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Truck Design Optimization Project Phase II

Performance Characterization of Type I Freight Car Trucks

Office of Research and Development

Washington, D.C. 20590

Wyle Laboratories
Scientific Services
& Systems Group

Colorado Springs Division
4620 Edison Avenue
Colorado Springs, Colorado 80915

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P.V. RamaChandran
M. M. ElMadany

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03 - Rail Vehicles & Components

NOTICE

NOTICE

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16. Abstract

TDOP/Phase II is part of a series of studies being conducted by the FRA to define the engineering options available to the railroad industry to improve the efficiency and productivity of rail freight operations. As part of this effort, experimental and analytic studies have been conducted to define the performance capabilities of the current freight car truck configurations. The results of these studies are used in arriving at quantitative characterization of performance of the standard, three-piece freight car truck under revenue service conditions. Field test data generated during TDOP/Phase I were supplemented with additional data gathered from field tests conducted during Phase II. These test data were reduced, analyzed, and interpreted in the light of physical reasoning as well as analytic simulations. Overall truck performance has been classified into four distinct and non-overlapping regimes, namely lateral stability, trackability, steady state curve negotiation, and ride quality. Performance indices, or measureable quantities typical of each performance regime, have been defined and quantified through the use of field test data and analytic simulations. Correlating the quantified performance indices within each regime with representative operating conditions such as speed, lading, and track quality, ranges of quantified performance levels have been arrived at as being characteristic of truck performance under the corresponding conditions of operation.

17. Key Words
Standard trucks, performance regimes, performance characteristics, lateral stability, trackability, ride quality, steady state curve negotiation, TDOP Phase II, freight car truck performance

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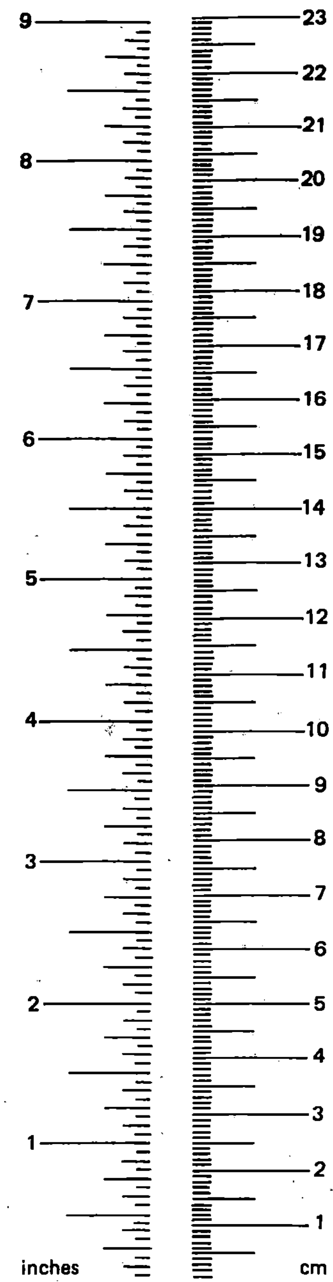
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

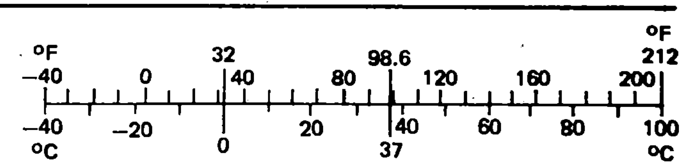
Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
	MASS (weight)			
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
	VOLUME			
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
	TEMPERATURE (exact)			
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

*1 in. = 2.54 cm (exactly). For other exact conversions and more detail tables see NBS Misc. Publ. 286. Units of Weight and Measures. Price $2.25 SD Catalog No. C13 10 286.



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	36	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



PREFACE

One of the stated objectives of TDOP Phase II is the establishment of "performance specifications" for freight car trucks. As commonly understood, the phrase "performance specification" carries connotations of precision and specificity with respect to both the qualitative aspects of performance and the level of performance in stated quantitative terms. The feasibility of attaining the stated goals for the project should be assessed in the context not only of the limited scope of the project but also the complexity of the freight car environment. The efforts within the project point to the "establishment of performance levels for freight car trucks, on the basis of interpretation of test results obtained under the project."

Proposed performance guidelines for Type I trucks (70-ton and 100-ton trucks) as represented by quantified characterizations, are presented in this report. The guidelines have been developed principally on the basis of performance test data generated during Phase I of TDOP and supplemented, wherever necessary, by Phase II data. In addition, an attempt has been made, in the interpretation of the quantitative data, to base the guidelines on substantive theoretical foundations by means of physical reasoning and comparative data studied through simple analytic and engineering models. Following this reasoning, the procedure adopted in the project is outlined briefly as:

a. A thorough evaluation of the procedures involved in the acquisition of test data;

b. Reduction of the data, followed by interpretation to ensure that the test results are consistent in terms of physical principles as well as of specific characteristics of the vehicle and test environment;

c. Comparison of test results with results obtained from comparable tests of similar vehicles, to identify and resolve any major discrepancies;

d. Simulation of test conditions by simplified models, to determine the sensitivity of the test results to variations in vehicle configuration and environment;

e. Extrapolation of performance to conditions not present in the tests, based on successful completion of step (d) which constitutes model validation;

f. Establishment of performance boundaries based primarily on the results of specific tests, but modified, and, where possible, amplified by analytic means as long as these can be validated with reference to the same test results, and corroborated by other verifiable information.

Performance of freight car trucks has been divided into four distinct regimes which, taken together as inclusive sets of conditions associated with predominant features, identify all aspects of truck behavior. These regimes are identified as lateral stability (hunting), trackability (harmonic roll, bounce, track twist, and curve entry/exit), steady state curve negotiation, and ride quality. Performance indices, which represent measurable quantities of typical performance, are defined in each of the performance regimes.

Quantitative performance characterizations for Type I trucks presented in this report are defined by ranges of performance indices in each performance regime, specifically related to operating conditions such as speed, track quality, degree of track curvature, and lading conditions. The quantified range of performance indices, developed from field test data, has been interpreted in the light of physical reasoning and tempered by comparative data studied by means of simple analytic and engineering models. Within the domain of statistical significance of the test data upon which the present characterizations are based, it is expected that tests involving similar equipment and conditions are likely to produce results comparable to the quantified ranges of performance presented in this report.

The results represent a comprehensive characterization of performance of the Type I freight car trucks, embodied in a range of quantified performance indices which are relatable to the economics of railroad operations. Therefore, it is believed that this body of results can be used by an operating railroad or by any regulatory or coordinating agency to provide the basis for a set of performance specifications for the Type I trucks which could be useful in railroad procurement and maintenance operations, as well as to provide a guideline or basis for equipment manufacturers.

While the characterization of performance of the Type I freight car truck presented here is considered to be comprehensive and representative of the equipment tested and operating conditions included, this document makes no claim with regard to representing the performance characteristics of all Type I trucks under all operating conditions; the results are to be taken within the context of the conditions governing the field tests, data collection and analysis procedures, and interpretations made therefrom.

Acknowledgment

The authors wish to express their gratitude to, and acknowledge the contributions from, several individuals and organizations associated with the Truck Design Optimization Project/Phase II as well as the industry at large. While expressing our thanks to the entire staff at Wyle Laboratories, special appreciation is due to Mr. Klaus Cappel, the Technical Director, without whose continuous guidance many of the finer points would have been lost in large volumes of data handled during the project; Messrs. David Gibson and Richard Peacock in charge of the conduct of the field test program; Messrs. Charles Bush and Robert Glaser for their valuable contributions on the data reduction software; Mr. Glenn Sheldon and his technical publications staff for a tireless effort; and Mr. Gordon Bakken, the Project Manager, for his patience and leadership through many a trying and frustrating moment during the program. We also wish to thank Mr. Arne Bang and Dr. Thomas Tsai of the Federal Railroad Administration; the Southern Pacific Transportation Company who provided the data generated during Phase I of TDOP; Mr. Garth Tennikait and the American Steel Foundries for their cooperation during the project, especially in making available to us their field test data on curving; and the many outstanding members of the industry who contributed through their participation in the TDOP Consultants Group.

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SECTION 1 - INTRODUCTION

As part of the effort under the Federal Railroad Administration-sponsored Truck Design Optimization Project (TDOP), experimental and analytical studies were conducted to define the performance capabilities of current freight car truck configurations. The experimental studies were, to a large extent, undertaken during Phase I of the project. During Phase II, the performance data base generated during Phase I was evaluated with a view to apply the results, in consultation with industry, to the development of guideline performance characterizations of Type I truck configurations. Wherever the Phase I data base was found to be inadequate for this purpose, additional field tests were conducted during Phase II.

Existing performance test data from Phase I, augmented by data from Phase II, were studied. Using a systematic methodology for the evaluation of freight car trucks, performance test data were reduced and analyzed; mathematical models were validated by comparative studies with respect to reduced test data; validated models were utilized in the simulation of additional performance data to aid in the interpretation of reduced test data; and a series of performance indices were quantified. By means of physical reasoning, the quantified performance indices were studied in relation to specific sets of operating conditions, such as speed, track quality, and lading conditions, to provide reasonable guidelines of performance that may be expected of Type I freight car trucks.

As indicated in the TDOP/Phase II Introductory Report (reference 1), four specific performance regimes were identified. These are lateral stability (hunting); trackability (harmonic roll, bounce, load equalization with respect to track twist, and curve entry/exit); steady state curve negotiation; and ride quality. Besides being distinct sets of conditions associated with predominant features of performance, this set of performance regimes is considered to identify inclusively all aspects of truck behavior.

In the performance regimes of lateral stability (hunting), trackability (harmonic roll), and ride quality, the test data considered consisted only of Phase I data. In the curve negotiation regime and trackability regime (load equalization and curve entry/exit), the Phase I data were considered inadequate. Therefore, data generated during Phase II were used in the characterization of truck performance.

Note: References can be found on the last page of this document.

SECTION 2

CLASSIFICATION OF TRUCK PERFORMANCE

With the objective of arriving at quantified characterizations of freight car truck performance, four major performance regimes have been identified. These performance regimes are:

- Lateral stability (hunting)
- Trackability (harmonic roll, bounce, load equalization with respect to track twist, and curve entry/exit)
- Steady state curve negotiation
- Ride quality

Each of these regimes is primarily defined as a set of conditions with predominant features which distinguish one from another. Measurable quantities of truck performance, defined as performance indices*, are identified within each regime. The overall characterization of truck performance consists of a range of quantified indices in each performance regime to which a truck is expected to conform under specified operating conditions. Performance data generated by means of field tests form the basis for quantification of the performance indices within each regime.

2.1 LATERAL STABILITY

The phenomenon of interest in this regime is hunting, which is a self-excited lateral and yaw oscillation of the truck and carbody occurring above a certain 'critical' speed. Wheel tread and rail contours, surface condition of the rail, design features of the truck, characteristics of the suspension system, and the mass and mass distribution of the carbody are all parameters which influence the range of the critical speed.

The performance indices identified in the lateral stability performance regime are:

- Critical speed
- Peak value of lateral acceleration (zero-to-peak)

In addition, the probability of exceedance of a given level of lateral acceleration has been studied to provide an additional useful tool to characterize performance in this regime.

2.2 TRACKABILITY

The ability of a truck to maintain a safe range of vertical load distribution on all four wheels under a range of track conditions and the responses they induce are of interest in this performance regime. Load equalization, periodic vertical rail irregularities, foundation modulus changes, as well as curve entry and exit are all conditions associated with this performance regime.

*The reader is referred to Section 5 for definitions of the performance indices.

Subclasses of the trackability performance regime and their associated performance indices are given below:

Performance Subregimes	Performance Indices
a. Harmonic roll	• Critical speed • Peak roll angle (zero-to-peak)
b. Bounce	• Critical speed • Peak vertical acceleration (zero-to-peak)
c. Load equalization	• Wheel unloading index (peak value)
d. Curve entry and exit	• Wheel unloading index (peak value)

The wheel unloading index (WUI) is identified as follows:

$$WUI = 1 - W_L/W_H/3,$$

where

W_L = force on most lightly loaded wheel

W_H = sum of forces on three most heavily loaded wheels

This definition of the wheel unloading index, in practical terms, implies that the higher the value of the index, the worse the condition of load equalization.

2.3 RIDE QUALITY

Ride quality as a performance regime refers to the acceleration environment in the carbody and is meant to encompass the capability of the truck to attenuate the excitation arising from track irregularities. The characteristic of a truck to function as a mechanical filter in isolating the carbody from the disturbances induced by the track is of primary interest in this performance regime. The principal performance index identified in this regime is:

- Transmissibility

2.4 STEADY STATE CURVE NEGOTIATION

In steady state curve negotiation, horizontal forces between the wheels and rails act to guide the truck along the curved track. For standard freight car trucks, these guiding lateral forces are likely to contribute to the resistance against the forward motion of the truck. Wheel and rail wear, and potential derailment of the vehicle, are consequences likely in this regime. The performance indices identified are:

- Lateral force on leading outer wheel
- Ratio of lateral to vertical forces (L/V ratio) on leading outer wheel
- Angle of attack of leading outer wheel

SECTION 3

QUANTIFICATION OF TRUCK PERFORMANCE

Performance test data gathered during TDOP Phase I and Phase II form the basis on which quantified performance characteristics were developed. Test data from Phase I were evaluated (reference 2) and additional performance testing on Type I trucks was conducted during Phase II to overcome inadequacies in the Phase I data.

Phase I data are used in the performance regimes of lateral stability, trackability (harmonic roll), and ride quality. Phase II data are used in the curve negotiation regime and trackability regime (load equalization and curve entry/exit).

The methodology used in arriving at the quantified performance levels includes the following chronological steps:

a. Review of the time history and selection of data to be used in each performance regime

b. Reduction of raw data to generate specific outputs which can be related to the performance indices in a given regime

c. Analysis and interpretation of reduced data:

- Quantification of performance indices directly where possible
- Use of reduced data in validating mathematical models
- Simulation of performance data by use of mathematical models
- Use of simulated data in interpolating, interpreting, and extending test data, and in quantifying performance indices

d. Characterization of performance through quantified performance indices correlated to specific operating conditions

With reference to the use of mathematical models in analytic simulations, the extent of such use has varied from one performance regime to another. In most cases, the use of models turned out not to be feasible for both technical and economic reasons. In all of the performance regimes, models were subjected to validation on the basis of test data (reference 3). In the regimes of lateral stability, harmonic roll, and curve negotiation, the results of the validation exercises proved very encouraging. In the regime of ride quality, the simple models which were used in simulations fell short of expectations and further efforts using more complex models were not attempted since ample test data were available for the task at hand and the cost of further analytic work could not be justified.

In the lateral stability regime, models were used primarily in an interpretative mode, i.e., addressing specific questions relating to the results from the test data. For example, the test data revealed a consistent tendency on the part of the box type freight cars to initiate the nosing phenomenon prior to the development of hunting. Since no apparent explanation was readily available from the test data or the operating conditions under which they were obtained, analytic simulations using models were used. Additionally, the hunting frequencies associated with results from test data were confirmed through analytic simulations. Analysis also helped address key parameters of influence in freight car hunting, especially as relating to wheel/rail contours and contact geometry.

In the case of harmonic roll, simulations were compared against field test results with good agreement. However, only a minimal amount of field test data were available for this regime and use of models to extend these sparse results was not considered judicious.

Simulations using a nonlinear curve negotiation model were compared against field test data from two different sources. The results were in good agreement with the test data under comparable conditions.

In the regime of ride quality, initial efforts centered around simple models in the vertical and roll modes which assumed that these modes were decoupled. Verification against test data proved this assumption unjustifiable. Restructuring the model to overcome the deficiencies could not be justified in light of the abundance of data available for use in the ride quality regime.

Thus, although analytic models were utilized in the simulation of truck behavior, field test data remain the primary basis of quantified performance characterizations presented in this report. The results from the test data were interpreted and correlated to appropriate operating conditions and parameters of significant influence through the use of analytic simulations, as well as through existing knowledge of the behavioral performance of freight car trucks.

Discussion of the performance quantification in each of the performance regimes follows.

3.1 LATERAL STABILITY

3.1.1 Quantification of Performance Indices

Performance test data on Type I trucks operating in combination with different carbodies on jointed rail under various speeds and lading conditions were reduced and analyzed to quantify the performance indices in this regime. The process of quantification of performance indices included the following steps:

a. Examination of time history data on the vehicle lateral motions at different locations on the carbody and the truck

b. Analysis of power spectral density functions on selected segments of data chosen on the basis of the time history data

c. Calculation of levels of lateral acceleration, evaluated for limited bandwidth

d. Extraction of peak values of lateral acceleration from the time history

e. Determination of the probability of exceedance of a given level of magnitude of lateral acceleration

The lateral acceleration data at the sill as well as the roof level of the carbody at the leading and trailing ends were statistically analyzed. The frequency range of analysis is 0 to 20 Hz. Each power spectrum in the range of 0 to 5 Hz is scanned, and the peak value of the spectrum is selected. Centered around this frequency, the root mean square (rms) acceleration is calculated for a frequency bandwidth of 1 Hz. The results of the test data analysis have been summarized for the two classes of trucks, namely, 70-ton and 100-ton trucks.

3.1.2 70-Ton Trucks

The empty flat car using worn wheels shows the earliest evidence of instability, at a speed range between 30 and 40 mph. The maximum acceleration level for this case increases sharply to 0.55 g at 40 mph, and to 1.1 g at 79 mph. No data are available on the same configuration in the loaded condition.

For the flat car using new wheels, test data for the loaded and the empty conditions exist. The analysis reveals that the loaded configuration shows no evidence of hunting and the vehicle remains stable through the entire range of operating speeds up to 79 mph. However, in the empty condition with new wheels, hunting is evidenced in the speed range between 70 and 79 mph. In general, for this configuration, the critical speed varies depending on the operating conditions, and, even more markedly so on the wheel profiles, with the predominant frequency of hunting in the range of 2.5 to 2.9 Hz.

The behavior of the mechanical refrigerator car and the box car can be placed into one category since the findings drawn from the performance test data indicate general conformity. Therefore, they will be grouped together and referred to as 'box type' cars. In the case of the empty box type cars with new wheels, hunting begins at a speed between 40 and 50 mph. In the case of the loaded box type cars, there is an indication of 'nosing,' i.e., hunting restricted to the leading end of the carbody only, initiated in the speed range between 60 and 70 mph, and continuing until 79 mph. To generalize, in this set of configurations, the critical speed increases with increasing loads; with regard to the effect of wheel profiles, the empty cars hunt at a lower frequency with worn wheels than with new wheels. While the empty box type cars hunt at a frequency slightly above 3 Hz with new wheels, those with worn wheels hunt at about 2.5 Hz. The effect of wheel profiles on amplitude response is not significant. For the empty box type cars, maximum acceleration levels range from 0.66 g at 50 mph to 1.25 g at 79 mph. From the very limited test configurations and results, which may not be typical, wheel profiles are seen to have no significant effect on performance in the case of the loaded cars. It is emphasized here that all of these observations regarding the influence of wheel profiles on truck performance are on the basis of limited test data; generalization to all freight car trucks is not intended.

3.1.3 100-Ton Trucks

The test data available in this area cover a 100-ton box car and a 100-ton covered hopper car equipped with new wheels. No data are available with respect to worn wheels. Analysis of reduced data indicates that empty cars on new wheels evidence hunting in the speed range between 70 and 79 mph. Further, the leading end of the carbody undergoes more pronounced motion as well as experiences lateral motion earlier than the trailing end (beginning in the speed range between 60 and 70 mph).

The maximum lateral acceleration level experienced is about 0.8 g in the 70 to 79 mph speed range. In contrast, the loaded car configuration remains stable through the entire range of operating speeds up to 79 mph.

3.1.4 Remarks

Narrow band rms acceleration levels calculated around the hunting frequency, peak acceleration levels, and probability of exceedances of given levels of lateral acceleration are used to quantitatively characterize performance in the lateral stability regime.

The test data considered generally covered 70-ton and 100-ton three piece trucks in combination with mechanical refrigerator cars, box cars, covered hopper cars, and flat cars (stac-pac) in the empty and loaded conditions on new and worn wheels over jointed rail track. The equipment details and test matrices are given in Appendix A.

Wheel profiles and lading conditions are the two significant parameters which influence performance in this regime. While the test data are generally considered adequate to investigate the effects of lading conditions on lateral stability performance, the data available for configurations using worn wheels are insufficient to examine in depth the effect of wheel profiles on hunting performance.

Analysis of test data in this regime indicates that (a) loaded cars have less tendency to hunt; (b) amplitudes of motion for the 100-ton configurations are much lower than those for the 70-ton configurations; and (c) the effect of wheel profiles on configurations with loaded box type cars is not seen to be significant.

A summary of the findings derived from the quantification of performance indices in the lateral stability regime using test data is given in Figure 3-1 and Table 3-1.

3.2 TRACKABILITY

3.2.1 Quantification of Performance Indices

The performance regime of trackability consists of the subregimes harmonic roll, bounce, load equalization, and curve entry/exit. The requirements of test data with regard to each of these subregimes were reviewed in light of the performance data available from Phase I tests. Phase I test data were judged to be inadequate to address the subregimes of bounce, load equalization, and curve entry/exit. As applied to the harmonic roll subregime, data from shimmed track tests were reduced and analyzed with a view to quantify performance. The shimmed track consists of twenty 39-ft rail lengths with joints uniformly staggered at approximately 19 ft, 6 inches. The rail opposite each joint in the test zone is shimmed upward to yield a cross level variation of 0.75 inches. The reduced test data are for a 70-ton mechanical refrigerator car and a 100-ton box car. The two test cars were loaded and equipped with cylindrical wheels. The height of the center of gravity of the 70-ton refrigerator car is 88 inches above the rail, and the truck center distance is 45 ft, 8 5/8 inches. The corresponding values for the 100-ton box car are 94 inches and 46 ft, 3 inches, respectively.

The process of quantifying the performance indices in the harmonic roll subregime included the following steps:

a. Examination of the time history data of the roll motions

b. Determination of peak values of roll angle and roll acceleration at different locations on the carbody

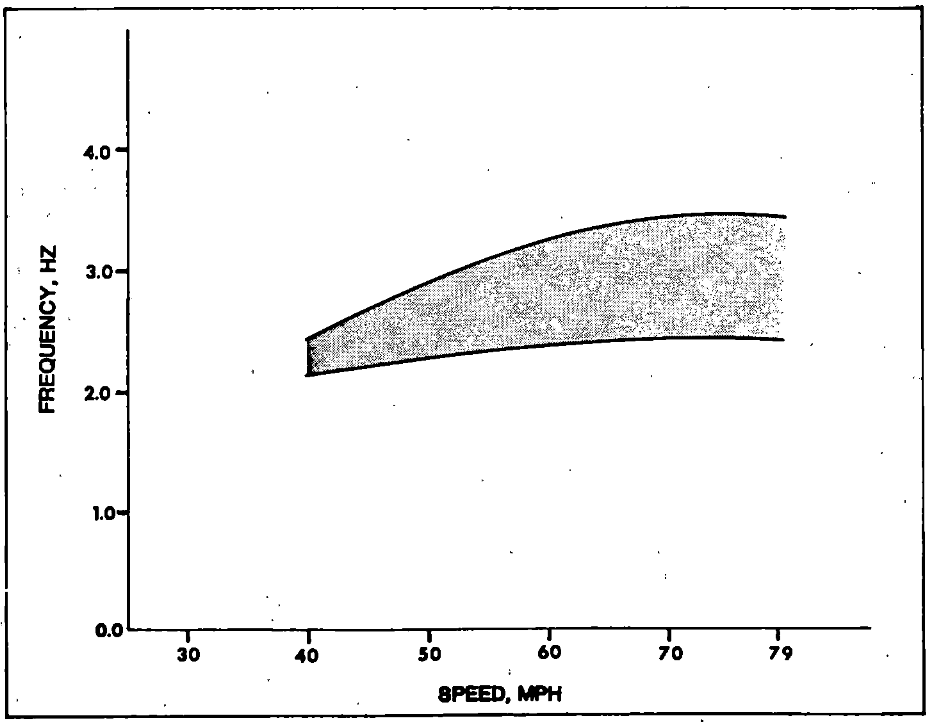


FIGURE 3-1. RANGE OF HUNTING FREQUENCIES VERSUS CRITICAL SPEED FOR TYPE I TRUCKS - ALL CONDITIONS TESTED

TABLE 3-1. SUMMARY OF RESULTS FROM ANALYSIS OF TEST DATA IN THE LATERAL STABILITY PERFORMANCE REGIME

Vehicle Configuration	Hunting* Yes/No	Critical** Speed Range mph	Hunting** Frequency Hz	RMS Lateral** Acceleration g	Peak Lateral** Acceleration g	Remarks
70-ton Trucks with Box Type Cars						
1. New Wheels/Empty	Yes	40-50	2.5-3.1	0.16-0.36	0.58-1.24	Initiation of nosing at 40-50 mph; fully developed hunting at 60-70 mph.
2. New Wheels/Loaded	No					Nosing initiated in the 60-70 mph speed range with an associated frequency range of 3.3 to 3.5 Hz, RMS acceleration of 0.27 to 0.29 g, and peak acceleration levels of 0.67 to 0.77 g. Nosing continued through the speed range up to 79 mph, the terminal speed for the test runs.
3. Worn Wheels/Empty	Yes	40-50	2.5	0.18-0.40	0.66-1.12	Leading end nosing & trailing end intermittent hunting at 40-50 mph. Both ends hunting with increasing speed.
4. Worn Wheel/Loaded	No	—	—	—	—	No evidence of hunting.
70-ton Trucks with Flat Cars						
1. New Wheels/Empty	Yes	70-79	2.8	0.11	0.59	Fully developed hunting at 70-79 mph.
2. New Wheels/Loaded	No	—	—	—	—	No evidence of hunting.
3. Worn Wheels/Empty	Yes	30-40	2.2-2.9	0.12-0.30	0.55-1.10	Fully developed hunting at 30-40 mph.
4. Worn Wheels/Loaded	—	—	—	—	—	No data available.
100-ton Trucks with Box Type Cars						
1. New Wheels/Empty	Yes	70-79	2.7	0.10-0.25	0.73-0.83	Fully developed hunting.
2. New Wheels/Loaded	No	—	—	—	—	No evidence of hunting.
3. Worn Wheels/Empty	—	—	—	—	—	No data available.
4. Worn Wheels/Loaded	—	—	—	—	—	No data available.

*"Hunting" denotes full-body hunting as differentiated from nosing.

**Includes nosing and full body hunting.

Results of the reduction and analysis of the test data on the shimmed track showed that the loaded refrigerator car has the ability to extract energy from the track input excitations, and the carbody reaches a state in which the rocking car exceeds its capability to dampen or absorb rolling motion. The peak roll angle at the leading end of the carbody is 2.9° for the refrigerator car at about 14 mph. The peak roll angle at the leading end of the carbody for the 100-ton box car is 2.4° at about 14.5 mph. The results also showed that the data are quite nonlinear, and contain higher frequency components, particularly when acceleration responses are considered. The carbody is rolling about the lower center; in other words, the mode excited is the lower center roll.

Data from the Phase II yard tests were reduced and analyzed with a view to quantify the truck performance in the load equalization subregime. The truck tested was a 100-ton capacity Type I truck running under a 100-ton open hopper car in both loaded and empty conditions. The yard track, which is classified as class 1 track, consists of two curves. The degrees of curvature are 16° (left-hand curve) and 15.75° (right-hand curve), and the corresponding superelevations are -0.26 inches and -0.3 inches, respectively. The speed limit is 10 mph. Appendix B contains a more detailed description of the conditions and procedures of this testing.

Results of the reduction and analysis of test data indicate that the dynamic components of the lateral forces and L/V ratios are high. The wheel unloading index for the loaded car has a mean value of 0.138 and a standard deviation of 0.065 on the 16° curve. The corresponding values for the 15.75° curve are 0.208 and 0.108, respectively. The mean values of the wheel unloading index for the empty car are 0.409 and 0.264 for the 16° curve and 15.75° curve, respectively, with standard deviations of 0.083 and 0.73, respectively. It is noted that the wheel unloading index is substantially higher for the unloaded car than that for the loaded car. This is mainly due to the friction snubber in the suspension which permits little motion between the truck components for the empty car. It may be noted, however, that the field test data considered here included only the constant friction snubber trucks.

For the curve entry/exit subregime, Phase II data were used. A 100-ton capacity Type I truck was tested under an open hopper car. Both loaded and empty cars were tested. Tests were made at the balance speed for track curvature ranging from 2.5° to 6.2°. Repeat tests were then made below the balance speed and above the balance speed. The "axle bending" technique was used to determine the vertical and lateral forces at the wheel/rail interface (see Appendix B for more details).

Analysis of the data indicates that, in general, the peak value of the wheel unloading index is increasing with increasing degree of curvature. The effect of speed on this index is not clear (i.e., does not have a constant pattern) from the results. This might be due in part to the dependence of this index on just one point extracted from the time history, and in part to the dependence of the car response on the track memory of the truck. Rail contamination and vehicle nonlinearities may also lead to this phenomenon. However, it has been noticed that the empty cars experience a higher wheel unloading index than the loaded cars on all curves tested.

3.3 RIDE QUALITY

3.3.1 Quantification of Performance Indices

The performance characterization of interest in this regime is the capability of the truck to attenuate track-induced vibrations. Extreme performance phenomena, such as resonance and other unstable conditions, are excluded from consideration in this regime since they have been accounted for within the other identified performance regimes. Performance test data from TDOP Phase I have been examined, reduced, and analyzed to quantify the performance indices selected for this regime (see Appendix A). The quantification on the basis of test data followed the sequence outlined below.

a. Examination of time history data on the vehicle vertical and roll motions at different locations on the carbody and truck

b. Selection of data to be analyzed, excluding resonant and unstable phenomena from consideration

c. Analysis of power spectral density functions on selected data

d. Calculation of rms levels of vertical and roll accelerations, evaluated for selected bandwidths

e. Calculation of transmissibility in both the vertical and roll directions, relating carbody response to track input as obtained from track geometry data, i.e., profile and cross level power spectral densities, respectively

Test data obtained over medium-speed (up to 45 mph) jointed track as well as high-speed (up to 79 mph) jointed track were reduced, analyzed, and compared over corresponding frequency spectra, wherever possible. The quantified levels from the high-speed track data were checked for continuity with the levels obtained from the medium-speed track data; on the basis of satisfactory comparison, the quantified levels in this report represent results obtained from the high-speed track test data.

It may be noticed that certain configurations, notably the 70-ton trucks with flat cars, and certain speed ranges are missing from the presentation of quantified levels. These omissions are by design and can be traced to resonant or unstable phenomena which are covered within one or more of the other performance regimes treated.

One of the more serious problems encountered during the calculation of the response/excitation transfer functions relates to the lack of correlation between the available track geometry data and the performance test data. This necessitated some manipulations during the processing of data to align the track geometry data with the performance test data. Although the alignment has been accomplished, the precision of this manipulation leaves much to be desired. In fact, the alignment can be justified only at the initial and final points of the segments of data chosen for analysis; intermediate to these points the track geometry data and the test data are not expected to show the desirable degree of alignment. Consequently, the defined performance index for this regime, namely, transmissibility thus generated, has associated with it a degree of imprecision. For this reason, it was felt desirable to provide a supplementary index: the rms acceleration levels calculated for the wide band frequency spectrum of 0 to 20 Hz in both the vertical and the roll directions. This index should provide a reasonable indication of the energy content in the responses over the entire frequency spectrum for which the test data are considered valid.

The results obtained from the reduction and analysis of the test data have been summarized for 70-ton and 100-ton trucks.

3.3.2 70-Ton Trucks

In general, loaded box type cars on 70-ton trucks indicate increasing rms values of vertical acceleration with increasing speed and a tendency to resonate in the vertical plane at about 50 mph. In the case of empty box type cars, the levels of vertical acceleration response are higher as compared to the response of the loaded cars, the implication being that loaded cars obtain better ride quality than empty ones. However, in one case the loaded box car indicated higher levels of vertical acceleration above 40 mph as compared to those of the empty car. This case is considered the exception rather than the rule, and one possible explanation for this phenomenon is the coincidence of the natural frequencies of the carbody with those of the excitations from the jointed track, as well as the coincidence of the truck center spacing with the spacing of rail joints.

In the case of the flat cars, only the loaded configuration has been analyzed since the empty configuration was extensively covered by the lateral stability regime by virtue of indications of hunting. The flexural modes of vibration of the car are believed to be significant contributors to the car response.

In the roll mode, the amplitude response of the loaded box type cars is lower than that of the empty cars and the principal reason is considered to be the lower level of friction damping in this mode. Analysis of data on the loaded flat car indicates that the contribution from the torsional mode of vibration is significant. At about 40 mph the response peaks, with the leading end undergoing higher amplitude response than the trailing end.

3.3.3 100-Ton Trucks

Once again, the loaded box type cars on the 100-ton trucks exhibit better vertical ride quality characteristics as compared to those of the empty cars. The difference in the responses between the empty and the loaded cases is attributable, at least in part, to the higher natural frequencies of the empty cars and the effect of friction snubbing.

Among the box type cars in the roll mode, the hopper cars indicate lower levels of amplitude response as compared to those of the box cars. The trailing end of the carbody undergoes higher levels of roll acceleration than the leading end.

3.3.4 Remarks

The role of train speed on the ride quality response of the carbody is clearly discernible; as the train speed is increased more track excitation is transferred to the car, resulting in higher amplitude response. Rail joint frequencies and the location of peaks in the power spectra are strongly related, indicating that the input excitation to the car arises mainly from the periodic rail joint spacing, with smaller contributions from the stochastic excitation from the random track irregularities. In general, response consists of rigid body modes and flexural modes; it is seen that the flexural modes of vibration of the carbody play a major role in the dynamic response of the freight cars. Lading conditions affect the ride quality in both the vertical and the roll modes, with the empty cars having higher amplitude response as compared to the loaded cars. In the roll mode, for loaded cars, 70-ton vehicles exhibit more desirable dynamic characteristics than the 100-ton vehicular combinations. In the vertical mode, the 100-ton trucks are more effective in attenuating the track excitations transmitted to the carbody than the 70-ton trucks.

3.4 STEADY STATE CURVE NEGOTIATION

Results presented in this section are derived from the Phase II field test data; testing procedures are described more fully in Appendix B.

The test data used have been gathered from tests on track consisting of both left- and right- hand curves ranging from 2.5° to 6.2°. The test vehicle configuration consisted of a 100-ton truck in combination with a loaded and empty 100-ton open hopper car, and AAR standard, 1/20 taper wheel profiles in the new condition. Test speeds were at balance, below balance, and above balance, and a run at balance speed in the reverse direction was conducted. The technique for measuring the forces at the wheel/rail interface is the "axle bending" technique. The axle bending technique makes use of the data gathered from strain gaging the axle for measuring axle bending and from the bearing adapters for measuring vertical forces. Data covering the track segments of constant curvature were used in quantifying performance in the steady state curve negotiation regime. In quantifying the lateral forces and L/V ratios on the leading outer wheel, the algebriac average and the standard deviation were computed from the test data.

The results of the reduced data show that the lateral forces and L/V ratios are increasing with increasing degree of curvature and they tend to have the same characteristics. For the moderate curves of 2.5° and 3°, the lateral forces on the leading outer wheel of the loaded car are comparable. However, these lateral forces show substantial increase in magnitude as the degree of curvature increases, reaching an approximate value of 14,000 lb at the 6.2° curve. The ratio of the dynamic lateral forces to the steady state lateral forces are lower for higher degree of curvature. The values of the lateral forces and L/V ratios in both the forward and reverse directions are comparable. By comparing the results for the loaded and empty cars, the following may be stated:

a. During curve negotiation, the L/V ratios are more critical for the loaded cars than for the empty cars. This conclusion is based on mean values of L/V ratios at balance speed without considering the associated time duration.

b. The rate of increase of the lateral forces and L/V ratios on the leading outer wheel with increasing degree of curvature is higher for the loaded cars than for the empty cars.

c. The ratio of the dynamic components of the lateral forces and L/V ratios to the steady state components are higher for the empty cars than for the loaded cars. This indicates that the dynamic effect of both curve entry and track irregularities is much higher for empty cars than for loaded cars.

3.4.1 Remarks

The test data enabled quantification of the effect of curvature and superelevation deficiency on lateral and vertical forces, as well as L/V ratios for both the loaded and empty cars.

On sharp curves (5^{o} - 6.2^{o}), due to the fact that the lateral forces required to guide the truck through the curve are larger than the primary guidance achieved by the tangential forces at the wheel treads, the wheel flanges come into action. In other words, the wheel flanges perform the primary role in curve negotiability in curves of small radii.

The leading outer wheel, which is the main guiding wheel when entering a curve, experiences and maintains large lateral forces and L/V ratios above balance speed than does any other wheel. Above balance speed, the leading outer wheel also seems to be more sensitive to track curvature.

Below balance speed, the trailing axle of the truck runs closer to the low rail of the curve, causing flange contact of the trailing inner wheel on sharp curves, and consequently experiences the highest forces.

Below and at balance speed, the trailing axle carries the greater part of the net lateral forces for all curves considered.

Past history of the truck (the initial configuration of the truck as it enters the curve) seems to have an influence on the level of the steady state lateral forces.

SECTION 4

CORRELATION OF ENGINEERING FACTORS

In the process of developing performance characterizations of new freight car trucks, the quantification of performance as described in Section 3 forms only a first step. Equipment deterioration with service and the effect of service-worn components on performance need to be considered in the translation of the quantified performance levels into meaningful and realistic performance characterizations of new equipment. In turn, this step demands that (a) sufficient information be available with respect to wear and deterioration patterns of components with service over the life cycle of the trucks, and (b) a reasonable body of knowledge be available with respect to the performance levels of the trucks with varying degrees of component wear and deterioration.

The Wear Data Collection Program being undertaken in Phase II of the Truck Design Optimization Project is geared, in part, to provide some of the information concerning wear and deterioration patterns of components with service. Analysis of maintenance data from operating railroads complements this information. To address wear and deterioration of components on performance, Series 1 and Series 2 of the tests conducted during Phase I examined the effects on performance due to variations in gib and side bearing clearances, snubber wear, and wheel wear. Effects due to some spring changes were also studied. Although this effort in Phase I was by no means comprehensive, it was a meaningful attempt and sheds some light on the problem.

The Wear Data Collection Program is still in its early stages and only three cycles of measurements, covering approximately 100,000 accumulated miles, have been completed to date. Thus, no trends can be determined from this program at this stage. Evaluation of published results from Phase I data indicates that, with the exception of wheel profiles, variations in the parameters studied have no significant effect on truck performance. Therefore, attention has been focused on the effect of worn wheel profiles on performance. Existing test data are too sparse to permit any in-depth examination of the effect of wheel profiles on truck performance.

SECTION 5 - PERFORMANCE CHARACTERIZATION

The intent of the guidelines presented in this section is to identify Type I freight car truck performance levels which can be correlated to savings associated with reduced maintenance, longer equipment life, and other tangible benefits in terms of railroad operations. The development of these guidelines has kept in perspective common industry practices and the Association of American Railroads (AAR) requirements for interchange service and regulatory safety requirements.

The quantified levels of performance given under each of the performance regimes represent the results of analysis and interpretation of quantified test data.

5.1 LATERAL STABILITY

Characteristic performance levels in the regime of lateral stability are given in terms of rms lateral acceleration and peak lateral acceleration (zero-to-peak). Figures 5-1 through 5-4 show overall envelopes of performance levels for all Type I trucks for inclusive sets of conditions identified, and are provided for convenient reference. Characteristic levels of performance, under the appropriately identified set of operating conditions, shall be limited by the upper bound of the bands of performance shown in the characterization charts given in Figures 5-5 through 5-14. Lower bounds of performance, provided wherever possible, are shown for reference indicating the most optimistic levels of performance that may be expected under the tested operating conditions.

The performance index, critical speed, used in the lateral stability regime is identified through the use of the root mean square (rms) lateral acceleration levels. The rms value is calculated for unit (1 Hz) bandwidth around the characteristic frequency of hunting; it is determined from the power spectral density analysis of the lateral acceleration response data measured at the carbody sill level or at the truck wheelsets.

The other measure used to characterize performance levels in the lateral stability regime is peak lateral acceleration, defined as the absolute peak value (zero-to-peak) of lateral acceleration; it is determined from an examination of the time history of the response data at the sill level of the carbody or the truck wheelsets.

Figures 5-15 through 5-19 provide a more detailed statistical view of the expected probability associated with the different levels of the magnitude of lateral acceleration for the vehicular configurations identified therein under the given operating conditions.

5.2 TRACKABILITY

The performance regime of trackability consists of the subregimes harmonic roll, bounce, load equalization with respect to track twist, and curve entry/exit.

Characterization of performance in the haromonic roll subregime is provided by means of quantified performance indices identified for the regime. These indices are:

a. Critical speed

b. Peak roll angle at the leading end

Besides these two indices, the roll accelerations at the two ends of the carbody are given. The peak roll angle and roll accelerations, as presented here, are defined as the zero-to-peak value extracted from the time history of the response data.

Figures 5-20 through 5-23 represent the quantified characterization in the harmonic roll subregime for both 70-ton and 100-ton box type cars.

Characterization of performance in the load equalization subregime is provided by means of the wheel unloading index (WUI), which is the zero-to-peak value extracted from the time history. Performance characteristics are shown in Tables 5-1 and 5-2.

Characterization of performance in the curve entry/exit subregime also is provided by means of the wheel unloading index. These characterizations are presented in Figures 5-24 through 5-28.

5.3 RIDE QUALITY

Characterization of performance in the ride quality regime is provided by means of quantified performance indices identified for the regime. These indices are: (a) transmissibility, and (b) rms response over the wide band spectrum.

Transmissibility, as presented here, is identified as the ratio of the rms value calculated from the response power spectral density within a specified frequency bandwidth to the rms value calculated from the track input power spectral density over a corresponding frequency bandwidth.

Transmissibility has been quantified in both the vertical and the roll directions. Vertical acceleration response at the sill level and roll acceleration response at either end of the carbody in the frequency bandwidths of 0 to 4 and 4 to 10 Hz have been considered. The corresponding input consisted of power spectral densities of track profile in respect to vertical response and track cross level in respect to roll response in the same frequency bandwidths.

The rms values of the response power spectral densities for both the vertical and the roll accelerations were computed over the frequency range of 0-20 Hz as an additional performance index and plotted as a function of speed.

Figures 5-29 through 5-37 represent the quantified characterization of the vertical motion, and Figures 5-38 through 5-46 represent the characterization of the roll motion.

5.4 STEADY STATE CURVE NEGOTIATION

Quantitative performance levels in steady state curve negotiation for 100-ton Type I trucks with box type cars equipped with AAR standard 1/20 taper, new wheels are shown in Figures 5-47 through 5-58 in both the loaded and empty conditions. Figures 5-47 through 5-56 show performance levels in terms of lateral forces and L/V ratios on the leading outer wheel as a function of speed; Figures 5-57 and 5-58 show performance levels in terms of lateral forces and L/V ratios on the leading outer wheel as a function of the degree of track curvature near balance speed.

Angle of attack measurements and results are given in Appendix C.

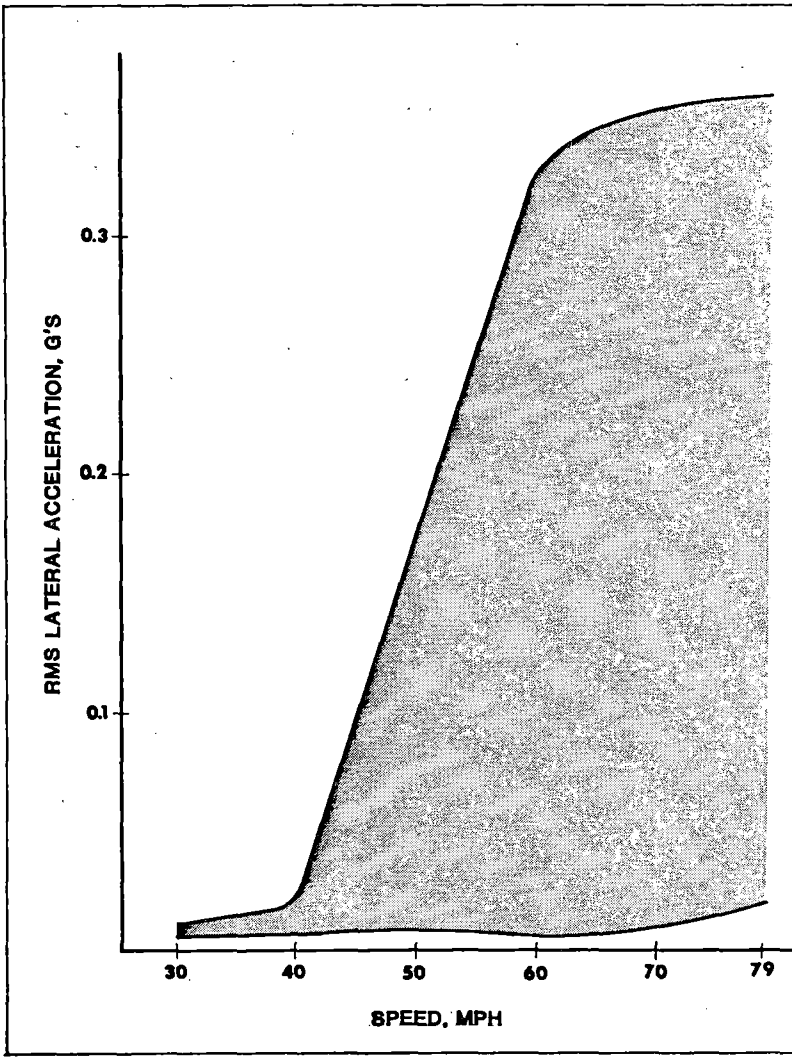


FIGURE 5-1. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - BOX TYPE CARS, ALL CONDITIONS TESTED

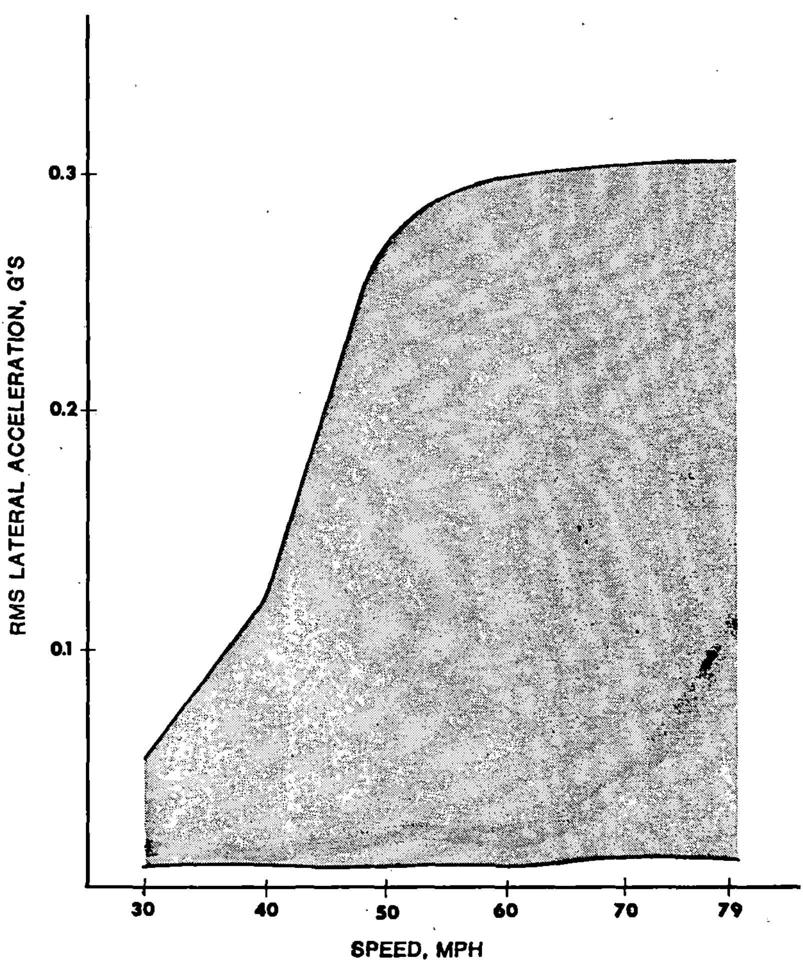


FIGURE 5-2. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - FLAT CARS, ALL CONDITIONS TESTED

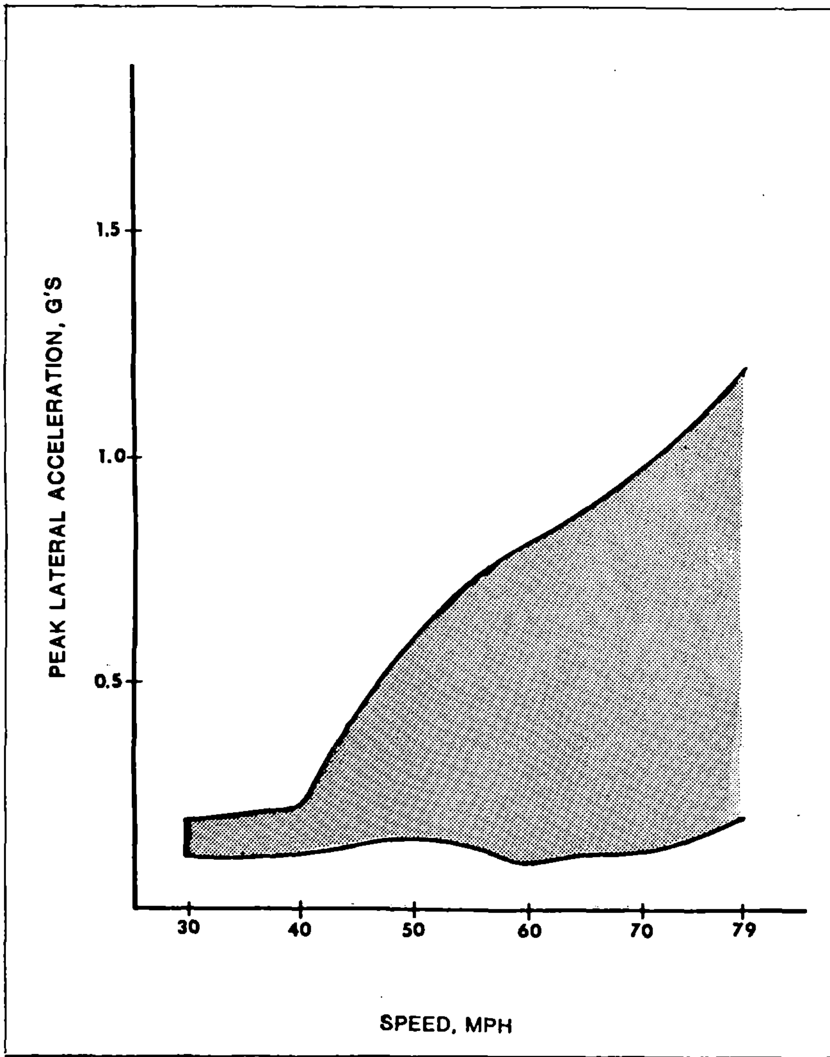


FIGURE 5-3. PEAK LATERAL ACCELERATION VERSUS SPEED - BOX TYPE CARS, ALL CONDITIONS TESTED

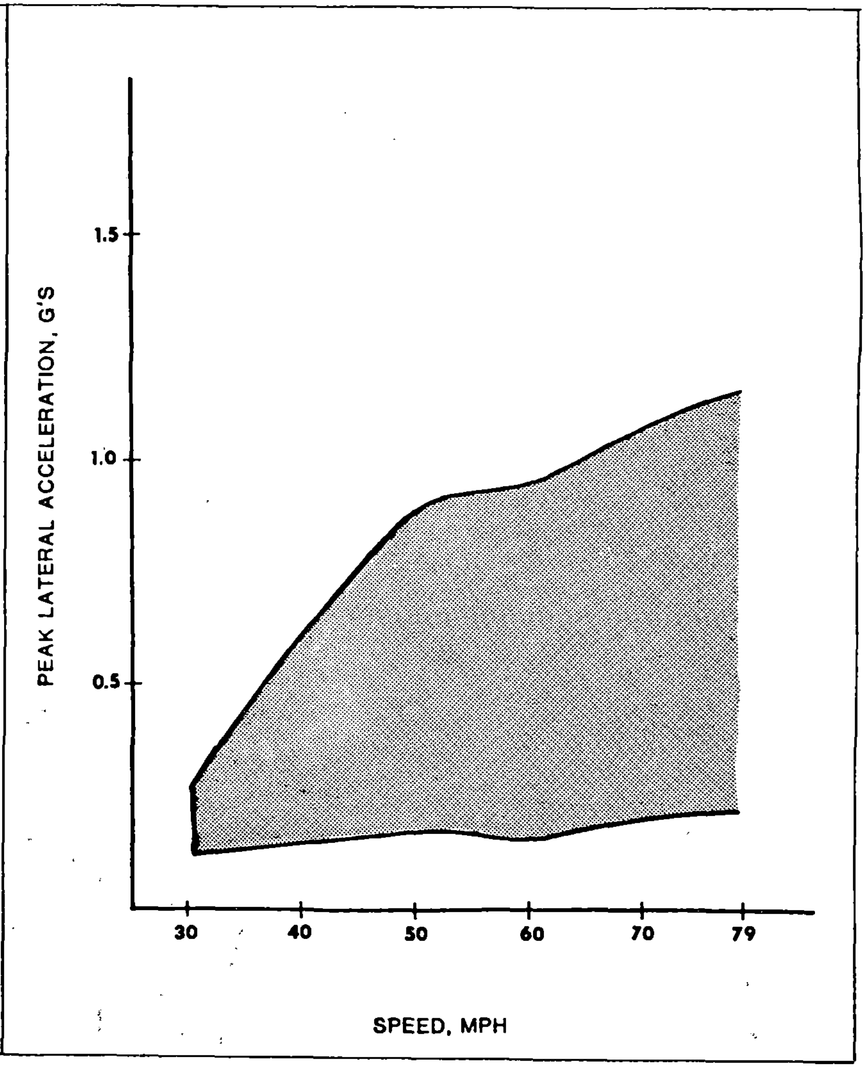


FIGURE 5-4. PEAK LATERAL ACCELERATION VERSUS SPEED - FLAT CARS, ALL CONDITIONS TESTED

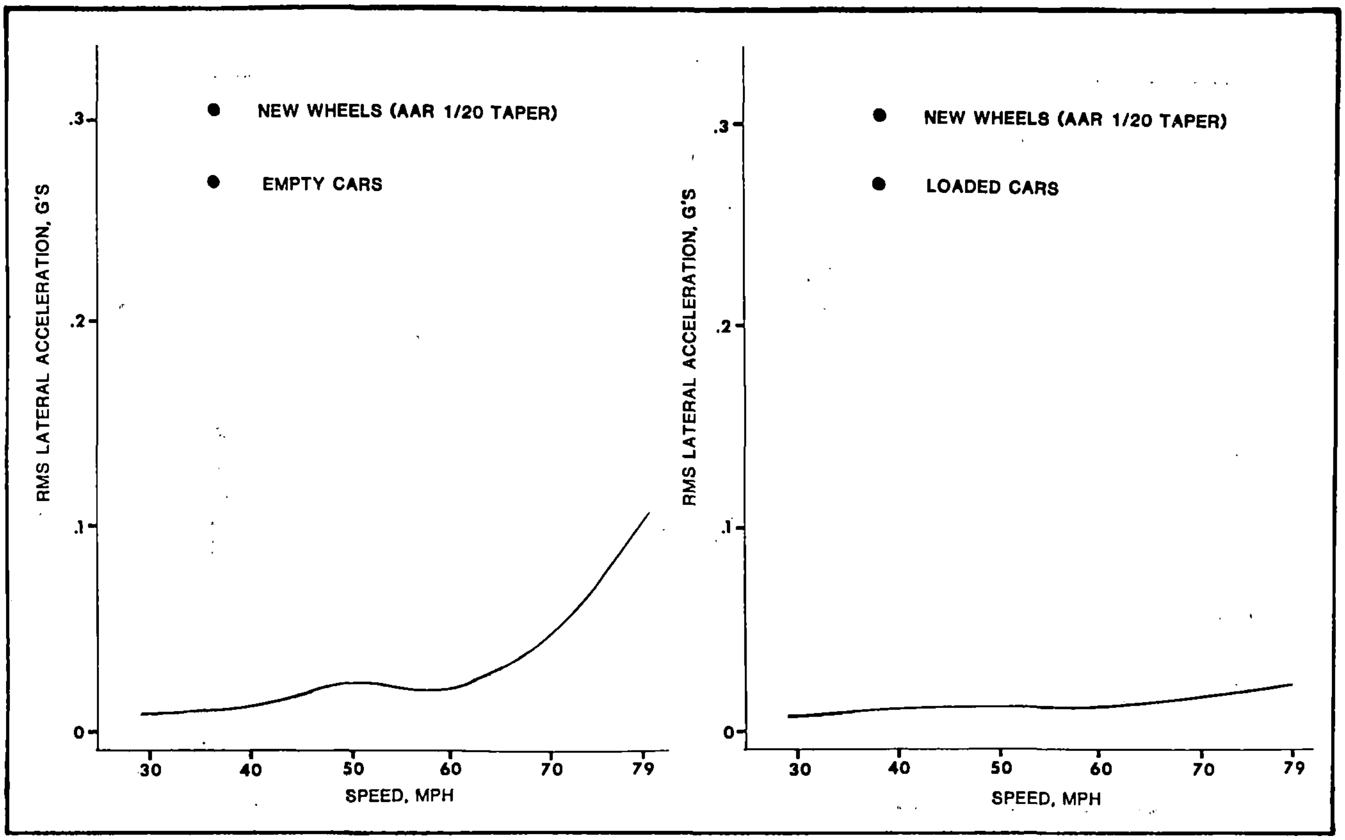


FIGURE 5-5. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, NEW WHEELS

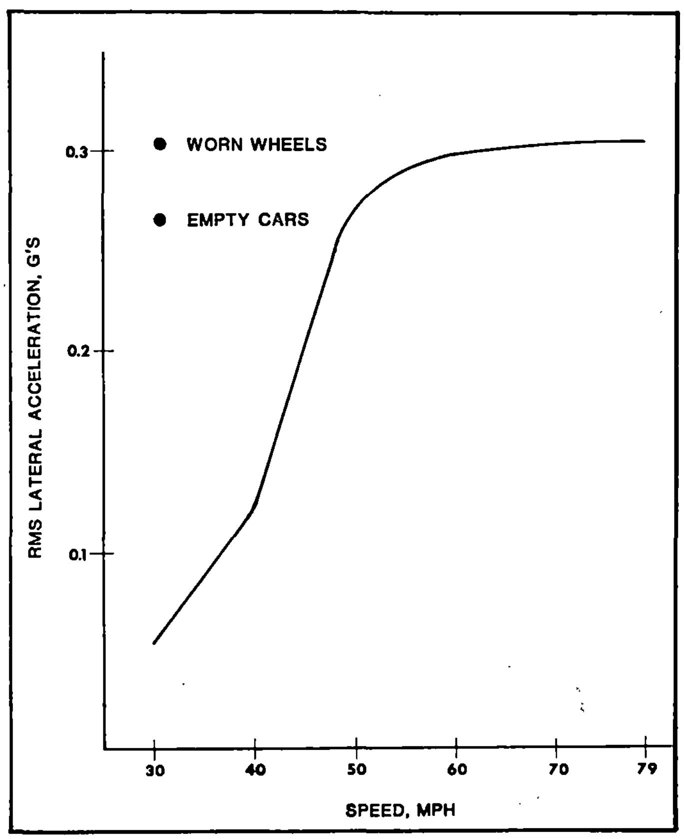


FIGURE 5-6. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, WORN WHEELS

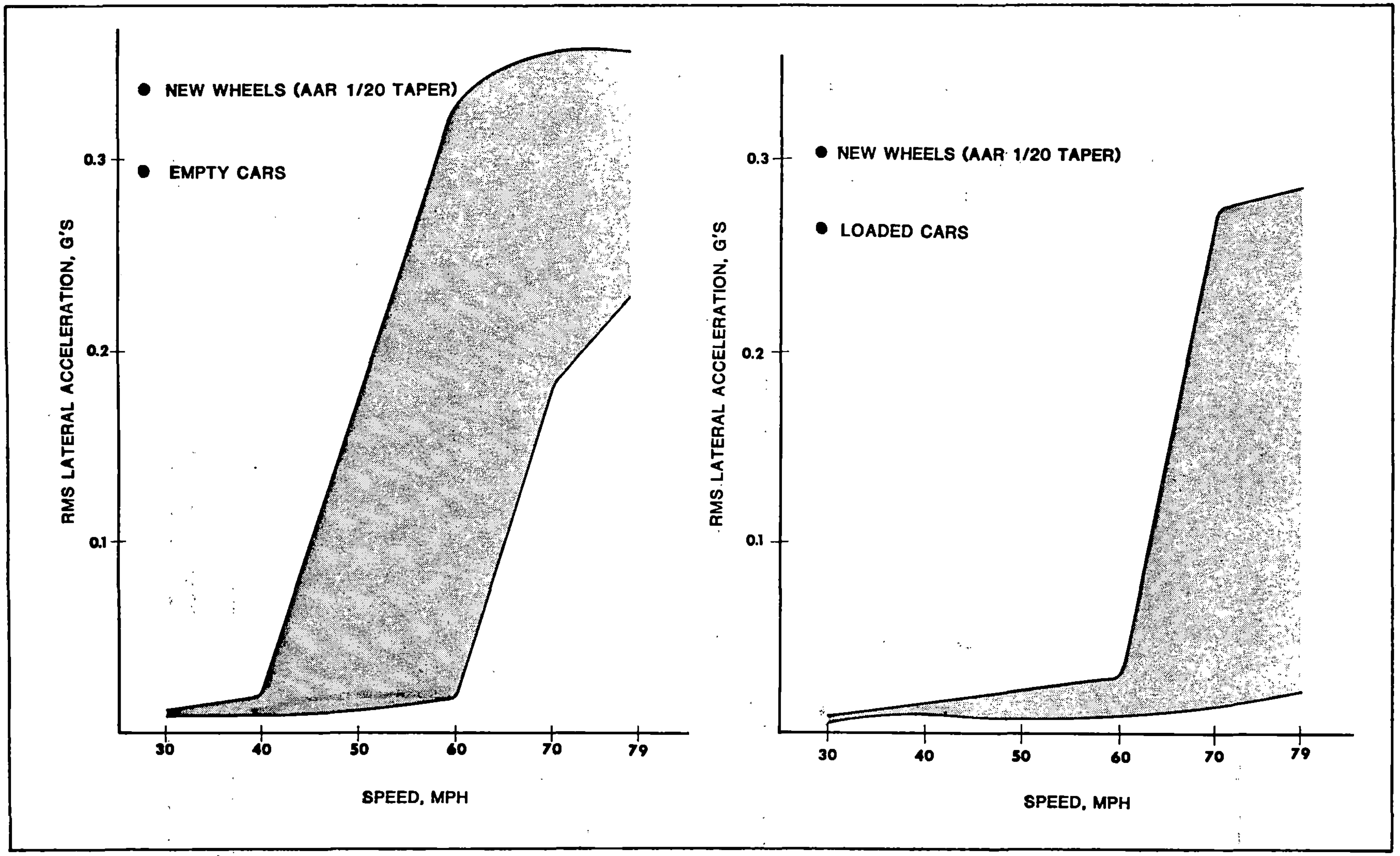


FIGURE 5-7. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, NEW WHEELS

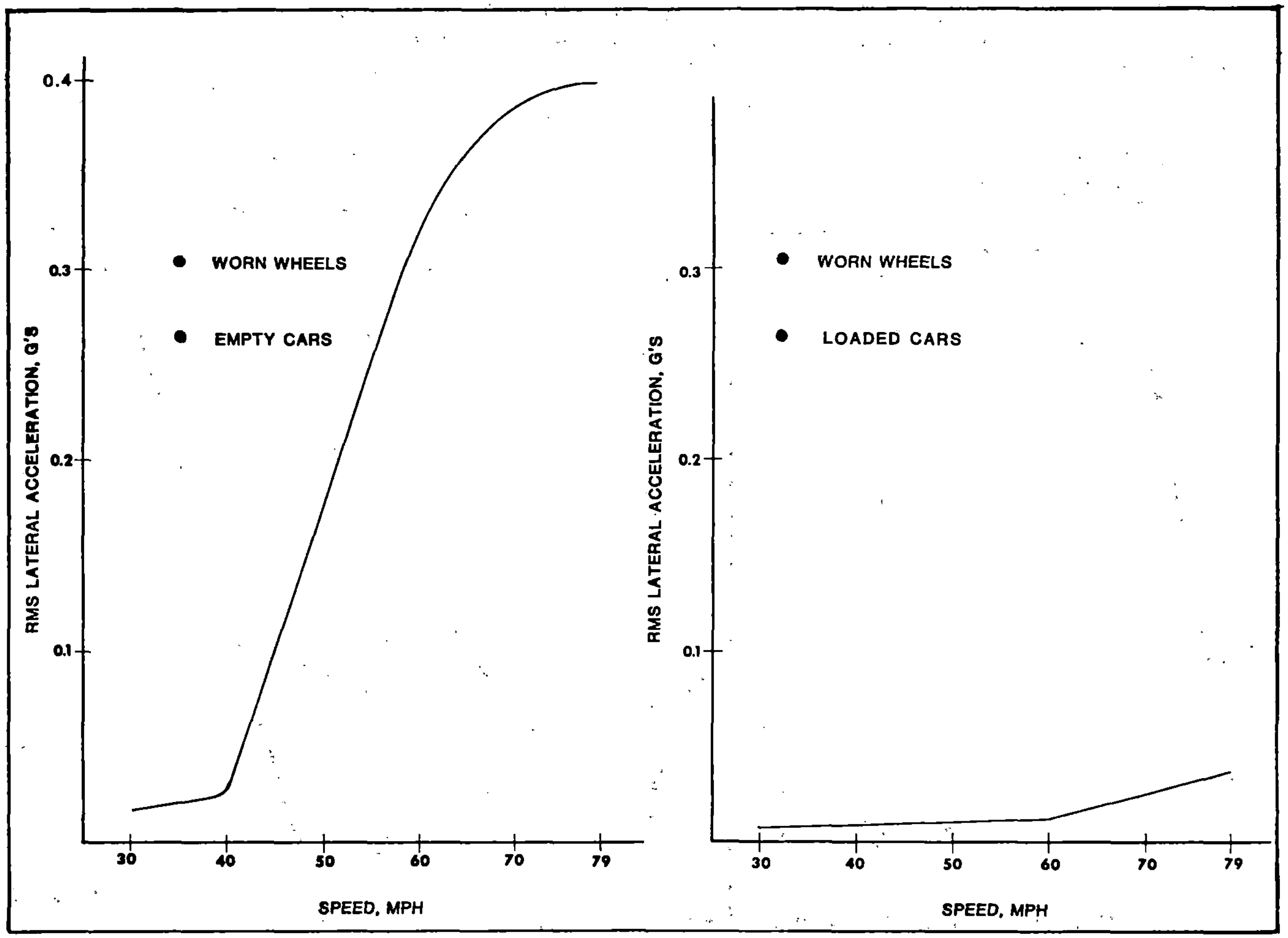


FIGURE 5-8. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, WORN WHEELS

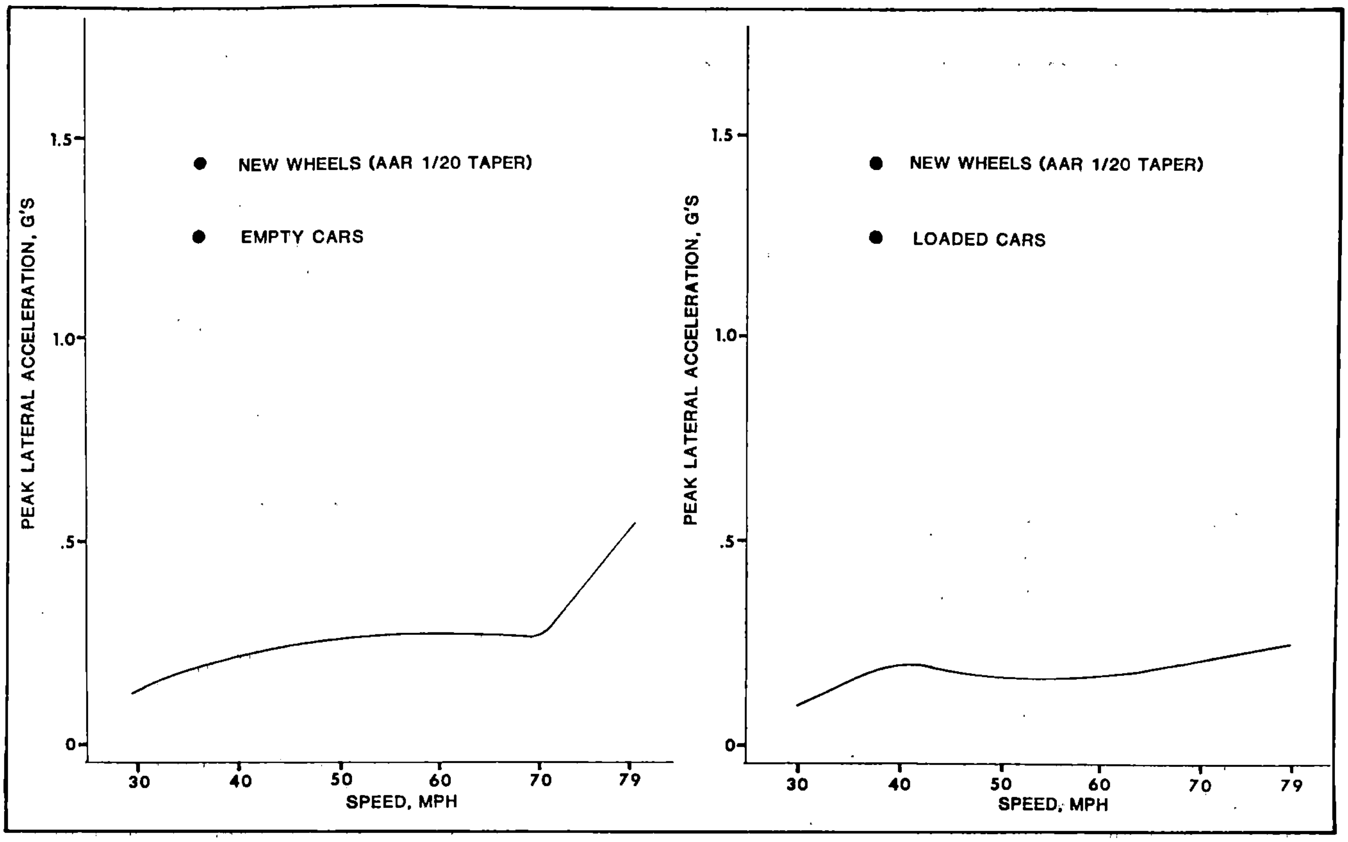


FIGURE 5-9. PEAK LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, NEW WHEELS

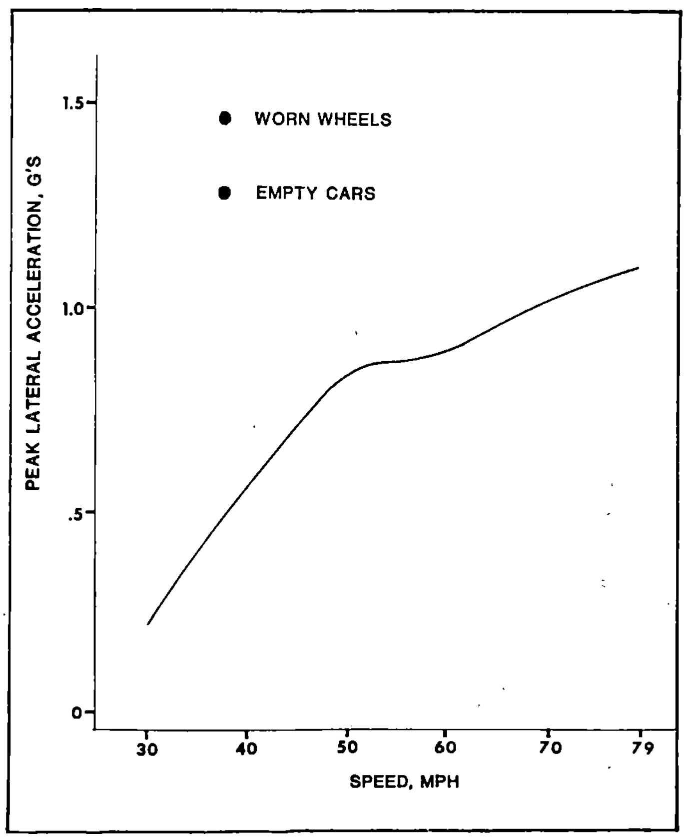


FIGURE 5-10. PEAK LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, WORN WHEELS

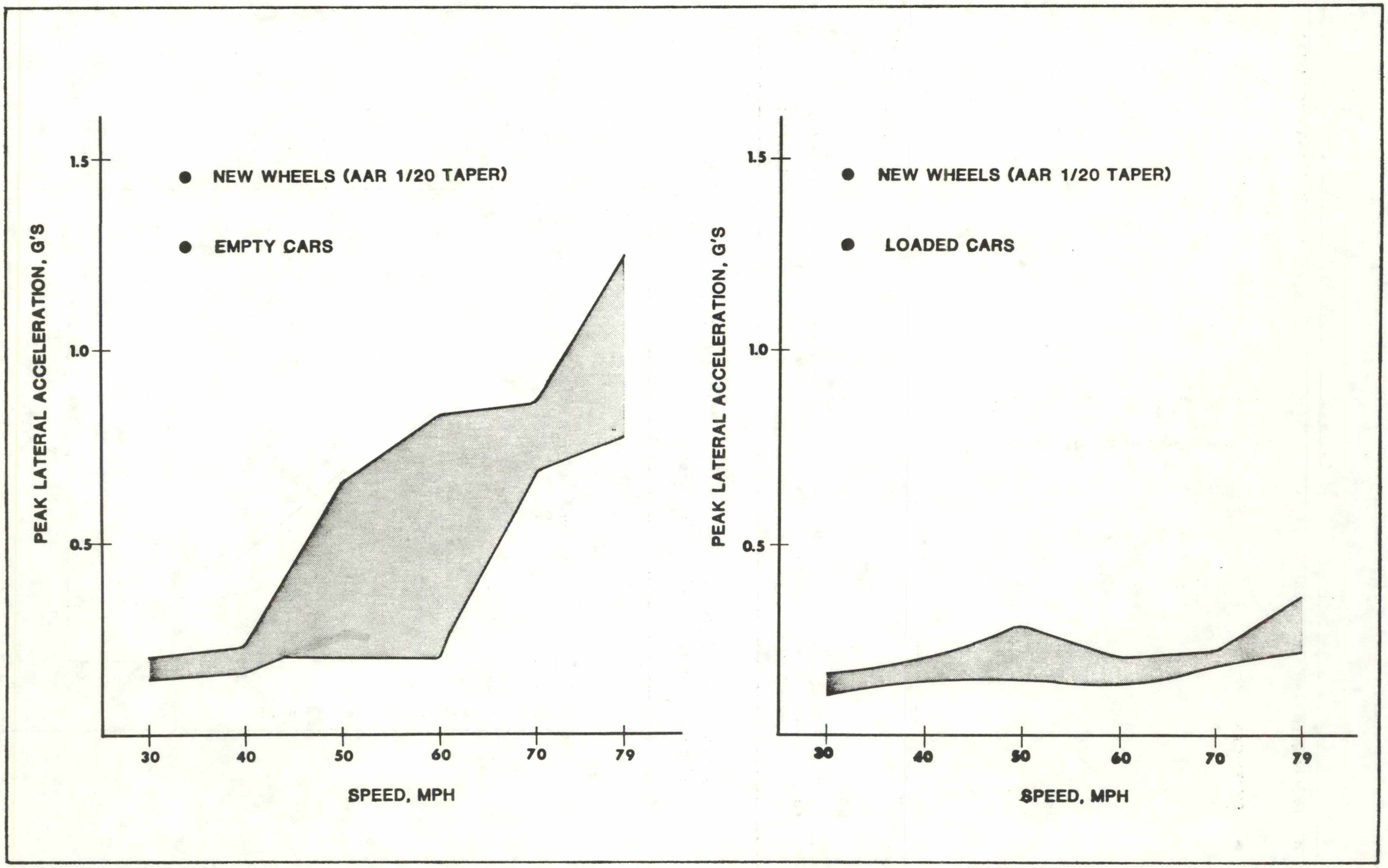


FIGURE 5-11. PEAK LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, NEW WHEELS

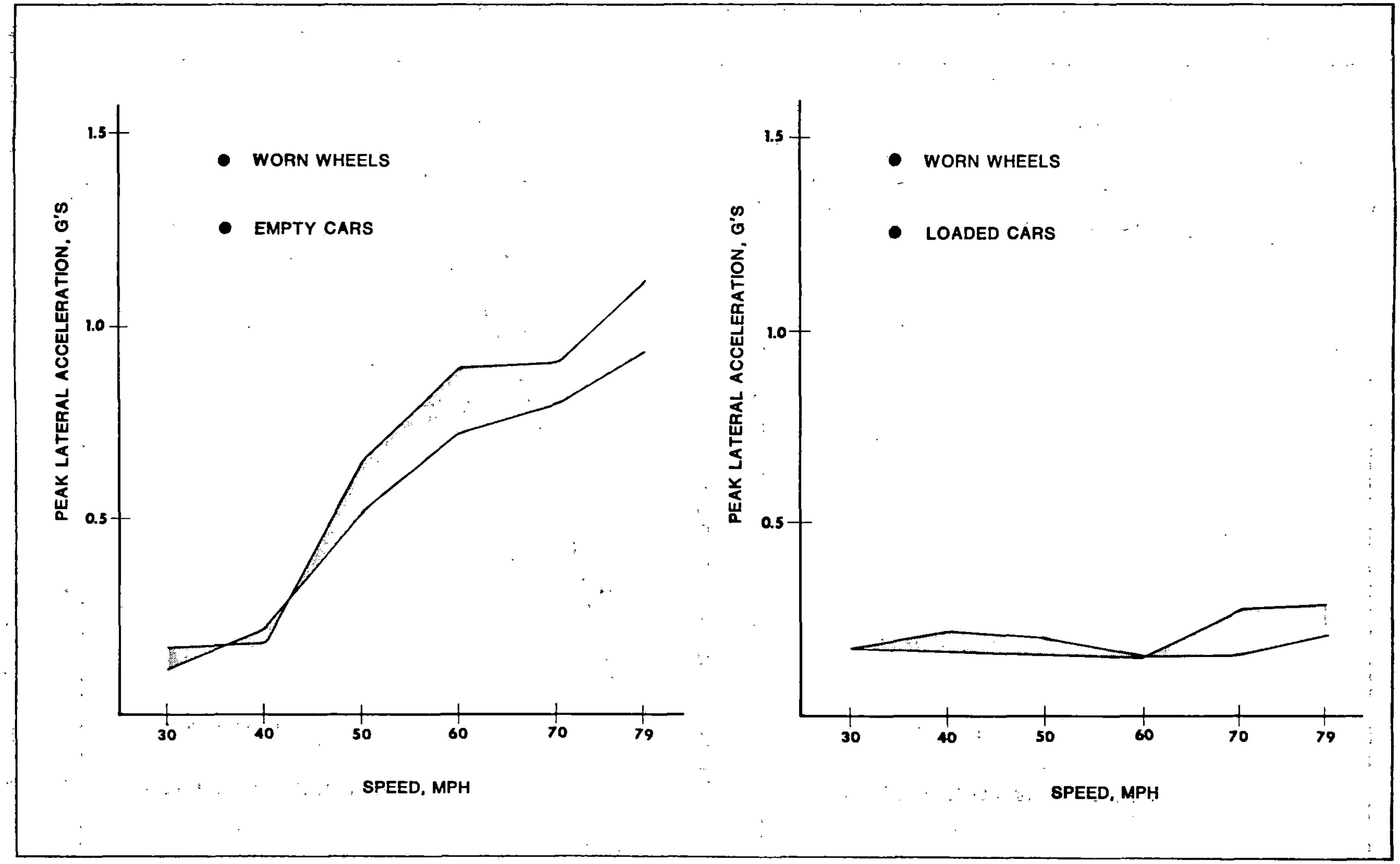


FIGURE 5-12. PEAK LATERAL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, WORN WHEELS

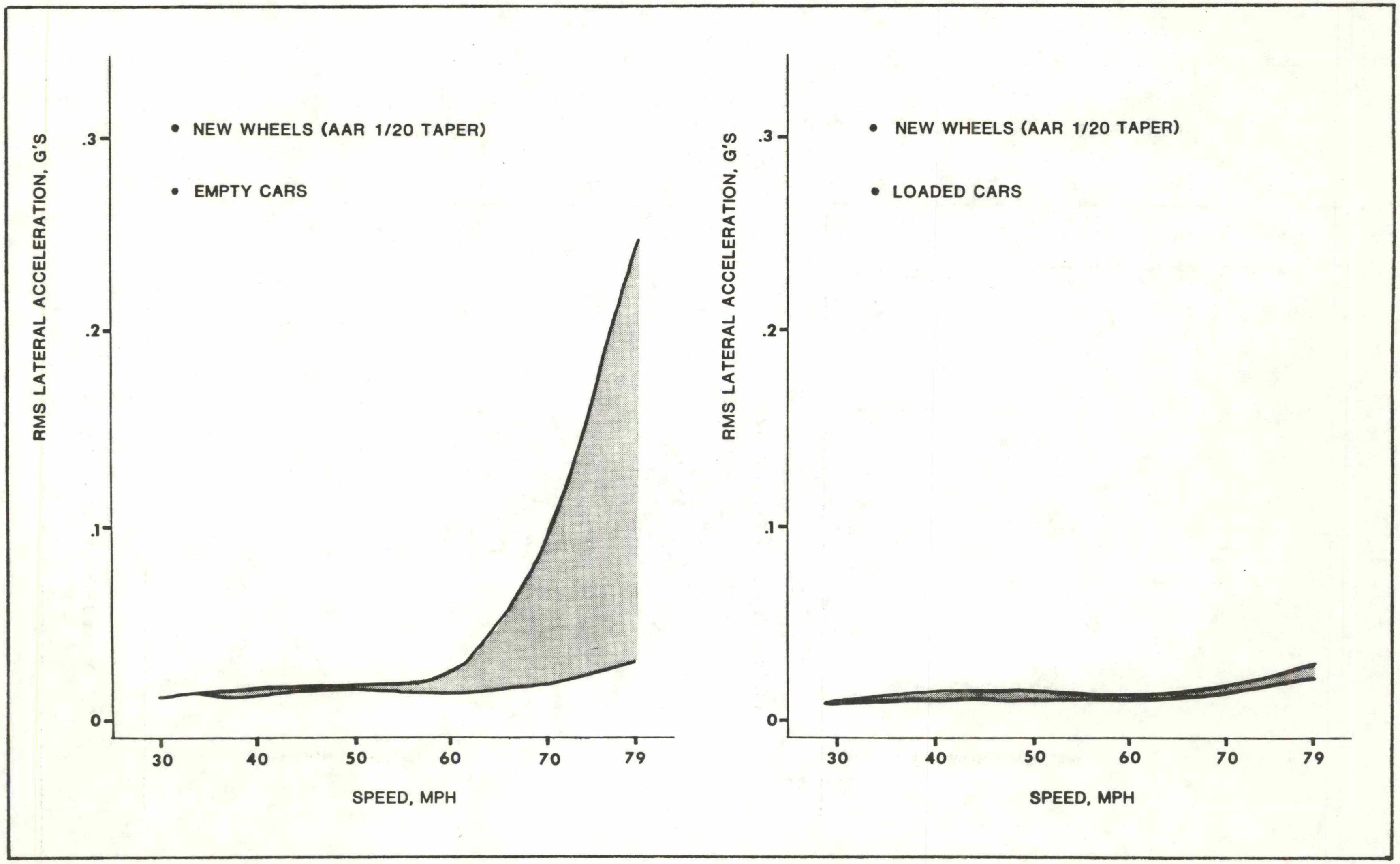


FIGURE 5-13. ROOT MEAN SQUARE LATERAL ACCELERATION VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, NEW WHEELS

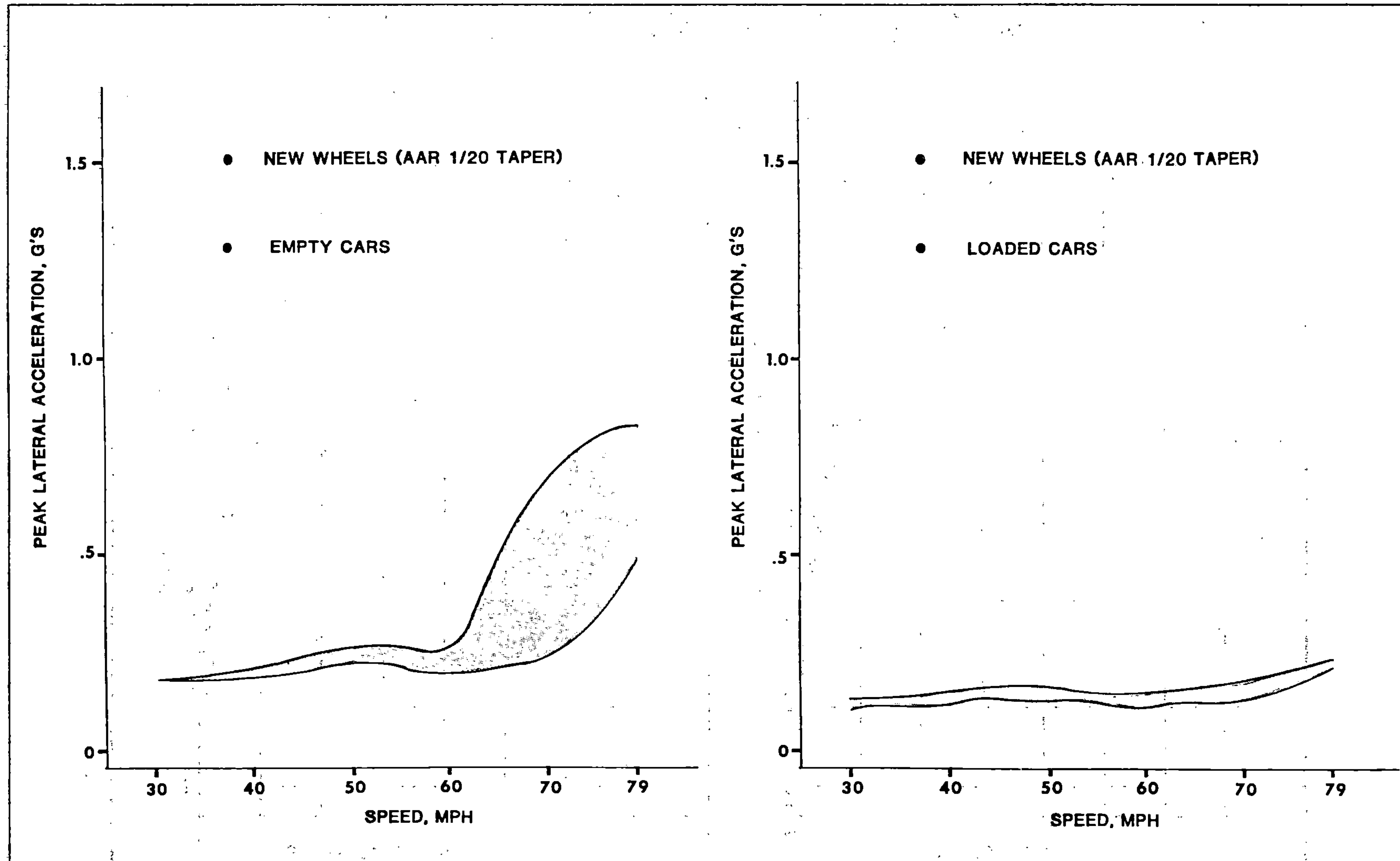


FIGURE 5-14. PEAK LATERAL ACCELERATION VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, NEW WHEELS

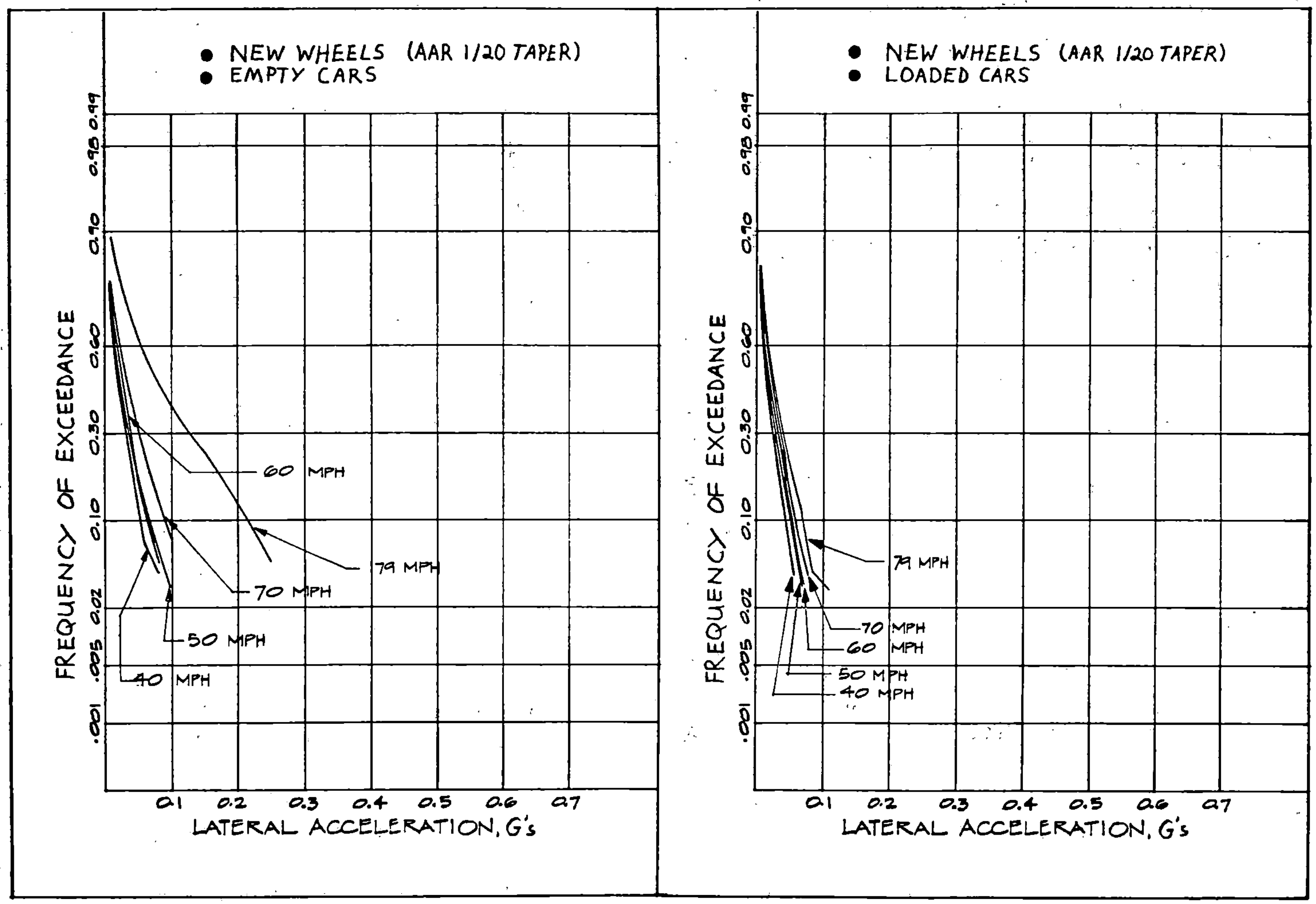


FIGURE 5-15. CUMULATIVE PROBABILITY DISTRIBUTIONS OF LATERAL ACCELERATION - 70-TON TRUCKS WITH FLAT CARS, NEW WHEELS

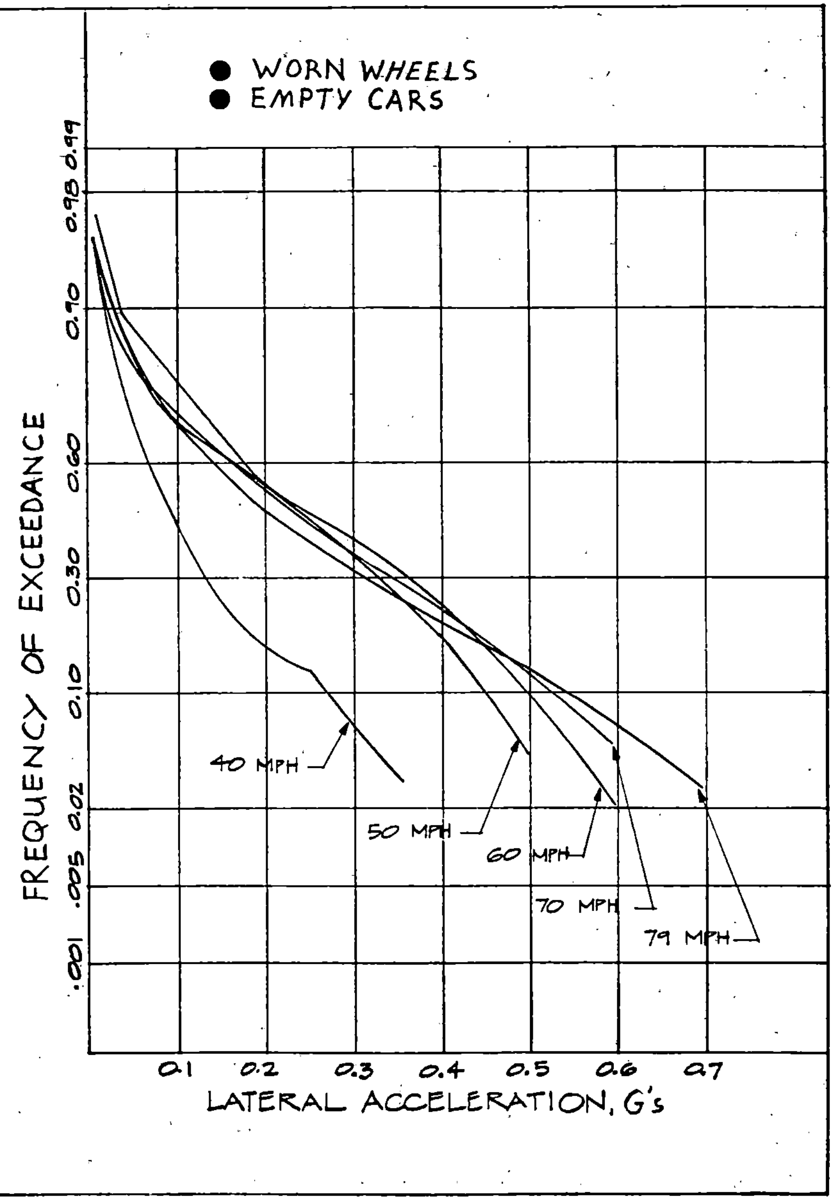


FIGURE 5-16. CUMULATIVE PROBABILITY DISTRIBUTION OF LATERAL ACCELERATION - 70-TON TRUCKS WITH FLAT CARS, WORN WHEELS

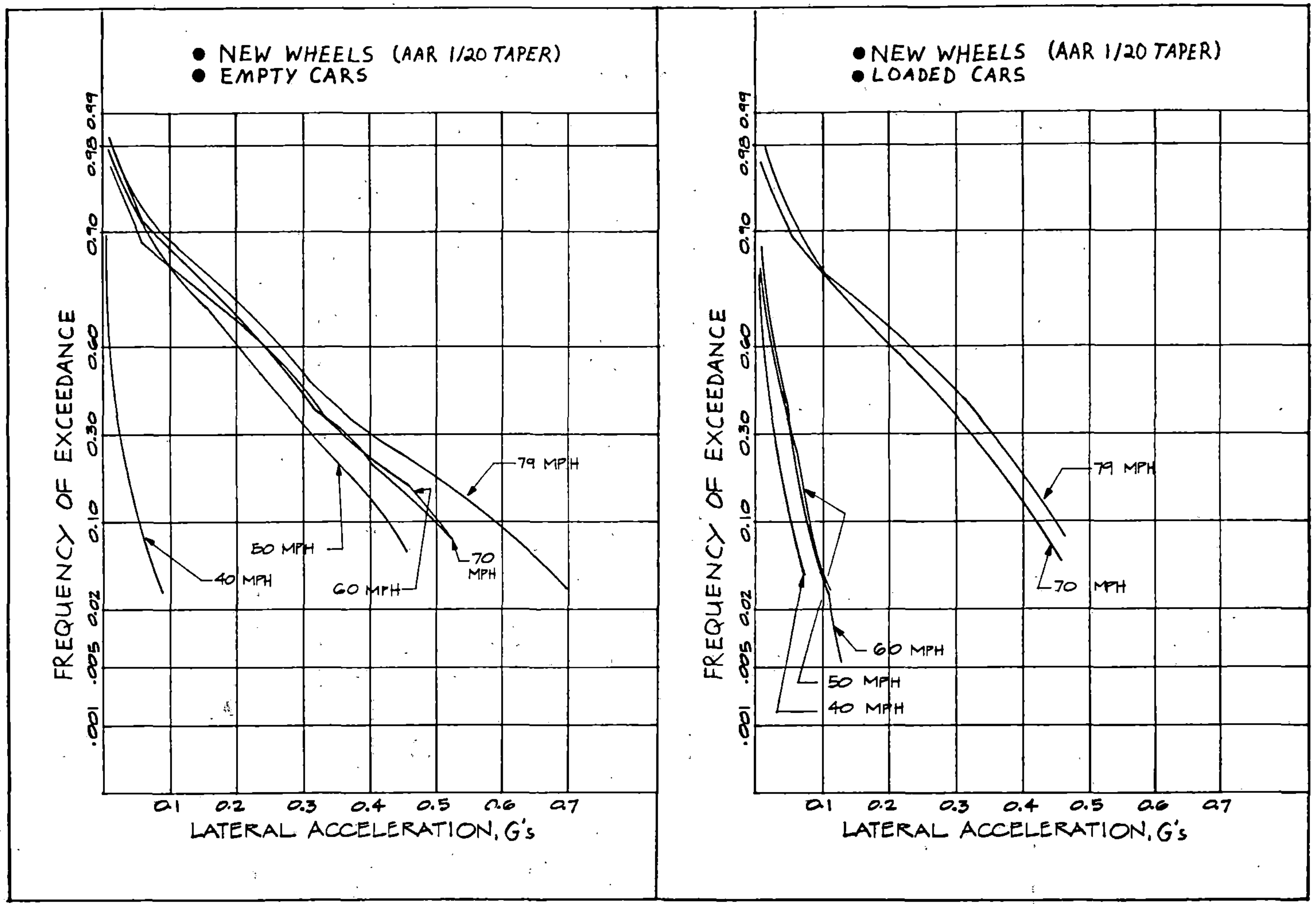


FIGURE 5-17. CUMULATIVE PROBABILITY DISTRIBUTIONS OF LATERAL ACCELERATION - 70-TON TRUCKS WITH BOX TYPE CARS, NEW WHEELS

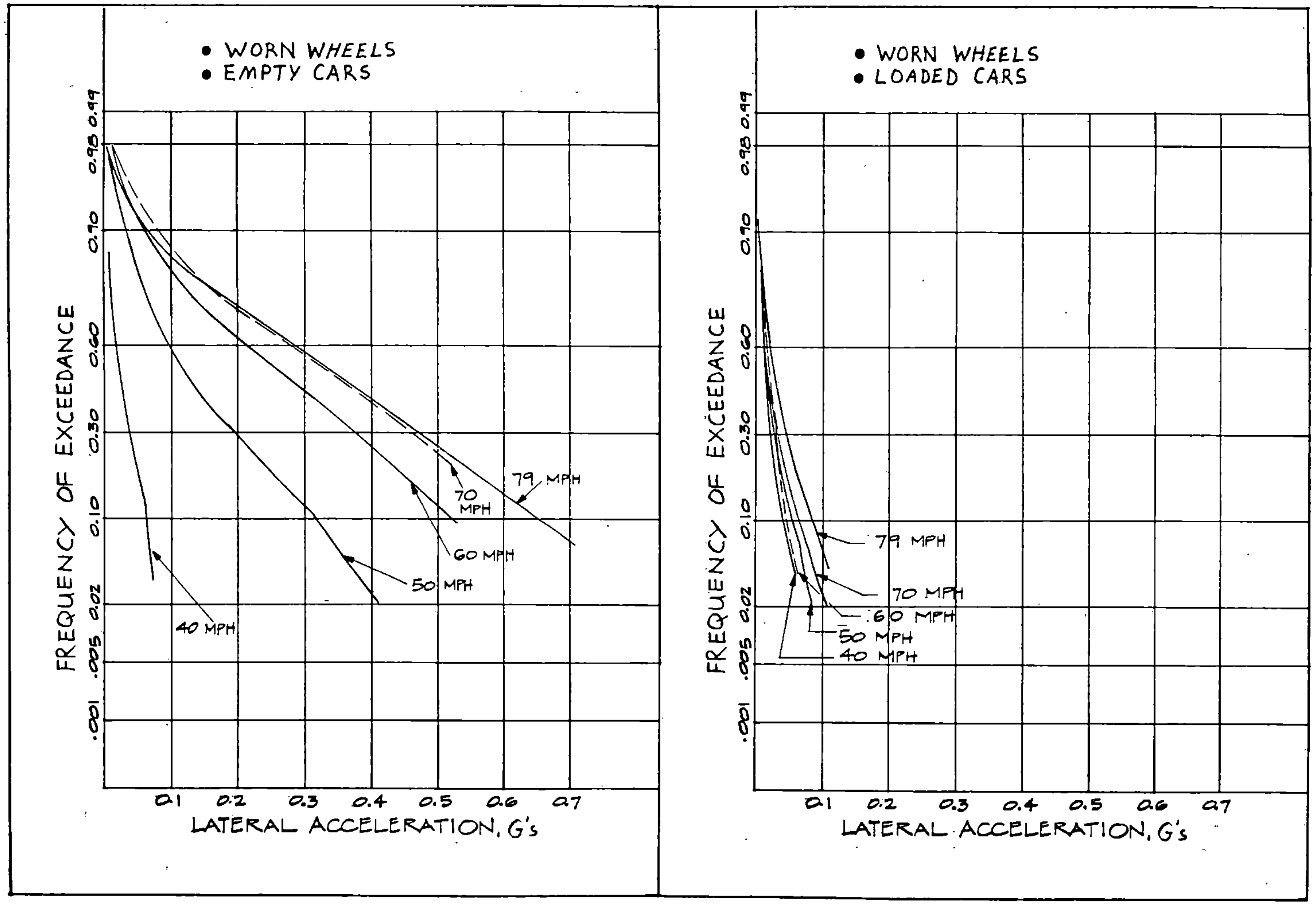


FIGURE 5-18. CUMULATIVE PROBABILITY DISTRIBUTIONS OF LATERAL ACCELERATION - 70-TON TRUCKS WITH BOX TYPE CARS, WORN WHEELS

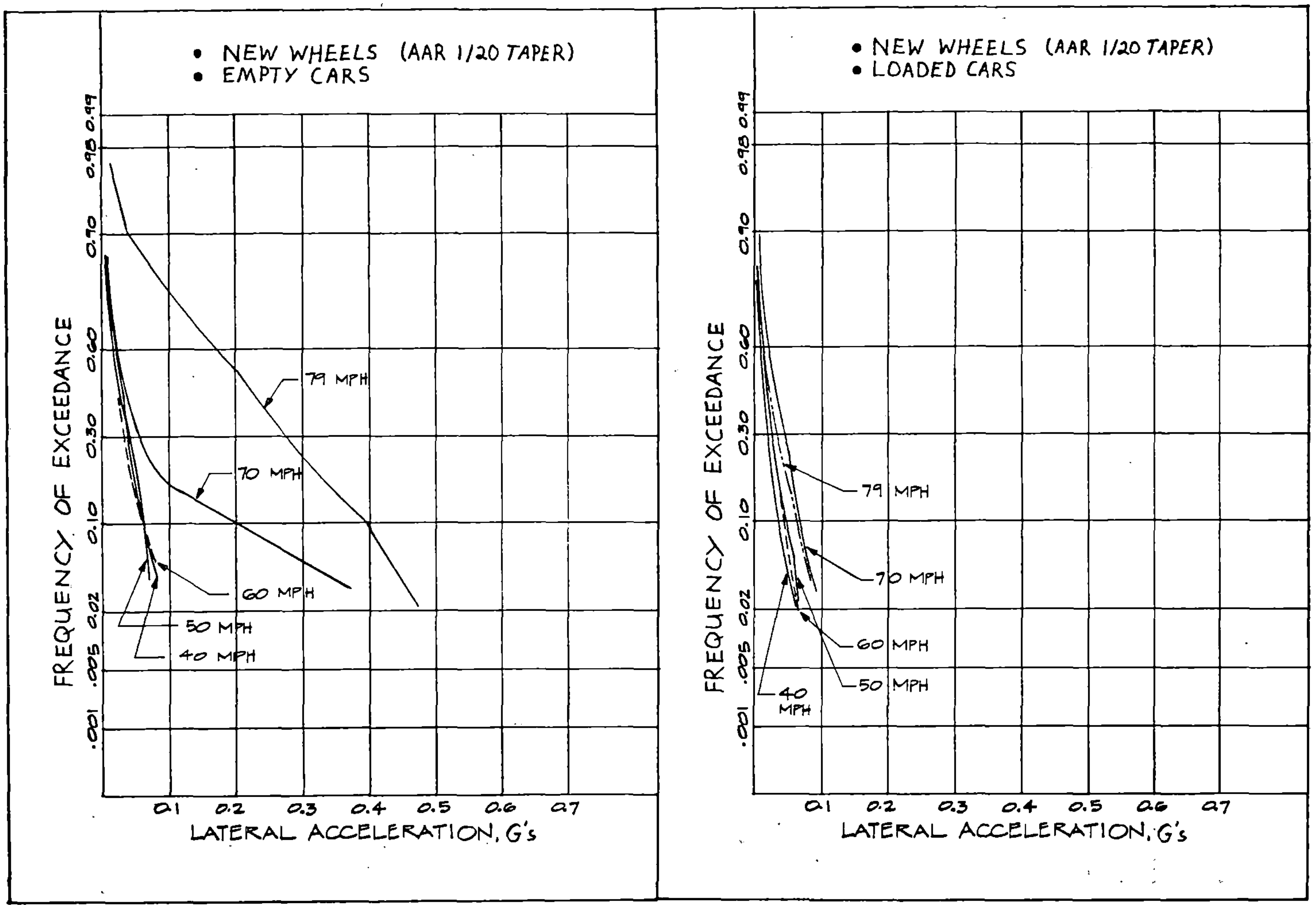


FIGURE 5-19. CUMULATIVE PROBABILITY DISTRIBUTIONS OF LATERAL ACCELERATION - 100-TON TRUCKS WITH BOX TYPE CARS, NEW WHEELS

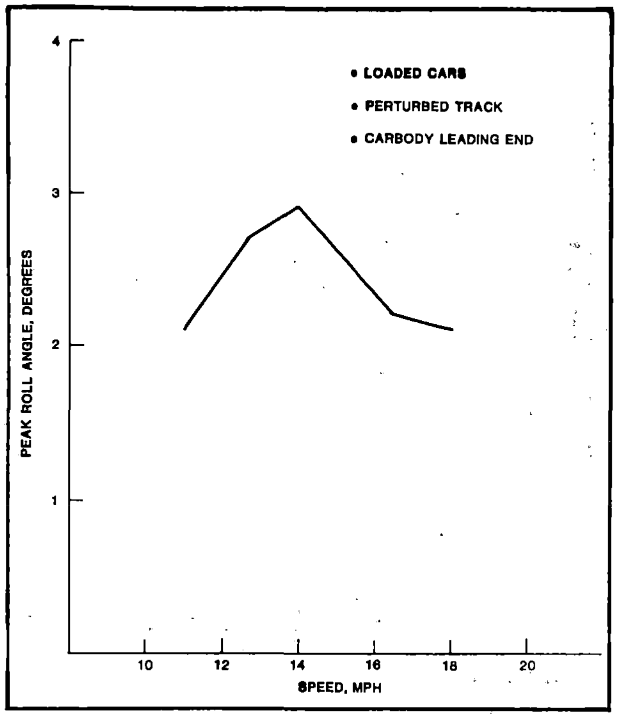


FIGURE 5-20. PEAK ROLL ANGLE (CARBODY TO SIDEFRAME) VERSUS SPEED, 70-TON TRUCKS WITH BOX TYPE CARS, CYLINDRICAL WHEELS

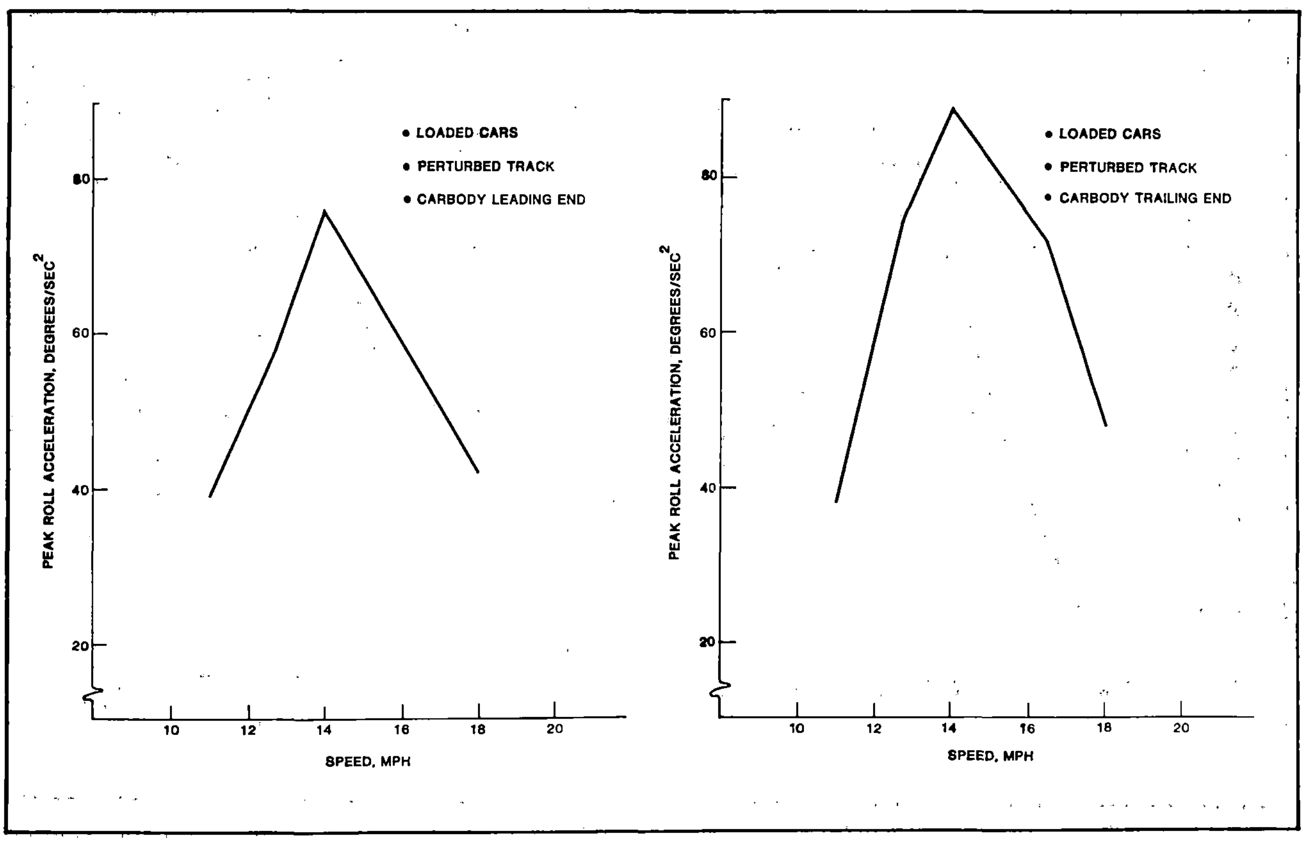


FIGURE 5-21. PEAK ROLL ACCELERATION VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, CYLINDRICAL WHEELS

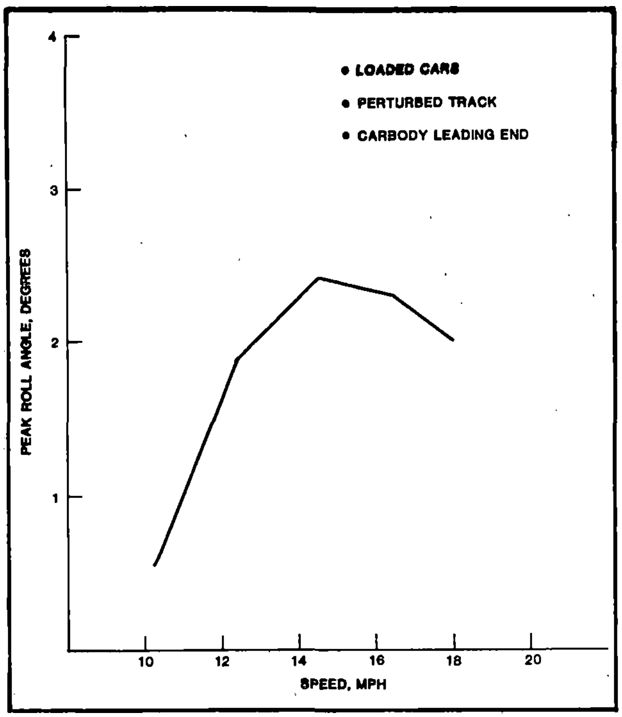


FIGURE 5-22. PEAK ROLL ANGLE (CARBODY TO SIDEFRAME) VERSUS SPEED, 100-TON TRUCKS WITH BOX TYPE CARS, CYLINDRICAL WHEELS

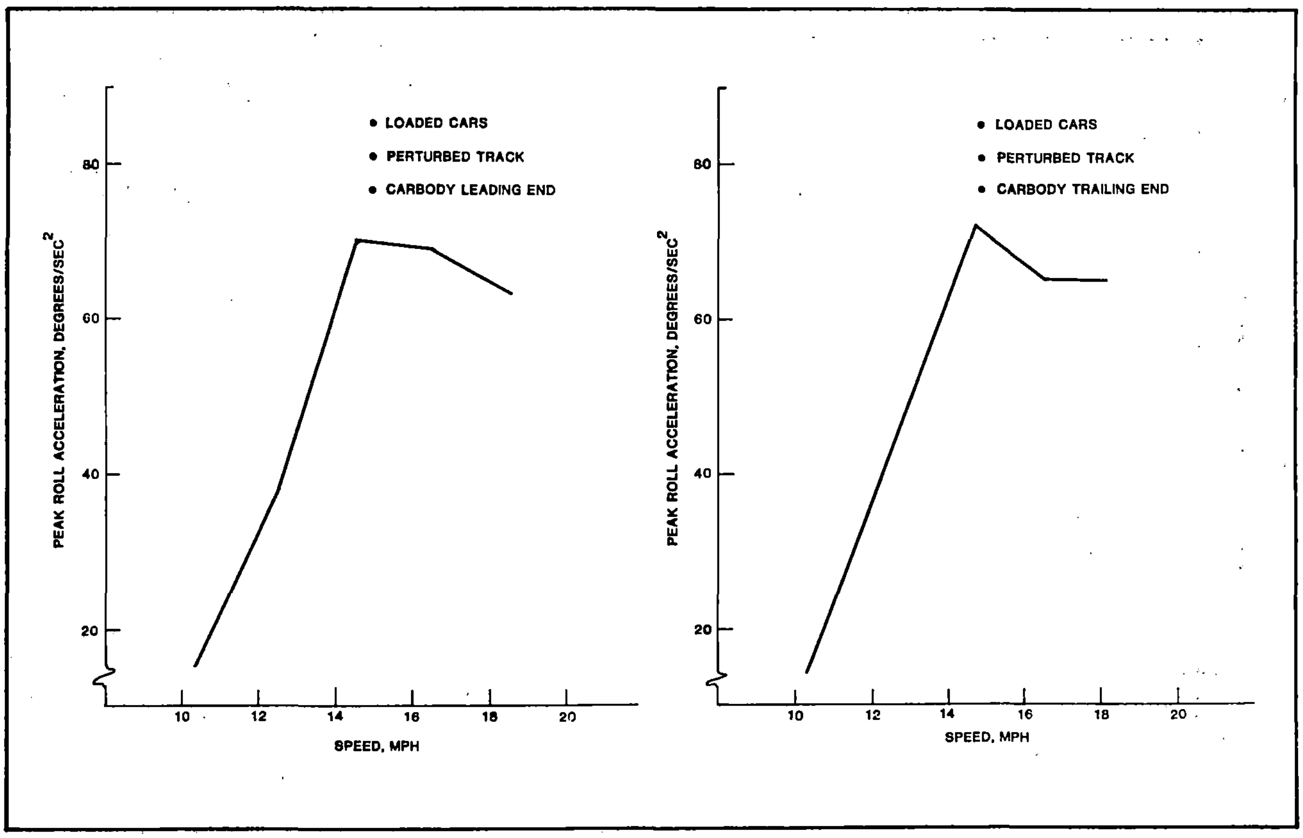


FIGURE 5-23. PEAK ROLL ACCELERATION VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, CYLINDRICAL WHEELS.

TABLE 5-1. STATISTICAL SUMMARY OF WHEEL/RAIL LOAD DATA OF 100-TON TRUCKS (WITH LOADED BOX TYPE CARS FROM CURVED YARD TRACK TESTS)

Test Condition	Vertical Load (lb)		Lateral Load (lb)		L/V Ratio		Wheel Unloading Index	
	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.
9.8 mph, 16° Curve -.26" Superelevation							0.138	0.065
Leading Outer Wheel	28300	2880	9790	4430	0.35	0.163		
Leading Inner Wheel	26570	2860	6340	1800	0.239	0.066		
Trailing Outer Wheel	29620	2960	5430	8800	0.175	0.191		
Trailing Inner Wheel	26840	2320	4960	8390	0.317	0.303		
8.55 mph, 15.75° Curve -.30" Superelevation							0.208	0.108
Leading Outer Wheel	24560	3700	8230	2000	0.342	0.0995		
Leading Inner Wheel	28730	3200	12460	5580	0.451	0.234		
Trailing Outer Wheel	25270	2830	9830	2070	0.396	0.103		
Trailing Inner Wheel	26200	5050	12400	3520	0.483	0.148		

TABLE 5-2. STATISTICAL SUMMARY OF WHEEL/RAIL LOAD DATA OF 100-TON TRUCKS (WITH EMPTY BOX TYPE CARS FROM CURVED YARD TRACK TESTS)

	Vertical Load (lb)		Lateral Load (lb)		L/V Ratio		Wheel Unloading Index	
	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.
9.3 mph, 16° Curve -.26" Superelevation							0.409	0.083
Leading Outer Wheel	9330	680	1850	1000	0.196	0.103		
Leading Inner Wheel	5440	710	2030	680	0.366	0.135		
Trailing Outer Wheel	8670	970	-15	1030	0.009	0.114		
Trailing Inner Wheel	9740	630	440	740	0.043	0.079		
9.3 mph, 15.75° Curve -.3" Superelevation							0.265	0.073
Leading Outer Wheel	7960	920	3210	1510	0.399	0.186		
Leading Inner Wheel	7730	1050	1720	790	0.222	0.105		
Trailing outer Wheel	10080	1280	1710	1780	0.043	0.079		
Trailing Inner Wheel	10430	1860	-1450	1680	-.106	0.130		

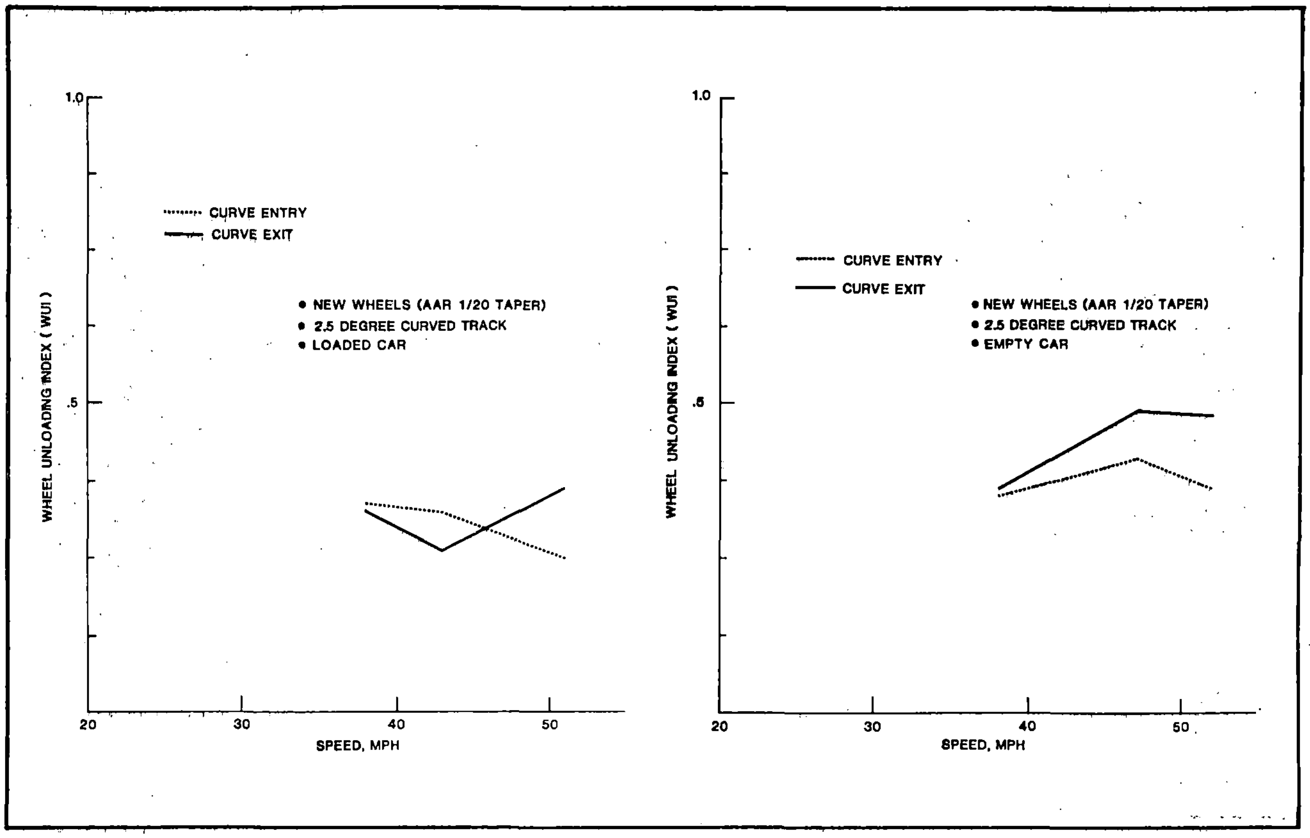


FIGURE 5-24. PEAK VALUE OF WHEEL UNLOADING INDEX VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 2.5° CURVED TRACK

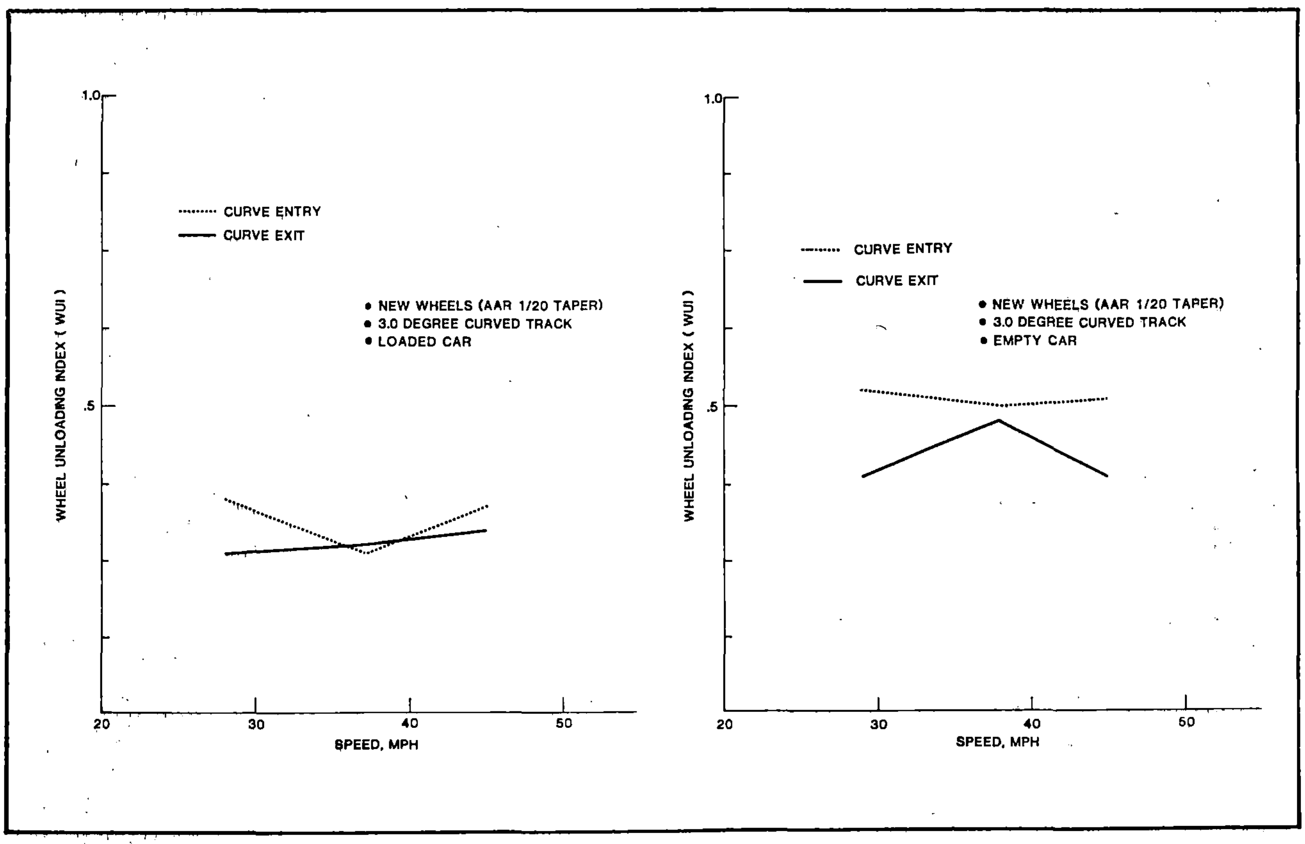


FIGURE 5-25. PEAK VALUE OF WHEEL UNLOADING INDEX VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 3.0° CURVED TRACK

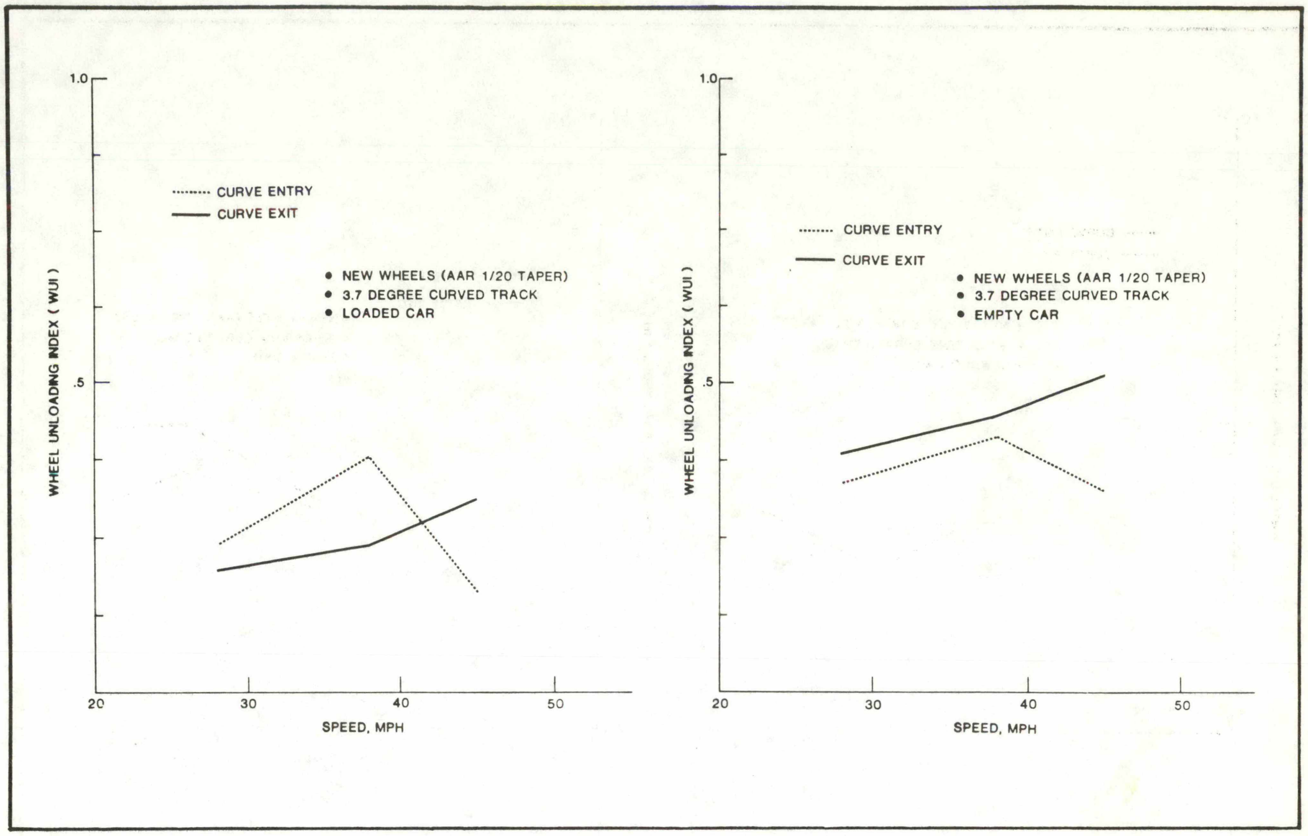


FIGURE 5-26. PEAK VALUE OF WHEEL UNLOADING INDEX VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 3.7° CURVED TRACK

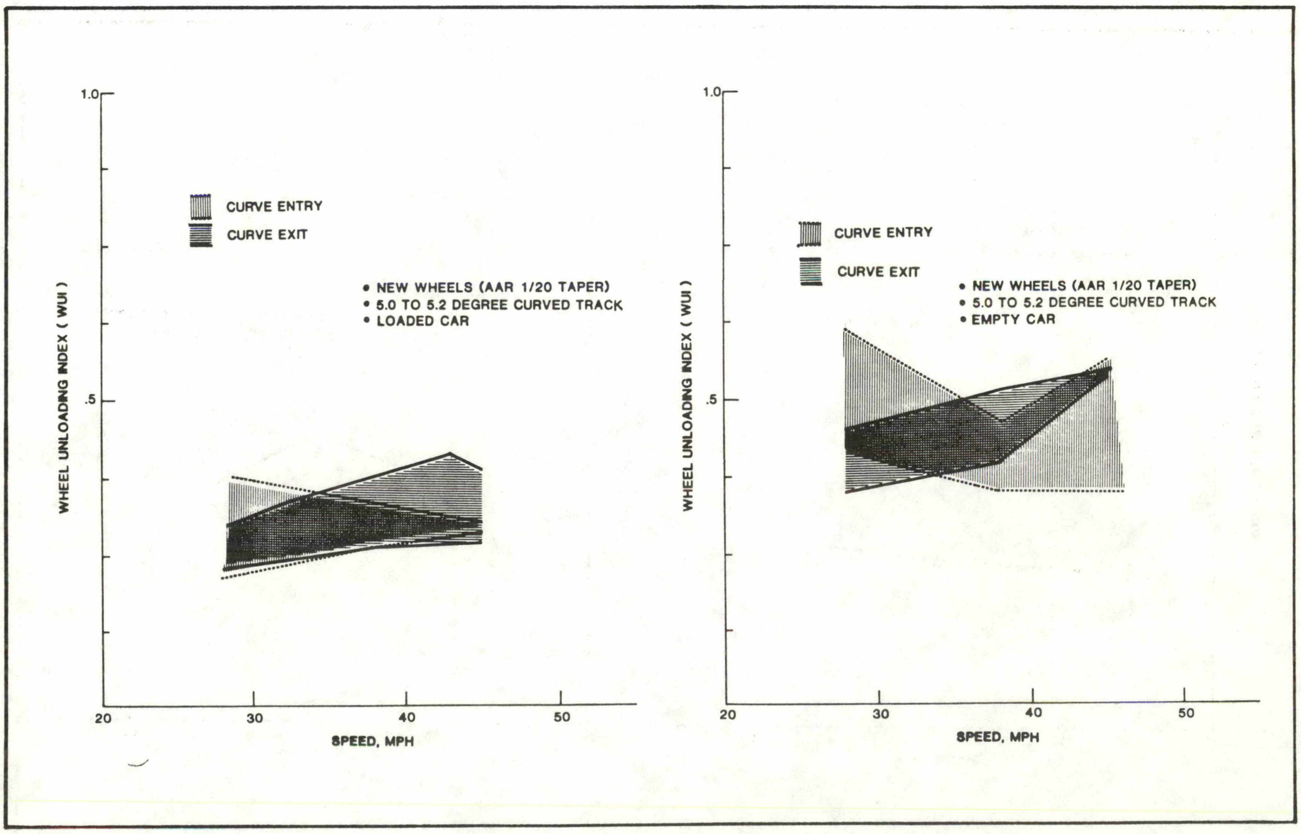


FIGURE 5-27. PEAK VALUE OF WHEEL UNLOADING INDEX VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 5.0° to 5.2° CURVED TRACK

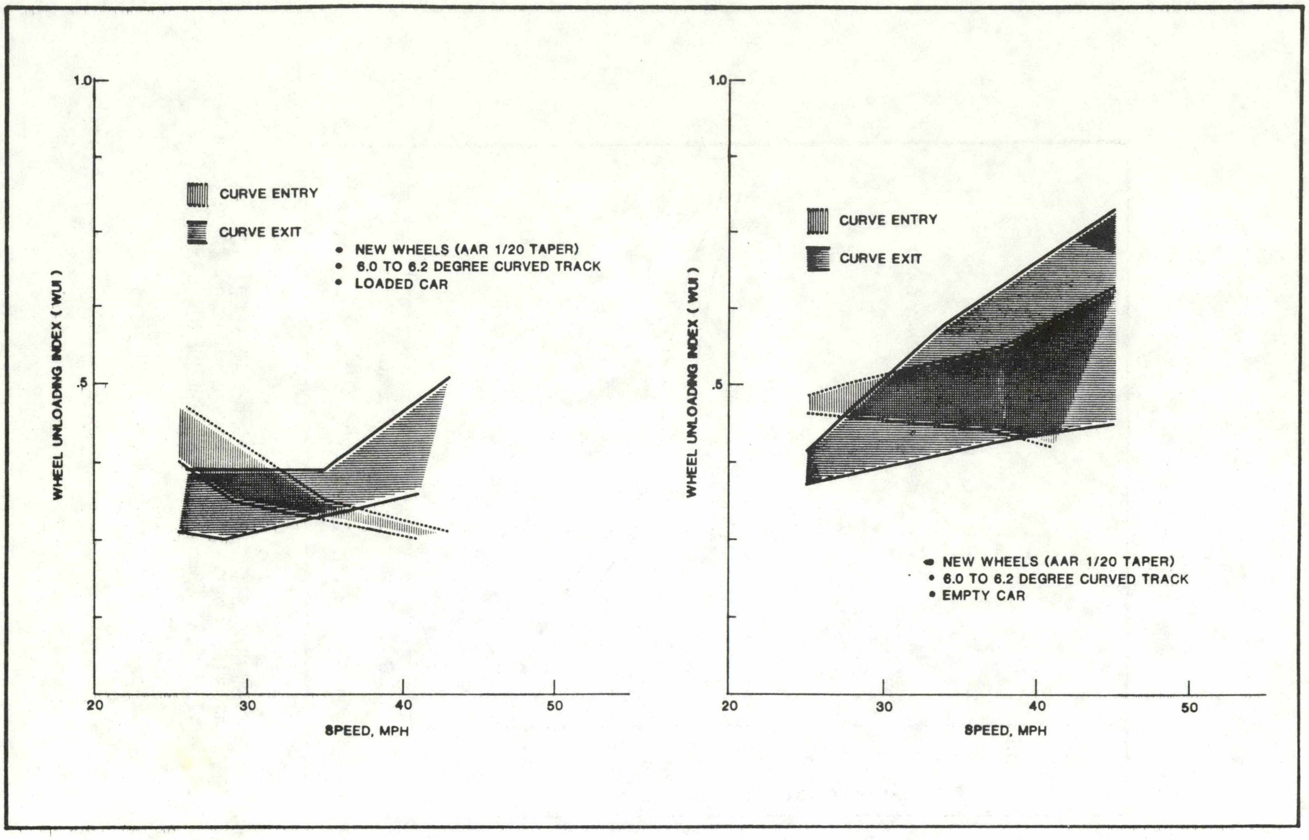


FIGURE 5-28. PEAK VALUE OF WHEEL UNLOADING INDEX VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 6.0^{0} to 6.2^{0} CURVED TRACK

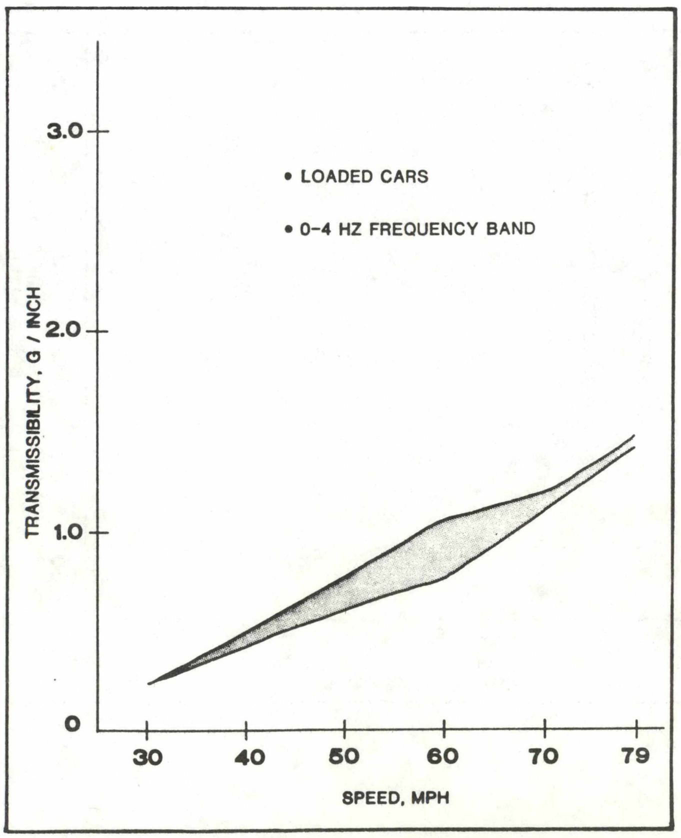


FIGURE 5-29. RATIO OF RMS VERTICAL ACCELERATION AT SILL LEVEL TO RMS TRACK PROFILE VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, 0-4 HZ FREQUENCY BAND

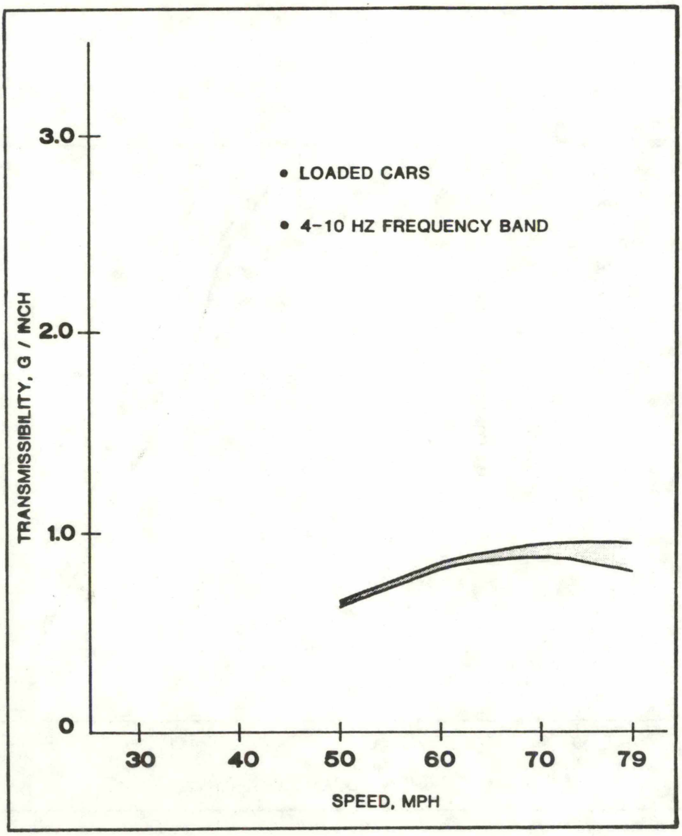


FIGURE 5-30. RATIO OF RMS VERTICAL ACCELERATION AT SILL LEVEL TO RMS TRACK PROFILE VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, 4-10 HZ FREQUENCY BAND

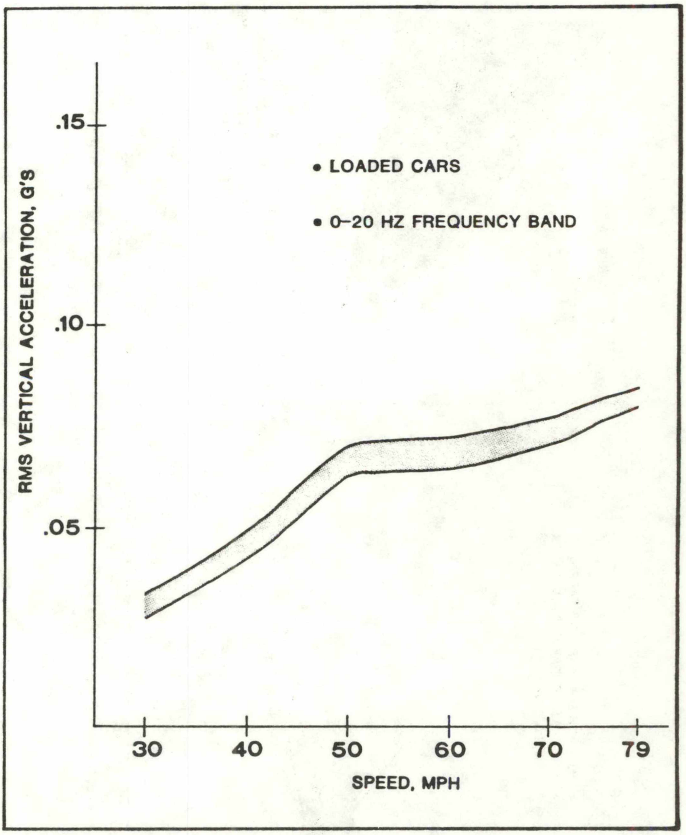


FIGURE 5-31. RMS VERTICAL ACCELERATION AT SILL LEVEL VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, 0-20 HZ FREQUENCY BAND

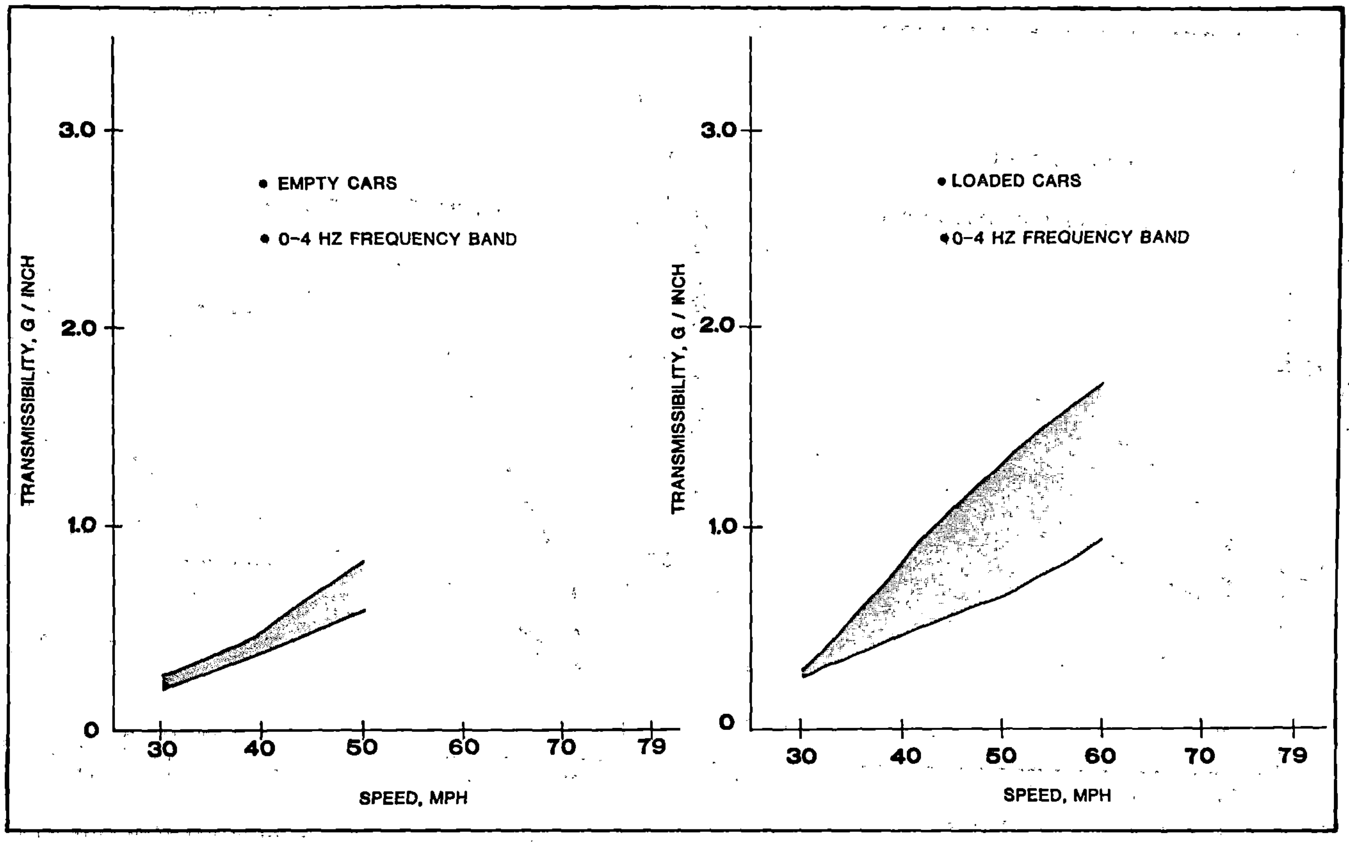


FIGURE 5-32. RATIO OF RMS VERTICAL ACCELERATION AT SILL LEVEL TO RMS TRACK PROFILE VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, 0-4 HZ FREQUENCY BAND

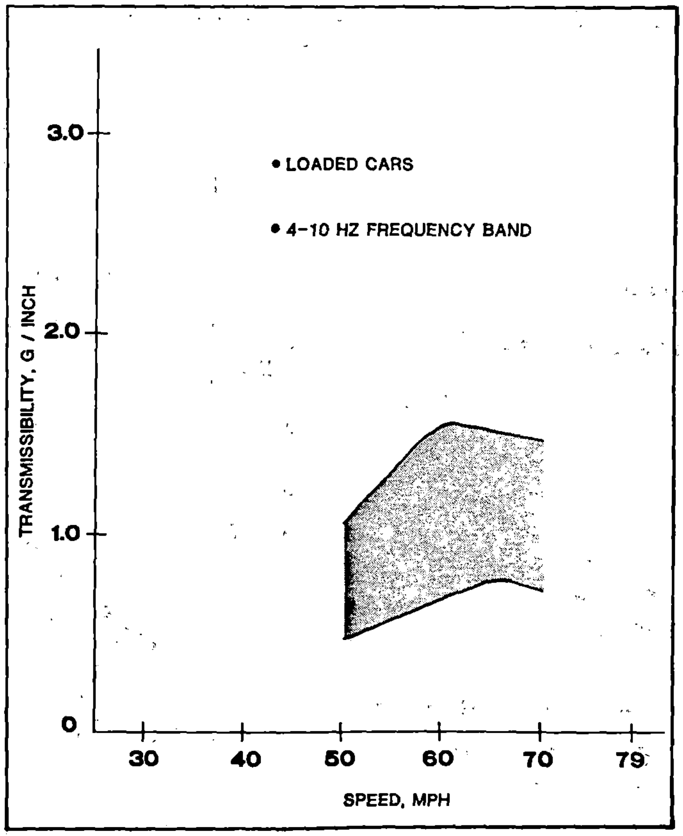


FIGURE 5-33. RATIO OF RMS VERTICAL ACCELERATION AT SILL LEVEL TO RMS TRACK PROFILE VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, 4-10 HZ FREQUENCY BAND

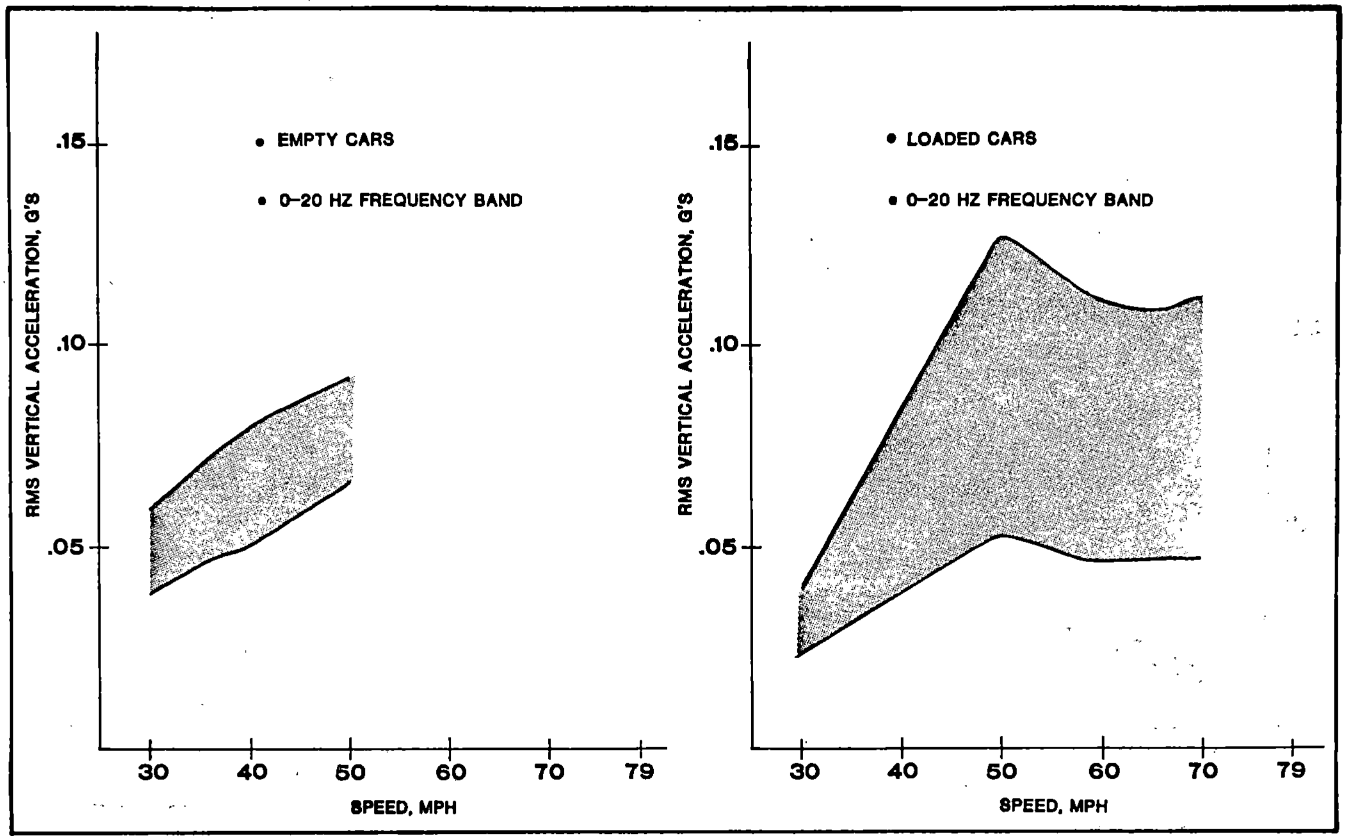


FIGURE 5-34. RMS VERTICAL ACCELERATION AT SILL LEVEL VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, 0-20 HZ FREQUENCY BAND

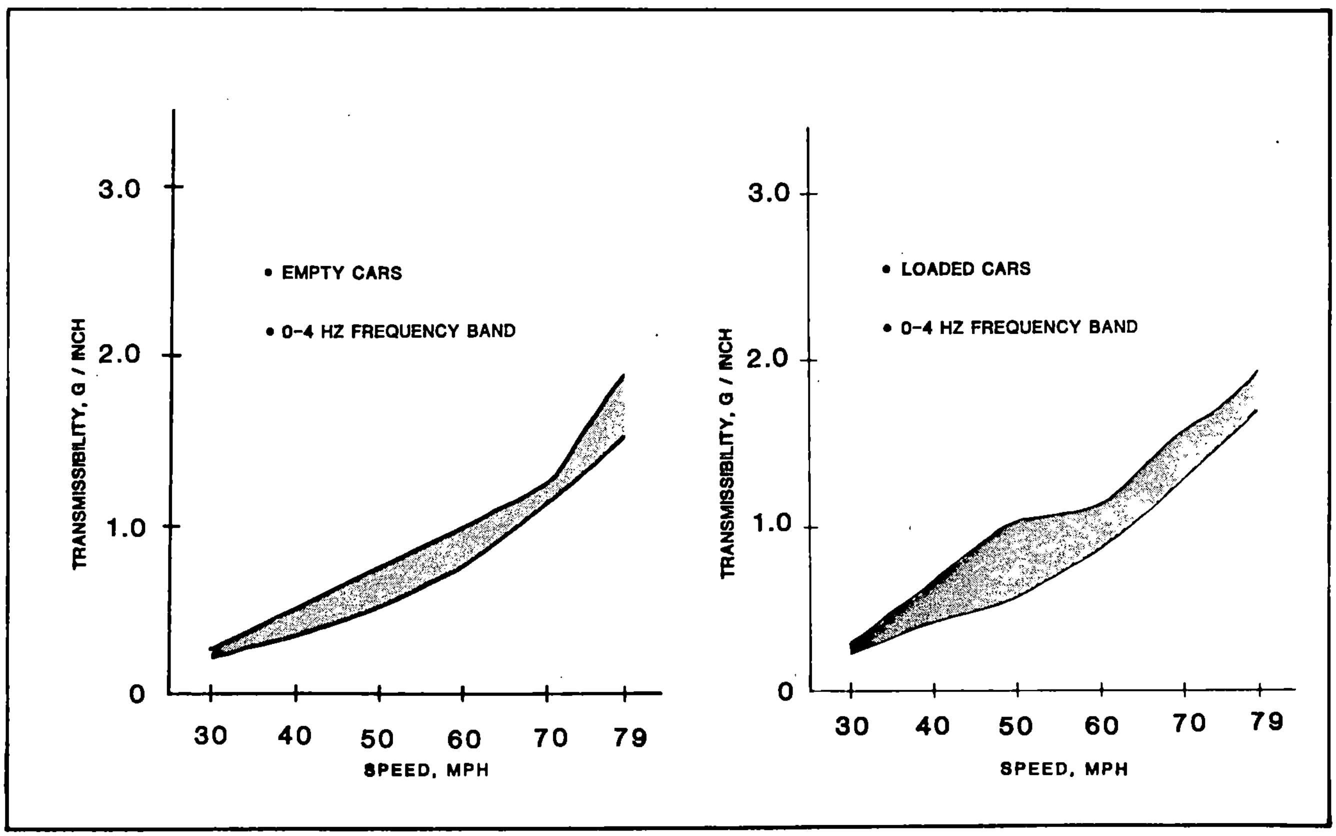


FIGURE 5-35. RATIO OF RMS VERTICAL ACCELERATION AT SILL LEVEL TO RMS TRACK PROFILE VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 0-4 HZ FREQUENCY BAND

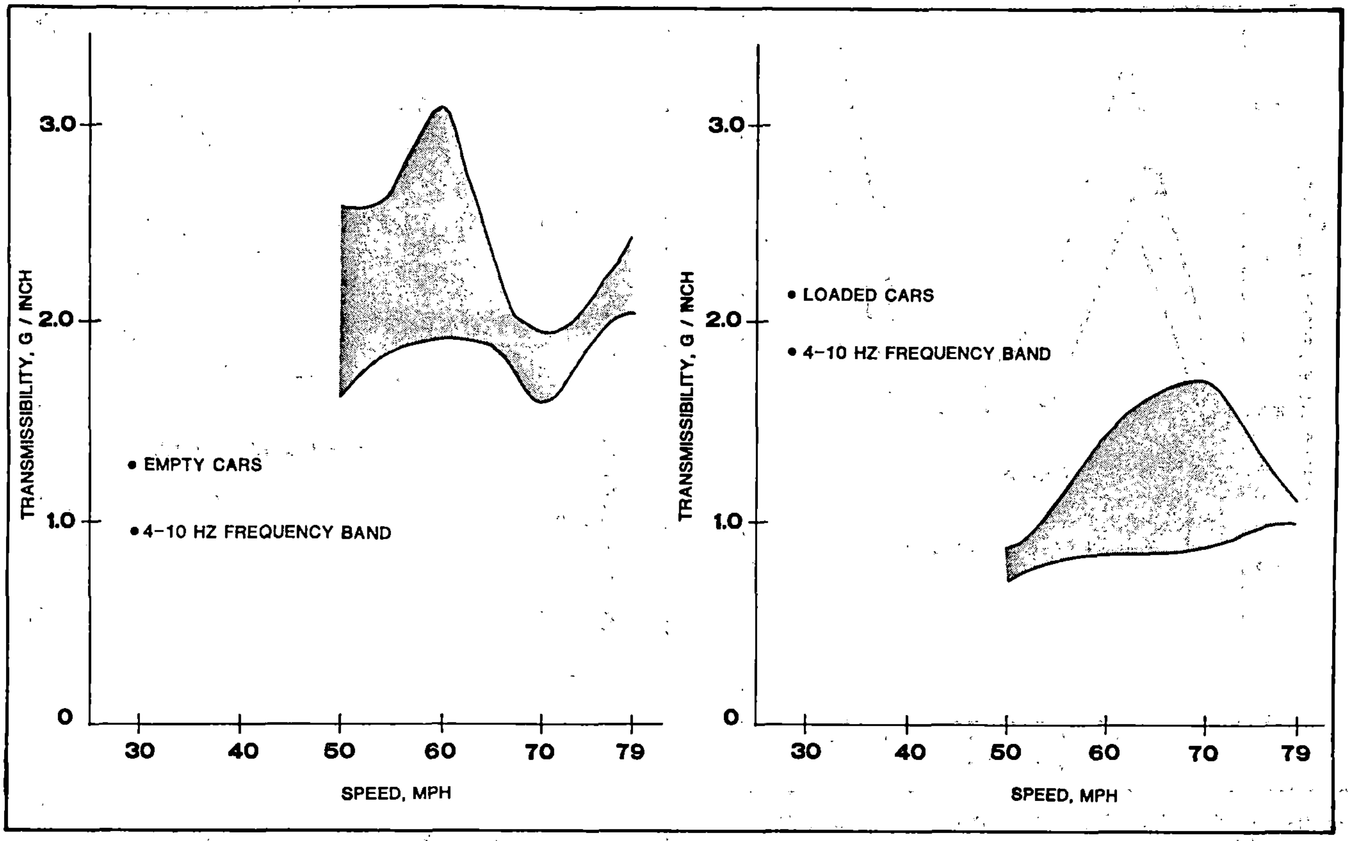


FIGURE 5-36. RATIO OF RMS VERTICAL ACCELERATION AT SILL LEVEL TO RMS TRACK PROFILE VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 4-10 HZ FREQUENCY BAND

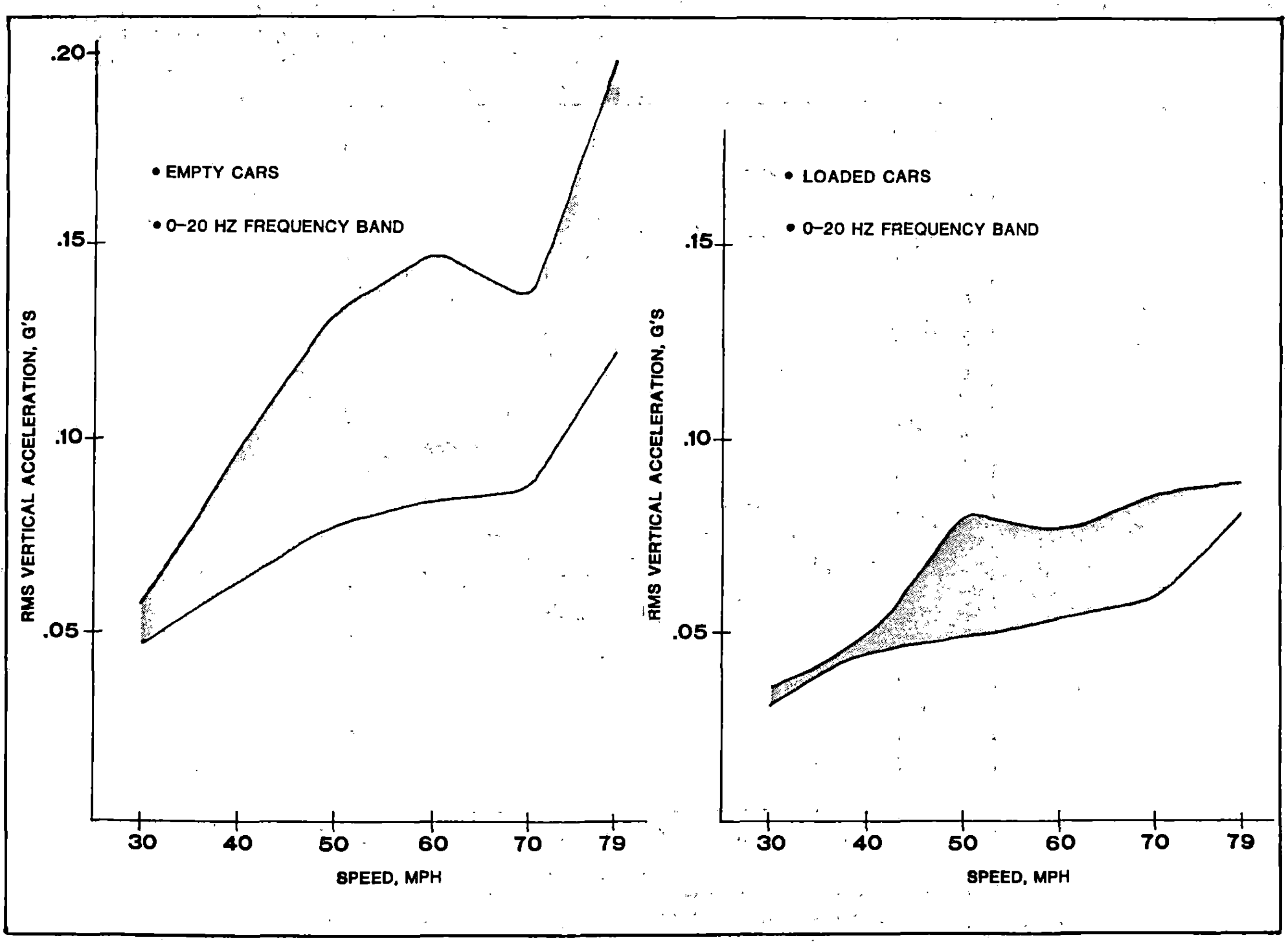


FIGURE 5-37. RMS VERTICAL ACCELERATION AT SILL LEVEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 0-20 HZ FREQUENCY BAND

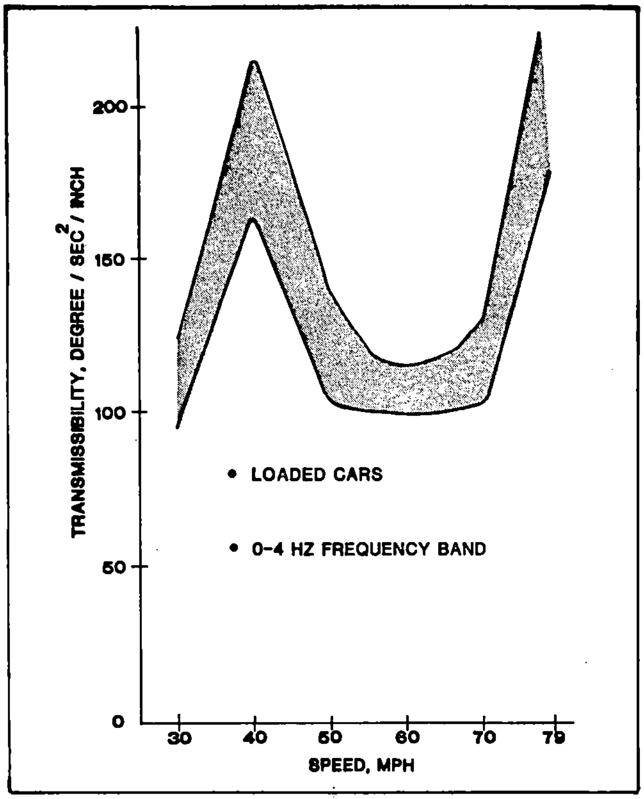


FIGURE 5-38. RATIO OF RMS ROLL ACCELERATION AT EITHER END OF CARBODY TO RMS CROSS LEVEL VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, 0-4 HZ FREQUENCY BAND

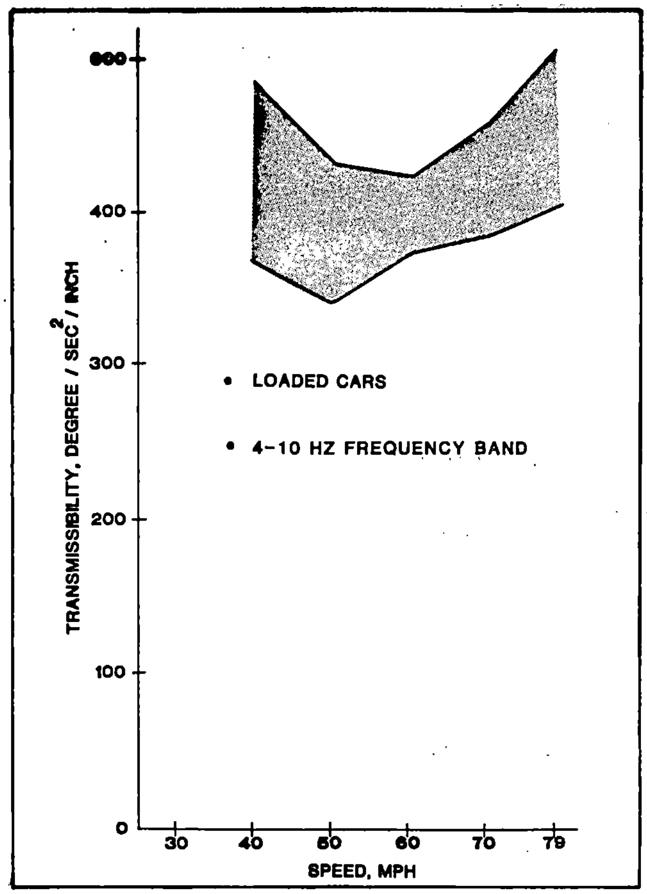


FIGURE 5-39. RATIO OF RMS ROLL ACCELERATION AT EITHER END OF CARBODY TO RMS CROSS LEVEL VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, 4-10 HZ FREQUENCY BAND

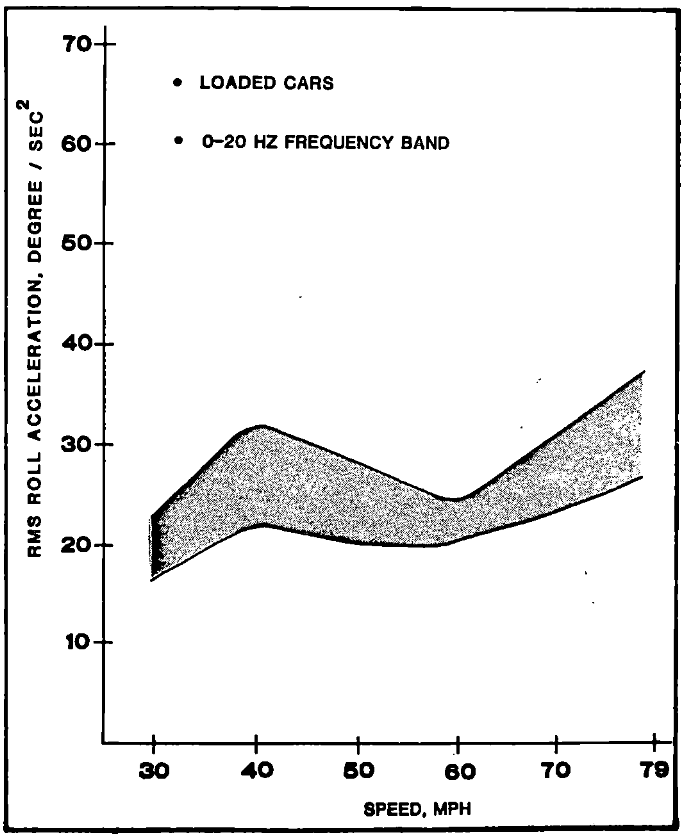


FIGURE 5-40. RMS ROLL ACCELERATION AT EITHER END OF CARBODY VERSUS SPEED - 70-TON TRUCKS WITH FLAT CARS, 0-20 HZ FREQUENCY BAND

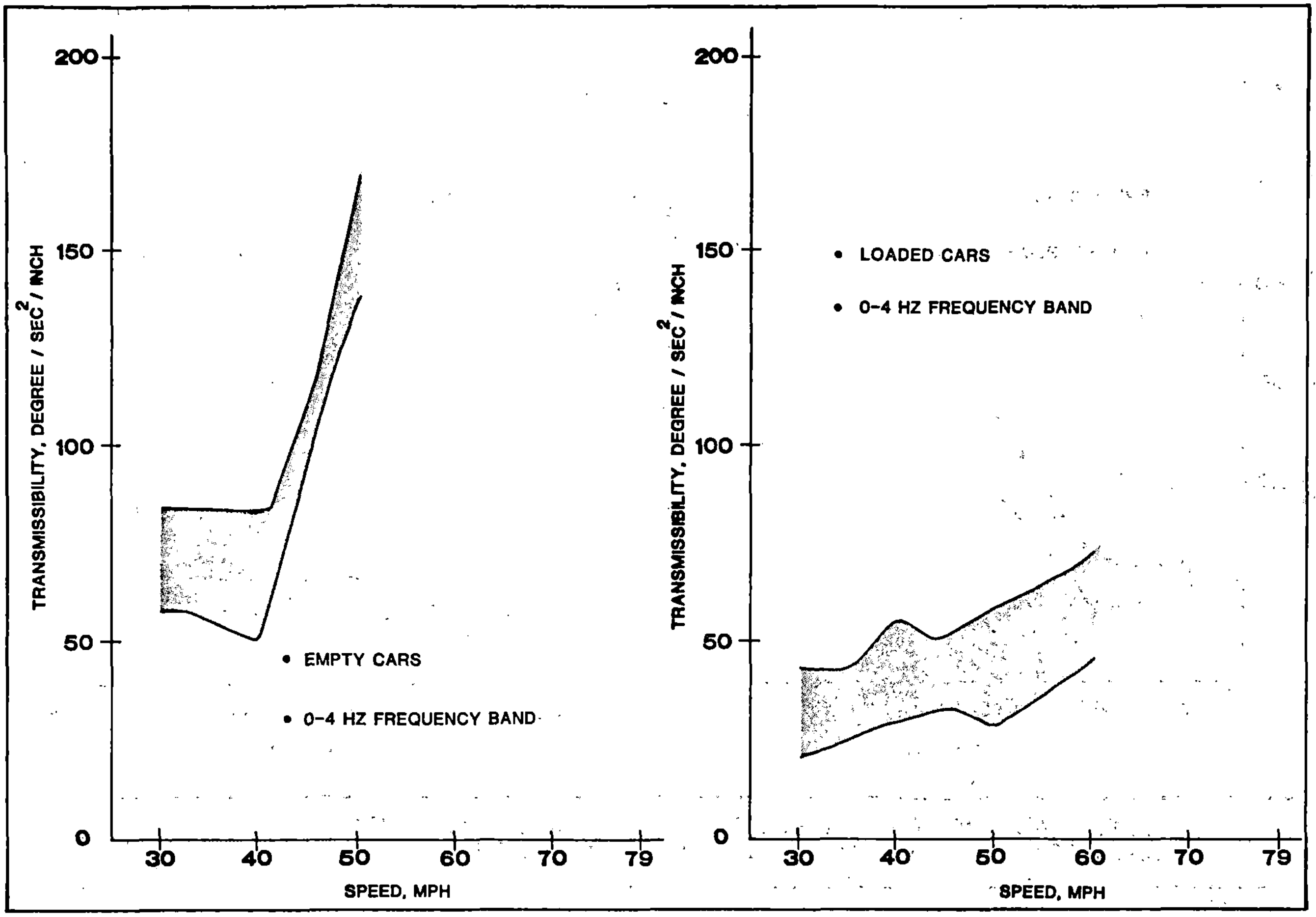


FIGURE 5-41. RATIO OF RMS ROLL ACCELERATION AT EITHER END OF CARBODY TO RMS CROSS LEVEL VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, 0-4 HZ FREQUENCY BAND

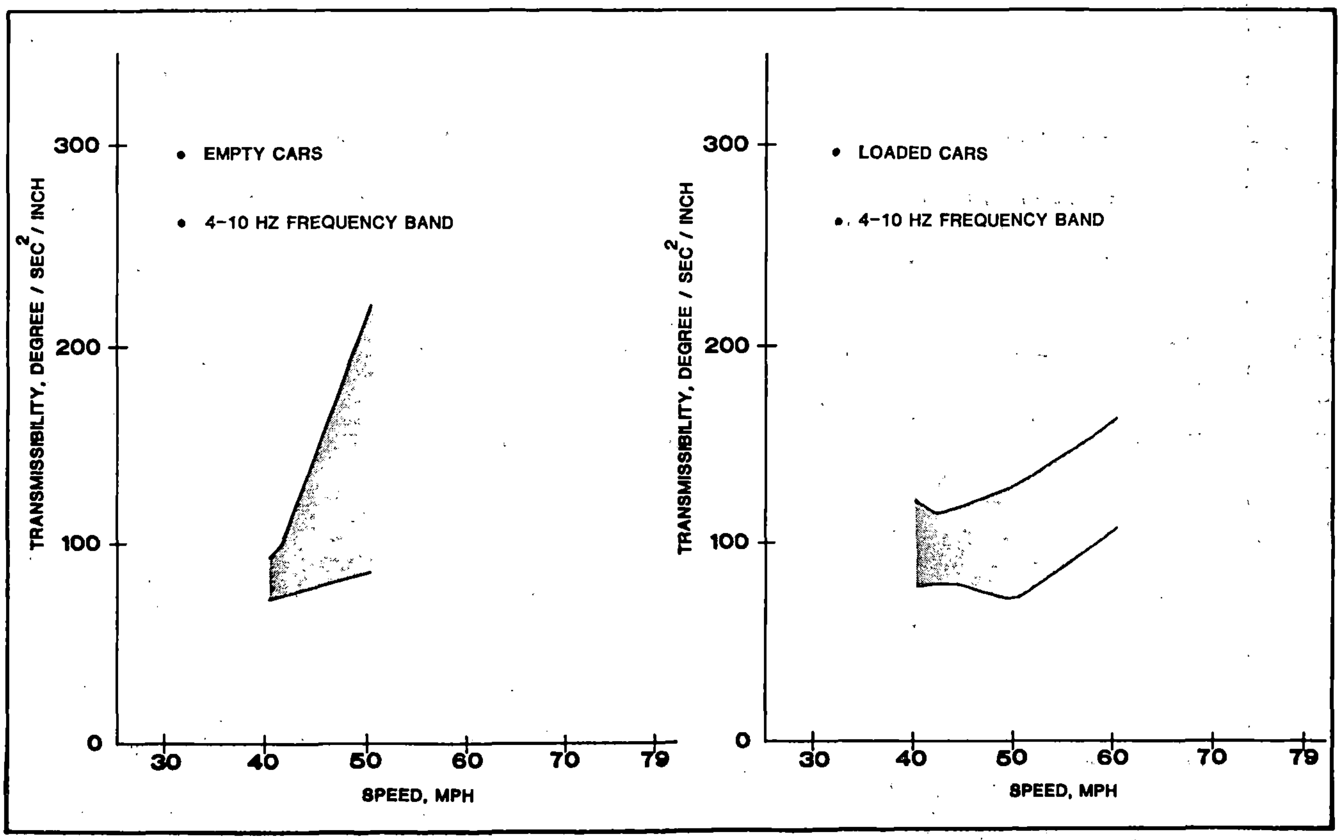


FIGURE 5-42. RATIO OF RMS ROLL ACCELERATION AT EITHER END OF CARBODY TO RMS CROSS LEVEL VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, 4-10 HZ FREQUENCY BAND

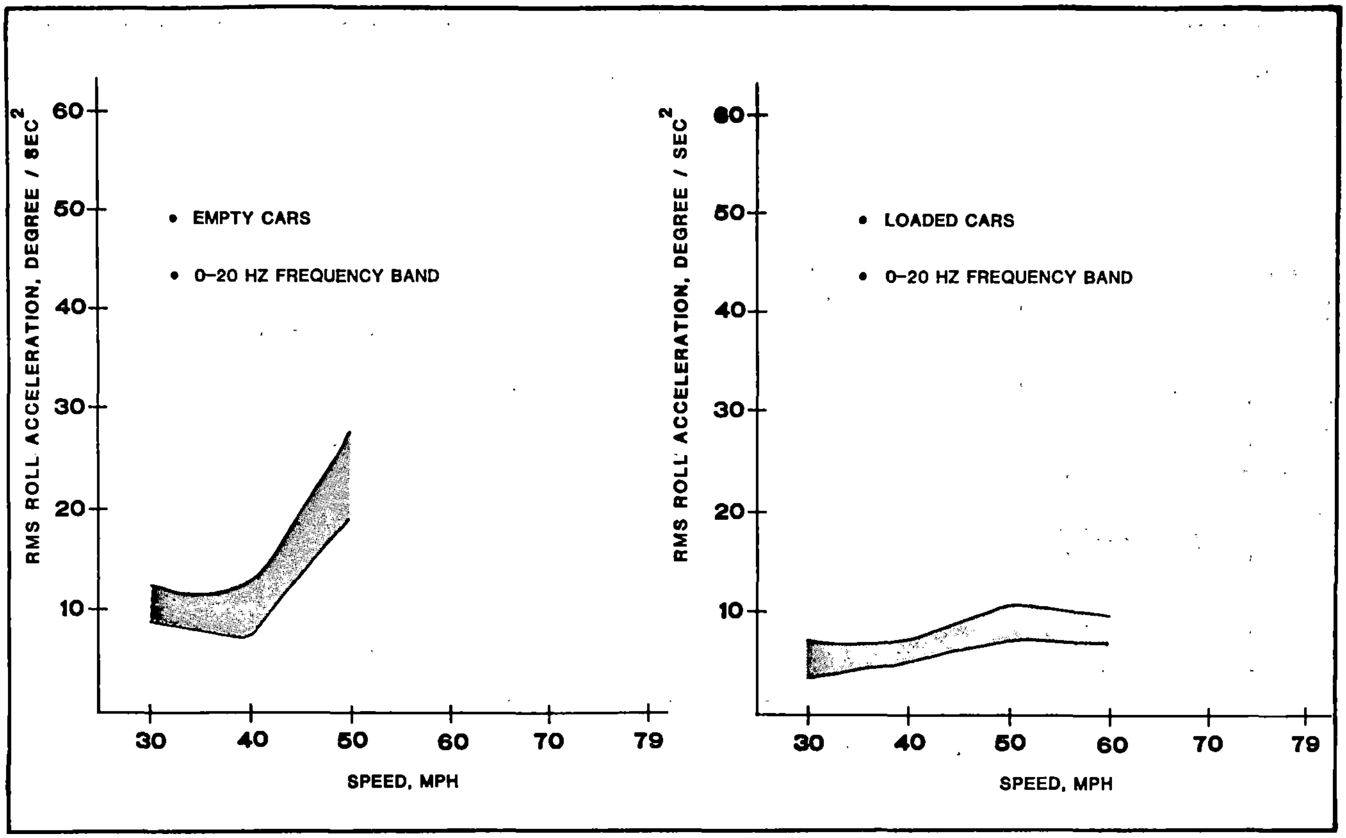


FIGURE 5-43. RMS ROLL ACCELERATION AT EITHER END OF CARBODY VERSUS SPEED - 70-TON TRUCKS WITH BOX TYPE CARS, 0-20 HZ FREQUENCY BAND

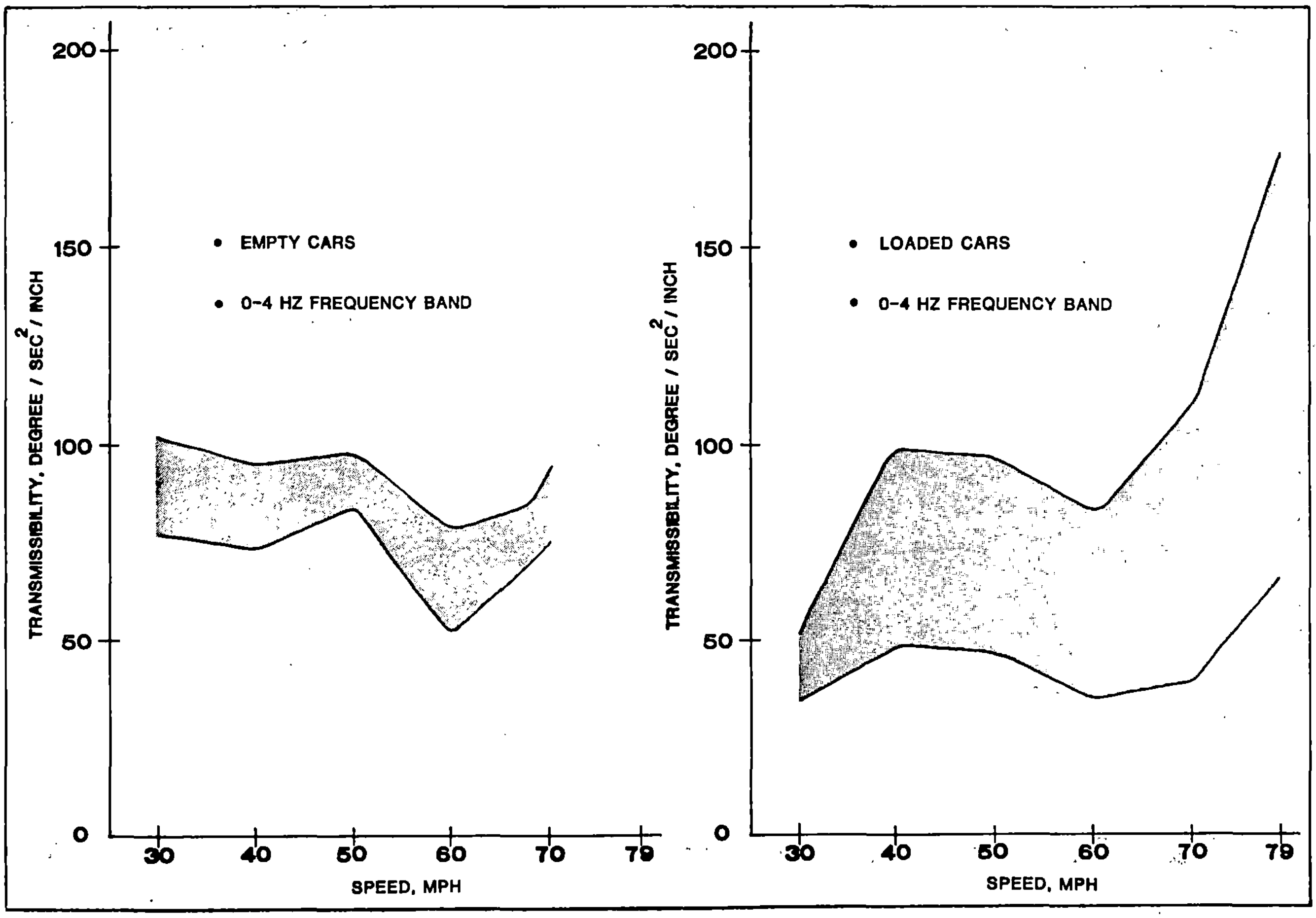


FIGURE 5-44. RATIO OF RMS ROLL ACCELERATION AT EITHER END OF CARBODY TO RMS CROSS LEVEL - 100-TON TRUCKS WITH BOX TYPE CARS, 0-4 HZ FREQUENCY BAND

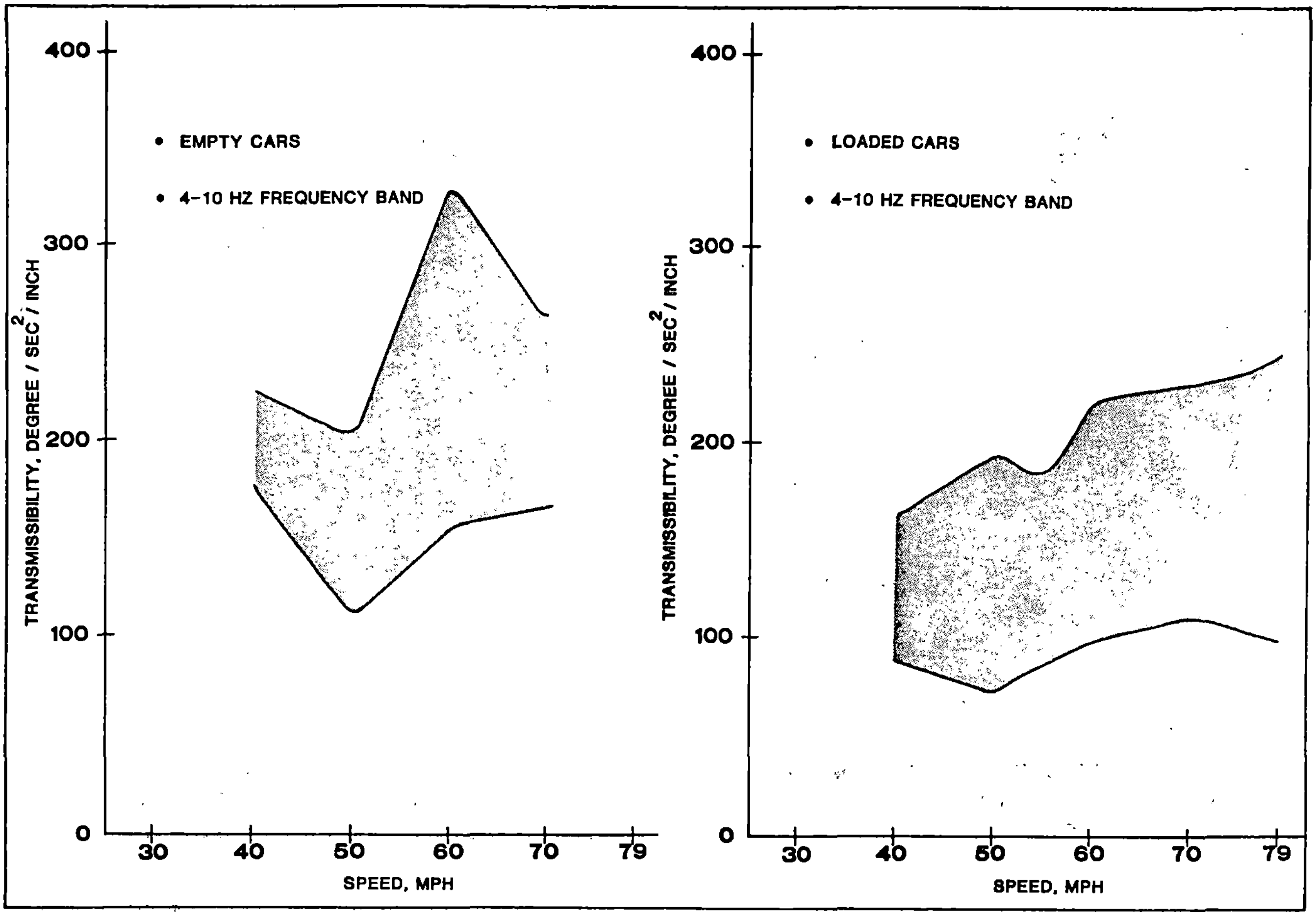


FIGURE 5-45. RATIO OF RMS ROLL ACCELERATION AT EITHER END OF CARBODY TO RMS CROSS LEVEL - 100-TON TRUCKS WITH BOX TYPE CARS, 4-10 HZ FREQUENCY BAND

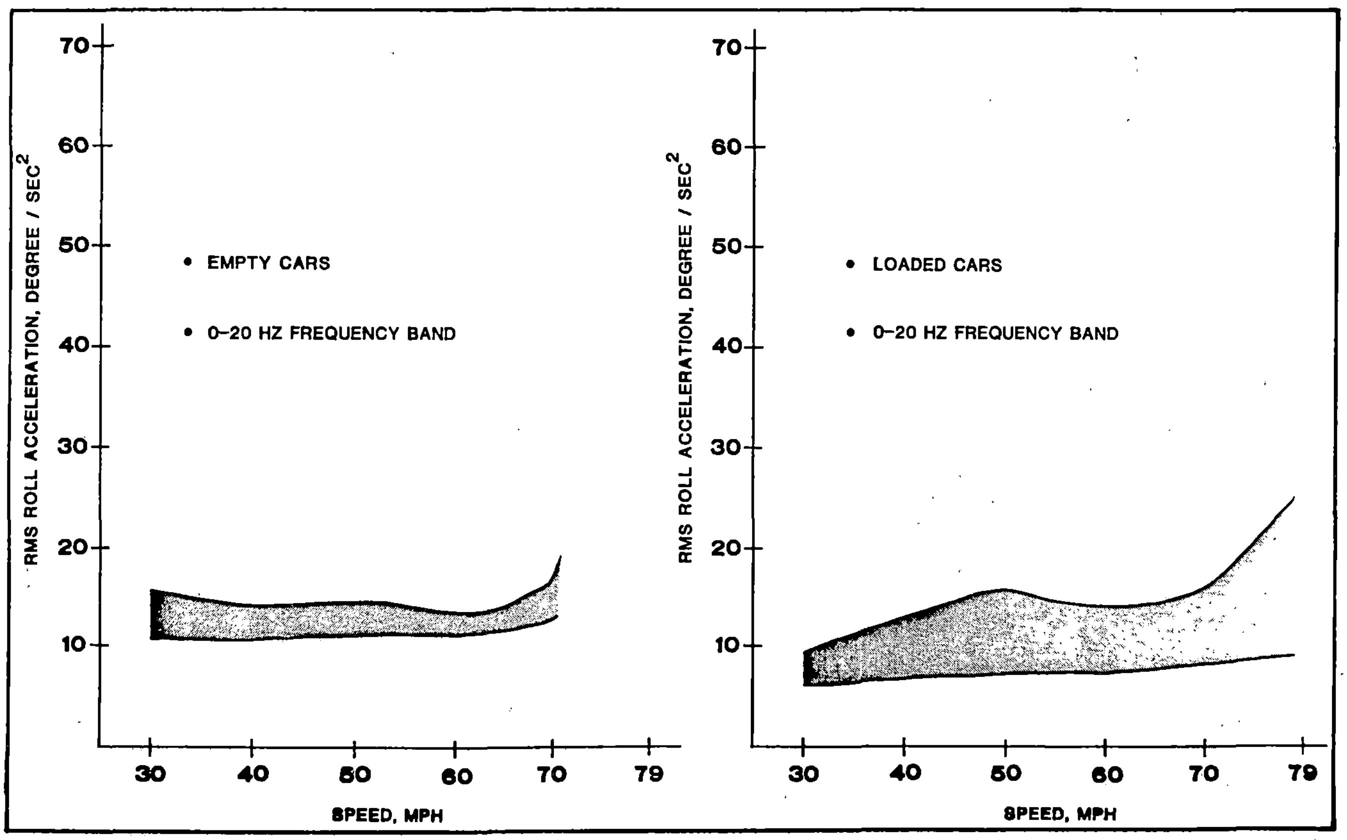


FIGURE 5-46. RMS ROLL ACCELERATION AT EITHER END OF CARBODY - 100-TON TRUCKS WITH BOX TYPE CARS, 0-20 HZ FREQUENCY BAND

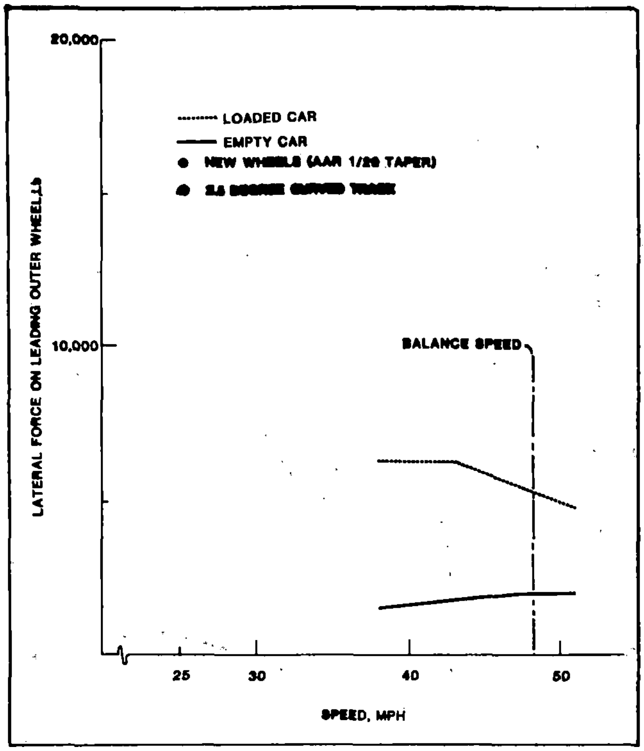


FIGURE 5-47. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 2.5^{0} CURVED TRACK

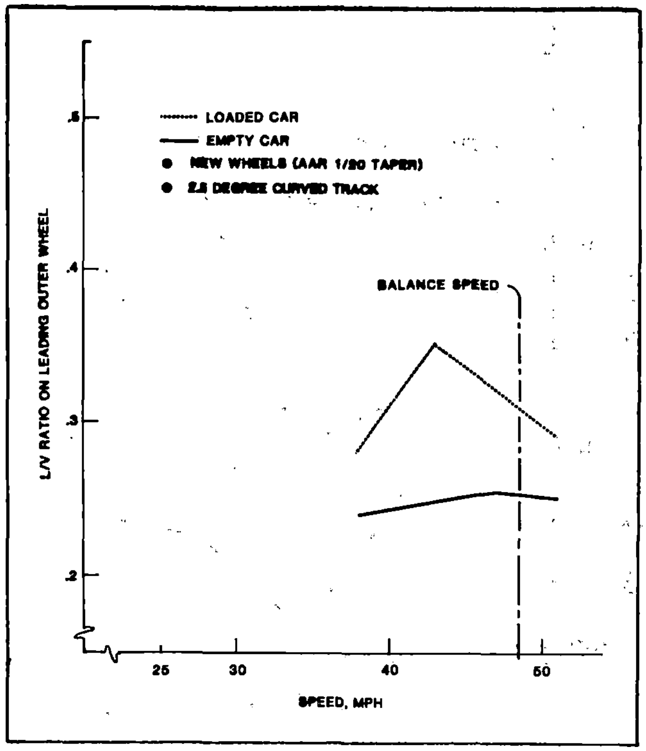


FIGURE 5-48. L/V RATIO ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 2.5^{0} CURVED TRACK

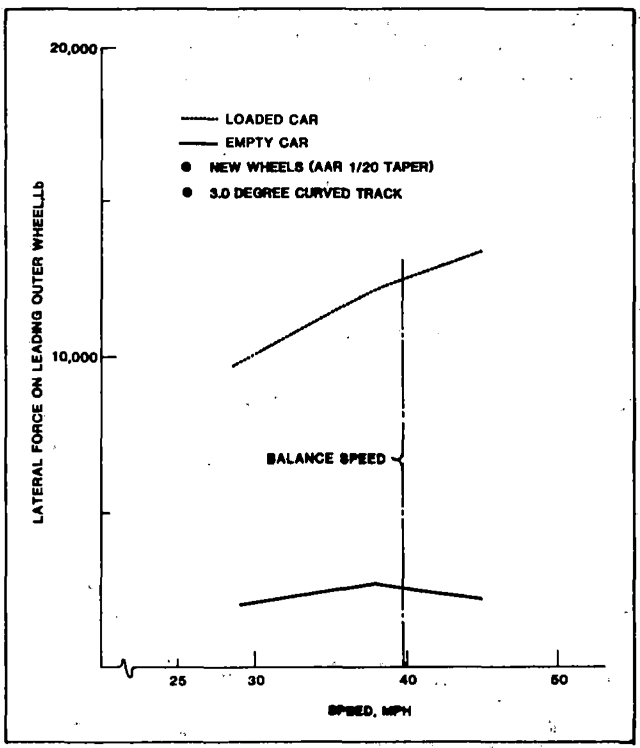


FIGURE 5-49. AVERAGE LATERAL FORCE ON LEADING OUTER WHEEL VERSUS SPEED - 100- TON TRUCKS WITH BOX TYPE CARS, 3.0^{0} CURVED TRACK

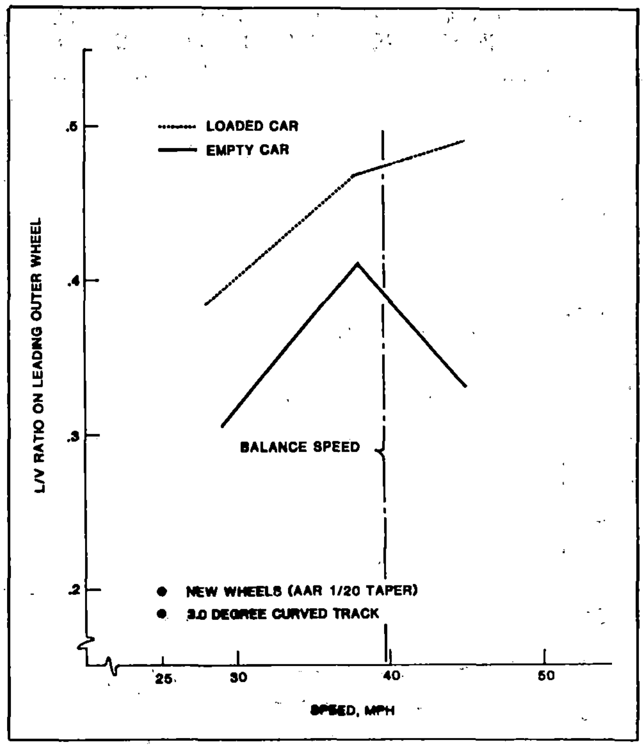


FIGURE 5-50. L/V RATIO ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 3.0^{0} CURVED TRACK

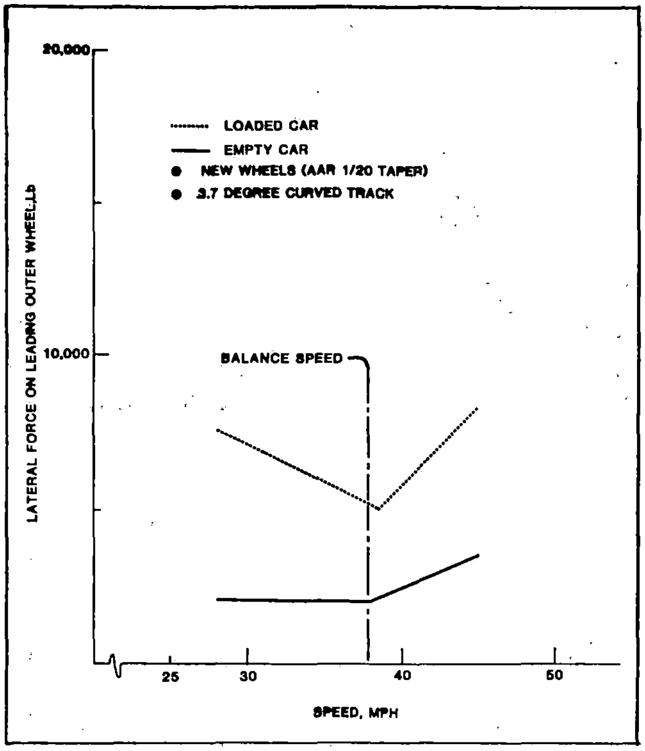


FIGURE 5-51. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 3.7° CURVED TRACK

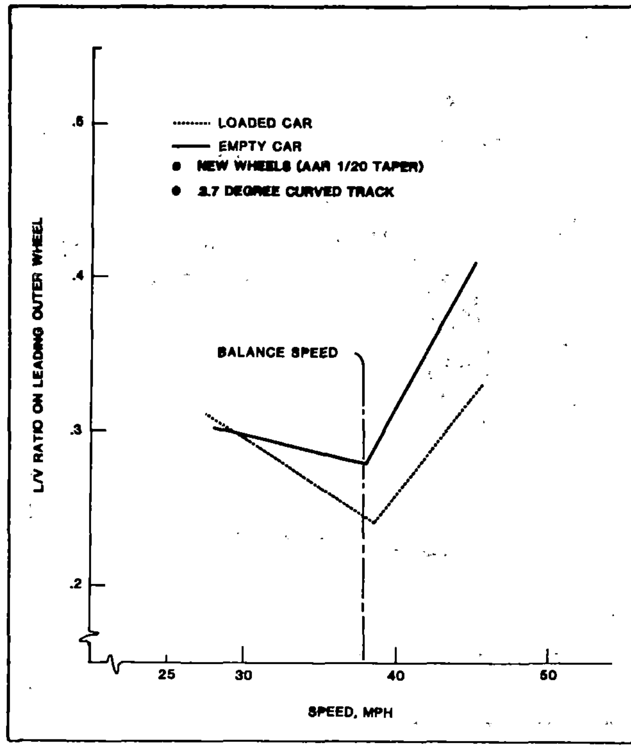


FIGURE 5-52. L/V RATIO ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 3.7° CURVED TRACK

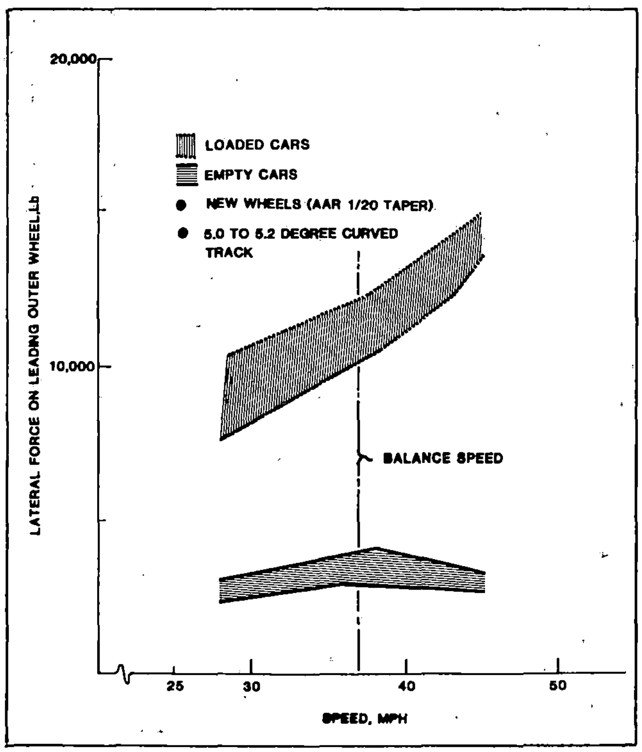


FIGURE 5-53. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS SPEED - 100 TON TRUCKS WITH BOX TYPE CARS, 5.0° to 5.2° CURVED TRACK

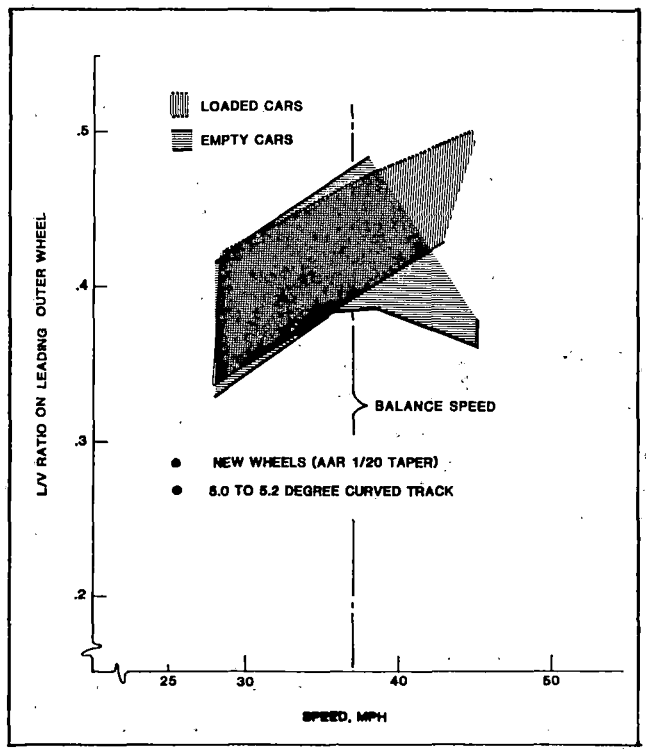


FIGURE 5-54. L/V RATIO ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 5.0° to 5.2° CURVED TRACK

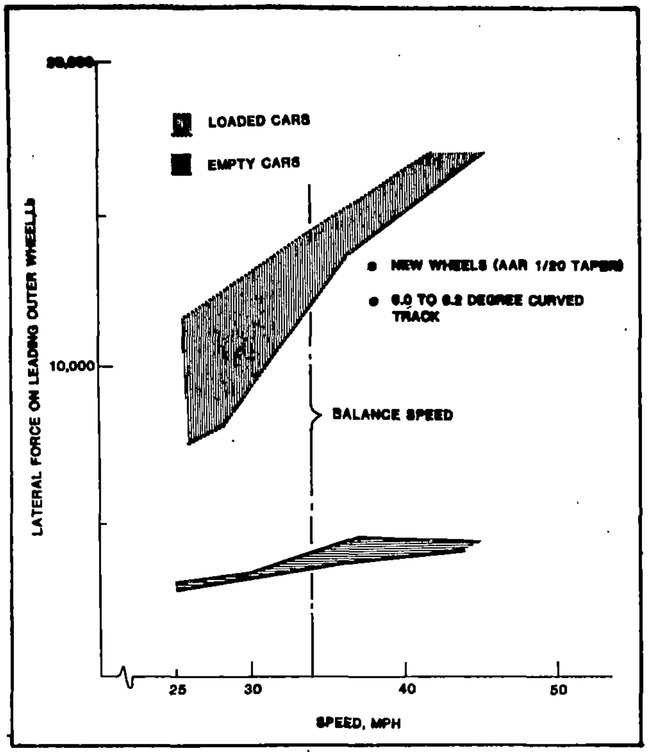


FIGURE 5-55. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 6.0^{0} to 6.2^{0} CURVED TRACK

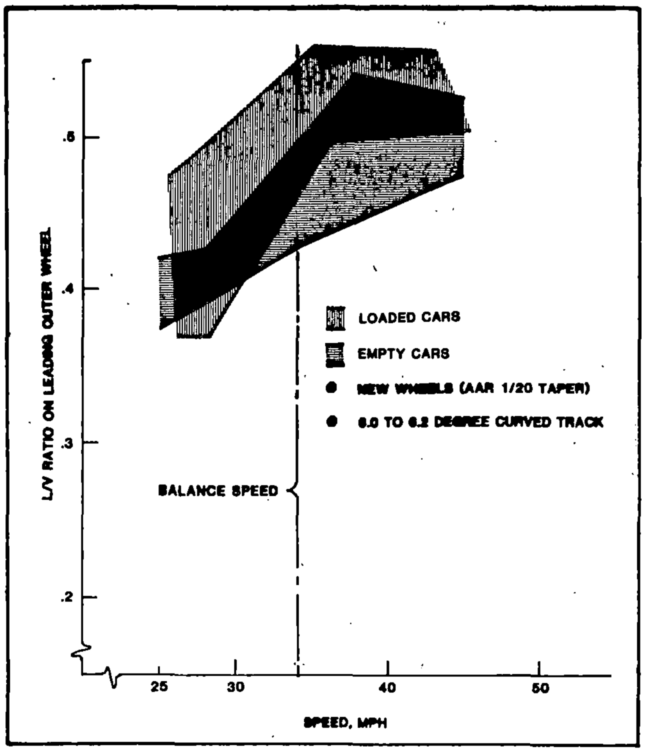


FIGURE 5-56. L/V RATIO ON LEADING OUTER WHEEL VERSUS SPEED - 100-TON TRUCKS WITH BOX TYPE CARS, 6.0^{0} to 6.2^{0} CURVED TRACK

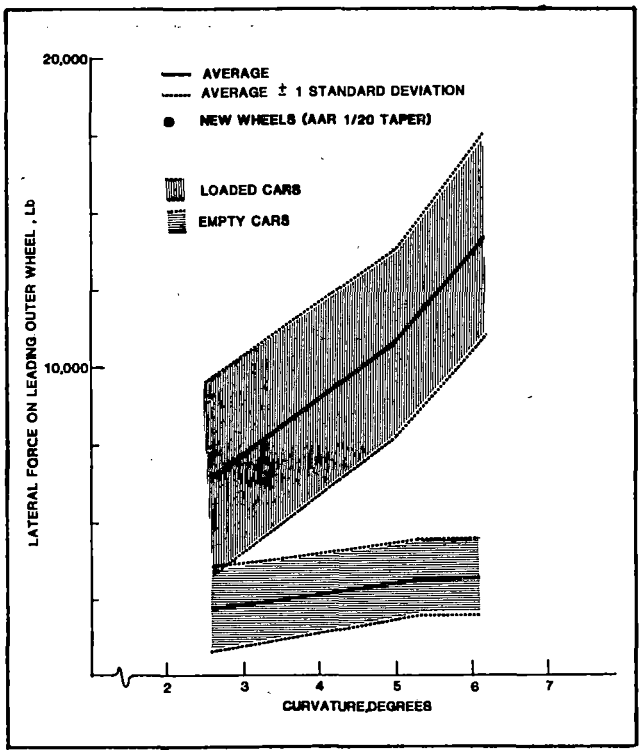


FIGURE 5-57. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS DEGREE OF CURVATURE NEAR BALANCE SPEED - 100-TON TRUCKS WITH BOX TYPE CARS

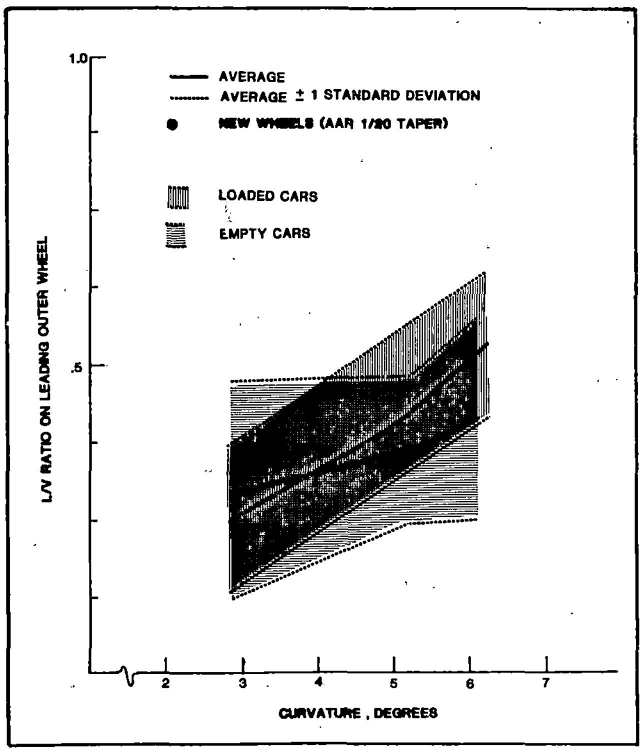


FIGURE 5-58. L/V RATIO ON LEADING OUTER WHEEL VERSUS DEGREE OF CURVATURE NEAR BALANCE SPEED - 100-TON TRUCKS WITH BOX TYPE CARS

APPENDIX A

PHASE I TEST EQUIPMENT AND CONDITIONS

INTRODUCTION

This appendix contains a brief summary of the equipment tested during TDOP Phase I to generate field test data. A complete description can be found the Phase I Final Report (reference 4).

TEST TRAIN

The test train was made up of a locomotive, the SP-250 instrument car, the test car, and a caboose, in that order. This consist reflects the intent to study freight car truck performance with the test car approximating a free body with no extraneously imposed longitudinal forces.

EQUIPMENT TESTED

Phase I of the Truck Design Optimization Project consisted of tests on five different types of cars equipped with different combinations of truck types and wheels.

The cars tested were a 70-ton mechanical refrigerator car, a 70-ton boxcar, a long, low-level "stac-pac" flatcar, a 100-ton boxcar, and a 100-ton covered hopper car. The data on these cars are given in Table A-1.

Trucks tested were 70-ton ASF Ride Control trucks, 70-ton Barber S-2-C trucks, 70-ton ASF low-level Ride Control trucks, and 100-ton Barber S-2-C trucks. The data on these trucks are given in Table A-2.

Wheel profiles used in the Phase I test program, data from which were used in quantifying performance characterizations under the Phase II effort, are listed below.

- CM-33 1/20 taper profile wheels on the 70-ton ASF Ride Control and Barber S-2-C trucks
- CM-33 worn profile wheels on the 70-ton ASF Ride Control trucks
- CJ-36 1/20 taper profile wheels on the 100-ton ASF Ride Control and Barber S-2-C trucks
- CD-28 1/20 taper profile wheels and CB-28 worn profile wheels on the 70-ton ASF Ride Control low-level trucks
- CM-33 TDOP cylindrical profile wheels on the 70-ton ASF Ride Control trucks
- CJ-36 TDOP cylindrical profile wheels on the 100-ton Barber S-2-C trucks

TEST TRACK

High-speed jointed rail test track consisted of a 7.8-mile westbound section of track between Suisun-Fairfield and Bahia (MP 48.5 to 40.7). This track has alternate staggered rail joints of 39-foot, 132-pound per yard rail.

Medium-speed jointed rail test track consisted of a 5-mile section of the Schellville branch beginning near Cordelia and ending near Suisun-Fairfield. This is a section of alternately staggered joints of 39-foot, 132-pound per yard rail (second-hand rail within serviceable limits).

A short section of the Schellville branch near Lombard was selected for distortion by instituting 0.75-inch cross level differences at the rail joints.

The track geometry cars were used to measure and record track characteristics at the high-speed and medium-speed test sites. The track geometry measured included profile, alignment, gage, cross level and curvature.

TEST MATRICES

The test matrices for high-speed and medium-speed jointed track used during Phase II in quantifying the performance characteristics of Type I trucks are given in Tables A-3 and A-4. For the shimmed track test, a loaded 70-ton mechanical refrigerator car equipped with a 70-ton ASF Ride Control truck, and a loaded 100-ton boxcar equipped with a 100-ton Barber S-2-C were used. The two test trucks were equipped with cylindrical wheels.

INSTRUMENTATION

The various test cars were instrumented to obtain information for quantifying ride quality, and for measuring track input, track energy transmission through the truck, and movement between truck components.

These objectives were accomplished by application of displacement transducers, accelerometers, and force transducers at strategic locations on the trucks. To obtain information on reaction of the carbody, accelerometers were placed at optimum locations to record body movement.

Truck-mounted instrumentation was heavily concentrated on the B-end truck, which was the leading truck in the direction of motion during all tests. A lesser amount of instrumentation was on the trailing truck.

TABLE A-1. CARBODY CHARACTERISTICS

	70-Ton Capacity Mechanical Refrigerator Car	70-Ton Capacity General Service Boxcar	70-Ton Capacity Long Low-Level Flatcar	100-Ton Capacity Auto-Parts Boxcar	100-Ton Capacity Covered Hopper Car
Light Weight, lb	89,100	61,200	56,300	87,300	64,500
Capacity, lb	130,900	154,000	122,000	174,000	197,500
Length Over Pulling Face of Coupler, ft	63.70	55.38	93.67	68.25	54.29
Truck Centers, ft	45.72	40.00	64.00	46.25	40.83
Car Wheel Base, ft	51.39	46.83	69.08	52.08	46.25
Overhang, ft	9.00	7.29	14.83	11.00	7.29
Center of Gravity-Loaded, ft	7.33	7.03	7.17	7.83	7.03
Center of Gravity-Empty, ft	5.55	4.58	1.97	5.17	4.58
Centerplate Diameter, ft	1.17	1.17	1.17	1.33	1.25

TABLE A-2. TRUCK CHARACTERISTICS

	70-Ton ASF Ride Control Truck	70-Ton Barber S-2-C Truck	70-Ton ASF Low-Level Truck	100-Ton ASF Ride Control Truck	100-Ton Barber S-2-C Truck
Wheel Base, ft	5.67	5.67	5.08	5.83	5.83
Wheel Diameter, ft	2.75	2.75	2.33	3.00	3.00
Bolster Centerplate Diameter, ft	1.15	1.17	1.17	1.25	1.33
Centerplate Height, ft	2.15	2.15	1.68	2.07	2.15
Weight, lb	9,080	9,100	7,600	10,540	10,560
Gross Rail Load, lb	220,000	220,000	179,000	263,000	263,000
Vertical Spring Rate (Per Car), lb/in	94,466	89,653	97,450	108,333	109,367
Lateral Spring Rate (Per Spring Nest), lb/in	4,665(at 9.47")* 7,795(at 7.56")*	3,470(at 9.47") 9,080(at 7.56")	4,755(at 9.06") 12,015(at 8.31")	3,655(at 9.47") 9,560(at 7.56")	2,705(at 9.47") 10,285(at 7.56")
Friction Snubber Column Load, lb	3,140	Variable (Load-Dependent)	3,110	Variable (Load-Dependent)	4,510

*Spring Nest Height

TABLE A-3. HIGH-SPEED JOINTED TRACK TEST MATRIX

TDOP PHASE I TEST MATRIX USED DURING PHASE II ANALYSIS

⊡ **TEST DATA AVAILABLE**

☐ **NO TEST CONDUCTED**

Truck	Carbody	Empty		Loaded	
		Wheel Profile			
		New AAR 1/20	Worn	New AAR 1/20	Worn
70-Ton ASF Ride Control	Refrigerator Car	●	●	●	●
70-Ton Barber S-2-C	Refrigerator Car	●		●	
	70-Ton Boxcar	●		●	
70-Ton Low Level ASF Ride Control	89-ft Flatcar	●	●	●	
100-Ton Barber S-2-C	100-Ton Boxcar	●		●	
100-Ton ASF Ride Control	100-Ton Covered Hopper Car	●		●	

TABLE A-4. MEDIUM-SPEED JOINTED TRACK TEST MATRIX

TDOP PHASE I TEST MATRIX
USED DURING PHASE II ANALYSIS

⊡ **TEST DATA AVAILABLE**

☐ **NO TEST CONDUCTED**

Truck	Carbody	Empty		Loaded	
		Wheel Profile			
		New AAR 1/20	Worn	New AAR 1/20	Worn
70-Ton ASF Ride Control	Refrigerator Car	●	●	●	●
70-Ton Barber S-2-C	Refrigerator Car	●		●	
	70-Ton Boxcar	●		●	
70-Ton Low Level ASF Ride Control	89-ft Flatcar	●		●	
100-Ton Barber S-2-C	100-Ton Boxcar	●		●	
100-Ton ASF Ride Control	100-Ton Covered Hopper Car	●		●	

APPENDIX B

PHASE II TEST EQUIPMENT AND CONDITIONS

INTRODUCTION

This appendix is a summarized description of the equipment tested, test conditions, and procedures used in generating the field test data acquired during Phase II in the characterization of Type I truck performance. Complete descriptions of Phase II field tests are documented elsewhere (references 5 through 8).

TEST OVERVIEW

The 100-ton ASF Ride Control truck with new AAR standard 1/20 taper wheel profiles was used in the Phase II tests. This truck was the identical one used in the TDOP Phase I test program and in the TDOP Phase II Wear Data Collection Program. The carbody used for this program was a 100-ton open hopper car in both an unloaded and loaded configuration.

Instrumentation for the test program consisted of 92 data channels. Forty-nine of the channels were used to obtain data for the computation of lateral and vertical (L/V) forces at the wheel/rail interface. The basic approach taken to measuring L/V forces was the strain-gaged axle technique. The vertical forces at the bearing adapter were measured using the strain-gaged bearing adapters from TDOP Phase I. Forty-three of the 92 channels of data provided measurements of rigid body car motions, longitudinal coupler forces, truck/carbody relative displacements, and angle of attack.

Curving tests were conducted over curved track on Union Pacific's mainline south of Las Vegas, Nevada. Angle of attack, lateral forces, and vertical forces at the wheel/rail interface were used to quantify the performance indices for the curve negotiation regime.

TEST TRAIN

A standard consist, made up of a locomotive, the Union Pacific car 210, two buffer cars, the test car, and a caboose, was established for all test runs and maintained throughout the test program.

The buffers were open hopper cars. Prior to the start of testing, each buffer car was loaded. To provide for easier interchange of test cars, the instrumented coupler was placed on the test car end of each buffer car. However, the coupler angle measurement was made on the test car.

EQUIPMENT TESTED

The carbody type used for this test program was a 100-ton open hopper car rented through UP. The lading for the loaded hopper was coal. The 100-ton hopper car was chosen because it is representative of the higher capacity cars being placed into service today. Further, the car was readily available, and facilities existed for loading it.

The Type I truck selected for testing was the 100-ton ASF Ride Control truck. The truck set was temporarily assigned to the Type I truck test program from the Wear Data Collection Program and was returned to that program at the completion of Type I truck testing. A detailed description of these trucks is contained in reference 5, along with a set of engineering parameters.

One set of new wheel profiles was used for the test program. The new wheel profiles were AAR standard 1/20 taper profiles.

The UP Mobile Laboratory Car 210 was used as the instrumentation car for all testing on Type I trucks.

TEST ZONES

The test sites used for the Type I truck testing were mainline and yard tracks of the UP's South Central District, California Division. Three test zones were selected for Type I truck testing and are described in Table B-1. Test zone 1 consisted of mainline track with 1.5° to 6° of curvature. Test zone 2 provided a section of tangent, jointed track over which high-speed (up to 79 mph) tests could be conducted. Test zone 3 was selected because it supplies a section of yard track over which load equalization tests could be conducted. An Automatic Location Detector (ALD) system was placed in each of the test zones, and track geometry measurements were obtained.

TEST VARIABLES

To bound the testing planned on Type I trucks, it was necessary to define the constant and variable quantities. Table B-2 presents the list of planned test variables. All other truck parameters were set to a "nominal" condition for that specific truck type in regards to spring group, gib clearance, snubbing, center plate, and side bearing for all test runs. This condition was maintained, as nearly the same as possible, throughout the test program. The exact condition of these parameters was noted in the tape header for each test.

In general, tests were run on days when track conditions were clear and dry. Other test conditions such as humidity, temperature, and wind, were noted on the tape header for the given test; any extreme conditions were avoided. The test vehicle was oriented so the B-end (instrumented truck) was always leading. Use of the available transducers and signal conditioning resulted in a cut-off frequency of 20 to 100 Hz.

INSTRUMENTATION

The primary objective of Phase II instrumentation was to obtain measurements required to calculate the forces at the wheel/rail interface. These critical measurements were not obtained during Phase I. In addition to the instrumentation required to measure the wheel/rail interface forces, transducers were installed to measure truck and carbody relative motion, rigid body car motion, coupler forces, and wheel/rail angle of attack. All instrumentation was installed prior to the start of testing. Then the 92 data channels were recorded for all test runs. Each measurement was assigned an alphanumeric identifier which was carried throughout the test program for that channel. After the system was installed, each measurement would have an A/D channel assignment number. At that time, the sequence number was dropped and the measurement list re-sequenced in order of A/D channel assignment. The location of the instrumentation is shown in Figures B-1 through B-9.

TABLE B-1. TYPE I TRUCK TEST ZONES

TEST ZONE	SITE DESIGNATION	DESCRIPTION
1	LOCATION	ARDEN TO SLOAN, NEVADA
	MILEPOSTS	321.5 TO 314 (7.5 MILES)
	TRACK TYPE	CLASS 4 - CURVED
	RAIL TYPE	133 LB JOINTED
	SPEED LIMIT	40 MPH
2	LOCATION	ARDEN TO BOULDER JUNCTION
	MILEPOSTS	321.48 to 326.5 (5.0 MILES)
	TRACK TYPE	CLASS 4 - TANGENT
	RAIL TYPE	133 LB JOINTED
	SPEED LIMIT	79 MPH
3	LOCATION	LAS VEGAS, NEVADA
	MILEPOSTS	YARD LIMITS (0.22 MILES)
	TRACK TYPE	12 AND 16^{o} CURVES
	RAIL TYPE	JOINTED
	SPEED LIMIT	10 MPH

TABLE B-2. PLANNED TEST VARIABLES

CARBODY TYPE

100-TON HOPPER CAR

TRUCK TYPES

100-TON ASF RIDE CONTROL

TRACK TYPES

YARD QUALITY CLASS 1
MAINLINE CURVED
MAINLINE CURVED

LADING CONDITIONS

EMPTY
LOADED

WHEEL PROFILES

NEW (STANDARD AAR PROFILE)

TEST PROCEDURE

Curving

The curving tests on the Type I trucks were run over test zone 1 (see Figures B-10 and B-11). This zone consisted of mainline, jointed track from the Arden junction to Sloan. The test zone started at MP 321.48 and continued to MP 314.5. All tests were made with the B-end truck in the leading position. The curving tests consisted of three passes through the test zone run in the uphill direction. For the loaded configurations, one curving test was conducted in the downhill direction. The downhill direction ran from MP 314.5 to MP 321.48.

The curves in the planned test zone varied from 1.5 to 6.2 degrees and had equilibrium speeds ranging from 34 to 48 mph. The characteristics for each of the curves are listed in Figure B-12, along with a plan view of the curved section of track. The speed profile shown in Table B-3 was used for running the curving tests. To control the speed of the consist, the forward observer in the locomotive used a display of car speed and milepost from the car 210 instrumentation to direct the train engineer. Speeds were maintained within ± 2 mph of those specified in Table B-3.

The first pass through the test zone was at speeds of 10 mph less than the equilibrium speed, the second pass was made at the speed profile in Table B-3, and the third pass at speeds 10 mph greater than those shown in the table. The pass through the test zone in the downhill direction was conducted at the equilibrium speed profile in Table B-3.

Load Equalization

The test runs for load equalization were conducted over class 1 track (zone 3) in the Union Pacific yard in Las Vegas. The test zone consisted of .22 miles of yard track with a 15.8° and 16° curve (see Figure B-13 for a profile map of zone 3). The test run consisted of one pass over the zone from point A to point B and back to point A. The test consist was moving at 10 mph as the train passed the start of the test at point A. It maintained this speed through the zone. After the test car (i.e., 100-ton hopper) was well past point B, the train stopped and immediately backed up through the zone from point B to point A at a constant speed of 10 mph. Data were collected and recorded throughout the entire test sequence.

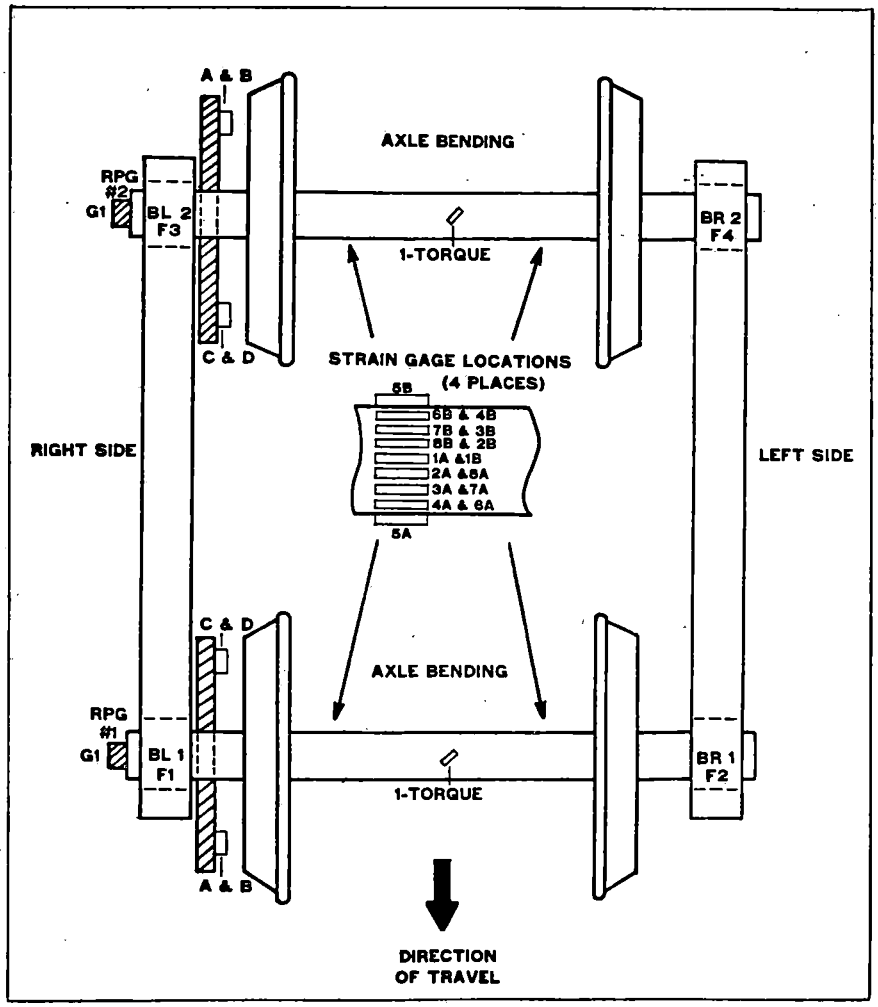


FIGURE B-1. WHEEL/RAIL MEASUREMENT INSTRUMENTATION

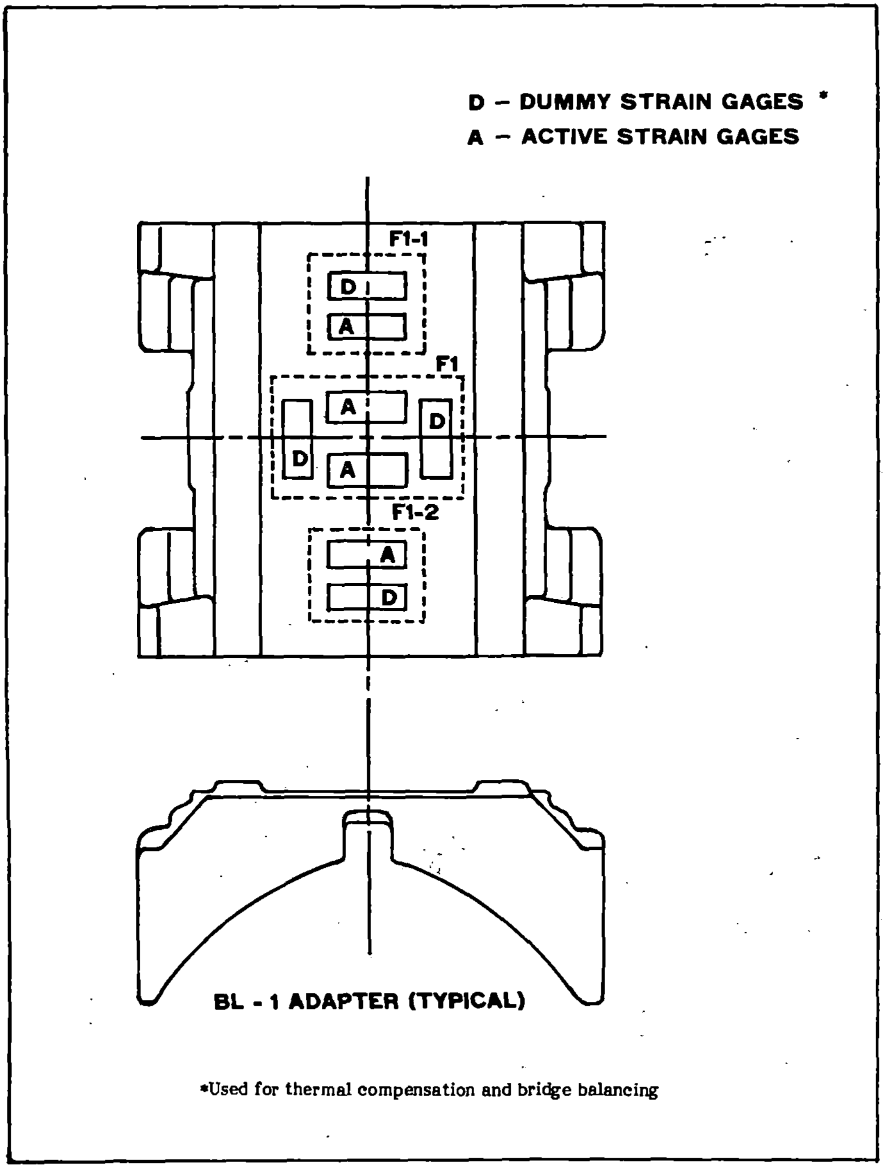


FIGURE B-2. FORCE TRANSDUCER ON BEARING ADAPTER

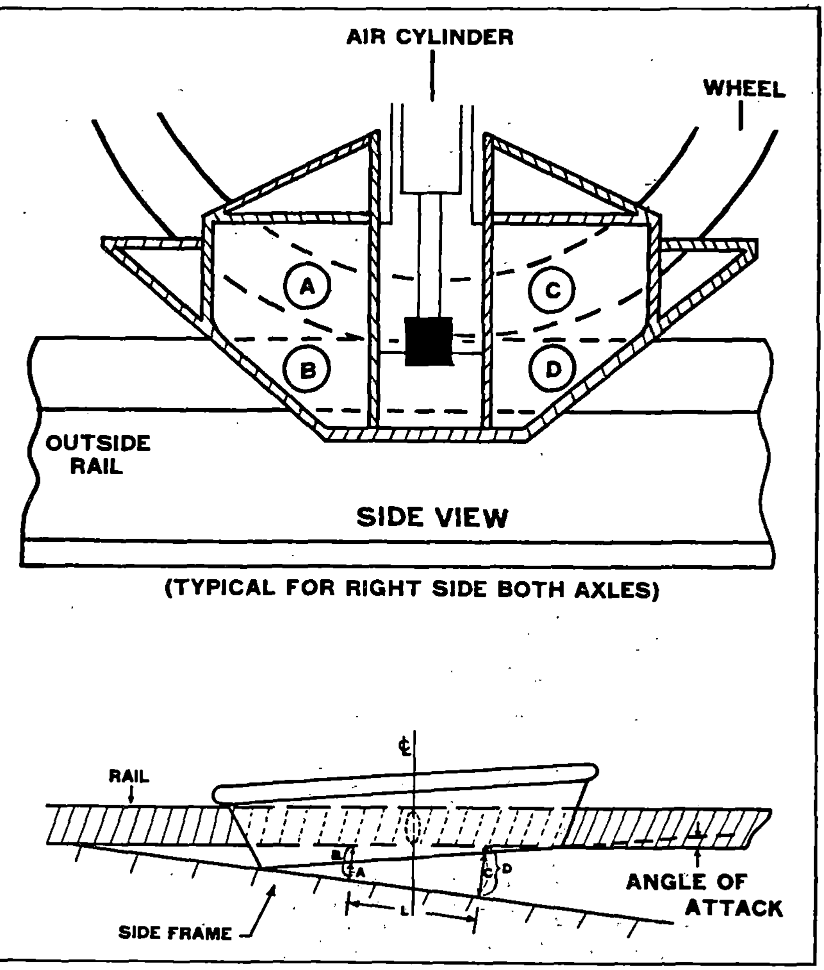


FIGURE B-3. WHEEL/RAIL POSITION MEASUREMENT

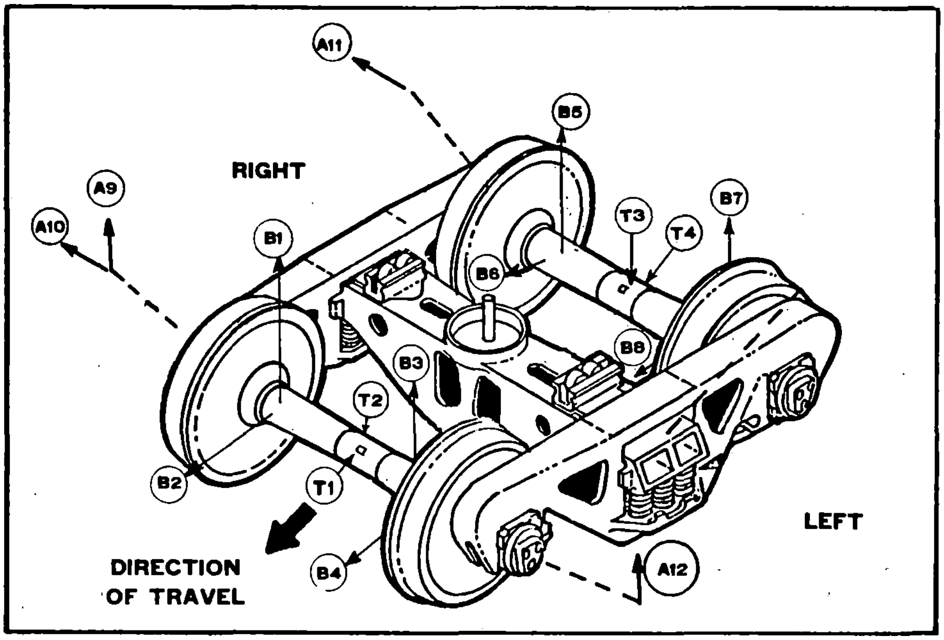


FIGURE B-4. TRUCK INSTRUMENTATION LOCATIONS, B-END

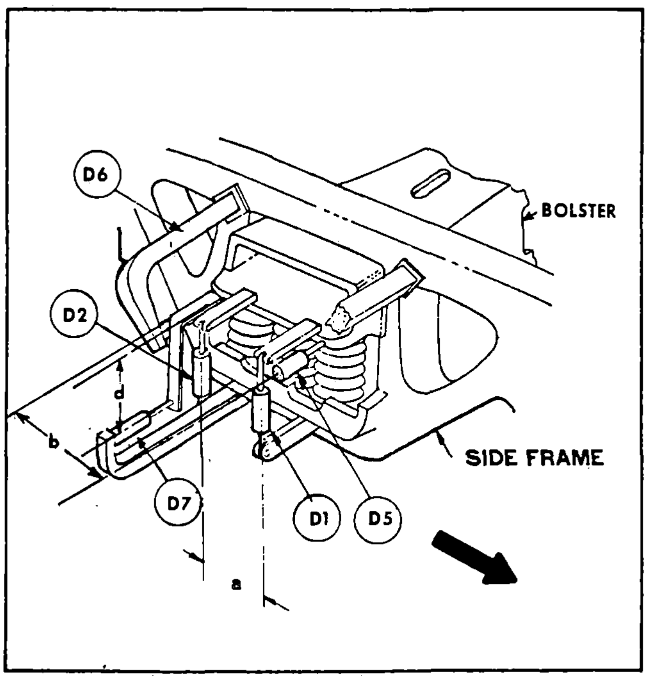


FIGURE B-5. BOLSTER INSTRUMENTATION, RIGHT SIDE

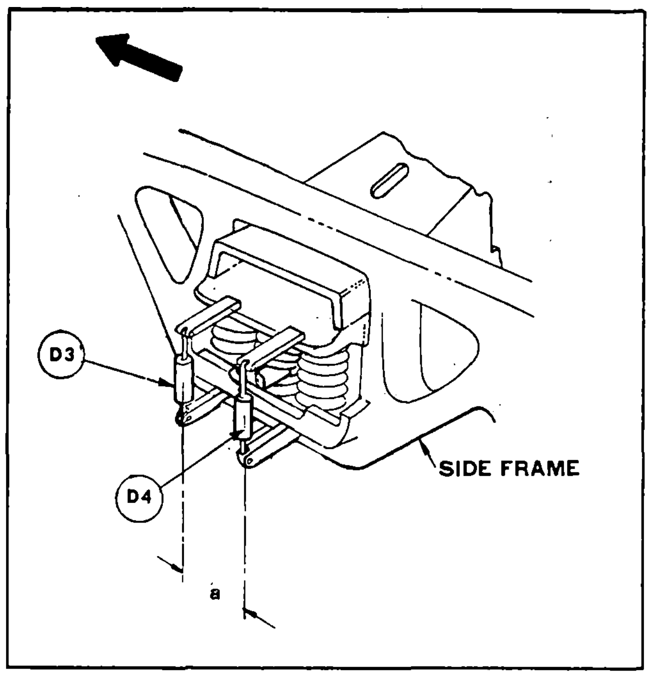


FIGURE B-6. BOLSTER INSTRUMENTATION, LEFT SIDE

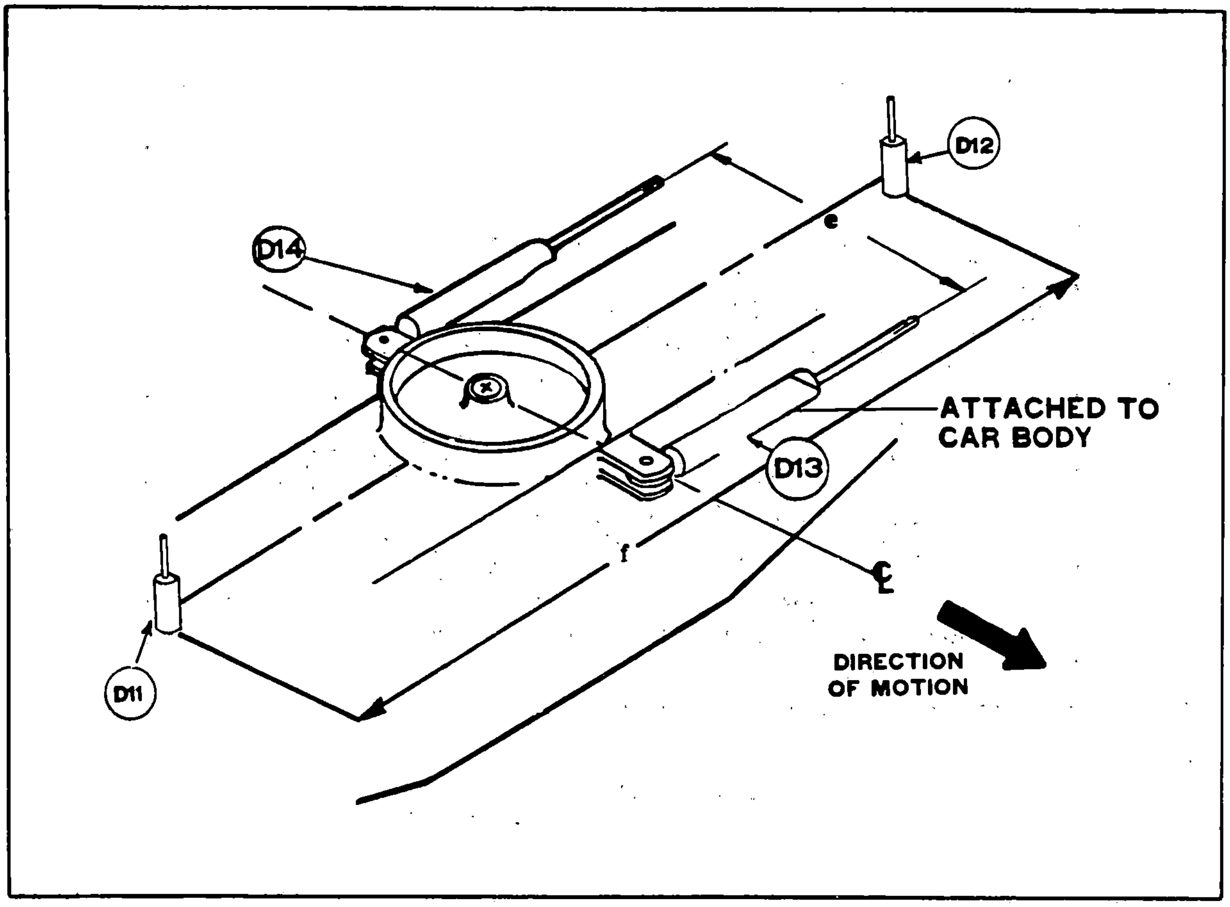


FIGURE B-7. CENTER PLATE INSTRUMENTATION

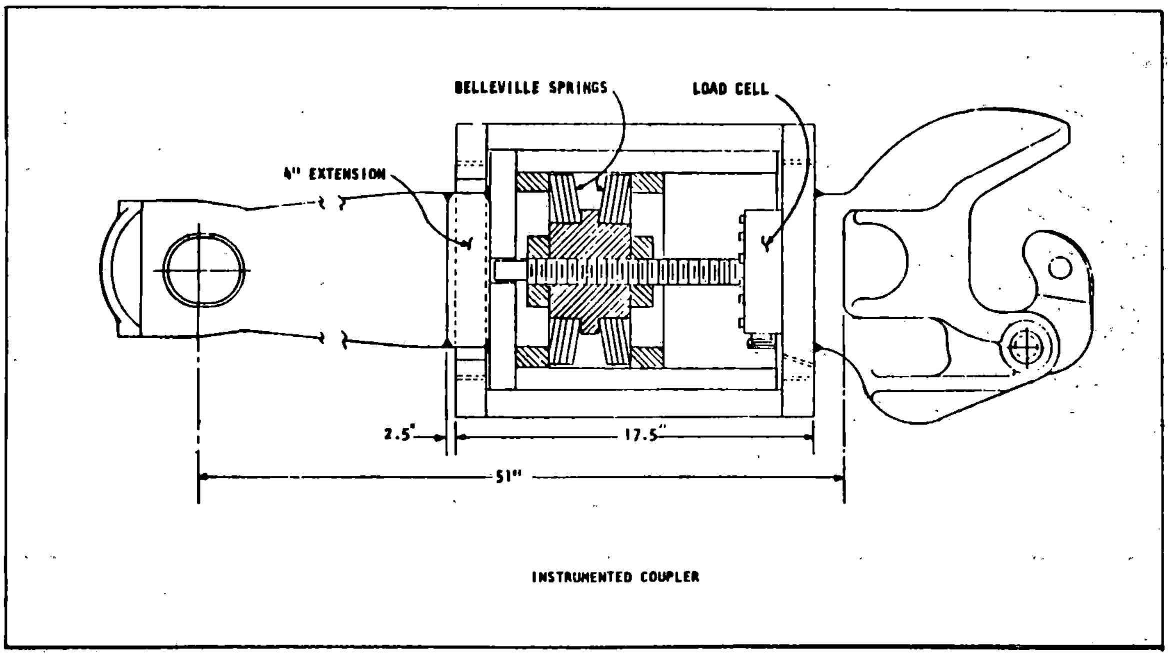


FIGURE B-8. INSTRUMENTED COUPLER SCHEMATIC

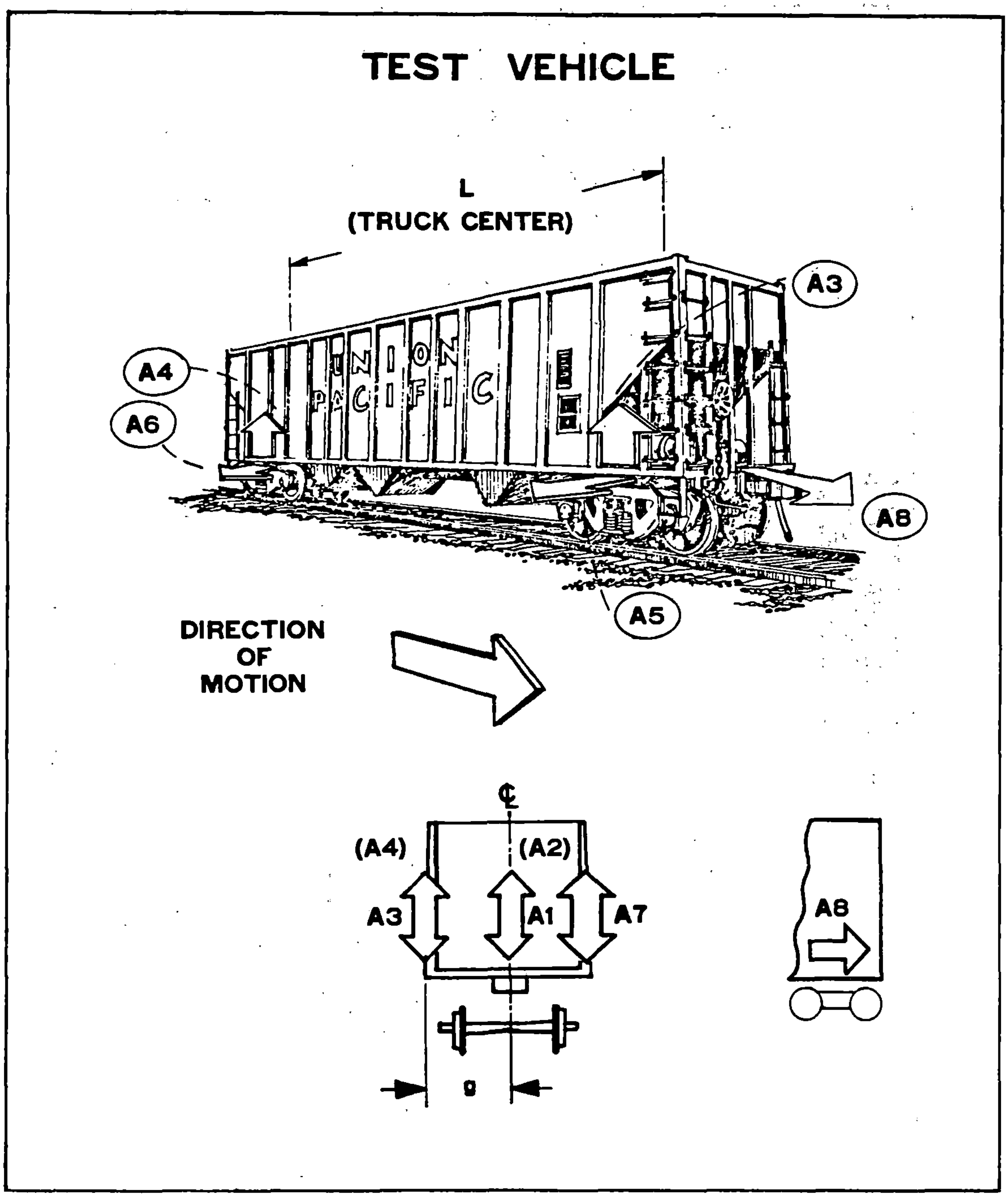


FIGURE B-9. CARBODY INSTRUMENTATION

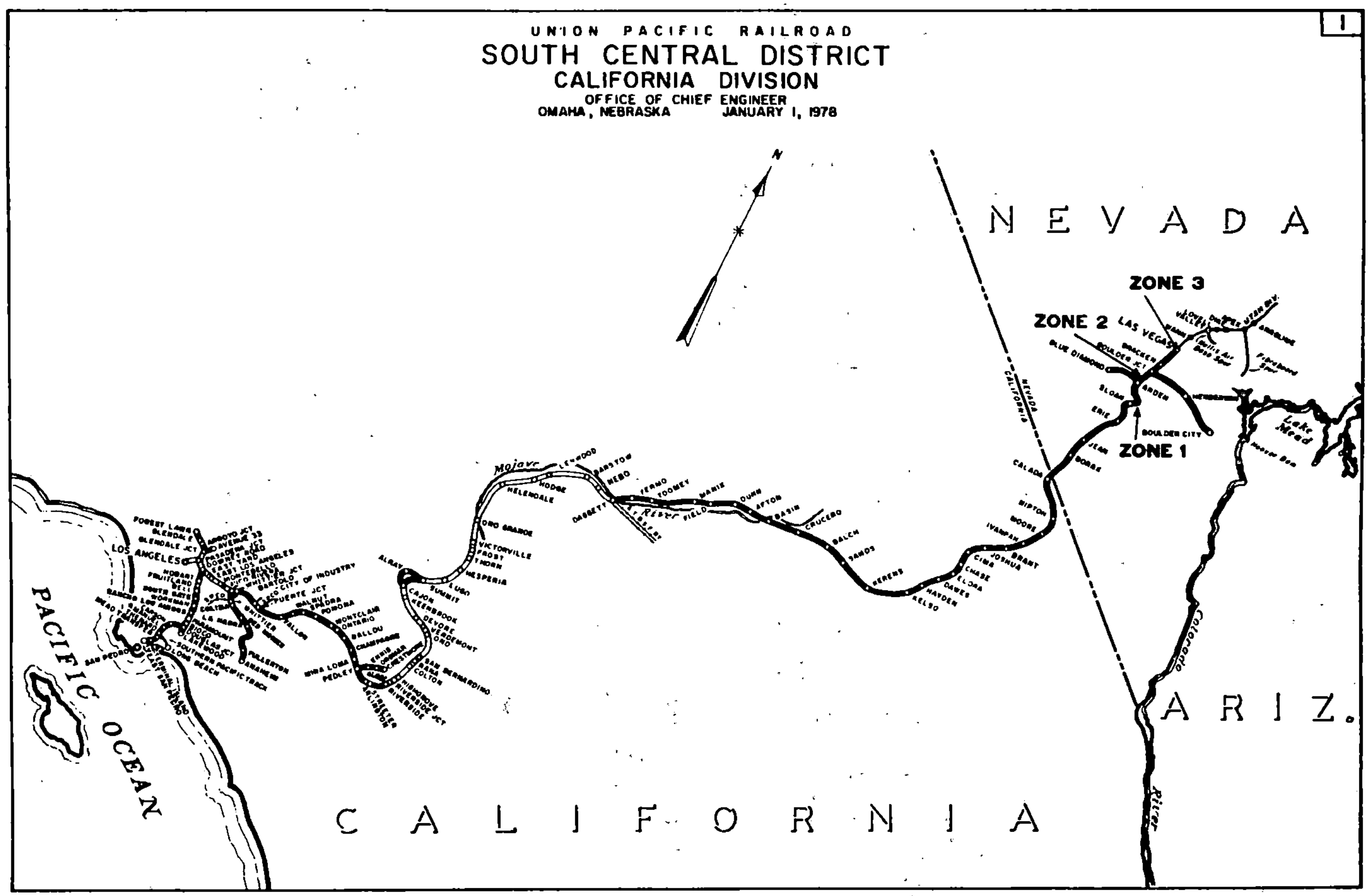


FIGURE B-10. TEST ZONE LOCATIONS

CALIFORNIA DIVISION MAIN LINE

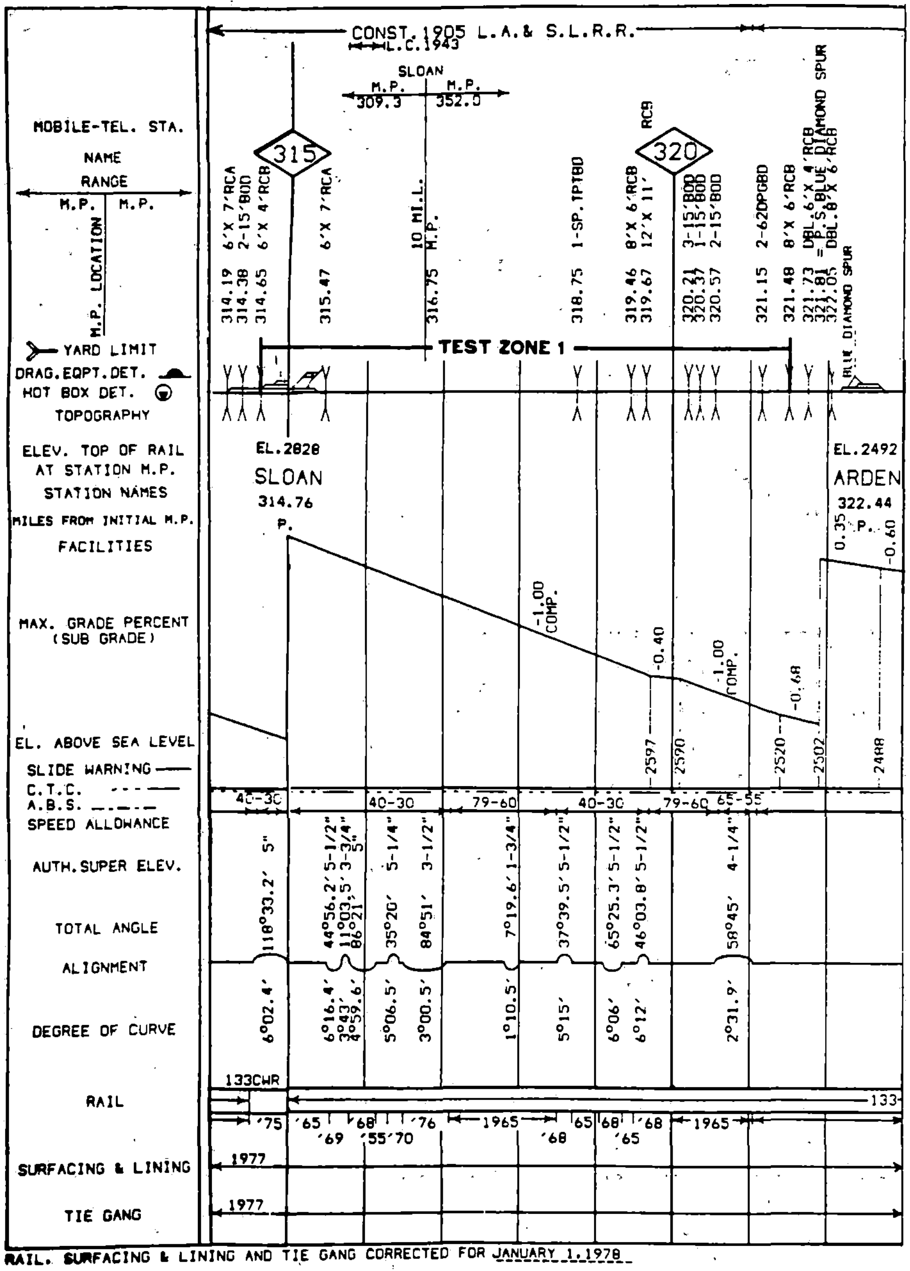
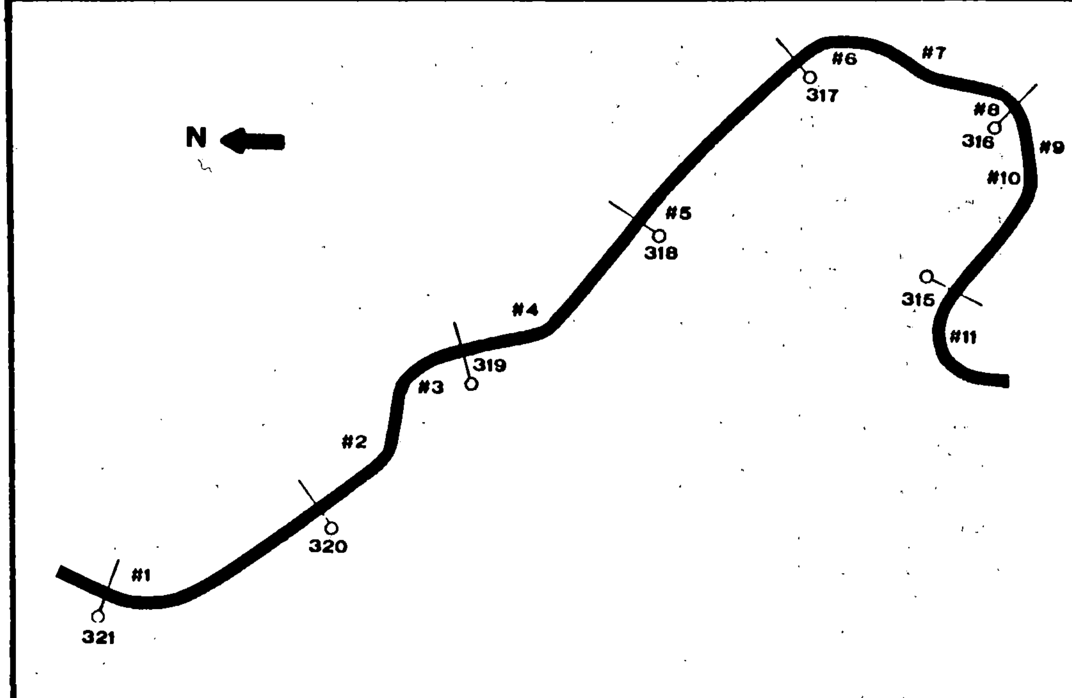


FIGURE B-11. TRACK PROFILE - TEST ZONE 1



Curve No.	Degree of Curve	Direction Curve	Length (ft.)	Super Elevation (in.)	Start MP	Stop MP	Equilibrium Speed
1	2.5	Left hand	2736	4.09	321.1	320.6	48.3
2	6.2	Left hand	1080	4.98	319.7	319.5	33.9
3	6.1	Right hand	1386	4.90	319.4	319.1	33.9
4	5.2	Left hand	1136	5.08	318.7	318.5	37.4
5	1.1	Right hand	934	1.41	318.0	317.8	42.8
6	3.0	Right hand	3118	3.32	317.1	316.5	39.8
7	5.1	Left hand	961	4.88	316.5	316.3	37.0
8	5.0	Right hand	2070	4.79	316.2	315.8	37.0
9	3.7	Left hand	492	3.72	315.8	315.7	37.9
10	6.2	Right hand	1037	4.94	315.7	315.5	33.7
11	6.1	Left hand	2420	4.92	315.0	314.6	34.1

FIGURE B-12. CURVE PROFILES - TEST ZONE 1

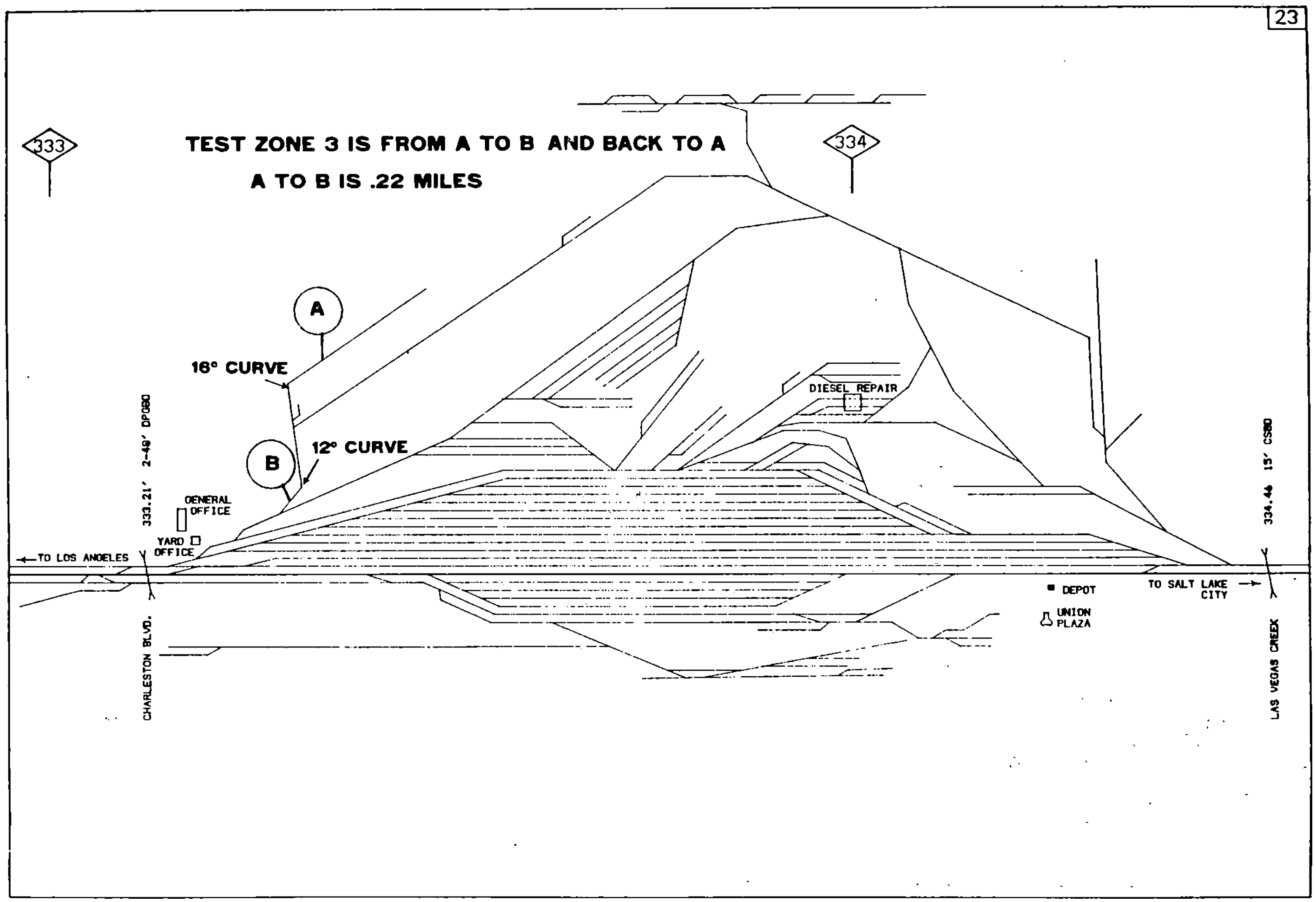


FIGURE B-13. LAS VEGAS YARD - TEST ZONE 3

TABLE B-3. EQUILIBRIUM SPEED PROFILE

MP	Speed (mph)
321.48 - 320.6	48
320.6 - 319.7	48 to 35
319.7 - 319.1	35
319.1 - 318.7	35 to 38
318.7 - 315.8	38
315.8 - 315.7	38 to 34
315.7 - 314.5	34
Tolerance on Speed	±2 mph

Pass Number	Speed	Duration
1	10 mph below profile	15.1 min.
2	At profile speed	11.1 min.
3	7 mph above profile	9.4 min.

APPENDIX C

ANGLE OF ATTACK MEASUREMENT

INTRODUCTION

Measurement of the wheel/rail angle of attack was one of the goals of the Phase II field test program. Considerable effort was expended in developing a vehicle-borne angle of attack measurement system, believed to be the first of its kind to be used in an extensive field test program.

The field test data acquired through this instrumentation package exhibit considerable scatter and it is difficult to discern characteristic trends of truck performance from the data. Although simplifications and theoretical assumptions can be used in the interpretation of the data, it is not considered desirable to use theoretical reasoning as the sole basis in arriving at "characterization" of truck performance. Therefore, the results from the test data are presented here in the interest of documentation, which may be useful in subsequent efforts.

INSTRUMENTATION SYSTEM

Angle of attack data are provided by non-contacting position sensors mounted on the right side of each axle of the leading truck. Two sensors measure the relative sideframe to wheel displacement, and two others measure the relative sideframe to rail displacement. The difference between the two sensors gives the relative angle; the difference between the sideframe to wheel and the sideframe to rail angles results in the angle of attack (see Figures C-1 and C-2). The sensors are of the eddy current type, which result in a signal based on the average distance from the sensor to a surface.

FIELD IMPLEMENTATION

Irregularities of the surface make it difficult to obtain a consistent reference. Also, the car must be stopped to make the measurement; this could set the truck up in unnatural operating positons.

Examination of the angle of attack data on several segments of track shows that it generally does not vary by more than ± 4 minutes on tangent track. Even though a particular truck can take a set on tangent track, it is believed that the average angle of attack on tangent track provides a better zero reference than the static calibration. Therefore, angle of attack data are referenced to tangent track. Although not providing an absolute reference, this method does have the advantage of removing the bias of a particular truck. The sensitivity calibration is believed to be more accurate than the zero calibration.

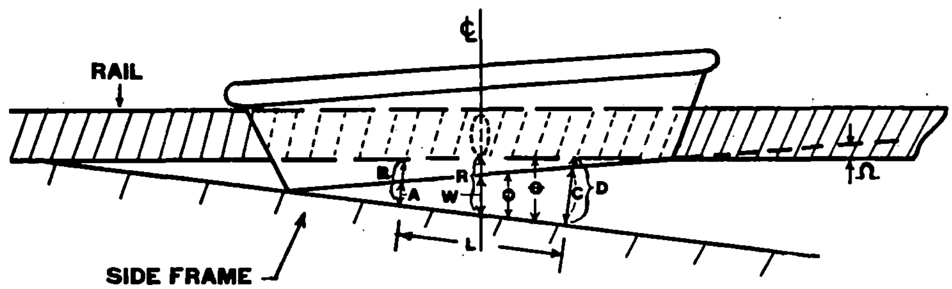
FIELD TEST DATA

The angle of attack data were analyzed using time history and statistical measurements (average and standard deviation of the signal). Figures C-3 and C-4 show the absolute average of the angle of attack as a function of speed, while Figures C-5 and C-6 show the absolute average angle of attack versus the degree of curvature near balance speed. Data from both loaded cars and empty cars are presented. After examining the time domain and the statistical properties of the angle of attack, the following remarks can be made:

a. Considerable scatter in the results has been noted, and no clear-cut trends can be established. This is true both spatially, in the sense that there is considerable variation along a fixed radius curve, and in terms of the averages between curves of the same radius. This scatter may be due to one or more reasons, namely, local variations in the track curvature; differences in truck set; large dependence on the track history (i.e., direction and preceding curve); and errors in the collected data.

b. The empty car data tend to have more scatter than the loaded car data.

c. There is apparent difference between left-hand and right-hand curves. This difference may be caused by nonlinearity of the eddy current transducers due to overranging.



Θ = RAIL TO SIDEFRAME $\measuredangle \cong \frac{C-B}{L}$ RADIANS $= \frac{C-B}{L}$ (57.3) DEGREES

Φ = WHEEL TO SIDEFRAME $\measuredangle \cong \frac{D-A}{L}$ RADIANS $= \frac{D-A}{L}$ (57.3) DEGREES

Ω = WHEEL TO RAIL $\measuredangle = \Theta - \Phi$

R = SIDEFRAME TO RAIL DISTANCE $= \frac{B+C}{2}$

W = SIDEFRAME TO WHEEL DISTANCE $= \frac{A+D}{2}$

R/W = RAIL WHEEL POSITION = R-W

FIGURE C-1. WHEEL/RAIL POSITION MEASUREMENT

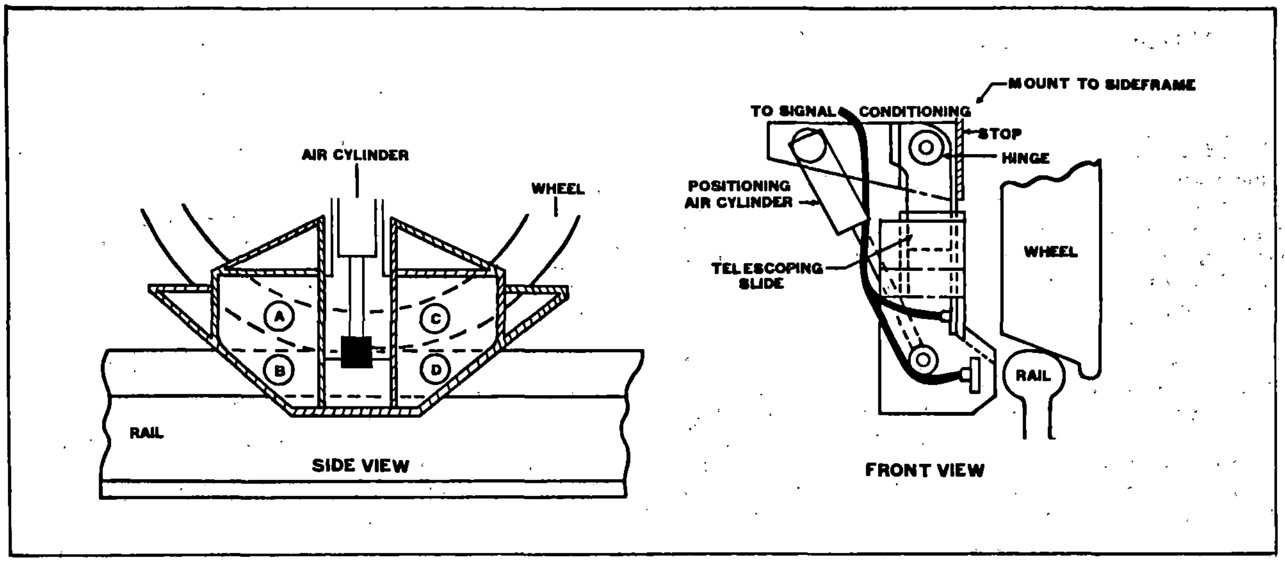


FIGURE C-2. WHEEL/RAIL MEASUREMENT SYSTEM

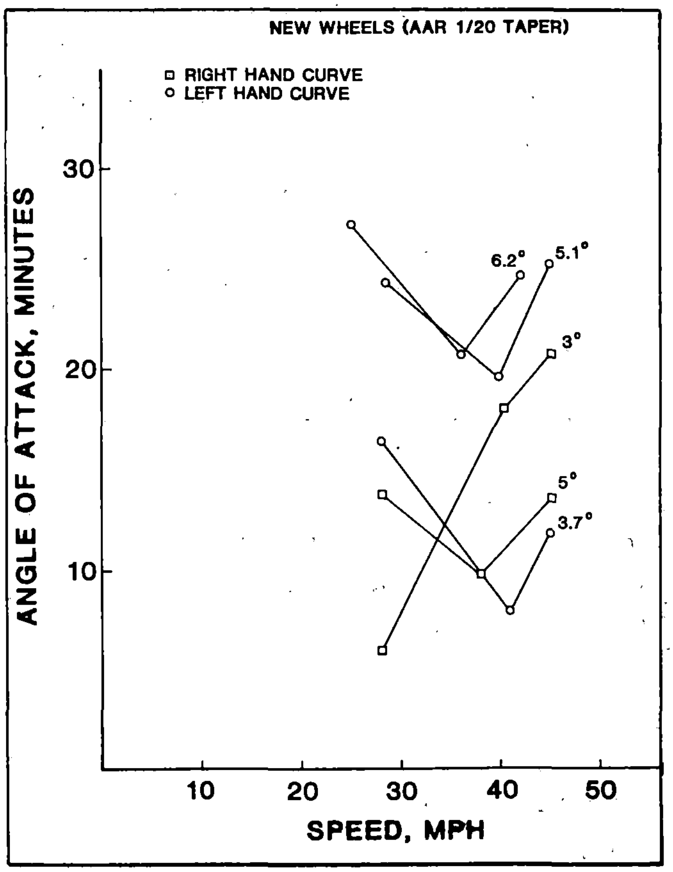


FIGURE C-3. ANGLE OF ATTACK VERSUS SPEED - 100-TON TRUCKS WITH LOADED BOX TYPE CARS

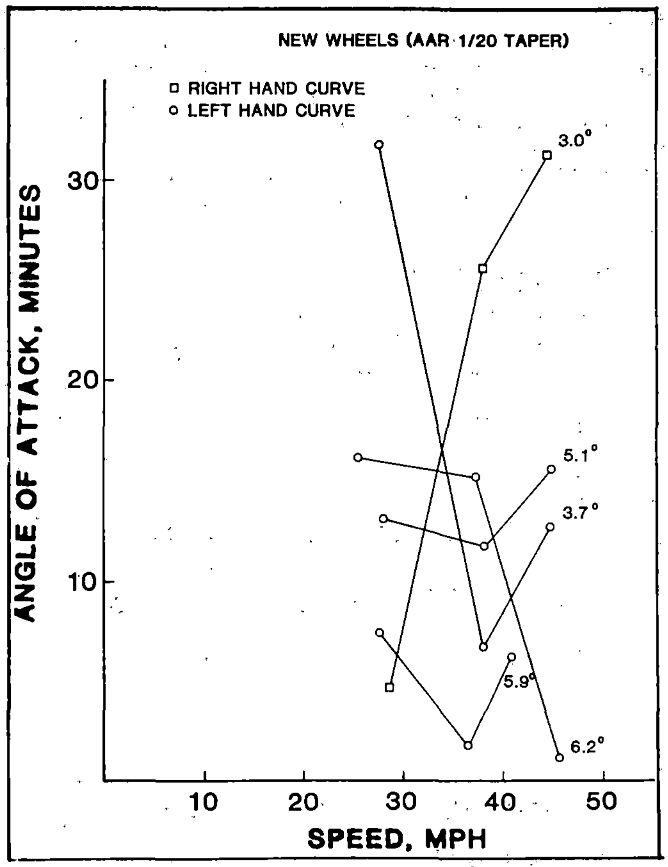


FIGURE C-4. ANGLE OF ATTACK VERSUS SPEED - 100-TON TRUCKS WITH EMPTY BOX TYPE CARS

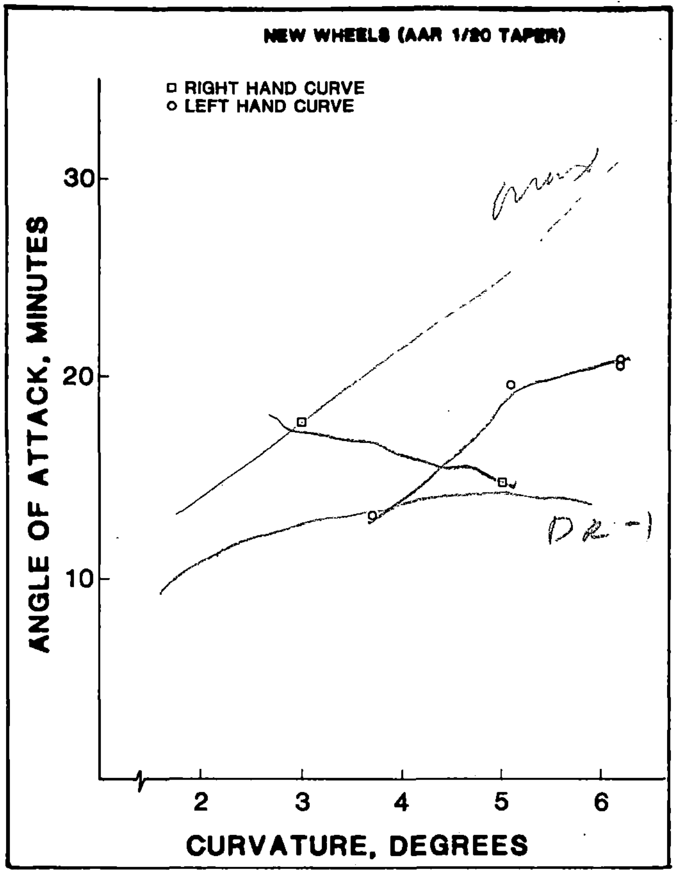


FIGURE C-5. ANGLE OF ATTACK VERSUS DEGREE OF CURVATURE NEAR BALANCE SPEED - 100-TON TRUCKS WITH LOADED BOX TYPE CARS

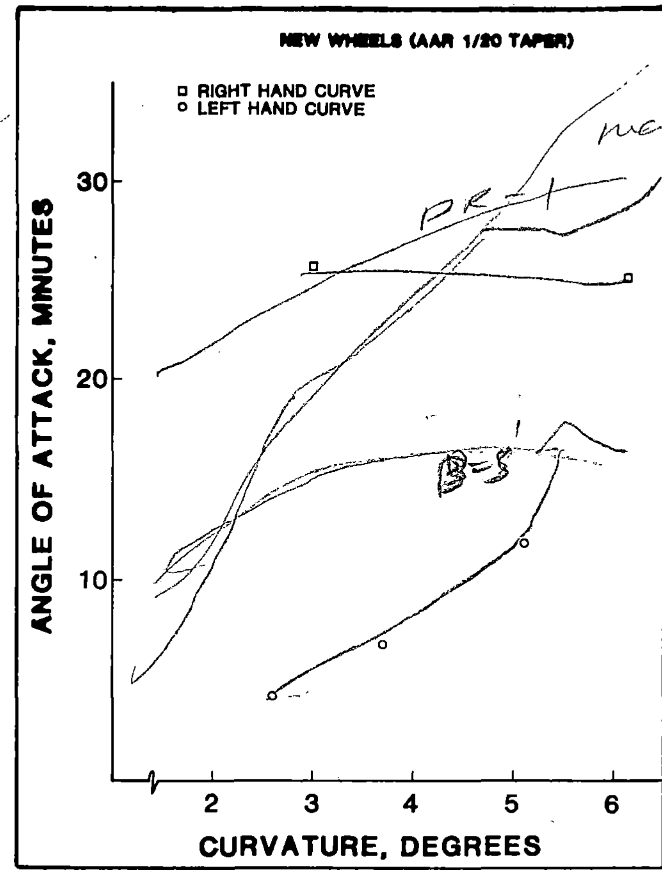


FIGURE C-6. ANGLE OF ATTACK VERSUS DEGREE OF CURVATURE NEAR BALANCE SPEED - 100-TON TRUCKS WITH EMPTY BOX TYPE CARS

APPENDIX D - GLOSSARY

ANALYTICAL TOOL

Refers collectively to a series of techniques used to study or predict the dynamics of a physical system, such as a freight car. Analytical tools are made up of mathematical models and computer programs. Models consist of a set of equations which can be used to mathematically study and predict the response of physical systems. The computer programs implement the sets of equations or models on a digital computer.

ANGLE OF ATTACK

Horizontal angle between the vertical plane of the wheel and the tangent to the rail at the point of contact.

"B" END OF CAR

The end on which the hand brake is located.

BALANCE SPEED

The speed with which a vehicle traverses a superelevated curve of constant radius when the centrifugal force exactly balances the horizontal component of the weight due to inclination.

BOUNCE

Vertical oscillation of the center of gravity of the sprung mass (carbody, truck bolster, etc.).

CLIMB

The process of the wheel flange frequently contacting and climbing the rail onto the railhead.

COULOMB DAMPING

A damping mechanism that depends on constant amplitude friction forces which always resist the direction of motion.

CURVE NEGOTIATION

The ability of a truck to enter, provide guidance through, and exit a curve.

CURVE

In the United States it is customary to express track curvature in degrees noted by the deflection from the tangent measured at stations 100 feet apart. The number of degrees of central angle subtended by a chord of 100 feet is the "degree curve." One degree of curvature is equal to a radius of 5,750 feet.

COULOMB DAMPING

A damping mechanism that depends on constant amplitude friction forces which always resist the direction of motion.

CREEP

The capability of two bodies to displace in their plane of contact without slipping. It is made possible by shear deformation of both bodies in the region of the interface, which can support tractive forces.

CRITICAL DAMPING

Amount of damping at which no oscillatory vibration occurs after a spring-mass has been released from a non-equilibrium position.

CRITICAL HUNTING SPEED

Minimum speed at which violent truck shimmy occurs.

CRITICAL SPEED

Excitation forces applied to the vehicle are related to forward speed. A critical speed is one at which a car-truck dynamic resonance occurs.

CURVE

In the United States it is customary to express track curvature in degrees noted by the deflection from the tangent measured at stations 100 feet apart. The number of degrees of central angle subtended by a chord of 100 feet is the "degree curve." One degree of curvature is equal to a radius of 5,750 feet.

CURVE NEGOTIATION

The ability of a truck to enter, provide guidance through, and exit a curve.

DAMPING COEFFICIENT

Number describing the energy-absorbing property of a physical system.

DATA BASE

A collection of interrelated data stored together to serve one or more applications; data are stored so that they are independent of programs using the data, a common and controlled approach to maintenance and retrieval of data.

ELEVATION

The higher position of one of the two rails.

FORCED FREQUENCY

Frequency imposed on the system by superimposed forces (rail joints, etc).

FORCED RESPONSE

A term used to describe the response of a system due to some external forcing function.

FRICTION SNUBBER

A device built into the secondary suspension of a truck to absorb energy. The standard designs rely on sliding friction to dissipate energy.

GAGE OF THE TRACK

The distance between the rails measured from the inside head of each rail at a right angle 5/8 inches below the top of the rail. The standard for this dimension on North American Railways is 4 feet, 8½ inches.

GRADE

Part of roadbed with changing elevation.

GROSS WEIGHT

The total weight of a car, including the lading.

HARMONIC ROLL

Periodic angular displacement of the vehicle body about its longitudinal axis, due to vertical track inputs close to the natural frequency of the carbody on its suspension, referred to as rock and roll.

HUNTING

Dynamic instability of sets of wheels or entire trucks consisting of a lateral translation along the axle and rotational vibration about a vertical axis.

HYSTERESIS

The dependence of the state of a system on its previous history, generally in the form of a lagging of a physical effect behind its cause.

KINEMATICS

The branch of mechanics that deals with motion without consideration of inertial forces.

KINEMATIC WAVELENGTH

The wavelength of the sinusoidal motion of a wheelset or truck along the track when inertial forces are negligible.

KINETIC FRICTION

Friction of motion, such as that between the brake shoe and wheel (when the wheel is turning) or as between a wheel and rail (during sliding or slipping). Kinetic friction is always less than static friction.

LATERAL STABILITY

Refers to the stability of a truck in the lateral direction. Trucks with a low degree of lateral stability tend to oscillate or hunt as speed is increased below some critical value.

LATERAL VIBRATION

Pure side to side movement in the horizontal plane.

LOAD EQUALIZATION

Ability of truck to maintain equal load distributions on all four wheels while accomodating the full range of in-service track geometry.

LOWER CENTER ROLL

Rotation of the carbody about a virtual longitudinal axis below its center of gravity.

L/V RATIO

Defined as the ratio of the lateral force to the vertical force of a car or locomotive wheel on a rail. It is an important indicator of wheel climb, shifting of the track structure, rail turnover, and/or derailments.

NATURAL FREQUENCY

The frequency at which the system tends to vibrate when released after being displaced from neutral position.

OPERATIONAL CONDITIONS

Refers to the conditions or physical environment in which a truck must operate. Typical operational conditions include speed, track condition, and loading.

PARALLELOGRAMMING

Relative longitudinal displacement of truck sideframes which causes the truck to go in and out of tram.

PARAMETER VARIATIONS

Refers to variations made on the parameters of a mathematical model to study the effect of component changes, configuration changes, or operational environment changes.

PEAK VALUE

Peak value is defined as the absolute peak value (zero-to-peak) of the signal determined from an examination of the time history of the signal.

PERFORMANCE CRITERIA

The aspects of truck behavior considered desirable in various performance regimes. Criteria may range from the most general, such as safety from derailment or low wear rates, to the specific, such as lateral stability or curve negotiability.

PERFORMANCE INDEX

A measurable physical quantity characteristic of performance in a particular regime. An example of a performance index for hunting would be the critical speed, and for curve negotiation, the lateral load on the outer leading wheel of the truck. Each performance index must be qualified by a statement of conditions for which it applies, and which may affect its magnitude to varying degrees.

PERFORMANCE REGIME

The characteristic way in which a railcar or truck responds to a combination of track and operating conditions (such as speed). Inherent in this definition is a comparison with stable vehicle behavior on "ideal" tangent track. Performance regimes selected for truck characterization should be sufficiently distinct to permit ranking of truck performance on non-overlapping scales. The four primary regimes chosen are lateral stability, trackability, curve negotiation, and ride quality.

PITCH

Angular motion in the vertical plane about the axle perpendicular to the direction of the track.

POWER SPECTRAL DENSITY (PSD)

This represents the distribution of energy in a vibrating system under defined test conditions over the frequency spectrum specified.

PRIMARY SUSPENSION

Suspension elements between the sideframe and wheelsets. For freight car trucks, these usually take the form of elastomeric pads.

RECTIFIED SINE WAVE TRACK GEOMETRY

The shape of the vertical track deviations used to analytically describe jointed rail. The joints correspond to the low points of the rectified sine wave and the mid points represent the peaks of the sine wave.

RESONANCE

The condition at which forcing frequency is equal to natural frequency; this usually results in violent motion.

ROCK AND ROLL

An informal term for the excessive lateral rocking of cars and locomotives, usually at low speeds and associated with jointed rail. The speed range at which this cyclic phenomenon occurs is between 10 and 25 mph, with the exact speed determined by such factors as the wheel base, height of the center of gravity of each individual car or engine, the spring dampening associated with the suspension system of each vehicle, and the relative difference in elevation between successive joints in jointed rail territory. In extreme cases, actual wheel lift can occur which can result in derailments (see also Harmonic Roll).

ROLL

Rotation of the carbody about a longitudinal axis through the center of gravity.

ROLLABILITY

The relative resistance of the truck to longitudinal motion.

ROOT MEAN SQUARE VALUE

The root mean square value (rms) of a signal is determined from the corresponding power spectral density by integration within a specified frequency band.

SECONDARY SUSPENSION

Suspension elements between the truck bolster and sideframe. This is the principal means of isolating vibration in a freight car truck.

SHIMMY

A synonym for hunting.

SNUBBERS

Damping devices which are used to attenuate oscillations of a car or truck. They may be similar to hydraulic shock absorbers. Friction devices are commonly used in rail vehicles.

SUPERELEVATION

The vertical distance between the heights of inner and outer edges of railroad rails.

SWIVELING

Angular oscillation about an axis; a symmetry, usually applied to truck action when the bolster oscillates around the center pin.

THROAT (CAR WHEEL)

The curved transition between the wheel tread and flange.

TRACKABILITY

Refers to the ability of a truck to maintain equal wheel loads under all extremes of operating conditions.

TRACK-TRAIN DYNAMICS

A term used to describe the dynamic motion and the resulting dynamic forces that result from the interaction of the vehicles coupled into a train interacting with the track, under given climatic conditions, train handling, train makeup, grades, curvature, and operating policies.

TRACK-TRAIN ENVIRONMENT

All the conditions which affect the track and/or the train, such as grades, curvature, locomotive and car characteristics, train handling, etc.

TRACK TWIST

Refers to cross-level variations which occur within the wheelbase of the truck.

TRAM

This term applies to the diagonal measurement of axle bearing locations. When, in a four-wheel truck, the two diagonal measurements are equal, the truck is said to be in tram.

TRANSMISSIBILITY RATIOS

A frequency-dependent function of amplitude ratios called a transfer function, or a sequence of root mean square ratios of output to input over selected frequency bands.

TRUCK CENTER

The center point of a truck. The distance between truck centers is that distance as measured from one truck center to the other truck center on a single car.

TRUCK CLASSIFICATIONS

TYPE I: GENERAL PURPOSE DESIGN (STANDARD THREE-PIECE)

This design is interchangeable with existing trucks so as to preserve the present truck coupler height, support the carbody on center plates, utilize air brakes which are compatible with existing systems, accept standard wheelsets and journal bearings, and whose components meet applicable Association of American Railroads (AAR) requirements.

TYPE II: SPECIAL PURPOSE DESIGN (PREMIUM)

This design utilizes current wheelset and journal bearing assemblies, is compatible with existing air brake systems, and preserves car coupler height. The Type II truck may employ mechanisms other than center plate and side bearings for support and stabilization of the carbody.

TRUCK WHEEL BASE

The horizontal distance between the centers of the first and last axles of a truck.

UNDULATING GRADE

A track profile with grade changes so often that an average train passing over the track has some cars on three or more alternating ascending and descending grades. The train slack is always tending to adjust as cars on descending grades tend to roll faster than those on ascending grades.

UPPER CENTER ROLL

Rotation of the carbody about a longitudinal axis above its center of gravity.

VERTICAL VIBRATION

Pure up and down motion often described as bounce.

WHEEL CLIMB

This term applies to the condition where the lateral (axial) force between the wheel flange and rail head is great enough so that the resulting friction force causes the wheel flange to climb up on the rail.

WHEEL CONICITY

The slope of the wheel tread at the point of wheel/rail contact. Wheel conicity is used in linearized models to determine the change of rolling radius and the restoring force resulting from the particular wheel/rail contact.

WHEEL FLANGE

The projecting edge or rim on the periphery of a car wheel for keeping it on the rail.

WHEEL LIFT

This term applies to the lifting of a lightly loaded wheel due to the moment resulting from the high vertical force on the opposite bearing. Such forces are encountered when rail vehicles are operated at speeds too great for the existing superelevation on a curve, from very slow speed operation on a high superelevation curve, from high draft (or buff) forces on a curve, or from harmonious rocking of a car on rough track.

WHEEL SLIDING

The situation where the wheel is rotating slower than longitudinal movement would dictate, and adhesion is lost.

WHEEL SLIPPING

The situation where the wheel rotates faster than longitudinal movement would dictate, and adhesion is lost.

WHEEL TREAD

The exterior cylindrical surface of a wheel which bears on the rails.

WHEEL UNLOADING

Reduction of vertical wheel reaction on the rail.

YAW

Angular motion in the horizontal plane about a vertical axis.

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US DOT, FRA, DV RamaChandran, MM El Madany