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16. Abstract

By means of computerized analysis and a review of existing literature, various aspects of the dynamic performance of long truck combinations were assessed. The vehicles of interest included Rocky Mountain and Turnpike doubles and a triples combination. The performance of each vehicle configuration was examined relative to that of a conventional five-axle tractor-semitrailer and a conventional five-axle doubles combination. Performance was considered in each of the following aspects: backing up, general braking performance, issues related to brake system air delivery, low-speed offtracking, high-speed offtracking, stability issues related to rapid steering maneuvers, roll stability, yaw stability of the power unit, and power requirements. In addition, the matter of alternative devices for coupling multiple-trailer combinations together is discussed.

The results of these analyses compare the performance characteristics of the respective configurations in areas thought to have implications for operating efficiency and traffic safety.

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AN OVERVIEW OF THE DYNAMIC PERFORMANCE PROPERTIES OF LONG TRUCK COMBINATIONS

Contract No. DTFH61-82-C-00054

Special Report

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July 1984

CONTENTS

1.0 INTRODUCTION

This document addresses the dynamic performance characteristics of alternative configurations of long combination commercial vehicles. The report was produced in partial fulfillment of the requirements of Contract Number DTFH61-82-C-00054, sponsored by the Federal Highway Administration of the U.S. Department of Transportation. This portion of the work is intended to provide a technical background in support of a DOT endeavor to study long combination vehicles as prescribed by Sections 138 and 415 of the Surface Transportation Assistance Act of 1982. In this study, a number of different performance attributes have been treated for vehicles configured as tractor-semitrailers, doubles, and triples. The discussion, based upon existing literature and upon a limited quantity of original computer simulations of dynamic response, addresses various truck configurations which are distinguished from one another by the length and number of trailers and the numbers of axles employed at tractor, trailer, and dolly elements.

The discussion is intended to provide insight into the state of knowledge which exists on each aspect of performance. In general, the selected performance categories involve presumed safety qualities, although the link to the actual potential for accidents has been more firmly established with some categories than it has with others. Certain additional aspects of performance pertain more to the ease of operation of the various vehicles than to safety issues, per se. Moreover, the discussion addresses a pot pourri of subjects to the degree that currently technology can support definitive generalizations on performance.

The following subjects have been addressed in this report:

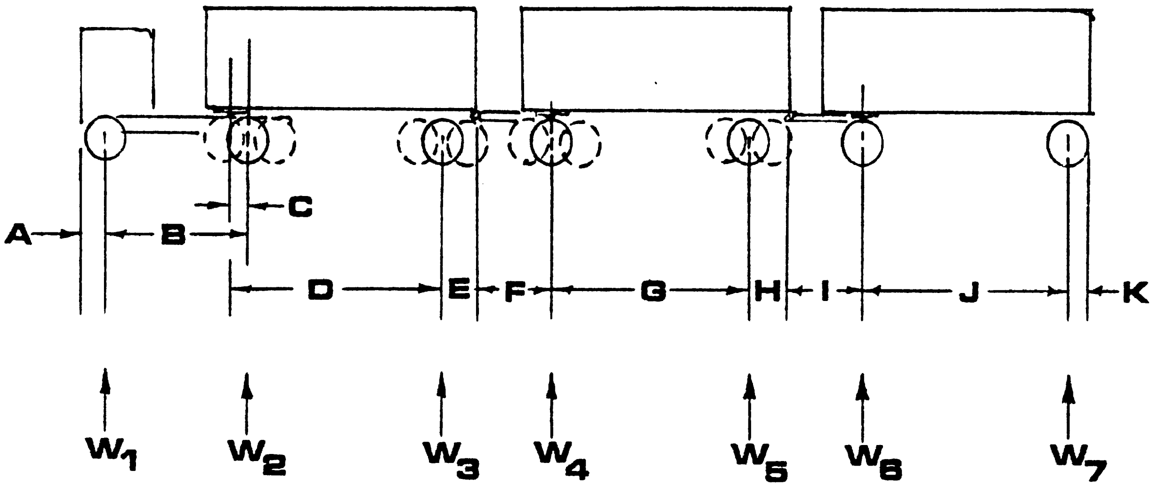
Alternative Coupling Devices
Backing Up
General Braking Performance
Issues Related to Brake System Air Delivery
Low-Speed Offtracking Performance
High-Speed Offtracking Performance
Stability Issues Related to Rapid-Steering Maneuvers

Roll Stability

Yaw Stability of the Power Unit

Power Requirements

The vehicle combinations of interest are defined in terms of the dimensional and loading parameters defined in Figure 1. The geometric data shown in the figure represent what are thought to be characteristic values for vehicles which are in service in the 1984 time frame. Axle loadings are selected such that the indicated gross vehicle weights are attained, while distributing the load in a simplified manner which roughly corresponds to popular practice. It should be recognized that many of the long combination vehicles currently in service in the U.S. are not typically loaded to the levels of gross weight shown. On the other hand, the specific loading conditions presented in the figure do not strongly influence the relative dynamic performance levels to be discussed, except with regard to braking and engine power requirements. Each of the above listed subjects will be addressed, in turn, in Sections 2.0 through 11.0. Section 12.0 summarizes observations pertaining to the various vehicle configurations.



(Longitudinal Dimensions, ft)

Vehicle	A	B	C	D	E	F	G	H	I	J	K	Overall Length
Trct/Semi-48	2.5	18	1.8	40	5.0							63.7
Double-28's	2.5	10	0.7	22.8	2.2	6.1	22.8	2.2				65.7
RMD-48/28	2.5	12	0.4	40	5.0	6.1	22.8	2.2				90.2
RMD-45/28	2.5	12	0.4	37	5.0	6.1	22.8	2.2				87.2
TPD-48/48	2.5	12	0.5	40	5.0	6.1	40	5.0				110.1
TPD-45/45	2.5	12	0.5	37	5.0	6.1	37	5.0				104.1
Triple-28's	2.5	10	0.7	22.8	2.2	6.1	22.8	2.2	6.1	22.8	2.2	99.0

(Axle/Tandem Loads, K-lbs)

Vehicle	W1	W2	W3	W4	W5	W6	W7	GVW
Trct/Semi-48	12	34	34					80
Double-28's	10	17.5	17.5	17.5	17.5			80
RMD-48/28	10	33	32	15.5	14.5			105
RMD-45/28	10	33	32	15.5	14.5			105
TPD-48/48	10	28.5	27	27.5	27			120
TPD-45/45	10	28.5	27	27.5	27			120
Triple-28's	10	17.5	17.5	17.5	17.5	17.5	17.5	115

Figure 1. Dimensional and loading parameters of combination vehicles.

2.0 ALTERNATIVE COUPLING DEVICES

Multiple-trailer combinations currently used in the United States employ a separate coupling device between adjacent trailers called a "converter dolly." The term "converter" reflects the fact that such dollies serve to "convert" a semitrailer to a self-supporting "full trailer." Vehicle combinations are then built up by coupling the first semitrailer directly to the tractor, via a "fifth wheel" coupling and by connecting subsequent trailer(s) via converter dollies. As shown in Figure 2, the conventional dolly device consists of a light frame which supports a fifth wheel for coupling the succeeding trailer and which employs a "pintle hitch" coupling at its forward end for connecting to the preceding trailer. The dolly can be configured with either one or two axles, depending upon the type of vehicle combination which is being assembled. During turning, the dolly pivots about the pintle hitch to permit the vehicle combination to track along a curved path. All of the long combination vehicles examined in this study employ this conventional style of dolly.

The converter dolly described here introduces two articulation points into the vehicle combination--namely, one at the pintle hitch and one at the fifth wheel connecting the dolly to the succeeding semitrailer. Because of certain dynamic phenomena, which will be discussed in Section 7.0, the stability of the combination vehicle in rapid maneuvers is reduced in combinations having a greater number of articulation points. The stability characteristic at issue, here, involves the tendency of the rear trailer in conventional multiple-trailer combinations to produce amplified lateral motions in response to rapid steering inputs, at worst causing rollover of the rear trailer in maneuvers having a moderate level of severity and, more typically, causing lightly damped "swaying" motions which may be evident during normal down-the-road travel. Also, since the pintle-hitch connection of the conventional converter dolly affords no resistance to the roll motion

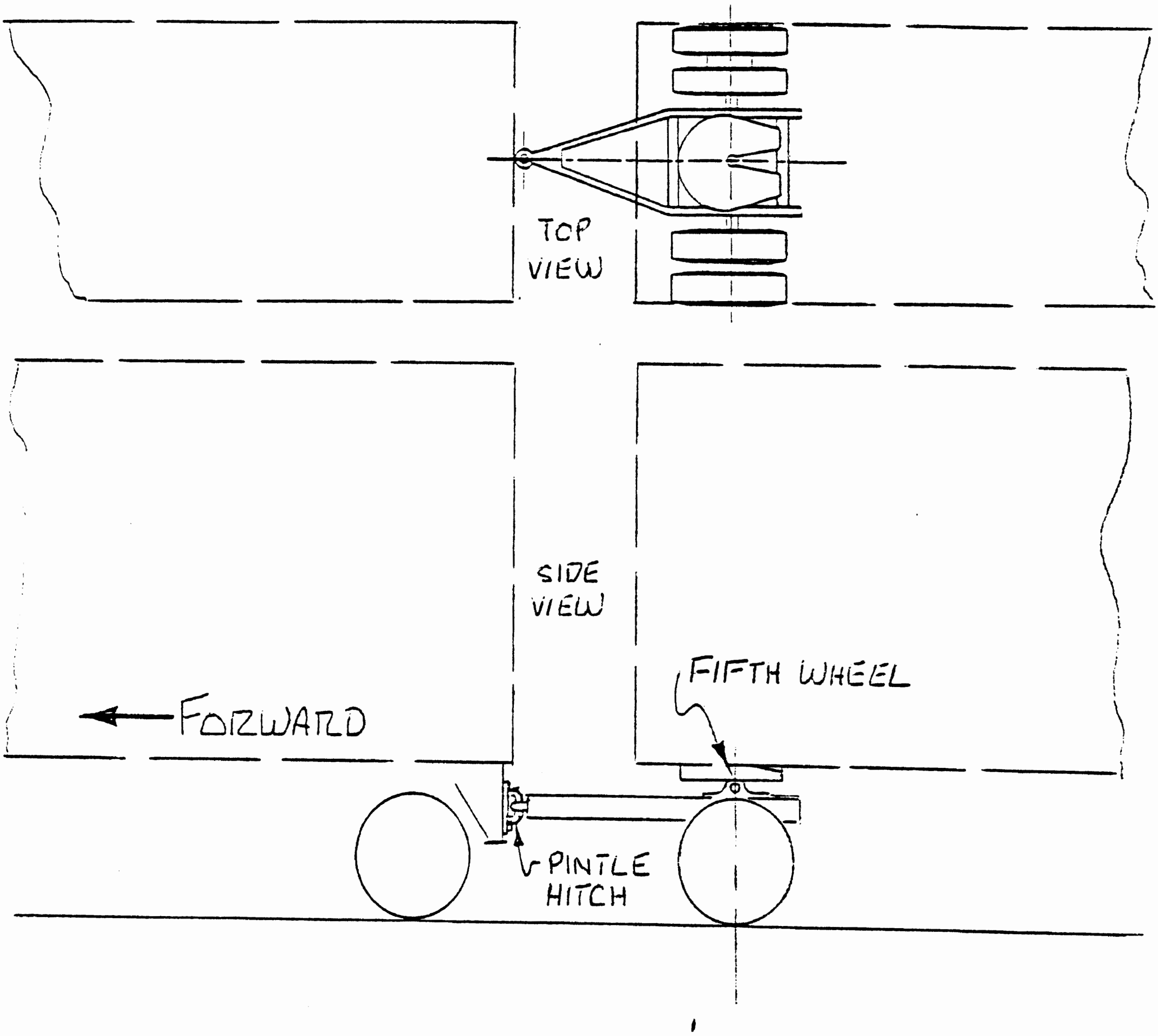


Figure 2. Conventional, or "A-type," converter dolly.

of one trailer with respect to the other, the rear trailer is free to respond to the amplified lateral motions by rolling over independently of the other vehicle units.

Another controllability issue arising with conventional dollies in a combination vehicle is the possibility of a very rapid "dolly jackknife" articulation motion following lockup of the dolly wheels during braking. The dolly becomes free to rotate about the pintle hitch, causing the following trailer to swing forward and to one side—usually impacting the rear corner of the preceding trailer. This mode of instability is most likely on slippery surfaces, when the vehicle is unloaded, and it can lead to complex subsequent motions of the coupled trailers, possibly resulting in rollover or collision with other objects.

Primarily due to these dynamic deficiencies, other schemes of inter-trailer coupling have been developed which either reduce the number of articulation points between trailers from two to one or which modify the functional mechanics of the coupling. At present, virtually none of these alternative coupling designs are in commercial service in the U.S. Most of the development in this area has been seen in Canada and a variety of alternative devices are found in commercial service there. These devices are discussed here to provide the reader with information which appears to suggest that some promising improvements in vehicle coupling technology are on the horizon. The prominent alternative concepts for coupling trailers together are described below, and the salient features of each are discussed.

2.1 B-Train Configurations

Since the conventional converter dolly, employing a single pintle hitch and a fifth wheel coupling, constitutes the de facto standard around the world for connecting successive trailers in combination, popular jargon terminology has labelled this device as an "A-Dolly." Correspondingly, the combination vehicles employing such dollies are called "A-Trains." The first conceptual variation on this configuration involves the simple elimination of the pintle-hitch connection altogether, constituting a so-called "B-Train," as shown in Figure 3. The B-Train incorporates an unusual first trailer design

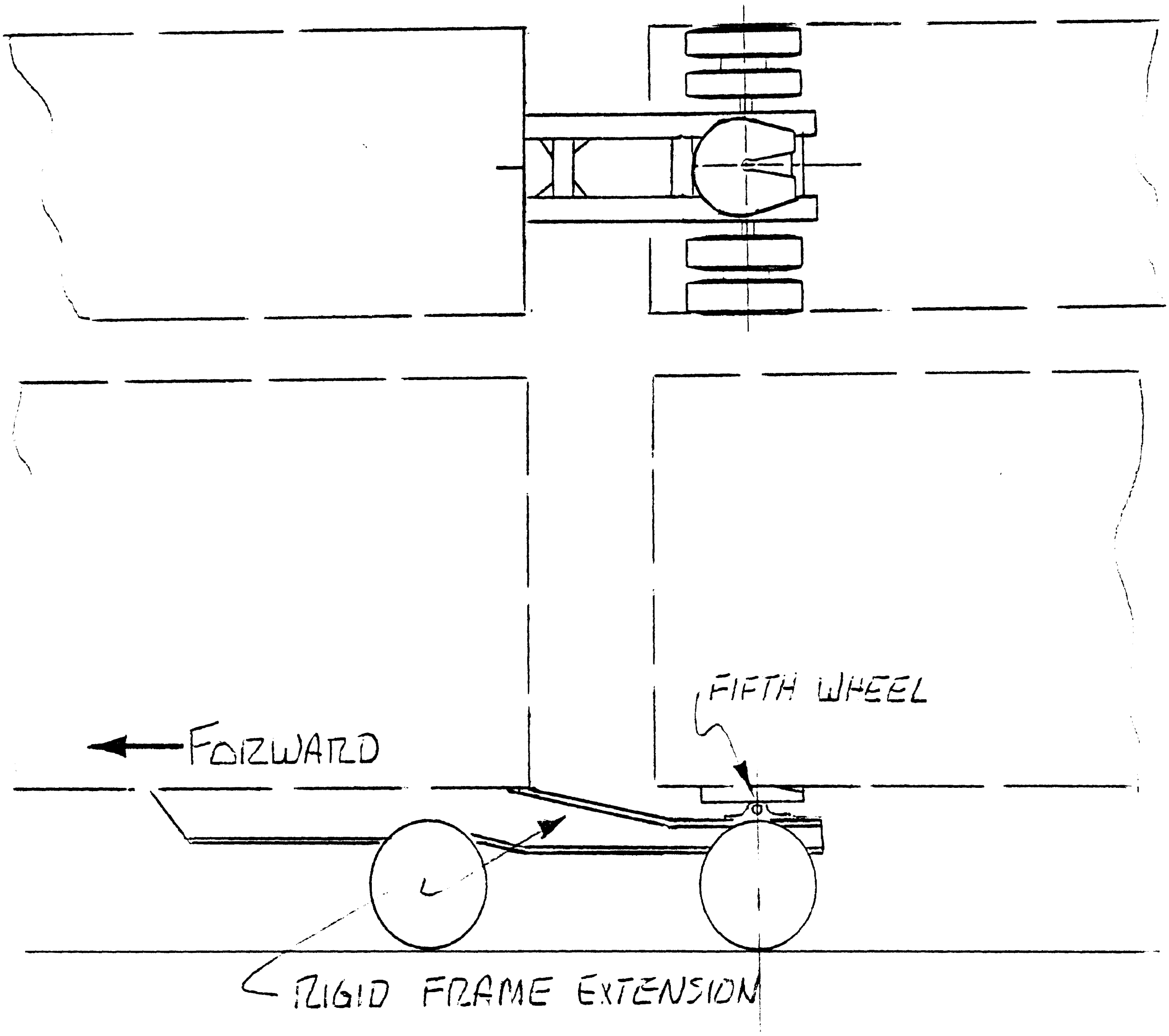


Figure 3. Coupling section between two trailers of a "B-train."

in which the trailer frame structure is extended beyond the rear of the payload bed so as to mount a fifth-wheel coupling for directly connecting the succeeding trailer. Not only does the B-Train design eliminate one articulation point, it also provides a roll-rigid connection between the successive trailers, thus assuring that the units can "assist" in assuring roll stability during rapid steering maneuvers. The salient features of B-Train performance are as follows:

1) The offtracking of the rearmost trailer tires in tight turns is somewhat greater than that of comparable A-Train layouts. The extent of increased offtracking is considerably less, however, than that which usually accrues from adding another trailer to a given vehicle combination.

2) The dynamic stability of B-Trains is comparable to that of tractor-semitrailers, in steady turning maneuvers, and can range from moderately lower to moderately higher than that of tractor semitrailers in dynamic steering maneuvers. Thus, the B-Train is regarded as essentially "solving" the dynamic stability shortcomings of A-type multiple-trailer combinations [1].

3) There is no equivalent to the rapid "dolly jackknife" mode of instability during braking. Articulation instabilities of the trailers, themselves, are still possible under severe braking conditions, but the motions are much slower than that associated with the jackknife of A-Dollies and thus are seen as constituting a reduced level of hazard.

4) The B-Train can be more easily backed up than A-Trains because of the elimination of the short dolly element from the combination.

5) The common use of either multiple axles or widely-spread axles beneath the inter-trailer coupling on the B-Train is likely to produce a greater level of tire wear, due to the "scrubbing" or "scuffing" motions which occur when such vehicles negotiate tight-radius curves.

A practical consideration which has made the B-Train concept rather unattractive for application to van-type trailers is the fact that the extended frame portion on the lead trailer prevents direct access to the rear of the trailer when backed up to a conventional loading dock. Further, the two B-Train trailers are not interchangeable in the combination. Thus, in Canada where a considerable usage of B-Trains has prevailed, the applications primarily involve either bulk commodity tankers and gravity hoppers or flat-bed configurations which are loaded by fork lift trucks from the side. In these applications, the two trailers are kept more or less permanently coupled together such that interchangeability is not an issue.

2.2 Rigid Dual-Drawbar Dollies

One concept which yields vehicle performance attributes which are virtually identical to those obtained with the B-Train employs a rigid, dual-drawbar dolly as the coupling between otherwise-conventional trailers. This device, shown in Figure 4, looks identical to the conventional "A-Dolly" in the side view, but incorporates two side-by-side pintle-hitch connections, as seen when viewed from above. The dual drawbar connection to the lead trailer eliminates the steer or "yaw" articulation which is conventionally afforded by the single pintle connection of A-Dollies. This concept differs from the B-Train in performance only insofar as the dolly is free to pivot in the pitch direction about its dual pintle hitches. Otherwise, all stability, tracking, backing-up, and tire-scrubbing characteristics are identical to those of a corresponding B-Train having the same geometric placement of axles and fifth wheel centers. When the rigid, dual-drawbar dolly is adapted to existing trailers, measures must generally be taken to strengthen the trailer structure so as to suitably handle the higher loads which are applied at the two pintle hitch constraints.

Operationally, the rigid, dual-drawbar dolly concept offers considerably more flexibility than the B-Train configuration such that it is more attractive in applications to conventional van-type doubles. Specifically, the dolly may be easily removed to facilitate access to each trailer from a loading dock and, additionally, trailer interchangeability is assured. By mounting pintle hitches for both the dual-drawbar and conventional single A-Dolly connections on the back of each trailer, a fleet can even interchange both dolly types during a transition period in which either dolly type must be accommodated in the operation.

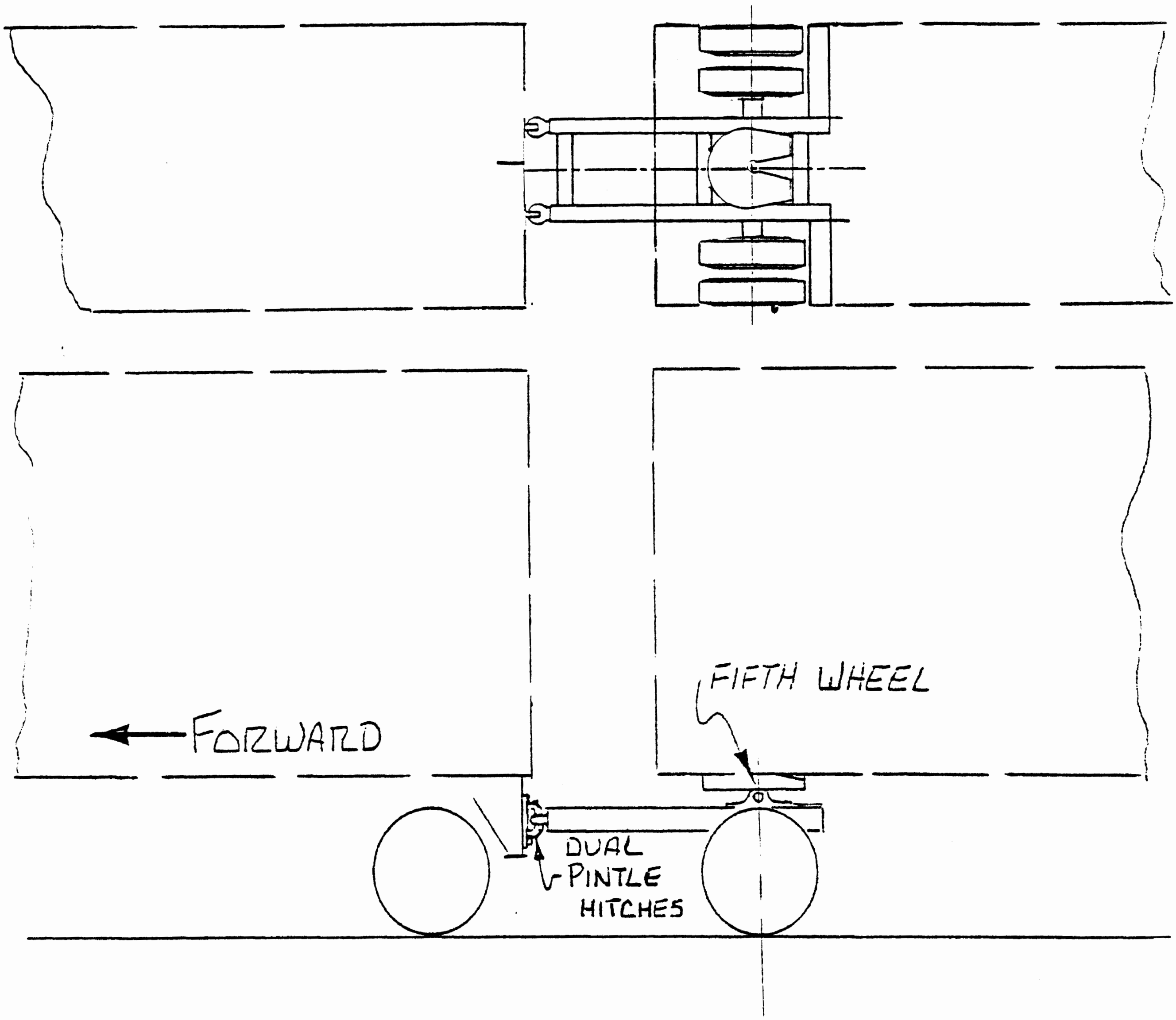


Figure 4. Rigid, dual-drawbar dolly (dolly axle does not steer).

2.3 B-Dollies

Another dolly variation which has been developed in Canada and is receiving rather wide application is the so-called "B-Dolly" concept. This device is identical to the rigid, dual-drawbar dolly defined above, except that another measure has been taken to reduce tire scrubbing as well as the level of reaction forces applied to the two pintle hitches during turning. Two varieties of B-Dollies are shown in Figure 5. Both concepts employ a caster-steering principle such that the wheels on the dolly axle(s) are steered to accommodate tight-radius turning with minimal tire scrub and with minimal degradation in tractor steering quality.

The "automotive-steer" type of B-Dolly employs an axle which is conceptually analogous to the steering axle on a truck tractor insofar as the wheels are supported on spindles which pivot about kingpins to provide a steering displacement. These steering motions are resisted, to some degree, by a "centering mechanism" whose purpose is to keep the wheels running straight during normal travel on the highway.

The other design is the "turntable-steer" B-Dolly in which a conventional solid axle is mounted on a central turntable. The center of the axle is located aft of the center of the turntable such that an effective caster or mechanical trail is available for "steering" the entire axle about the center of the turntable. Again, a centering mechanism is employed to keep the wheels running straight during normal operations on the highway.

A recent Canadian study [2] has established that performance of a doubles combination equipped with a B-Dolly is essentially the same as that of an equivalent vehicle combination equipped with a dual-drawbar dolly, with non-steerable wheels or axles, with the following exceptions:

1) Tire scrubbing is reduced in tight-radius turns, to the extent that the centering mechanism allows the dolly wheels to steer. By the same mechanism, tractor steering quality is improved during tight-radius turns, relative to the unit having a rigid, non-steering dolly.

2) Low-speed offtracking is moderately improved over that exhibited by an

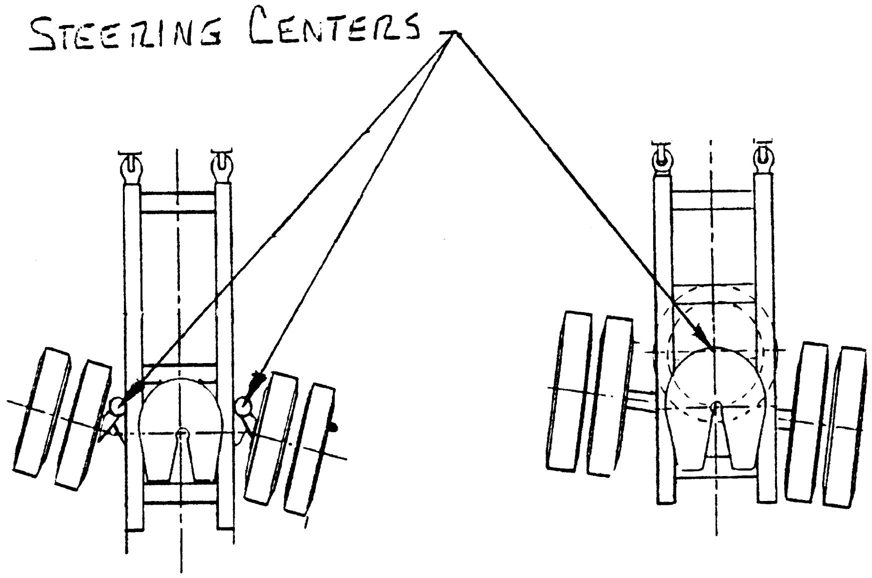


Figure 5. B-dollies having (a) automotive- and (b) turntable-steering mechanisms.

equivalent A-Train combination.

3) The potential for amplified lateral motions of the vehicle combination in rapid steering maneuvers may range from equivalent to B-Train performance to worse than A-Train performance. The determining factor involves the ability of the dolly's centering mechanism to resist the lateral forces generated by the tires in such maneuvers. When too much steering freedom is allowed by the centering mechanism, poorer dynamic performance is obtained. Thus, the achievement of the "low tire-scrub" quality is fundamentally in conflict with the achievement of good dynamic stability. Moreover, the acceptability of a given B-Dolly, from a dynamic stability point of view, is dependent upon design details.

4) The B-Dolly exhibits a sensitivity to right/left differences in the torque output of the brakes installed on the dolly axle. If an excessive amount of torque imbalance exists, or if the vehicle is braked while the dolly tires are running over a surface having large differences in friction level, left-to-right, the wheels may experience an anomalous steer displacement. This steer response may induce a sufficient motion disturbance that vehicle control is threatened. Again, the acceptability of the B-Dolly, from the viewpoint of insensitivity to imbalanced braking forces, is dependent upon design details. It is clear, however, that the B-Dolly design with knuckle-mounted wheels is inherently less sensitive to imbalanced braking forces than is the turntable variety, simply as a result of the differences in the relative proximity of the wheel centers to the steering pivot.

Many of the B-Dolly designs being currently marketed in Canada incorporate a locking feature whereby the wheels may be locked on center. The vehicle has a switch available in the tractor for engaging the "locked" steering mode. In this mode, the B-Dolly-equipped vehicle can be backed up with the same facility as a comparable B-Train. Also, one may adopt the practice of locking the dolly steering system whenever the vehicle is operated at highway speeds, thereby circumventing any of the above-cited dynamic deficiencies.

Since the B-Dolly couples via pintle-hitch and fifth-wheel hardware in a fashion identical to the rigid, dual-drawbar dolly described above, vehicles employing the B-Dolly accrue the same advantages of trailer interchangeability and ready access at loading facilities. At the same time, the B-Dolly can impose large levels of loading to the pintle hitches, such that the structural integrity of the selected couplings deserves close scrutiny. Although the B-Dolly concept offers substantial promise as a means to optimize the dynamic performance characteristics of multi-trailer combinations, dolly characteristics must be carefully constrained. Further development of this concept is known to be currently underway [3] and standardization of dolly characteristics has been recommended [2,4].

2.4 Active Linkage Couplings

Another conceptual area which has produced commercial hardware for coupling trailers together has involved dollies in which the single tongue, or drawbar portion, of a conventional A-Dolly is replaced by a set of active linkages which control articulation between the dolly and the preceding trailer. An example of such an arrangement is shown in Figure 6. Linkage designs of this type, which do improve dynamic properties, serve to locate an "instant center" of dolly rotation significantly forward of the conventional pintle-hitch location [5]. Thus, they also cause the effective length of the drawbar of the dolly to be longer. Such linkage couplings do tend to produce a greater amount of offtracking, although the ability to back up and the resistance to rapid articulation instabilities during braking is less than with B-Trains and properly designed B-Dolly equipment.

Variations on such active linkage concepts have also been proposed in which additional features of the design have served to modify the articulation function so as to provide better offtracking performance in tight-radius turns [6].

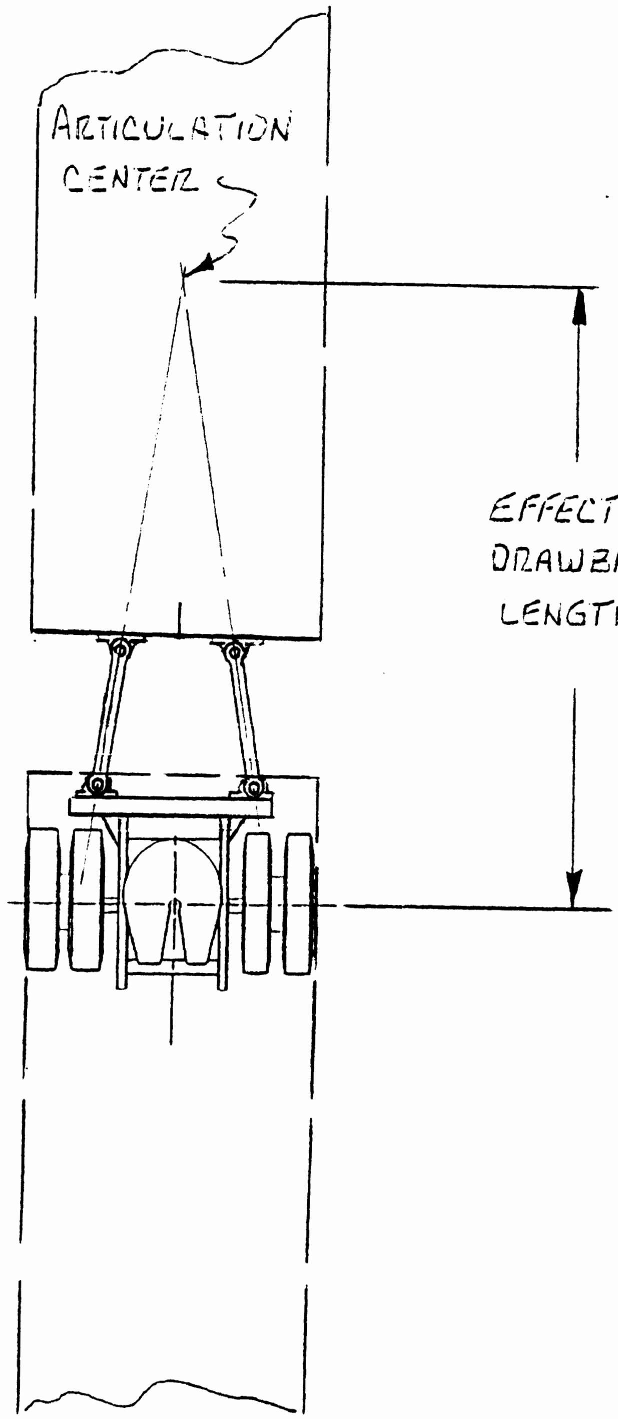


Figure 6. Trapezoidal, active-link dolly coupling

3.0 BACKING UP

The process of backing up an articulated vehicle is a basically unstable maneuver. Without any driver steering action when the vehicle operates in reverse, any deviations in the longitudinal alignment of the vehicle train will grow with distance traveled. The mechanism is illustrated by the simple model of a semitrailer operating in reverse in Figure 7. As the vehicle moves backward (in the Z direction at the hitch), the magnitude of the deviation angle, θ, measured between the centerline of the trailer and the direction of travel at the hitch, will grow with increasing distance of travel. The governing equation for the motion takes the form:

$$Z/L = \log[\mathrm{Tan}(\theta/2)/\mathrm{Tan}(\theta_0/2)]$$

where

Z = the distance traveled in reverse

L = trailer wheelbase (hitch to rear axle)

θ_0 = initial deviation angle (at $Z = 0$)

θ = deviation angle as a function of distance

As the vehicle moves backward, the deviation angle will grow continuously until the trailer is perpendicular to the Z direction, and in the absence of any interference with the towing vehicle, would commence to trail the hitch point in a forward attitude. Although the above equation would indicate that, with zero initial angle, the backing distance is theoretically infinite, in reality no vehicle can ever achieve a true zero alignment angle. If nothing else, lash in the hitch and asymmetry in the structure will result in some initial angle.

The above equation tells the story of uncontrolled backing maneuvers, where the driver does not or cannot exercise any control over the vehicle's lateral deviations. In such cases, the backing distance that can be achieved with the front of the vehicle train (denoted by Z) is proportional to the wheelbase of the trailer and to the allowable maximum deviation angle (the point at which vehicle interference would occur). For the very small initial deviation angle of 0.01 degrees and an allowable deviation angle of 45

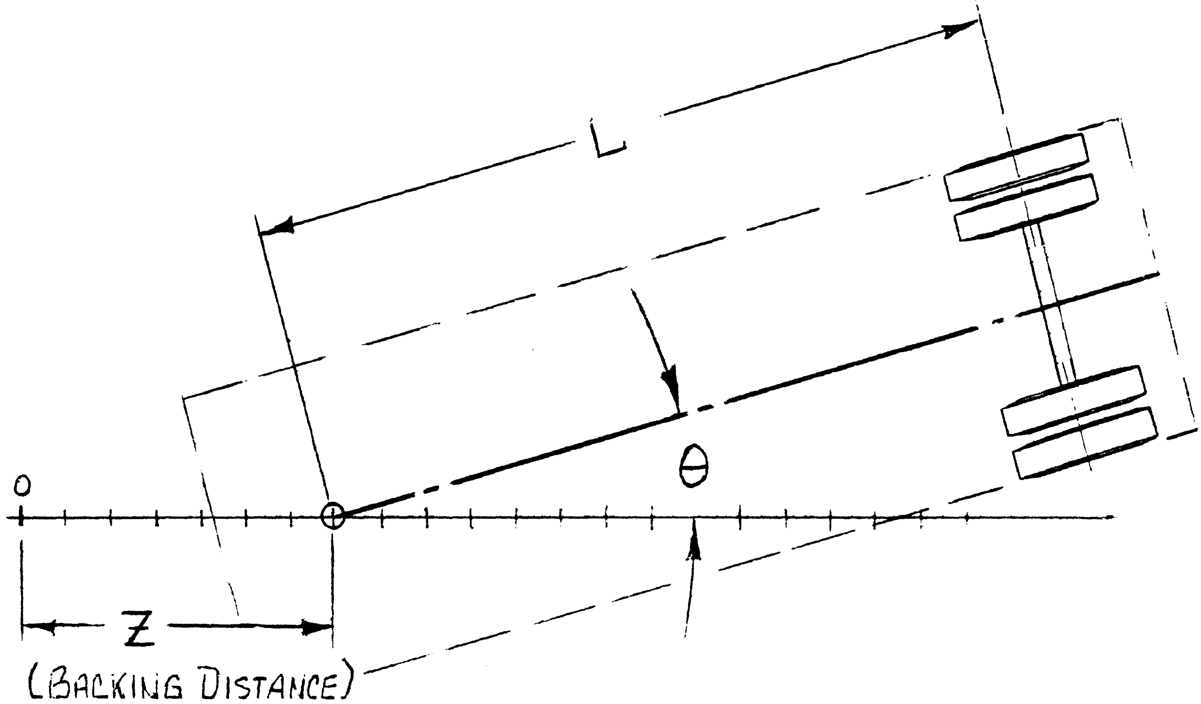


Figure 7. Schematic for analyzing the articulation of a semitrailer being backed up.

degrees, the calculated backing distance would be approximately 3.7 times the wheelbase. At an initial angle of 1.0 degrees, the backing distance decreases to approximately 1.7 times the wheelbase; and for a relatively large initial angle such as 10 degrees, the distance decreases to approximately 0.7 times the wheelbase.

In the case of vehicles having multiple articulation points, such as full trailers, doubles, and triples, the potential distance for which the vehicle can be backed in the "uncontrolled" mode becomes dependent on the shortest link in the train. For example, the dolly hitch used with these vehicles has an effectives wheelbase of only 6.1 feet. Hence, the backing maneuver will be limited in length by the likelihood that the longitudinal alignment of the train will first collapse at this point. Depending on the initial conditions, the backing distance limit is predicted to be anywhere from about 1.0 to 3.0 times the dolly wheelbase. Experience in observing such vehicles would indicate that these predictions are reasonable.

3.1 Controllability During Braking

The extent to which multiply articulated vehicles can successfully back up for indefinite distances is, practicably speaking, an issue of controllability; i.e., can the driver observe the alignment of the essential (shortest wheelbase) elements of the train; does he have the skills to compensate; and is the power unit (or intermediate units) capable of making the necessary maneuvers to compensate for the collapsing unit in the train. Answers to these questions must largely come from experience and observation.

There is no question that most singly articulated vehicles are capable of successful backing for indefinite distances with an experienced driver. Exceptions to this case are of no interest or consequence here. Tractor-semitrailers, which are the common example of single articulation, are, from experience, all capable of unlimited backing.

Vehicle trains with two articulations are also generally capable of unlimited backing, assuming a skilled driver is at the wheel. This class would include such vehicles as full trailers used on farms, full trailers pulled behind straight trucks, the B-Train configuration, and doubles units in which the articulation of the dolly unit can be locked out during backing maneuvers.

Once the number of articulation points reaches three, however, successful backing in a controlled fashion is only infrequently possible. Hence, the backing constraints defined in the earlier paragraph will become the governing influence. In the case of the doubles units, the shortest link is the dolly unit, effectively limiting the backing maneuver to a range of 6 to 18 feet, depending on luck.

The equation would suggest that the same backing limits apply to triples units as well. In fact, the range of successful backing distances would be expected to be shorter, on the average, for triples than for doubles. The reason is that it takes more forward distance of travel to get a triples unit initially aligned. Hence, from an arbitrary stopped situation, it is likely that higher initial deviation angles will be present between the respective elements of a triples configuration. Therefore, there is a higher probability of misalignment in a triples, and more articulation points available at which the train can collapse, halting the backing maneuver. The addition of articulation-lockout features on the dolly units of a triples combination will improve backing distance, simply because the wheelbase of the shortest unit now becomes lengthened to approximately the wheelbase of the trailer. For triples employing 28-foot trailers, the "lockout" feature improves the backing distance to a range of 20 to 60 feet, depending on conditions. With three articulation points still active in the vehicle train, it would be the very unusual driver who could successfully back up a triples for any distance significantly longer than that predicted, and even then it would require a very slow and deliberate maneuver.

3.2 Conclusion

Applying this analysis to the long combination vehicles listed for consideration leads to conclusions regarding backing ability, as tabulated in Table 1.

Table 1

Backing Ability of Long Combination Vehicles

Vehicle	Expected Backing Distance
Tract/Semi-48	Infinite
Double-28	6 to 18 feet (without dolly lockup)
	20 feet to infinity (with dolly lockup and skilled driver)
RMD-48/28 and 45/28	6 to 18 feet (without dolly lockup)
	20 feet to infinity (with dolly lockup and skilled driver)
TPD-48/48 and 45/45	6 to 18 feet (without dolly lockup)
	40 feet to infinity (with dolly lockup and skilled driver)
Triples-28	6 to 18 feet (without dolly lockup)
	20 to 60 feet (with dolly lockup)

4.0 GENERAL BRAKING PERFORMANCE

Three regimes of braking can be conveniently distinguished, namely, (1) "normal" stops, (2) downhill speed control, and (3) emergency stops.

The measures used to evaluate performance in each of these three regimes of braking are entirely different. During normal braking one may be mainly concerned with brake wear and with obtaining uniform wear rates so that all brakes on a unit need maintenance at approximately the same time.

In normal braking, the level of friction at the tire/road interface is not an issue because the brake-force demand does not exceed the available friction-force capability. Thus, except for surfaces having very low friction levels, such as ice- and snow-covered pavements, there is no safety issue involving the low-level braking applications which typify all normal driving. There are special concerns, however, with the low-level, but long-termenergy-absorption demands associated with downhill speed control--say, on a mountain grade.

On the other hand, emergency braking performance (whether on high- or low-friction surfaces) may be limited by an unfavorable distribution of brake torques among the axles. In emergency braking, directional control problems arise if wheels on certain axles lock up due to brake-torque levels exceeding the available frictional capability of the tire/road interface. Consequently, performance in emergency stops is expressed in terms of the wheels-unlocked stopping distances (or decelerations) attainable at various levels of tire/road friction.

In this section, the downhill and emergency braking process will each be addressed from the viewpoint of distinctions between differing types of truck combinations.

4.1 Downhill Speed Control

The heat flow into a brake during a mountain descent can cause its temperature to rise significantly. If the temperature becomes excessively high, the brake's torque capability will be greatly reduced--possibly to the point where a heavy vehicle may run away. Even if the brakes do not reach temperatures that cause them to fade appreciably, brake wear increases dramatically as temperature increases, leading to uneven brake wear throughout a vehicle if the brakes are not "temperature balanced." Accordingly, to prevent brakes from overheating, trucks generally descend mountain grades at low speeds, thereby reducing the power-absorption demands on the brakes.

The selection of a braking system for a heavy combination vehicle represents a compromise among the qualities of lining wear, emergency stopping performance, and downhill speed control. With regard to vehicle configuration, wear and temperature balance requirements must be met by all types of combination vehicles. As long as there are not large differences in performance at low levels of brake pressure, the brakes on each unit of a combination may be balanced (sized) to prevent excessive wear or high temperatures from occurring on any one unit. Given typical brakes and presuming that all brakes on a vehicle will actually be applied (that is, not just the trailer brakes, for example), then no particular type of combination vehicle has an <u>inherent</u> advantage over the others with respect to normal brake wear or in descending mountains using the foundation brakes to control speed.

Since some of the longer truck combinations have axle loads which are less than those seen in more conventional practice, however, these longer units would present certain benefits for downhill braking on mountain grades. For example, if we observe that each of the vehicle configurations being studied here has a specified gross weight and a specified number of installed brakes, it is possible to estimate the steady downhill braking capability deriving simply from the horsepower-absorption capacity of the brakes. Using a nominal value for power absorption rating of 20 horsepower per full-sized truck brake [7], the respective vehicle combinations from Figure 1 give the values shown in Table 2 for "pounds of vehicle weight per horsepower of brake absorption capacity."

Table 2

Properties Related to Downhill Speed Control Using the Foundation Brakes Only.

Vehicle	Gross Weight, lbs	Number of Full-Sized Brakes*	Nominal Energy Absorption Rating lbs/hp**
Trct/Semi-48	80,000	9.3	430
Double-28's	80,000		
RMD-48/28	105,000	13.3	395
RMD-45/28	105,000		
TPD-48/48	120,000	17.3	347
TPD-45/45	120,000		
Triple-28's	115,000	13.3	432

*Counting each brake on tractor drive axles and trailer axles as 1.0 and each brake on the tractor steering axle as 0.65.

**Assuming 20 hp per brake.

Clearly, lower values of the (lbs/hp) measure, in the right-hand column, suggest that the vehicle is able to more adequately handle the demands of downhill braking. Thus, the Rocky Mountain and Turnpike doubles provide an improvement over conventional vehicles which are loaded to the full gross weight allowance. Also, the triple provides an "absorption rating" which is essentially the same as that obtained with conventional vehicles.

As a supplement to (or possibly, a substitute for) the use of foundation brakes as a means to control downhill speeds, some operators employ a so-called "retarder" device. The retarder is typically an engine-mounted device which permits the harmless absorption of energy while applying a retarding torque to the drive wheels of the tractor.

Standard procedures for selecting a retarder on the basis of its horsepower rating are based on (a) gross vehicle weight, (b) the slope of the steepest downgrade of interest to the operator, and (c) the minimum descent speed the operator is willing to accept [8]. These selection procedures are well suited for tractor-semitrailer vehicles weighing less than 80,000 lbs. However, experience shows that retarders should be turned off on icy roads to prevent the jackknifing that may result from overbraking the drive wheels of the tractor.

For heavily-loaded long combinations, the standard procedure for selecting a retarder is inappropriate since the indicated horsepower level of the retarder would so overbrake the tractor drive wheels that hazardous jackknife-threatening conditions may be encountered even on moderately-slick road surfaces such as wetted pavement. On the other hand, if the retarder power were selected simply in proportion to the load carried on the drive wheels of the tractor, the directional control problem would be relieved. However, the heavier vehicle could not achieve the same level of downhill speed control as that achieved by a tractor-semitrailer having the same drive-axle load. As a first approximation (assuming retarder-only braking), heavy vehicles might be expected to operate with a "speed penalty" that is inversely proportional to the weight ratio; specifically,

$$V_H = (W_L/W_H)V_L$$

where V_H is the speed of the heavy vehicle of weight, W_H

and V_L is the downhill control speed for a lighter vehicle of weight W_L with comparable load on the drive wheels

Based on this approximation, the heavier configurations suffer an appreciable loss in downhill speed control compared to another combination having equal loading of the drive wheels. (The triple 28's, for example, should travel at a maximum descent speed equal to (80/115) of the speed of the retarder-equipped double 28's.)

Note that these differences could be compensated for by using trailer-axle retarders if the costs were acceptable to the owners. Although these additional retarders may pay for themselves in terms of reduced brake wear and added safety, they are still an added expense to the buyer of new equipment.

In summary, the downhill braking performance of the long combination vehicle has been discussed in two parts. Firstly, if the foundation brakes are used to control downhill speeds, it was seen that: (a) the triple provided approximately as high a level of capability as conventional vehicles and (b) the Rocky Mountain and Turnpike doubles provide substantially higher-than-conventional levels of capability. Secondly, if a tractor-mounted retarder is to be used as the <u>sole</u> means for controlling downhill speeds, the respective long combinations will exhibit <u>reduced</u> downhill capability according to the ratio of the gross weights of conventional vehicles (say, 80,000 lbs) to the gross weights of each of the long combinations. Thus, the overall picture is mixed. Table 3 lists the percent difference in downhill speed control capacity, with respect to five-axle tractor-semitrailers and doubles at 80,000 lbs GCW, for each of the long combinations.

It is difficult to generalize on the tradeoff between differences in the energy capacities of the foundation brakes versus retarders in various vehicles. We see in the far right column of Table 3 that when the foundation brakes <u>and</u> 300 hp of engine retardation are combined to control downhill speed, the three long combination vehicles are all deficient in total capability relative to conventional 80,000-lb vehicles. Moreover, it is

Table 3

Percent Difference in Speed Control Capacity from that of Conventional Vehicles

(A positive percentage indicates that the longer combination provides a higher level of downhill braking capacity)

Vehicle	GCW, lbs	Using Foundation Brakes Only	Using Tractor-Mounted Retarder Only	Using Foundation Brakes and 300 hp of Engine Retardation
Rocky Mountain Double	105,000	+9%	-24%	-12%
Turnpike Double	120,000	+24%	-33%	-13%
Triple	115,000	-0.4%	-31%	-24%

apparent that there are both "give and take" distinctions in downhill capacity with the Rocky Mountain and Turnpike doubles, while the triples combination is basically registering a net loss in all of the speed-control scenarios.

4.2 Emergency Braking

At the outset, it must be noted that it is hard to make general statements about emergency braking performance because the torque capabilities of specimens of the same type of truck brake are so highly variable. Brakes are wonderful devices in that they can absorb large amounts of energy in short periods of time. They can do this over and over without failing or wearing out prematurely. However, the price of these astounding virtues is found in the torque variability of the brake [e.g., 9,10,11].

Given that truck brakes are highly variable, and also sensitive to adjustment and past work history, it is not surprising that the performance of any particular truck in service may deviate considerably from that of another seemingly identical vehicle.

The statements that follow are based on the estimated average performance of vehicles with properly adjusted brakes that have not been degraded by high temperature operation or other abuses.

By "emergency braking," we are referring to the situation in which the driver is trying to stop as quickly as possible without losing control of the vehicle. For fully laden trucks with well-adjusted brakes, the maximum brake torque available is usually enough to provide decelerations of from 0.4 to 0.5 times the acceleration of gravity (0.4 to 0.5 g). On a good road, the maximum deceleration capability of a typical loaded heavy truck is not limited by tire/road friction but by the brakes themselves. As long as the number of brakes is in proportion to the weight of the vehicle, all combinations will have virtually the same maximum deceleration capability on a good road. In practice, some tractors have less braking capability than trailers so that combinations with more trailers tend to have slightly better maximum deceleration capability than that of combinations with fewer trailers (fully loaded triples, for example, tend to be very slightly better than doubles which tend to be slightly better than tractor-semitrailers if all units are comparable).

On slippery surfaces, or in the case of an empty combination, the maximum torque capabilities of the brakes can exceed the frictional coupling at the tire-road interface. This "overbraked" situation can cause extreme directional responses such as tractor jackknifing, trailer swinging, or dolly jackknifing. Under these circumstances, the ratio of braking force to vertical load at each wheel determines the frictional potential needed to prevent wheel lock. Ideally, for "perfect proportioning," the ratio of braking force to vertical load at each axle would be equal to the frictional potential of the roadway.

Fully loaded trucks typically have close to ideal proportioning on roads with frictional potentials less than approximately 0.4 because their brake torque capabilities are based on gross axle weight ratings. Braking efficiency (that is, the ratio of deceleration divided by the frictional potential required to prevent the least favorable wheels from locking up) may be used to quantify the ability of a vehicle to utilize the available tire/road friction. The braking efficiency of a loaded combination vehicle will be less than unity due to the load transfer caused by deceleration. This load transfer tends to unload the rear wheels of the units in a combination vehicle and load the front wheels of those units. In a tractor-semitrailer, the semitrailer wheels are unloaded, the tractor's rear wheels stay at approximately their static load, and the tractor's front wheels become more heavily loaded during a stop. For a full trailer, the rear wheels are unloaded and the dolly wheels are loaded. The consequence of this load transfer effect is a reduction in braking efficiency as the deceleration level is increased.

This load transfer effect is reduced as individual units are made longer. In this regard, 48-ft trailers are slightly more efficient than 45-ft trailers; however, the wheelbases of both of these trailers are so long that load transfer is not large and the effect on braking efficiency is small. The load transfer effect is nearly twice as large for a 28-ft trailer as it is for a 48-ft trailer with a comparable center of gravity height. Hence, combination vehicles with 45-ft or 48-ft trailers tend to have better braking efficiencies than those attainable by combinations employing 28-ft units.

These contrasts can be illustrated by calculations of friction utilization (that is, the ratio of braking force divided by vertical load at each axle during constant deceleration) for the fully loaded long combinations listed in Figure 1. In particular, results have been obtained for a deceleration of 0.4 g, corresponding to the deceleration required to stop in approximately 300 feet from an initial velocity of 60 mph--a level of emergency braking performance expected of heavy vehicles.

As shown in Table 4, the Rocky Mountain double has the lowest braking efficiency of the vehicles listed and, accordingly, requires the highest tire-road friction in order to make a 0.4 g stop. This vehicle has the lowest efficiency for two reasons. First, the 28-ft rear trailer contributes to a lower efficiency than a longer trailer would (see the results for the Turnpike double); but second, and equally importantly, the typical axle loading assumed for the 28-ft trailer in the Rocky Mountain double (see Figure 1) is less than that assumed for the double 28-ft or triple 28-ft combinations. In addition, the load on the rear axle in the Rocky Mountain double is less than that on the dolly axle of the 28-ft trailer. This loading practice, which is believed to be common, may aid directional performance in response to external disturbances; however, it is a detriment to braking efficiency.

In this example, the tractor-semitrailer has the highest braking efficiency of those studied. This is because (a) the semitrailer is long and (b) the tractor is much longer than those selected for the other combination vehicles (18-ft versus 10-ft and 12-ft wheelbases). Again, longer length produces improved braking efficiency for vehicles with simple braking systems. In this case, the effect of tractor length more than compensates for the reduced braking capability selected for the tractor.

For economic reasons, owners attempt to operate their vehicles in a fully loaded condition. By careful scheduling, some fleets are able to operate their vehicles in the fully loaded condition over 80% of the time. In other hauling operations, the vehicles must return empty so that they are fully loaded only approximately 50% of the time. Braking systems used in the U.S. do not adjust for partial or empty loads except for some systems which change the brake proportioning on a "bobtail" tractor. Since braking systems are

Table 4

Fully Laden Vehicles

Emergency Stopping Performance - Deceleration = 0.4 g

Vehicle Configuration	Braking Efficiency at 0.4 g	Friction Required for 0.4 g
Rocky Mountain Doubles		
45' semi + 28' trailer	.68	.59
48' semi + 28' trailer	.68	.59
Double 28's	.76	.52
Triple 28's	.76	.52
Turnpike Doubles		
45' trailers	.82	.49
48' trailers	.80	.50
Tractor-Semi		
48' semi (long tractor)	.85	.47

currently proportioned in favor of the loaded vehicle, directional control problems during braking are more critical for the unloaded or partially loaded vehicle.

Extreme variations in loading from one trailer to another can be particularly undesirable. Partial unloading is seen to consistently degrade the stopping capability of combination vehicles. For example, as explained in Reference [1], "The worst (partial loading) case, from the viewpoint of stopping distance performance, involves the removal of freight from the rear half of trailers." In such cases, the braking efficiencies of partially loaded vehicles can be so low that stopping distances are approximately double those attained with a fully loaded vehicle.

Clearly, then, a very important issue in the braking of heavy vehicle combinations involves operating with partial loads or with some trailers full and others empty. Unfortunately, information is not available to determine whether one type of vehicle configuration is more likely to experience partial loading than another type. Also, we do not know if some configuration is more likely to be used for transporting empty trailers. Accordingly, we are not able to distinguish between various types of long combinations with regard to partial loading--probably the most important of the emergency-braking considerations.

In summary, vehicles with longer units tend to have higher braking efficiencies than vehicles with shorter units; however, the loading of vehicles may be more critical than the length effects. Among the various long combinations of interest, variations in assumed tractor wheelbase and axle load distributions impose stronger influences on emergency stopping performance than are established simply by the "inherent" differences in configuration.

In Europe, load-sensing proportioning valves are required for heavy commercial vehicles. Theoretically, these valves would provide ideal proportioning if they worked perfectly. In the U.S., the ultimate solution to the problem of emergency braking performance--antilock systems--was tried, but the courts found existing devices to be unreliable. It may be that reliable advanced braking systems will be developed in the future. In that case, braking problems arising from both the loading and geometric configuration of combination vehicles will be largely reduced.

5.0 ISSUES RELATED TO BRAKE SYSTEM AIR DELIVERY

This discussion centers on the effects of (1) brake-timing on jackknifing and (2) pushout and/or valve-opening pressures on downhill retardation. In discussing these effects, attention will be paid to comparing long combinations (Rocky Mountain doubles, turnpike doubles, and triples) with shorter vehicles (tractors with 48-ft semitrailers and doubles with 28-ft trailers).

A typical air delivery system used in heavy truck braking consists of (1) a treadle (foot) valve for initiating and modulating braking effort, (2) air lines (both control and supply lines) for delivering signals and actuating brakes, (3) reservoirs filled with high pressure air, (4) brake chambers, (5) relay valves, and (6) other valves and connectors not pertinent to this discussion of timing and low-level braking (retardation). The treadle valve usually supplies reservoir air directly to the front brake chambers, and supplies control signals to relay valves which, in turn, modulate the air to the tractor rear brakes and to other relay valves that supply air to trailer brakes along the vehicle combination. The instantaneous pressure in a brake chamber depends upon the level of actuation of the treadle valve, the dead time associated with the transportation of air signals, and the rise (or fall) time of pressure in the brake chamber. The pressure versus time curves measured on an air system are characterized by dead time, pressure rise rate, the 0-to-60 psi ("apply time"), a "release time" needed to drop from 95 psi back to 5 psi, and the reservoir pressure drop.

In long combination vehicles, special devices are sometimes used to decrease the 0-to-60 psi apply time for the brakes on rearward axles in the "train." Electrically actuated valves may be used to reduce the delay time in the system. These valves pre-charge appropriate air lines to approximately 10 psi so that the rear brakes can respond more quickly to the level of pressure set at the treadle valve. Also, relay valves may be inserted in the control lines to increase pressure rise rates. However, the pressure out of the relay valve follows its input pressure and does not exceed the input pressure. In some cases, relay valves may add to the dead time in the system because it may take a pressure rise of 3 to 5 psi to open (crack) the relay valve. Although the use of electrical signals and relay valves can reduce the apply time, they do not prevent delays in brake application from propagating rearward throughout the brake system; rather, they reduce the amounts of these delays.

Ideally, the order of brake applications, from one axle to the next, and the associated delays are selected to (1) shorten stopping distances and (2) avoid instabilities (i.e., jackknifing the tractor or the dollies in doubles and triples). To shorten braking distances, one wants to apply all brakes as quickly as possible. Instabilities due to brake timing (as contrasted to those due to brake proportioning) are due to overrunning forces generated while leading units are braking and trailing units have not started to brake [12]. To reduce overrunning forces, one would like the rear brakes to be applied as soon as possible.

At the start of a braking operation, large overrunning forces may prevail until the braking on the trailing units catches up. If the braking capabilities of the trailing units (full trailers) are greater than leading units (tractor-semitrailers and full trailers), the combination vehicle will be in tension. This is sometimes referred to as "stretching the train." From a directional stability standpoint (as long as the wheels remain unlocked), stretching the train is viewed by many vehicle operators to be preferable to compressing the vehicle. However, the consequences of locking wheels are far more serious than the consequences of the overrunning forces that are due to typical timing relationships.

In the interest of reducing the overrunning forces generated at the beginning of a stop, experiments have been performed with the apply times of the tractor's brakes purposely extended to allow more uniform timing between tractor and semitrailer. Clearly, this compromise would increase stopping distance slightly if everything else remained equal. However, under these circumstances, drivers have a tendency to overbrake, causing wheel lock on slippery surfaces if the apply times of the tractor brakes are extended to approximately 0.6 seconds or more. Since the vehicle does not start to decelerate as soon as expected, the driver is likely to ask for more braking than the tire/road interface can support, thereby locking wheels, losing directional control, and requiring quick modulation of the brakes. Upon modulating the brakes to regain directional control, the braking distance is greatly increased over that which would be required if it were not necessary to modulate. The results of experimenting with delaying the application of tractor brakes indicate that tractor apply times should be kept short.

On the other hand, if the tractor is much faster than the semitrailer (or the dolly is much faster than its semitrailer), then braking in a turn on a slippery surface might lead to jackknifing. For tractor-semitrailers in severe turns near the limit of vehicle turning capability, extreme differences in timing between tractor and semitrailer (for example, apply times of 0.35 sec for the tractor and 1.4 sec for the semitrailer) may cause jackknifing that the driver cannot control. Although experimental results are not available to guide us with respect to dolly jackknifing, extreme differences between the brake timing at the dolly and semitrailer are believed to be necessary for timing problems, per se, to cause a dolly to jackknife. For typical timing performance levels attained with properly maintained and designed air systems, the differences in timing between tractors and their semitrailers and between dollies and their semitrailers are not large. For example, brakes at the respective units of a double with 27-ft trailers could have the following apply times (under extremely good circumstances):

tractor	0.35 sec
front semitrailer	0.46 sec
dolly	0.50 sec
rear semitrailer	0.53 sec

These differences in timing are very small and the associated overrunning forces do not pose significant control difficulty.

With regard to the five primary types of combinations that are the focus of this study, brake timing does not appear to constitute a substantial issue. On the one hand it is certainly clear that longer vehicles will have longer dead times than shorter vehicles, all other things being equal. A triple, for example, may take 0.1 to 0.2 seconds longer to begin actuating its rear brakes than a comparable double, however, this will only increase stopping distance by a few feet (less than approximately 3 feet in a stop from 55 mph). Similarly, turnpike doubles may be slightly slower than Rocky Mountain doubles which, in turn, will be slightly slower than a conventional double with 28-ft semitrailers (all things other than length being equal).

It seems that the primary sources of timing problems come from poor maintenance or mismatched systems. Research has shown that the diameters of air lines can either be too large, causing volume problems, or too small, causing flow resistance problems [11]. Depending upon the elements incorporated in a particular air system, there is an optimum diameter for air lines. In addition, apply times increase as the brakes wear and the amount of stroke of the push rod in the brake chamber increases, thereby requiring a greater volume of air for actuating the brakes. Unfortunate combinations of these effects might occur, leading to extreme differences in timing. Nevetheless, one type of long combination is not seen as more susceptible to this type of problem than the other types are.

During snubbing to reduce speed, or low-level applications such as those required for mountain descents, the differences between the application characteristics of the front and rear brakes can be important. In a snub (a brief application of the treadle valve), there is a pressure pulse that travels through the brake system causing some activation of the rear brakes, but this signal is usually attenuated as it travels to the back of the vehicle such that the front brakes tend to do more than their share of the work during snub applications with doubles and triples. This phenomenon can cause the brakes on the rearward axles to wear slightly less rapidly than the forward brakes.

A more important phenomenon can occur during mountain descents, particularly in cases where relay valves have been installed in the control lines in order to "accelerate" the pressure rise in the control lines. These relay valves may require up to 5 psi as a "cracking pressure" needed to operate. This pressure level may be such that the brakes before the relay valve do almost all of the braking while maintaining speed in a mountain descent, while those after the relay valve are not involved because the combinations of their push-out pressures and the relay valve cracking pressure are enough to limit their braking effort to a small level. This causes the forward brakes to overheat and wear even more rapidly than expected.

This problem can be worse for a triple than a double if a second relay valve is installed in the triple to accelerate the pressure signals. In this case, the brakes on the rearmost trailer may not be actuated unless the pressure at the treadle valve exceeds the sum of the pressures needed to open both relay valves.

Although this phenomenon involving relay valves has contributed to past problems in trucking operations, it is believed to be broadly recognized today such that operators generally avoid using valves with large pressure drops. Moreover, while measurable delays and attenuations are experienced in the brakes along a combination vehicle, these mechanisms are not seen as substantially discriminating among the various long combinations being examined in this study. Indeed, other design and operating influences are known to introduce such large variability into truck braking performance that the timing distinctions can be simply neglected in analyzing alternative vehicle configurations.

6.0 LOW-SPEED OFFTRACKING

When a conventional articulated vehicle tracks a steady-state circular path at low speed, each axle of the vehicle train follows a path which lies inside of that inscribed by the preceding axle. The difference between the radii inscribed, respectively, by the tractor front axle and the rearmost trailer axle constitutes the so-called "offtracking" dimension. The extent to which the paths of the successive axles differ depends upon the lengths between axle locations and articulation points along the train of vehicle elements. Longer elemental lengths in the combination introduce disproportionately greater amounts of offtracking. The "steady-state circular path" condition is attained in actual service only for long, sweeping highway curves and on exit ramps. When the radii of such curves becomes rather small, the offtracking behavior of the vehicle may be of concern because the rearmost trailer tires will tend to track off of the inside edge of the roadway.

Using a reasonable radius value of 300 ft, which is found on the lower speed section of many expressway exit ramps, the steady-state offtracking performance of the various vehicle configurations of interest have been calculated. (For the mathematical relation, see Ref. [13].) Listed below are the offtracking dimension and also the "Maximum Swept Path" dimension which is equal to the sum of the offtracking value plus the vehicle width. The swept-path measure indicates the overall width of pavement which must be provided to accommodate the passage of all of the vehicle's tires. Swept-path values assume that the vehicle is 8.5 ft (102 inches) in overall width.

Table 5

Maximum (Steady-State) Offtracking

Rank	Vehicle	Offtracking, Ft.	Max. Swept Path, Ft.
1	Double-28	2.0	10.5
2	Triple-28	2.9	11.4
3	Tr/Semi-48	3.2	11.7
4	RMD-45/28	3.4	11.9
5	RMD-48/28	3.8	12.3
6	TPD-45/45	4.9	13.4
7	TPD-48/48	5.6	14.1

When traveling initially in a straight line and then entering a curved path, the trailers of a vehicle combination begin to offtrack to an increasing degree, depending upon the length of the curve. This transient portion of the offtracking response is long enough, for example, that common tractor-semitrailers and doubles combinations do not arrive at their maximum, or steady-state, offtracking condition in the course of passing through a typical street intersection. There is, then, some point in such maneuvers at which the transient offtracking maximizes before the trailer(s) begin to straighten out again. This maximum extent of offtracking is the primary determinant of the roadway clearance space which must be provided in order to adequately accommodate a given vehicle combination at typical intersection layouts and in other tight maneuvering areas.

In order to compare the transient offtracking performance of the various long vehicle combinations which are of interest here, computerized calculations [14] of the wheel paths subtended during a 90° right-hand intersection maneuver were conducted. The tractor of the vehicle combination was guided such that its left front tire tracked a 45-ft-radius curve though the intersection. The paths, then, of the inboard-tracking right-side wheels of the combination were computed as the means for determining the extent of offtracking. As shown in Figure 8, the respective paths inscribed by the inboard edges of the right rear wheel of the tractor and the two trailers of a Double-28 combination produce successively greater amounts of maximum

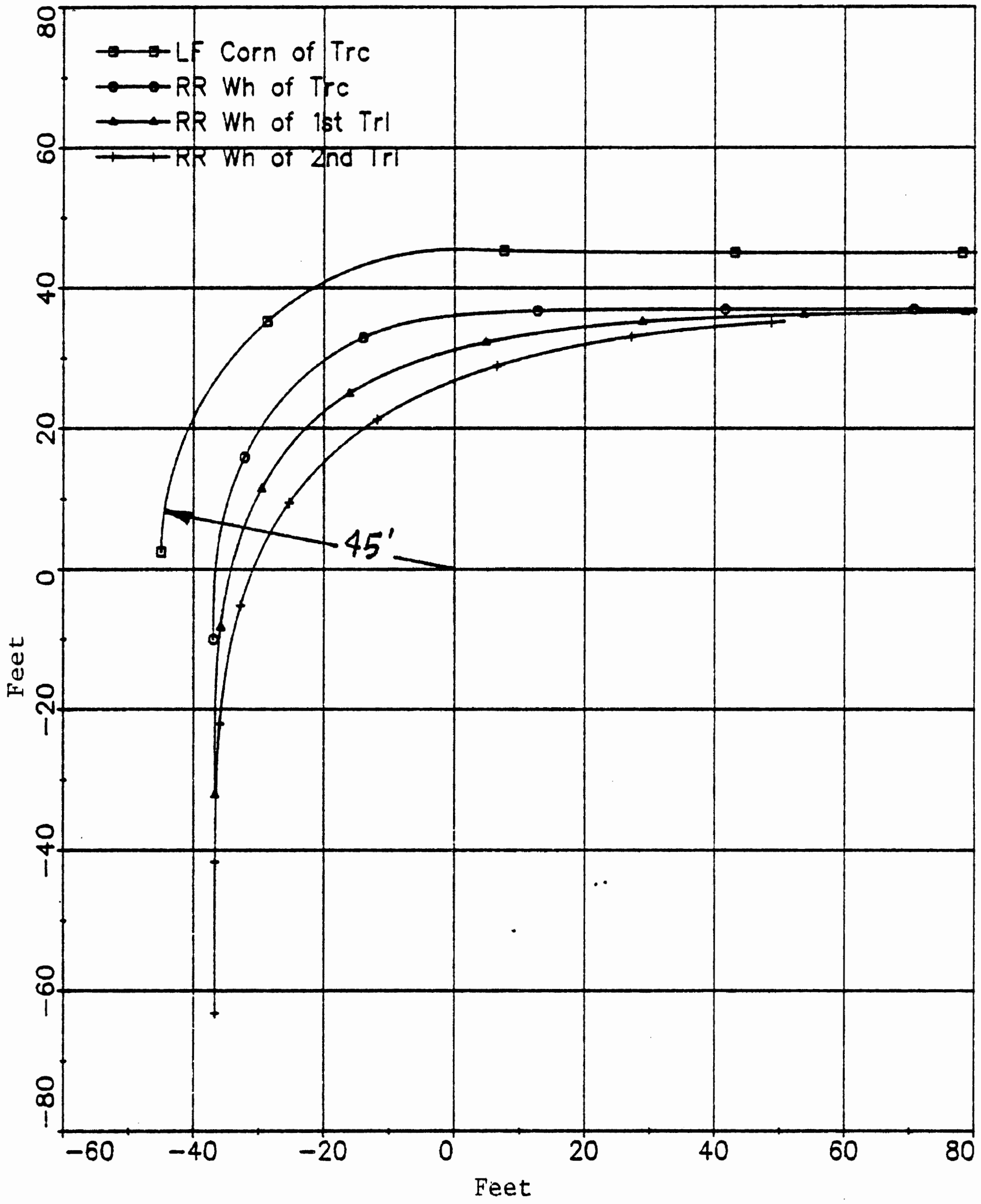


Figure 8. Trajectories inscribed by points on a doubles-28 ft combination while maneuvering around a 90° intersection, with the left front tractor tire following an arc of 45 ft radius.

offtracking in the maneuver.

Listed below, in rank order, are the values of offtracking and maximum swept path which were obtained for each of the study vehicles in the described intersection turn.

Table 6

Offtracking in a 90^{o} Intersection Turn

Rank	Vehicle	Offtracking, Ft.	Max. Swept Path, Ft.
1	Double-28	12.5	21.0
2	Triple-28	16.9	25.4
3	Tr/Semi-48	17.5	26.0
4	RMD-45/28	18.5	27.0
5	RMD-48/28	20.1	28.6
6	TPD-45/45	24.4	32.9
7	TPD-48/48	27.1	35.6

The differences in offtracking which distinguish these vehicle configurations from one another are clearly rather profound, considering that the lateral clearance dimensions available at intersections are often just sufficient for passage of the tractor-semitrailer, (Tr/Semi-48). In fact, the Tr/Semi-48 vehicle can only negotiate right-hand turns at normal urban intersections of four-lane streets by means of a technique in which the tractor begins in the left-hand lane of the entry road and actually crosses over the centerline of the exit roadway to some degree in order to assure clearance between the trailer tires and the curb. Thus, the accommodation of Rocky Mountain Doubles and, especially, Turnpike Doubles in a given road system would seem to require a careful analysis of the geometric conflicts which might be created.

The offtracking behavior of longer combinations is also seen as having some safety relevance since automobiles will tend to become "entrapped" in the space which the truck combination will "consume" during offtracking at, say, an intersection. In the aforementioned right-hand intersection turn, for example, the truck driver's need to maneuver toward the left of the entry road tends to admit right-turning autos into the adjacent, curb-side lane. As the maneuver progresses around the corner, the offtracking trailer wheels may intrude upon the location of the auto. It is suspected that such problems would magnify with the operation of RMD and TPD combinations on the conventional system of surface roads. Additionally, of course, the occurrence of offtracking motions which exceed the space provisions of the highway may cause damage to the shoulders and to the roadside appurtenances.

7.0 HIGH-SPEED OFFTRACKING

While the trailers of articulated vehicles track inboard of the tractor during slow speed turns, the tracking relationships change as speed is increased. When such vehicles travel around a curved path at increasing speed, the offtracking begins to diminish and actually becomes zero at some speed. At higher values of speed beyond this point, the trailer tires track to the outside of the path of the tractor tires [15]. This outboard, or "high-speed," offtracking phenomenon is thought to be of potential significance to traffic safety insofar as the potential exists for the rear of the trailer to strike an object on the outside of the curve or for trailer tires to encounter an outboard curb. The latter case is seen as an occurrence which could precipitate rollover. Also, it is apparent that truck drivers are generally unaware of this outboard tracking behavior and thus may not be likely to place the tractor sufficiently inboard in such turns to compensate for the displaced path of the trailer wheels.

The extent to which the trailer tires track outboard of those on the tractor is dependent upon the speed of travel and the radius of turn. The most likely type of road section in which a sufficiently tight radius curve is available and in which vehicles are likely to be travelling at higher speeds is an expressway ramp. On assuming a rather severe cornering condition, namely a 55-mph speed and a 600-ft radius curve, calculations of high speed offtracking have been made for the long combinations of interest. The results are given in Table 7 below.

Table 7

High-Speed (55 mph) Offtracking in a 600-ft Radius Curve

Rank	Vehicle	Hi-Speed Offtracking, ft.
1	Tr/Semi-48	0.52
2	TPD 48/48	1.10
3	TPD 45/45	1.25
4	RMD 48/28	1.33
5	Double-28	1.43
6	RMD 45/28	1.45
7	Triple-28	2.13

Although there are substantial percentage differences among the high-speed offtracking values listed above, the absolute values of the offtracking dimensions are not large. While the tabulated values of high-speed offtracking assume that radial tires are installed, vehicles equipped with bias-ply tires will exhibit on the order of twice the offtracking magnitudes shown here.

Moreover, the accident potential posed by the high-speed offtracking phenomenon is thought to be very low, although individual incidents have undoubtedly occurred due to this mechanism.

8.0 STABILITY ISSUES RELATED TO RAPID STEERING MANEUVERS

In transient turning maneuvers, the rear unit of a multi-articulated vehicle may experience a maximum level of lateral acceleration that substantially exceeds the maximum level of lateral acceleration of the lead unit of the vehicle. This phenomenon, referred to as "rearward amplification," is the dominant performance property distinguishing the yaw response of multi-articulated vehicles from that of other commercial vehicles [16].

Intuitively, rearward amplification may be thought of as the propensity for multi-articulated vehicles to "crack the whip" in sudden obstacle-avoidance maneuvers. An important consequence of this amplified motion is the tendency for the rear trailer of a multi-articulated vehicle to roll over (even if the driver manages to avoid the obstacle which precipitated the evasive maneuver) [17].

In general, maximum levels of lateral acceleration in such maneuvers increase from one unit to the next unit, starting from the front of a "train" of articulated units. For example, in a double the semitrailer typically has nearly the same lateral acceleration level as the tractor, but the full trailer may have a much higher lateral acceleration level than that of the semitrailer. In a triple, the second full trailer may have a much higher lateral acceleration level than that of the first full trailer. Hence, as the number of identical articulated units is increased, the amount of rearward amplification is increased (where rearward amplification is quantified as the ratio of the lateral acceleration of the rearmost unit divided by the lateral acceleration of the tractor).

The amount of rearward amplification depends upon (a) the frequency of steering input, (b) the speed of the vehicle, (c) the lengths of the units involved, (d) the locations of the pintle-hitch connections between units, and (e) the ratio of tire lateral stiffness properties to the weight of each unit [18]. With regard to the long vehicle combinations being considered in this study, the lengths of the units are highly significant while the other factors (a, b, d, and e, above) are less important in explaining differences between vehicle types. All else being equivalent, longer trailers contribute less to rearward amplification than short trailers do. As shown in Figure 9, the triple-28s configuration has a predicted rearward amplification (amplification gain) of almost 2.5, which is much larger than the rearward amplification for the double-28s combination or for any of the other combinations employing 45-ft or 48-ft trailers. This result is to be expected because the triple has (a) the greatest number of articulation joints and (b) the shortest units in this set of vehicles.

The results presented in Figure 9 clearly indicate that doubles combinations with 28-ft trailers have higher rearward amplifications than those attained by longer doubles combinations. The differences in amplification obtained with 45-ft versus 48-ft semitrailers are small. The maximum values of amplification gain for each of the individual units in the Turnpike Double, for example, are all close to 1.0, such that the overall amplification gain of the Turnpike Double combination is nearly perfect, from a rearward amplification standpoint.

The analytical results that have been discussed here apply to vehicles which are loaded rather uniformly per the schemes laid out in Figure 1. In contrast to these results that apply to uniformly loaded vehicles, various types of undesirable rearward amplification phenomena can be brought about by unusual and non-uniform partial loading arrangements [1,17]. In particular, rearward-biased load distributions can create not only large amounts of rearward amplification, but also yaw stability problems in extreme situations. Since it is not known whether one type of configuration is more prone to rear-biased loading than another type, it is not possible to distinguish the various types of long combination vehicles on the basis of partial or non-uniform loading (other than to observe that vehicles having lower values of rearward amplification in the uniform loading state offer more of a "cushion" for tolerating non-uniform loading conditions than do vehicles exhibiting high levels of amplification).

Moreover, rearward amplification performance constitutes the primary basis for distinguishing the <u>dynamic</u> performance quality of one multi-trailer combination from the next. It is fundamental that the vehicles exhibiting the higher levels of amplification gain are also those that rate relatively better

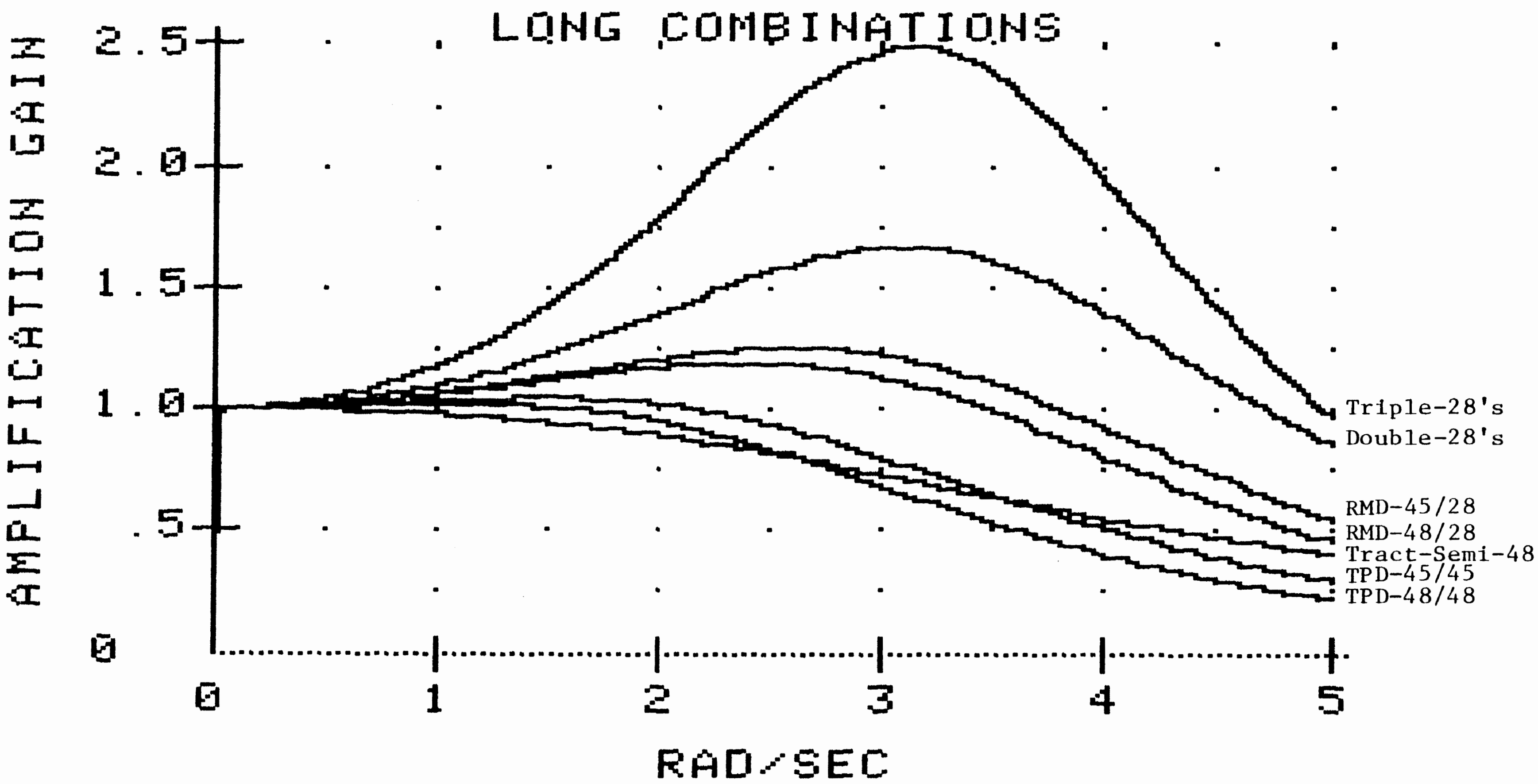


Figure 9. Rearward amplification at the rear trailer of differing combinations as a function of tractor steer input frequency.

in terms of low-speed offtracking. It is also known that new coupling devices are being developed to mitigate against rearward amplification, while also providing good offtracking performance. These innovative approaches have been discussed in Section 2.0.

9.0 ROLL STABILITY

For purposes of this discussion, we will express the roll stability level of a given truck combination in terms of the "rollover threshold," which is defined as that maximum level of steady lateral acceleration which can be tolerated without suffering the rollover of any of the units in the vehicle combination. The "lateral acceleration" level would equate, for example, to a certain value of speed for a vehicle travelling through a curve of fixed radius. The rollover threshold measure is of interest since this expression of inherent roll stability has been seen to correlate to a remarkable degree with the actual involvement of vehicles in rollover accidents [19]. Thus, truck roll stability is looked upon as a key property influencing vehicle safety.

The roll stability level of a given vehicle is known to be determined by the obvious parameters of height of center of gravity (c.g.) and width of the tire track, as well as more subtle properties concerning the stiffness and geometric details of the suspension designs and the stiffness of the tires. In multiple-unit vehicle combinations, the distribution of suspension properties from one axle to another is also known to be of significant importance to roll stability [19]. The fact that typical tractors employ relatively softer suspensions than trailers, for example, plays an important role in limiting the roll stability of tractor-semitrailer combinations. Since converter dollies generally employ suspension stiffness levels which are comparable to those found on the attached trailers, full trailers can be viewed as being reasonably well "balanced" in behalf of roll stability, as far as suspension considerations go.

The primary operating variable which distinguishes the relative roll stability of heavy commercial vehicles on the road is simply the weight of the payload and its c.g. height. Clearly, the parameters of the payload depend entirely upon the type of commodity being transported. Trucks hauling flat steel plate will typically exhibit a higher rollover threshold, for example, than trucks hauling lumber, since the high density of steel yields a payload c.g. height which is substantially lower than that yielded by the lighter-density wood products. Thus, to meaningfully compare the roll stability of two differing types of truck combinations, such as the Tr/Semi-48 vs. the RMD 48/28, one would need information on the differences in the typical payloads which will be carried in the two vehicles, as well as knowledge of any hardware distinctions pertinent to the vehicle designs.

Since the longer truck combinations clearly offer greater cubic capacity than the more conventional vehicles, but with generally lighter axle loadings, one would imagine that wider usage of the longer vehicles might be focused predominantly upon the transportation of lower-density commodities. Depending upon the payload densities and weights which would actually typify the service applications of these vehicles, however, the longer vehicles might exhibit a greater or lesser level of roll stability than more conventional vehicles carrying the same commodities.

Regarding the suspension hardware employed on the various long truck combinations, there are no known distinctions among basic vehicle configurations which would suggest differences in inherent roll stability.

Accordingly, these reflections suggest that the roll stability of the various vehicle combinations of interest in this study cannot be meaningfully compared without explicit knowledge of the distinctions in the payloads which would be carried in the respective vehicle types. Since such information is unavailable, no definitive projections of roll stability limits can be produced here. In the authors' view, however, the typical practices employed in the design of tractors, trailers, and dollies are such that no large distinctions in roll stability should be expected if comparable types of payloads are involved in the comparisons.

10.0 YAW STABILITY

When heavy-duty trucks and tractor-trailer combinations are driven around a curved path at elevated speeds, it is possible that a so-called "yaw instability" may occur prior to reaching the level of maneuver severity at which rollover occurs. The "yaw" motion of a vehicle pertains to its rotation about a vertical axis. This is the primary mode of motion which is involved in cornering maneuvers. A "yaw instability," then, pertains to a condition in which the yawing rotations tend to grow at an increasing rate, perhaps causing the vehicle to "spin out," with the vehicle pointing well away from its direction of motion as the spinout progresses. Clearly, a yaw instability would threaten vehicle control insofar as the driver is challenged by the task of steering to obtain a stable motion from an unstable system [20].

The yaw instabilities which are known to be possible during the pure cornering operation of heavy-duty combination vehicles involve only the tractor. That is, with combination vehicles, this phenomenon simply involves a tendency for the tractor to jackknife with respect to the semitrailer. The rapidity of such a jackknife is much slower than that which occurs during overbraking of the tractor drive wheels. In fact, in many cases it may be that tractor yaw instability is effectively compensated by capable driver action at the steering wheel such that loss-of-control is averted and no particular note is made of the event. Nevertheless, the popular wisdom in the vehicle dynamics community holds that such unstable vehicle behavior is undesirable and should be avoided, if possible.

Fortunately for the purposes of the study of long combinations, the mechanisms serving to cause tractor yaw instability are virtually unaffected by the configuration of the trailing elements aft of the tractor. In fact, considering the Rocky Mountain Doubles, Turnpike Doubles, and triples of interest here, we could generalize that there are no first-order influences on tractor yaw stability distinguishing these vehicle combinations. Probably of greater significance than trailer configurations, per se, are the axle load values being borne at the respective tractors and the c.g. height which follows from the prospect that certain vehicle configurations may become more popular in hauling commodities which differ in density from those carried on other types of vehicles. Insofar as the commodity-preference trends are unknown, it is fair to say that the yaw stability subject does not provide a basis upon which to obtain any meaningful comparisons of the alternative long truck combinations.

11.0 TRUCK POWER REQUIREMENTS

As the size and weight limits on trucks are relaxed, allowing larger and heavier vehicle combinations, the power of the engine installed in the vehicle must be increased, or reductions in the performance of the vehicle will result. Reductions in performance which are of public concern include those which lead to interference in the traffic stream, either diminishing the traffic capacity of the highway, or diminishing the safety of operations. An analysis is performed in this section in which the comparative power requirements of larger and heavier vehicles is assessed under certain selected situations.

11.1 Analytical Method and Assumptions

The power required to propel a vehicle on the road is proportional to its speed and the road load drive force. That power is derived from the engine, although the portion applied to the road as tractive effort is reduced by various engine accessories and by friction in the driveline. The road load forces are caused by rolling resistance, aerodynamic drag, and drag arising from grade of the road [21]. The rolling resistance is most directly proportional to the gross weight of the vehicle, although speed has a secondary effect. For vehicles in the future, the rolling resistance is best estimated by equations developed by the SAE and others for radial tires. In this analysis, the following equation for rolling resistance coefficient is used:

$$CRR = .001 \cdot (4.1 + .041 \cdot V)$$

where

CRR = rolling resistance coefficient

V = travel speed in miles per hour

The aerodynamic drag is the result of the dynamic pressure of the relative wind acting on the front of the tractor and trailers. Thus, it is related to the square of the speed of the relative wind approaching the vehicle. In the case of headwinds, the headwind component adds directly to the velocity of travel in the determination of the relative wind speed. The frontal area on which it acts is the simple product of the width and height of the vehicle, which for tractors with van trailers is about 100 square feet (8 feet wide by 12.5 feet high). The aerodynamic properties of the vehicle are taken into account by an aerodynamic drag coefficient which is multiplied by the dynamic pressure and the area to obtain the force. The common wisdom is that the aerodynamic drag coefficients vary between 0.7 and 0.9 for large trucks and tractor-trailers. The higher value of 0.9 was selected for these calculations, however, to account for the influence of the cross-wind component which seems to more properly represent the majority of travel on the highway.

The final drag contributor is the gravity component of the vehicle weight which acts in opposition to the direction of travel when a vehicle proceeds on an upgrade. The force developed is the product of the grade (expressed as rise over run) and the vehicle weight.

Using these assumptions, the power requirements were calculated for different vehicle combinations operating at different speeds and road conditions. The primary factor by which the vehicles are differentiated is the gross combination weight. All vehicles are assumed to be comprised of van trailers with the same frontal areas and aerodynamic drag coefficients, and are equivalent in all other relevant respects.

11.2 Travel on Level Roads

The power required to move the vehicle at a constant 55-mph speed on a level road was determined for the case of no headwind, and for a headwind of 25 mph. The results are presented in Table 8, below.

Table 8. Horsepower Requirements for Level Road Travel

Gross Weight	No Wind	25 mph Headwind
80,000 lbs	227 HP	383 Hp
105,000	255	411
117,000	268	425
120,000	271	428
150,000	304	461
200,000	359	516

The 80,000-lbs gross weight combination is included in this table to give the reader a reference point. The 80,000-lb vehicle is the current maximum allowed in all states, and represents a major segment of the tractor-semitrailers on the roads today.

11.3 Travel on Grades

In a similar fashion, the power requirements for negotiating grades at speeds of 20 mph and 30 mph were also calculated. The results are shown in Table 9, below.

Table 9. Horsepower Requirements on Grades

	20 mph					30 mph				
	Grade					Grade				
Gross Weight	2%	3%	4%	6%	8%	2%	3%	4%	6%	8%
80,000 lb	132	182	229	332	433	213	289	364	515	665
105,000	170	236	302	434	566	273	372	470	668	866
117,000	190	263	337	483	630	302	412	522	742	961
120,000	194	270	334	495	646	308	421	535	760	986
150,000	241	336	430	617	806	380	522	662	945	1227
200,000	319	445	571	822	1071	499	688	876	1252	1628

11.4 Acceleration Performance

The acceleration performance when a truck starts out is similarly influenced by power level of the engine. Consequently, the distance along the road needed for the truck to get up to speed is affected by engine power. Assuming that the engine is operated at wide-open throttle (for maximum power output), the power in excess of that needed to overcome rolling resistance, aerodynamic drag, and grade is applied to accelerating the vehicle. Thus, a portion of the energy delivered from the engine is transformed into an increasing kinetic energy of the moving vehicle. On steep grades (4% or more) where the maximum attainable speed is low (15 to 30 mph), the kinetic energy associated with the vehicle speed is low in comparison to the energy that is dissipated in rolling resistance and in overcoming the grade. Hence, the distance required for accelerating a vehicle to the maximum speed on steep grades is relatively short for all trucks. Typically, 90% of the final speed can be achieved in distances measured in fractions of a mile. However, as the steepness of the grade diminishes from 4%, the attainable speeds become much larger, approaching the limit of 55 mph. In those cases, the acceleration distances can become very large, depending on the ratio of engine power to gross weight. Acceleration on a shallow grade is, therefore, the situation in which the most profound differences in performance will be observed, as a function of engine power and gross vehicle weight.

A series of calculations were performed to examine the relative performance of different trucks when accelerating from a stop on 0% and 3% grades. The overall distance required to achieve a given speed can be very sensitive to the engine power level; i.e., on a vehicle powered to achieve a maximum speed of 55 mph on a level roadway, at least in theory, it takes an infinite distance for the vehicle to actually reach that speed. Thus, it is useful to assess the acceleration performance of candidate future vehicle combinations by comparison with that of typical vehicles now on the road. For the purpose of comparisons here, the baseline is an 80,000-lb GCW tractor-van trailer powered by a 250-horsepower engine. In the comparative calculations all other vehicle properties were held constant except for gross weight and engine power. The objective was to determine the horsepower required at different gross combination weights to achieve speed-distance performance comparable to the 80,000-lb vehicle. On a level roadway, the 80,000-lb vehicle requires approximately 6,250 feet to accelerate from a standing start to 50 mph. The horsepower required to achieve this same speed at 6,250 feet is calculated for the other values of gross vehicle weight. Also, on a 3% upgrade, the maximum speed achievable by the baseline 80,000-lb vehicle is about 27 mph. From a standing start, the 80,000-lb vehicle can accelerate on the 3% grade to 20 mph in a distance of 625 feet. The horsepower required to achieve this same speed and distance objective was determined for the other vehicle combinations as listed in Table 10.

Table 10. Horsepower Requirements for Acceleration

	Horsepower	
Gross Weight	50 mph in 6250 ft	20 mph in 625 ft (3%)
80,000 lb	250 HP	250 HP
105,000	305	320
117,000	330	355
120,000	335	365
150,000	400	460
200,000	505	605

It may be noted for the case of acceleration to 20 mph that equivalent performance is obtained when all vehicles have approximately the same weight-to-horsepower ratio (330 lb/hp). The reason is that the major fraction of the energy goes into acceleration, overcoming rolling resistance and overcoming grade. The magnitude of all of these factors is directly dependent on the vehicle mass (gross weight), hence, the power must increase in proportion to gross weight.

By contrast, in accelerating to 50 mph, aerodynamic drag consumes a larger percentage of the energy output of the engine. However, the aerodynamic drag is not directly related to gross vehicle weight. Thus, the heavier vehicles do not require proportionate increases in horsepower to achieve the same performance when accelerating to high speed. Whereas the 80,000-lb vehicle has a weight-to-horsepower ratio of nearly 320 lb/hp, a ratio of nearly 400 lb/hp is adequate to achieve equivalent high-speed acceleration performance in a 200,000-lb vehicle.

11.5 Discussion and Conclusions

The suggested benchmark for assessing appropriate power levels for various truck combinations is the 80,000-lb unit included in the calculations. The minimum engine power for tractor-trailers in this GCW range is typically about 250 hp in order to be capable of 55 mph travel on normal, level road conditions (see Table 9). Achieving comparable performance from other tractor-trailer combinations is most demanding in the hill-climbing mode, where an equivalent power-to-weight ratio is the governing factor. Applying this rule leads to the following conclusions with regard to appropriate minimum engine power for various long vehicle combinations:

Rocky Mountain Doubles (105,000 lb)	- 325 hp
Turnpike Doubles (120,000 lb)	- 365 hp
Triples (115,000 lb)	- 350 hp

Special vehicles in the GCW ranges of 150,000 lbs and 200,000 lbs would require 460 and 600 hp, respectively.

12.0 SUMMARY OF OBSERVATIONS

In this section, the various observations which have distinguished the dynamic performance of the differing long combinations from one another will be summarized. The digest of observations will be ordered by performance category.

BACKING-UP

1) Performance Contrasts -- All doubles and triples were found to be impossible to back up for distances in excess of 18 feet or so, unless a locking feature were employed on the installed dollies. The triple-28 combination was found to be more limited, with locking-dolly employed, than were the various doubles combinations. Nevertheless, the ability of all of the multi-trailer combinations to be backed up is severely constrained, at best. The key feature of these vehicles which tends to normalize all of the configurations toward the same, rather poor, backing performance is the short length of the dolly drawbar. If any of these units were to be built in B-train configurations, backing-up performance would improve dramatically. By any measure, however, the conventional tractor-semitrailer is vastly more amenable to being backed up than is any multi-trailer combination.

2) Significance -- The ability to back up a vehicle is seen as an issue of operational flexibility more than a safety matter. It is recognized, however, that in certain scenarios, the inability to back up multi-trailer combinations may serve to create a traffic blockage problem. Overall, it is the authors' view that there is little significance to the small distinctions in the backing ability of the various long combinations studied here. That is, all of the longer combinations offer little opportunity for backing up more than, say, a car length.

SPEED CONTROL DURING MOUNTAIN DESCENTS

1) Performance Contrasts -- Since the Rocky Mountain and Turnpike Double provided relatively more foundation brakes, per pound of gross vehicle weight,

the downhill speed control capability of these vehicles (assuming the use of foundation brakes, only) is enhanced over that of conventional tractor-semitrailers and doubles loaded to 80,000 lbs. The triples combination, loaded to a gross weight of 115,000 lbs, was seen as providing nominally the same level of dowhill braking capacity as is afforded by conventional vehicles. Also, the transmission of the air brake signal from axle to axle in long combination vehicles is done in such a way that no strong distinctions in downhill braking of the various longer combinations should accue due to air delivery mechanisms.

Considering the use of retarders as a substitute means for controlling downhill speeds, all of the longer combinations are deficient in comparison with conventional vehicles. This deficiency derives from the fact that the retarder typically acts only upon the drive wheels of the tractor. Since there is a practical limit to the amount of retarding horsepower which can be "handled" through these drive wheels, regardless of the gross weight of the vehicle, the heavier weights posed by the longer combinations render such vehicles "under-retarded" in the mountain descent scenario.

2) Significance -- For the Rocky Mountain and Turnpike Doubles, there are "give and take" distinctions which show them to be either more or less capable of downhill speed control than conventional vehicles. From a safety point of view, considering the foundation brakes to be providing either the primary energy absorption device or the "backup" system during retarder usage, these two larger vehicle combinations offer an advantage over conventional vehicles. Thus, it would seem hard to argue that the likelihood of runaway accidents with such vehicles on mountain highways would be greater than is experienced with conventional vehicles. The triples combination, on the other hand, offers essentially no advantage in the capacity of its foundation brake system. Thus, with the strong reduction in effectiveness of an engine retarder, the triple is simply more marginal in downhill capacity. Of course, these obersvations hinge entirely upon the assumed values of gross vehicle weight, as outlined in the text.

EMERGENCY STOPPING PERFORMANCE

1) Performance Contrasts -- Calculated braking performance results showed

differences between differing long vehicle combinations. These differences were noted to be primarily due to specified loading conditions and selections of tractor wheelbase. Since insufficient data exist to provide a solid case for these specifications, the calculated results are seen as more illustrative than conclusive regarding truly representative behavior. Further, it is known that truck brake systems are tremendously variable in their stopping performance in actual service, such that the definition of a specific performance level for any given configuration is an unrealistic expectation. Also, the timing delays associated with propagating the brake system air signals are not seen to be of such a magnitude that they discriminate to a significant degree among the various long combinations.

Thus, the authors observe that there is insufficient basis for concluding an inherent deficiency in the emergency stopping performance of one long combination relative to another. Further, the existing state of knowledge does not suggest that any strong distinctions in this performance area should be expected, were complete data to be made available.

2) Significance -- It seems intuitively clear, if not demonstrable from accident studies, that vehicles should be able to stop in acceptably short distances, and in a controllable manner. In this regard, it is known that heavy-duty trucks are generally deficient in comparison with passenger cars. Thus, the stopping performance of heavy vehicles, generically, seems clearly a safety issue. Nevertheless, there does not seem to be a substantive basis for concluding that the emergency stopping capability of various types of truck combinations differ in a generally significant way.

LOW-SPEED OFFTRACKING

1) Performance Contrasts -- There are profound differences in the low-speed offtracking performance of the longer combinations. While the triples combination was seen to offtrack somewhat less than even the conventional tractor-and-48-foot-semitrailer combination, the Rocky Mountain and Turnpike Doubles exceed the offtracking of this conventional vehicle by 6 to 55%.

2) Significance -- Low-speed offtracking on tight-radius turns and at

intersections risks intrusion of the trailer wheelpaths into the space occupied by highway appurtenances and other vehicles. Thus, large differences in vehicle offtracking are significant for reasons of property damage and maintenance of the highway system as well as traffic safety. The magnitude of the differences posed by the Turnpike Double are clearly so large that a road network intended for usage by this vehicle would require special geometric provisions. Similarly, the calculated results suggest that the Rocky Mountain Double cannot be readily accommodated on most of the U.S. road system.

HIGH-SPEED OFFTRACKING

1) Performance Contrasts -- The high-speed offtracking response is greater with vehicle combinations having the higher number of short-wheelbase trailers. Thus, results showed that the triple, with three 28-ft trailers, registered the highest value of high-speed offtracking. Although the Rocky Mountain Double with the 45-ft lead trailer was the next highest in this unfavorable characteristic, also high on the list was the conventional double having two 28-ft trailers.

2) Significance -- The magnitude of the high-speed offtracking dimension is highly sensitive to certain stiffness properties of the installed tire. Given the inexorable trend toward radial (i.e., stiffer) tires in the U.S. trucking fleet, the high-speed offtracking behavior of typically-equipped vehicles will involve rather small offtracking dimensions such that a minor level of safety significance is suspected. Thus, this measure does not discriminate among the various longer combinations in a manner which is thought to imply strong differences in accident potential. On the other hand, for vehicles equipped with bias-ply tires, the distinctions in high-speed offtracking for the examined vehicles would appear to be important to safety.

STABILITY IN RAPID STEERING MANEUVERS

1) Performance Contrasts -- The measure of most concern in rapid steering maneuvers is the rearward amplification gain which describes the potential for premature rollover of the rear trailer in a multi-trailer combination. Clearly, the triple poses a much greater hazard in such dynamic maneuvers, by this measure, than do any of the combinations incorporating one or more long

(45- or 48-ft) trailers. Although the conventional double exhibits an amplification level which is considerably below that of the triple, these two vehicles basically stand in a class by themselves relative to the rest of the vehicles which were considered.

2) Significance -- There is ample evidence in the accident record that various examples of multiple-trailer configurations having high levels of rearward amplification have suffered an extraordinarily high incidence of accidents in which only their rearmost trailer has overturned [17,22,23]. Although the absolute frequency of such accidents is low, (being measured in, say, incidents per 100 million vehicle miles), there seems little question that high levels of rearward amplification portend increased likelihood of rollover involvement. Thus, it is the authors' view that the relatively high levels of amplification of the conventional double, and certainly the triple, call for the development of improved vehicle hardware to mitigate the problem. Section 2.0 of this report spoke to such developments.

ROLL STABILITY

1) Performance Contrasts -- The "roll stability" subject was addressed in terms of the "static" stability which pertains to the resistance to rollover in smooth, steady, turns (in contrast to the dynamic rollover conditions prevailing during rapid steering maneuvers, such as discussed above. The inherent static roll stability of the various long combinations, considering some normalized loading arrangements, are essentially equivalent. Thus, if these vehicles were actually to be transporting commodities which rather uniformly loaded each of the respective axles, with a uniform height of the center of gravity of the payload, no significant distinctions in roll stability would be expected. Since data are not available to suggest that any anomalous kinds of loading patterns or commodity densities should be expected in the various types of combinations, no further findings on relative roll stability level can be made here.

2) Significance -- The rollover of heavy commercial vehicles is seen to be a very important part of the accident picture with such vehicles. It is rather widely recognized, for example, that over half of all truck driver fatalities occurring each year derive from rollover accidents. Further,

research has shown a very powerful relationship between the static rollover threshold of tractor-semitrailers and the frequency of their involvement in rollovers [19]. Thus, it is of significance that no inherent distinctions in the static roll stability of the various long combinations can be identified. This is not to say that the roll stability level of the long combinations, or any other heavy-duty commercial vehicle, for that matter, is up to the level at which rollover would be a small problem. Rather, these observations establish that no grounds exist for discriminating against the use of differing long combinations simply on the basis of static roll stability.

YAW STABILITY

1) Performance Contrasts -- The yaw stability issue pertains to the control qualities of the tractor in response to steering--particularly under more severe cornering conditions. No distinctions in this property, among the various long combinations, could be identified. As with the roll stability subject, explicit information on the density and distribution of payload within the trailers would be required, for each combination type, for definitive distinctions to be drawn. In the absence of such data, one can only observe that there are no inherent distinctions in vehicle design by which to discriminate the yaw stability of differing long combinations.

2) Significance -- The significance of the yaw stability subject to traffic safety is unknown. One can only observe that the popular wisdom of the technical community holds that avoidance of unstable yaw behavior is highly desirable. Since it is clear that properly trained drivers can maintain control even when the vehicle manifests an unstable yaw response characteristic, the matter is not clear cut. Moreover, as in the case of the findings on roll stability, no grounds exist for discriminating among the various long combination vehicles on the basis of yaw stability properties.

POWER REQUIREMENTS

1) Performance Contrasts -- Since any tractor can be coupled to any set of trailers, there is no inherent connection between a given multiple-trailer configuration and its "powering capabilities." Thus, the subjects of speed capability on grades and acceleration performance reduce to requirements for

engine horsepower, given the gross vehicle weight. Calculations showed that the respective long combinations would be capable of at least the minimum performance levels of conventional vehicles if they incorporated the following engine power ratings:

Rocky Mountain Double (105,000 lb, GVW)	-- 325 HP
Turnpike Double (120,000 lb, GVW)	-- 365 HP
Triples (115,000 lb GVW)	-- 350 HP

2) Significance -- The ability of commercial vehicles to achieve highway speeds and to maintain reasonable, although reduced, speeds on grades influences both the efficiency with which other traffic can use the highway and the potential for accidents due to speed differentials between vehicles. Regarding the latter, it has been shown that the accident rate will increase strongly when the difference between truck speed and the average running speed of other traffic exceeds approximately 10 mph [24]. Thus, the issue of engine power is seen as having a clear significance. Since engine ratings such as those listed above are readily available, however, the achievement of a minimum powering capability would not constitute an impediment to the use of long combinations.

REFERENCES

1. Ervin, R.D., et al. "Influence of Size and Weight Variables on the Stability and Control Properties of Heavy Trucks." Final Rept., Contract No. FH-11-9577, Transp. Res. Inst., Univ. of Mich., Rept. No. UMTRI-83-10, March 1983.

2. Woodrooffe, J.H.F., Billing, J.R., and Nisonger, R.L. "Improved Stability and Handling of Truck Combinations with the Double Drawbar Dolly." SAE Paper No. 831162, August 1983.

3. Winkler, C. "Techniques for Improving the Dynamic Ability of Multi-Trailer Combination Vehicles." Contract No. DTFH61-84-C-00026, Transp. Res. Inst., Univ. of Mich. (research currently in progress).

4. Woodrooffe, J.H.F. "Accident Investigation Report - Rollover of Kleyson Unit No. 815, Approximately 13 km North of Whitewood, Saskatchewan, April 26, 1984." National Research Council Canada, Rept. No. TR-RW-1, April 1984.

5. Sharp, R.S. "The Steering Responses of Doubles." IAVSD Symposium, Berlin, September 1979.

6. Fancher, P.S. Pavlyuk, A., and Segel, L. "Trackless Train Hitch." Disclosure of Invention, Univ. of Mich., 1980.

7. Newcomb, T.P. and Spurr, R.T. Braking of Road Vehicles. Chapman and Hall Ltd., London, 1967.

8. Fancher, P.S. and Winkler, C.B. "Retarders for Heavy Vehicles: Phase III Experimentation and Analysis; Performance, Brake Saving, and Vehicle Stability." Final Rept., Contract No. DOT-HS-9-02239, Transp. Res. Inst., Univ. of Mich., Rept. No. UMTRI-84-4, January 1984.

9. Office of Heavy Duty Vehicle Research. "Stopping Distance Test Data on Air Braked Vehicles." National Highway Traffic Safety Admin. U.S. Dept. of Transp., February 7, 1980.

10. Federal Register 39:221, November 4, 1974, pp. 4016-4019.

11. Radlinski, R., Machey, J., and Williams, S. "The Importance of Maintaining Air Brake Adjustment." SAE Paper No. 821263, November 1982.

12. Vehicle Research and Test Center. "Development of a Truck-Tractor Pneumatic Simulator and Optimization of Trailer Air Brake Plumbing." National Highway Traffic Safety Admin., U.S. Dept. of Transp., Memorandum Rept. CAR-5, April 1981.

13. "Turning Ability and Offtracking--Motor Vehicles." SAEJ695b, SAE Recommended Practice, Soc. of Auto. Engrs. Handbook, 1981.

14. Sayers, M. "FHWA/UMTRI Vehicle Offtracking Model and Computer Simulation--User's Guide." Transp. Res. Inst., Univ. of Mich., June 1984.

15. Bernard, J.E. and Vanderploeg, M. "Static and Dynamic Offtracking of Articulated Vehicles." SAE Paper No. 800151, February 1980.

16. Hazemoto, T. "Analysis of Lateral Stability for Doubles." SAE Paper No. 820973, August 1982.

17. Ervin, R.D., et al. "Ad Hoc Study of Certain Safety-Related Aspects of Double-Bottom Tankers." Final Rept., Highway Safety Res. Inst., Univ. of Mich., Rept. No. UM-HSRI-78-18, Sponsored by Mich. State Office of Highway Safety Planning, Contract No. MPA-78-002A, May 7, 1978.

18. Fancher, P.S. and Segel, L. "Offtracking Versus Stability and Dynamic Response of the Trackless Train." 8th IAVSD Symposium, Cambridge, Mass., August 15-19, 1983.

19. Ervin, R.D. "The Influence of Size and Weight Variables on the Roll Stability of Heavy Duty Trucks." SAE Paper No. 831163, August, 1983.

20. Ervin, R.D. and Mallikarjunarao, C. "A Study of the Yaw Stability of Tractor-Semitrailer Combinations." Proceedings, 7th IAVSD Symposium, Cambridge, England, September 7-11, 1981.

21. Huff, T.S. and Scrivner, F.H. "Simplified Climbing-Lane Design and Road-Test Results." Highway Res. Bd. Bulletin 104, 1955.

22. Heath, W.M. "California Tank Truck Accident Survey - February 1, 1980 to January 31, 1981." California Highway Patrol, December 1981.

23. Consolidated Freightways Corp. of Delaware vs. Larson, et al., Civil No. 81-1230 (M.D.PA., Filed Oct. 27, 1981), Exhibits 1010 through 1024 and 1070 through 1094.

24. Glennon, J.C. "An Evaluation of Design Criteria for Operating Trucks Safely on Grades." Highway Res. Record 312, 1970.