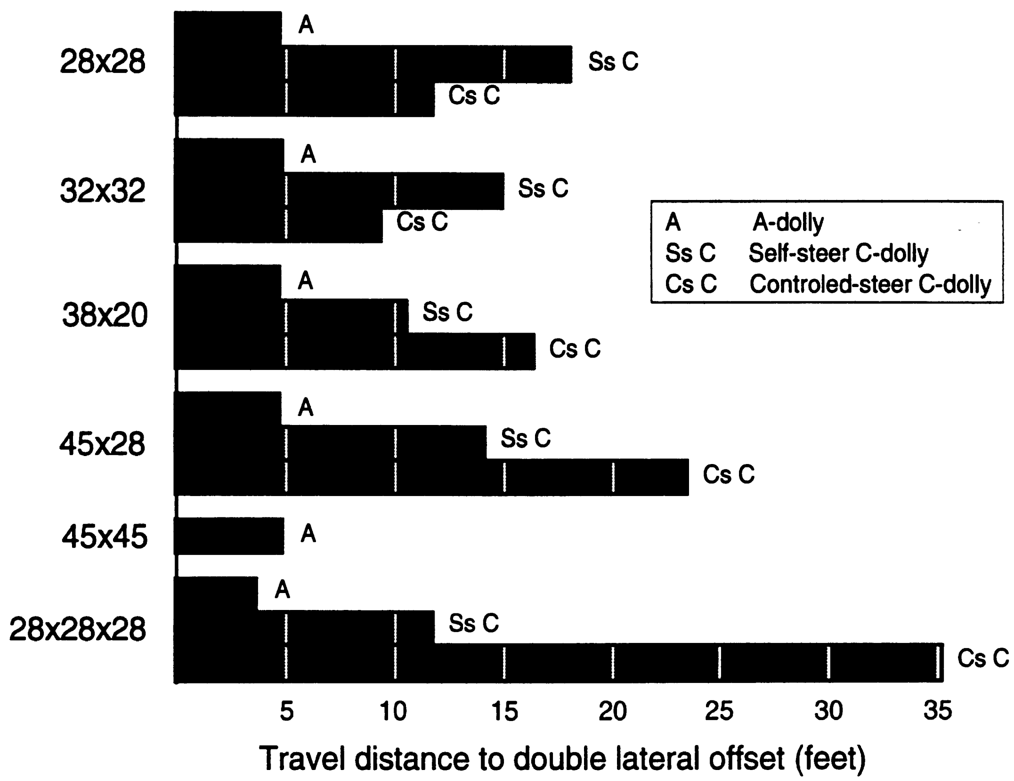

Figure 21. Stability in backing for various vehicle types: lateral offset doubling measure

permits free articulation in yaw at the coupling between itself and its preceding trailer. The yaw freedom derives from being connected to the towing trailer by means of a single pintle hitch. The freedom to move in yaw at the hitch causes the A-dolly itself to behave as a semitrailer of very short wheelbase. Since the rate of divergence of a semitrailer in backing is proportional to the inverse of its wheelbase, the short A-dolly contributes powerfully to the instability of a multitrailer combination.

By way of illustration, figure 22 shows a 28x28-foot A-train double that has backed into a limiting condition within a rather short distance. In this case the A-dolly, acting as a short trailer, reached a high articulation angle before either of the 28-foot trailers diverged significantly from a straight backward path. The behavior shown here is typical of all the A-train doubles simulated in this study. In the A-train triples case, both dollies acted as short trailers and reached a high articulation angle in a short distance. The second dolly of the triples reached a high articulation angle more rapidly than the first because its towing trailer was guided off the straight path by the first dolly.

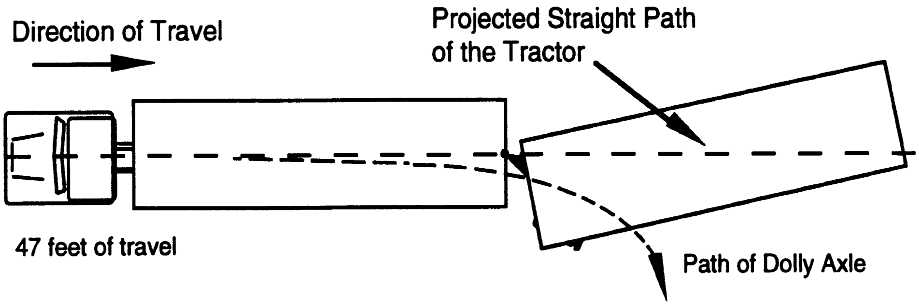

Figure 22. Backing of 28x28-foot A-train doubles

<u>The Self-steer C-dolly</u>

In contrast to A-trains, C-train doubles act as two serial semitrailers in backing, with the dolly incorporated as an extension of the first trailer by means of the double drawbar connection. This arrangement causes the C-dolly and first semitrailer to act as one long (and thus rather stable) trailer. When backing a C-train with a self-steering C-dolly, the axle steering mechanism of the dolly must be locked on center to prevent divergent steer behavior due to negative caster effect. Thus, the wheels of the simulated C-dolly were held straight with respect to the dolly when it was backed.

Figure 23 shows a 28x28-foot C-train double with a self-steering C-dolly that has backed from an initial articulation angle until an undesirable condition was reached. With the steering of the dolly locked, the C-dolly and first trailer were moved along the projected straight path of the tractor. The second trailer, however, slowly diverges from the straight path. As shown in the left portion of the figure, the articulation angle between the dolly and the pup trailer becomes significant after the vehicle has been backed for a while. If the

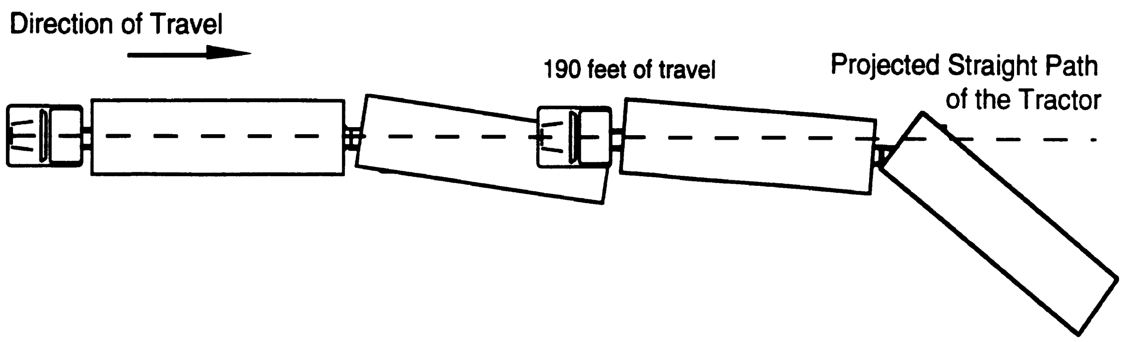

Figure 23. Backing of 28x28-foot C-Train doubles with a self-steering dolly

vehicle continues to back, the lead trailer and the C-dolly remain relatively straight, while the pup trailer ends up at a large articulation angle relative to the dolly.

All of the doubles combinations simulated with self-steer C-dollies showed this type of behavior. The effective extension in the length of the first trailer, together with the fact that the locked axle of the C-dolly forms a wide-spread tandem pair with the first trailer's axle, yields a semitrailer package that is quite resistant to yaw motion. These effects make the first trailer more stable during backing than the second. Thus, the second trailer shows a high articulation angle and lateral offset before the first trailer begins to diverge. In the case of the triples combination with a self-steering C-dolly, the third trailer acts in the same manner as the second trailer of the doubles combinations described above.

Controlled-steer C-dollies

As mentioned in the discussion of self-steering C-dollies, C-trains permit no yaw articulation at the coupling between the dolly and the towing trailer. The distinguishing property of the controlled-steer C-dollies in backing is that, unlike self-steer C-dollies, they are able to steer as they back. The steer angle of the dolly tires is a function of the articulation angle between the dolly and the following trailer. The ability to steer in this manner contributes stability to the vehicle during a backing maneuver.

Figure 24 shows a 28x28-foot C-train double with a controlled-steer C-dolly that has

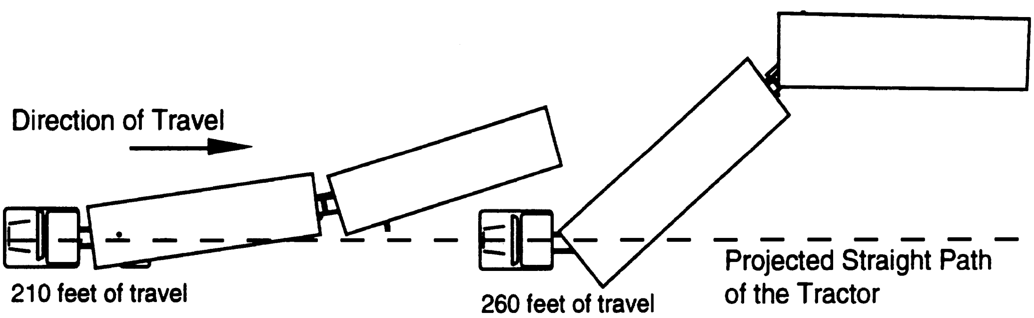

Figure 24. Backing of 28x28-foot C-train doubles with a controlled-steering dolly

backed from an initial articulation angle. As the vehicle begins to back, articulation angle between the dolly and the pup trailer grows. As a result, the dolly wheels steer in a manner tending to straighten out the articulation, as shown in the left portion of the figure. Eventually, the dolly steers too far and the pup trailer articulates in the other direction. Although the steer angle of the dolly reverses as the articulation angle changes polarity, the reverse articulation angle grows too rapidly for the dolly to compensate. This results in the folding effect shown in the right portion of figure 24.

Loading Demands Placed on C-Dollies and Hitching Hardware

The additional constraints that come into play when an C-dolly replaces an A-dolly (that is, the constraints on yaw and roll motion at the connection to the towing trailer) imply substantial new loads. Indeed, we have observed here that the introduction of C-dollies substantially alters the motion of rearward placed trailers. Altering the motion of large, heavy objects obviously requires some large change in forces or moments. Thus, the simple observation that C-dollies appear to make a substantial difference in vehicle behavior suggests we should expect substantial new loads.

The single-point hitch of the A-dolly results in the development of three forces at the hitch point between the dolly and its towing trailer. Figures 25 through 27 illustrate these three forces and label them as F_x, F_y, and F_z in accordance with their direction. The longitudinal force (F_x) resulting from either towing or braking forces is typically the most severe. The largest vertical forces (F_z) typically occur during braking and result from the pitch moment placed on the dolly by the combined action of fifth wheel overrun forces and brake forces at the dolly tires. The lateral tongue force (F_y) on the A-dolly is typically small. That is, the wagon-tongue-steering mechanism is very effective and requires only relatively small lateral forces to steer the dolly axle. The large lateral forces needed to actually motivate the trailer are developed at the dolly tires. In the C-dolly the elimination of yaw articulation establishes a whole new situation in which the lateral forces at the hitch are no longer just steering forces but are much larger forces involved in directly controlling trailer motion.

For the C-dolly, longitudinal and vertical loads, *which result from straight-ahead towing and braking,* are essentially the same as they would be for the A-dolly, except that they may be shared by two hitches. However, this potential for reducing individual hitch loads pales in comparison to the new loads imposed due to the new yaw and roll motion constraints.

Figures 25 and 26 illustrate the moments between the C-dolly and the towing trailer. These are represented by the yaw moment, M_z, and the roll moment, M_x. The absence of either yaw or roll motion results in the development of large yaw and roll moments. In practice, of course, these moments actually exist as a force *couple*, that is, a pair of forces acting in opposite directions at the two hitches. The yaw moment actually exists as a pair

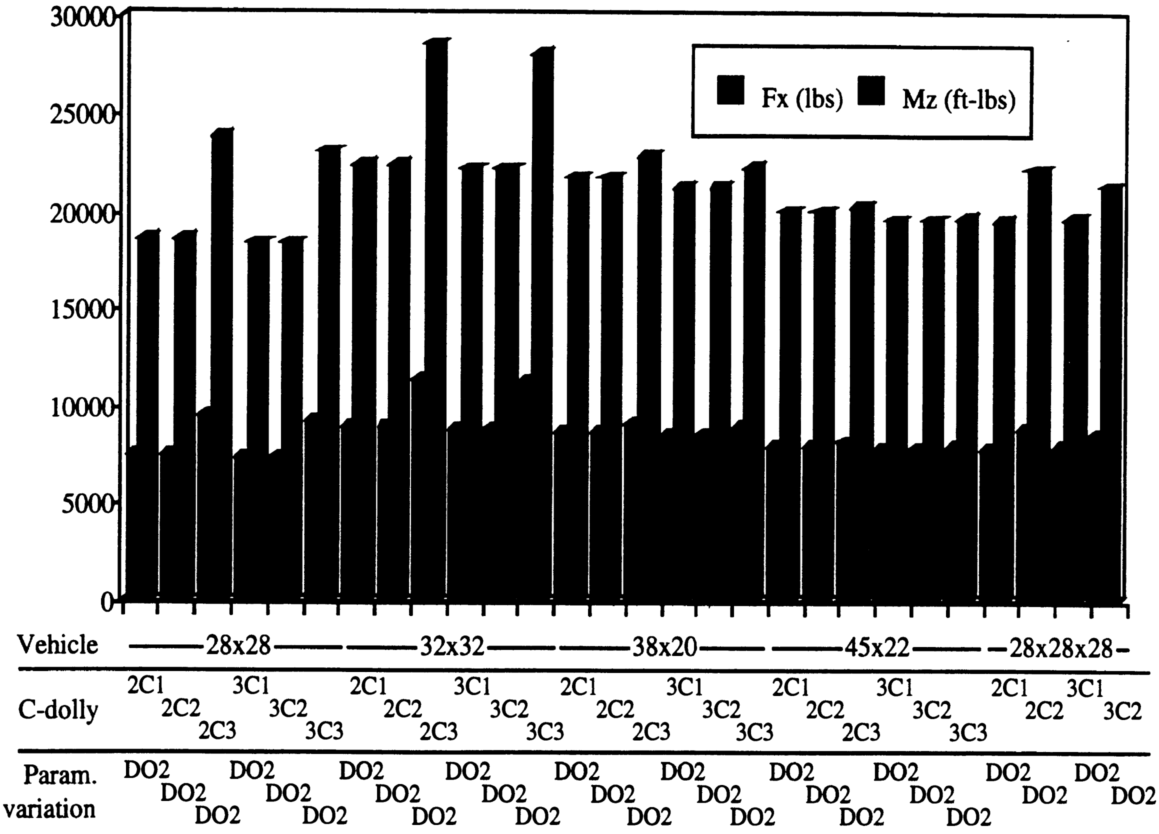

Figure 25. Peak longitudinal force, Fx, and yaw moment, Mz

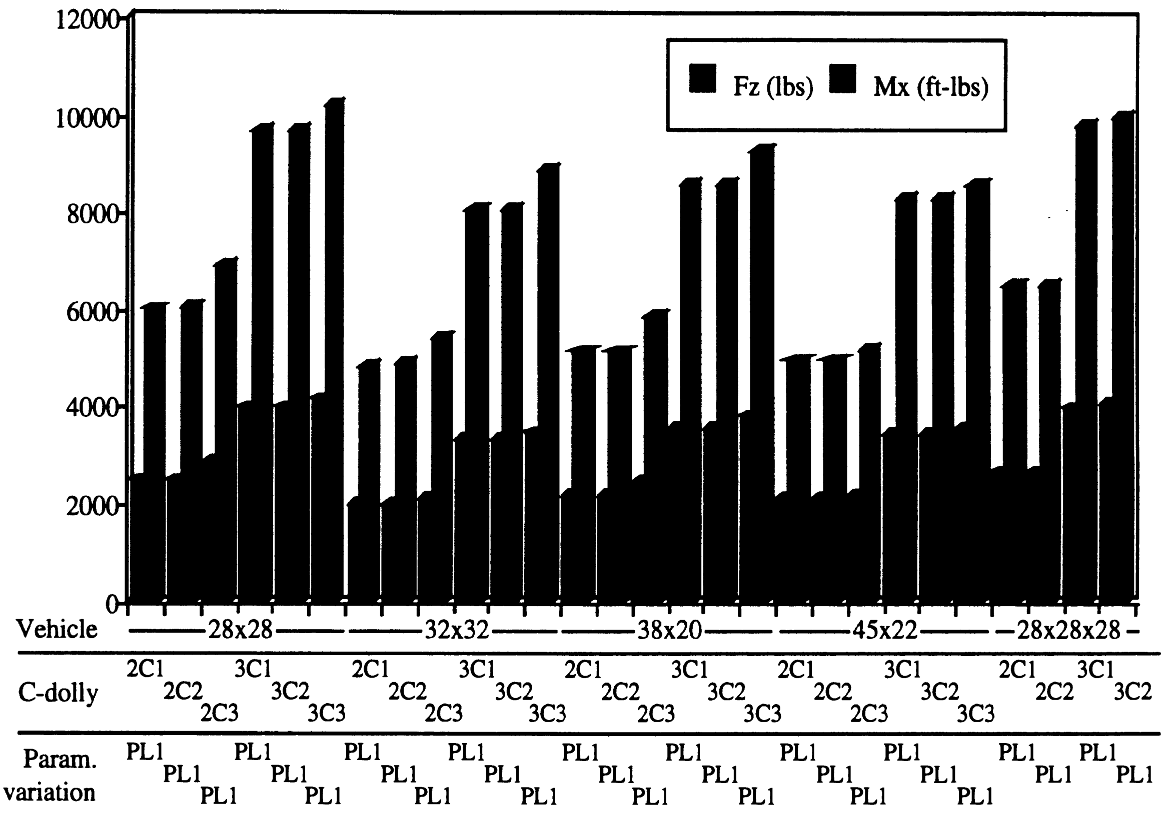

Figure 26. Peak vertical force, Fz, and roll moment, Mx

of longitudinal forces acting in opposite directions (F_{x1} and F_{x2}), and the roll moment exists as a pair of vertical forces (F_{z1} and F_{z2}).

The forces and moments presented in this section are the peak values of lateral force (F_y), yaw moment (M_z), and roll moment (M_x) observed during all of the simulated maneuvers conducted in which the simulated vehicle *did not* rollover. Presented along with peak values of M_z and M_x, are the related peak values of the individual longitudinal (F_x) and vertical (F_z) hitch forces that would make up the force couple needed to develop M_z and M_x, respectively, given a 30-inch spread between the hitches. These forces and moments could serve as estimates of the *minimum* level of additional loads (over and above those normally experienced in A-trains) for with the hitches, frames, and fastening hardware of C-dollies and towing trailers should be designed. It should also be noted, that previous research and publications on the subject have reported higher loadings than these, and have recommended higher minimum design loads.[1,9,17,18,19]

As expected, the largest hitch loads observed in this project occurred in the dynamic lane-change maneuvers. Thus, all of the peak loading values reported here come from the RTAC-B maneuvers. (These maneuvers are discussed in appendix A.) For each vehicle, dolly, and parameter variation, hitch loadings during the three different RTAC-B maneuvers were scanned to capture the peak forces and moments in each maneuver. The complete set of these results is reported in appendix F. For the triple combinations, individual results for each of the two dollies are reported.

An abbreviated set of the hitch forces and moments results from appendix F are given in the three figures which follow. Each figure presents the greatest load experienced by each combination of vehicle configuration and dolly type. The particular parameter variation condition under which that load was developed is identified. (Appendix F includes the peak loads for *all* parameter variations. Also, see table 4 for the parameter variation code definitions.) Figure 25 shows peak yaw moment (M_z) and the associated peak longitudinal hitch force, F_x. Figure 26 presents peak roll moment (M_x) and the associated peak longitudinal hitch force, F_z. Peak lateral force, F_y, is given in figure 27.

Some general observations that can be drawn from these figures follow.

- From figure 25, the largest yaw moments, and related longitudinal hitch loads, occur with the longer tongue length dolly (DO2). This is true regardless of vehicle configuration or C-dolly type. Since the yaw moment is generally a result of lateral force from the trailer sprung mass acting at the dolly fifth wheel, this finding is no particular surprise.
- Figure 26 shows that the highest levels of roll moment and related vertical hitch loads occur with the high center-of-gravity loading condition (PL1). Again, this is no surprise since the roll moment is generated by the relative roll motions of the two trailers.

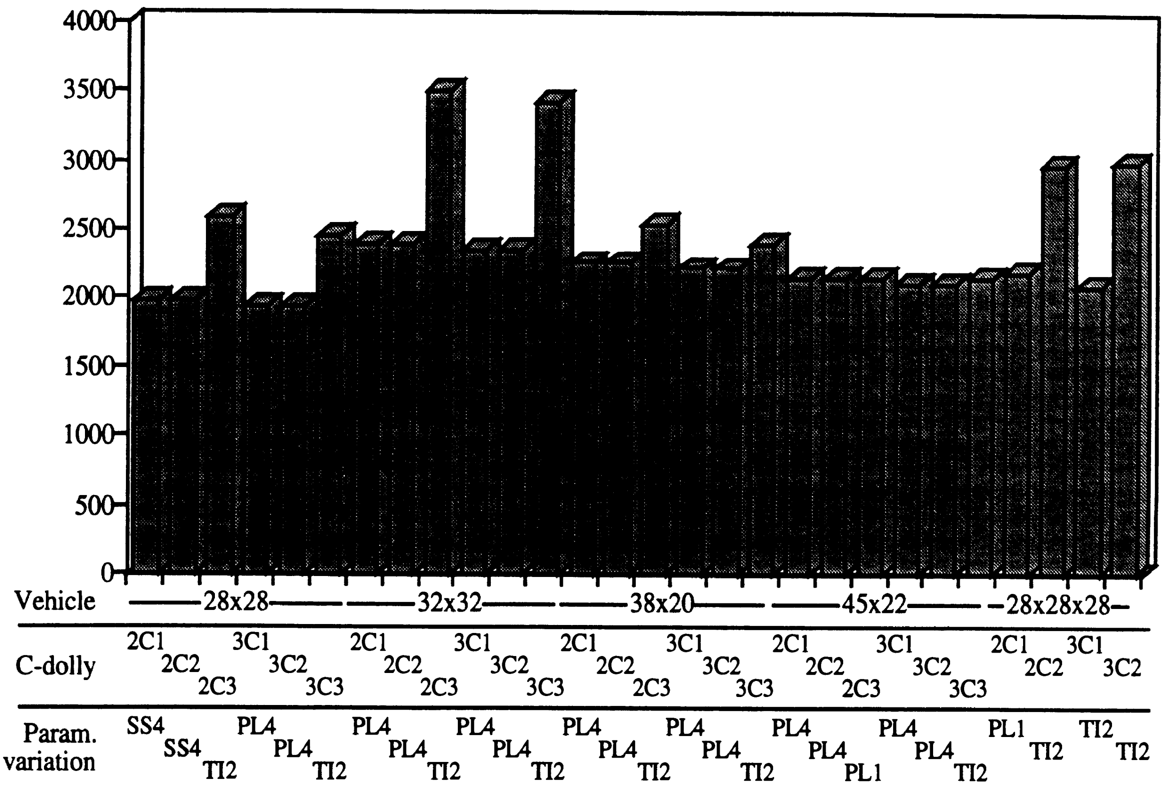

Figure 27. Peak lateral force, Fy

- From all three figures, it can be seen that loading is generally more severe for the controlled steering dolly (C3) than for the self-steering dolly types (C1 and C2). This seems in line with the earlier finding that the self-steering dollies suppress rearward amplification somewhat better than do the controlled steering dolly.

Economic Analysis

The issue of dolly economics addresses both the benefits gained from C-dollies in reducing traffic accidents and the costs to be borne from the purchase and operation of such equipment. The presentation is in two parts, with the *bottom line* tradeoff between benefits and costs appearing at the end of the second part, titled *Costs to be Borne.*

Accident Reduction Benefits Due to Innovative Dollies

The objective of this portion of the study is to determine the safety benefits of an innovative C-dolly, employing existing statistical information on truck accidents. All currently available accident data on multitrailer combinations represent almost exclusively A-dolly equipment; therefore, it is impossible to measure directly the safety improvements to be expected from widespread conversion to C-dollies.

Nevertheless, an appealing methodology of accident analysis arises from the observation (based upon engineering analyses and full-scale tests) that C-dollies improve the stability of double-trailer combinations (called doubles in this discussion) so that they approximate the stability level of tractor-semitrailers (i.e., singles). The most important dimension of this improvement is the additional resistance to rollover provided by C-dollies. That is, C-dollies afford lateral and roll constraints between successive trailers that are roughly equivalent to the constraints afforded by fifth-wheel coupling between a tractor and semitrailer. Thus, since doubles using the new dollies respond similarly to singles in accident situations, accident data that have been collected on the common tractor-semitrailer combination can serve as a convenient surrogate for data actually showing the accident experience of double combinations using the innovative dolly. Accident data related to doubles become the reference data for multitrailer combinations equipped with A-dollies.

The accident analysis is presented fully in appendix G and is divided into three parts. The first section describes the data sources that have been employed. The second section compares the accident experience of singles with that of doubles. Accident rates are compared for different operating environments, such as type of highway and day and night operation. Differences in how singles and doubles operate, as well as environments where doubles are overrepresented, are identified. A particular focus is accident types that should be helped by the innovative dolly. In the final section, the economic benefit that should be expected from C-dollies, is estimated and expressed as the dollar value of accident reductions.

Three data sets—two accident files and one travel file—were used to estimate accident rates and accident frequencies for singles and doubles. The accident files derive from the *Trucks involved in Fatal Accidents* (TIFA) file, produced and maintained by UMTRI, and the *General Estimates System* (GES) file, developed by the National Highway Traffic Safety Administration (NHTSA). TIFA data were used covering the years 1980 through 1988, providing the desired national census on all fatal accidents involving heavy duty trucks in the U.S. The file provides extensive information on vehicle configuration, as well as very accurate accident counts. GES is a sample file covering all levels of accident severity for both singles and doubles, allowing the analysis to be expanded beyond fatal accidents. The accident files are fully described in appendix G.

The travel data used to calculate accident rates are from the UMTRI effort to document truck usage called the *National Truck-Trip Information Survey* (NTTIS). The data from NTTIS provide detailed estimates of travel broken down by vehicle type, road type, area of operation (urban or rural), and time of day. The use of the NTTIS file, together with the nationally representative accident files, allows the calculation of accident involvement rates for selected vehicle types, on a per-mile-traveled basis.

Data files from FHWA's Office of Motor Carriers (OMC) and the National Accident Sampling System (NASS), developed by NHTSA, were used primarily to estimate the economic benefits of an improved dolly. The OMC file has information on costs of different types of accidents. These figures are used to calculate one part of the economic benefits of reducing or eliminating certain accidents. The NASS file is also used in that section to estimate the dollar savings that associate with injury severity.

Recognizing that the underlying assumption behind the entire analysis is that C-dolly-equipped doubles will exhibit an accident rate rather like that of tractor-semitrailers, it is useful here to discuss briefly a sample of the results that show the contrast between doubles and singles. Table 14[8], for example, shows accident rates, normalized to the total number of accidents for both singles and doubles, by road type for fatal involvements where truck rollover occurred. The percentage columns for both singles and doubles show the portion of travel in each category. The mile totals are annualized; fatal involvement numbers represent totals over the time covered by the data files. The involvement rate column, at the right, is determined by dividing the percent involvements by the percent of travel. The involvement rate figure allows direct comparison of a particular category to the population. Rates higher than 1.0 are overinvolved, less than 1.0 are under involved. Overall, the fatal rollover rate for doubles is significantly higher than that for singles, 1.20 compared with 0.99. Clearly, the population of doubles as currently configured have greater tendency to roll over than singles. On limited access roads the respective rollover rates are closer—0.70 for doubles compared with 0.61 for singles. On other types of roads, doubles exhibit the much higher rollover rate, 2.49 compared to 1.52.

Table 14. Travel, rollover fatal involvements, and involvement rates by road type, singles and doubles—NTTIS and 1980 through 1986 TIFA data

Road Type	*Miles (10^8)*	*Percent*	*Fatal Involvement*	*Percent*	*Involvement Rate*
			Singles		
Limited Access	193.97	55.20	1,239	33.6	0.61
Other	138.30	39.30	2,200	59.7	1.52
Single Subtotal	332.28	94.50	3,439	93.4	0.99
			Doubles		
Limited Access	13.96	4.00	103	2.8	0.70
Other	5.40	1.50	141	3.8	2.49
Double Subtotal	19.35	5.50	244	6.6	1.20
Grand Total	351.63	100.00	22,063	100.0	1.00

8 Table G-7 from appendix G. Numbers may not add directly due to rounding.

Table 15[9] uses data from the combined 1988–90 GES file, providing a view of property-damage-only (PDO) accidents. The table shows that a higher proportion of PDO accidents involving doubles (over 8 percent) are rollovers than is the case with singles (3.7 percent). This result appears to indicate that rollover, primarily of the rear-most trailer, is the mechanism that causes doubles to be in this category. That is, research on the dynamics of conventional doubles shows that in rapid steering maneuvers the rearmost trailer tends to amplify, or exaggerate, the motions of the forward units (rearward amplification), causing a crack-the-whip response that leads to rollover of the last trailer. Since it is less likely that an injury or fatality will accompany such an event, confirmation of the rearward amplification problem should show up prominently in PDO data—and it does.

Table 15. Combination vehicle involvements by rollover and number of trailers property-damage-only accidents—1988 through 1990 GES data

	Single		*Double*	
No Roll	306,107	96.3	4,749	91.8
Rollover	11,798	3.7	427	8.2
Total	317,905	100.0	5176	100.0
Number of Cases	2,956		93	

In terms of total rollover experience, the analyses in appendix G explains how the numbers in table 15 are modified by the more reliable data in the TIFA files to show that approximately 305 doubles rolled over in PDO accidents each year. If doubles rolled over at the same rate as singles, there would be 137 PDO rollovers, thus eliminating 168 rollovers of this type. Additionally, the analyses show that property damage costs can be saved by avoiding another 128 double rollovers per year that have previously incurred injury or loss of life. As a total savings in property damage, then, it is concluded that a C-dolly could prevent 296 (168 plus 128) rollovers, at an estimated property damage cost of $2,918,481.

The dollar value of the injuries and fatalities due to double rollovers that could be avoided with C-dollies can be stated by direct costs[10] equal to $3,874,374. Beyond the direct costs many investigators have sought to quantify the social costs that are implicit with the *pain and suffering* outcome of human casualties. By quantifying what people would be *willing to pay* to avoid a given injury, a dollar value can be placed on the intangible

9 Table G-15 from appendix G.

10 Direct costs include medical care and emergency services, lost wages and household production, costs for workplace disruption, insurance costs, and legal proceedings.

component of injury. If these social costs, which include both the direct and the pain and suffering "costs" of injury and fatality are considered, the cost savings in lower casualty rates from the advanced dolly are estimated at $16,131,024.

In sum, the total savings in direct costs due to property damage and casualty losses, deriving from the use of innovative dollies, are found to be $6,792,855. This includes the PDO of $2,918,481 and the direct medical costs of $3,874,374. Including the larger social costs together with PDO costs, the total cost savings are $19,049,505. With an estimated rate of 19.35×10^8 miles traveled annually by all doubles, the potential total cost savings (social plus direct) from reduced doubles rollovers by means of C-dollies are $.0098 per dolly mile.

Costs to be Borne from the Purchase, Maintenance, and Operation of Innovative Dollies

The analysis of *costs to be borne* is designed to permit comparison of the dollar benefits discussed above with a corresponding set of dollar costs involved with introducing innovative dollies into a hypothetical trucking fleet that currently uses conventional A-dollies. An earlier study of innovative dollies [1] was used as a benchmark and format basis for this analysis. The current analysis, presented in appendix G, is a condensed version of that done previously, with similarities and differences to that study described, but without the background philosophy being restated. The reader is referred to the previous report to put this updated analysis into full perspective.

Because innovative dollies remain relatively rare in the doubles segment of the trucking industry, related operational information is still somewhat limited. The majority of advanced dolly usage is in Canada where federal and provincial regulations favor C-dollies in certain applications. Updated information from these fleets was used in this analysis with due consideration being given to the influence of regulation. Virtually all C-dollies in commercial use are of the self-steering variety. No attempt was made in this analysis to distinguish between particular C-dolly designs since it was judged that the available data base could not support the level of fidelity that would thereby be implied.

Along with a baseline financial analysis, which used the best estimate value for each parameter, a companion set of sensitivity calculations was conducted to illustrate the influence of various cost parameters on the net tradeoff of costs and benefits of C-dollies. Key parameters that have been shown to significantly influence costs can then be examined in various scenarios by which future changes in size and weight allowances, could invoke a C-dolly requirement in a manner that makes the package cost-beneficial.

To gain useful numerical values of cost elements, based upon current industry practices, trucking operators and manufacturers of innovative dollies were contacted and requested to fill out an informal questionnaire relative to this study. Questionnaires were mailed to 24 manufacturers and 31 users of innovative dollies. With the aid of follow-up phone interviews, information was gathered from 16 viable manufacturers and 14 users. The gathered data were a mix of both statistically useful and anecdotal information.

Starting with a situation that tries to approximate the current U.S. operating environment, a financial model was used to analyze the hypothetical decision by a fleet operator to buy six innovative dollies. The model considered the cost impacts of the following differentiating characteristics, in switching from A- to C-dollies:

- Initial cost of the dolly (the controlled-steer C-dolly costs approximately $5500 more).
- Converting existing equipment (it costs an estimated $1500 to equip a trailer for coupling with a C-dolly).
- Major overhauls (a C-dolly must be overhauled twice as often as an A-dolly)
- Preventative maintenance (a C-dolly requires some 50 percent more in preventative maintenance costs).
- Tire wear (a C-dolly tends to wear the dolly tires 10 to 15 percent faster than does an A-dolly).
- Scheduling costs (a small additional cost is incurred by a fleet having mixed A- and C- type equipment since it must schedule the circulation of the dollies and trailers to assure a match in the hitching equipment).
- Training (a small cost is incurred in training operators to use the new dolly equipment plus a short period of lower productivity while changes in trailer hitching practices are learned).
- Backing (a reduction in operating costs arises from the greater ease of backing a doubles combination when a C-dolly is installed).
- Weight penalty (because a C-dolly typically weighs some 460 lbs more than a A-dolly, payload weight is displaced and thus shipping revenue is lost whenever the combination vehicle would otherwise be running at the fully-allowed level of Gross Vehicle Weight).
- Accident savings (as developed above, the savings in accident costs is incorporated into the total financial model).
- Ability to operate on secondary roads (assuming that regulatory or legislated changes were made acknowledging the stability benefits of a C-dolly, broadening of access privileges to allow the operation of doubles on secondary roads would afford a cost savings).
- Permit to increase axle loads (the prospect for an increased weight allowance to nullify the weight penalty associated with the heavier C-dolly was included as an optional scenario).

In the baseline case of changeover to C-dolly equipment, the Net Present Value (NPV)[11] of such a decision results in a total negative cash flow (i.e., a loss) of $205,894 to purchase and operate six C-dollies. It is important to emphasize that this represents an *incremental* loss due to a decision to buy and operate the six C-dollies instead of a A-dollies. For example, if there were an underlying decision (with an NPV of at least +$205,000) to use twin-trailer combinations instead of tractor-semitrailers, then the further decision to outfit those twin trailers with C-Dollies would render the original decision unprofitable.

If the reference fleet were to increase its shipping charges to cover its incremental loss, the freight charges would have to be increased by $0.0000858 per 100 lb (45 kg) per mile (1.6 km). The rate increase was determined for six controlled-steer C-dollies, observed over a 10-year period, traveling 100,000 miles (160,934 km) per year and carrying 40,000 lb (22,500 kg) of cargo per trip. By way of illustration, the increase in freight charges would translate into an increase of $203.60 in the cost of shipping 100,000 lb (45,359 kg) of cargo, in small lots over a period of time, from Ann Arbor, Michigan to San Diego, California—an increase of 7.4 percent.

A sensitivity analysis using the economic model (for details, see appendix G) reveals that the dominant factor in determining this result is weight—a finding that will come as no surprise to many involved in trucking.

The additional weight of a C-dolly over an A-dolly, and the accompanying loss of payload capacity on many trips, accounted for 85 percent of the incremental cost per vehicle mile resulting from using C-dollies rather than A-dollies (as predicted by the model). Conversely, the net result was found to be rather insensitive to higher out-of-pocket costs (higher purchase price, cost of modifying trailers, greater maintenance costs, etc).

This finding leads to the observation that increased weight allowances for vehicles using C-dollies could make C-dolly use financially attractive. Under the baseline conditions assumed, an allowance that offset the weight penalty of the C-dolly (assumed to be 500 pounds) and granted an additional 191 pounds would render the decision to operate C-dollies a break-even proposition. A total increase of 1000 pounds would result in C-dolly use being distinctly profitable. Given that the reference weight limit is currently 80,000 pounds, it would appear that C-dolly use could be effectively promoted through modest increases in the legally authorized weight allowances.

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11 "NPV" is defined as the sum of the incremental annual cash flows over the life of the project reduced by the inflation rate to current dollars.

SUMMARY OF THE RESEARCH FINDINGS AND CONCLUSIONS PERTAINING TO DOLLY SPECIFICATIONS

SUMMARY AND DISCUSSION OF THE RESEARCH FINDINGS

As was noted in the introduction of this report, the goal of the study was to establish a method for specifying an appropriate C-dolly based on the performance properties of the vehicle on which it would be used. This approach grew from the recognition that the double- and triple-trailer vehicles in the U.S. come in a large variety of configurations and, thus, have a large variety of performance properties. The method developed for specifying dollies was to be *practical* in that it should be usable by people in the trucking community who are not familiar with vehicle dynamics analysis methods. The approach taken was broken down into the following tasks:

- Establish a set of relevant (i.e., influenced by dolly properties) vehicle-performance measures and related minimum vehicle-performance goals.
- Establish a *simple* means for predicting these performance measures for specific multitrailer vehicles when equipped with conventional dollies.
- Establish a simple means for predicting the improvement in the performance measures attainable with innovative dollies based on relevant specifications of the dolly.

Accomplishing these three tasks would allow people in trucking both to establish warrants for the use of innovative dollies and to specify dollies appropriate to their vehicles and performance needs.

The study has been partially successful in fulfilling its intentions. Regarding the first step, drawing on the state-of-the-art understanding of multitrailer vehicle dynamics, a number of appropriate performance measures have been put forward and their relevance explained. Also, drawing from a knowledge of regulatory practices in Canada and New Zealand and the judgment of the authors, a set of minimum performance goals has been suggested.

Next, efforts to develop simple means for predicting the critical performance measures for A-trains were remarkably successful. The regression models developed as simple predictors of the performance of A-trains were found to predict performance measures with a remarkably high degree of correlation to the "actual" performance as determined by complex simulation. These linear formulations are clearly simple enough in form to be readily used in the field, and, in a statistical sense, their accuracy is better than we would have expected.[12]

The most serious limitation to the potential practical application of the simple predictors is their need for certain parameters that often are not readily available. The two most important examples of this are tire-cornering stiffness and center-of-gravity height. These two parameters show up repeatedly among the most important in the simple predictor formulations. This, of course, is unfortunate in a practical sense since they are not broadly available and they require special effort or equipment to obtain. The other message from these results, however, is one more confirmation of the simple fact that these two parameters matter. As much as we would like them to go away for practical reasons, the fact is they will not. They are important, even fundamental, to vehicle behavior and that will not change because they are inconvenient.

Success in developing simple improvement factors that would aid in a flexible *method for specifying* C-dollies has been more limited. Improvement factors of relatively good statistical quality were found for the performance measure associated with emergency evasive maneuvering, i.e. the RTAC-B maneuver. These are rearward amplification, dynamic-load-transfer ratio, and high-speed, transient offtracking. Since the specific purpose of C-dollies is the improvement of this particular performance regime, success here and not elsewhere is not particularly surprising. For example, from the outset it was not expected that C-dolly design would influence static rollover threshold in the matrix of vehicles studied. On the other hand, the absence of consistent improvement in yaw damping through the application of self-steering C-dollies is confounding.

A striking quality of the improvement factors that were identified is their lack of sensitivity to the dolly parameters that were varied in the study. For example, while a difference was found between self-steering and controlled-steering C-dollies, all four variations of self-steering C-dollies examined showed a remarkably consistent ability to suppress rearward amplification. This quality of consistency is a valuable finding in itself, but it tends to defeat the goal of the study. The notion of specifying C-dollies to *meet the need*, so to speak, as defined by the difference between the baseline performance of the vehicle as an A-train and the stated performance goals, does not remain viable. Rather, we must settle for merely predicting performance achievable with a C-dolly and observing whether or not that performance meets a standard.

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12 We must reiterate, here, that the statistics for these models, and indeed the models themselves, presented earlier were based on a sample of vehicles selected through a mechanistic rationale, not on a random sample of the fleet. Thus, while we believe the statistics are meaningful indicators of the general quality of the approach, they should not be interpreted as precise measures of the ability of the models to predict performance of the population in general.

Finally, the economic portion of this study has shown that modest incentives in the form of increased weight allowances could make the broad application of C-dollies economically attractive. This study confirmed the early finding (see [1]) that the economics of C-dolly use is dominated by weight, not by price or other out-of-pocket costs. The higher weight of a C-dolly relative to an A-dolly imposes an economic burden that offsets the costs savings which might result from fewer accidents. But the influence of weight is so powerful that even a modest increase in weight allowance (on the order of 1000 pounds) could make the decision to use C-dollies a profitable one.

CONCLUSIONS PERTAINING TO DOLLY SPECIFICATIONS

The conclusions and recommendations presented below are based on the results of this study, the authors' overall understanding of the dynamic performance of C-dollies, and the authors' practical experience in dealing with the various elements of the U.S. trucking industry. Some of the observations made pertain to dolly characteristics that aid in mitigating the problems inherent in A-trains. Others are intended to ensure that the C-dolly does not introduce new undesirable attributes. The presentation will cover the issue of dolly specification by means of five individual subjects, as follows:

- A-train performance problems meriting solution via C-dollies
- Distinction among dolly configurations that tend to mitigate these problems.
- The one critical dolly specification that must be satisfied for any C-dolly to be of benefit.
- Other significant dolly properties whose specification impacts upon safety improvement in a secondary way.
- Dolly properties that, while not related to the achievement of safety qualities, nevertheless merit specification for the sake of hardware compatibility.

As a preamble to this presentation, it is useful to comment on the scenario by which dolly specifications are expected to be used. That is, as in most engineering problems, tradeoffs are present whenever specific, absolute, values are selected for an application. Almost invariably, the selections would be swayed one way or another depending upon the application that is envisioned. The authors' understanding of the U.S. trucking situation and the setting for LCV application, in general, is that while very few C-dollies exist in the U.S. today, a suitable set of specifications might help facilitate the adoption of such hardware by industry. It may also be that government at either state or federal level may establish certain rules that encourage or mandate C-dolly usage, whereupon dolly specifications could play a role in regulation. Whether by voluntary adoption or legislative encouragement, the assumption is that dolly specifications must be prudent so that safety is enhanced without undue penalty on the efficiency and economy of trucking practice. Insofar as the authors of this report have studied the multitrailer vehicles since they first arose in 1977, relative to double-bottom gasoline tankers in the State of Michigan, the following comments on dolly specification represent a cumulative view of the prudent tradeoffs.

A-train Performance Problems

The rearward amplification and yaw-damping responses of A-trains in common use in the U.S. differ widely from one vehicle configuration to another. At one extreme, the triple 28-foot combination amplifies tractor steering motions to a very high degree; at the other end of the spectrum, the turnpike doubles combination is essentially benign and is an insignificant amplifier of dynamic yaw motion. Thus, the first observation is that not all multiarticulated vehicles exhibit undesirable dynamic behavior. This study has produced the first *simple* method of determining whether a given vehicle configuration does exhibit a problem meriting mitigation by C-dolly. The method is practicable insofar as a sound assessment of the magnitude of the problem can be done using very simple formulas and a limited number of vehicle parameters. Generally, these parameters can be obtained using a tape measure. Thus:

> *It is recommended that A-train configurations be prequalified using the Simple Predictors and Performance Goals developed here. Comparison of the predicted performance with the performance goal could establish the warrants for C-dolly application.*

Basic Distinction Among Dolly Configurations

The simulation results show that all the different C-dollies studied help in mitigating the dynamic stability problems of double- and triple-trailer combinations. Among these dolly types, however, an important distinction is noted. Namely, it is observed that the control-steering C-dolly does not uniformly improve performance and, in general, is not as strong in its level of improvement compared with the self-steering C-dolly. Accordingly, we suggest that it be discouraged from general usage.

> *The self-steer C-dolly is recommended as the configuration of choice, when a C-dolly is warranted.*

Further, regarding the self-steering C-dolly, we have noted the remarkably low level of sensitivity of performance measures to the range of design parameter variations of the dolly examined here. This result confirms the basic principle that a C-dolly achieves most of its performance improvement simply by eliminating yaw articulation at the pintle hitch, without introducing excessively free-steering behavior at it axle. Since no self-steering dolly was represented in this study with excessive steering freedom, all of the simulated dollies provided a major improvement in performance, other parameter values notwithstanding.

At a more detailed level, however, it should be noted that the numerical value of dynamic performance measures depends significantly on the level of maneuver severity, as well as on the properties of the dolly itself. This is a fundamental point that applies to any dynamic system with significant nonlinear characteristics. In the specific case of a vehicle with a self-steering C-dolly, nonlinear elements play a significant role (1) in the case of maneuvers that cause self-steering dolly wheels to achieve a significant steer displacement, (2) at higher levels of lateral acceleration in which nonlinearities in tire shear force response predominate, or (3) whenever wheel lift-off events occur as a vehicle approaches rollover. All of these conditions clearly depend on maneuver severity.

As a result of the complexity of such nonlinear sensitivities, maneuvers simulated in this study were not necessarily as demanding of one dolly parameter as they were of another. Thus, we noted a general *insensitivity* of many performance characteristics to dolly parameters. In many cases, a more severe maneuver would have caused the dolly to operate across one of the nonlinear boundaries mentioned above, tending to increase the impact of one parameter or another.[13]

One Critical Dolly Specification

If only one parameter were to be specified for a self-steering C-dolly, it would certainly be the so-called break-out force, i.e., the level of tire side force required to initiate significant steering of the dolly wheels. If the break-out force value is too low, the tires on the dolly axle will be unable to contribute the level of side force needed to stabilize trailer yaw response, and exceedingly unfavorable dynamic behavior may result. On the other hand, a minimum threshold value will guarantee that the dolly achieves a major improvement in the dynamic behavior of the combination vehicle, assuming that it is structurally sound and does not simply fail as a trailer-coupling mechanism during severe maneuvers.

An extensive amount of research prior to this study had established that a threshold value of 0.25 for the ratio of side force to rated axle load would ensure the provision of needed side forces, while also serving to avoid excessive levels of tire wear due to scrubbing in tight radius turns. Although threshold values up to 0.30 were also examined here, the lack of any substantial improvement over the increment, 0.25 to 0.30, establishes that the 0.25 value appears to be sufficient. Further, this value matches the figure selected

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13 It must be acknowledged that an iterative method of searching for uniformly demanding maneuvers, regardless of the installed parameter values, would yield the most broadly meaningful measures of parametric sensitivity. However, the approach tends to increase the magnitude of the simulation matrix by an order of magnitude. While this approach was used in previous research on C-dollies employing a small matrix of study vehicles, it was found to be beyond the scope of this effort since the simulation matrix covered so many vehicles.

in the regulations that now apply across Canada for application of C-dollies in inter-provincial transport. Accordingly:

One C-dolly specification transcends all others in the assurance of good basic performance (given the assumption of structural integrity). This specification requires that the steer-displacement threshold be equal to a total tire side force of 0.25 of the vertical load or higher, and that this level of side force be maintained throughout the steering range.

Other Significant Dolly Properties

A number of additional dolly parameters warrant specification in order to attain high levels performance, while also ensuring the needed structural strength. Each of these will be discussed in turn.

Torsional Stiffness of the Dolly, as a Trailer-to-Trailer Link

It is well understood that the secondary benefit of C-dollies, after their reduction in rearward amplification through the elimination of an articulation point, is afforded by the ability to couple successive trailers together in roll. Thus, when a trailer unit tends to roll-over prematurely in a severe steering maneuver, the roll-coupling that derives from a dual-drawbar connection enables the lead trailer to help hold up the successive unit. The torsional stiffness of the dolly structure—effectively the spring that becomes wound up during this helping process—is instrumental in determining the net roll stability of the combination, insofar as it helps determine the maximum amount of roll motion that the rear trailer will experience. A lower level of torsional stiffness allows a larger roll motion, thus tending to reduce the stability of the combination and render it less tolerant of severe steering maneuvers.

In this study and the previous FHWA research [1], values of 30,000 and 60,000 in-lbs per degree of torsional displacement were studied as parametric variations. The previous study also included C-dollies with zero torsional stiffness in order to elucidate the importance of rearward amplification, per se, in the absence of roll coupling between trailers. Consideration of these earlier results for the baseline Western doubles combination shows that, even with a zero value of torsional stiffness, dynamic rollover performance improves 47 percent due to the basic C-dolly. If the torsional stiffness is set at 30,000 in-lbs per degree, a 56 percent improvement accrues. At 60,000 in-lbs per degree, an 87 percent improvement is seen. Clearly, the largest increment in performance comes simply with the dual-drawbar dolly configuration, but large additions in performance level accrue as the torsional stiffness parameters rises in value.

Further, in the real world, the relationship between the severity of vehicle behavior and the actual occurrence of accidents is highly nonlinear. That is to say, a specific incremental improvement in performance is more effective in reducing accidents when applied to severe performance. For example, if one were to implement a series of five-percent improvements in a particular performance quality one by one over time, the greatest accident reduction would come from the first improvement, and the last change would yield the smallest decrease in accidents. (For rigorous applications of this general notion, see for example, [15,16].) In this context, the 47 percent improvement in dynamic performance, which derives from the C-dolly devoid of torsional stiffness, is the "first" performance improvement. The additional increments of improvement available from torsional stiffness can be seen as less significant to actual accident reductions than would be implied by their proportional size.

The principal trade-off issue deserving consideration here derives from the field experience gained with C-dollies in actual service. Field usage indicates a great variety of structural problems arising when stiff dolly frames are employed. In particular, it is known that torsional stiffness levels in the vicinity of 30,000 in-lbs per degree have been difficult to build, may have imposed substantial weight penalties (see also the preceding *Economic Analysis* section, which identifies dolly weight as the primary detriment), and have induced serious challenges relative to fatigue failure of trailer structures.

By way of explanation, the typical combination vehicle encounters daily situations in which the level of torsional stiffness in the C-dolly will determine whether high levels of structural stress will be imposed upon trailer bodies—with implications for fatigue loading and the prospect for structural cracking over time. For example, the successive trailers in a combination undergo significantly differing roll angles while mounting curbs and entering raised aprons at fueling sites, industrial facilities, and marshaling areas, while executing a right-angle turn through slow-speed intersections having significant variations in grade on the entrance and exit legs, and while traveling roadways where the crown geometry of crossroads is not blended well into the travel lanes. When roll angles are induced due to road profile variations along the vehicle, torsional moments are borne across the dual-drawbar coupling in proportion to the level of torsional stiffness. Under these everyday conditions, the resulting stresses that follow directly from torsional stiffness level constitute a distinct down-side to this otherwise beneficial parameter.

In light of the preceding, the authors believe that the best specification for torsional rigidity depends on the marketplace and the regulatory environment. Assuming a specification, which is intended to be only advisory (and promotional), we favor no requirement on torsional stiffness. We believe that specification of this parameter at a level that is sufficiently high to achieve a performance benefit would serve to discourage, rather than encourage, the use of C-dollies. The discouragement would follow both from the fatigue issues due to the stiffness level and the increased dolly weight that follows from the structural robustness. (Indeed, the *Economic Analysis* shows the use of C-dollies to be a commercial burden, especially due to the weight, unless some accompanying relief is provided via increased weight allowances.) On the other hand, were the specification to be implemented in a regulatory scenario including compensating weight allowance increases, then a torsional stiffness in the 30,000 to 60,000 in-lb/deg range would be appropriate. Given that no regulatory actions is contemplated in the foreseeable future:

No specification on torsional stiffness of the dolly frame is explicitly recommended. The best specification policy appears to call for exclusion of this parameter from a list of requirements.

Tongue Length

The distance from the dolly axle centerline to the center of the pintle hitches of a C-dolly constitutes the dolly wheelbase, or tongue-length, parameter. If the tongue length is excessive, given the configuration of the vehicle combination, side force needed at the tires of the *towing trailer* to maintain stability can become excessive in maneuvers severe enough to cause self-steering of the C-dolly wheels. While no simple rule exists for stipulating how long a tongue is acceptable for each configuration, the principle is clear: the shorter the better. In all of the calculations conducted here and in previous U.S. and Canadian studies of the subject, and in the general understanding of the authors, undesirable yaw oscillation has never been seen when dolly tongue length is equal to the 80-inch U.S. convention for A-dollies and the steering resistance requirement is met. It is further noted that this value (actually, the rounded figure, 2.0 meters) has been adopted in the Canadian specification. Further, in this study, tongue lengths of 100 and 120 inches were not found to cause difficulty in the configurations of vehicles studied.

Accordingly:

A dolly tongue length of no more than 80 inches is recommended as broadly applicable and desirable. Longer values up to 120 inches are acceptable for use with the specific vehicle configurations of this study.

Strength Specifications

This study did not undertake structural modeling or any form of explicit estimation of the loads that will be incurred over the lifecycle of a C-dolly. It is straightforward, however, to compute maximum values of loading that will accrue under various severe loading conditions that are plausible and demonstrably among the worst that may develop. What is not well recognized is the full impact that a given strength requirement will have on the practicability of dolly design. As one authoritative benchmark, C-dollies have been built and employed in Canada according to a number of strength specifications stipulated in the Canadian requirements [17,18,19]. Insofar as these specifications are the object of daily practice in Canada, they do represent an important point of reference. It is also clear that each of the strength specifications can be associated with either the total weight of a towed trailer or the maximum rated load to be carried on the dolly axles, such that a proportional adjustment of strength levels to account for the specific vehicle of interest could be done.

Accordingly, four individual strength specifications appear warranted, as scaled against the absolute values stipulated in Canadian rules. They are as follows:

(1) Torsional strength of the dolly structure (assuming a roll moment transmitted between the fifth wheel plate in the rear and the two pintle eyes in the front). Since the maximum torsional moment in service will accrue during the near-rollover events, which have been extensively studied here, the pertinent trailer parameter by which to apply a proportional adjustment to the torsional strength specification is the *total weight of the towed trailer.* This choice of normalizer derives from the observation that the full weight of the towed trailer determines the magnitude of roll moment transmitted across the dolly to the preceding trailer, when the axles of the towed unit and its dolly have lifted off the pavement. The definition of the torsional strength specification is thus,

$$\text{Torsional Strength} = \text{Canadian Value } [W_t/W_r]$$

where:

W_r is the weight of the Canadian-reference trailer.
W_t is the weight of towed trailer in question.

In absolute terms:
Torsional Strength of the Dolly Frame Structure =
400,000 in-lb/deg [W_t / 50,000]

(2) Fore-aft strength of the pintle hitch couplings. The normalizer for this value is again the total trailer weight since the primary determinant of fore-aft loading of a pintle hitch derives from the dynamic forces deriving from "chugging" of the trailer mass across the pintle lash space. Accordingly, the normalized specification is expressed as:

$$\text{Fore-Aft Strength} = \text{Canadian Value } [W_t/W_r]$$

where:

W_r is the weight of the Canadian-reference trailer.
W_t is the weight of towed trailer in question.

In absolute terms:
Fore-Aft Strength of Each Pintle Hitch Coupling =
90,000 lbs each [W_t / 50,000]

(3) Vertical strength of the pintle hitch couplings. The normalizer for this value is the total trailer weight since the primary determinant of vertical loading of a pintle hitch derives

from the torsional moment that was presented as item (1) among the strength specifications, above. Accordingly, the normalized specification is expressed as:

$$\text{Vertical Strength} = \text{Canadian Value } [W_t/W_r]$$

where:

W_r is the weight of the Canadian-reference trailer.

W_t is the weight of towed trailer in question.

In absolute terms:
Vertical Strength of Each Pintle Hitch Couplings =
22,500 lbs [W_t / 50,000]

(4) Lateral strength of the pintle hitch couplings. The normalizer for this value is the dolly axle weight since the primary determinant of lateral loading of a pintle hitch derives from the maximum tire side forces that will derive at the dolly tires whenever a low speed maneuver is generated while the dolly's steering motion is locked out. In such conditions, a lateral force is borne by the hitches equal to the sum of the tire side forces on dolly tires. Accordingly, the normalized specification is expressed as:

$$\text{Lateral Strength} = \text{Canadian Value } [W_{ta}/W_{ra}]$$

where:

W_{ra} is the maximum load rated for the Canadian-reference dolly axle.

W_{ta} is the maximum load rated for the dolly axle in question.

In absolute terms:
Lateral Strength of Each Pintle Hitch Couplings =
9,000 lbs [W_{ta} / 20,000]

Specifications for Hardware Compatibility

In addition to dolly specifications whose purpose it is to ensure acceptable safety performance, two additional factors are obviously in need of standardization, although their absolute values have no bearing upon performance. These factors address the vertical placement of pintle hitch connections, above the ground and the lateral spacing between pintle hitches. While the authors believe that such figures should be set on the basis of a formal inquiry of the impacted industry, the following can be said:

Re: vertical placement. A-dollies in the U.S. are employed with pintle hitch heights of nominally 32.5 inches above the ground. In Canada, the standard height is stipulated as the range from 35 to 36 inches. No objective basis for any particular selection is seen nor is any relationship recognized between the prior height of single pintle hitches for A-dollies and the preferable height of the dual pair of hitches for a C-dolly.

Re: lateral spacing: The commonly cited value, also incorporated as the standard in Canada, is 30 inches. The largest American manufacturer of C-dollies, however, (Independent Trailers in Yakima, Washington) prefers to mount pintle hitches at a lateral spacing of 28 5/8 inches on the assertion that this dimension provides the best match-up with the spacing of frame rails in the construction of American trailers. Again, no objective basis for this parameter can be stated by the authors.

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