U.S. Department
of Transportation

Federal Railroad
Administration

Laboratory Tests of Two 100-Ton Covered Hopper Cars

G Gyons

Office of Research and
Development
Washington, D.C. 20590

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DOT/FRA/ORD-85/13

December 1985

MITRE

14 January 1986

W102-7276
1926E

Mr. Alfred G. Bowers
Office of Research and Development
Federal Railroad Administration
400 Seventh Street, S.W., Room 5423
Washington, D.C. 20590

Subject: Submission of Final Report Under Task 2.4.3,
DOT/FRA Contract No. DTFR53-82-C-00087

Dear Mr. Bowers:

Enclosed please find 30 copies and one unbound reproducible copy of the final report, "Laboratory Tests of Two 100-Ton Covered Hopper Cars," MITRE Report Number MTR84W00253, submitted in fulfillment of contract deliverable requirements. The report also carries Department of Transportation document number FRA/ORD-85/13. This completes the activity on Task 2.4.3.

If you have any questions, please call me at (703) 883-6824.

Sincerely,

Peter Wood
Department Head
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Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
FRA/ORD-85/13		

4. Title and Subtitle	5. Report Date / 6. Performing Organization Code / 8. Performing Organization Report No.
LABORATORY TESTS OF TWO 100-TON COVERED HOPPER CARS	5. Report Date: December 1985 6. Performing Organization Code: 8. Performing Organization Report No.: MTR84-W00253
7. Author(s): George Kachadourian	
9. Performing Organization Name and Address: The MITRE Corporation, METREK Division 1820 Dolley Madison Boulevard McLean, Virginia 22102	10. Work Unit No. (TRAIS) 11. Contract or Grant No.: DTFR53-82-C-00087
12. Sponsoring Agency Name and Address: Federal Railroad Administration 400 Seventh Street, S.W. Washington, D.C. 20590	13. Type of Report and Period Covered: Technical Report 14. Sponsoring Agency Code: RRD-10

15. Supplementary Notes

16. Abstract

The Federal Railroad Administration (FRA) and the Association of American Railroads (AAR) have jointly sponsored an experimental program to study the 100-ton covered hopper car with the end objective of improving its safety performance. The total program includes track and laboratory testing of two base vehicles and two prototype vehicles. This report presents the laboratory results of the two base vehicles. The laboratory tests consist of quasi static tests on each truck from both base vehicles and vibration tests of each vehicle on the Vibration Test Unit (VTU) at the Transportation Test Center (TTC) in Pueblo, Colorado. The dynamic properties of the test vehicles were measured including several configuration variations. The responses of the vehicles to simulated track conditions were measured with track conditions carried to the point of wheel lift. The objectives of the laboratory tests were to define the dynamic properties of the test vehicles and the effects of these properties on safety performance.

17. Key Words	18. Distribution Statement
Covered Hopper Car Harmonic Roll Vibration Testing Truck Properties Safety Performance	Available to the Public through the National Technical Information Service, Springfield, Virginia 22161

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

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

METRIC CONVERSION FACTORS

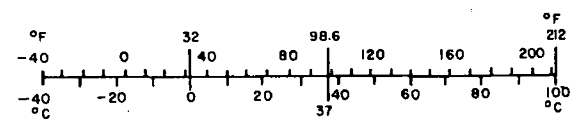
Approximate Conversions to Metric Measures

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 (exactly). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights 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	35	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



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ABSTRACT

The Federal Railroad Administration (FRA) and the Association of American Railroads (AAR) have jointly sponsored an experimental program to study the 100-ton covered hopper car with the end objective of improving its safety performance. The total program includes track and laboratory testing of two base vehicles and two prototype vehicles. This report presents the laboratory results of the two base vehicles. The laboratory tests consist of quasi static tests on each truck from both base vehicles and vibration tests of each vehicle on the Vibration Test Unit (VTU) at the Transportation Test Center (TTC) in Pueblo, Colorado. The dynamic properties of the test vehicles were measured including several configuration variations. The responses of the vehicles to simulated track conditions were measured with track conditions carried to the point of wheel lift. The objectives of the laboratory tests were to define the dynamic properties of the test vehicles and the effects of these properties on safety performance.

ACKNOWLEDGEMENT

The contents of this report are the result of activities by the Systems Development Department of The MITRE Corporation as part of a program to develop and evaluate prototype 100-ton covered hopper cars. The work was done as Task 2.4.3 of Contract No. DTFR53-82-C-0087 with the Department of Transportation, Federal Railroad Administration, under the direction of Claire L. Orth and Alfred G. Bowers of the Office of Research and Development. The tests were performed at the Transportation Test Center, Pueblo, Colorado by the Association of American Railroads personnel under the supervision of the RDL Test Engineer, Dan Inskeep, and with the very valuable contribution of Technical Analyst Firdausi Irani in over-seeing the data processing and computerized data analysis. Special thanks are also in order to Diane Boone of the MITRE Technical Staff for her assistance in analysis of the test data.

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EXECUTIVE SUMMARY

INTRODUCTION

The 100-ton covered hopper car has historically experienced a larger than average derailment tendency and has been identified as having an apparent safety problem which is compounded by the fact that covered hoppers represent a large and increasing portion of the national fleet.

In addressing this problem, the Track-Train Dynamics (TTD) program has been working to develop a high-cube covered hopper car with improved operating safety and improved dynamic performance characteristics. Performance Guidelines have been issued, [1]*, that define design requirements and improvement objectives, and two manufacturers, the Thrall Car Manufacturing Company and the Budd Company, have developed and built prototype covered hopper cars using these guidelines.

A test program was also arranged with the objective of evaluating the degree of improvement achieved by the prototype hopper cars, with two, in-service 100-ton covered hopper cars being used as the basis for comparison. The test program, which is being performed at the Transportation Test Center (TTC), is a cooperative effort involving Thrall and Budd for the use of their prototype cars, the Missouri Pacific Railroad for the loan for the two 100-ton hopper cars in current service, and the Federal Railroad Administration (FRA) and the Association of American Railroads (AAR) for funding and conducting the test program. Testing of the two base cars and the Thrall prototype have been completed. The testing of the Budd prototype is expected to be performed in late 1985.

The results of the track testing performed on one of the base vehicles and on the Thrall prototype have been prepared by the AAR and are contained in References [2] and [3]. The purpose of this report is to present the results of the laboratory tests performed on the two base vehicles. Laboratory testing of the Thrall vehicle was not performed.

* Referenced reports, designated by bracketed numbers, are listed at the end of the report.

TEST OBJECTIVES

Track test objectives were to measure the performance of the base and prototype hopper cars as to hunting, flange climbing, harmonic roll, vertical and lateral wheel loads, load equalization, and curve negotiation. The track testing was a demonstration of vehicle performance and provided the basis for measuring the degree of improvement achieved by the prototype hoppers.

The laboratory test objectives were to obtain measures of the stiffness and damping in each truck configuration, to measure the modal dynamic response characteristics of each vehicle, and to measure the dynamic response of each vehicle to a range of track condition. The laboratory tests would provide data on the dynamic response characteristics of the vehicles and how these characteristics are affected by component or configuration changes in each vehicle. This information would then provide a basis for understanding how improvements were achieved and how further improvements might be possible.

An add-on laboratory test objective, separate from the tests to evaluate hopper car performance, was to obtain data for the evaluation of a track geometry measurement parameter that has been named Cross Level Index, or CLI.

SUMMARY RESULTS OF LABORATORY TESTING ON BASE VEHICLES

A summary of results of the laboratory tests performed on the two base vehicles are presented in this executive summary for the Truck Characterization Tests, the Vehicle Modal Tests, and the Track Simulation Tests. The results presented are for the most part qualitative statements. Numerical and graphic results are presented in detail in the body of the report.

Truck Characterization Test Results

Base Vehicle No. 1 was received with Barber S-2-C trucks with load variable friction snubbers, a D-5 spring group, inner and outer side springs, and constant contact resilient side bearings. Base Vehicle No. 2 was received with Ride Control trucks with constant column friction snubbers, a D-5 spring group, and double roller side bearings. Characterization tests were performed on all four trucks. The effect of load condition from empty to 80% and the effect of moisture on the snubbers was measured. The Barber trucks were tested with both roller and resilient side bearings.

Test procedure was to vary the applied loads with a sine function at the frequency of 0.2 Hertz. This cyclic application of load was used to eliminate errors that would have otherwise been caused by the difference between static and sliding friction forces in the snubbers. The 0.2 Hertz frequency is low enough to avoid any dynamic effects. The loads were applied in three modes: vertical, to obtain vertical spring rates; lateral, to obtain lateral spring rates; and roll to obtain roll spring rates.

There was nothing unusual found in the spring constants for the suspension systems. Comparison between the Barber and Ride Control trucks showed a consistency of values as did comparison to three computer models. It was found that the constant contact resilient side bearings cause the truck to be stiffer in roll in such a way as to increase the tendency for center plate lift with roll motions of the car body.

The outstanding difference between the Barber and Ride Control trucks was in the friction forces developed in the snubbers. Some of the difference was due to the difference between load variable (Barber) and constant column (Ride Control) snubbers. However, the A end Barber truck was found to have a binding condition that caused it to have considerably higher friction forces. A summary tabulation of snubber forces is given in Table A.

Modal Test Results

The modal testing was performed on the Vibration Test Unit (VTU), the shaker facility at the TTC for vibration testing of rail vehicles. The VTU is a computer controlled system of fourteen hydraulic actuators capable of supporting the full weight of a loaded 100-ton freight car while imposing pre-programmed vertical and lateral motions at each wheel and longitudinal motions at the couplers.

The modal testing consisted of imposing sinusoidal motions at the wheel-rail interface with motions and frequencies intended to excite the vehicle resonant frequencies of interest. There were eight modes found and investigated: first roll, yaw, bounce, second roll, pitch, torsion, lateral bending, and vertical bending. The effects of eleven configurational changes were investigated.

TABLE A
SUMMARY OF SNUBBER FORCE DATA FROM TRUCK CHARACTERIZATION TESTS

TRUCK	COLUMN FRICTION FORCE - POUNDS			
	DRY SNUBBERS		WET SNUBBERS	
	EMPTY	90%	EMPTY	90%
Ride Control	4,300	3,500	2,900	2,300
Barber B Truck (a)	1,600	4,400	1,400	3,200
Barber B Truck (b)	2,850	7,800	2,150	4,820
Barber A Truck (b)	3,300	13,300	(not tested)	

(a) with outer side springs only
(b) with outer and inner side springs

Track Simulation Tests with Base Vehicles on the Vibration Test Unit

There were three basic track conditions simulated on the VTU: staggered rail, grade crossing, and vehicle yaw motions representative of body hunting conditions. The staggered rail cross level variations were also combined with alignment variations and super elevation. Several different profile shapes were used for the staggered rail simulation. Of these, two were the most significant: the "shimmed rail" profile was a duplication of the triangular shimming pattern used for the track testing; and the "rectified sine" profile was a duplication of profiles found in service.

Comparison of the VTU and Track Test Results for Staggered Rail

Comparison of the VTU and track test results for staggered rail showed wheel lift to occur at a larger cross level input (.85 compared to .75 inches) and with smaller car body roll angles (7.3 compared to 10.6 degrees peak-to-peak) in the VTU tests. It was postulated that this difference was due in part to the track test being with slack conditions while the VTU tests were with about 45,000 pounds draft load. This theory should be investigated in some future test. A second probable contributor to the difference in results is that the VTU does not account for the effects of flexibility of the track and road bed.

Variation of Profile Shape in Staggered Rail

Variation of profile shape in the staggered rail indicated that the shimmed rail profile, as used in the track tests, was not as severe a test as with a rectified sine profile shape. That is, where the shimmed rail resulted in wheel lift with 0.75-inch cross level the rectified sine profile resulted in wheel lift with 0.60-inch.

Safety Margin with Staggered Rail, VTU Test

It was found, in the comparison of track and VTU test results, that there was very good agreement when rectified sine rail profile was used in the VTU test and when using wheel lift as the basis for comparison. The safety margin testing for staggered rail was, consequently, done with the rectified sine simulation.

There were eight configurational variations made for these tests. The cross level of 0.75 inches was used as the dividing line between safe and unsafe performance. That is, if a limit condition (wheel lift or center plate lift) was obtained with cross level input less than 0.75 inches, the configuration was considered unsafe. Based on this criterion, the following configurations were found to be unsafe:

B1 - Vehicle No. 2, empty, Barber trucks with double side springs. Critical cross level: 0.50 inches.

B2* - Vehicle No. 1, 101-ton load, Barber trucks with double side springs. Critical cross level: 0.70 inches.

RC2 - Vehicle No. 2, 99-ton load, Ride Control trucks, Critical cross level: 0.65 inches.

B3 - Vehicle No. 2, 72-ton load, Barber trucks with double side springs. Critical cross level: 0.60 inches.

B7 - Vehicle No. 2, 99-ton load, Barber truck with double side springs and constant contact resilient side bearings. Critical cross level: 0.70 inches.

The following configurations reached at least 0.75-inches cross level without experiencing a limit condition and were considered safe:

B4 - Vehicle No. 2, 99-ton load, Barber trucks with single side springs. Critical cross level: 0.75 inches.

B2 - Vehicle No. 2, 99-ton load, Barber trucks with double side springs. Critical cross level: 0.80 inches.

B5 - Vehicle No. 2, 99-ton load, Barber trucks with single side springs, and auxiliary hydraulic snubbers. Critical cross level: 0.80+ inches.

The B2 configuration was shown by the VTU testing to have an adequate margin of safety for harmonic roll performance. The following configurational changes were shown to lower the car's performance to an unsafe condition. (Unsafe performance being defined as wheel lift or center plate lift with staggered rail cross levels of less than 0.75 inches.)

- Truck spacing of 39 ± 5 feet
- Vehicle loading less than 75 tons
- Snubber friction force of 4000 pounds or less
- Use of constant contact resilient side bearings

Bounce Test Results

The purpose of the bounce testing was to obtain a measure of the 100-ton covered hopper car response to track hard spots such as grade crossings. The critical speed was found to be between 60 and 65 miles per hour. However, there were no responses measured that would indicate a safety problem.

In comparing the several configurations tested, the Ride Control truck, configuration RC2, had the largest responses and the Barber truck with reduced friction and auxiliary hydraulic snubbers, configuration B2, had the smallest responses. This excludes the empty condition where the critical speed was above 80 miles per hour.

Yaw Test Results

The yaw tests were an attempt to duplicate motions and loads that result from a body hunting condition. The results were expected to indicate the frequencies at which body hunting will occur and a general indication of the severity of the motions.

The yaw frequency was found to fall between 2.5 and 3.0 Hertz. The maximum conditions tested, which were with a lateral sinusoidal input of 1.20 inches peak-to-peak, resulted in lateral carbody accelerations of about 0.28 g RMS which is representative of a full flange-to-flange hunting condition. Lateral wheel loads reached maximum values in excess of 40 kips with maximum L/V values of about 0.90.

The conclusion drawn is that it is possible to create motions of the freight car that are representative of on-track hunting conditions and that the VTU can be a useful tool in studying loads and motions caused by body hunting.

1. INTRODUCTION

The purpose of this report is to present the results of laboratory tests performed on two 100-ton covered hopper cars. These cars were obtained from the Missouri Pacific Railroad's fleet of hopper cars as being representative of cars in current service. The tests were performed as part of a program to improve the safety performance of 100-ton covered hopper cars that included track testing of prototype covered hopper cars as well as the in-service cars used as a base for comparison.

An overview of the test program is included in the introduction with a review of why the program is needed, what the program is, and descriptions of the four covered hopper cars (two base cars and two prototype) to be tested.

The laboratory testing was divided into four parts both as to test performance and results presentation in this report:

1. Truck characterization tests of the Barber trucks from Base Vehicle No. 1.

   Test dates: April 20 - June 5, 1983.

2. Truck characterization tests of the Ride Control trucks from the Base Vehicle No. 2.

   Test dates: September 15-19, 1983.

3. Vibration testing of the loaded 100-ton covered hopper with Barber S-2-C trucks identified as the MoPac Vehicle (Base Vehicle No. 1).

   Test dates: December 14, 1982 - January 13, 1983.

4. Vibration testing of Base Vehicle No. 2 both with Ride Control trucks and with Barber trucks from Base Vehicle No. 1.

   Test dates: August 4 - September 9, 1983.

A summary of data from all four parts is presented in Section 7.

1.1 Background

The 100-ton covered hopper car has historically experienced a larger than average derailment tendency. From 1974 to 1977, 1216 derailments involved 100-ton covered hopper cars. These cars had 1.7 accidents per 10 million car-miles and 4.0 derailments per billion net ton-miles while the average for all car types was 0.9 and 3.2, respectively. Nayak and Palmer [4], following a compilation and analysis of accident data, make the statement that "the single prominent type of car with an apparent safety problem is the covered hopper." The problem is compounded by the fact that covered hoppers represent a large and increasing portion of the national fleet going from 14.2% in 1977 to 19.5% in 1982, based on data from the Association of American Railroads (AAR) Yearbooks of Railroad Facts. When loaded, covered hopper cars are the heaviest cars, as a class, with the highest center of gravity.

In addressing this problem, the Track-Train dynamics (TTD) program has been working to develop a high-cube covered hopper car with improved operating safety and dynamic performance characteristics. As part of the TTD Program the AAR issued the document Performance Guidelines--High Performance/High Cube Covered Hopper Cars [1] and two car manufacturers have developed prototype covered hopper cars using these guidelines: the Thrall Car Manufacturing Company and the Budd Company.

1.2 Test Program

The next step in the TTD Program for this project was to generate and implement an experimental program to quantify the improvements achieved by the two prototype covered hopper cars. This experimental phase of the program, which at this writing is still in progress, is a cooperative effort involving the Thrall and Budd Companies for the loan of their prototype cars, The Missouri Pacific Railroad for the loan of two typical in-service 100-ton covered hopper cars, and the FRA and AAR for funding and conduct of the test program.

The primary objective of the testing was to measure the performance of the prototype covered hopper cars and to show the degree of improvement that has been achieved over 100-ton covered hopper cars in current service. A second general objective of the test program was to measure the dynamic characteristics of each of the four vehicles through tests performed in the Rail Dynamics Laboratory. These consisted of static load type tests to measure truck stiffness and friction properties and vibration tests to measure modal properties of the vehicles as well as vehicle performance under simulated track conditions. A final test objective was to perform tests to be used in the validation of a cross level index developed by the Transportation System Center [5].

With this writing, only the testing of the Budd prototype covered hopper car remains to be done. Tyrell, Ehrenbeck and Weinstock of TSC have written a report [6] that presents the results of tests performed in evaluation of TSC's cross level index. The AAR has prepared two reports that present summaries of track test results on one of the base hopper cars [2] and on the Thrall prototype (the THETA-80) [3]. The THETA-80 report also presents comparisons that show the performance improvement achieved by the THETA-80.

Table 1-1 presents a Program Test Matrix showing the four vehicles under test, the three categories of truck, VTU and track testing indicating what has been completed, the reports containing results, and what remains to be done.

1.3 Test Vehicle Description

Configurational data is given in Table 1-2 for the two base vehicles. Additional base vehicle information is given below along with configuration descriptions of the two prototype vehicles.

1.3.1 Base Vehicle No. 1

Base Vehicle No. 1 was pulled from the Missouri Pacific Railroad fleet of covered hopper cars. The vehicle had been received with constant contact resilient side bearings--these were replaced with double roller side bearings. In the course of testing this vehicle, there were four conditions found which, although they will occasionally be found in a fleet of vehicles, are not according to AAR specifications. These were as follows:

1. Friction snubbers binding in A end truck. The truck had apparently been refurbished with new column wear plates, in the side frames, with the plates welded in place and positioned so that the edge of the plate would gouge the friction face of the friction casting. This resulted in very noisy operations and very high effective friction forces.

2. Center plate on the body bolster, A end, rotated 90° so that tapered edges were at the front and back edges. There was no measurable affect on car characteristics due to this error. Its only likely impact would be on the wear life of the center bowl.

3. No vertical wear liner in the B center bowl. No noticeable effects.

TABLE 1-1
PROGRAM TEST MATRIX

TESTS	VEHICLES			
	BASE NO. 1	BASE NO. 2	THETA-80	BUDD
Truck Characterization	Completed (a)	Completed (a)	Not to be done	To be done 1985
VTU	Completed (a) & (b)	Completed (a)	Not to be done	To be done 1985
Track	Not to be done	Completed (c)	Completed (c)	To be done 1985

(a) Results presented in this report
(b) Results presented in Reference [5]
(c) Results presented in References [2] and [3]

TABLE 1-2
BASE VEHICLE CONFIGURATION DATA

ITEM	UNITS	BASE VEHICLE NO. 1 MP 712074	BASE VEHICLE NO. 2 MP 723288
Capacity	cu. ft.	4,427	4,750
Light Weight	lbs.	60,200	60,700
Load Limit	lbs.	202,800	202,300
C.G. Height Empty	in.	63	61
C.G. Height Loaded	in.	97	95
Length	ft.-in.	50-0	57-4
Truck Spacing	ft.-in.	40-5	45-9
Axle Spacing	ft.-in.	5-10.5	5-10
Truck Type		Barber S-2-C	Ride Control*
Friction Snubber		Load Variable	Constant Column
Spring Group		D5: 7 Outer, 6 Inner 432 Outer: 2 433 Inner: 2	D5: 7 Outer, 9 Inner
Side Bearing		Double Roller	Double Roller
Center Plate Diameter	in.	16 (Tapered)	16 (Tapered)
Wheel Diameter	in.	36	36
Journal	in.	6 1/2 x 11	6 1/2 x 11

* Laboratory tests were performed on Base Vehicle No. 2 with both the Ride Control and Barber S-2-C trucks.

4. Mismatch of B end center plate and center bowl evidenced by the right side end of the truck bolster always being lower than the left end.

Base Vehicle No. 1 was introduced into the program primarily for TSC's Cross Level Index testing and because of schedule pressures. Only laboratory tests were performed on this vehicle.

1.3.2 Base Vehicle No. 2

This vehicle was also from the MoPac fleet. Configurational differences between the two base vehicles are shown in Table 1-2. Both vehicles were conventional 100-ton covered hopper cars with three-piece trucks. The significant differences were that Base Vehicle No. 2 had Ride Control trucks with constant column damping rather than the Barber S-2-C with load variable damping. Vehicle No. 2 had a longer truck center distance (45' 9" against 40' 5") and lower center of gravity (95" against 97" loaded). Base Vehicle No. 2 did not evidence any of the anomalies found in Vehicle No. 1 as discussed above. These differences between Vehicles No. 1 and 2 are within the variabilities to be found between 100-ton covered hopper cars in the fleet.

Base Vehicle No. 2 was intended to be the vehicle to be used as the comparison base for evaluating the prototype vehicles. Both laboratory and track tests were performed on this vehicle.

1.3.3 THETA-80 Prototype 100-Ton Covered Hopper Car

The THETA-80 car (Thrall Engineered Transportation Advancement for the 80's) is the Thrall Car Manufacturing Company's response to the challenge of meeting the railroad's demand for improved dynamic performance. The carbody of the THETA-80 has reduced weight through the use of light weight materials and efficient use of structure. The carbody is also configured for low center of gravity in both the empty and loaded conditions.

The modified three-piece trucks have a three-stage suspension designed to improve curving and hunting performance. High-conicity wheels are used to effect further improvement in curving. The secondary suspension has modifications that are intended to improve harmonic roll and bounce performance. More detailed descriptions of the THETA-80 can be found in [3] and [7].

Only track tests were performed on the THETA-80. Laboratory tests had been tentatively planned, but the Thrall company was satisfied with the track testing and chose not to continue with the laboratory tests.

Results of the THETA-80 track testing with comparisons to Base Vehicle No. 2 are presented in Reference [3].

1.3.4 Budd Prototype Covered Hopper Car

At the writing of this report, only preliminary information is available on the Budd Prototype vehicles. It is a two-body articulated vehicle with three, three-piece trucks. Detailed information will not be released by Budd until later in 1985.

2. TEST OBJECTIVES

The primary goal of the overall program is to achieve and demonstrate improved safety performance of covered hopper cars. The plan to meet this goal is to: (1) measure the dynamic properties and the safety performance of current design "base" 100-ton covered hopper cars; (2) measure the dynamic properties and safety performance of the two prototype cars that have been made available; and (3) make comparisons between the prototype and the base vehicle to show the degree of improvement achieved.

The program includes laboratory and track tests with all tests being performed at the TTC by AAR personnel. This report is limited to presenting results of laboratory testing of the base vehicles.

There were three general objectives to be accomplished in the laboratory tests of the base 100-ton covered hopper cars. The first was to determine their performance with respect to the following properties:

- derailment tendencies, with emphasis on harmonic roll,
- vehicle dynamic response to a variety of track conditions,
- impact of off-specification components, and
- impact of lading center of gravity height on derailment tendencies.

The second general test objective was to determine the dynamic characteristics of the 100-ton covered hopper car. This included quasi static testing on each truck to measure stiffness and damping, and modal testing of the complete vehicle to identify the major resonant mode frequencies.

The third objective of the tests was to evaluate a Cross Level Index (CLI) formulated by the TSC.

These primary objectives for the base vehicle laboratory tests support the following set of second level objectives:

- providing a basis for understanding relationships between vehicle dynamic properties and vehicle safety performance,

- defining the degree to which vehicle configuration changes cause changes in dynamic properties and safety performance, and

- providing a basis for understanding the performance of the prototype cars as to why improvements were or were not achieved and also providing a basis for deciding what changes to the prototype could result in additional improvements.

3. BARBER S-2-C TRUCK CHARACTERIZATION TESTS

Quasi static tests were performed on both Barber S-2-C trucks from the Base Vehicle No. 1. The testing consisted of applying loads at the center bowl and making measurements to obtain vertical, lateral and roll spring rates and associated friction forces. Vertical load conditions were varied to cover a range of vehicle gross weight conditions. Tests were also performed with Rail Dynamics Incorporated (RDI) hydraulic snubbers installed. The tests were conducted in the two-week period from April 26 through May 6, 1983, at the Transportation Test Center (TTC), Pueblo, Colorado.

3.1 Test Objectives

1. To determine the stiffness, in terms of spring constants, and friction forces of the test trucks.

2. To determine the effect of load variations on the truck stiffness and friction properties.

3. To determine effects of the friction casting variables, of one or two side springs per casting, and of dry or wet friction surfaces.

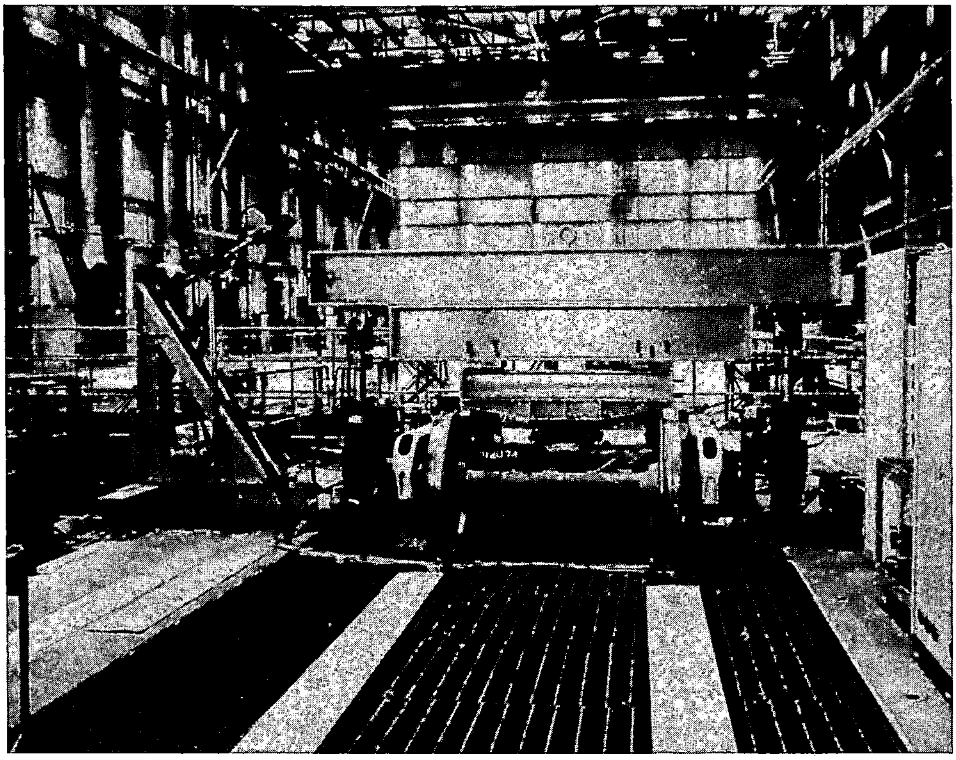
4. To investigate the unusual behavior of the A truck friction snubbers.

5. To investigate the apparent weakness of the right side of the B truck.

6. To evaluate the benefits of a hydraulic snubber.

3.2 Test Fixture and Instrumentation Description

The proposed test procedure was to support the truck assembly on rails, to apply loads through the carbody-to-truck interface points of center plate and side bearings, to measure applied forces and resulting displacements, and to subsequently obtain spring rates and friction forces. Using existing RDL facilities to a maximum practical extent, the test set up was located on the service pit in the RDL. The existing rails were used to support the truck, and the pit provided the space needed for the steel structure to support the vertical actuators. Figure 3-1 is a photograph showing the general arrangement of the truck, the fixture and actuators in the RDL.



View Looking East

FIGURE 3-1
TRUCK CHARACTERIZATION TEST FIXTURE

The wheels were blocked and clamped to the rails to prevent the truck from rolling longitudinally or sliding laterally on the rails. There was a tendency for the truck to move longitudinally, and it was necessary to occasionally realign the truck and loading fixture.

There were some welding failures at several points in the steel structure supporting the hydraulic actuators. The failures occurred in the course of testing and were evidenced by excessive displacement at the actuator attachments. The failed welds were repaired and additional steel plates and straps were welded at high stress points.

Measurement notation and location are defined in Table 3-1 and Figure 3-2.

3.3 Truck Configurations Tested

The series of tests covered in Section 3 were performed on the two trucks from Base Vehicle No. 1, the MoPac 100-ton covered hopper car number MP712074. VTU tests on this car were performed in December 1982 and January 1983, the results of which are presented in Section 5 of this report. The trucks, identified A and B according to their location on MP712074, were Barber Stabilized freight trucks, type S-2-C, with load variable friction furnished by the stabilizer friction casting. Some details of the trucks are given in Figure 3-3.

There were some peculiarities of these trucks noted during the vehicle tests.

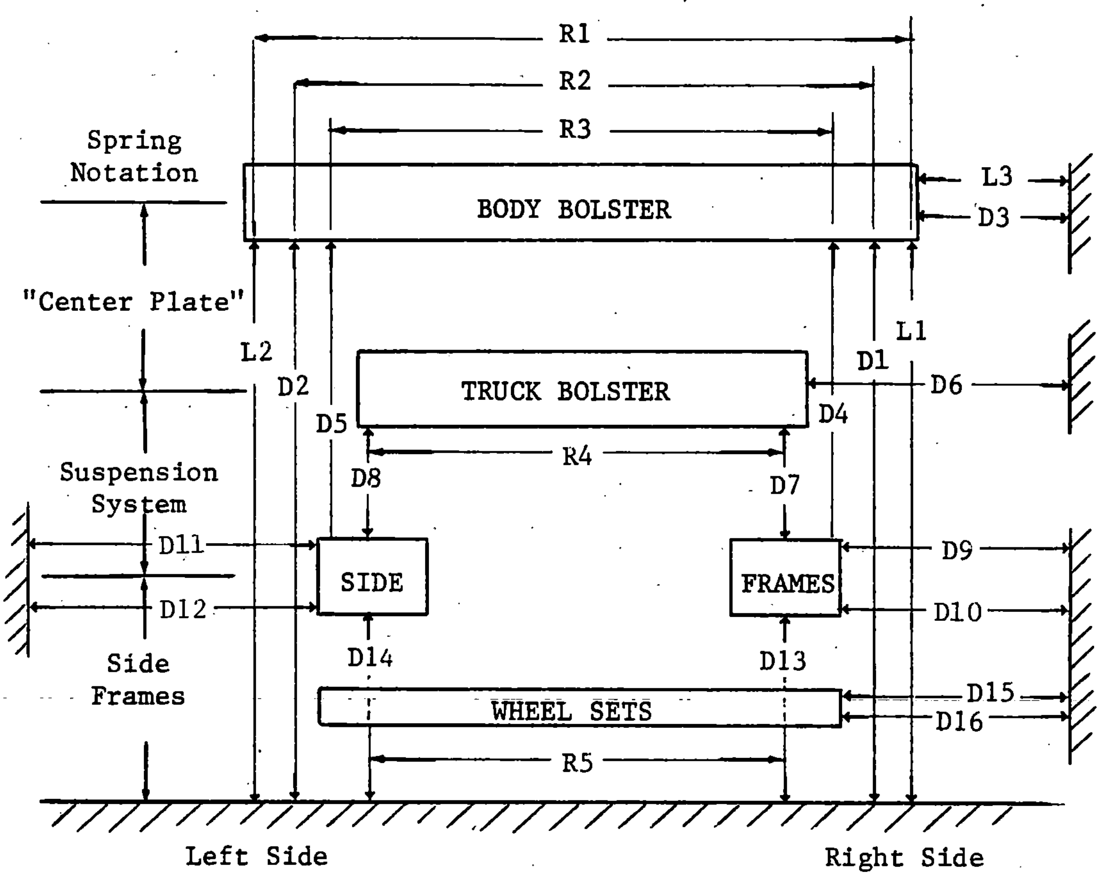
- The body bolster at the A end was improperly oriented in that the tapered edges were fore-and-aft instead of side-to-side. This is shown in the photograph of Figure 3-4.
- The A truck had the binding, high friction characteristic of the "new car syndrome" described by McLean and Weber [8].
- The B truck had no wear liner in the center bowl.
- In the assembled vehicle the B truck right side spring nest had larger displacement than the other three corners.

TABLE 3-1
TRUCK CHARACTERIZATION TEST INSTRUMENTATION
BARBER S-2-C TRUCKS

MEASUREMENT NUMBER	LOCATION	INSTRUMENTS
L1	Right Vertical Actuator Force, North Pegasus	Interface, 50 kip Load Cell, Ser # 16651
L2	Left Vertical Actuator Force, South Pegasus	Interface, 50 kip Load Cell, Ser # 16650
L3	Lateral Actuator Force	Lebow, 5.5 kip Load Cell, Ser # 4459
D1	Body Bolster to Gnd Vertical Right Side	Celesco 10" String Pot Ser # 10075
D2	Body Bolster to Gnd Vertical Left Side	Celesco 10" String Pot Ser # 10371
D3	Body Bolster to Gnd Lateral	Celesco 10" String Pot Ser # 10372
D4	Body Bolster to Side Frame, Vertical Right	Celesco 10" String Pot Ser # 10065
D5	Body Bolster to Side Frame, Vertical Left	Celesco 10" String Pot Ser # 10367
D6	Truck Bolster to Gnd Lateral	Celesco 10" String Pot Ser # 10430
D7	Truck Bolster to Side Frame, Vertical Right	Trans-Tek $\pm$ 1.75 in LVDT, Ser # 10826
D8	Truck Bolster to Side Frame, Vertical Left	Trans-Tek $\pm$ 1.75 in LVDT, Ser # 00323
D9	Side Frame to Gnd, Top, Lateral, Right	Celesco 10" String Pot Ser # 10156
D10	Side Frame to Gnd, Bottom, Lateral, Right	Celesco 10" String Pot Ser # 10376

TABLE 3-1
TRUCK CHARACTERIZATION TEST INSTRUMENTATION
BARBER S-2-C TRUCKS
(Concluded)

MEASUREMENT NUMBER	LOCATION	INSTRUMENTS
D11	Side Frame to Gnd, Top, Lateral, Left	Celesco 10" String Pot Ser # 10081
D12	Side Frame to Gnd, Bottom, Lateral, Left	Celesco 10" String Pot Ser # 10067
D13	Side Frame to Gnd, Vertical, Right	Celesco 10" String Pot Ser # 10368
D14	Side Frame to Gnd, Vertical, Left	Celesco 10" String Pot Ser # 10076
D15	Front Wheel Set to Gnd, Lateral	Celesco 10" String Pot Ser # 10074
D16	Aft Wheel Set to Gnd, Lateral	Celesco 10" String Pot Ser # 10370



R1 = 127.5 = Distance between vertical actuator center lines (L1, L2).
R2 = 110.75 = Distance between measurements D1 and D2.
R3 = 78.75 = Distance between D4 and D5.
R4 = 99.75 = Distance between D7 and D8.
R5 = 90.25 = Distance between D13 and D14.

All distances in inches.

FIGURE 3-2
TRUCK SCHEMATIC SHOWING MEASUREMENT AND HYDRAULIC ACTUATOR LOCATIONS AND SPRING NOTATION, BARBER S-2-C TRUCKS

Truck Characterization Data

16 inch diameter center bowl

Constant contact resilient side bearings (also tested with double roller)

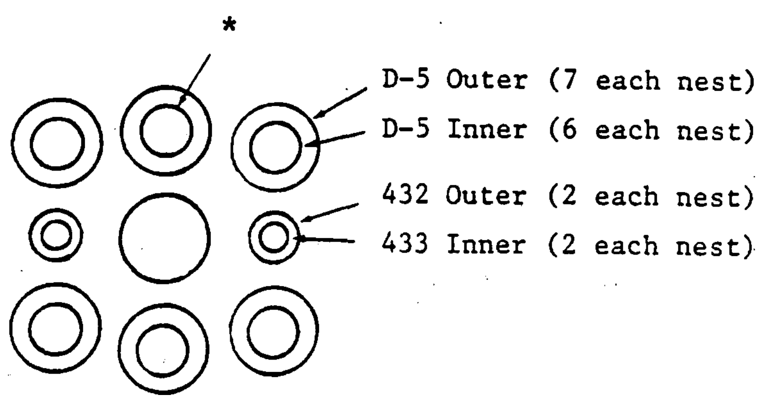
D-5 springs

Double side springs

6 1/2 x 11 inch journals

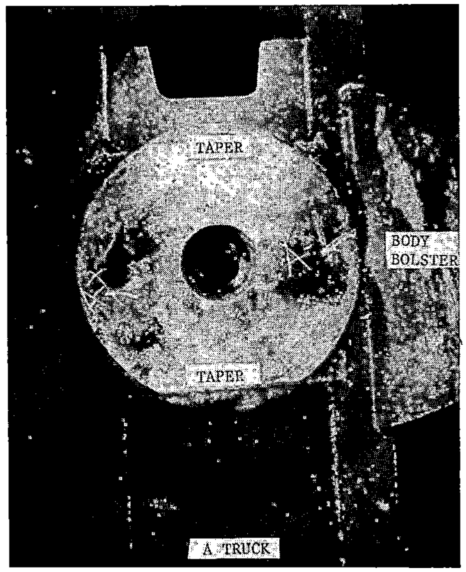
36 inch diameter wheels

Spring Nest Configuration



* The nest on the right side of the B truck has a D-6 inner spring at this location in place of a D-5 inner.

FIGURE 3-3
THE BARBER STABLIZED FREIGHT TRUCK, TYPE S-2-C, CONFIGURATION DATA



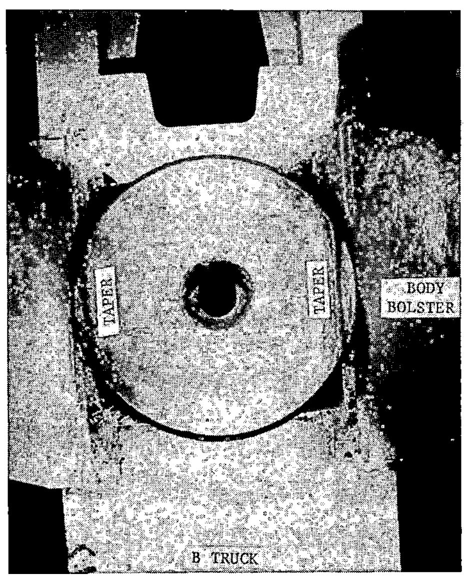


FIGURE 3-4
BASE VEHICLE NO. 1 BODY BOLSTERS SHOWING DIRECTION OF TAPER

There were a total of 14 configurations tested. The tests performed, the sequence of testing, run numbers, and configuration descriptions are listed in Tables 3-2 and 3-3.

3.4 Test Procedures

The general test procedure divided the testing into the three separate cases of vertical, lateral and roll as indicated in Table 3-2. In each case, loads were applied by the hydraulic actuators to simulate car weight conditions from empty to maximum actuator capacity, which was less than full load equivalence. (For example, 55,000 pound maximum capability in each vertical actuator is the equivalent of about an 88-ton load condition.)

In the vertical and roll tests, the lateral actuator was disconnected. In all tests the loads were applied in a 0.2 Hertz sinusoidal variation with the one exception that when testing with the RDI hydraulic snubbers, the cycling was done at 0.5 Hertz. A cycling motion was used in order to avoid the errors introduced by the difference in sliding and breakaway friction if a step load variation had been used. The 0.2 Hertz rate of cycling was used since this was judged to be low enough so as not to introduce any dynamic effects. The 0.5 Hertz cycling rate was used in evaluating the hydraulic snubbers since their force output is velocity dependent and since 0.5 Hertz is representative of the hopper car first roll mode. In the lateral test, the vertical actuators were held to a fixed displacement at a neutral condition load corresponding to the desired lading conditions while the lateral actuator was cycled at 0.2 Hertz. For the lateral cases where the friction snubber breakaway force was greater than the lateral actuator capability (5500 lb), a 5.0 Hertz motion was superimposed by the vertical actuators to get the friction snubbers moving by a "dither" condition.

Data recording consisted of real-time X-Y plotting of D1 vs L1 and D2 vs L2 for the vertical and roll cases and D3 vs L3 for the lateral cases and of recording of all measurements on digital tape. The X-Y plotter was located on the test floor and was used to verify that desired conditions were obtained before recording. The general procedure followed was to set desired load conditions on the actuator control panel, verify conditions with the X-Y plots and record several cycles of motion on the plotter and on tape.

TABLE 3-2
TEST MATRIX FOR TRUCK CHARACTERIZATION TESTS
BARBER S-2-C TRUCKS

CONFIGURATION	1		2			3	4
TEST AND	DRY	WET	DRY	WET	HYD	---	DRY
LOAD RANGE	A & B	B	B	B	B	A & B	B
Vertical Test							
20-110 kip total	x	x	x	x	x	x	x
Lateral Test							
± 5.5 kip lat. with:							
a. 100 kip Vert. total	x		x		x	x	x
b. 60 kip Vert. total	x					x	
c. 24 kip Vert. total	x		x		x	x	x
Roll Test							
a. 40 ± 15 kip	x	x	x		x	x	x
b. 30 ± 25 kip	x					x	x
c. 25 ± 20 kip	x	x	x		x	x	x
(each actuator)							

Configurations: 1 - Roller side bearings, normal friction (double side springs)
2 - Roller side bearings, reduced friction (single side springs)
3 - Roller side bearings, no friction
4 - Constant contact side bearings, normal friction

Notes:
- A and B refer to truck A and truck B.
- Normal Friction: friction casting with 432 outers and 433 inners. Reduced Friction: friction castings with 432 outers only. No Friction: friction castings removed.
- The A truck was tested with original castings and center plate in correct orientation and then check runs were made using a replacement set of castings and with the center plate rotated.
- The B truck tests were without center plate liner except for check runs (see Table 3-3).

TABLE 3-3
SEQUENCE OF BARBER TRUCK CONFIGURATIONS TESTED

SEQUENCE NO.	RUN NO.	CONFIGURATIONS
1	1 - 8	Test check out runs
2	9 - 18	B truck (1), Config. 1, Dry
3	19 - 20	" , " , Wet
4	21 - 27	" , Config. 2, Wet
5	28 - 32	" , " , Dry
6	33 - 46	" , " , Dry,w/RDI HS(2)
7	48 - 54	" , Config. 3,
Side bearing gap measured: 0.255", right 0.125", left		
8	55 - 59	B truck w/wear liner, Config. 1, Dry
9	60 - 67	" " , Config. 4,
Failed welds in actuator support structures were repaired.		
10	68 - 74	A truck, Config. 3
Side bearing gap measured: 0.279", right 0.298", left		
11	75 - 77	A truck, original castings, Config. 1, dry
12	78 - 82	" , "New" castings, Config. 1, dry
13	83 - 88	" , original casting, Config. 1, dry
14	89 - 92	" , " " , " , dry center plate rotated to have taper fore & aft
Side bearing gap measured: 0.327", right front 0.264", right rear 0.270", left front 0.248", left rear		

(1) B truck test was without center plate wear liner unless noted otherwise.
(2) Railroad Dynamics, Inc., MDA Control/Master hydraulic snubber.

3.5 Summary Observations

The quantitative results from the truck testing are presented in Section 3.6. However, there are several general qualitative observations which should be made and which will help explain some of the data spread shown in 3.6. These are listed below.

1. The right side spring nest in the B truck had the appearance of being defective (when assembled with the vehicle) by virtue of larger static deflections than the other three spring nests. However, in the truck characterization testing this difference was not apparent; all four spring nests were essentially equal. This would indicate that there was probably a misfit or obstruction in the carbody center plate-to-bolster center bowl interface that did not permit proper mating in the vehicle assembly.

2. The binding and grabbing of the friction castings in the A truck were concluded to be due to the position and orientation of the wear plates on the side frames. This condition is not uncommon with reconditioned trucks. Photographs of the friction surfaces of the castings in this truck are shown in Figure 3-5. The wear pattern indicates that the castings were not flat against the side frame bearing plate and that there was heavy gouging.

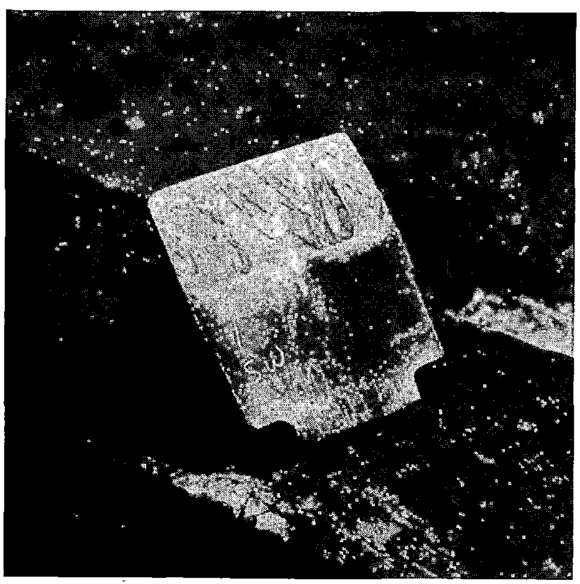
3. A new set of friction castings was used in the A truck for a series of roll and vertical tests. The new castings worked smoothly for a short time and then began to bind and grab much like the old castings. Upon removal and inspection, the new castings evidenced the same general uneven wear and gouging pattern found in the original casting. Photographs of the new casting are shown in Figure 3-6.

4. Water on the friction casting bearing surfaces resulted in a significant reduction of the friction forces. Only a small amount of moisture was needed to effect this change.

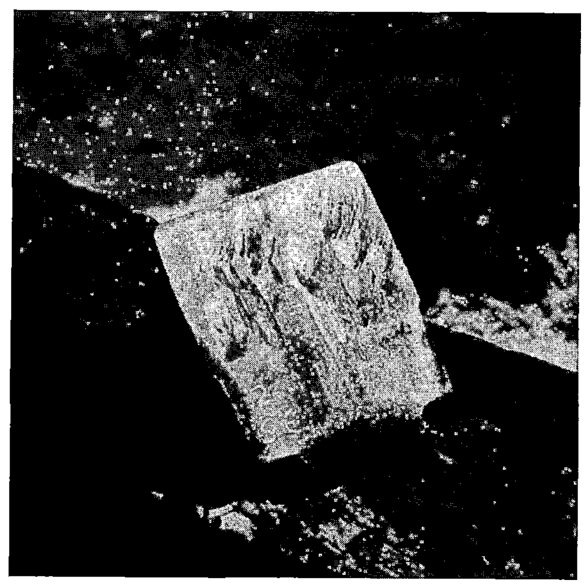
5. The effects of some of the test configuration variations were too subtle to quantify. These variations include: (1) 90° reorientation of the center plate tapered edges; (2) the use of the RDI hydraulic snubbers; (3) the B truck with and without center bowl wear liner; and (4) constant contact side bearings.

Right Side Front

Right Side Rear



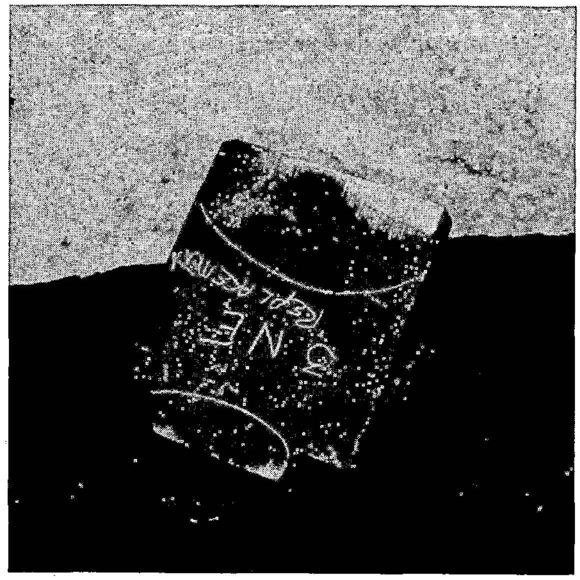
Left Side Front



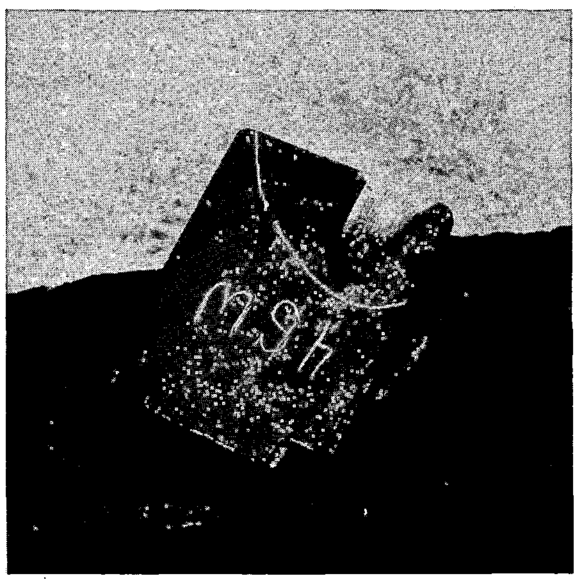
Left Side Rear

FIGURE 3-5
FRICTION CASTINGS FROM THE A-END TRUCK OF BASE VEHICLE NO. 1 SHOWING UNEVEN WEAR AND GOUGING

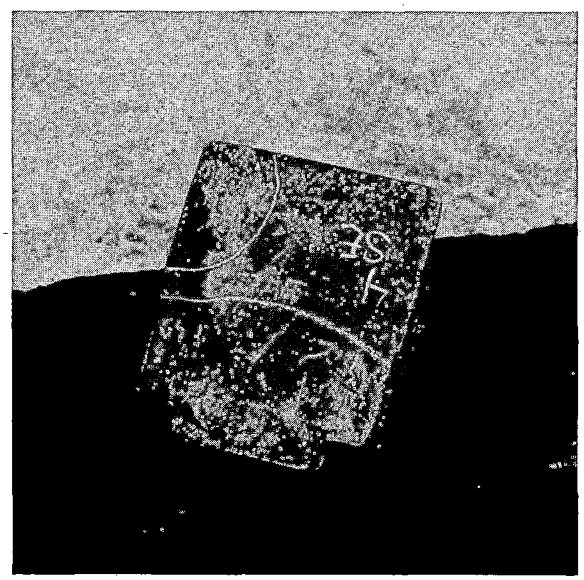
Right Side Front



Right Side Rear



Left Side Front



Left Side Rear

FIGURE 3-6
NEW FRICTION CASTINGS USED IN CHARACTERIZATION TEST OF THE A-END TRUCK, BASE VEHICLE NO. 2

3.6 Test Results, Barber Truck

The pervasive nonlinear characteristics of the three-piece freight car truck dictate that single valued properties cannot be assigned. Instead properties are best given in terms of variables such as lading weight, spring nest displacement, whether the center plate is seated or rocking, and whether or not side bearing contact has been made. Consequently, the stiffness and friction force properties are presented as variables plotted against vertical load in most cases. A tabulation of results, for empty and 90% load conditions, is also presented as a summarization.

The results are presented according to the test load conditions: vertical, lateral, and roll.

In the analysis process, the data was first output in the form of load/deflection plots which had the typical hysteresis loop associated with friction damping. Even with the friction snubbers removed, the hysteresis shape was present. Spring rates were determined from the slopes of the load/deflection paths. Damping force was determined from the height of the hysteresis loop.

3.6.1 Vertical Test Results

A tabulated summary of results is given in Table 3-4 for spring rates for the two load conditions of empty and 90%. Table 3-5 summarizes friction snubber forces also for the same two load conditions. Spring rate and friction force results are also plotted against vertical applied load in Figures 3-7 through 3-10.

Several things should be noted in the data:

- First, there is a significant difference between the A and B trucks: truck A is 10% to 25% stiffer than truck B, and truck A friction force is 16% to 74% greater than truck B.

- Second, the suspension system effective stiffness is increased by the friction snubber. In truck B the increase is between 8% and 9%. In truck A it runs between 26% and 46%.

- Third, there is significant difference both in stiffness and friction force between the empty and full conditions.

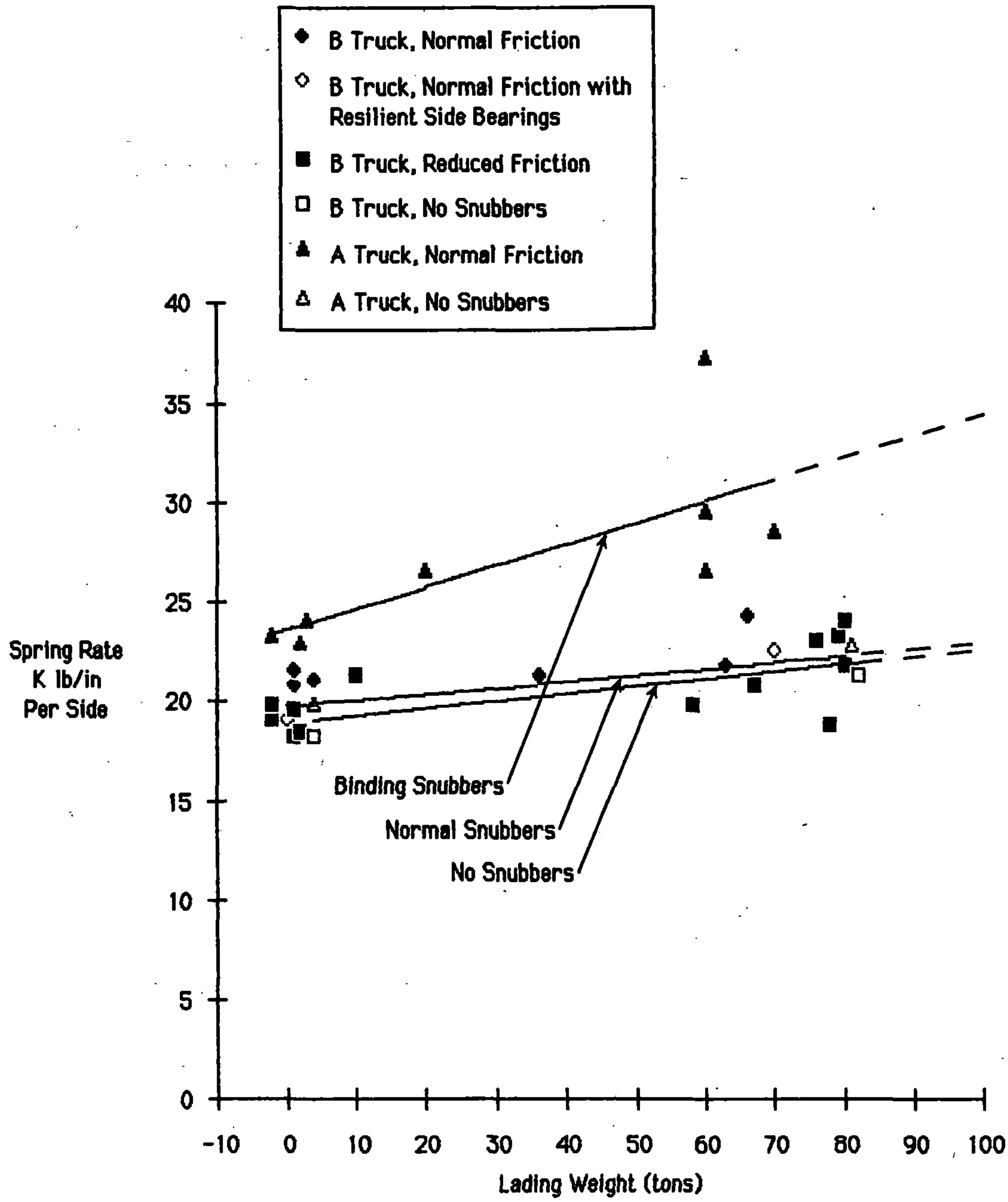
TABLE 3-4
VERTICAL SPRING RATES, BARBER S-2-C TRUCK,
TRUCK CHARACTERIZATION TEST RESULTS

TRUCK COMPONENT	LOAD CONDITION EMPTY K LB./IN. PER SIDE	LOAD CONDITION 90% K LB./IN. PER SIDE
From no-snubber tests, Averaged A & B trucks		
Overall (body bolster to ground)	18.7	21.5
Center Plate (body bolster to truck bolster)	230.0	230.0
Secondary Suspension (Truck bolster to side frame)	21.0	24.8
Suspension and Center plate	19.2	22.4
Side frames to Ground	527.0	647.0
A Truck with snubbers		
Overall (increase due to snubbers)	23.6 (26%)	32.0 (49%)
Secondary Suspension	27.7	39.4
B Truck with snubbers		
Overall (increase due to snubbers)	20.4 (9%)	24.4 (13%)
Secondary Suspension	23.4	28.5
B Truck with snubbers and resilient side bearings		
Overall	19.2	23.5
Secondary Suspension	21.8	27.3

TABLE 3-5
SNUBBER FORCE FROM VERTICAL TEST DATA
BARBER S-2-C

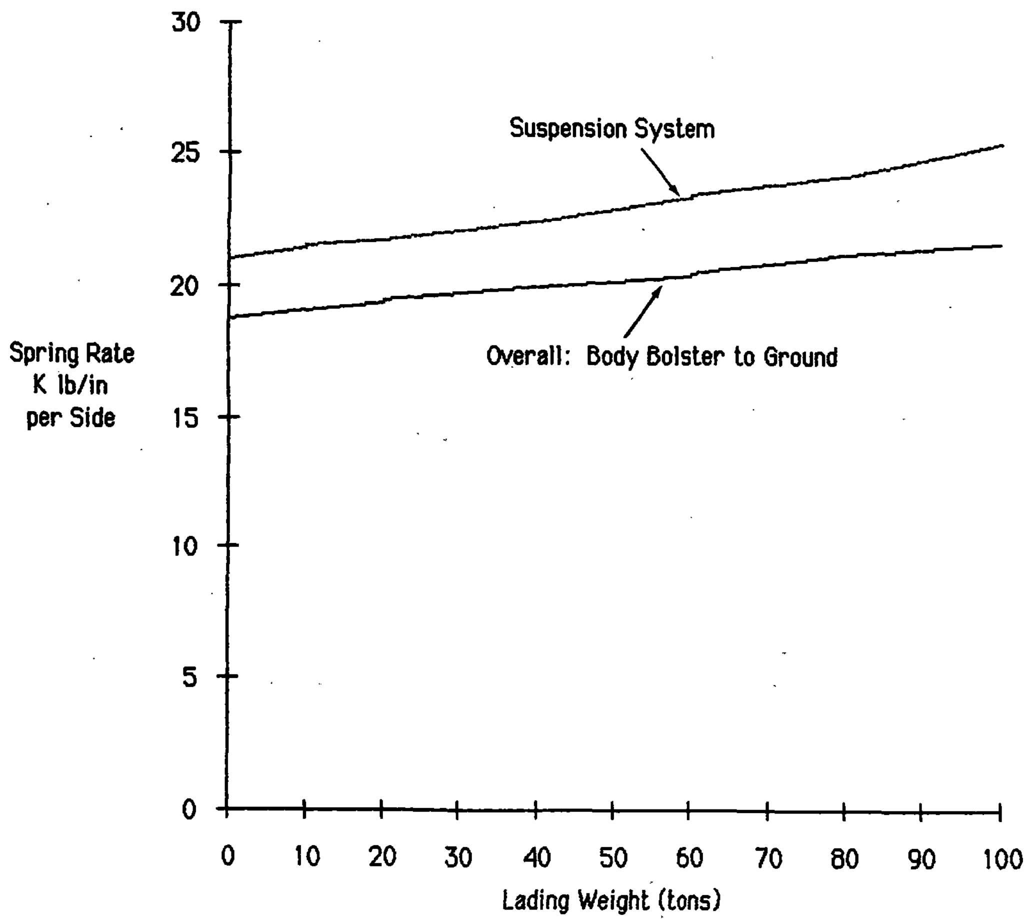
TRUCK AND CONDITION	LOAD CONDITION EMPTY, POUNDS, PER SIDE	LOAD CONDITION 90%, POUNDS, PER SIDE
A TRUCK		
Double side spring, dry	3,300	13,300
No snubber	350	500
B TRUCK		
Double side spring, dry	2,850	7,800
Double side spring, wet	2,150	4,820
Single side spring, dry	1,600	4,400
Single side spring, wet	1,400	3,200
No snubbers	350	500
Hydraulic Snubber Max force at 0.5 Hz.	14,500	

Note: Friction snubber force is taken as half the height of the hysteresis loop. Hydraulic snubber force is on the down stroke only; consequently, its force is the full height (less friction snubber force.)



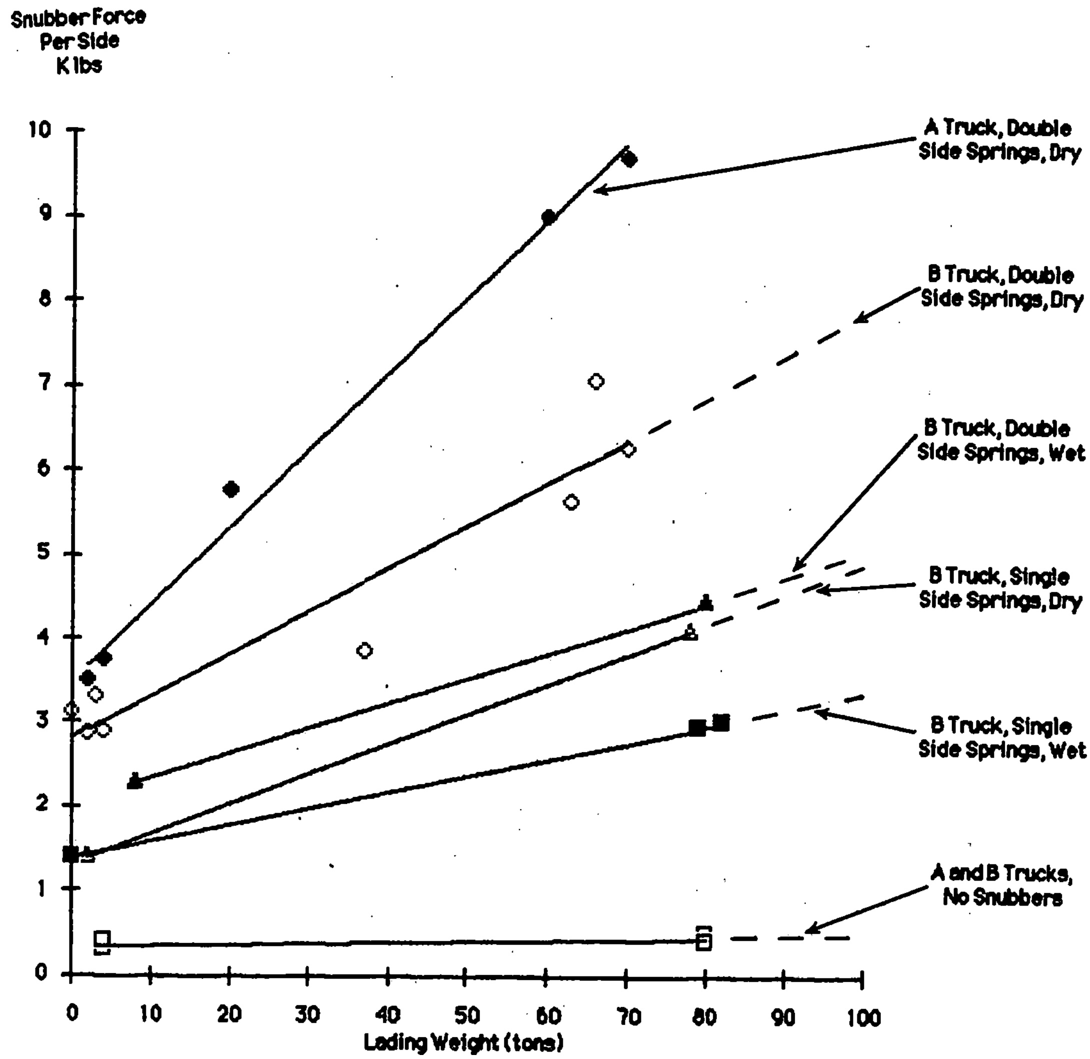
Obtained using D1 and D2, the displacements between body bolster and ground.

FIGURE 3-7
BARBER S-2-C TRUCK, OVERALL VERTICAL STIFFNESS, PER SIDE



Based on data from B truck tests.

FIGURE 3-8
BARBER S-2-C TRUCKS
VERTICAL SPRING CONSTANTS, PER SIDE, FROM NO-SNUBBER TESTS



Values are hysteresis loop half height.

FIGURE 3-9
BARBER S-2-C TRUCKS
FRICTION FORCE VARIATION, VERTICAL TESTS, PER SIDE

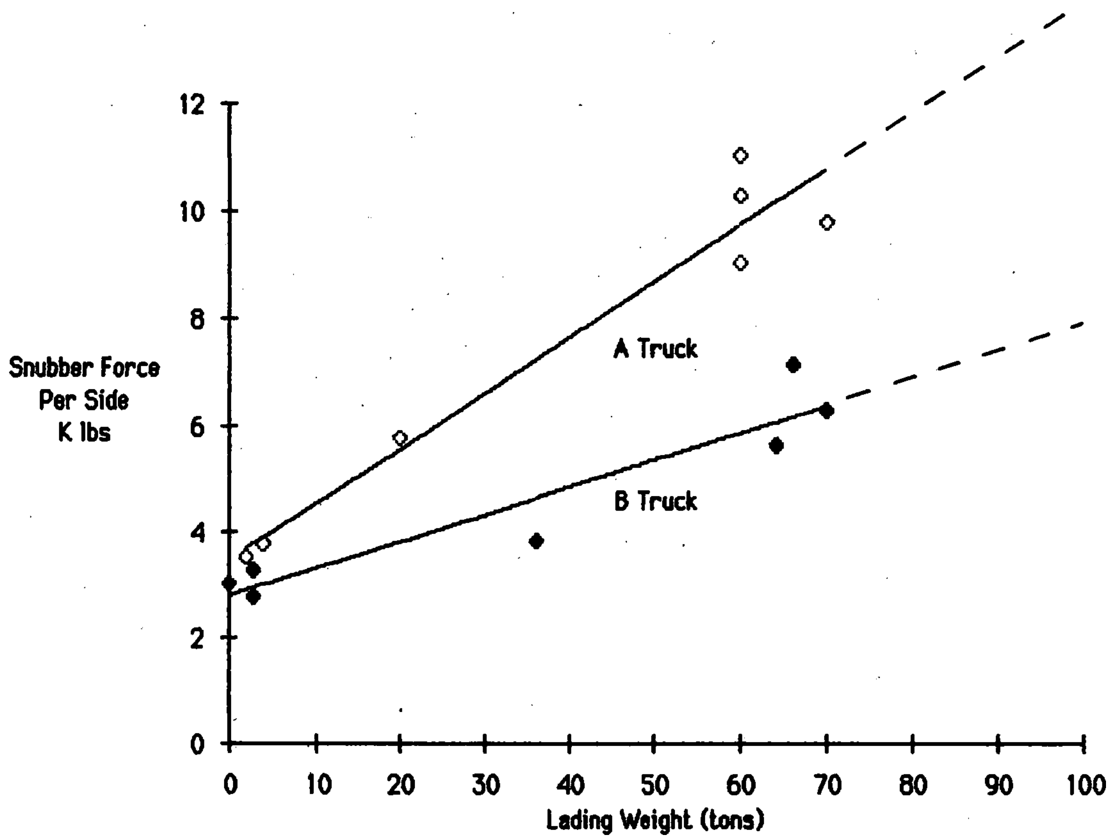


FIGURE 3-10
EFFECT OF BINDING FRICTION CASTINGS
BARBER S-2-C TRUCK WITH DOUBLE SIDE SPRINGS

- Fourth, moisture on the snubber friction surfaces resulted in reduced friction force ranging from 60% to 90% of dry friction.

The snubbing force provided by the RDI hydraulic snubber tested is also listed in Table 3-5 and plotted in Figure 3-11. As the plot in Figure 3-11 shows, the hydraulic snubber does not begin to act until the height of the spring nest is about 9.0 inches and then reaches a maximum value of about 14,500 pounds. The maximum velocity of the piston stroke, based on the 1.7-inch displacement at 0.5 Hertz used in the test, was 2.67 inches per second.

3.6.2 Hydraulic Snubber

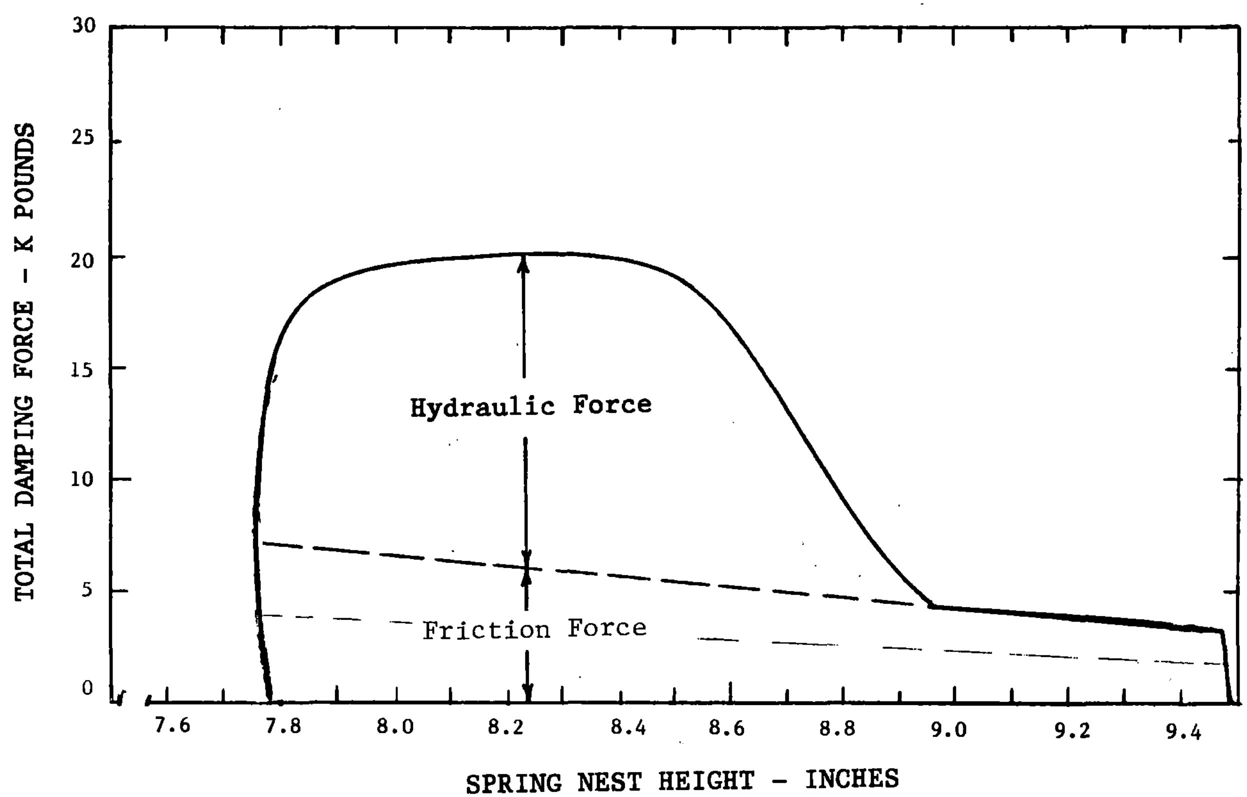
The hydraulic snubbers used in these tests were manufactured and supplied by Railroad Dynamics, Inc., designated as MDA Control/Master model no. D-5 standard units. The combined friction and hydraulic snubber force variation with stroke, obtained from load/deflection data, is shown in Figure 3-11. The hydraulic force peaks out at about 14,500 pounds. This occurred with a 0.5 Hertz input at a stroke of 1.7 inches which, if the motion were a pure sinusoid, would result in a peak velocity of 2.7 inches per second. Figure 3-12 has this point plotted on the hydraulic snubber's performance curve. The likely reasons the 14,500 pound data point is high relative to the performance curve are: (1) the manufacturer's performance curve is probably a minimum and (2) the actuator motion was not a true sinusoid and the peak velocities were probably closer to 4.0 in./sec.

The results show that the hydraulic snubber is inactive for light weight conditions and small motions (by design) but can absorb large amounts of energy for loaded vehicles with large motions in the spring nest.

3.6.3 Lateral Test Results, Barber Truck

The lateral stiffness of the three piece truck can be divided into the three sections of: (1) the spring nest; (2) what is above the spring nest; and (3) what is below it. These three sections are, in this report, referred to as the center plate, the suspension system, and the side frames as a convenience, it being understood that each includes flexibility due to other parts such as body and truck bolsters, journals, and wheel sets.

A summary tabulation of these three "springs" is given in Table 3-6 for empty and 90% load conditions. The side frame and suspension spring constants increase with lading weight:



From vertical test with 0.5 Hertz input motion

FIGURE 3-11
DAMPING FORCE OF HYDRAULIC SNUBBER, TRUCK B
WITH REDUCED FRICTION SNUBBER

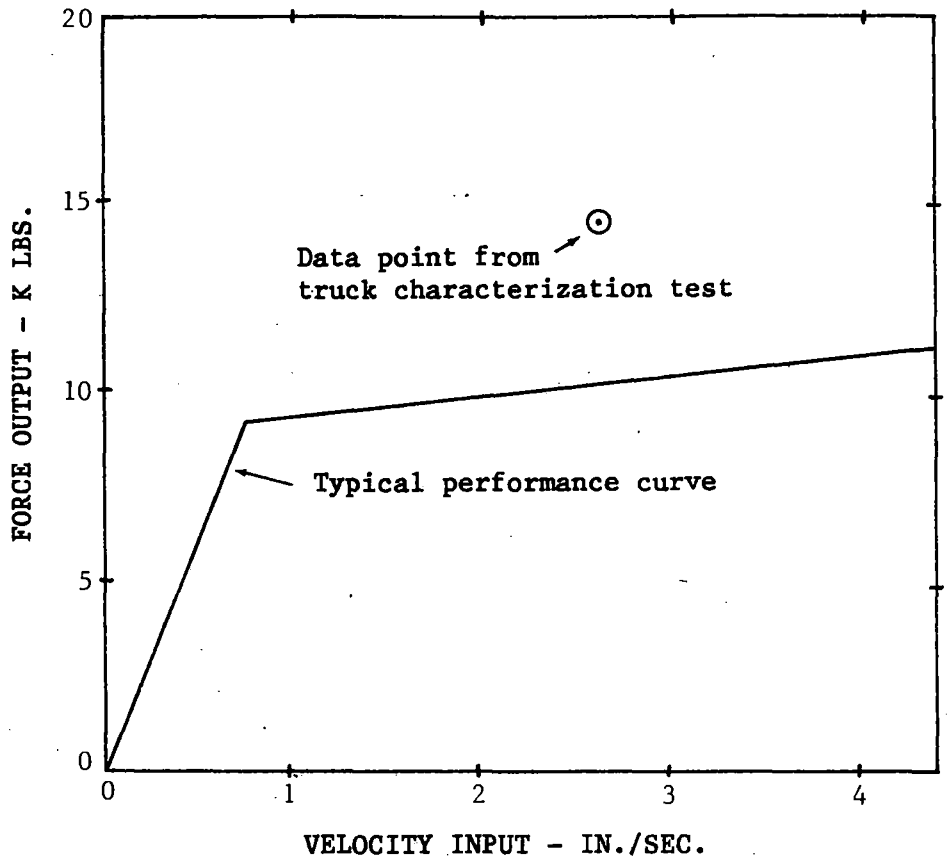


FIGURE 3-12
FORCE OUTPUT OF HYDRAULIC SNUBBER

TABLE 3-6
LATERAL SPRING RATES, BARBER S-2-C TRUCK

TRUCK COMPONENT	LOAD CONDITION	
	EMPTY	90%
Center plate (body bolster to truck bolster)	169.0*	169.0*
Suspension System (truck bolster to side frame)	20.0	70.0
Suspension System and Center plate	17.9	49.5
Side Frame	118.0	160.0
Suspension System with no snubbers	16.0	27.0

* Units are K lb./in. per truck

the side frames from 118 to 168 K lb./in., a 42% change, and the suspension from 20 to 70 K lb./in., a 250% change. This unusually large change in the lateral suspension system is, as will be shown, due to the action of the load variable snubbers. The lateral stiffness of the center plate is given as a constant 169.0 K lb./in.; however, the data had enough scatter to hide possible variations with load.

The load/displacement plots for the lateral suspension system were typically variations of the four example plots shown in Figure 3-13. In this figure, there are four different kinds of motion identified by the following labels:

A. load change without deflection,

B. load/deflection with snubbers not sliding,

C. load/deflection with snubbers sliding, and

D. deflection change without load change (a test peculiar condition).

The diagonal lines labeled "E" were used to determine effective stiffnesses that were various combinations of Types A, B, and C.

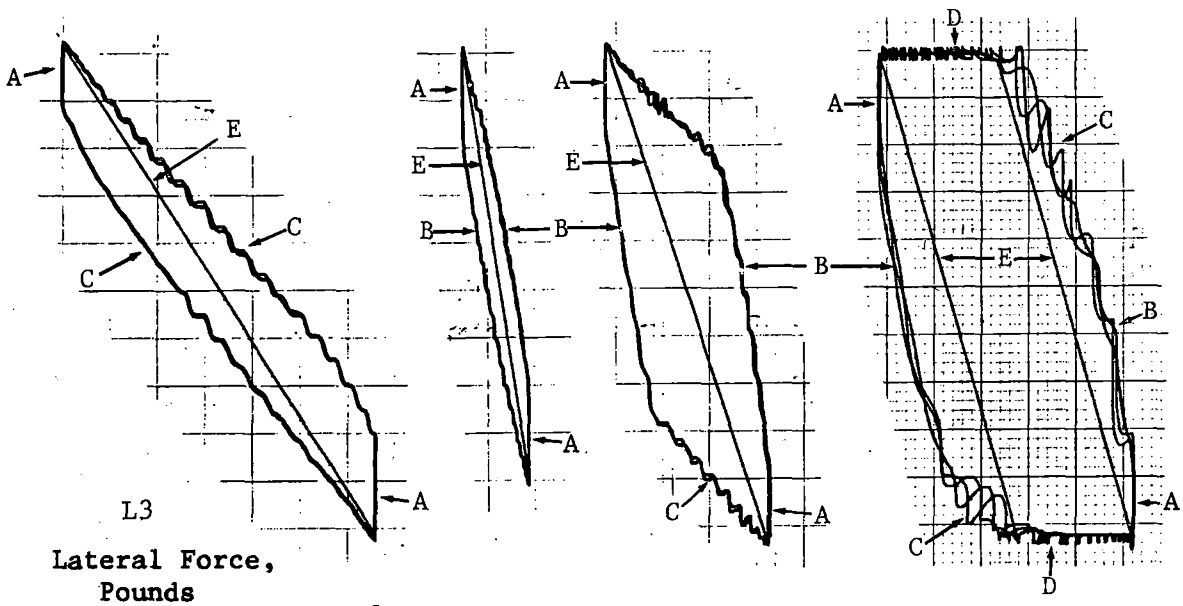
Analysis of the lateral suspension system in this manner resulted in the three curves, shown in Figure 3-14, for (1) without snubbers; (2) with a range of snubbing and the snubbers sliding; and (3) with the snubbers locked. The curves are shown as bands due to the data scatter.

It is important to note, in looking at the results shown in Figure 3-14, that the lateral effective spring in the three-piece truck with load variable snubbers is very non-linear, being largely dependent on lading weight.

Another factor not obvious in Figure 3-14 is that whether the snubber is locked or sliding is also dependent on the magnitude of the dynamic forces, both lateral and vertical. That is, with small dynamic forces the lateral suspension will appear to be stiffer and with larger dynamic forces (lateral or vertical) will appear to be less stiff.

The side frame and the center plate lateral spring rates are shown in Figure 3-15 and 3-16 in plots of spring rate against lading weight. The side frame stiffness increases slightly with load.

1. No Snubbers 2. Reduced Friction Snubbers 3. Full Friction Snubbers



Lateral Force, Pounds

D3

Lateral Displacement, Inches

Explanations

A Load build-up before displacement starts at the beginning of return stroke. Typical of three-piece truck.

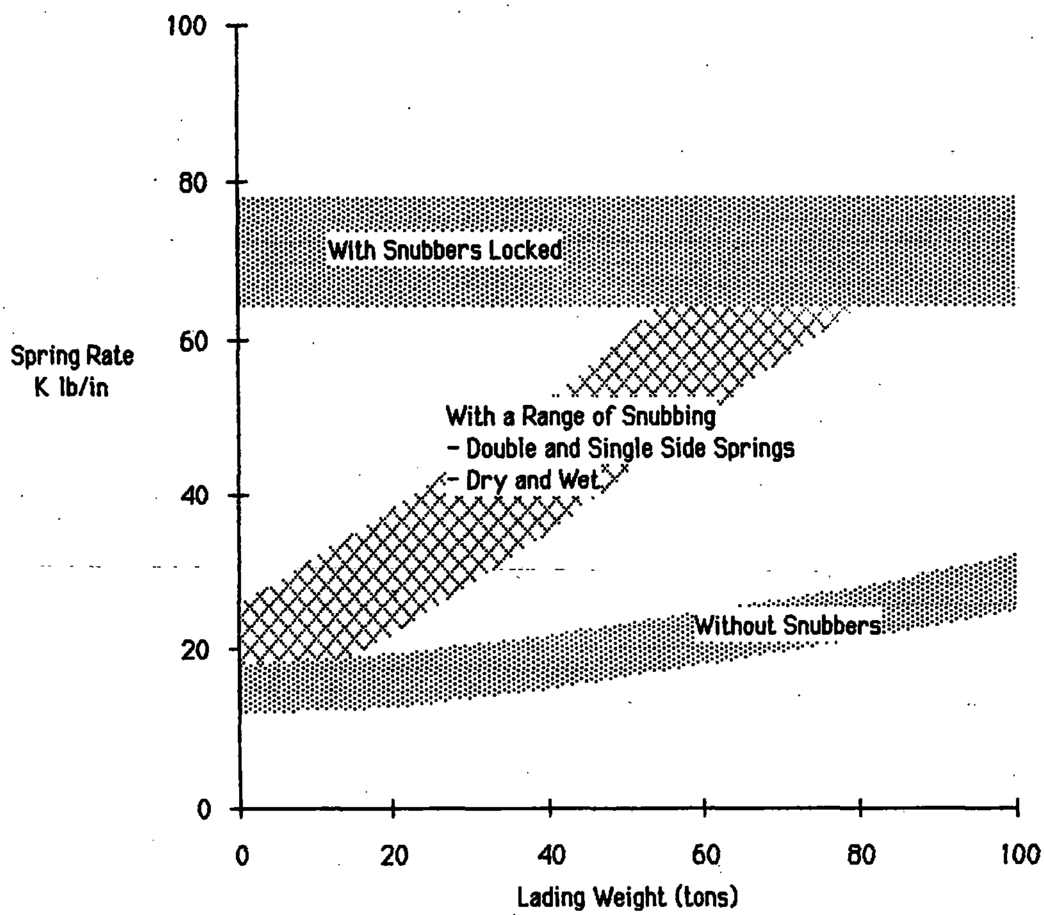
B Friction snubbers prohibits relative motion between bolster and sideframes. Stiffness is of truck structure.

C Relative motion exists between bolster and sideframes. Stiffness is dependent on truck secondary suspension system.

D Test peculiar condition. Force limit of lateral actuator is reached but lateral displacement continues when vertical force variations (5.0 Hz.) overcome friction snubber force.

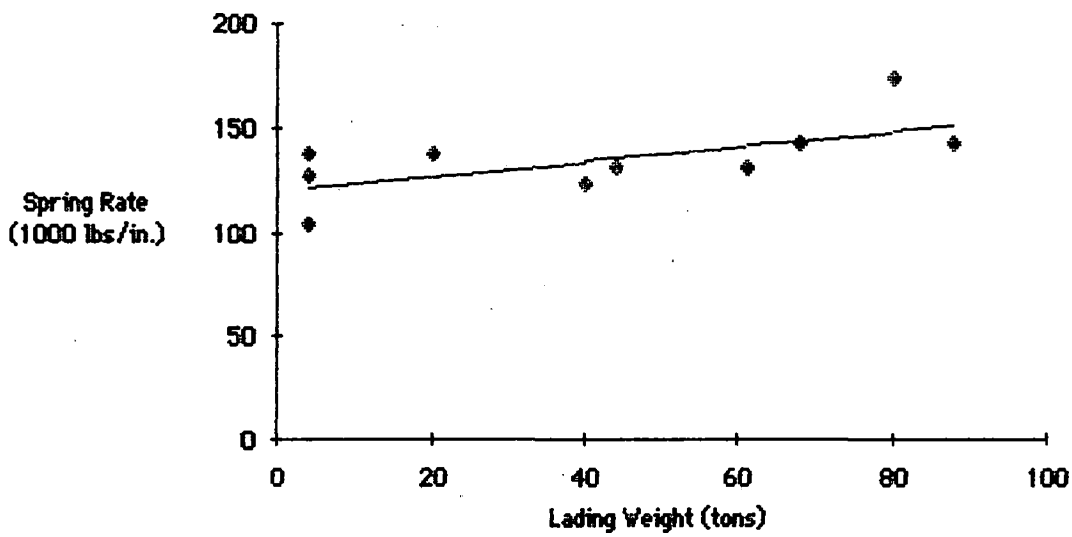
E Diagonal lines for determining effective combined stiffness.

FIGURE 3-13
LOAD DEFLECTION DIAGRAMS TYPICAL OF LATERAL TESTS



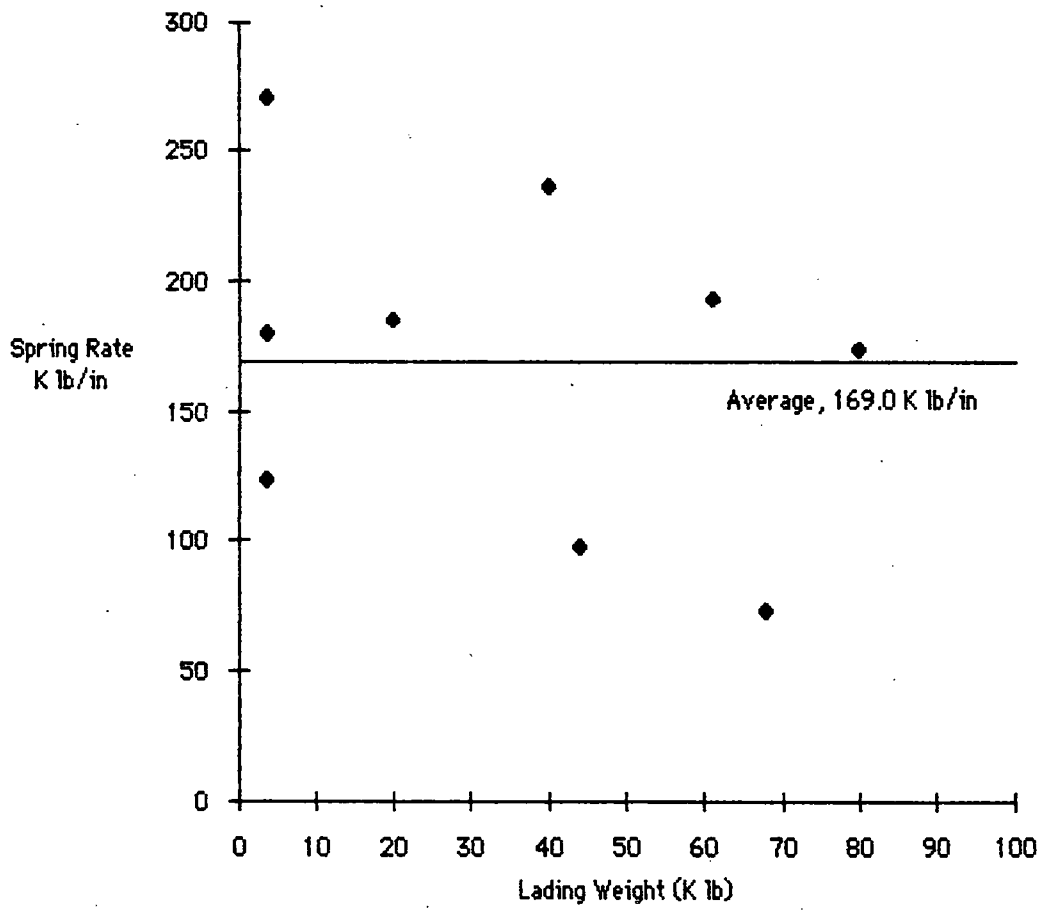
(Based on diagonal lines "E" shown in Figure 3-13)

FIGURE 3-14
BARBER S-2-C TRUCK SUSPENSION SYSTEM LATERAL SPRING RATES PER TRUCK



Side frame to ground, B truck data

FIGURE 3-15
BARBER S-2-C TRUCK, SIDE FRAME LATERAL SPRING RATES PER TRUCK



Body bolster to truck bolster, B truck data

FIGURE 3-16
BARBER S-2-C TRUCK, CENTER PLATE LATERAL SPRING CONSTANT PER TRUCK

There is an unusual amount of scatter in the center plate lateral stiffness data which may be due to experimental error but more likely is due to the non-linear nature of the three-piece truck. The 169 K lb./in. average value shown in Figure 3-17 has an estimated ±20% variability associated with it.

3.6.4 Roll Test Results, Barber Truck

The non-linear characteristics of the three-piece truck are most apparent in the roll stiffness properties. There are four positional conditions that cause changes in roll stiffness: (1) when the center plate is seated; (2) when the center plate is rocking; (3) when there is side bearing contact; and (4) when the body bolster has lifted off the center plate and the only contact is at the side bearing.

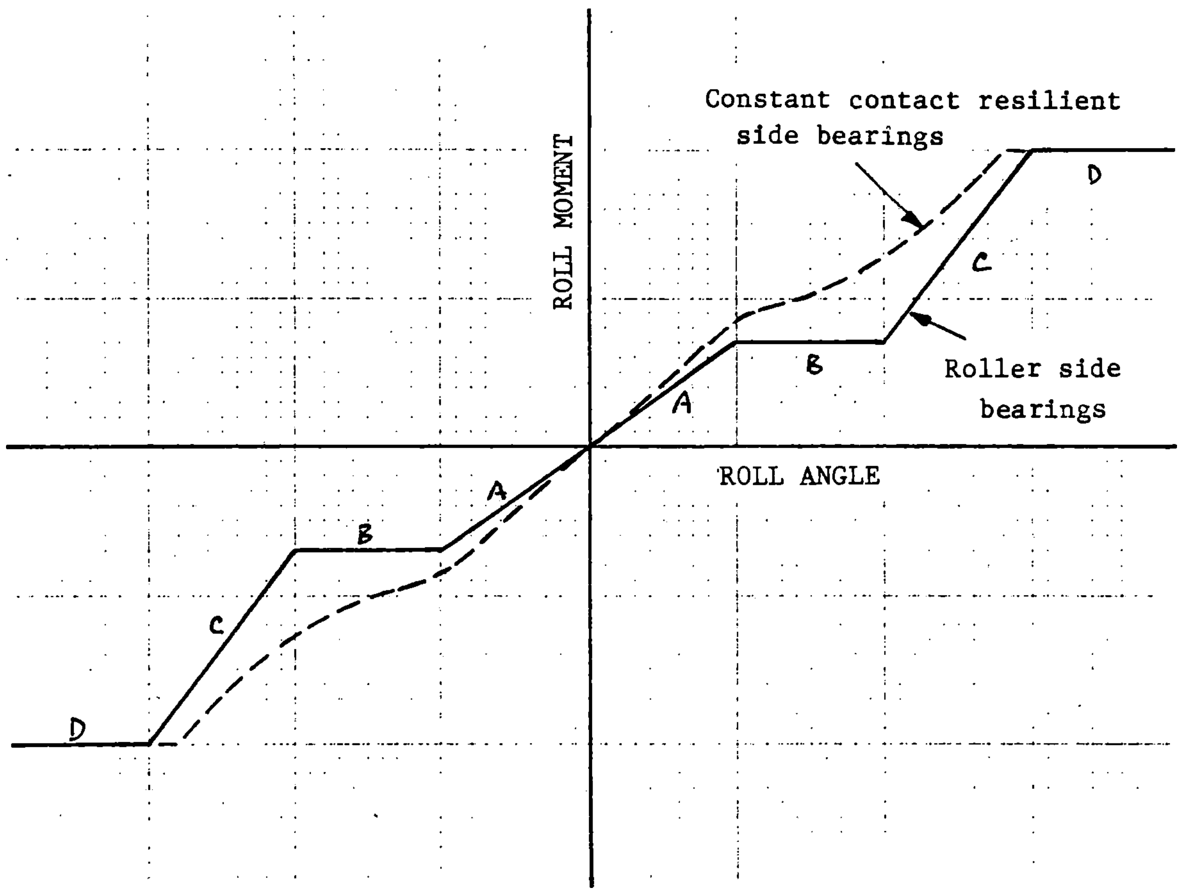
For most normal conditions, the rolling motions of a freight car will be in the amplitude range where the center plate will be seated, with occasional larger amplitudes where the center plate rocks but side bearing contact is not made.

Under conditions of unusually large motions, side bearing contact will be made. The fourth condition of body bolster lift can occur when the vehicle is empty. (With loaded vehicles, the weight and c.g. conditions are such that wheel lift will generally occur before body bolster lift.)

This very non-linear behavior of the three-piece truck in roll is illustrated in the moment/deflection diagram in Figure 3-17. The regions B and D are where there is rotation about a pivot--for B this pivot is the edge of the center plate, for D the pivot is the side bearing--in which case the only restoring moment is the weight of the vehicle times the horizontal distance between the body c.g. and the pivot point. The effective spring constant is assumed zero.

With constant contact resilient side bearings, region A stiffness is slightly higher because the side bearings are carrying some load, region B no longer has zero spring rate and region C is slightly softer because the resilient side bearing is in series with the main suspension springs.

The Barber truck roll test results are summarized for both with roller and constant contact resilient side bearings in Figure 3-18 for the overall truck and Figure 3-19 for the secondary suspension. The secondary suspension data is also given in Table 3-7 for empty and 90% load conditions.



A = center plate seated
B = center plate rocking
C = side bearing in contact
D = body bolster lifted at center plate

FIGURE 3-17
HYPOTHETICAL MOMENT/ANGULAR DEFLECTION DIAGRAM FOR A THREE-PIECE TRUCK IN ROLL

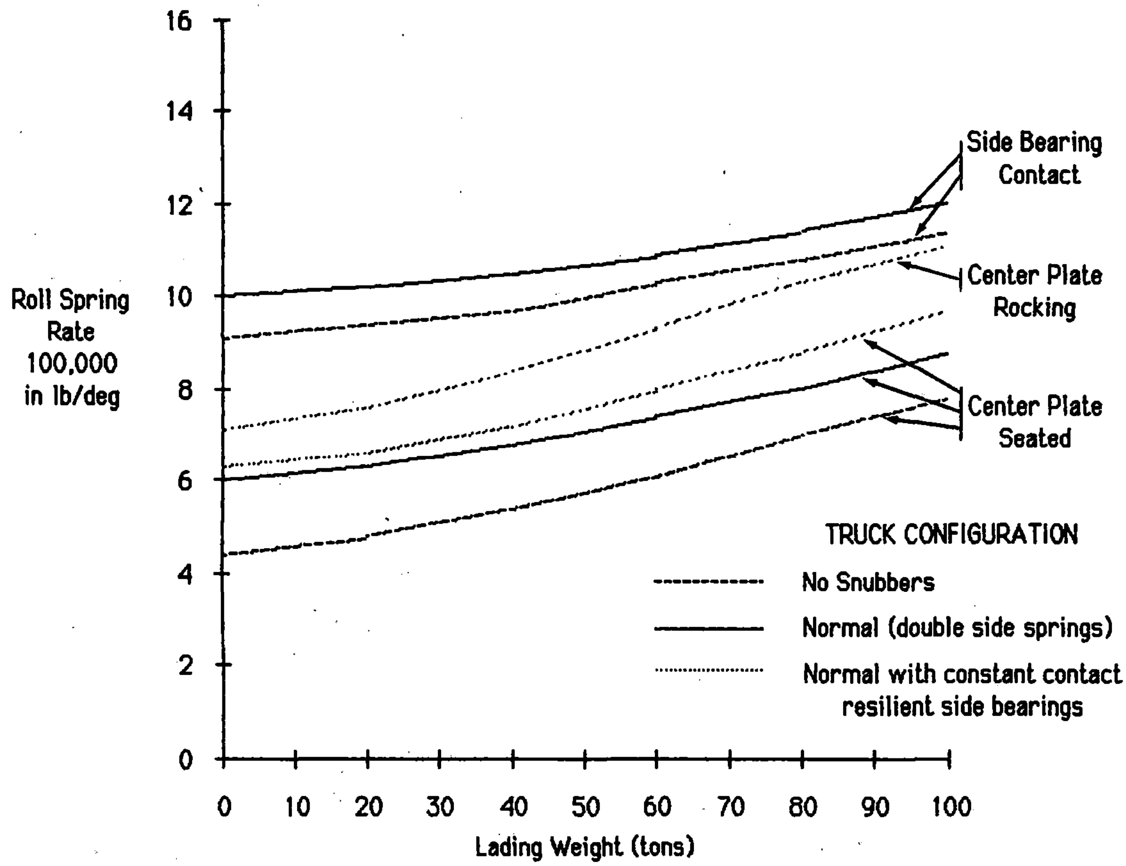


FIGURE 3-18
OVERALL TRUCK ROLL SPRING RATES, PER TRUCK
BARBER S-2-C

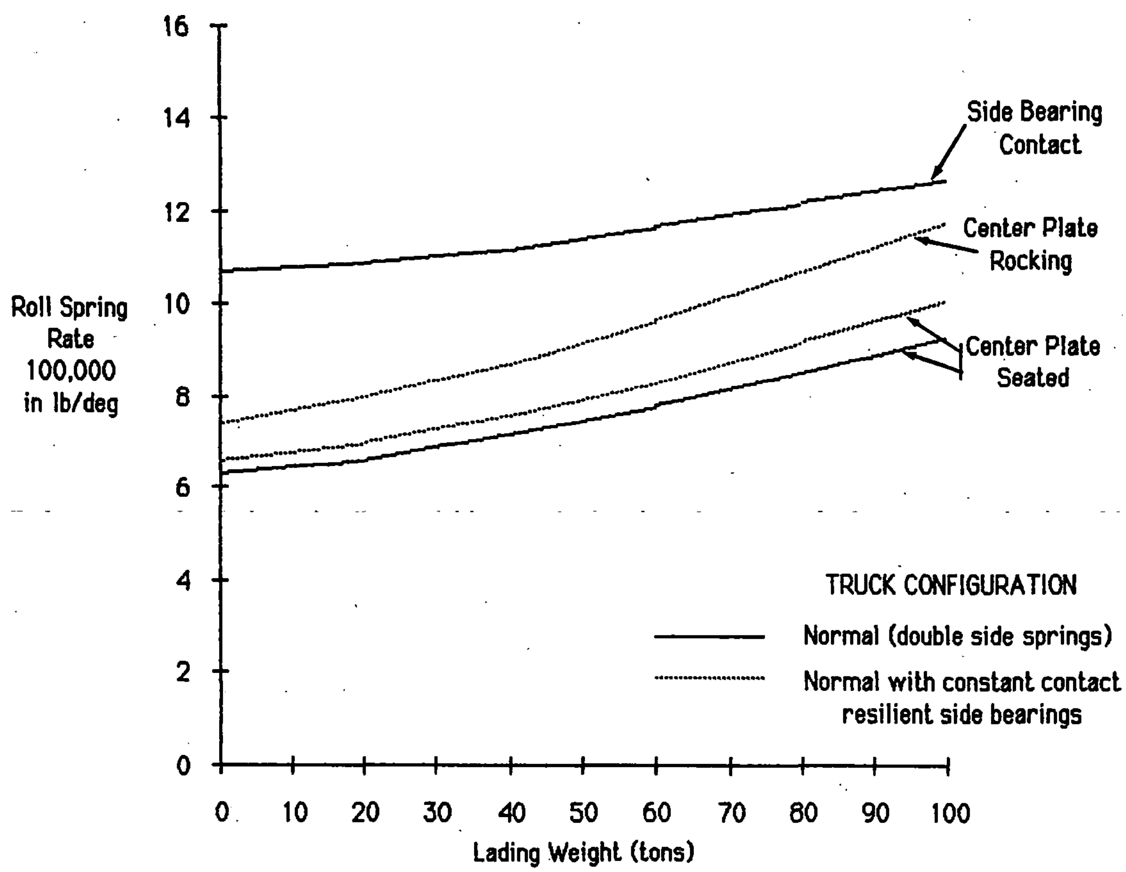


FIGURE 3-19
SUSPENSION SYSTEM ROLL SPRING RATES, PER TRUCK
BARBER S-2-C TRUCK

TABLE 3-7
ROLL SPRING RATES, BARBER S-2-C TRUCK
Carbody bolster to Side Frames, Per Truck

TRUCK CONDITION	LOAD CONDITION EMPTY K IN LB./DEG.	LOAD CONDITION 90% K IN LB./DEG.
Roller Side Bearings		
Center plate Seated	623	890
Center plate Rocking	0	0
Side Bearing Contact	1067	1250
Center plate Lifted	0	0
Resilient Side Bearings		
Center plate Seated	645	960
Center plate Rocking	732	1120
Center plate Lifted	0	0

4. RIDE CONTROL TRUCK CHARACTERIZATION TESTS

Quasi static tests were performed on both Ride Control trucks from the Base Vehicle No. 2. The testing was similar to the tests performed on the Barber S-2-C trucks as described in Section 3. The same test set up was used and the same test procedures were followed. Measurements differed in that D9 and D10 were changed and D11 and D12 were dropped. The measurements list is given in Table 4-1. The only configurational variation was to test with snubbers both dry and wet. The tests performed and loading conditions applied are listed in Table 4-2. A schematic of the truck is shown in Figure 4-1 showing location of measurements and the spring notation of primary, secondary, and center plate. Configuration and characteristics data are given in Figure 4-2.

4.1 Test Results, Ride Control Truck

As in Section 3, the truck test results are presented according to test load conditions: namely, vertical, lateral, and roll. The analysis process was to first output test results on load/deflection plots. Spring rates were determined from the slopes of the load deflection paths and damping forces were determined from the heights of the hysteresis loops.

The truck stiffness data are again divided into and presented in three parts: the suspension system, consisting of the spring nest and some local structure effects; the center plate, which also includes flexibility of the truck bolster; and the side frames, which also includes wheel sets and joints. The combined stiffness of the suspension system and center plate/truck bolster is also given since this is a common treatment in math models.

4.1.1 Vertical Test Results

Results from the vertical tests performed on the Ride Control truck are presented in Table 4-3 as spring rates from the empty and 90% load conditions. These data are also plotted in Figure 4-3 for the secondary suspension, the secondary combined with the center plate, and the overall. Note that what is termed as center plate includes the flexibility of the truck bolster. Figure 4-3 shows that: (1) the secondary suspension is soft in comparison to the center plate and primary suspension (by a factor of between 20 and 30); and (2) that the suspension stiffness increase with load is nearly linear. The differences in stiffness between the A and B truck were small and the wet or dry snubbers did not change the truck spring rates.

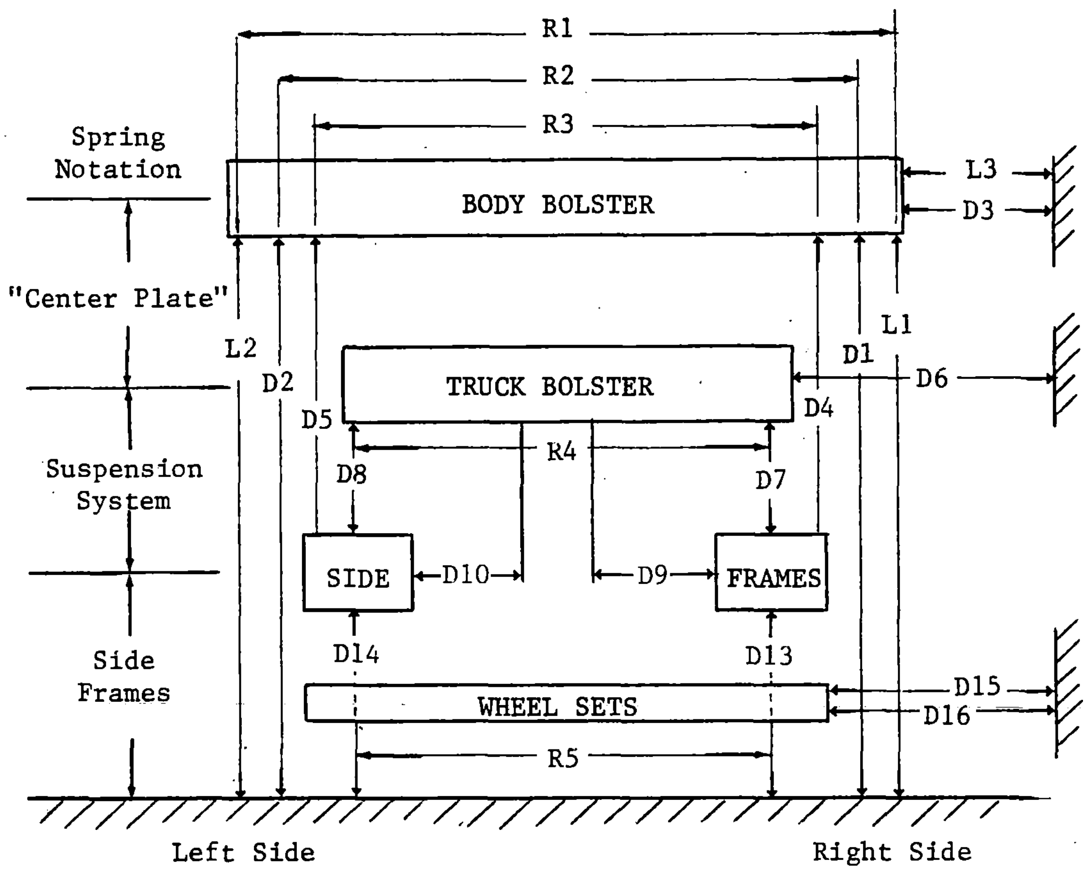
TABLE 4-1
MEASUREMENT LIST, RIDE CONTROL TRUCK CHARACTERIZATION TEST

Measurement Number	Locations
L1	Right Vertical Actuator Force, South Pegasus
L2	Left Vertical Actuator Force, North Pegasus
L3	Lateral Actuator Force
D1	Body Bolster to Ground Vertical Right Side
D2	Body Bolster to Ground Vertical, Left Side
D3	Body Bolster to Ground Lateral
D4	Body Bolster to Side Frame, Vertical, Right
D5	Body Bolster to Side Frame, Vertical, Left
D6	Truck Bolster to Ground, Lateral
D7	Truck Bolster to Side Frame, Vertical Right
D8	Truck Bolster to Side Frame, Vertical Left
D9	Truck Bolster to Right Side Frame, Lateral
D10	Truck Bolster to Left Side Frame, Lateral
D13	Side Frame to Ground, Vertical, Right
D14	Side Frame to Ground, Vertical, Left
D15	Front Wheel Set to Ground, Lateral
D16	Aft Wheel Set to Ground, Lateral

TABLE 4-2
TEST MATRIX FOR RIDE CONTROL TRUCK CHARACTERIZATION

TEST AND LOAD RANGES	DRY A & B	WET A & B
Vertical Test 20-100 kip	x	x
Lateral Test 100 kip Vert. ± 5.5 kip Lat.	x	x
60 kip Vert. ± 5.5 kip Lat.	x	
24 kip Vert. ± 5.5 kip Lat.	x	x
Roll Test 40 ± 15 kip each actuator	x	x
30 ± 25 kip each actuator	x	
25 ± 20 kip each actuator	x	x

Notes: A and B refer to truck A and truck B.
Dry and wet refer to condition of snubbers.



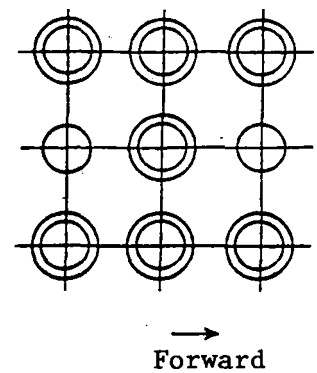
R1 = 127.5 = Distance between vertical actuator center lines (L1, L2).
R2 = 110.75 = Distance between measurements D1 and D2.
R3 = 78.75 = Distance between D4 and D5.
R4 = 99.75 = Distance between D7 and D8.
R5 = 84.00 = Distance between D13 and D14.

All distances in inches.

FIGURE 4-1
TRUCK SCHEMATIC SHOWING MEASUREMENT LOCATIONS AND SPRING NOTATION, RIDE CONTROL TRUCKS

A. General Configurational Data

- Spring nest - 7 D5 outers
  9 D5 inners
- Snubbers - constant column friction
- Center plates - 16 inch tapered
- Side bearings - double roller



B. Rated Characteristics, New Conditions, per nest

• Solid Capacity	95,683 lb.
• Loaded Spring Rate	25,781 lb./in.
• Column Load	4,740 lb.
• Height, Light Car	9.82 in.
• height, Loaded Car	7.88 in.

C. Measured Spring Nest Heights, Inches

NEST LOCATION	CAR REMOVED	LOADED CAR
B end, right side	9.75	7.875
B end, left side	10.125	7.75
A end, right side	10.00	7.375
A end, left side	10.00	7.75

FIGURE 4-2
RIDE CONTROL TRUCK CONFIGURATION AND CHARACTERISTICS DATA

TABLE 4-3
VERTICAL SPRING RATES, RIDE CONTROL TRUCK

TRUCK COMPONENT	LOAD CONDITION	
	EMPTY	90%
Center Plate, et. al.	605	605
Suspension System	21.5	29.2
Suspension and Center Plate	20.8	27.9
Side Frames	650	650
Overall (body bolster to ground)	20.1	26.5

Units are K lb./in., per side

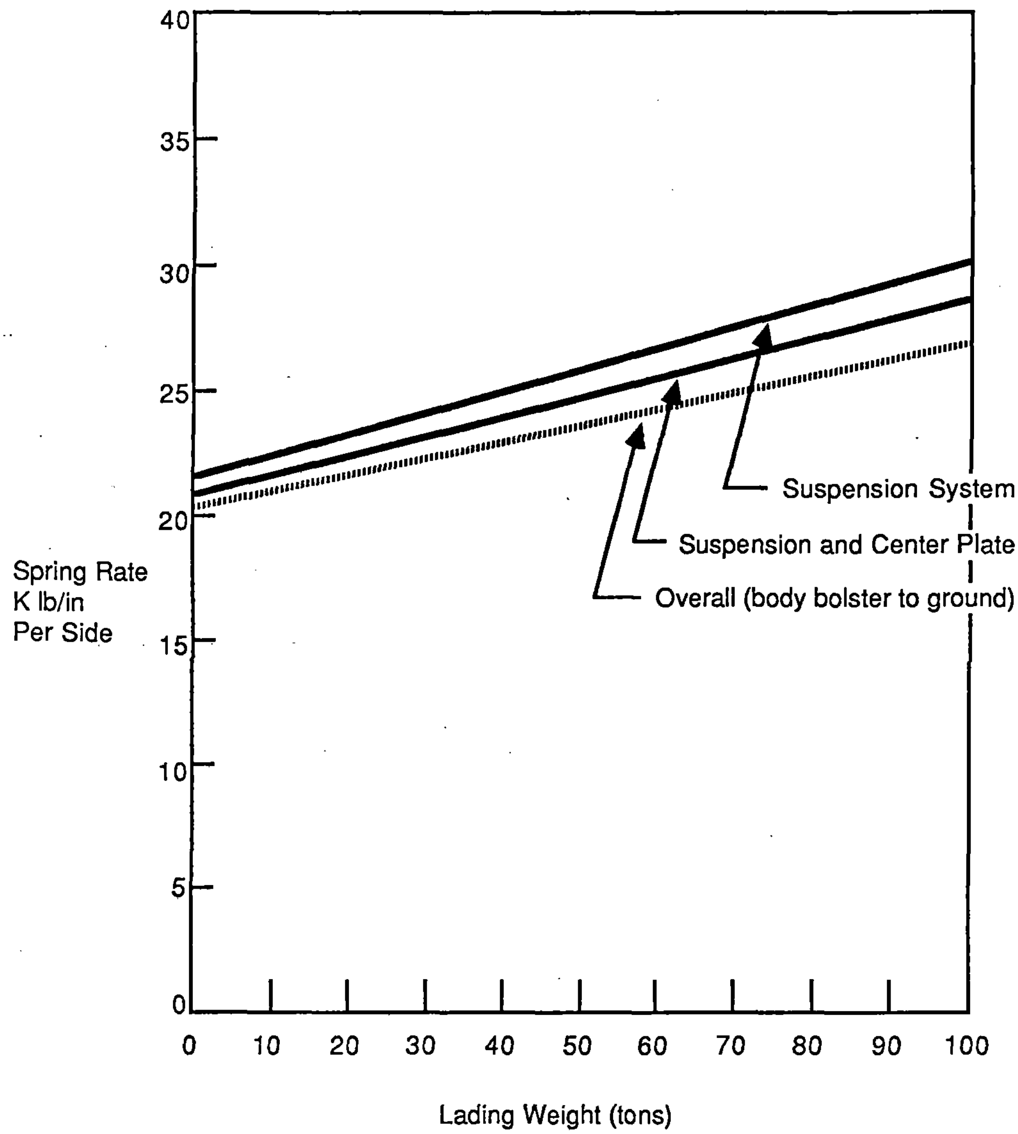


FIGURE 4-3
RIDE CONTROL TRUCK VERTICAL SPRING RATES, PER SIDE

The friction forces developed by the snubbers are shown in Figure 4-4 plotted against weight of lading. Moisture on the snubbers caused a 30 to 40% loss in friction force. The friction force was also seen to be reduced with increase in lading weight. This latter condition was not expected. However, an hypothesized explanation would be that there was uneven wear on the column friction plate, see Figure 4-5.

That is, most of the wear life of the trucks was probably under fully loaded conditions resulting in greater wear on the lower portion of the friction plate which would result in reduced friction force at the lower portion of the friction plate. The amount of reduction in the friction force will vary depending on whether or not the shoe moves up in the pocket as a result of the dimensional changes. With the long strokes (the equivalent of empty to 80 tons) used in the test procedure, it is more likely that the shoe stays in one portion of the pocket. In normal operating conditions, motions will be smaller and the shoe will probably move down into the lower positions. There are, accordingly, two sets of curves shown in Figure 4-4: one for the long stroke conditions in test and the other for the short stroke assumed for normal operational conditions.

4.1.2 Lateral Test Results, Ride Control Truck

As in the Barber Truck results, the lateral stiffness is presented in three sections and named center plate, suspension system, and side frames, where each includes flexibility from adjacent structures. A summary tabulation of these three "springs" is given in Table 4-4. There is also a fourth spring value given that is the combined stiffness of the center plate and suspension. These four stiffness are also presented in Figures 4-6 and 4-7 as functions of lading weight.

4.1.3 Roll Test Results, Ride Control Truck

The results of the roll tests on the Ride Control trucks are presented in Table 4-5 and Figure 4-8. The roll stiffness was again, as in the Barber Truck data, divided in the three regional conditions of (a) center plate seated, (b) center plate rocking, and (c) side bearing contact. However, the data scatter covered differences between (a) and (c) so that these were lumped into one relatively wide band rather than two curved lines.

The Ride Control truck differed from the Barber truck in that it did not have zero spring rate for condition (b), center plate rocking. The data are presented as analyzed although the

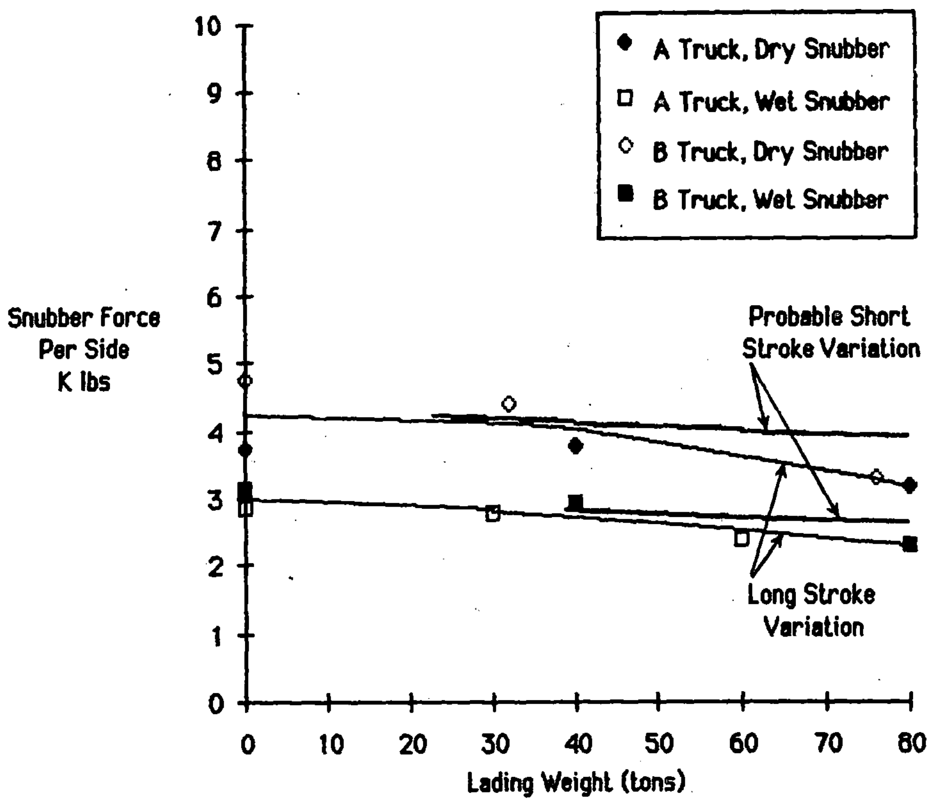


FIGURE 4-4
RIDE CONTROL TRUCK SNUBBER FORCE

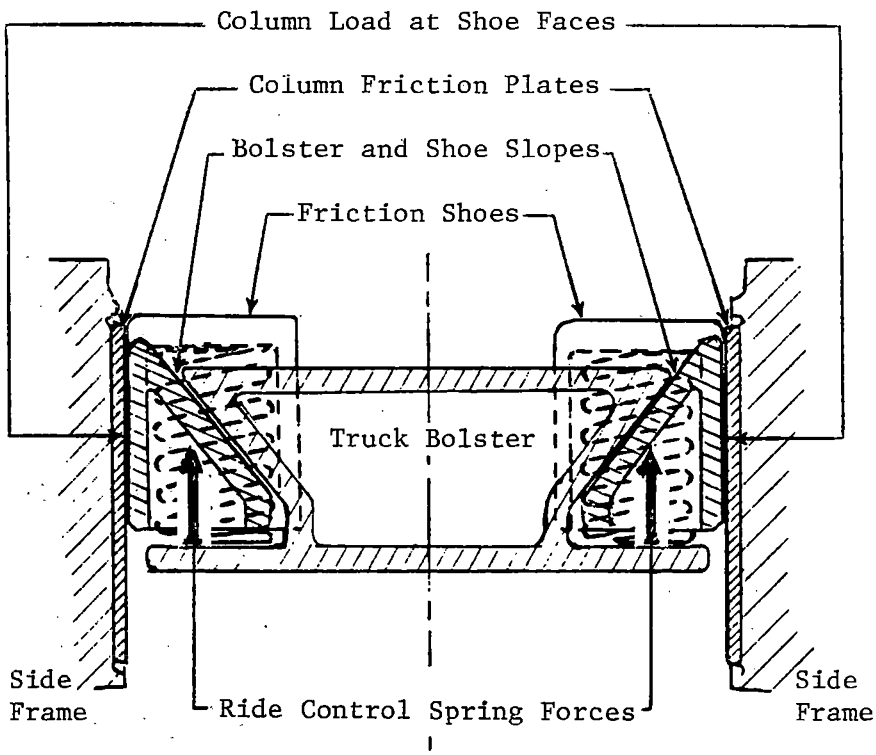


FIGURE 4-5
RIDE CONTROL TRUCK FRICTION SNUBBER ASSEMBLY, CROSS SECTION

TABLE 4-4
LATERAL SPRING RATES, RIDE CONTROL TRUCK

TRUCK COMPONENT	LOAD CONDITION EMPTY	LOAD CONDITION 90%
Center Plate (body bolster to truck bolster)	40	43.5
Suspension System (truck bolster to side frame)	45	83
Suspension and Center Plate	21.2	28.5
Primary (side frames to ground)	100	1025

Units are K lb./in., per truck

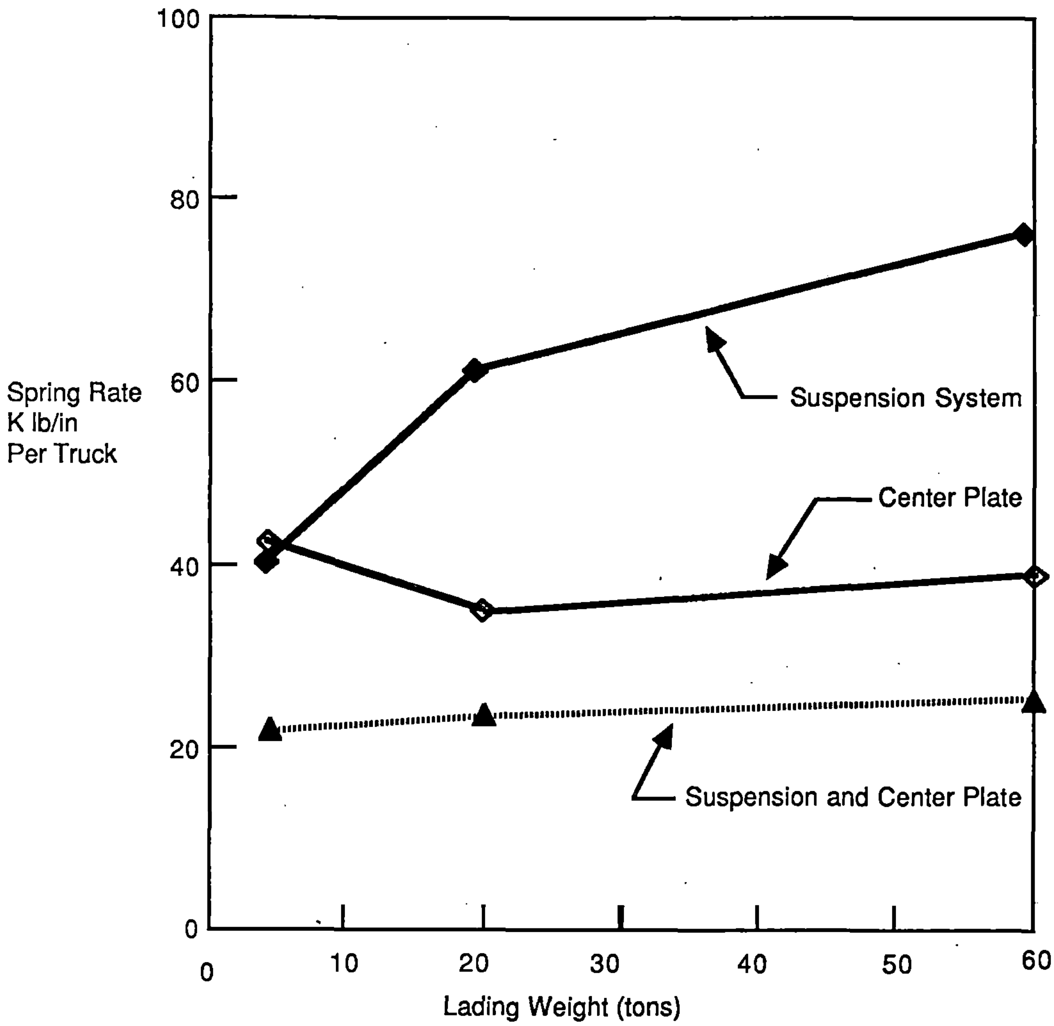


FIGURE 4-6
RIDE CONTROL TRUCK SUSPENSION SYSTEM
LATERAL SPRING RATES, PER TRUCK

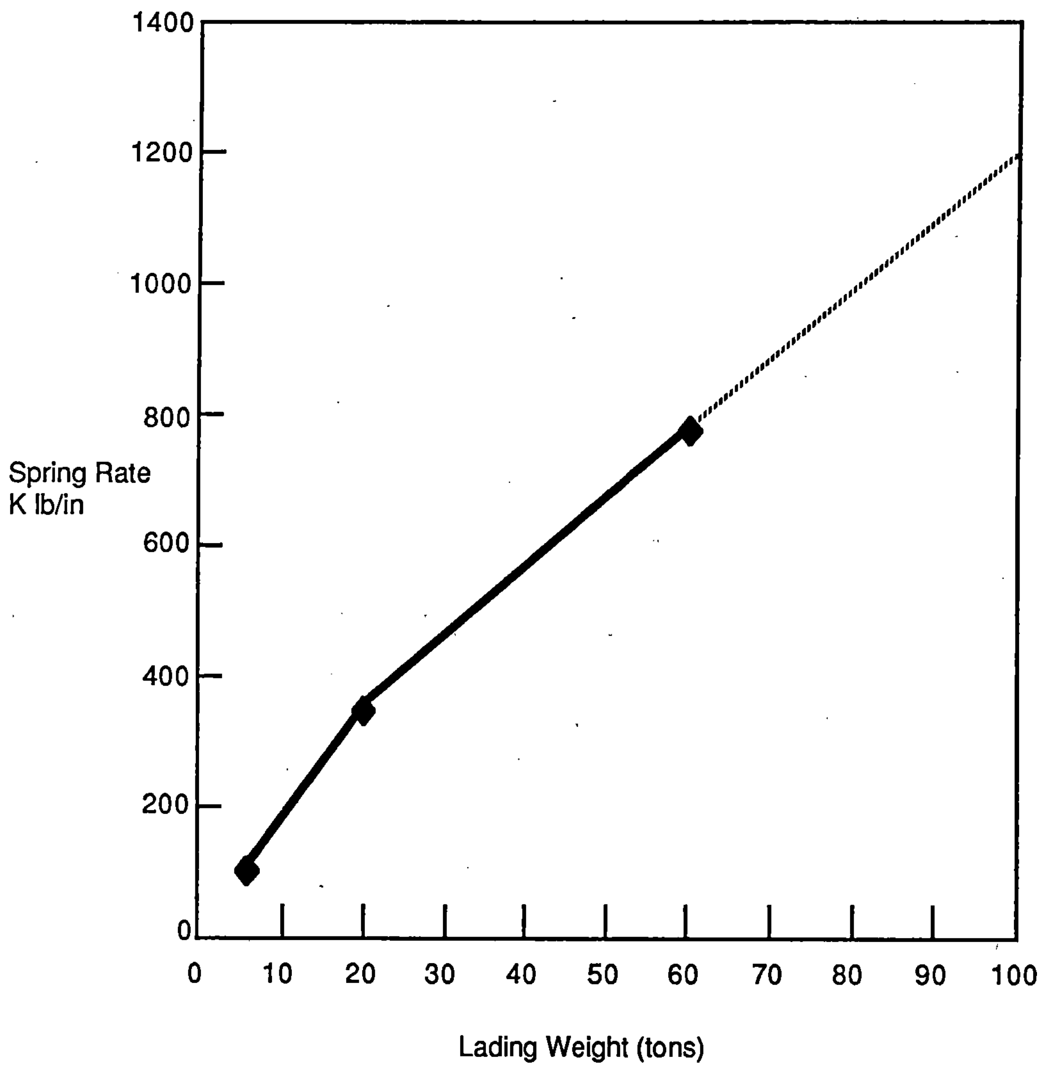


FIGURE 4-7
RIDE CONTROL TRUCK LATERAL SPRING RATE,
SIDE FRAMES, PER TRUCK

TABLE 4-5
SUSPENSION SYSTEM ROLL SPRING RATES,
RIDE CONTROL TRUCK, PER TRUCK

TRUCK CONDITION	LOAD CONDITION	
	EMPTY	90%
Center plate seated or side bearing contact	8.5×10^5	11.2×10^5
Center plate rocking	3.6×10^5	4.3×10^5

Notes:
- Units are in lb./deg., per truck.
- Values are from average of data scatter band in Figure 4-8.
- Data are for overall spring rates (body bolster to ground) however, data scatter is large enough that values are also applicable to suspension system.

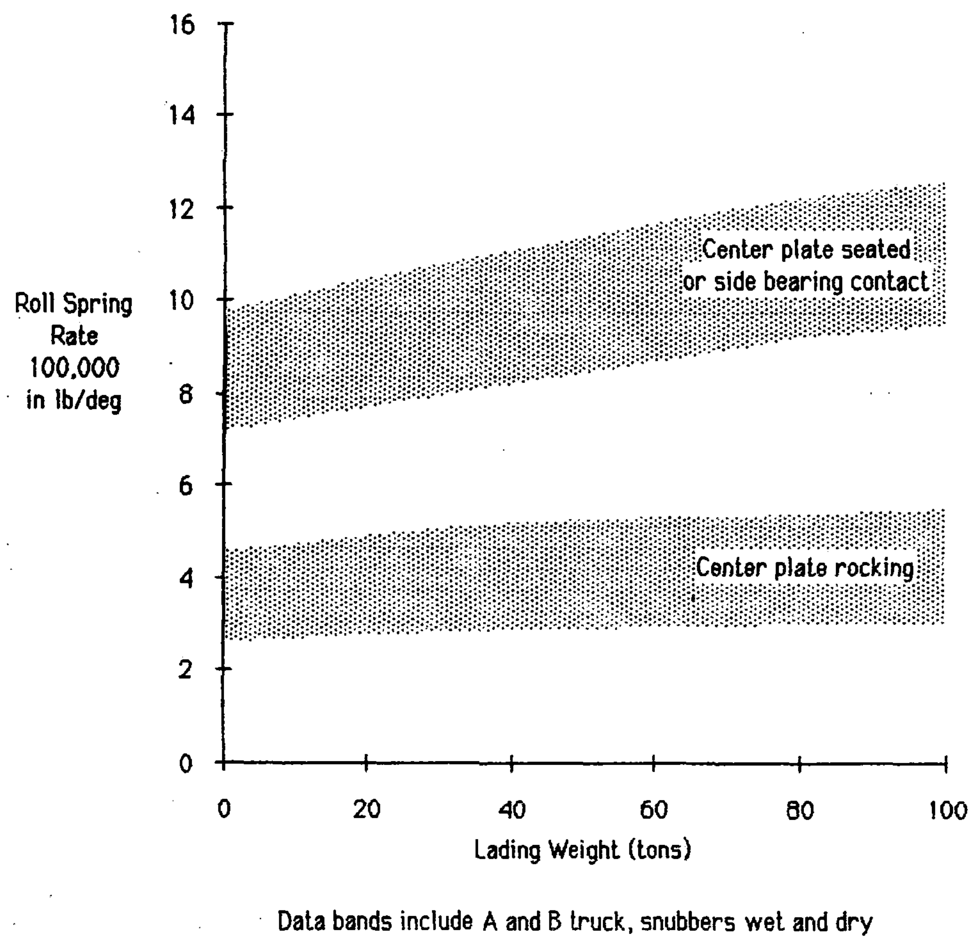


FIGURE 4-8
SUSPENSION SYSTEM ROLL SPRING RATES, RIDE CONTROL TRUCK, PER TRUCK

question of why the rocking spring rate is not zero could not be answered. The amount of data scatter also detracts from confidence in the accuracy of the data: that is, is the data scatter due to the erratic nature of the truck or is it due to experimental inaccuracies?

5. VIBRATION TESTS OF BASE VEHICLE NO. 1

The primary purpose for the tests performed on Base Vehicle No. 1 was for evaluation of the Cross Level Index, a root-mean-square value of cross level which, as proposed by the Transportation Systems Center [5], represents a safe limit of track geometry for harmonic roll of freight cars. Of the 959 runs made on this vehicle, 940 were for CLI evaluation. The TSC has, in a separate effort, performed analyses on the CLI tests and has reported their findings [6]. Consequently, the purpose of this report is to present only the dynamic response properties test results.

5.1 Test Objectives

The test objectives, other than CLI evaluation, were to obtain modal data on the first roll mode of the vehicle and on the torsion mode of the vehicle body and to obtain harmonic roll response characteristics with staggered rail inputs. These results would then be used to augment the results of more extensive dynamic response testing performed on Base Vehicle No. 2.

5.2 Description of Base Vehicle No. 1

The Base Vehicle No. 1 configuration is shown in Figure 5-1 with notations on vehicle dimensions and orientation. Other configurational data are given in Tables 5-1 and 5-2 and Figure 5-2. Center plate dimensions are given in Figure 5-3. Absence of the center bowl liner in the B truck is noteworthy since it results in a clearance of only 1/8 inch between the truck and body bolsters at the top edge of the bowl. Note also, that the deflection under load of the right side B truck spring nest is greater than the other three spring nests by about 3/4 inch.

5.3 Test Procedures

Testing on Vehicle No. 1 was in two phases, one for TSC requirements and the other for FRA requirements; however, the TSC tests were primary and comprised the major portion of testing.

All tests performed had the commonality of imposing motion at the wheel-rail interface and recording loads and responses of the vehicle. On this common base there were three variations: frequency, wave form, and wave amplitude. These are discussed below.

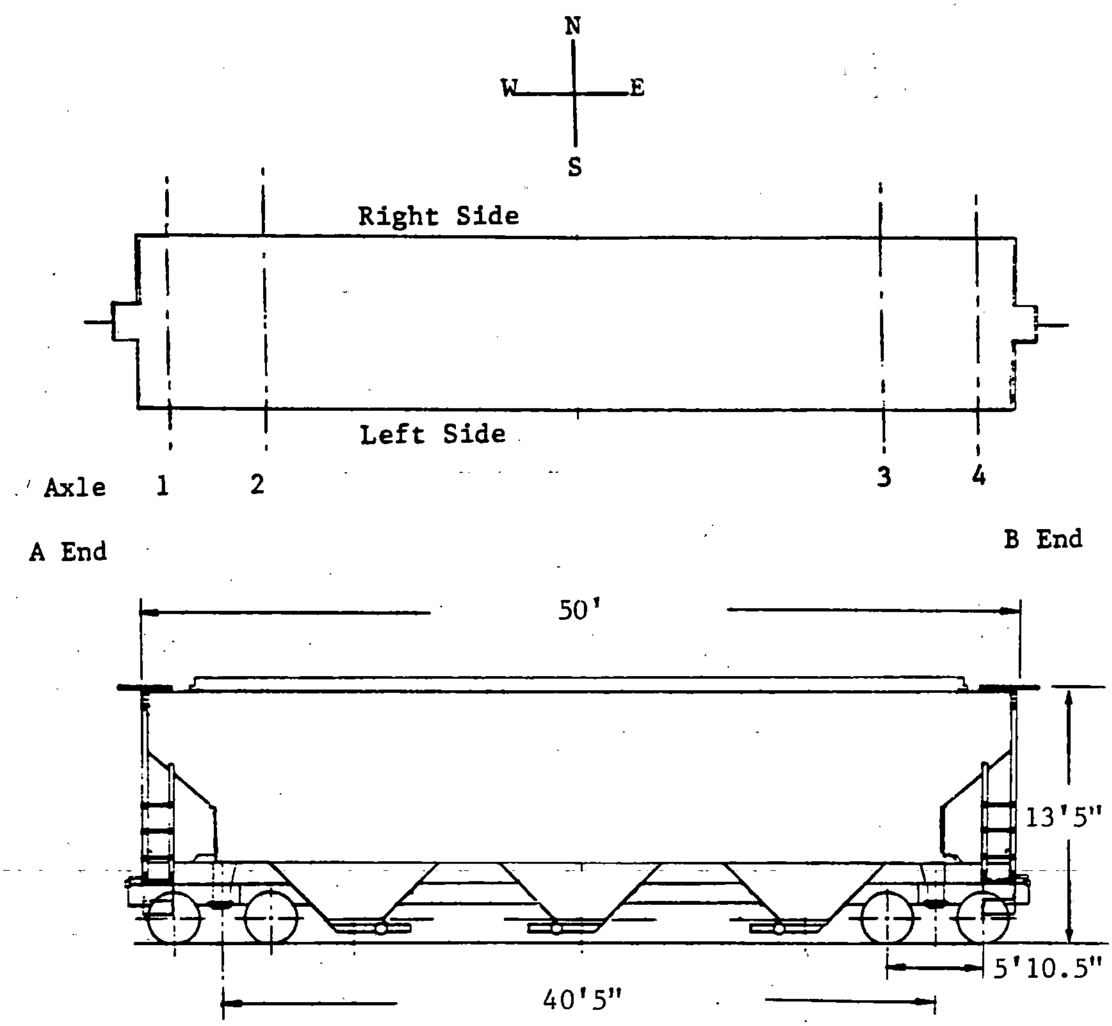


FIGURE 5-1
ORIENTATION OF BASE VEHICLE NO. 1
ON THE VIBRATION TEST UNIT

TABLE 5-1
BASE VEHICLE NO. 1 CONFIGURATION DATA[1]

Number	- MP 712074
DES date	- 11-82
Capacity	- 202,000 lbs
Load Limit	- 202,800 lbs
Light Weight	- 60,200 lbs
Volume	- 4,427 cu. ft.
Wheel Diameter	- 36"
Truck	- Barber S-2-C
Snubbers	- Load Variable Friction
Springs	- 28 Outer D-5
	- 24 Inner D-5[2]
	- 8 Outer 432
	- 8 Inner 433
Axle Spacing	5 ft., 10 1/2 in.
Truck Spacing	40 ft., 5 in.
Test Gross Weight	A end 131,140 lbs (at the rails)
	B end 131,640 lbs
	Total 262,780 lbs

Side Bearings
A end: side bearing load cells - 1/4" clearance each
B end: roller side bearings - contact left side
1/2" clearance right side

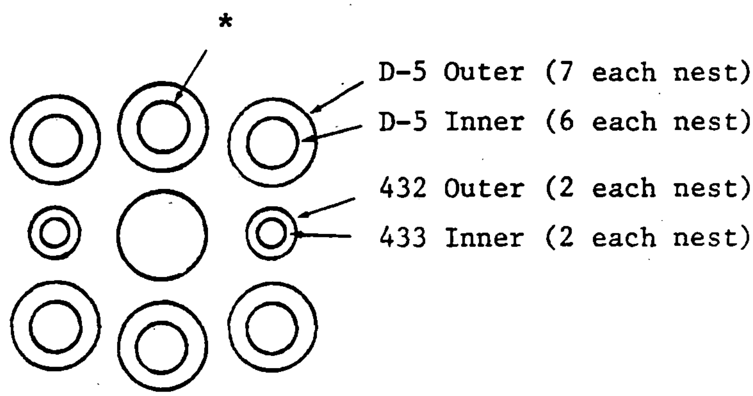
Center of gravity Height
Empty Vehicle: 63.0 in. (from top of rail)
Loaded Vehicle: 97.0 in.
(the car was very close to maximum volume)

[1] Base Vehicle No. 1 was delivered with constant contact resilient side bearings which were replaced with roller bearings at the B end load cells at the A end.

[2] One of the Inner D-5 springs in the right side, B track nest was actually, and obviously by mistake, an Inner D-6 (See Figure 5-2).

TABLE 5-2
SPRING NEST HEIGHT AND SIDE BEARING CLEARANCE DATA

	A TRUCK		B TRUCK	
	LEFT SIDE	RIGHT SIDE	LEFT SIDE	RIGHT SIDE
Side Bearing Clearance Pre Test	1/4"	1/4"	1/2"	0"
Spring Nest Height Pre Test	7-7/8"	8"	7-7/8"	7-3/8"
Spring Nest Height Post Test	7-7/8"	8"	7-5/8"	7-1/4"
Free Height Post Test	10-11/16"	10-7/8"	10-7/8"	10-3/4"



*The nest on the right side of the B truck has a D-6 inner spring at this location in place of a D-5 inner.

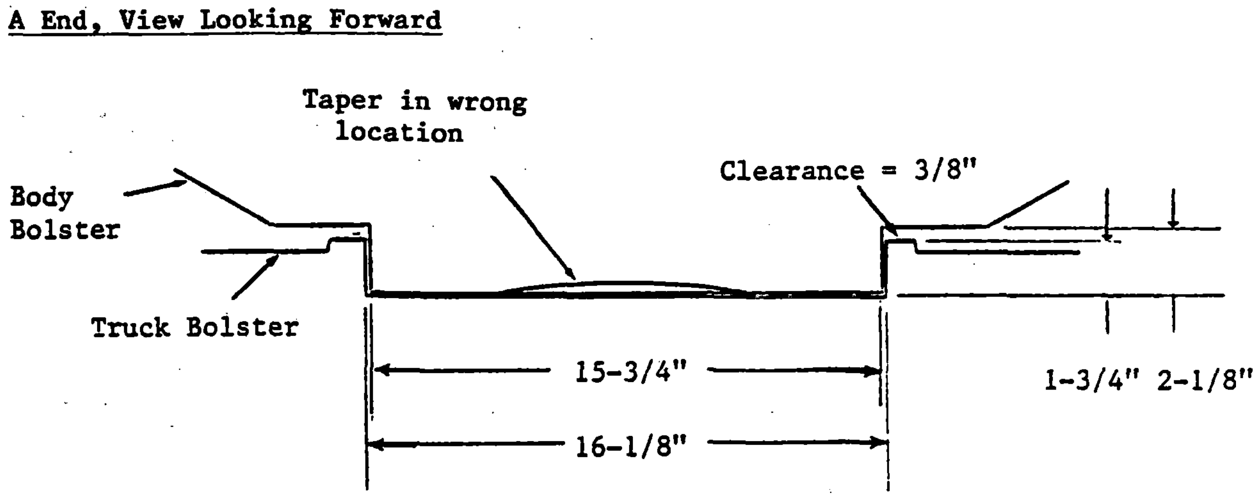
Total Spring Complement

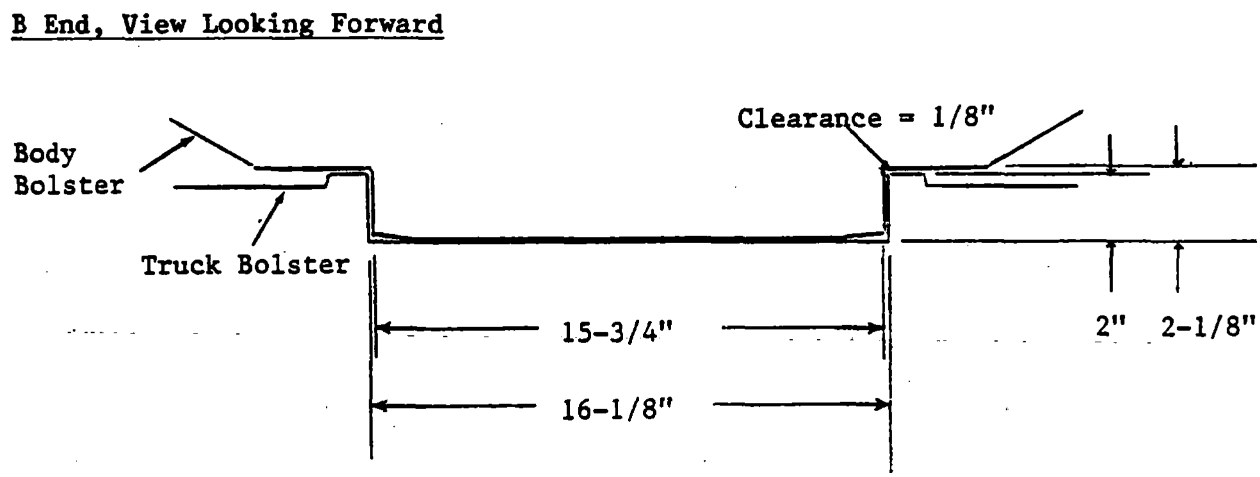
28	Outer	D-5
23	Inner	D-5
1	Inner	D-6
8	Outer	432
8	Inner	433

Reduced Snubber Configuration

- All 433 inner springs were removed. This results in approximately 33% reduction of friction force.

FIGURE 5-2
SPRING NEST CONFIGURATION, BARBER TRUCK





NOTE: There was a liner in the A bowl but not in the B bowl.

FIGURE 5-3
TRUCK AND CARBODY CENTER PLATE DIMENSION

Frequency Variation in Test Procedure

Frequency, wave length, and track speed are related by the equation:

$$f = V/\lambda$$

where:

f = frequency, Hertz
V = track speed, feet per second
λ = wave length, feet (rail length)

Assuming a 39-foot rail length and using miles per hour (mph) for track speed, the equation becomes:

$$f = MPH/26.591$$

The general test procedure was to vary the frequency and amplitude of input to find the frequency (speed) of maximum vehicle response and the amplitude of input resulting in a critical vehicle response.

Frequency variation was accomplished in two ways: sweeps, where a predetermined range of frequencies is covered in a single run at a constant amplitude of input, and dwells, where the frequency and amplitude are constant for each run.

Wave Form of Input Motions

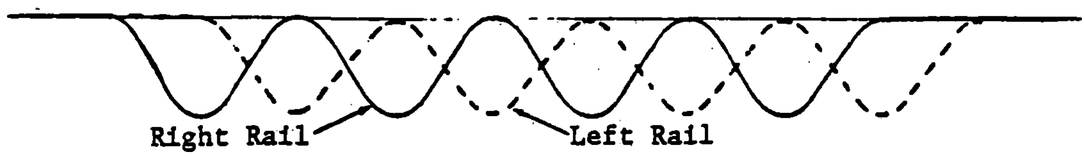
There were three basic wave forms used in the testing:

- exponential,
- sinusoidal, and
- rectified sine.

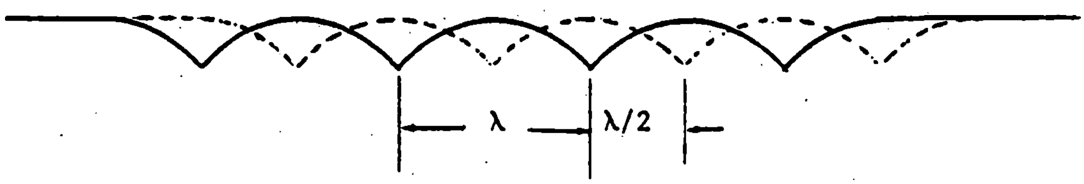
These are pictured in Figure 5-4.

The exponential wave forms were always applied with a preset number of wave lengths and with the amplitude of each joint controlled by a Shape designation. In the Shape designation each low joint was assigned a relative amplitude that varied between 0 and 1. Shapes A through F are sketched in Figure 5-5. Shapes G, H, and I are sinusoidal with wave lengths of 39, 50, and 75 feet.

a. Sinusoidal



b. Rectified Sine



c. Exponential

Exponential shape is of the form $e^{-x/\alpha}$

where x = distance on track, feet
α = decay length, feet (Use $\alpha = 8$)

NOTE: Profile wave forms shown are sketched and not to scale.

FIGURE 5-4
TEST WAVE FORMS FOR STAGGERED RAIL

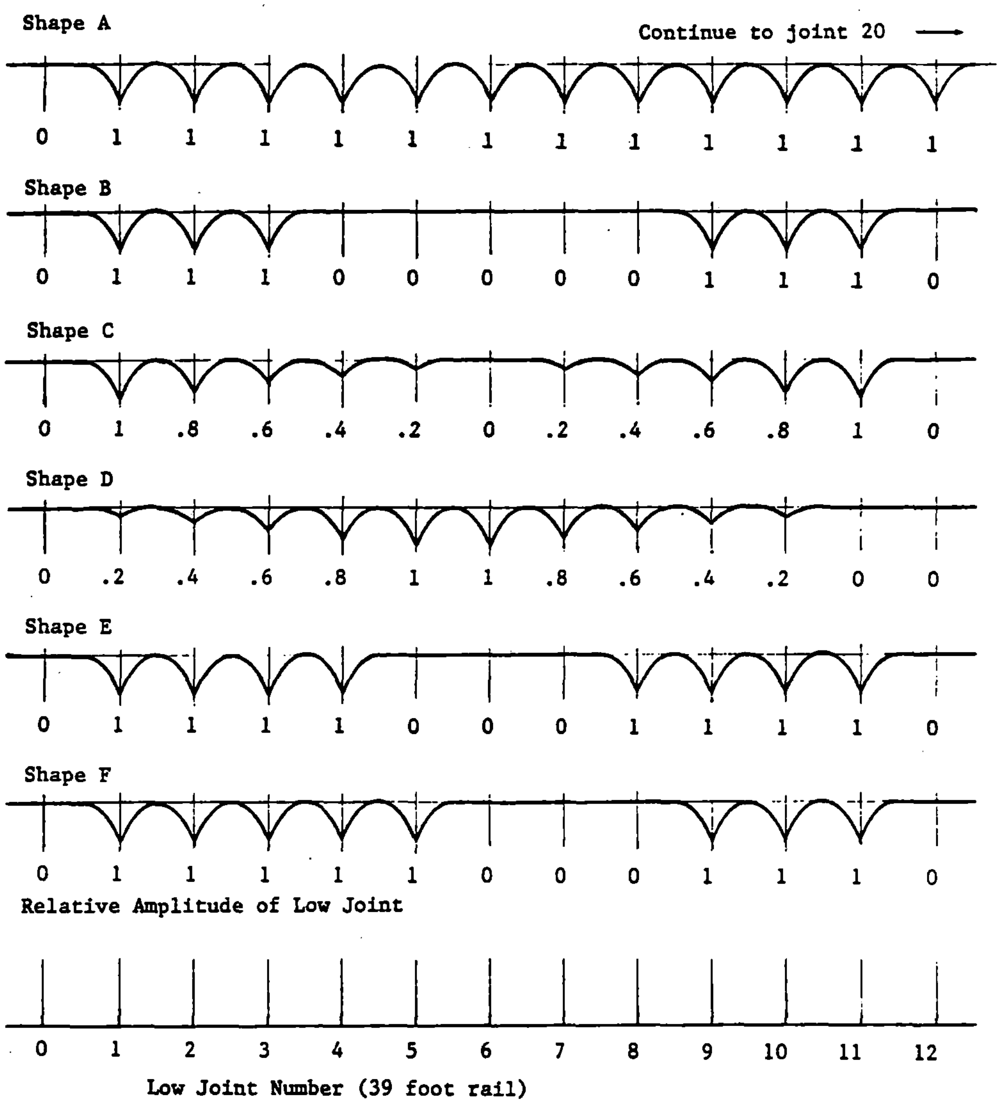


FIGURE 5-5
SHAPES OF EXPONENTIAL LOW JOINT PROFILES

Shapes A through I were used in the simulation of staggered rail in that the right and left rails were shifted relative to each other such that the low joint of each fell midway between the low joints of the other. This is shown in Figure 5-4. The simulation also time phased the profile between wheel sets corresponding to axle and truck spacing and the track speed being simulated.

Shape J was a repeat of the Shape D staggered rail profile and also included a lateral motion of each wheel set that was a function of the difference in profile of the right and left rail. A vertical (profile) and lateral input trace from a Shape J test is shown in Figure 5-6. The vertical and lateral motions were set such that the lateral motion was maximum to the right when the right rail was at a low joint, and maximum to the left when the left rail was a low joint.

5.3.1 Test Procedures for Validating Cross Level Index

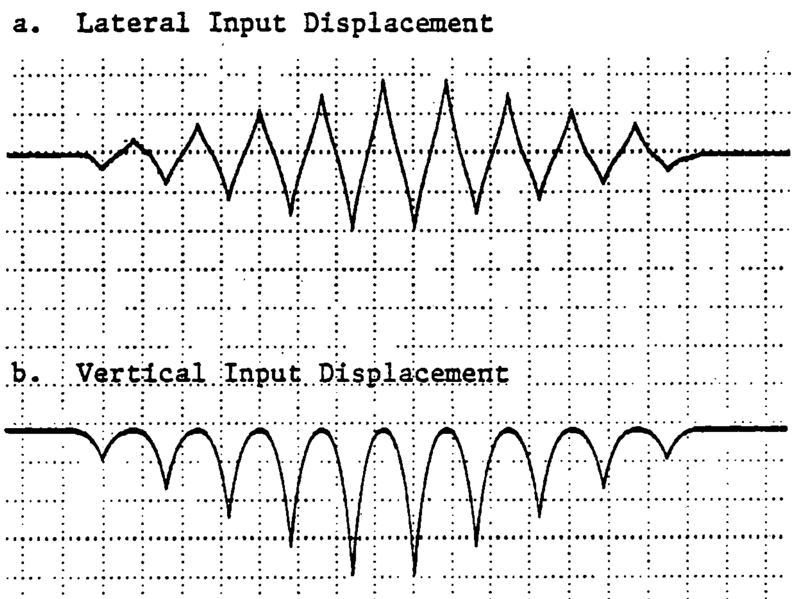
The objective of the CLI testing was to determine the critical conditions of speed and cross level amplitude for each of the simulated track conditions or Shapes. Carbody roll angle, wheel-rail loads and wheel lift were the primary measures monitored. However, the critical limiting condition was to be wheel lift of 0.5 inches.

Each test was run at a constant speed with a preset amplitude. The test procedure involved making sequential runs, changing speed and/or input amplitude between runs until the speed and amplitude of the critical condition were identified. In the process sufficient data were obtained to define plots of carbody roll and wheel load as a function of speed for several input amplitudes.

In the process of testing it was found that limit wheel load conditions (70,000 lb) were exceeded before 0.5-inch wheel lift was reached. Also, the rail sections interfacing the wheels and shaker actuator were found to be experiencing fatigue failures. Because of these factors, the critical condition was changed to incipient wheel lift.

5.3.2 Test Procedures for Dynamic Response Evaluation

The dynamic response test objectives were similar to the CLI testing in that critical conditions of speed and input amplitude were to be obtained. The procedures differed in that



Lateral motion maximum to the right when right rail is at low joint.

FIGURE 5-6
SHAPE J LATERAL AND VERTICAL INPUT DISPLACEMENTS

speed (frequency) sweeps were used in place of constant speed runs (dwells). The other procedure difference was that the wave forms used were sine and rectified sine.

5.4 Discussion of Vehicle Problems, Base Vehicle No. 1

There were two peculiarities that were noted in the vehicle at the very start of the VTU tests. One was that the spring nest on the right side of the B truck had the appearance of being weak. That is, that spring nest would always have larger deflections, under both static and dynamic conditions, than the other three spring nests. Along with this, under static conditions the truck bolster would be tilted, with the right end deflected down relative to the left end to the point that the left side bearing was in contact.

A second oddity was that the snubbers in the A truck would stick and bind in a very severe manner. That is, the snubbers would not move until large dynamic amplitudes were imposed on the vehicle and then they would release with a jerk and a bang (an explosive sound that was a little heavier than the crack of a 22-caliber rifle).

The weak-appearing spring nest was disassembled and examined without finding any visible faults. There was a D-6 inner where there should have been a D-5 inner, but this results in an increased spring rate of the nest of about 1.5% which is insignificant. The unloaded spring heights were at specification. No explanation for the B truck behavior was found.

In a closer examination of the A snubbers it was found that cotter pins used in assembly were still in the snubbers. The cotter pins were removed but snubber performance remained the same. Some light cutting oil was put on the A snubbers resulting in reduced friction force and smooth movement of the snubbers. The vehicle was exercised at 0.8 Hertz (near its roll resonance) with a 0.75-inch cross level input at the rail for about 14 minutes (600-700 cycles) in an attempt to accelerate the break-in of the A snubbers. The cutting oil was then washed off with Freon and the vehicle exercised in roll again.

These operations appeared at first to result in a satisfactory break-in of the A snubbers with snubber action being relatively smooth and with the snubber force in a nominal range. However, near the end of testing (about two weeks of testing and two weeks for Christmas and New Year shut-down), the A snubbers again changed back to their original behavior of binding, banging, and larger friction force. It was concluded that there had been some residue of the cutting oil and Freon acting as a lubricant and after it finally wore off and/or dried up, the snubbers returned to their original state.

There were two other non-standard conditions found after testing was completed and the carbody was lifted off the truck: (1) the B truck did not have a center plate liner (it had been noticed during the testing that the B center plate was low in the center bowl compared to the A end), and (2) the A center plate was rotated 90° from its proper position--that is, the tapered edges were on the front and back of the center plate instead of on the sides.

Because of the unusual behavior of the A snubbers and the B truck, right side spring nest, it was decided that a static load test be effected by lifting the carbody and measuring spring deflection and wheel load. The resulting load deflection plots would give a measure (with limited accuracy) of spring rates and snubber forces. This was accomplished by lifting the vehicle at the couplers, one end at a time. Sample values were taken, after the break-in exercises described above, through to the end of testing and averaged to obtain the values shown in Table 5-3.

Although the cotter pins may have prevented the snubbers from having normal and proper break-in, the snubber performance is a better fit to the behavioral pattern of new snubbers with non-parallel wear plates (the new car syndrome [8]). These trucks were remanufactured and, upon further examination, it was concluded that the wear plates were not correctly located and that this was the root of the uncharacteristic performance of the snubbers.

5.5 Test Results

Modal Sine Test

Sinusoidal input motions were used in these tests with the actuator phased first to excite the first roll mode of the vehicle and then to excite the body torsion mode. Several sweeps were made from 5.0 to 20.0 Hertz in unsuccessful attempts to find the body torsion modes. Results of the first roll mode testing are summarized in Figures 5-7 and 5-8. The first roll mode was found to vary between 0.59 and 0.69 Hertz depending on the input amplitude. The maximum cross level input before wheel lift was 0.50 inches with a carbody roll response at 6.2 degrees, peak-to-peak. At 0.60-inch cross level input there was an abrupt increase in response, the carbody roll angle going

TABLE 5-3
AVERAGE SPRING RATE AND FRICTION FORCE VALUES
FROM BASE VEHICLE NO. 1 TEST RUNS 284-557

	VERTICAL SPRING RATE PER SIDE (LB/IN)	FRICTION FORCE PER SIDE (LBS)
A Truck	29,600	14,200
B Truck	21,200	6,400
Average	25,400	10,300
Ref. 12 100-ton covered hopper	25,783	2,000
Ref. 10 100-ton covered hopper	24,900	4,000
Ref. 13 70-ton boxcar	24,800	5,800

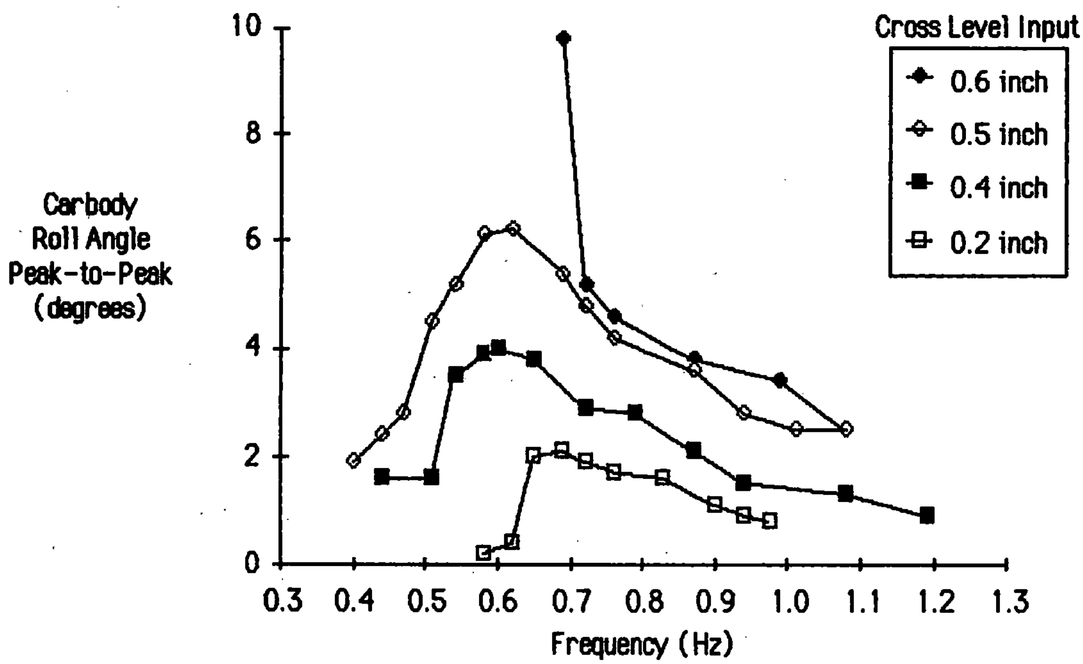


FIGURE 5-7
MODAL SINE TEST, CARBODY ROLL RESPONSE, BASE VEHICLE NO. 1

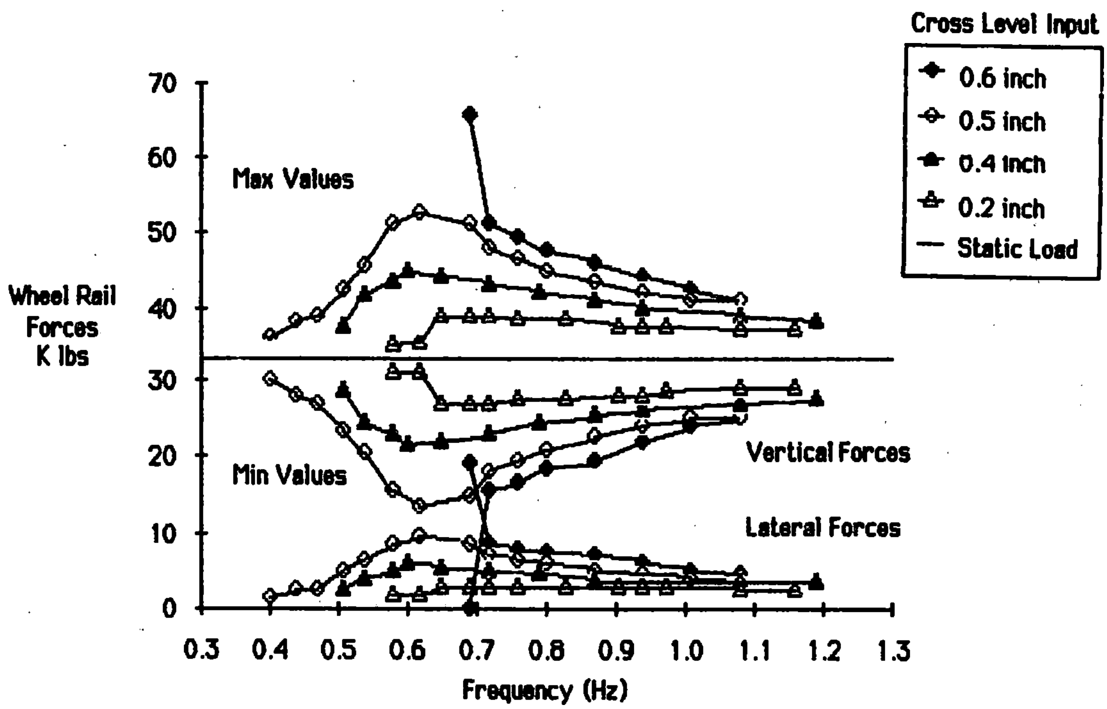


FIGURE 5-8
MODAL SINE TEST, WHEEL-RAIL LOADS, BASE VEHICLE NO. 1

from 6.2 degrees to something greater than 10.0 degrees, peak-to-peak. The test was stopped by the automatic shut-down, triggered by a wheel lift condition, before reaching the critical frequency.

The first roll mode frequency is seen to decrease and then increase as the cross level input is increased from 0.20 to 0.60 inches. This frequency range, 0.69 to 0.59 Hertz, corresponds to critical speeds of 18.3 to 15.7 miles per hour with 39-foot staggered rail.

Rectified Sine Simulation of Staggered Rail

Results of the tests using a rectified sine simulation of staggered rail are summarized in Figure 5-9, carbody roll angle, and in Figure 5-10, wheel-rail loads, in plots against input frequency (speed). The results are very similar to the sine test results with the wheel lift condition reached at 0.70-inch cross level for rectified sine compared to the 0.60-inch for sine. One significant difference was that the carbody roll angles were smaller in the rectified sine test while the wheel loads were about the same. This can be seen in the data comparison in Table 5-4.

Effect of Side Bearing Gap Change

Side bearing clearance was set at 0.25 inch at each of the four side bearings shortly before the start of the Dynamic Response testing. The actual configuration had a set of load cells on the A truck in place of roller side bearings. The B truck had the appearance of a weak spring nest on one side, resulting in no clearance on the left side and 0.50 inch on the right. However, near the end of the Dynamic Response tests it was noted that the A gaps had widened, and were in fact measured to be 0.32 inch right and 0.30 inch left and that the B gaps had narrowed and were measured to be 0.375 inch right and 0.0 inch left.

The gaps were reset to 0.25 inch right and left on the A truck and 0.50 inch on the right side of the B truck (the left side bearing was always in contact in static conditions). The 0.60 inch rectified sine test was repeated and the results are given in Figures 5-11 and 5-12. No significant differences can be seen in results comparing before and after the side bearing gap change.

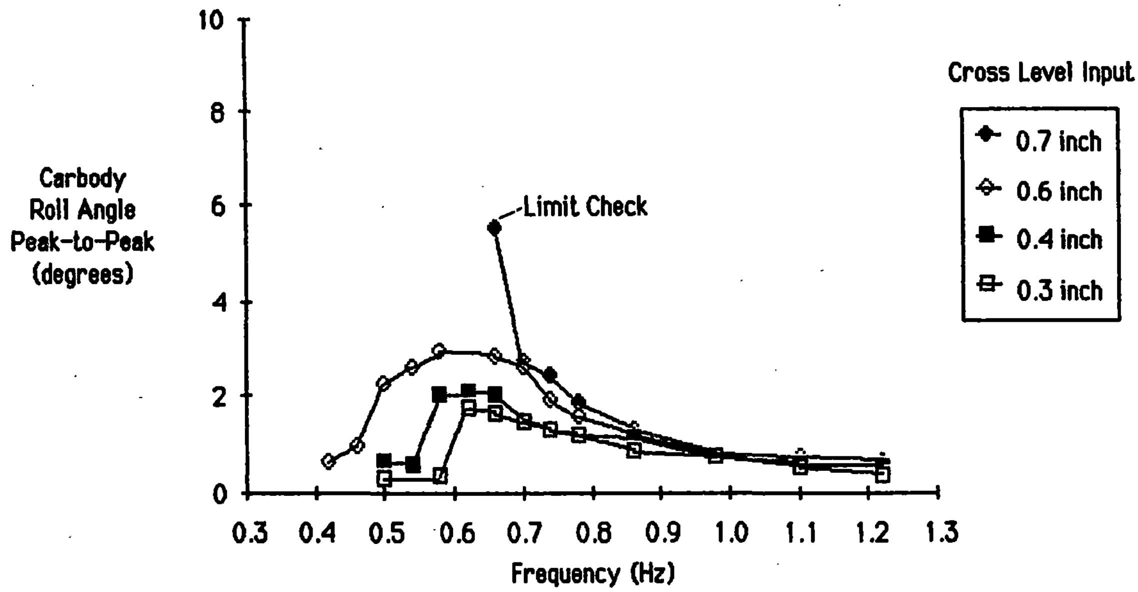


FIGURE 5-9
RECTIFIED SINE SIMULATION OF STAGGERED RAIL,
CARBODY ROLL RESPONSES, BASE VEHICLE NO. 1

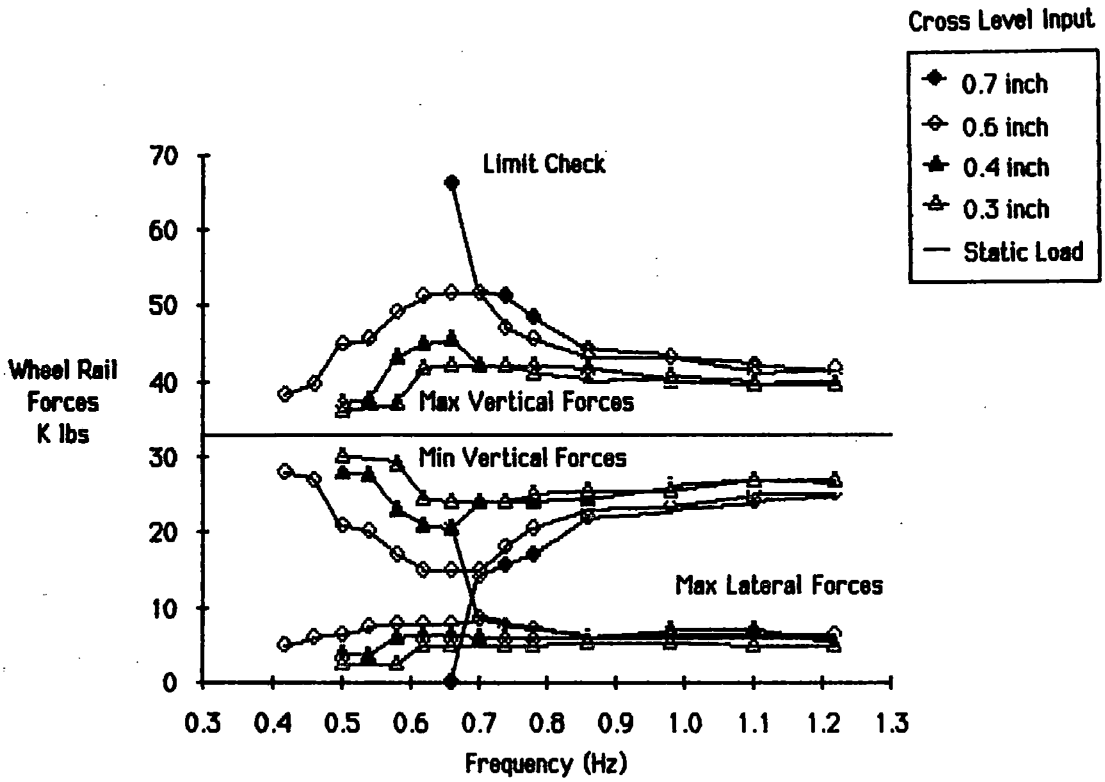


FIGURE 5-10
RECTIFIED SINE SIMULATION OF STAGGERED RAIL,
WHEEL-RAIL FORCES, BASE VEHICLE NO. 1

TABLE 5-4
COMPARISON OF SINE AND RECTIFIED SINE TEST RESULTS

TEST	ROLL ANGLE DEGREES PEAK-TO-PEAK	DYNAMIC WHEEL LOAD	
		VERTICAL KIP	LATERAL KIP
Sine Test 0.50-inch cross level	6.2	19.5	9.4
Rectified Sine Test 0.60-inch cross level	3.0	18.8	8.8

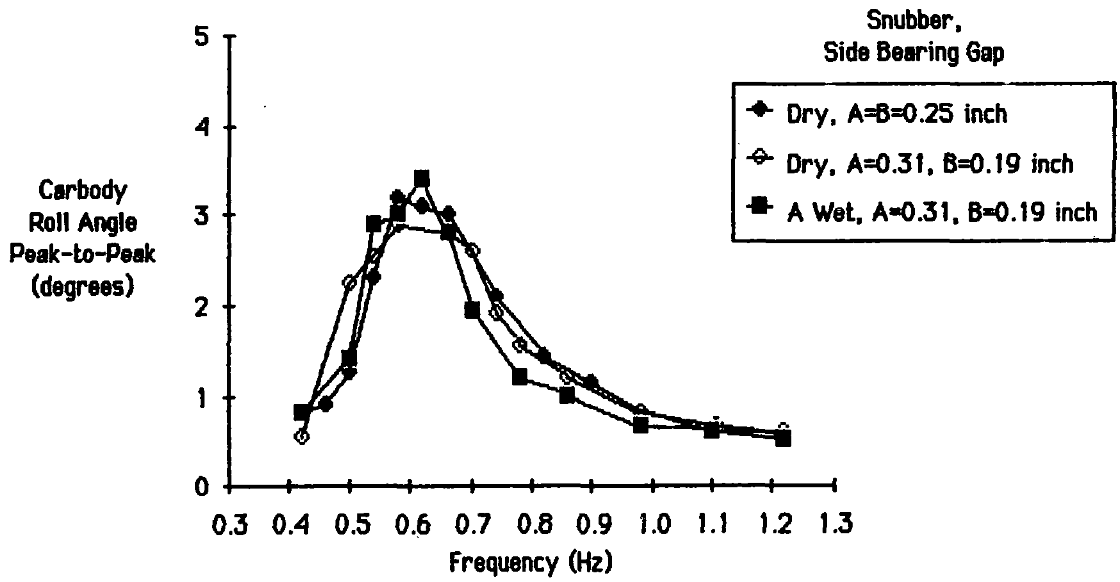


FIGURE 5-11
EFFECTS OF WET SNUBBERS AND SIDE BEARING GAP CHANGE ON CARBODY ROLL, 0.60-INCH CROSS LEVEL, RECTIFIED SINE, BASE VEHICLE NO. 1

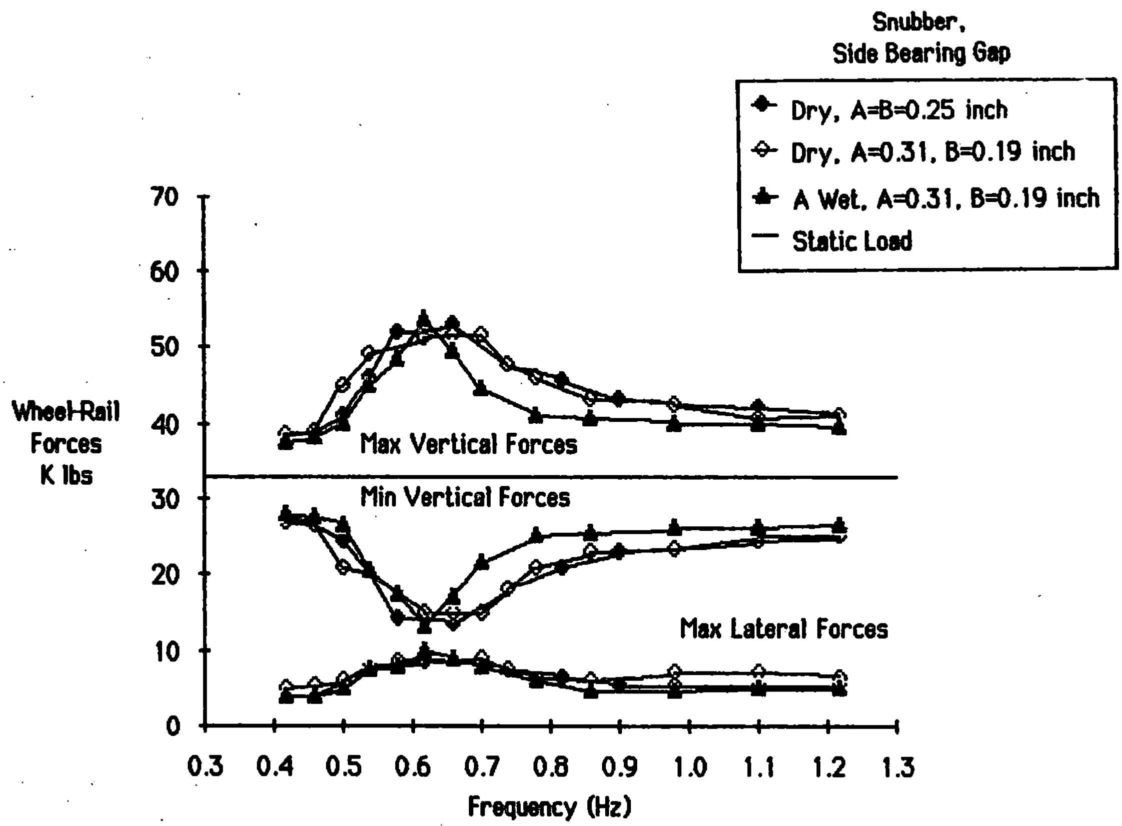


FIGURE 5-12
EFFECTS OF WET SNUBBERS AND SIDE BEARING GAP CHANGE
ON WHEEL RAIL LOADS, 0.60-INCH CROSS LEVEL, RECTIFIED SINE,
BASE VEHICLE NO. 1

Effect of A Truck Snubbers Binding

The peculiarities of the friction snubbers in the A truck were discussed earlier in this report. (Refer to Sections 3.5 and 5.4.) In the course of the Dynamic Response testing it was noted that the A snubbers were gradually returning to their original behavior of severe binding and explosive release. The motions of the B truck at this time were larger than the A truck, indicating the A friction to be larger than the B friction.

In order to evaluate the effects of the A snubber behavior, a repeat run was made for 0.60-inch cross level rectified sine with water sprayed on the A snubbers. The effect of water was very noticeable in the A snubber behavior: the binding was essentially eliminated and motions in the A and B truck were about equal. The effect on carbody roll and wheel-rail loads is shown in Figures 5-11 and 5-12. The plots show that with binding snubber the response is reduced by about 10% at critical speeds and is increased above critical speeds by varying amounts.

Test Results - Cross Level Index Evaluation

Limited amounts of the data from the CLI testing were studied and are summarized in Figures 5-13 and 5-14. A comparison of CLI and Dynamic Response test results is given in Figure 5-15.

The major portion of the CLI testing was done for staggered rail with an exponential profile shape. The exponential and rectified sine profiles look very much the same (refer back to Figure 5-4), the significant difference being that the exponential profile has a sharper cusp at the low joint than does the rectified sine. The other and probably more significant difference between CLI and Dynamic Response testing is that the latter tests were all sweeps whereas the CLI tests were constant speed with a preset number of low joints. (See Figure 5-5.)

There are a number of observations that are noteworthy in Figures 5-13, 5-14, and 5-15. First, the effect of reduced snubber is shown to result in an increased carbody roll angle of 20% to 30% and for wheel lift to occur at an input cross level that is 10% to 15% lower. A second observation is that once a wheel lift condition is reached there usually is an abrupt increase in the harmonic roll motions.

A third observation, from Figure 5-15, is that the carbody responses for rectified sine sweeps and exponential shape A are essentially the same, with exponential shape A reaching larger

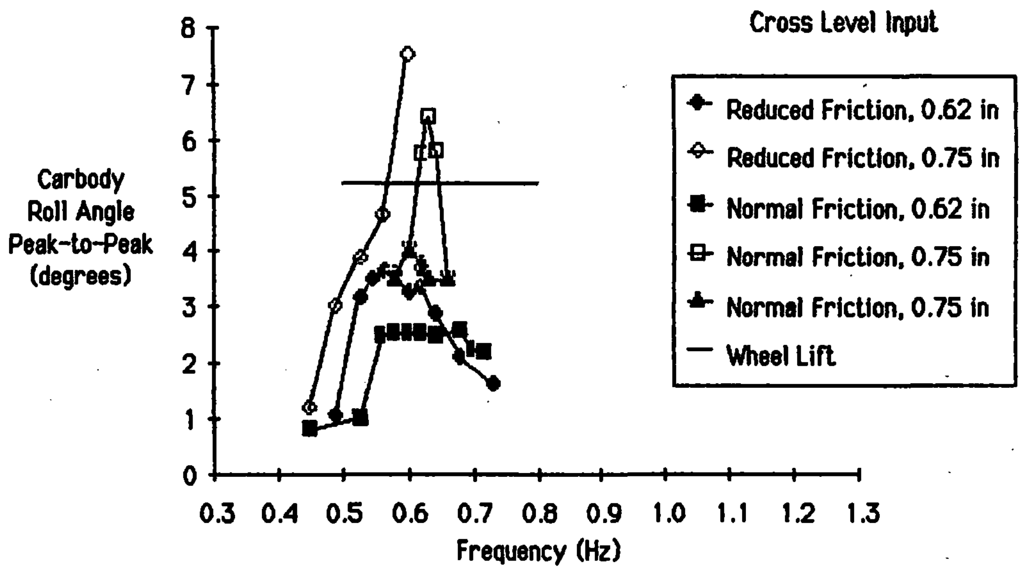


FIGURE 5-13
EXPONENTIAL LOW JOINT SIMULATION OF STAGGERED RAIL, CARBODY ROLL RESPONSE, BASE VEHICLE NO. 1

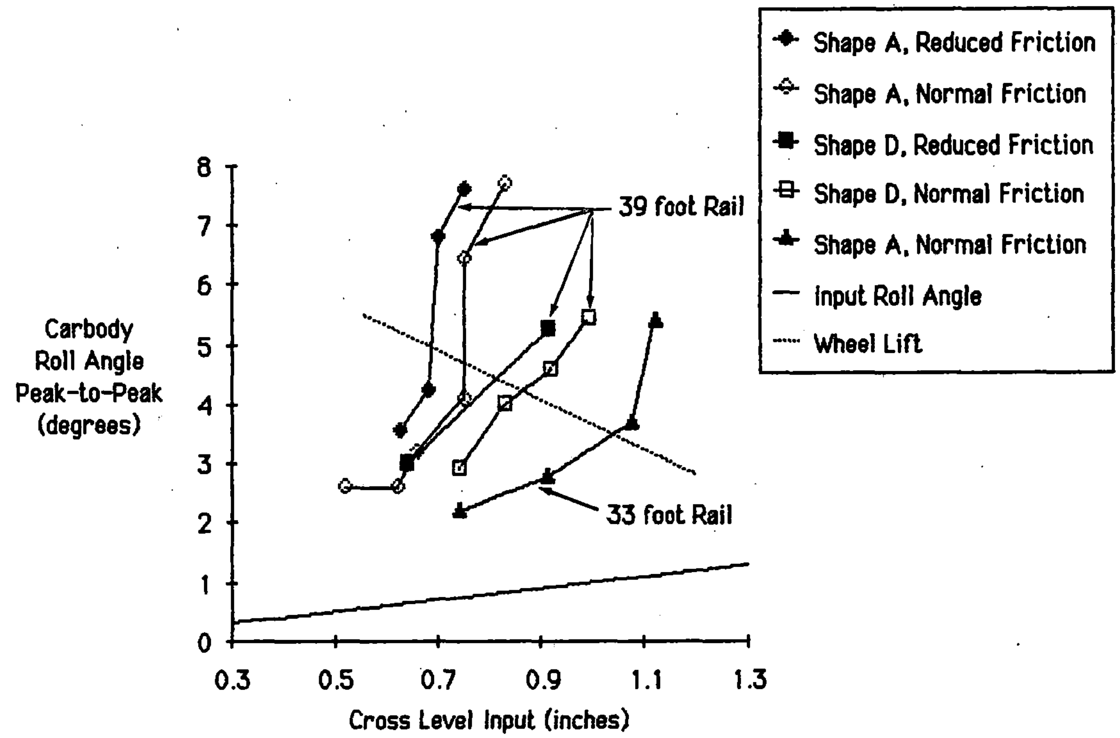


FIGURE 5-14
COMPARISON OF CARBODY ROLL RESPONSE FOR EXPONENTIAL PROFILE SHAPES, BASE VEHICLE NO. 1

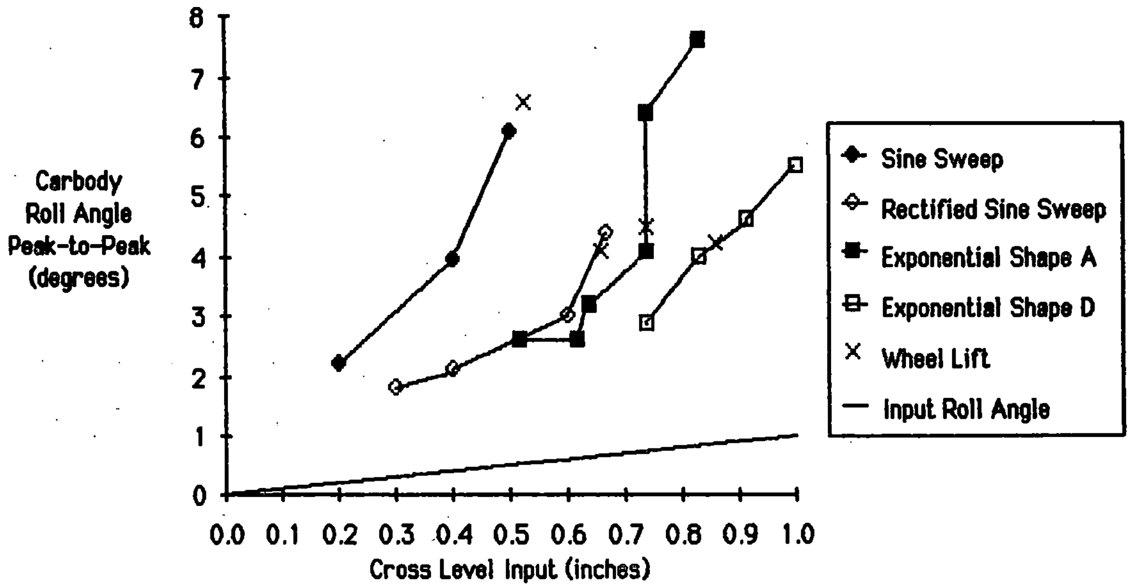


FIGURE 5-15
CARBODY ROLL RESPONSE FOR SEVERAL PROFILE SHAPES OF STAGGERED RAIL, 39-FOOT LENGTH, BASE VEHICLE NO. 1

cross level input before wheel lift (0.75 compared to 0.67 inch). However, this difference is probably because the rectified sine tests were performed with down sweeps, equivalent to speed slow-downs, and speed slow-downs are known to result in larger responses than constant speed runs for the same cross level conditions. A reasonable conclusion would be that the rectified sine and exponential low joints are essentially equivalent and that a slow down will result in larger roll responses than a constant speed run.

Sine sweep modal tests resulted in the largest carbody roll angles for the profiles and shapes tested. This is due partly to the sinusoidal profile shape being a more efficient driver and partly because the A and B end inputs were in-phase (that is, there was no phase lag due to difference between rail length and truck spacing). As expected, responses are lower when the cross levels are not constant in the staggered low joint simulation, or when the truck spacing does not match the rail lengths. Both of these effects are shown in Figure 5-14 in the shape D results and in the 33-foot rail results.

6. VIBRATION TESTS OF BASE VEHICLE NO. 2

Base Vehicle No. 2 was a 100-ton covered hopper that was 55 feet long with truck center spacing of 45 feet 7 inches. The car was received with Ride Control Trucks, which had constant column load friction snubbers and double roller side bearings. However, the car was tested both with the Ride Control trucks and with the Barber S-2-C trucks from Vehicle No. 1. In addition, a number of configurational variations were tested with the Barber trucks.

There were four sets of tests performed:

- Modal tests to define the resonant mode characteristics of the vehicle in each of the configurations tested.

- Sine Wave Track Geometry tests performed for the AAR to their requirements, consisting of sinusoidal cross level in combination with alignment variations.

- Safety Margin tests consisting of cross level, bounce and yaw input motions to determine critical track conditions and the effects of configuration variation. The sensitivity of roll response to profile shape was also investigated.

- CLI Track Geometry tests that repeated a limited set of tests performed on Base Vehicle No. 1 for Cross Level Index evaluation.

The results of this testing are presented in the following sections.

6.1 Test Objectives, Base Vehicle No. 2

The underlying objective of the testing of the Base Vehicles was to define their safety performance and to provide a basis for measuring the improved safety performance of prototype covered hopper cars.

The objective of truck characterization tests, vehicle modal tests, and the evaluation of the effects of configurational variations was to enable a better understanding of vehicle safety performance and how to effect improvements.

The objective of the CLI testing was to provide data in evaluation of a Cross Level Index developed by TSC [5].

6.2 Description of Base Vehicle No. 2

The Base Vehicle No. 2, 100-ton covered hopper car is pictured in Figure 6-1 showing its orientation on the VTU and giving some dimensional information. Base Vehicle No. 2 was five feet greater in length and three inches less in height than Vehicle No. 1; both of these differences result in Vehicle No. 2 having better performance than Vehicle No. 1 in harmonic response to 39-foot staggered rail.

Additional basic data for Base Vehicle No. 2 are presented in Table 6-1. Note that because the grain lading used (Sorghum) has higher specific weight than the red cracked wheat used in Vehicle No. 1, Vehicle No. 2 was not filled to its volumetric capacity at the full load condition. This resulted in Vehicle No. 2 center of gravity being at 89.2 inches compared to 97.0 inches for Vehicle No. 1 with both vehicles at approximate full load.

The configurations tested are identified in Table 6-2. The two side bearing configurations used, resilient constant contact and double roller, are shown in Figure 6-2. Descriptions of the trucks are given in Sections 3 and 4 and are not repeated here except to note that the friction snubbers in the A end Barber truck continued to bind and to have higher friction forces than in the B truck. Table 6-3 is a matrix identifying the tests performed on each configuration and also showing run numbers.

6.3 Modal Tests of Base Vehicle No. 2

The modal tests consisted of inputting sinusoidal motions at the vehicle wheels, vertically and laterally, to cause the vehicle to respond in its several modes of resonance. Resonance was identified by maximum response and 90° phase between response and input.

The results of the modal tests are of interest since these properties determine the dynamic behavior of the vehicle on the track. For example, the vehicle motion driven by staggered rail is in the first roll mode; consequently, the vehicle first roll mode characteristics will determine the vehicle's performance in harmonic roll. In a similar fashion, the yaw mode relates to body hunting; the bounce and pitch mode relates to the negotiation of grade crossing and other track profile perturbations; and the body torsion influences curve entry and exit.

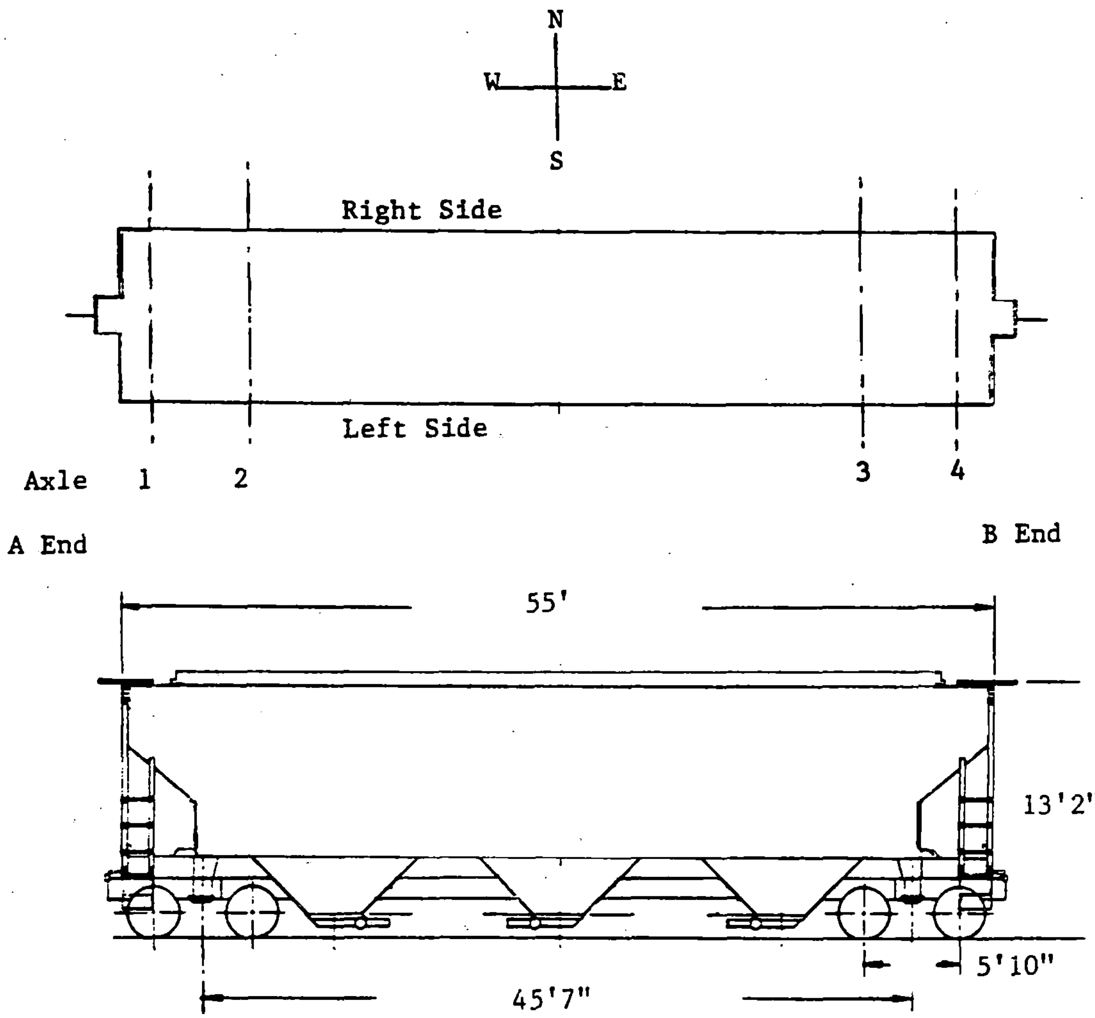


FIGURE 6-1
BASE VEHICLE NO. 2, ORIENTATION ON THE VIBRATION TEST UNIT

TABLE 6-1
VEHICLE NO. 2 BASIC DATA

Number	MP 723288	
DES Date	6/82	
Capacity	202,000 lbs.	4,750 cu. ft.
Load Limit	202,300 lbs.	
Light Weight	60,700 lbs.	
Wheel Diameter	36 in.	
Journal	6 1/2 x 11 in.	
Axle Spacing	5 ft., 10 in.	
Truck Spacing	45 ft., 7 in.	
Trucks	Ride Control Constant Column Friction Snubbers Double Roller Side Bearings D-5 Spring Group - 7 Outers and 9 Inners (Vehicle No. 2 was tested with the Barber as well as Ride Control trucks. See Table 6-2 for configurations tested)	
Center Plate	16 in. diameter, tapered	

Test Vehicle Measured Weights with Sorghum Lading

Condition (1), Full Load	
Gross Weight	258,650 lb.
Lading Weight	197,950 lb.
Condition (2), Reduced Load	
Gross Weight	205,530 lb.
Lading Weight	144,830 lb.

Center of Gravity Heights, from top of rail

Empty Vehicle	61.0 in.
Capacity Weight, 262,700 lb. (cubed)	95.0 in.
Test Condition (1), 258,650 lb.	89.2 in.
Test Condition (2), 205,530 lb.	81.6 in.

TABLE 6-2
TEST CONFIGURATIONS, BASE VEHICLE NO. 2

NOTATION DESCRIPTION

With Ride Control Trucks

RC2	Loaded Car

With Barber S-2-C Trucks

B0	Empty Car, no snubbers
B1	Empty Car, normal truck
B2	Loaded Car, normal truck
B2.1	Loaded Car, no snubbers
B3	27.6% off load, normal truck, lowered c.g.
B4	Loaded Car, reduced friction
B5	Loaded Car, reduced friction, auxiliary hydraulic snubbers
B6	Loaded Car, 5/16" side bearing clearance
B6.1	Loaded Car, 1/4" side bearing clearance
B7	Loaded, Car, constant contact side bearings

DEFINITIONS

Loaded car: 197,950 lb. of grain lading (sorghum), vehicle c.g. at 89.2 in.

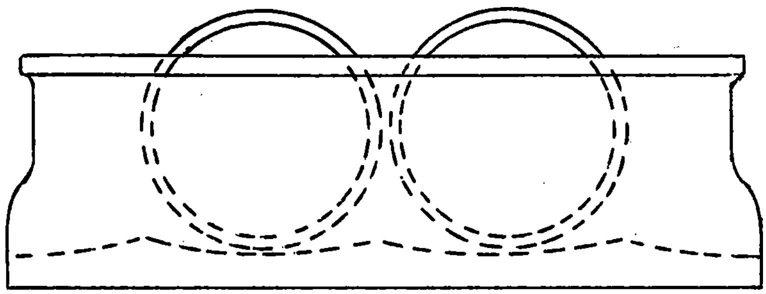
Lowered c.g.: 144,830 lb. of sorghum lading, vehicle c.g. at 81.6 in.

Normal truck: Inner and outer side springs

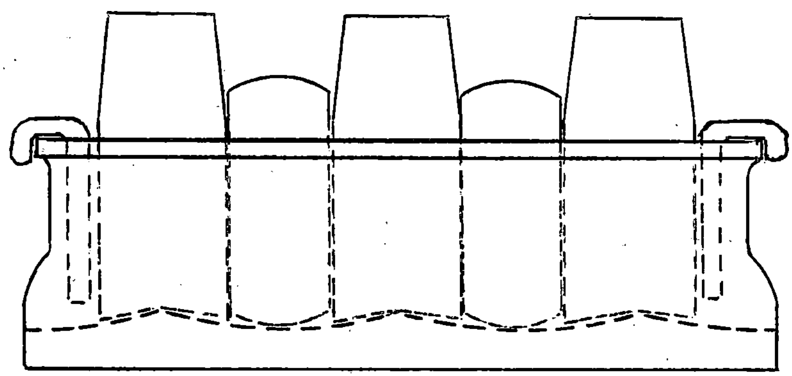
Reduced friction: Outer side springs only

Hydraulic Snubbers: Railroad Dynamics Inc., MDA Control Master model no. D-5 standard unit.

Side Bearing: Double roller or constant contact (see Figure 6-2)



A. Double Roller



B. Constant contact resilient

FIGURE 6-2
SIDE BEARING CONFIGURATIONS

TABLE 6-3
BASE 100-TON COVERED HOPPER CAR VTU TEST 1983:
RUN NUMBERS OF TESTS PERFORMED

TEST	CONFIGURATIONS										
	RC2	B2	B4	B2.1	B5	B6	B6.1	B7	B3	B1	B0
Modal	1-24	186-203	357-360	401-411	412-416			535-541	583-592	670-684 732 783-786	787-798
Shimmed Rail	53, 54, 79-99 115-128	227-254									
Rec. Sine Tangent	33-52	255-266	361-370		417-428	470-480	504, 506-513	542, 543 545-559	593, 597-605 618, 619	685-699	
Rec. Sine W/lateral	55, 129-143	267-280	371-379		429-439	481-491	505, 514-524	560-570 544	594, 595 596 607-617	700-712	
Rec. Sine W/super el.	100-114 154, 155	282-292	380-387		440-450	492-503	526-534	571-582	620-632	713-731	
Bounce	25-32 144-150 153, 156-159	293-304 330-356	388-397		451-465				633-638	753-764	
Yaw	151, 152 161, 162 163	204-207	398-400		466-469				649-652		
AAR Sine	56-77	208-226								733-752	
TSC Track Shape D	164-185	308-329							639-648 653-669	765-781	
Overall	1-185	186-356	357-400	401-411	412-469	470-503	504-534	535-582	583-669	670-786	787-798

The purpose of the modal testing, consequently, is to define these dynamic properties of the vehicle so that they can be used in the study of why the vehicle's dynamic performance is as it is and what changes are likely to effect improvement.

A summary of the modal frequencies obtained from the testing is given in Table 6-4 in a matrix of mode and configuration. In most instances, the modal frequency is given as a frequency band (for example, the first roll mode of configuration RC2 is between 0.53 and 0.75 Hertz). In most instances this is due to the resonant frequency changing with changes in input amplitude. In some cases, the response peak is broad and is best defined by a frequency band rather than the center frequency. In some cases, resonance identification is difficult and again a frequency band must be used.

The frequency data in Table 6-4 are also plotted in Figure 6-3 to illustrate the total range of frequency variation of each mode in the empty and loaded configurations. Table 6-5 lists the total range of frequencies for each mode and makes comparison to pre-test estimates.

6.3.1 First Roll Mode Data, Base Vehicle No. 2

The first roll mode has greater influence on safety performance of rail cars than any of the other modes and, consequently, more attention was given to this mode. Results are shown in Figures 6-4 through 6-12 in plots of carbody roll angle for nine of the configurations tested and one plot of wheel-rail vertical loads. Finally, Figure 6-13 is a cross plot of maximum response points from Figures 6-4 through 6-12, omitting empty vehicle configuration.

Based on carbody roll response data plotted in Figure 6-13, Configuration 5, the Barber truck with reduced friction snubbing and auxiliary hydraulic snubbers, is the best performer. Configurations with the largest, and about equal responses are RC2, B4 and B7. (RC2 is with Ride Control trucks, B4 is Barber with reduced friction, and B7 is Barber with resilient side bearings.)

Generally the better performers at large input motions are the poorer performers at small input, with the cross over being at about 0.30 inches cross level. This verifies the general observation that while increased snubbing will effectively limit the larger amplitudes, it also causes a rougher ride on track with smaller geometry variations.

TABLE 6-4
MODAL FREQUENCIES BY CONFIGURATION, BASE VEHICLE NO. 2

MODE	CONFIGURATIONS (FREQUENCY AT MAXIMUM RESPONSE - HERTZ)								
	RC2	B0	B1	B2(1)	B2.1	B3	B4	B5	B7
First Roll	.53-.75	.65-.85	.65-1.00	.59-.73	.50-.53	.66-.73	.55-.65	.53-.70	.48-.75
Yaw	2.7-3.0	2.6-2.7	5.6	2.4-3.1	2.1	3.4-4.5	2.4-3.0	2.6-3.4	x
Bounce	2.3-2.5	4.2	3.9	2.6-2.8	2.4	(2)	(2)	3.01	x
Second Roll	3.4-3.6	2.7	3.9+	3.7-3.9	x	3.1-3.5	x	x	x
Pitch - 2 end	3.6-4.0	4.9	5.6	4.3-4.6	2.8-3.4	4.25	3.0	x	x
Pitch - 1 end	2.4	x	4.0	3.0	x	x	2.4	x	x
Torsion	below 6.0	x	8.5	4.0-4.3 6.8-7.0	x	x	x	x	x
Bending - Lateral	7.5	x	x	8.7-10.0	x	x	x	x	x
Bending - Vertical	7.3,14.8	x	x	7.4,14.3	x	x	x	x	x

(1) B6 and B6.1 were assumed to be the same as B2
(2) No bounce mode apparent
x Not tested for

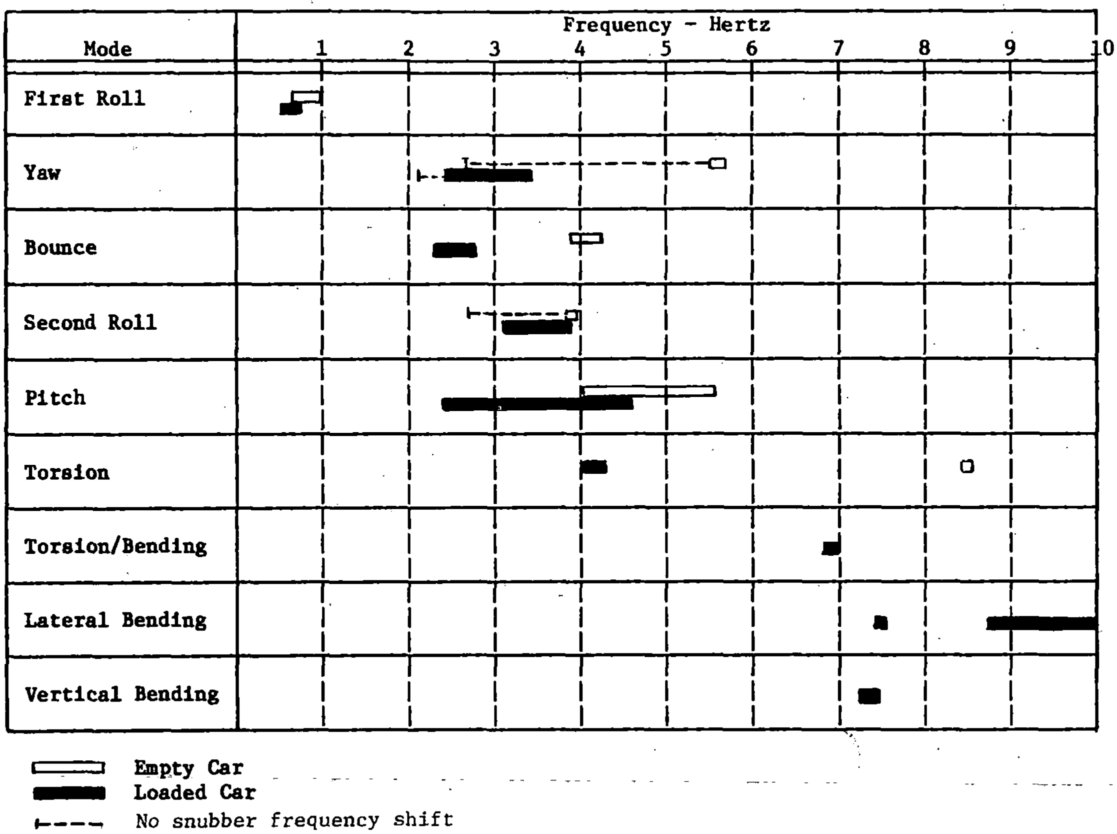
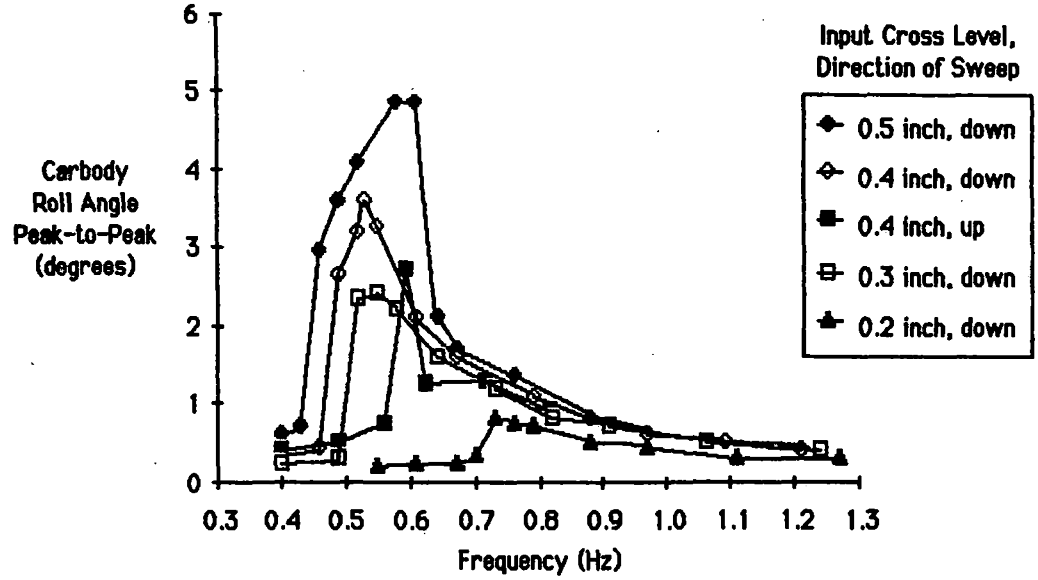


FIGURE 6-3
RANGE OF MODAL FREQUENCIES FOR BASE VEHICLE NO. 2

TABLE 6-5
100-TON COVERED HOPPER CAR RESONANT FREQUENCY SUMMARY

MODE	ESTIMATED FREQUENCY (HERTZ)	TEST FREQUENCY (HERTZ)
First Roll	.70 - .90	.47 - 1.00
Yaw	2.0 - 3.0	2.1 - 3.4
Bounce	2.5 - 3.0	2.3 - 4.2
Second Roll	3.0 - 3.5	2.7 - 3.9
Pitch - Two end	3.5 - 4.0	2.8 - 5.6
Pitch - One end		2.4 - 4.0
Body Torsion	8.0 - 14.0	4.0 - 4.3
Coupled Torsion -Lateral Bending		6.8 - 7.0
Body Bending, Lateral	10.0 - 16.0	8.7 - 10.0
Vertical	20.0 - 25.0	7.4 - 14.3

a. Carbody Roll Angle



b. Vertical Wheel Load

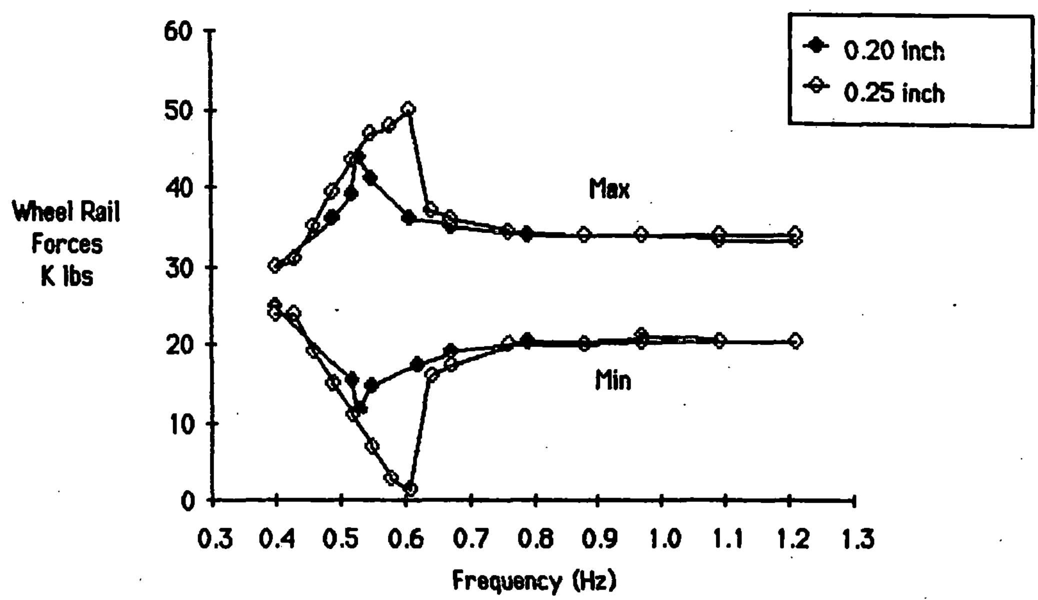


FIGURE 6-4
CONFIGURATION RC2, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

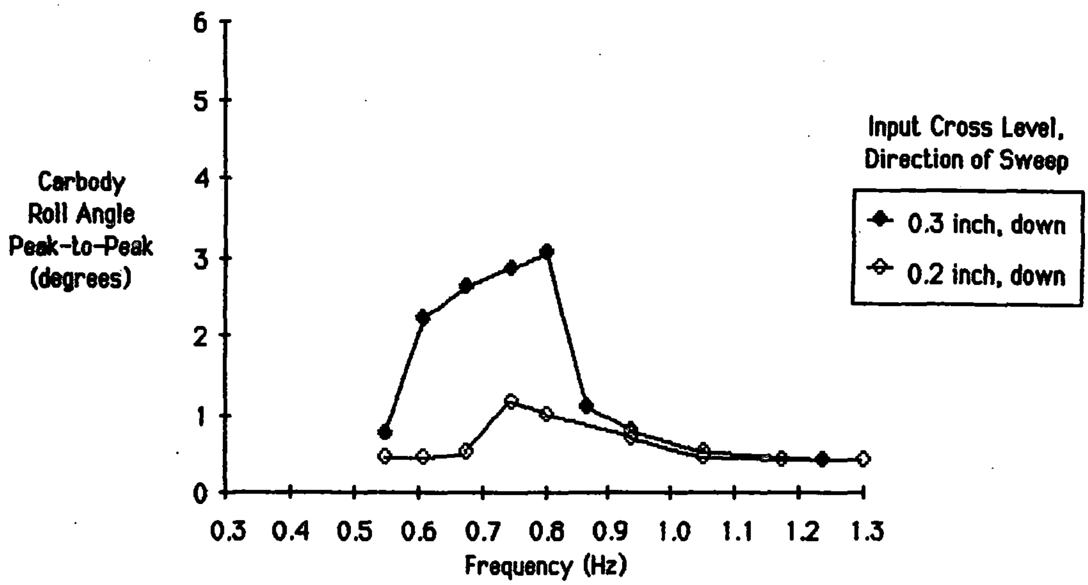


FIGURE 6-5
CONFIGURATION B0, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

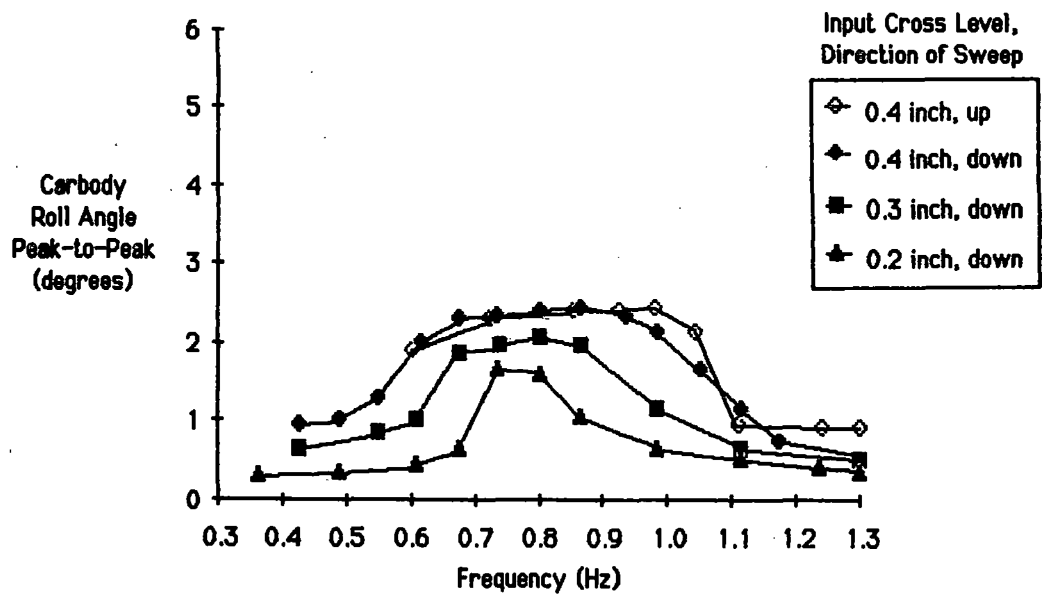


FIGURE 6-6
CONFIGURATION B1, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

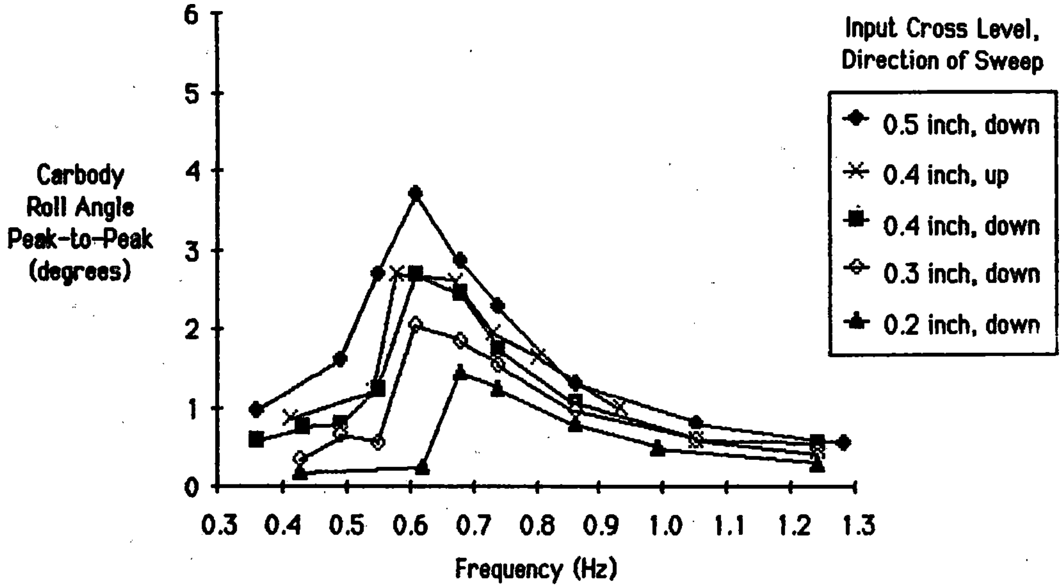


FIGURE 6-7
CONFIGURATION B2, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

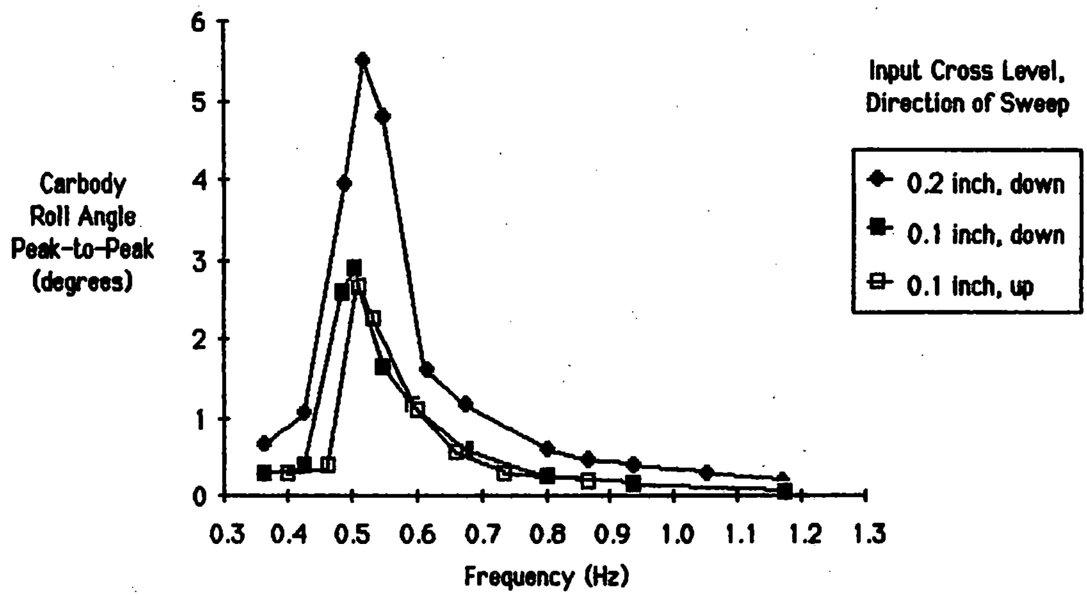


FIGURE 6-8
CONFIGURATION B2.1, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

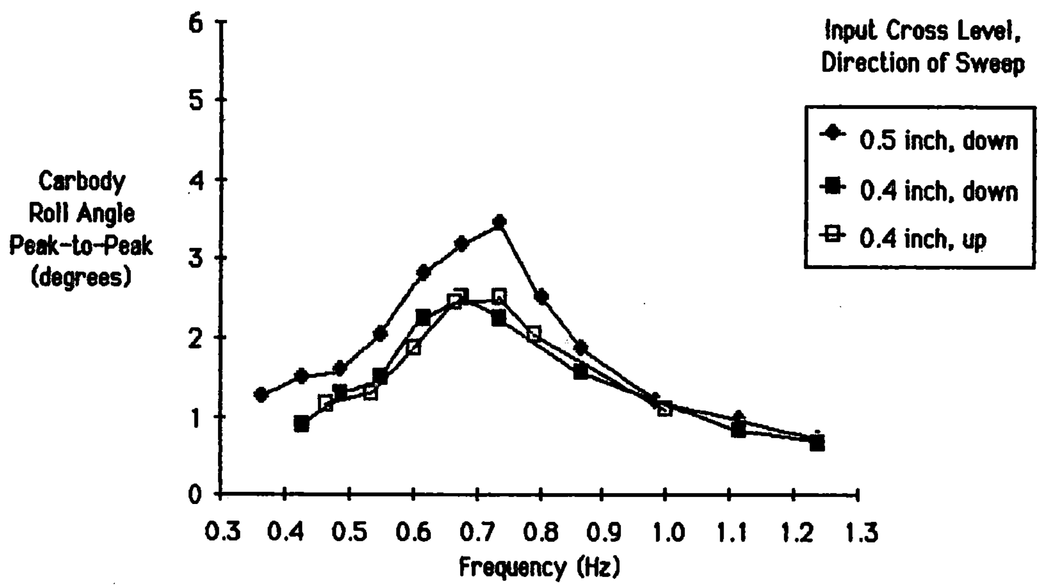


FIGURE 6-9
CONFIGURATION B3, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

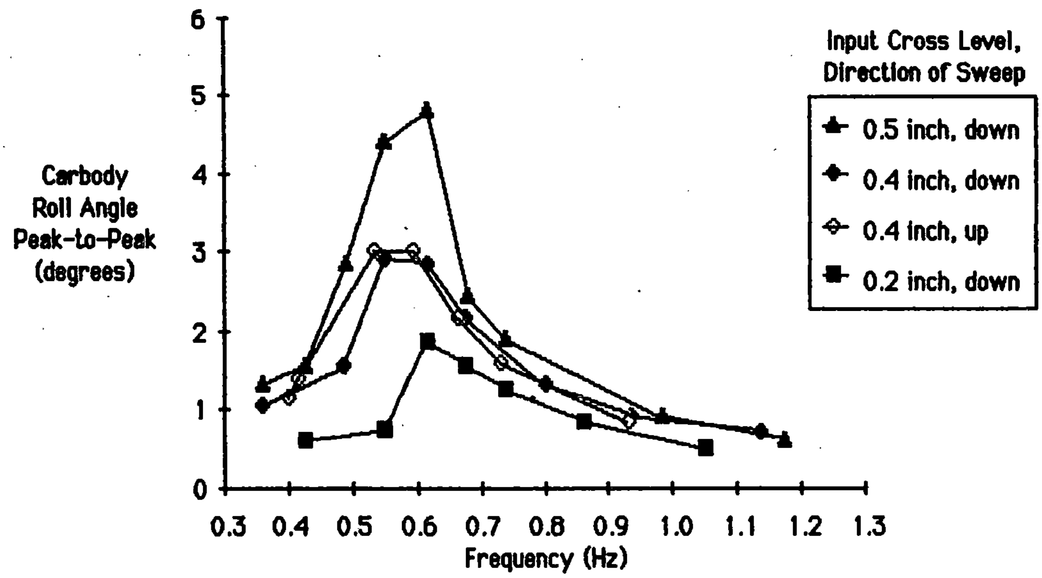


FIGURE 6-10
CONFIGURATION B4, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

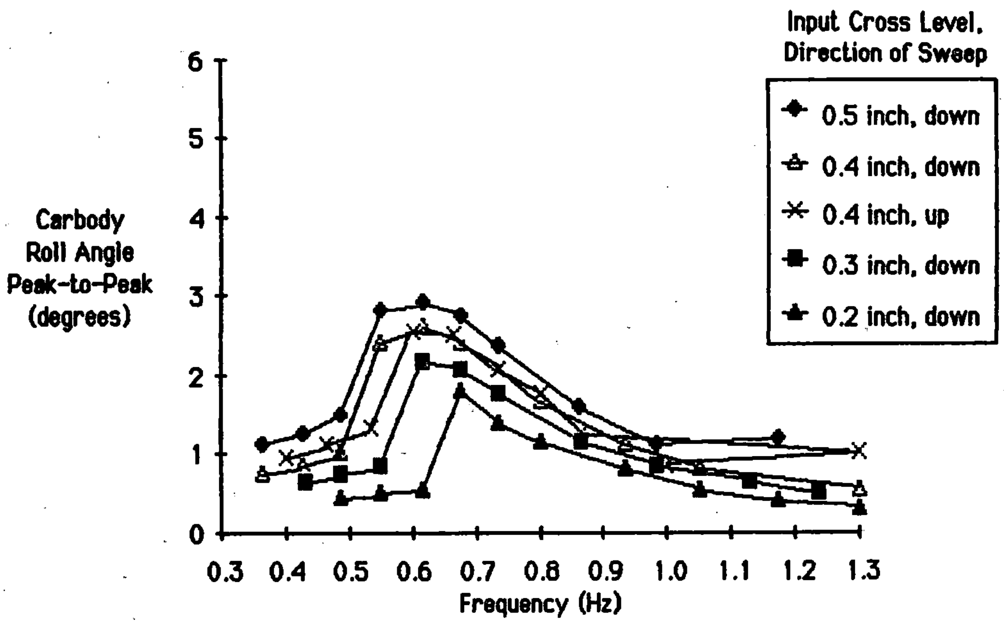


FIGURE 6-11
CONFIGURATION B5, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

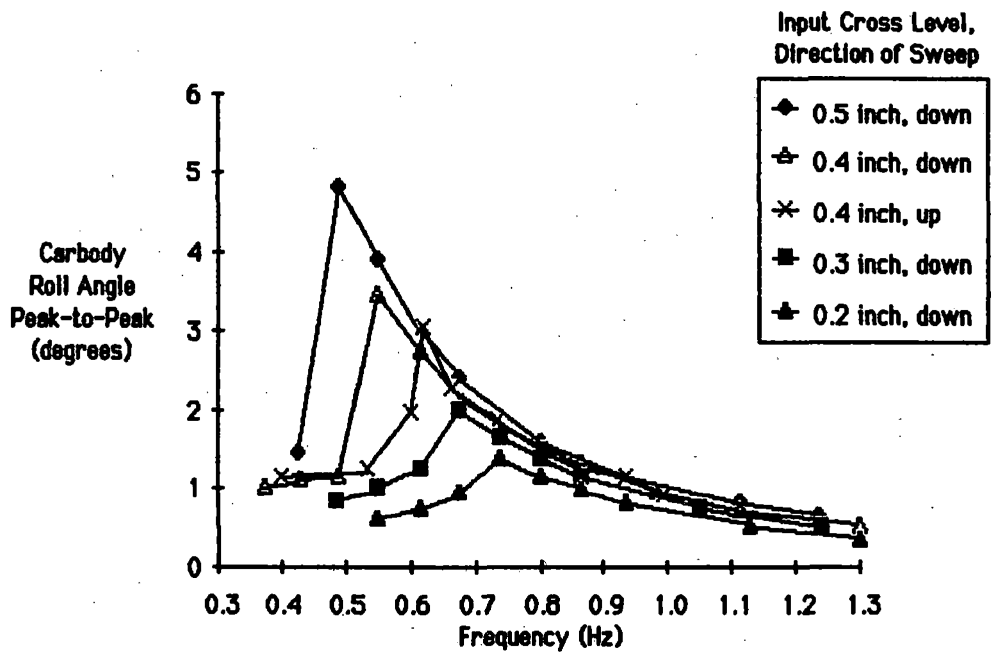


FIGURE 6-12
CONFIGURATION B7, FIRST ROLL MODE RESPONSE, SINE SWEEP TEST

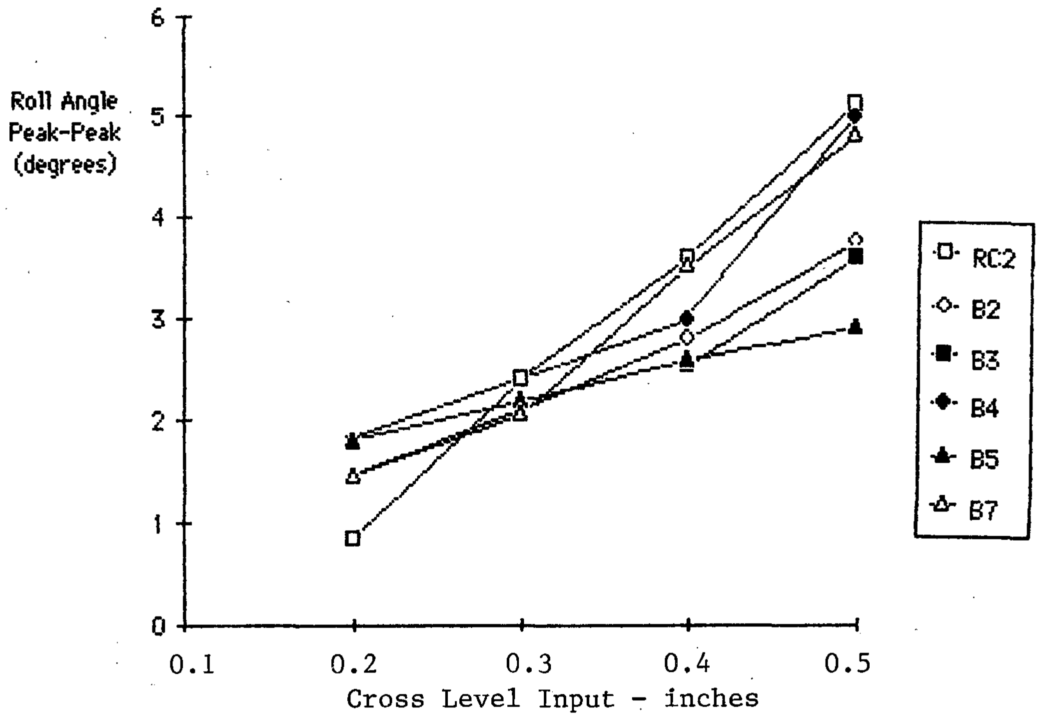


FIGURE 6-13
FIRST ROLL MODE MAXIMUM RESPONSE, COMPARISON BY CONFIGURATION

6.3.2 Observations on Modal Test Results

The comparison of modal frequencies from pre-test estimates and test results in Table 6-5 shows general agreement for most of the modes. However, there are several noteworthy differences. For one, most of the modes spread over a broader band of frequencies than had been expected. The first roll mode, for example, spreads from 0.47 to 1.00 Hertz compared to the estimated 0.70 to 0.90. One cause of these broadened modal frequency bands is the high friction forces in the Barber truck snubbers: for example, 13,300 pounds for the A truck and 7,800 pounds for the B truck sliding friction force at 90% load. These high snubber forces will act to both increase and decrease the modal frequencies. They will cause an increase with small motions because the snubbers are not sliding and the suspension system is, as a result, very stiff. Conversely with large motions, the larger snubber force cause a slow down of the motions and reduced frequencies. (This is similar to viscous damping causing the damped natural frequency of a system to be less than the undamped natural frequency.)

A second observation to be made from Table 6-5 is the presence of a second pitch mode noted as "one end" pitch. Because the Barber A end truck had larger snubber forces than the B end, there were conditions of input motions where the B snubbers would break (that is, start to slide) while the A snubbers would not. This resulted in a pitching motion with the B end at larger amplitudes and the A end at small amplitudes.

A third observation is that the flexural modes of the carbody, torsion and bending, are at lower frequencies than had been estimated.

Finally, the observation based on amplitude of carbody roll response is that increased snubber forces will result in improved performance for track cross level variations greater than 0.30 inches but will result in a poorer performance at smaller cross levels. This is discussed again in further detail in the staggered rail test results covered in Section 6.5.

6.4 Sine Wave Track Geometry Tests, Base Vehicle No. 2

These tests were requested by the Association of American Railroads Technical Center, Chicago, Illinois with the objective of defining the base vehicle's derailment tendencies. The AAR/TC would then use these results to correlate with their analytical predictions and to evaluate the relative performance of the prototype hopper cars. Consequently, the objective of the reporting of this document is only to present test results. The AAR will report separately on their evaluations.

The tests consisted of sinusoidal variations of cross level and alignment with wave lengths of 39 feet for cross level and 62.4 and 39 feet for alignment. The speeds of maximum response were found by performing step-sweeps from 11 to 30 miles per hour. A step-sweep consists of acquiring data at a constant speed, stepping up to the next speed and acquiring data at the new speed. The speed increment of 1.0 miles per hour was used and record time was set to include eight cycles of cross level variation. Amplitudes of cross level and alignment were increased to find the critical condition of wheel lift. Table 6-6 presents the matrix of cross level and alignment amplitudes used. The phase relationship between cross level and alignment are defined in Figure 6-14 for 62.4 foot alignment wave length and in Figure 6-15 for 39-foot alignment.

The alignment wave length of 62 feet was requested by the AAR based on track geometry measurement data. The 62 feet length was modified to 62.4, with AAR agreement, in order to have integer wave numbers of 5 and 8 for the alignment and cross level variations. This enabled an inclusion of a full range of phase relationships between cross level and alignment that was repeated every 312 feet of track length.

The testing was performed on three configurations of the Base Vehicle No. 2: RC2, B2, and B1. The RC2 configuration is the loaded vehicle with the Ride Control trucks. The B2 configuration is the same load configuration with the Barber S-2-C trucks. The B1 configuration is the empty vehicle with the Barber trucks.

Results of the Sine Wave Track Geometry testing are presented in the summary plots of carbody roll angle, wheel loads and carbody displacements in Figures 6-16 through 6-24. Comparisons of results between the three configurations tested are presented in Figures 6-25 and 6-26.

Notation is made on each plot for points at which the test was stopped because derailment margin criteria was reached (zero vertical wheel load or center plate lift).

Wheel lift (actually zero vertical wheel load) was used as the stop-test condition as a test convenience. It is a condition that can be measured with repeatable accuracy. It avoids the damaging loads, to test equipment, that accompany wheel lift

TABLE 6-6
INPUT AMPLITUDES FOR SINE WAVE TRACK GEOMETRY TESTING

TEST SEQUENCE	SINE HALF AMPLITUDE OF ALIGNMENT	SINE HALF AMPLITUDE OF PROFILE OF EACH RAIL				
		0	P1= .185 IN.	P2= .2775 IN.	P3= .37 IN.	P4= .4625 IN.
1	0		P1	P2	P3	P4
2	A1=0.5 in	A1	P1,A1	P2,A1	P3,A1	P4,A1
3	A2=1.0 in	A2	P1,A2	P2,A2	P3,A2	P4,A2
4	A3=1.5 in	A3	P1,A3	P2,A3	P3,A3	P4,A3

Notes:

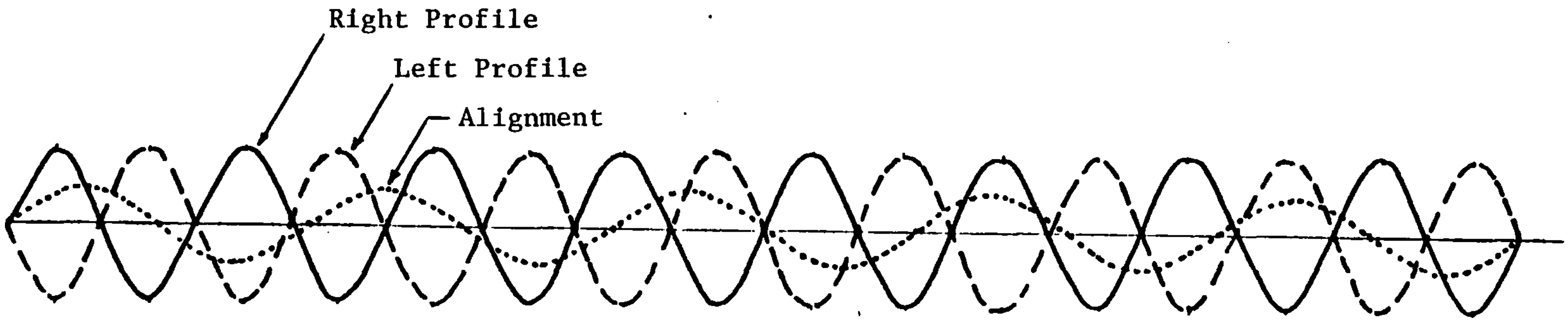
1. Rail profile variation will be 39-foot sinusoidal with right and left rails at 180° phase.

2. Phase lag between axles and trucks will be based on Vehicle No. 2 axle and truck spacings of 5' 10" and 45' 7", respectively.

3. Phasing of profile and alignment will be as shown in Figures 6-14 with 62.4-foot alignment wave length and as shown in Figure 6-15 for 39-foot alignment wave length. 62.4-foot alignment will be used in test sequences 2, 3, and 4. 39-foot alignment will be used in test sequence 2 only.

4. Frequency--speed relationship is:
   MPH = 26.591 x f. . . (39-foot rail)

5. Speed Sweep - 11 mph to 30 mph in 1 mph steps
   Sample Rate = 64 sps
   Dwell Time = 8 cycles of 39-foot wave



Profile Wave Length = 39 feet

Alignment Wave Length = 62.4 feet

Phase repeat every 312 feet (5 x 62.4 and 8 x 39)

Right Profile : $ZR = Pj \sin\left(\frac{2\pi V}{39} t\right)$

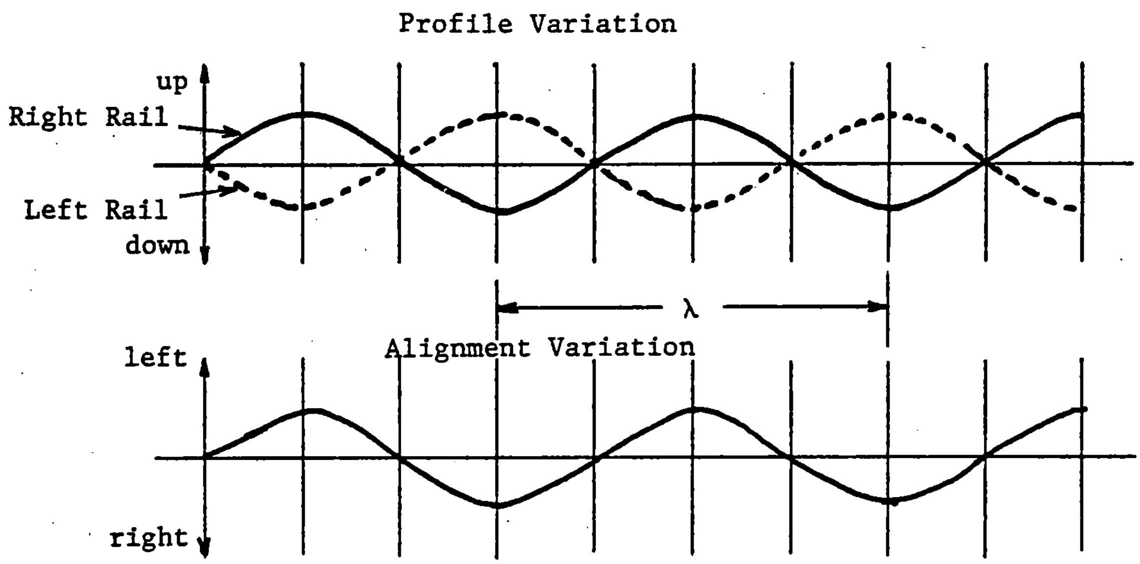
Left Profile : $ZL = -Pj \sin\left(\frac{2\pi V}{39} t\right)$

Alignment : $X = Aj \sin \frac{2\pi V}{62.4} t$

Pj = profile variation amplitude (Table 6-6)

Aj = alignment variation amplitude (Table 6-6)

FIGURE 6-14
SINUSOIDAL STAGGERED RAIL WITH 39-FOOT PROFILE AND 62.4-FOOT ALIGNMENT VARIATIONS



λ = 39 feet

Table 6-6, Test Sequence 2 only

FIGURE 6-15
SINUSOIDAL STAGGERED RAIL SIMULATION WITH ALIGNMENT VARIATION OF SAME WAVE LENGTH

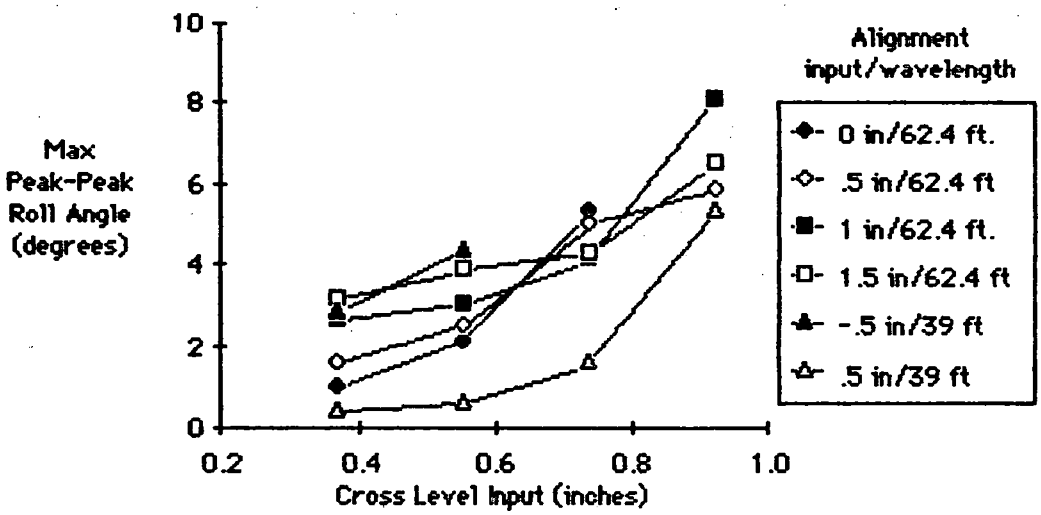
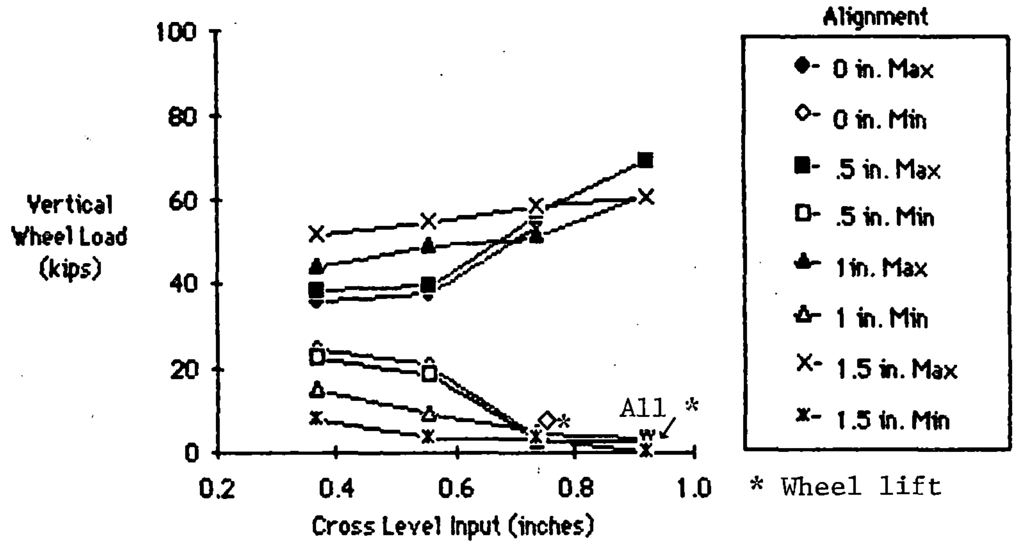


FIGURE 6-16
SINE TEST, CARBODY ROLL ANGLE, RC2 CONFIGURATION

a. Average of 29FZ and 31FZ, 62.4 ft. Alignment



b. 29FZ and 31FZ, 39 ft. Alignment

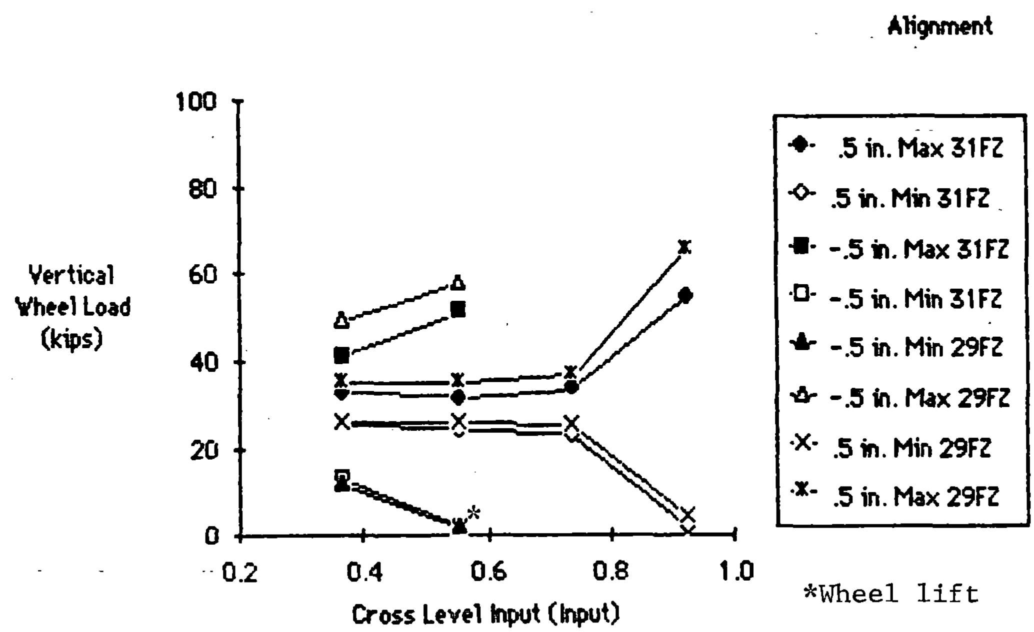
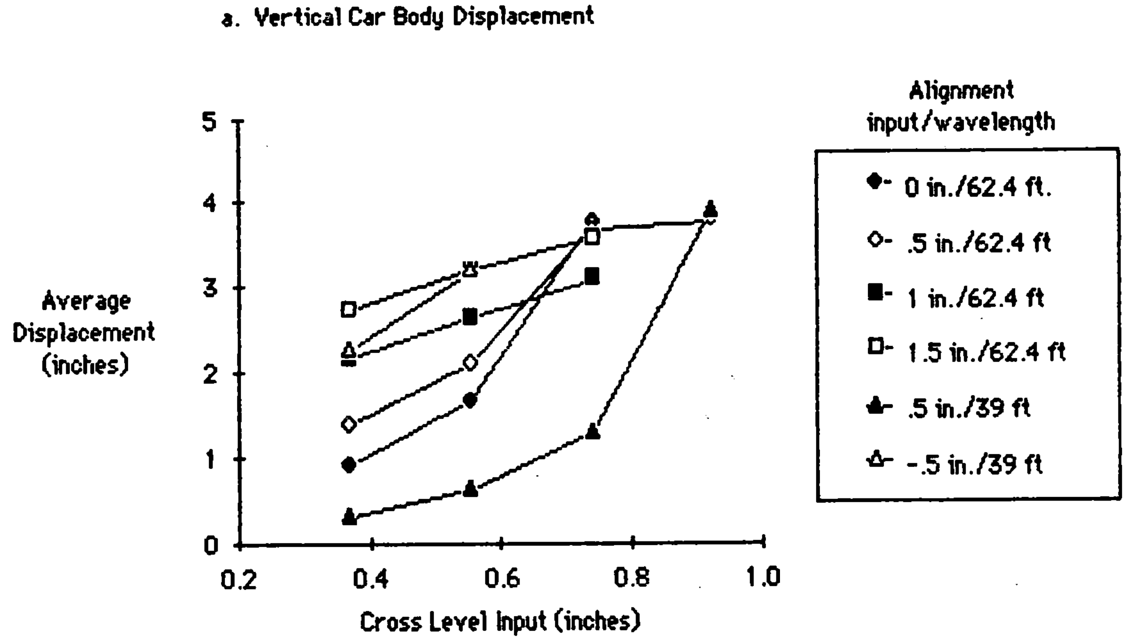


FIGURE 6-17
SINE TEST, WHEEL VERTICAL FORCES, RC2 CONFIGURATION



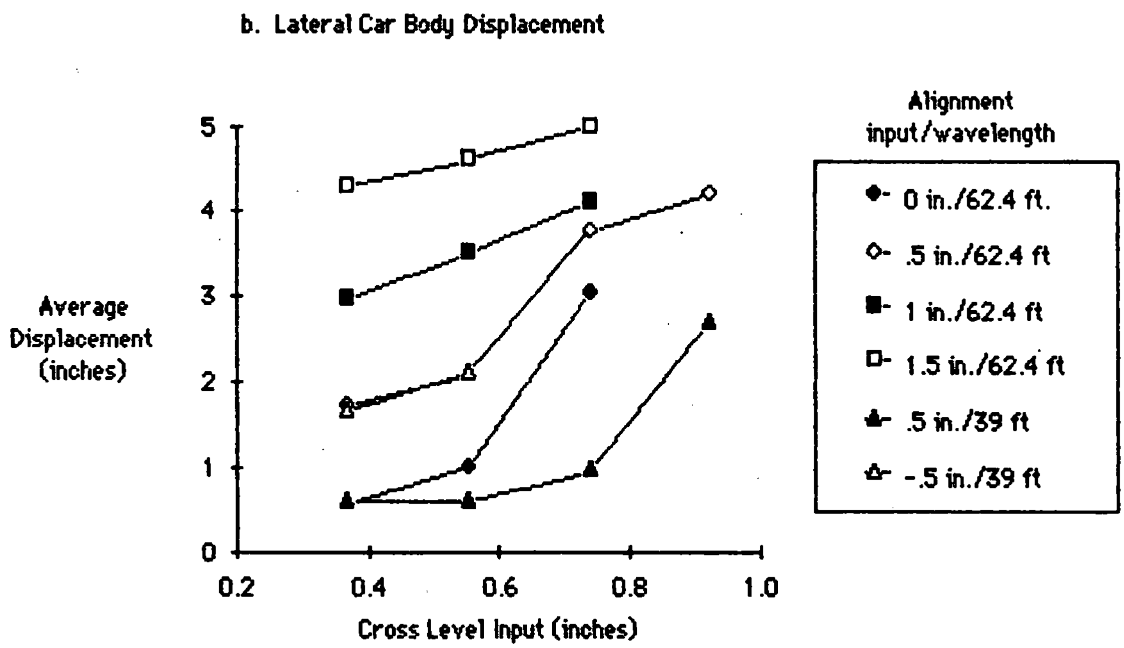
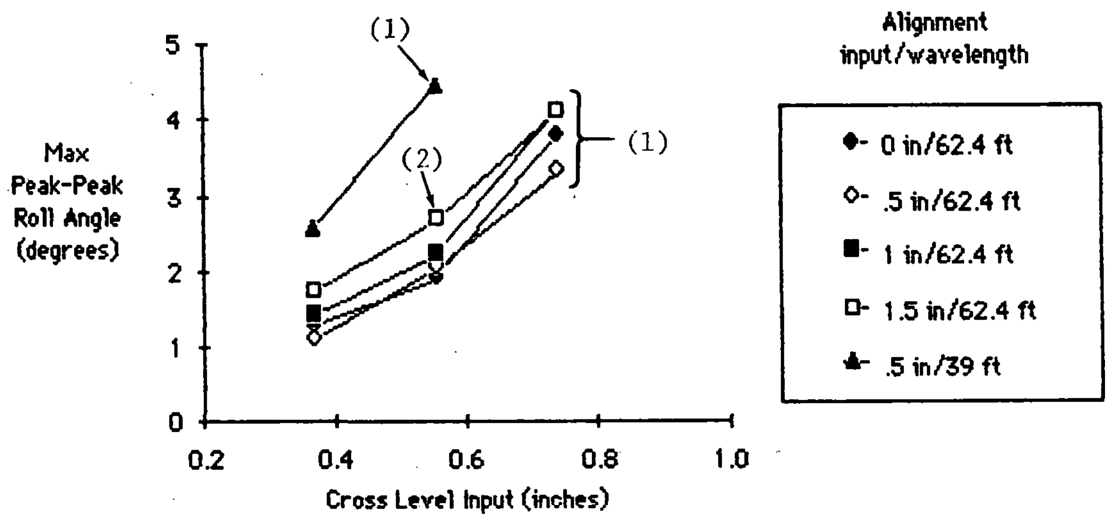


FIGURE 6-18
SINE TEST, CARBODY DISPLACEMENTS, RC2 CONFIGURATION

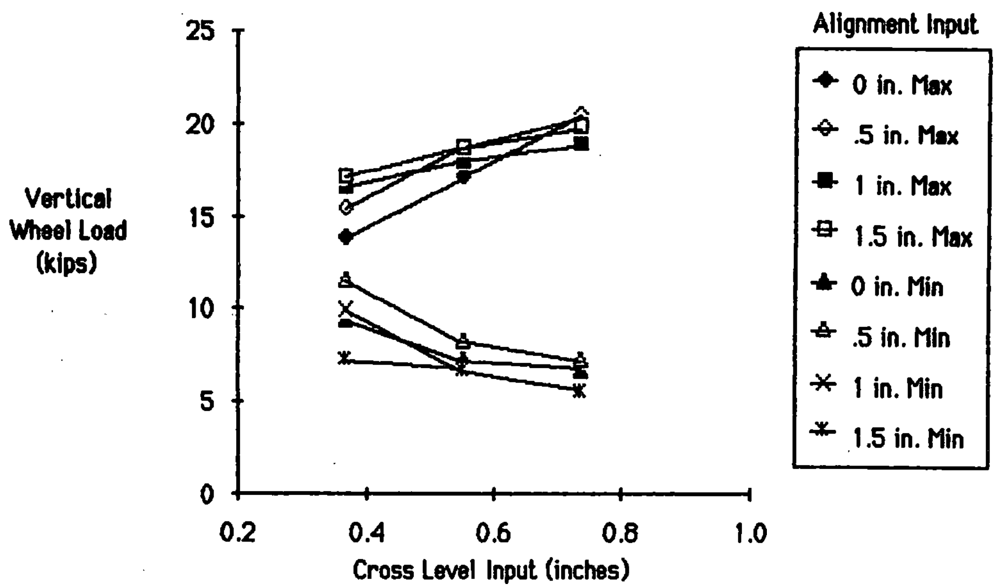


(1) Center plate lift, both trucks.

(2) Center plate lift, A truck only.

FIGURE 6-19
SINE TEST, CARBODY ROLL ANGLE, B1 CONFIGURATION

a. 62.4 ft. Alignment



b. 39 ft. Alignment

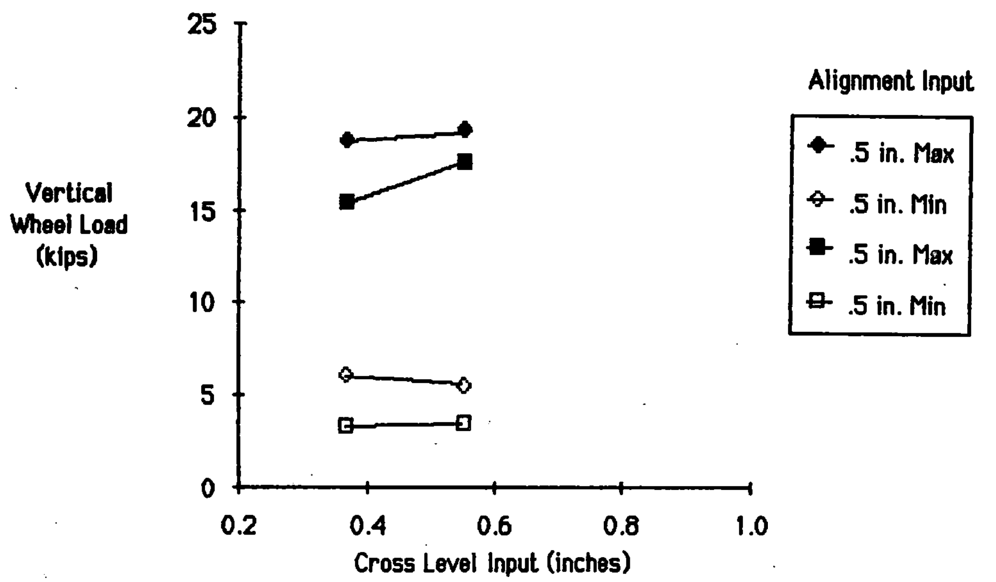
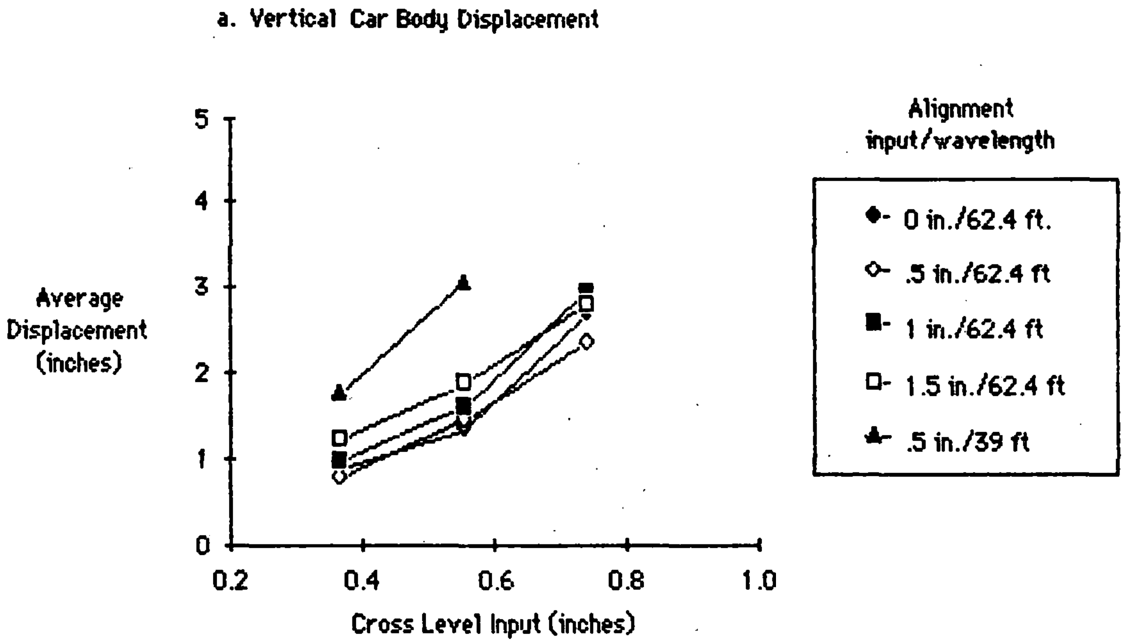


FIGURE 6-20
SINE TEST, WHEEL VERTICAL FORCE, B1 CONFIGURATION



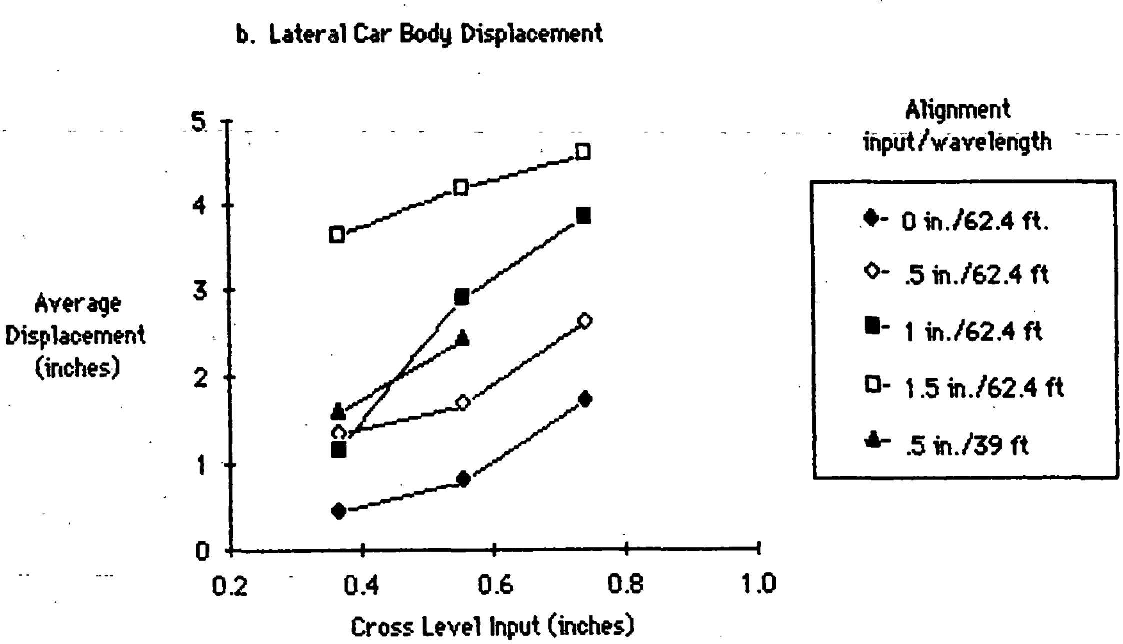


FIGURE 6-21
SINE TEST, CARBODY DISPLACEMENTS, B1 CONFIGURATION

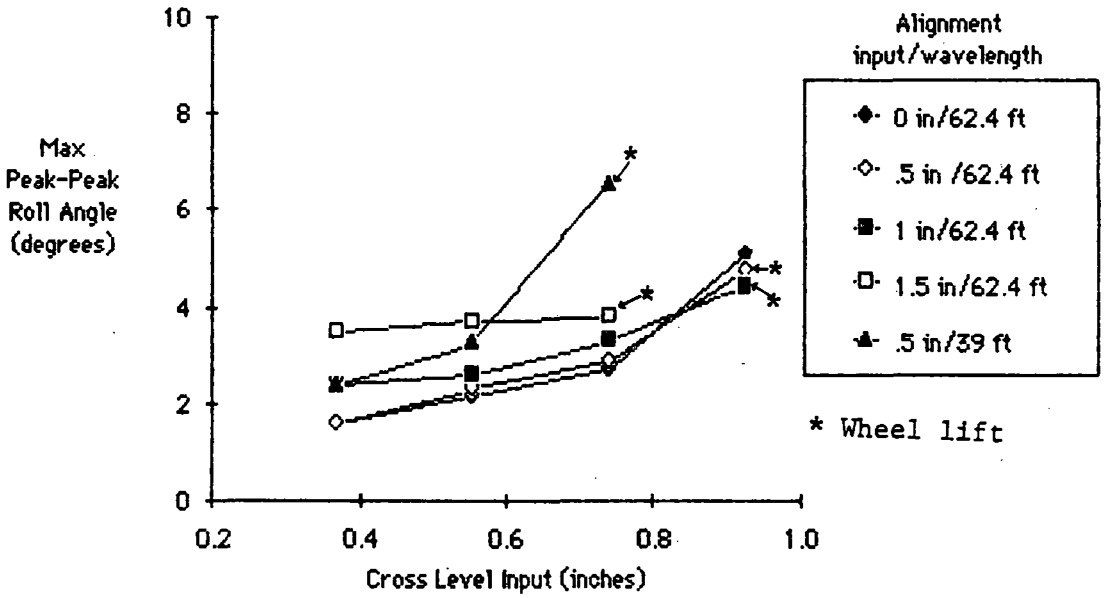
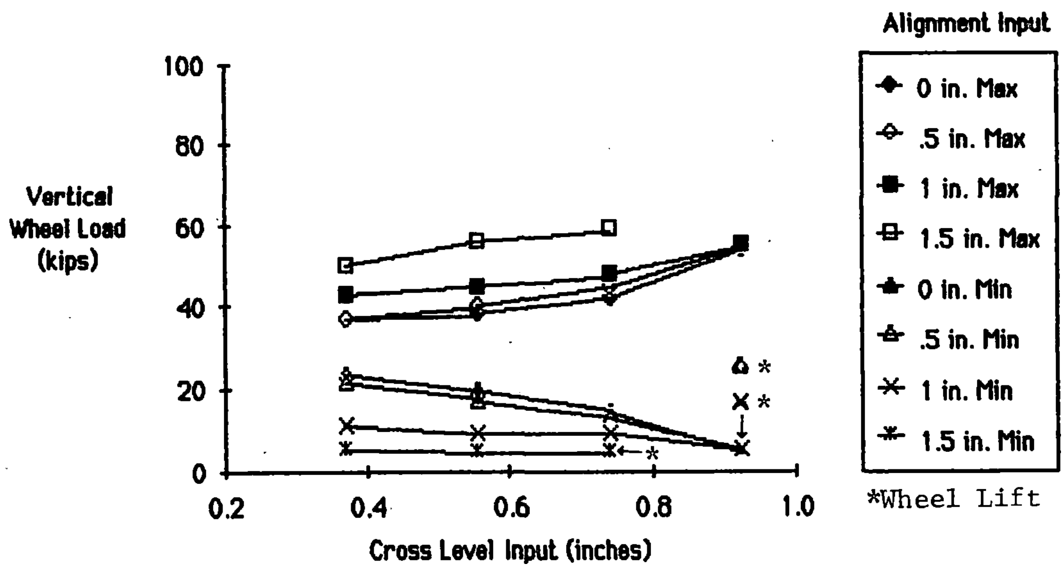


FIGURE 6-22
SINE TEST, CARBODY ROLL ANGLE, B2 CONFIGURATION

a. 62.4 ft. Alignment



b. 39 ft. Alignment

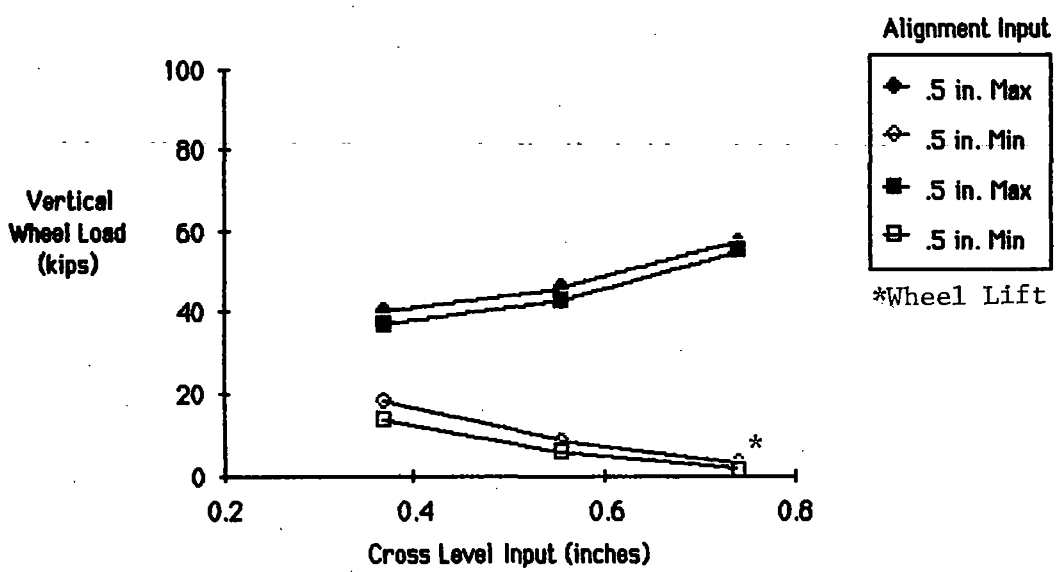
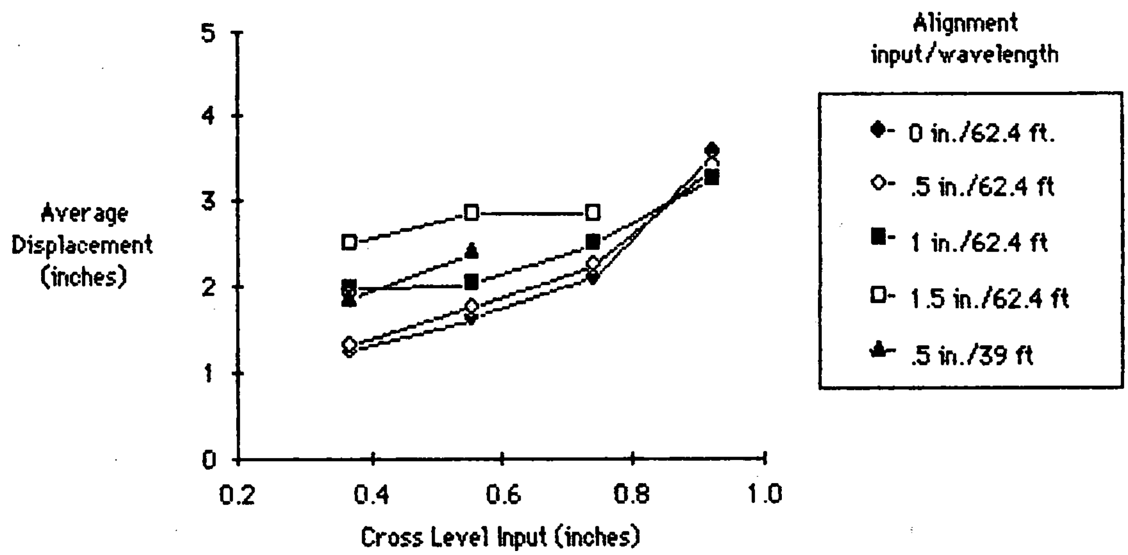


FIGURE 6-23
SINE TEST, WHEEL VERTICAL FORCE, B2 CONFIGURATION

a. Vertical Car Body Displacement



b. Lateral Car Body Displacement

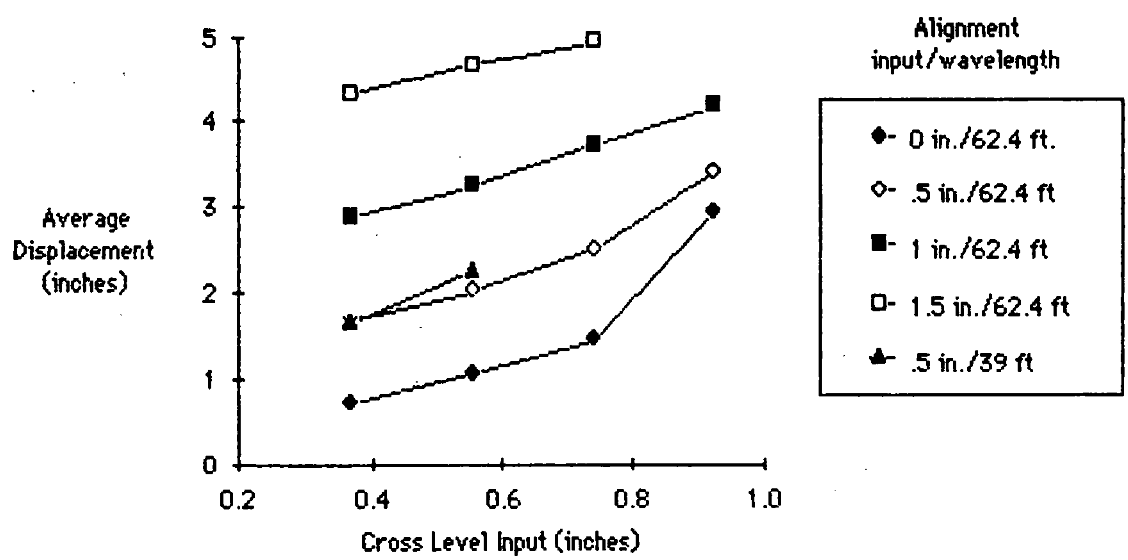
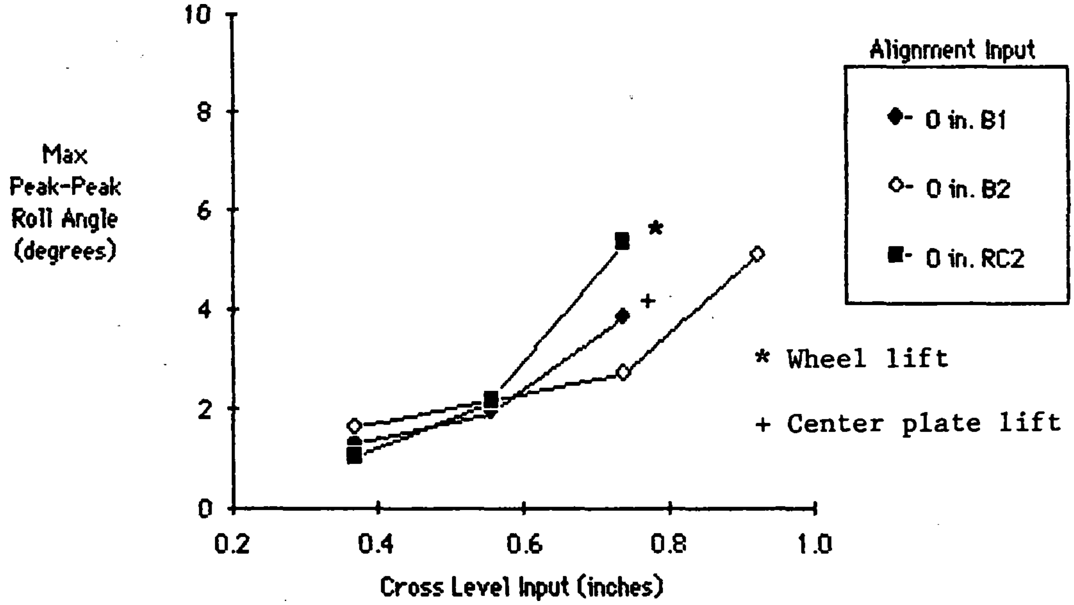


FIGURE 6-24
SINE TEST, CARBODY DISPLACEMENTS, B2 CONFIGURATION

a. No Alignment Input



b. Alignment input equal to 2.0 inches peak-to-peak, 62.4 foot wave

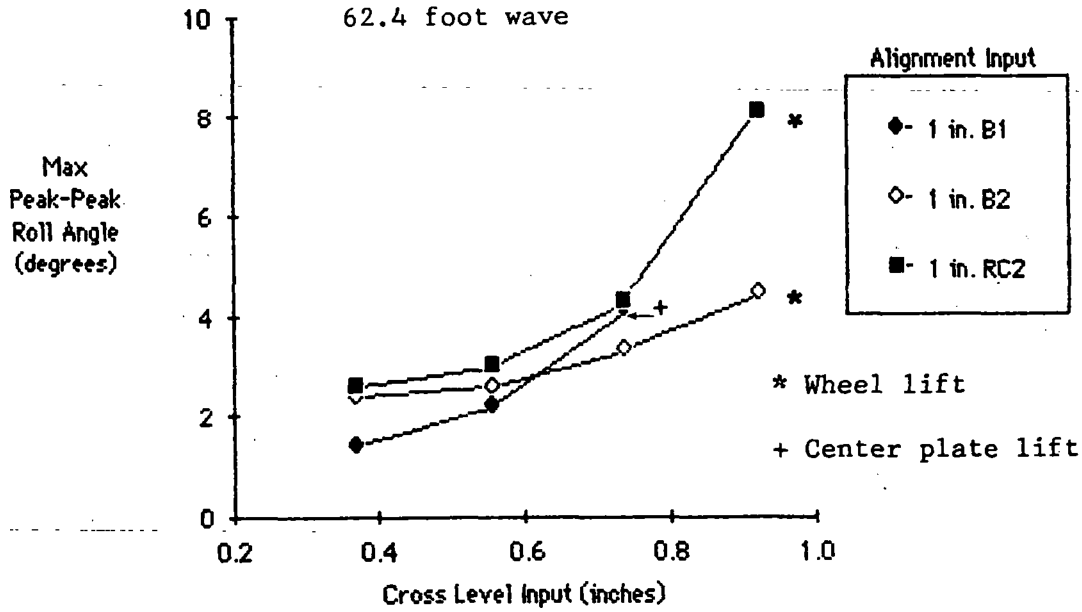
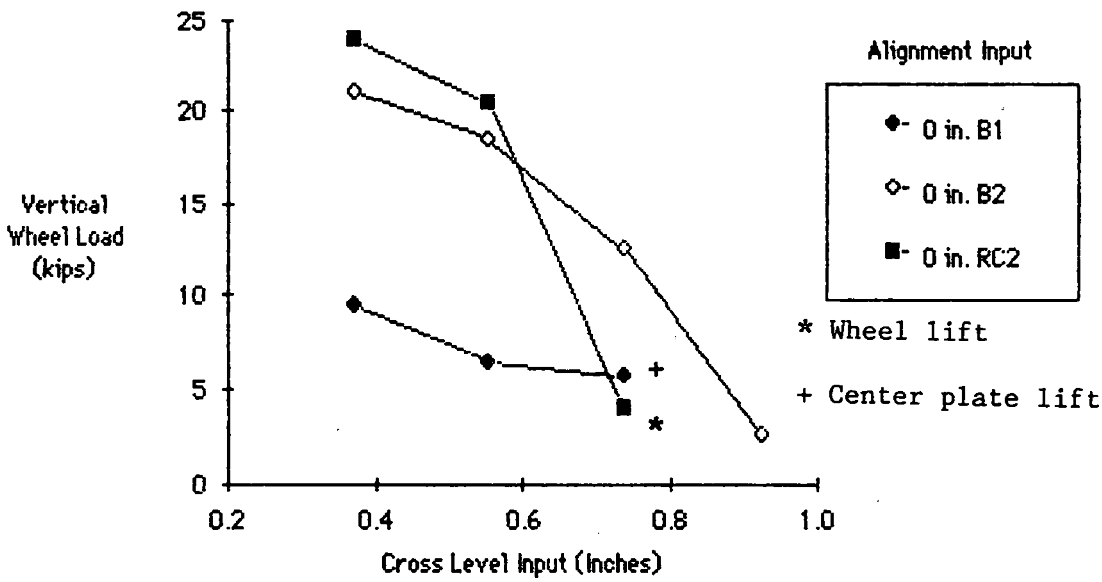


FIGURE 6-25
SINE TEST, COMPARISON OF CARBODY ROLL ANGLES

a. No alignment input



b. Alignment input equal to 2.0 inches peak-to-peak, 62.4 foot wave

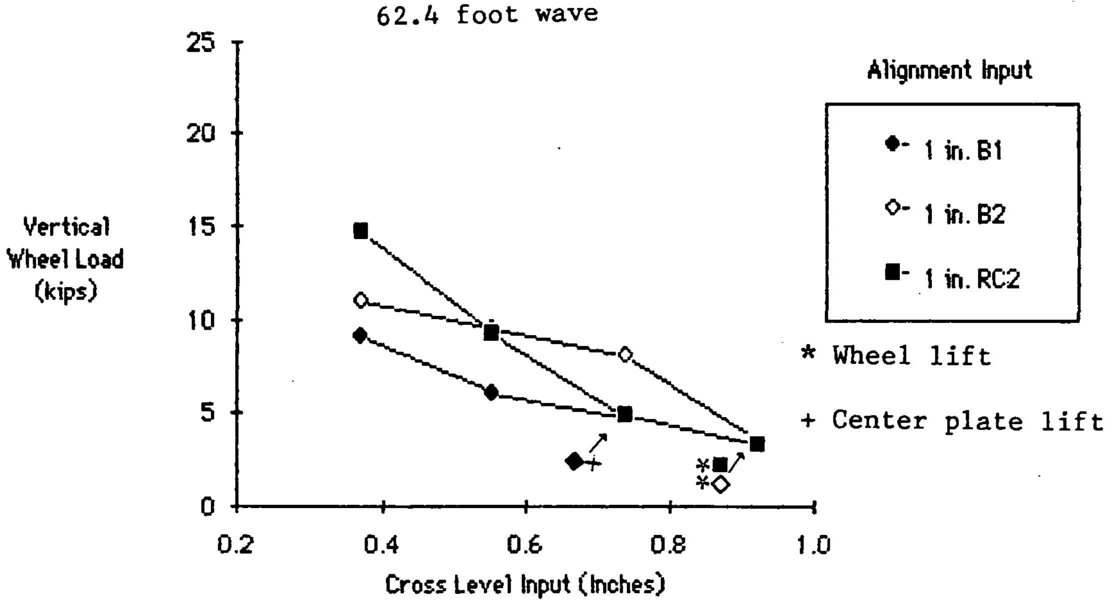


FIGURE 6-26
SINE TEST, COMPARISON OF MINIMUM WHEEL VERTICAL FORCES

and return impact conditions. With empty vehicles, the body bolster lifted out of the center bowl before wheel lift conditions were reached. Consequently, for the empty vehicle, center plate lift was used as the stop-test condition.

In comparing the effects of the various inputs, the condition of 39-foot wave length for both cross level and alignment was the most severe. With 0.5-inch (0-peak) alignment, a stop-test condition was reached at 0.56-inches cross level for both RC2 and B1 configurations and at 0.74-inches cross level for B2.

The tests with combined 39-foot cross level and 39-foot alignment were performed with the following two phasings between cross level and alignment:

- alignment to the right when right rail is low and alignment to the left when left rail is low (as shown in Figure 6-15), and

- alignment to the right when right rail is high and alignment to the left when left rail is high.

The results showed the first condition to result in more severe vehicle response than the second condition. Also the second condition resulted in less severe vehicle response than with no alignment input.

The response to 39-foot staggered rail was not significantly, nor consistently, changed by the inclusion of 62.4-foot alignment variation. For Configuration B1, the stop-test cross level dropped from 0.74 inches to about 0.65 inches as the alignment was varied from 0 to 1.5 inches. For Configuration B2, the stop-test cross level dropped from about 1.0 inches to 0.74 inches. For Configuration RC2, the stop-test cross level was 0.74 inches with no alignment variation and increased to 0.925 inches with alignment variation.

In comparing the three configurations, the empty vehicle, B1, had the poorest performance. The performance with the Barber trucks, B2, was slightly better than with the Ride Control Trucks, RC2.

6.5 Safety Margin Tests, Base Vehicle No. 2

The primary purpose of the Safety Margin Testing was to measure the safety performance of the Base Vehicles with the three conditions of:

- harmonic roll response to staggered rail,
- bounce response to profile perturbations, and
- yaw response representative of body hunting conditions.

The tests were directed at finding the track geometry and speed conditions that resulted in unsafe car response conditions. A secondary purpose of these tests was to evaluate the relative effects of several profile shapes used to simulate staggered rail. Wheel lift (or actually the point at which vertical wheel load reaches zero) and center plate lift were criteria used to define the stop-test condition.

6.5.1 Safety Margin Test Matrix

The types of tests performed and the configurations on which they were performed are listed in the test matrix shown in Table 6-7. The notation for the test configurations is defined in Table 6-2. In each case the test objective was to find the speed (or frequency) and track geometry condition that resulted in marginal safety conditions. Searches for critical conditions were made with frequency sweeps and constant speed (frequency) runs over 10 rail lengths of prescribed track geometry variations. The frequency sweeps were generally run first to locate the critical speed. The constant speed runs were then made to probe for the critical amplitude of track geometry. The results are presented in plots of vehicle response and wheel-rail loads against speed, or frequency, and response at critical speed against track geometry amplitude.

6.5.2 Discussion of Staggered Rail Profile Simulation

The AAR test specification, "Special Devices to Control Stability of Freight Cars," calls for the shimming of 20 consecutive joints (10 on each rail) to 3/4 inches higher than the general elevation with 39-foot rail length representation. The shim dimensions and shim arrangement are shown in Figure 6-27. The length of each perturbation is about 16.25 feet with successive perturbations placed every 39 feet. These perturbations are a simulation of low joints characteristic of bolted rail in revenue service. The simulation differs from the actual profiles in two respects; (1) the simulation provides a perturbation in the "upward" direction whereas the service perturbation is downward and (2) the simulation provides a smooth variation whereas the service condition is usually a depressed bolted joint that forms a cusp, the sharpness of which may be further emphasized by a gap in the rail.

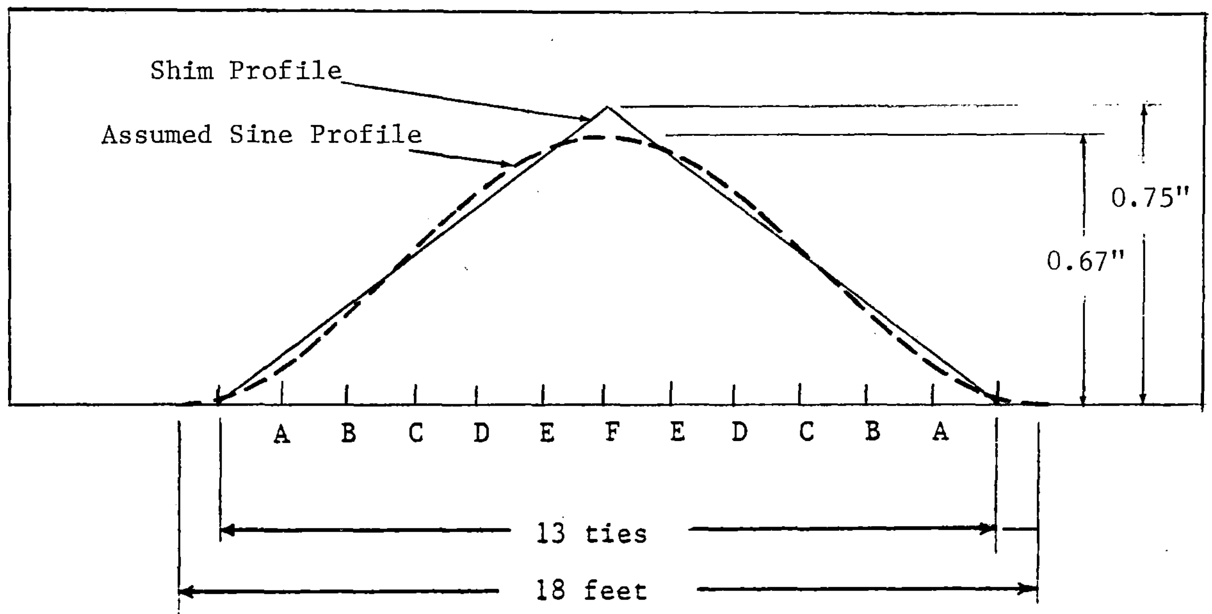
TABLE 6-7
VTU TEST MATRIX

TEST	CONFIGURATIONS TESTED								
	BASE VEHICLE NO. 2								
	RC2	B1	B2	B3	B4	B5	B6	B6.1	NO. 1
Shimmed Rail	x		x						
Inverted Shimmed Rail	x		x						
Rectified Sine -tangent track	x	x	x	x	x	x	x	x	x
-with lateral	x	x	x	x	x	x	x	x	
-with super elevation	x	x	x	x	x	x	x	x	
Bounce	x	x	x	x	x	x			
Yaw	x		x	x	x	x			

Base Vehicle No. 2 Configuration Notation

RC2 - loaded car Ride Control Trucks
B1 - empty car
B2 - loaded car
B3 - 27.6% off load (lowered c.g.)
B4 - loaded car, single side spring snubbers
B5 - same as B4 with auxiliary hydraulic snubbers
B6 - B2 with 5/16 in. side bearing clearance
B6.1 - B2 with 1/4 inc. side bearing clearance
B7 - loaded car, constant contact side bearing

- All B configurations - Barber S-2-C trucks with double side spring snubbers unless noted otherwise.



Shim Specification

Tie	A	B	C	D	E	F
Shim Height (inches)	1/8	1/4	3/8	1/2	5/8	3/4

Assumptions

1. Because the rail has stiffness and because the road bed deforms, it is assumed that the actual profile is more accurately defined by a sine wave then the triangular shimming specification.
2. The actual max height of rail, with 0.75 inch shim, is assumed to be 0.67 inches.
3. The length of the haversine is assumed to be 18 feet.

FIGURE 6-27
HAVERSINE REPRESENTATION OF SHIMMED RAIL

Haversine Simulation of Shimmed Track on the VTU

Since rail stiffness prevents the rail from conforming to the triangular shim pattern, the rail will assume a curved shape rather than an angular bend. Further, the road bed is not rigid but will deform, particularly at the high point of the perturbation. It was assumed that the triangular shim specification actually results in a smooth shape that is probably close to a haversine. The haversine, used to replace the triangular shape, was obtained by making the haversine longer than the triangle (18 feet instead of 16.25) and the height less (0.67 inches instead of 0.75). The resulting haversine is superimposed on the triangular shim specification in Figure 6-27 to show the closeness of the two shapes.

The total simulation of the perturbed track consisted of 10 haversine shapes separated by 21-foot flat sections, each 18-foot haversine and 21-foot flat section being the equivalent of one 39-foot rail. In the staggered rail simulation, the left and right rails are located so that the joints of one side are opposite the mid-spans of the other side.

The three wave forms used are shown in Figure 6-28; (a) is the haversine simulation of the shimmed track, (b) is the haversine simulation inverted, and (c) is the rectified sine simulation. The (d) curve is the alignment variation used in the combined cross level--alignment test and generated from the rectified sines used in the profile variations. These four wave forms are referred to in this report as follows:

- shimmed rail,
- inverted shimmed rail,
- rectified sine, and
- rectified sine with lateral.

6.5.3 Safety Margin Test Results - 39-Foot Staggered Rail

The safety margin test results are presented and discussed in four perspectives. First, comparison is made between VTU and track tests of staggered rail. Second, comparison is made between the rail profile simulations. Third, performance of the 100-ton covered hopper car is presented with comparison between the several configurations tested. Fourth, the effects of side bearing clearance are shown.

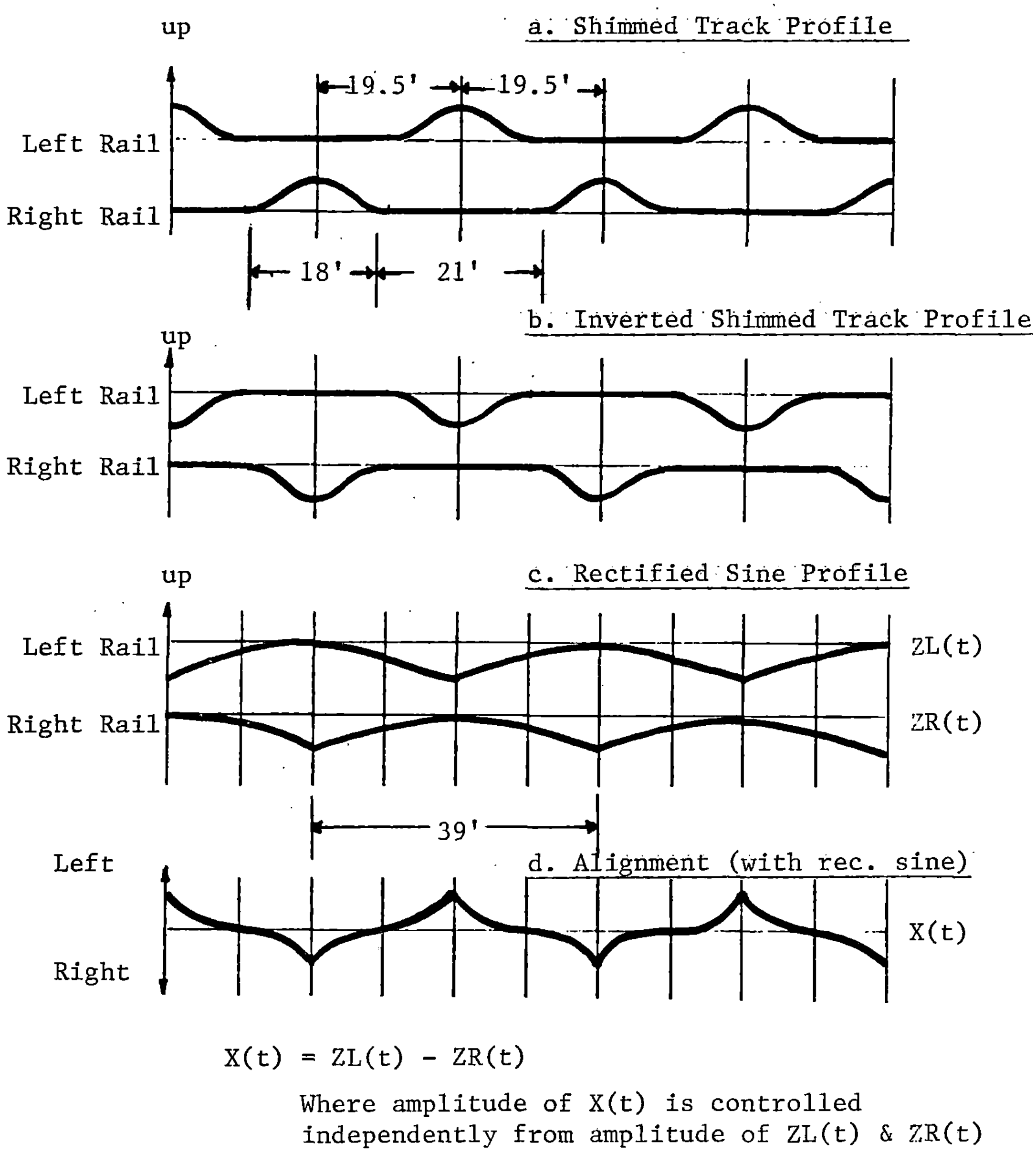


FIGURE 6-28
WAVE FORMS TO BE USED IN SAFETY MARGIN STAGGERED RAIL TESTS

VTU to Track Test Comparison

The track tests [2] were performed with Base Vehicle No. 2 in the RC2 configuration. The staggered rail test consisted of running the test consist over track that had been shimmed for 10 rail lengths of 39 feet with the 0.75-inch triangular shimmed patterns. Comparison is made here to four staggered rail profiles used in the VTU testing discussed in paragraph 6.5.2.

Figure 6-29 compares the VTU shimmed rail test with track test results for the RC2 configuration. Comparisons of results show that:

- critical speeds are within 0.5 miles per hour,
- track test carbody roll angles are greater than VTU test by, approximately, a factor of two at critial speed and a factor of six above critical speeds, and
- track test wheel lift occurs with lower cross levels; the track test has strong wheel lift at 0.75-inches cross level; the VTU test has incipient wheel lift at 0.85 inches.

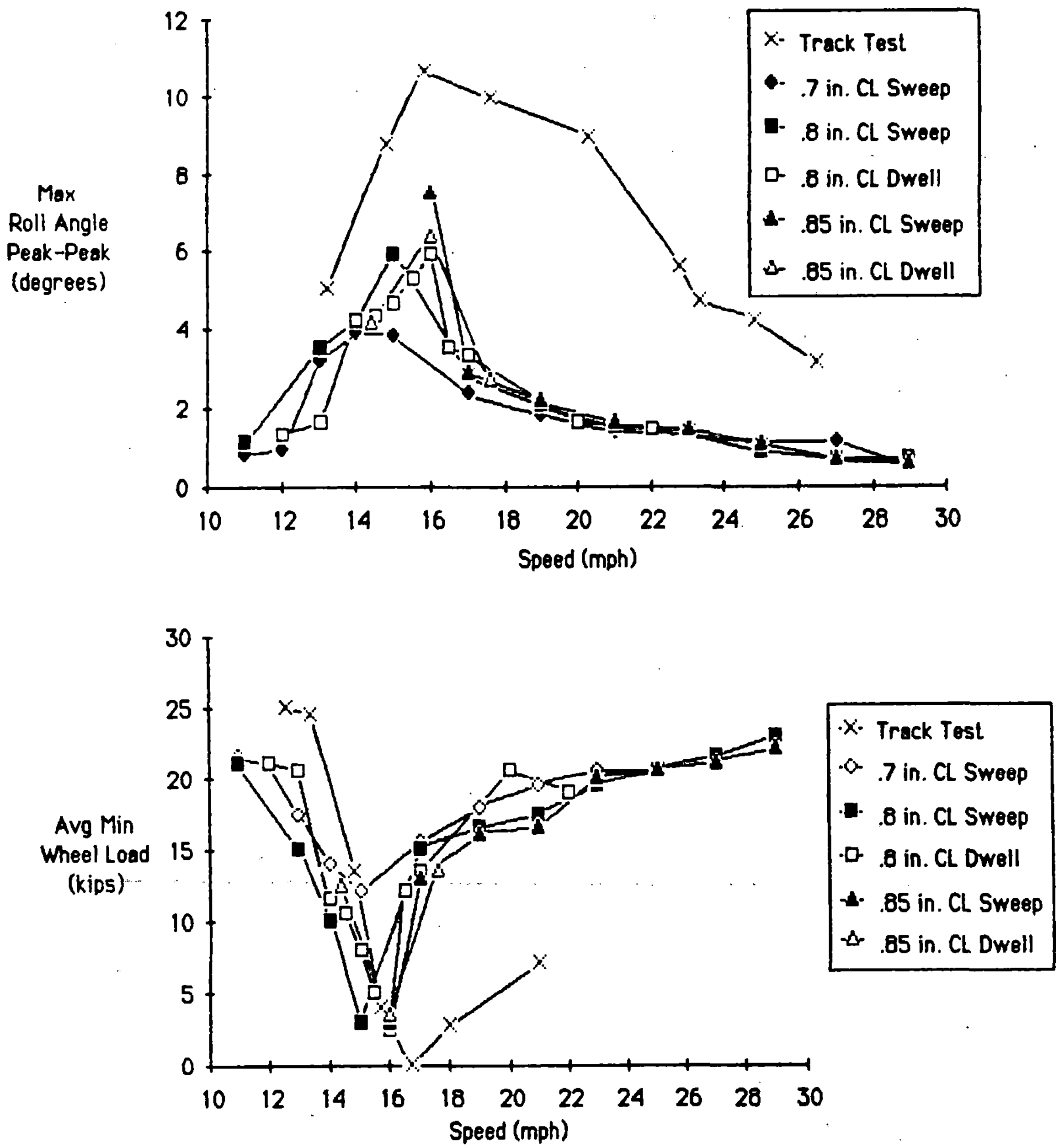
Overall, the VTU shimmed rail test results are close to but somewhat less severe than the track test results. The carbody roll angles are significantly larger in the track test.

Figure 6-30 compares track test with VTU test results including all four staggered rail profiles used in the VTU tests. The results comparison shows that:

- critical speeds are within 0.5 miles per hour,
- carbody roll angles are greater in the track test, and
- wheel lift cross level is less with rectified sine profiles and greater with shimmed rail profiles in comparison to the track test at 0.75-inches cross level.

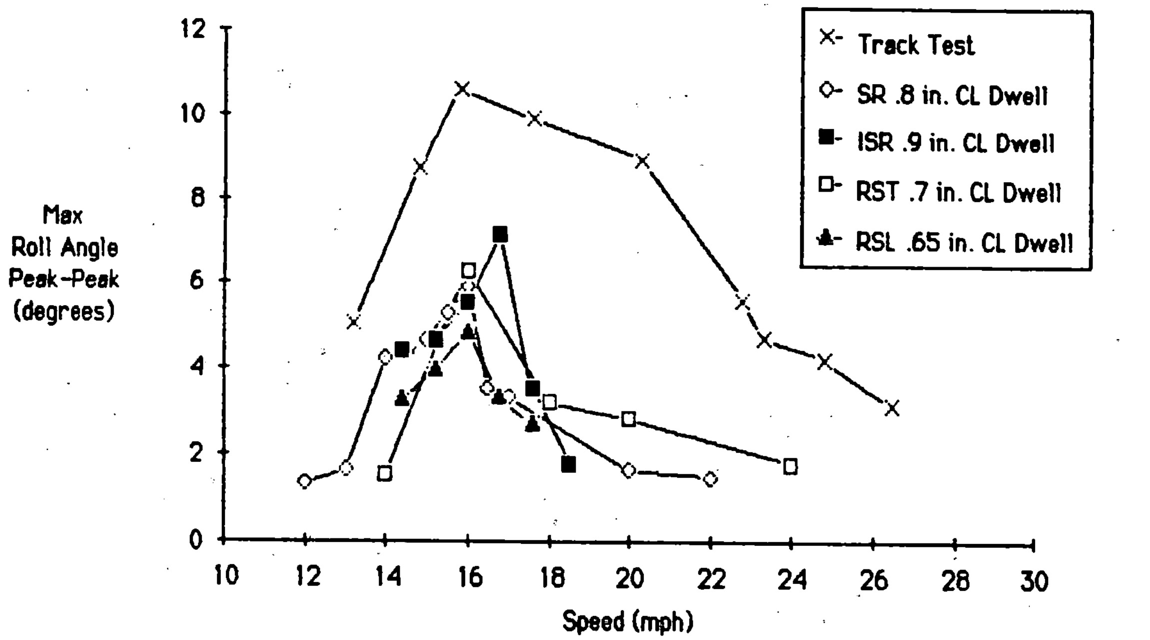
Figure 6-31 is a comparison of track and VTU test results where rectified sine profile cross level in combination with lateral input motion are used in the VTU test. The results show that:

- critical speeds are equal,
- carbody roll angles are greater in the track test, and



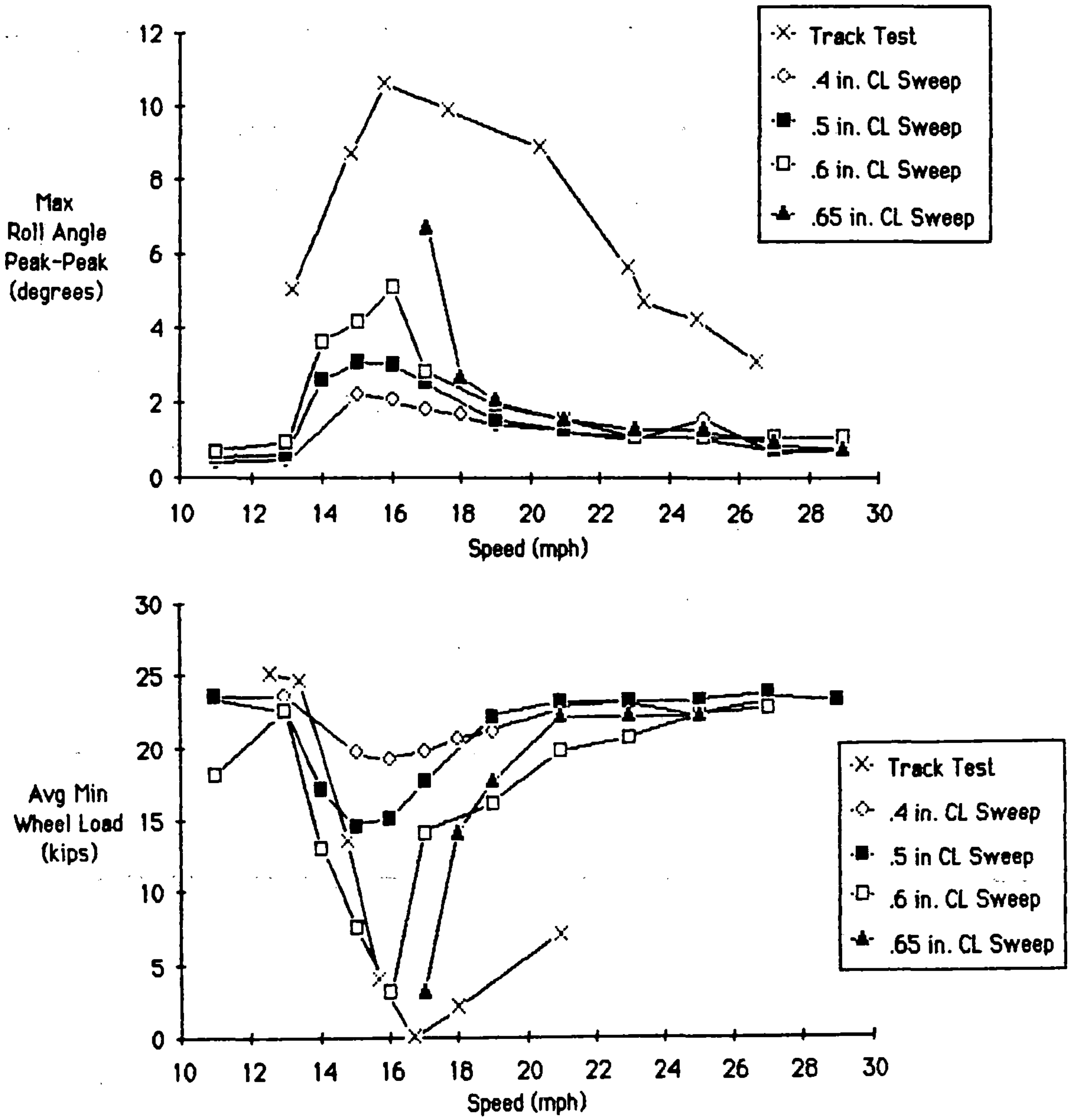
Track test data from Reference [2]
CL = Cross Level
Sweep = speed slow-down, VTU shimmed track test
Dwell = constant speed VTU shimmed track test

FIGURE 6-29
RC2 RESPONSE TO SHIMMED TRACK STAGGERED RAIL
COMPARISON OF TRACK AND VTU TESTS



Track test data from Reference [2]
CL = Cross Level
SR = Shimmed rail, constant speed VTU test
ISR = Inverted shimmed rail, constant speed VTU test
RST = Rectified sine tangent track, constant speed VTU test
RSL = Rectified sine with lateral input, constant speed VTU test

FIGURE 6-30
RC2 RESPONSE TO STAGGERED RAIL, COMPARISON OF TRACK TEST TO SEVERAL PROFILES USED IN VTU



Track test data from Reference [2]
CL = Cross Level
Sweep = Speed slow-down VTU test, rectified sine profile with lateral input

FIGURE 6-31
RC2 RESPONSE TO STAGGERED RAIL, COMPARISON OF TRACK TEST TO VTU RECTIFIED SINE WITH LATERAL TEST

- wheel lift occurs at 0.65 inches cross level on the VTU compared to the 0.75-inch track.

Figure 6-32 plots carbody roll angles and minimum wheel loads as functions of input cross level. Configurations RC2 and B2 are shown for shimmed rail, inverted shimmed rail and rectified sine profiles. The results show that:

- the VTU rectified sine test and track test results compare very well if it is considered that the VTU test point was zero wheel load while the track test was actual wheel lift. Extrapolation in Figure 6-32 of the VTU roll angle curve to 0.75-inch cross level results in excellent correspondence with the track test. Similarly, if the track test were to run with 0.70-inch shimmed track there would probably also be incipient wheel lift,

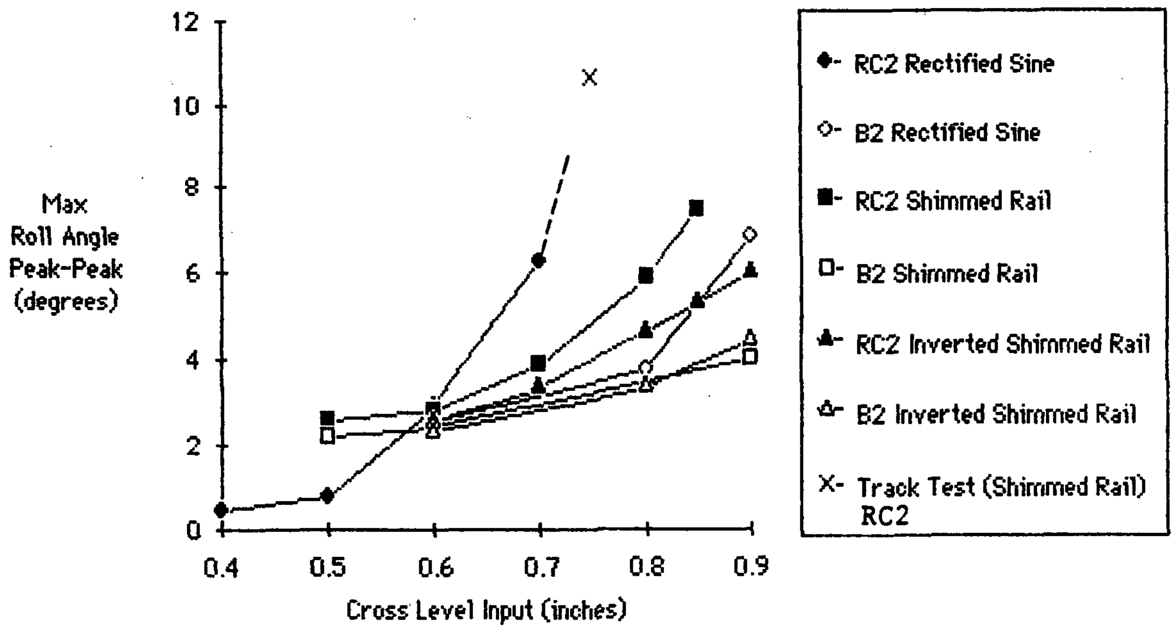
- the VTU shimmed and inverted shimmed rail simulations consistently showed less response than the rectified sine and the track test, and

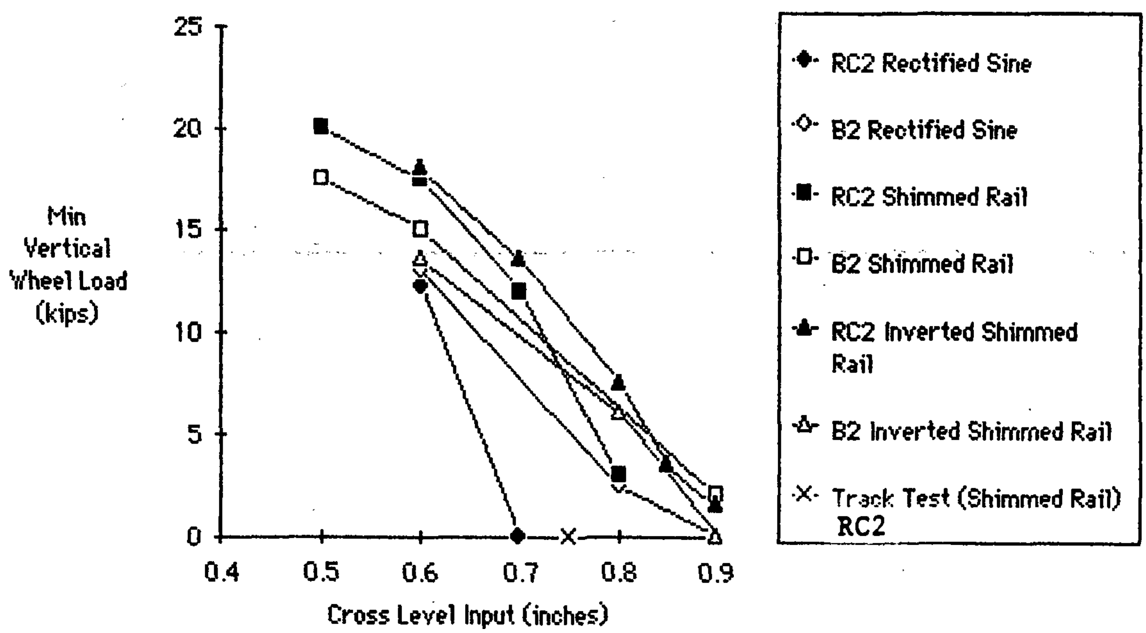
- the B2 configuration has better performance than the RC2 in that carbody roll angles are smaller and in that wheel lift occurs with larger input cross levels.

The conclusion is that the rectified sine and the shimmed rail profiles show very good comparison between VTU and Track test results with respect to conditions for the onset of wheel lift.

A second conclusion is that the carbody roll angle comparison is not as good, with track test angles being considerably larger than VTU. It is postulated that this roll angle difference may be caused by the difference in draft conditions. On the track, the test consist is in a constant speed condition, on level track, with essentially zero draft. On the VTU there is a constant draft force of about 45,000 pounds, which probably acts to restrain rolling motions of the vehicle. Verification of this theory should be made on some subsequent VTU testing by testing with and without draft forces.

The final conclusion is that, if the track shims were a closer representation of actual bolted rail profile, the track test responses would probably become more severe.





RC2 Rectified sine and Track test constant speed
All other speed slow-down tests

FIGURE 6-32
RC2 AND B2 RESPONSE VS. CROSS LEVEL
COMPARING TRACK TEST AND SEVERAL VTU PROFILES

Staggered Rail Profile Comparison

Figures 6-33, 6-34, and 6-35 compare VTU test results with the shimmed rail, inverted shimmed rail, and rectified sine profile of staggered rail. The results show the rectified sine to result in larger carbody roll angles and reach wheel lift with slightly smaller cross level input. The shimmed and inverted shim result in about the same response conditions with the inverted shim being slightly more severe.

The conclusion is that the rectified sine profile shape is preferred in the VTU simulation of staggered rail because its results are closer to shimmed track test results than the other two profiles tried and since it is also a more severe test.

Comparison of Vehicle Performance, Staggered Rail

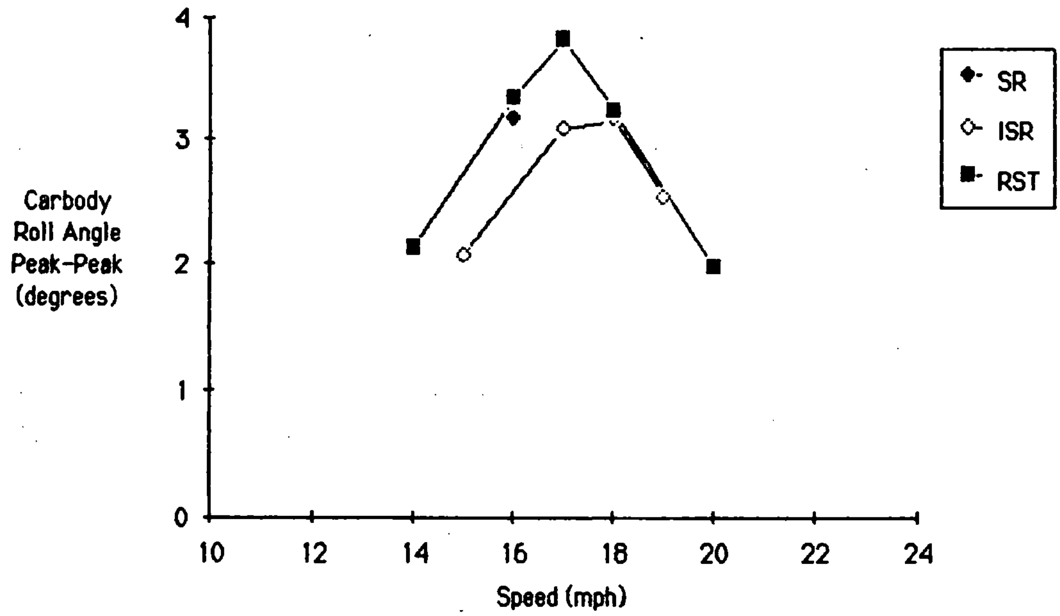
Comparisons of vehicle performance over staggered rail are presented in Figures 6-36 through 6-43. Figures 6-44 and 6-45 show the effects of side bearing clearance. Although the multiple plots in each figure require some effort in reading, they are presented to provide direct comparisons.

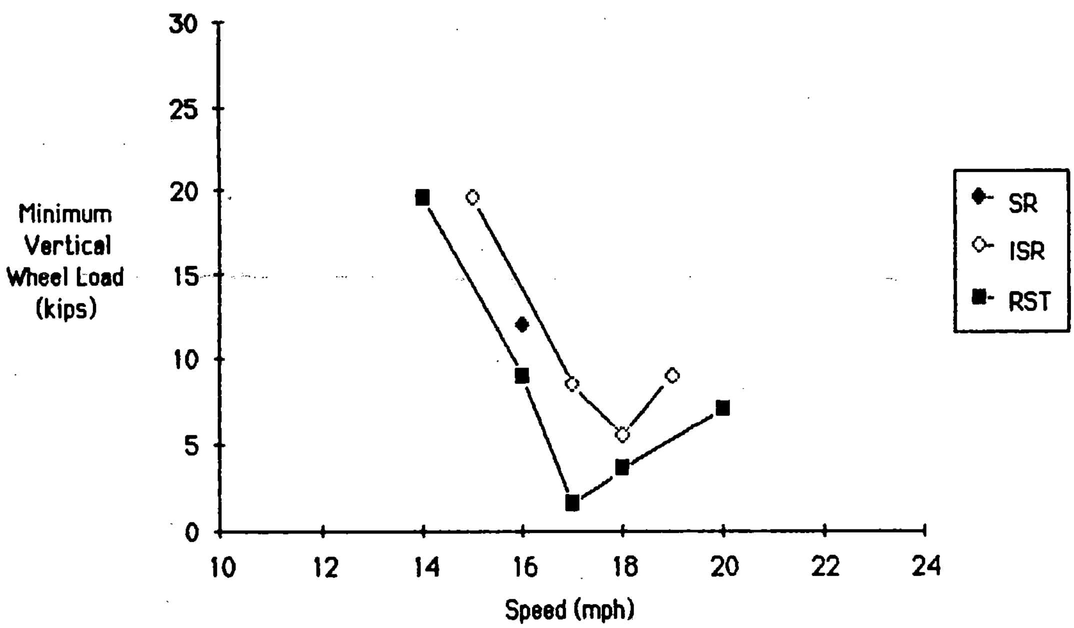
The effects of gross weight are seen in comparison of vehicle B1, empty, B2, at 98% of load capacity and B3 at 72% of load capacity. The following results are noted:

- critical speeds are 16 mph at 98%, 20 mph at 72%, and 25 mph empty,
- B1 is the poorest performer because of center plate lift at 0.60 inches cross level, and
- B3 (72%) does not perform as well as B2 (98%) because it has wheel lift at about 0.72 inches cross level compared to B2's 0.90 inches.

Five truck configuration changes were compared, not including side bearing clearance, with the vehicle loaded at 98% of weight capacity:

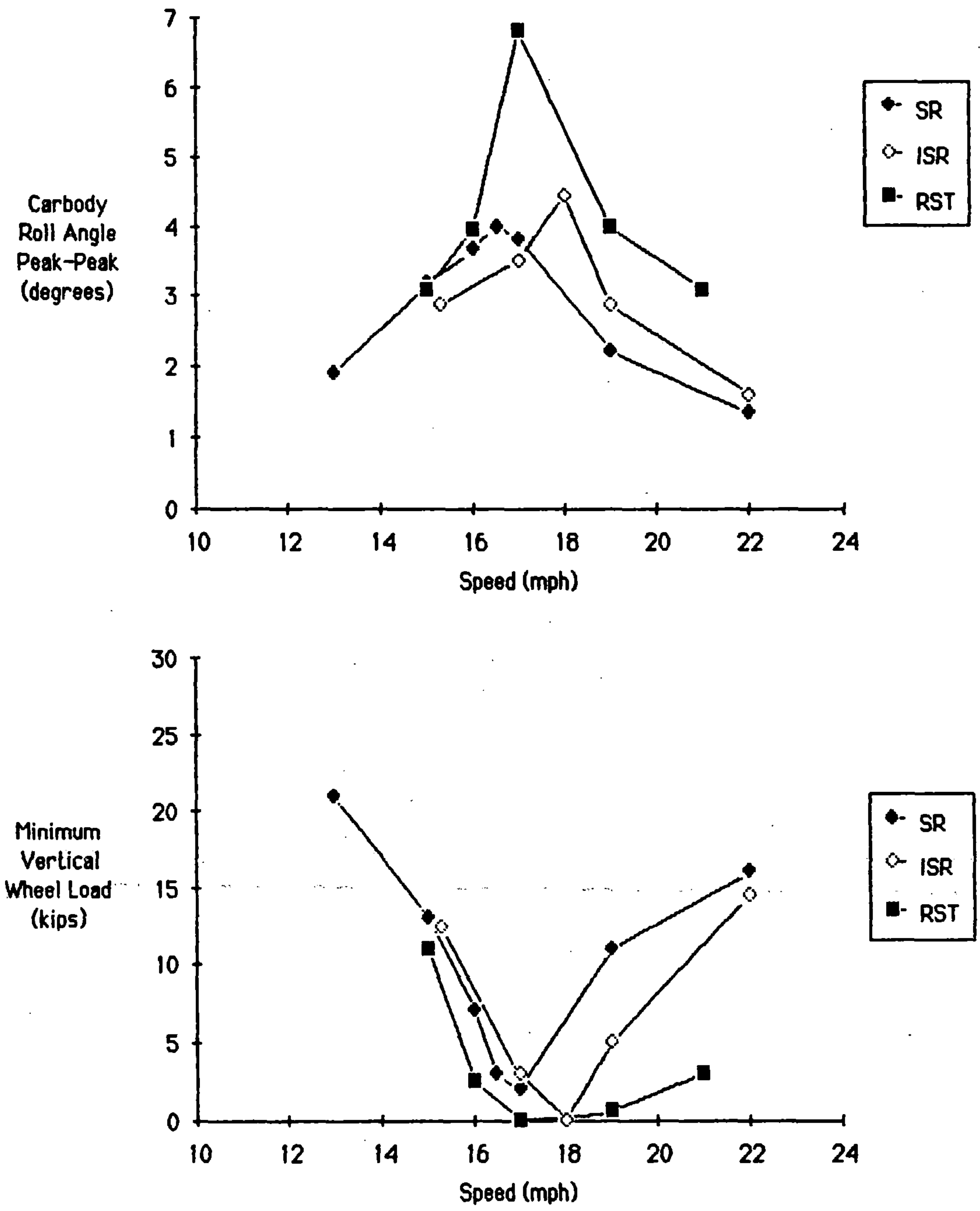
- RC2 - with Ride Control truck,
- B2 - the Barber S-2-C in its normal configuration,
- B4 - the S-2-C with single side springs,





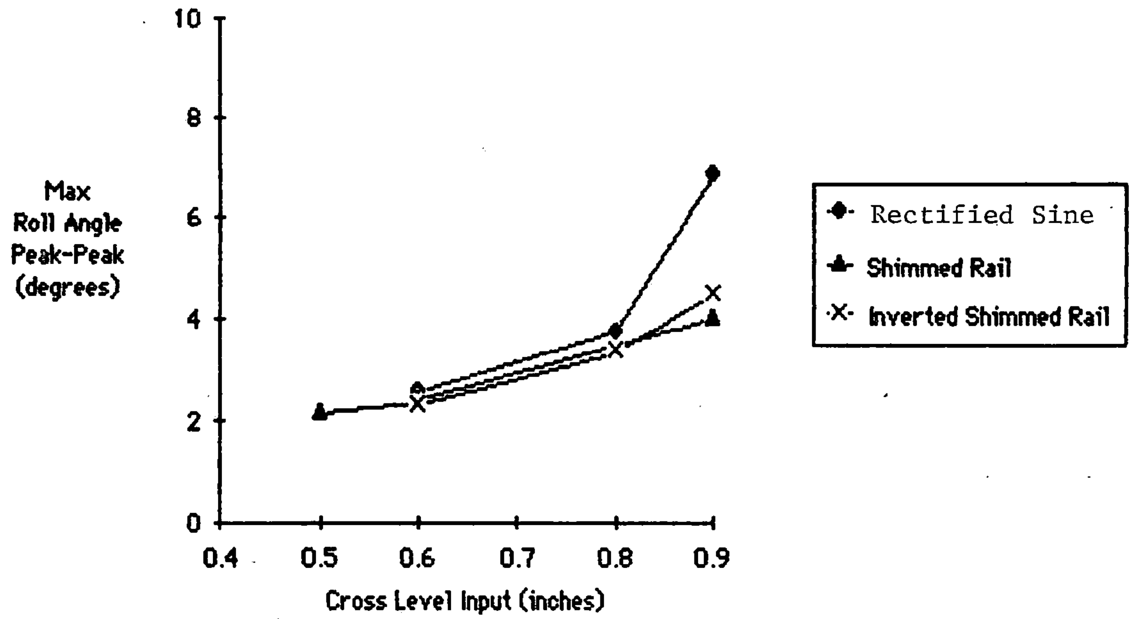
SR = Shimmed rail
ISR = Inverted shimmed rail
RST = Rectified sine tangent track

FIGURE 6-33
B2 RESPONSE TO STAGGERED RAIL, COMPARISON OF THREE PROFILE SHAPES AT 0.80-INCH CROSS LEVEL



SR = Shimmed rail
ISR = Inverted shimmed rail
RST = Rectified sine tangent track

FIGURE 6-34
B2 RESPONSE TO STAGGERED RAIL, COMPARISON OF THREE PROFILE SHAPES AT 0.90-INCH CROSS LEVEL



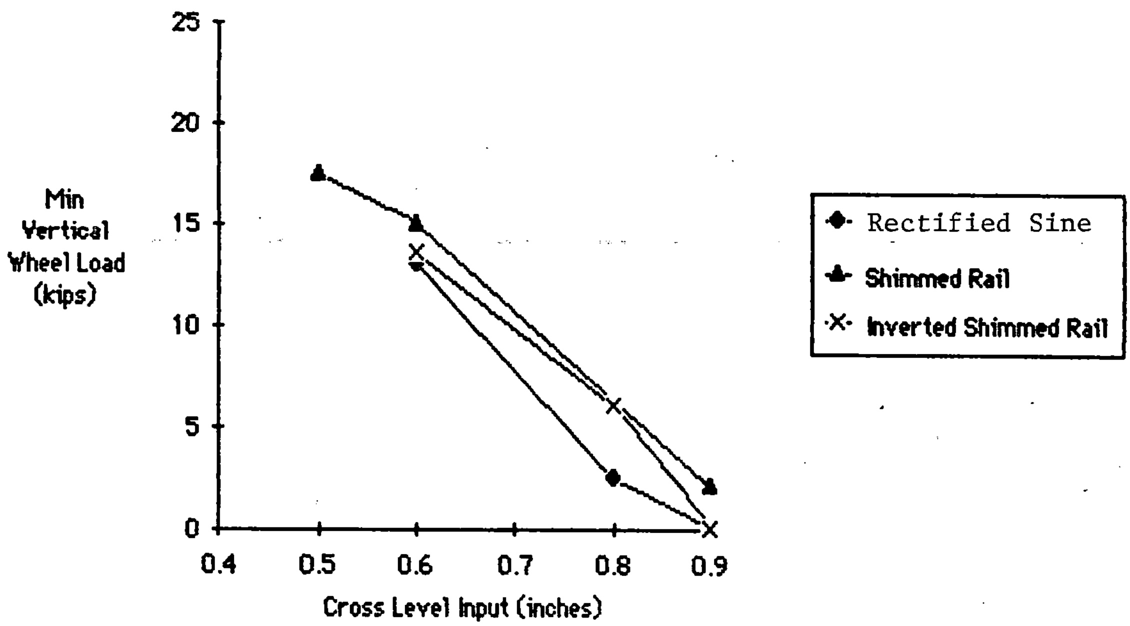


FIGURE 6-35
B2 RESPONSE VS. CROSS LEVEL, COMPARISON OF THREE, STAGGERED RAIL PROFILE SHAPES

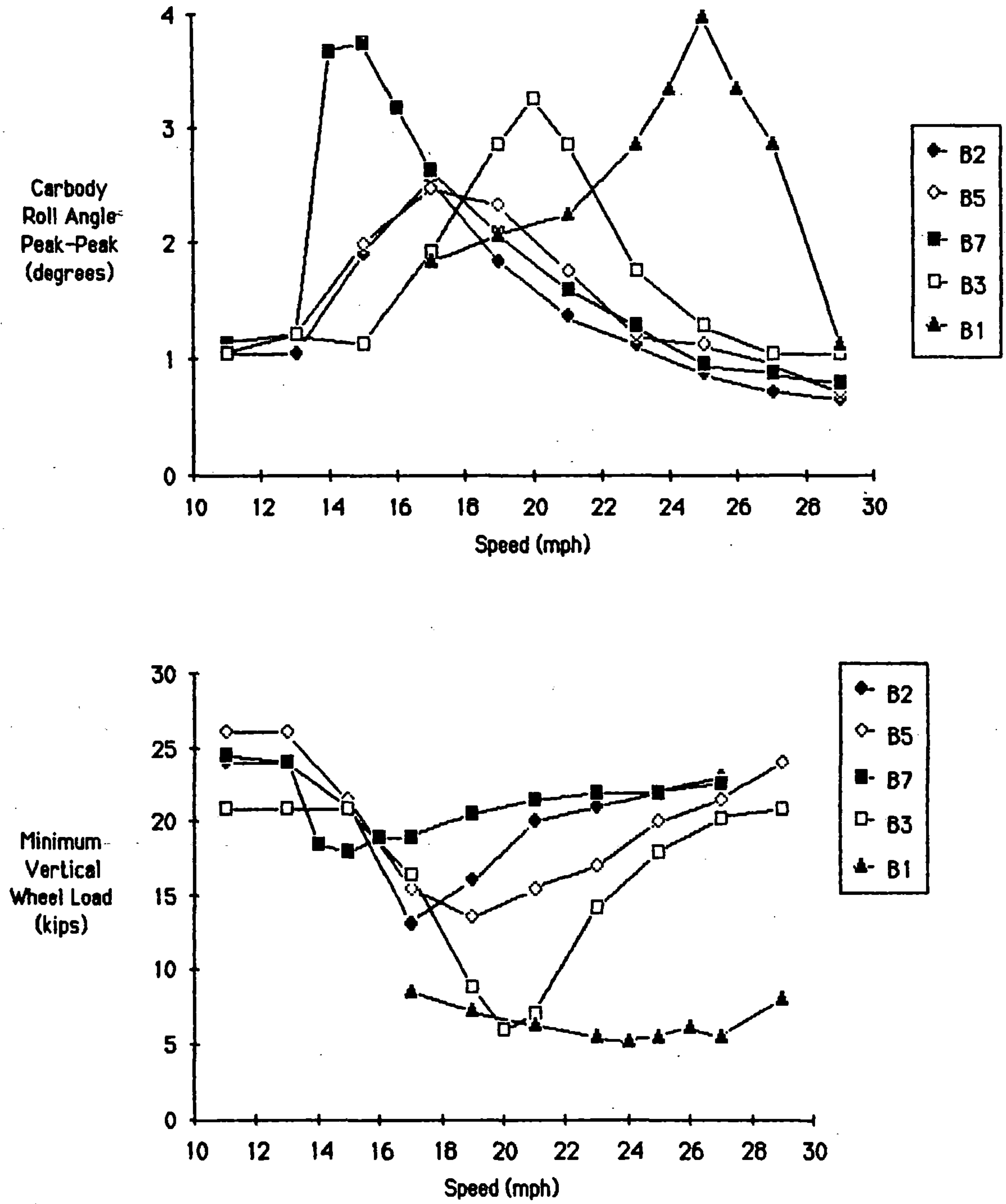


FIGURE 6-36
COMPARISON OF BARBER TRUCK CONFIGURATIONS
0.60-INCH RECTIFIED SINE STAGGERED RAIL

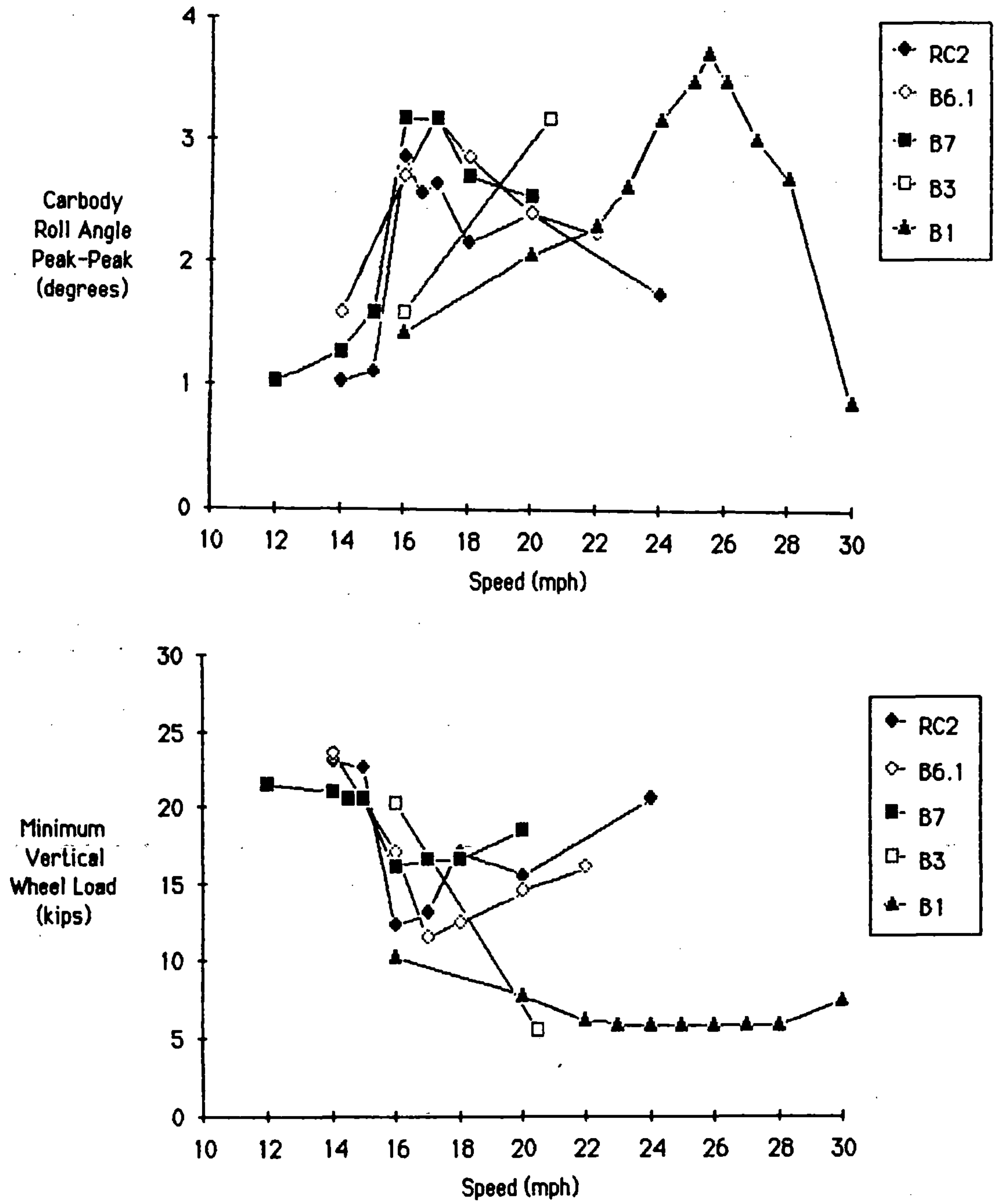
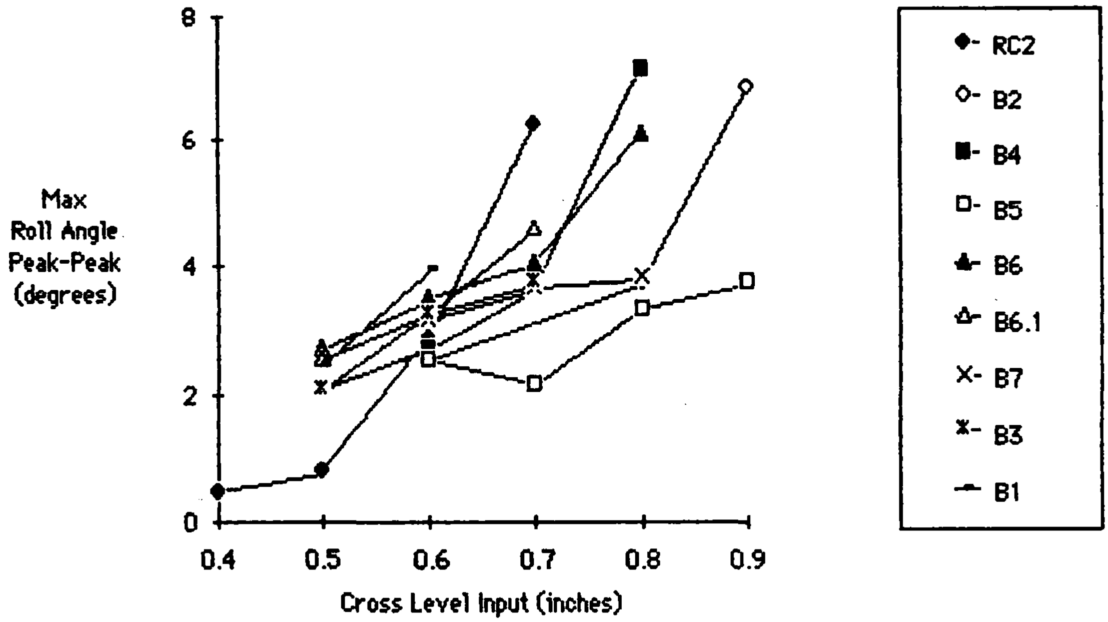


FIGURE 6-37
COMPARISON OF RESPONSE TO STAGGERED RAIL, 0.60-INCH RECTIFIED SINE, CONSTANT SPEED



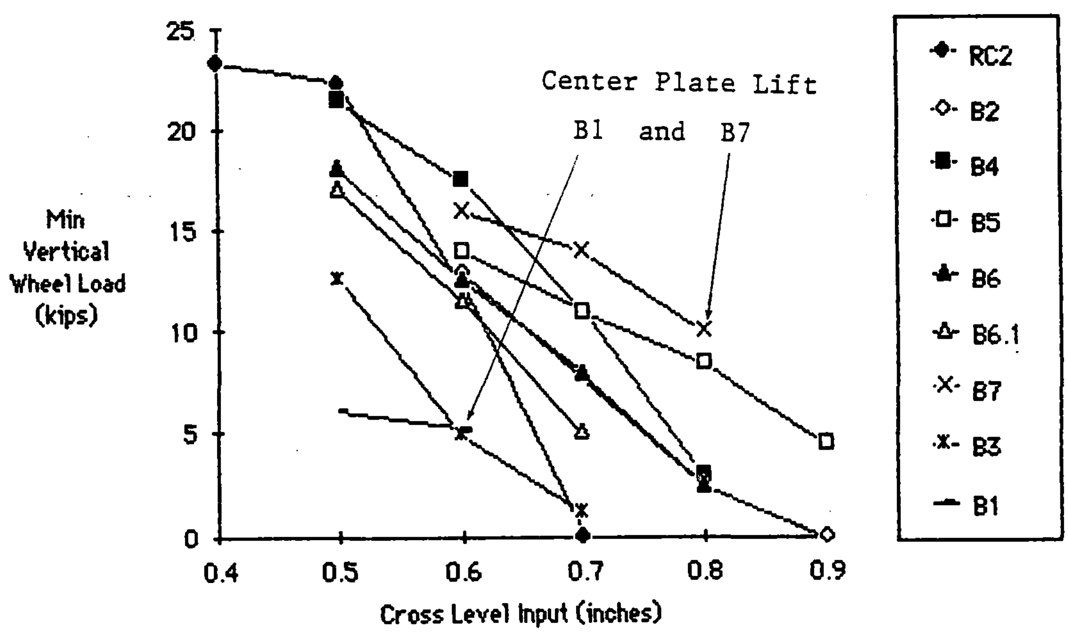
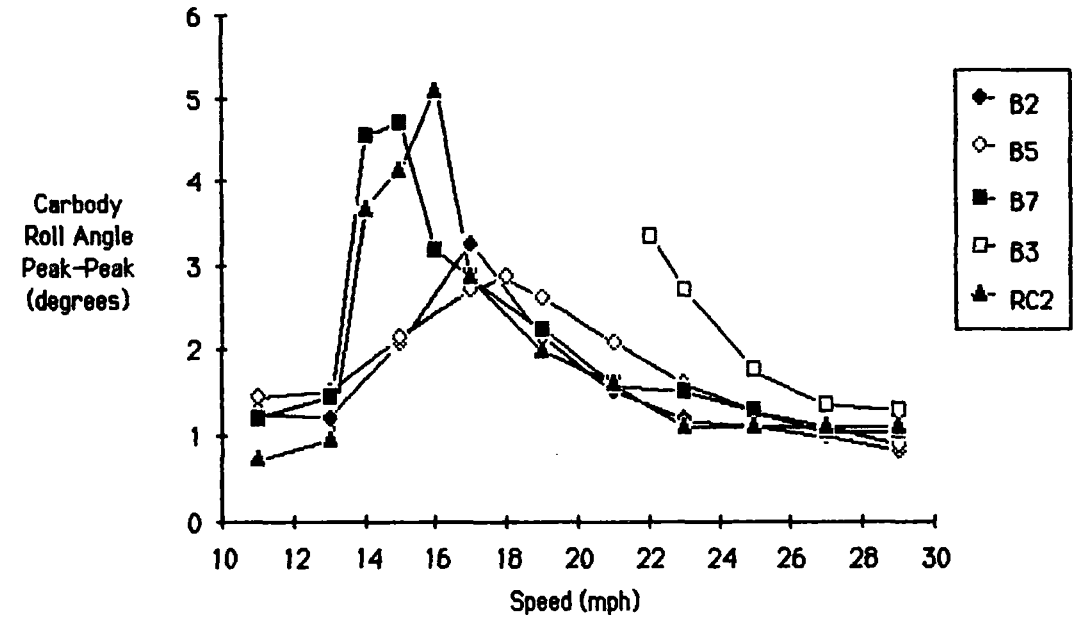


FIGURE 6-38
RESPONSE VS. CROSS LEVEL, ALL CONFIGURATIONS, RECTIFIED SINE INPUT



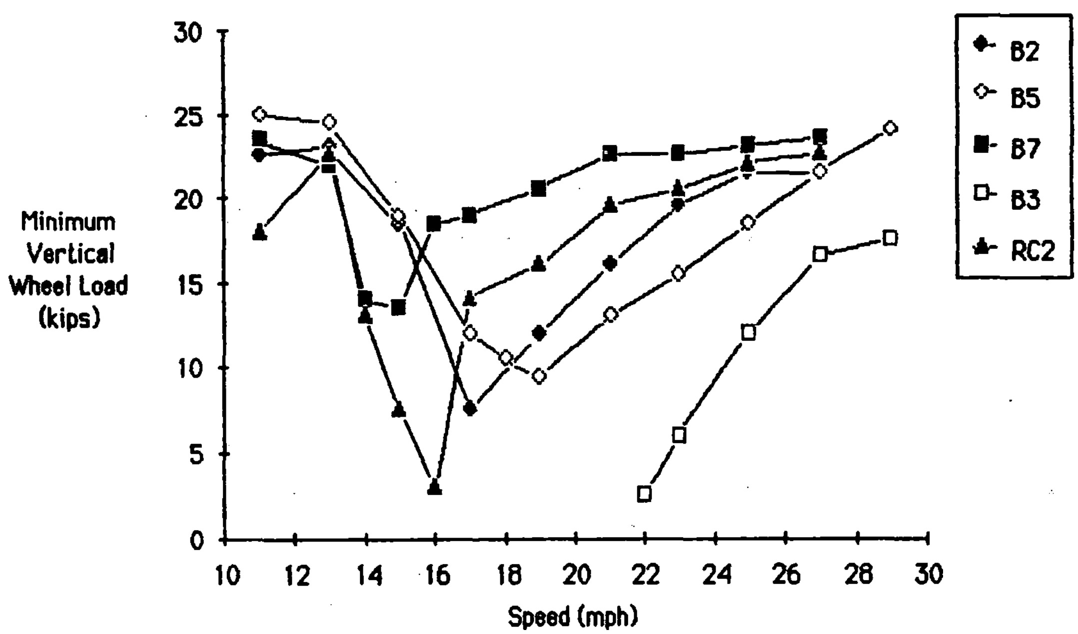
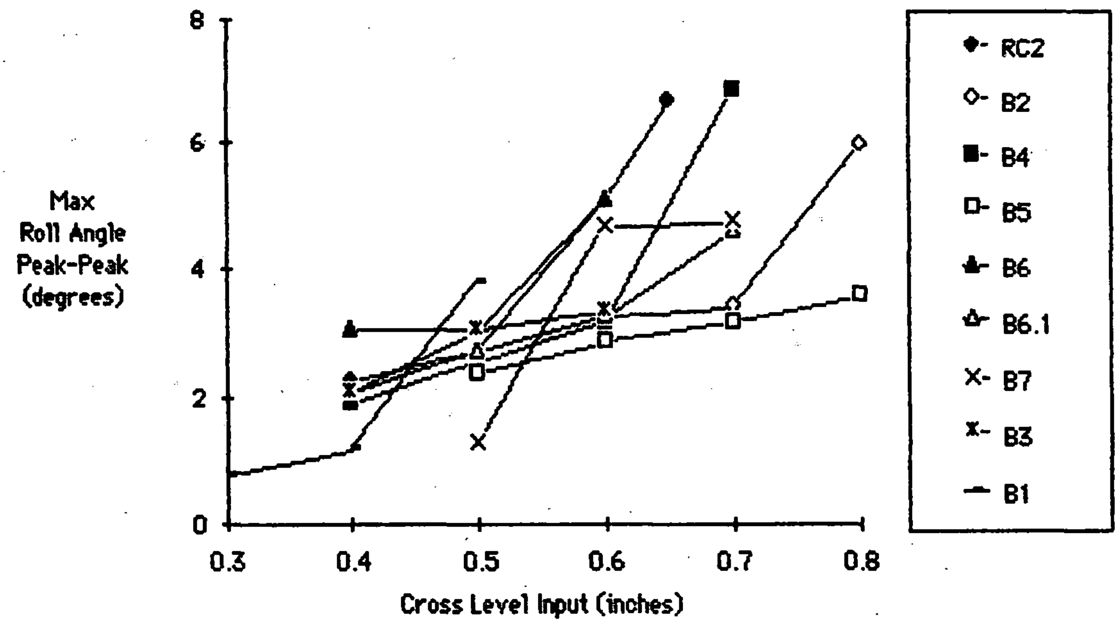


FIGURE 6-39
CONFIGURATION COMPARISONS, 0.60-INCH CROSS LEVEL RECTIFIED SINE WITH LATERAL



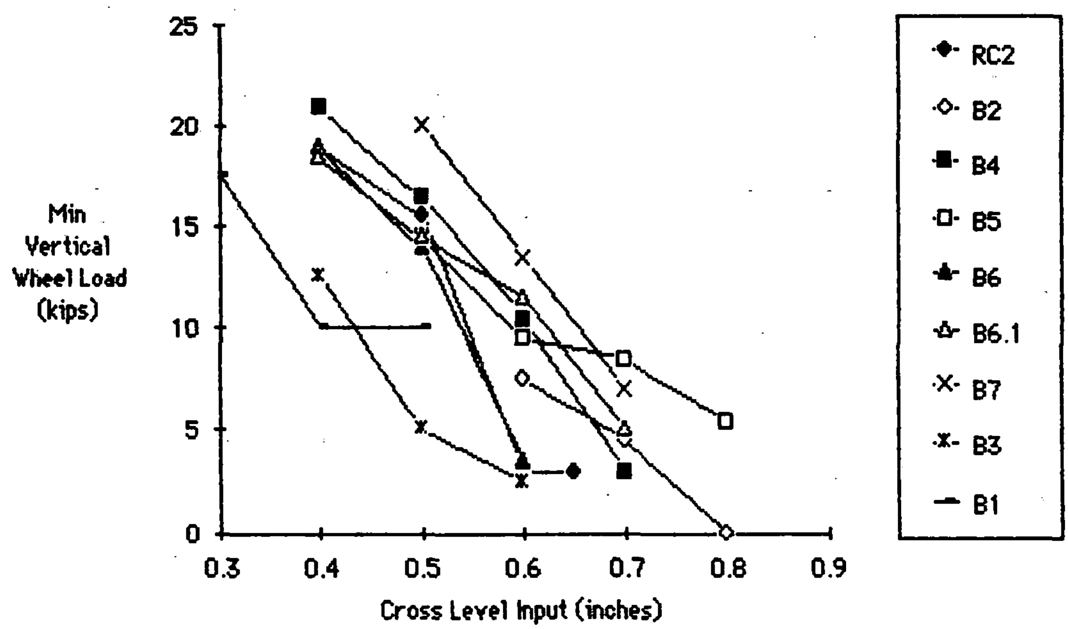
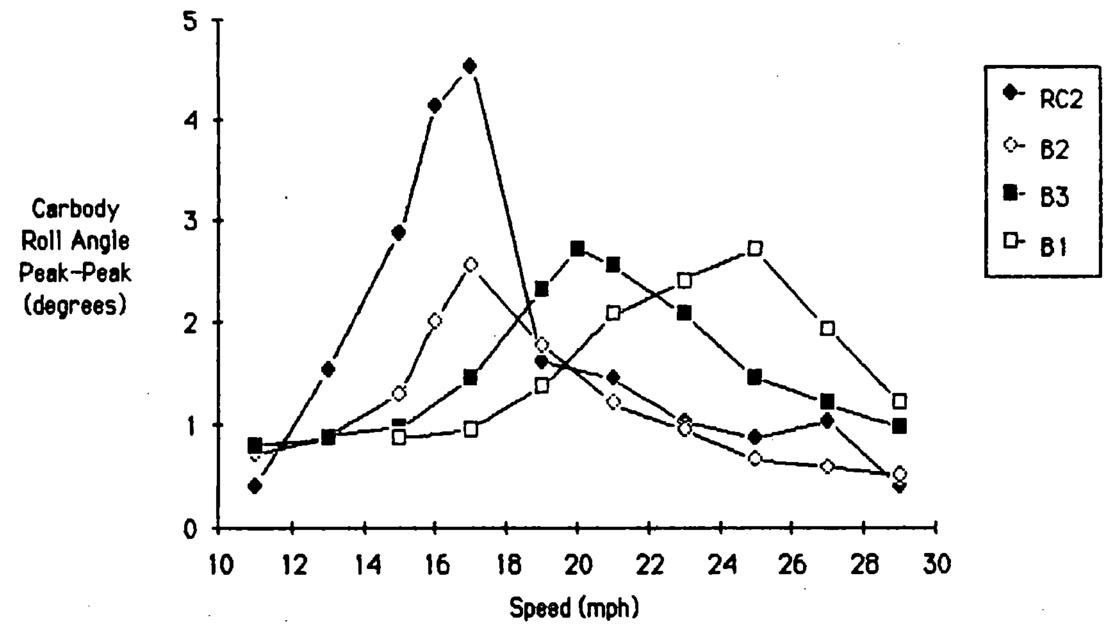


FIGURE 6-40
RESPONSE VS. CROSS LEVEL, ALL CONFIGURATIONS
RECTIFIED SINE STAGGERED RAIL WITH LATERAL



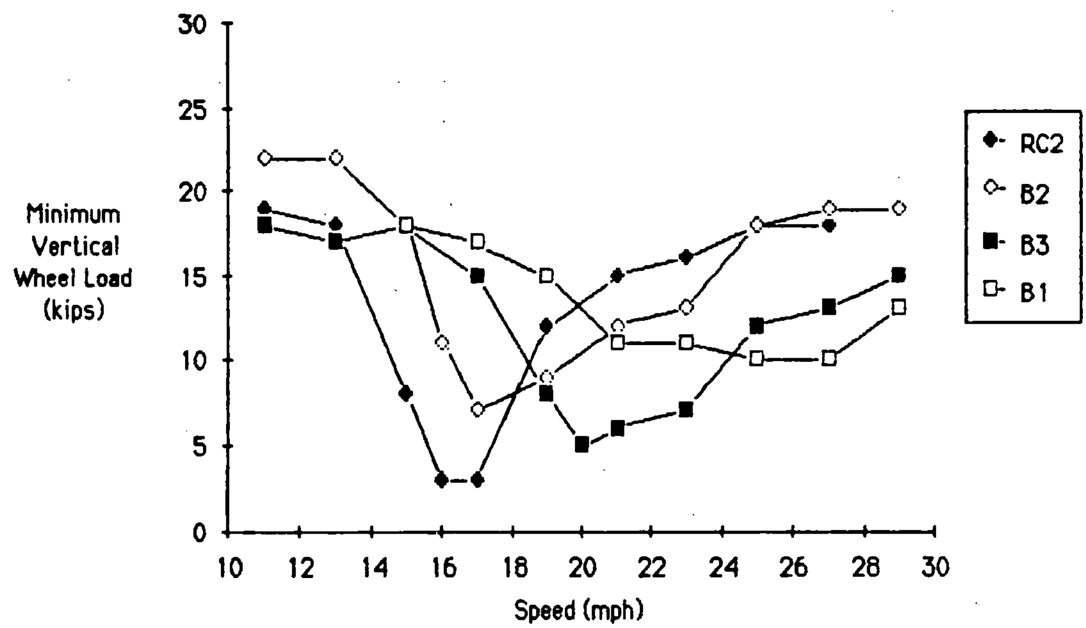


FIGURE 6-41
CONFIGURATION RESPONSE COMPARISON
0.50-INCH CROSS LEVEL RECTIFIED SINE STAGGERED RAIL
WITH 2.0-INCH SUPER ELEVATION

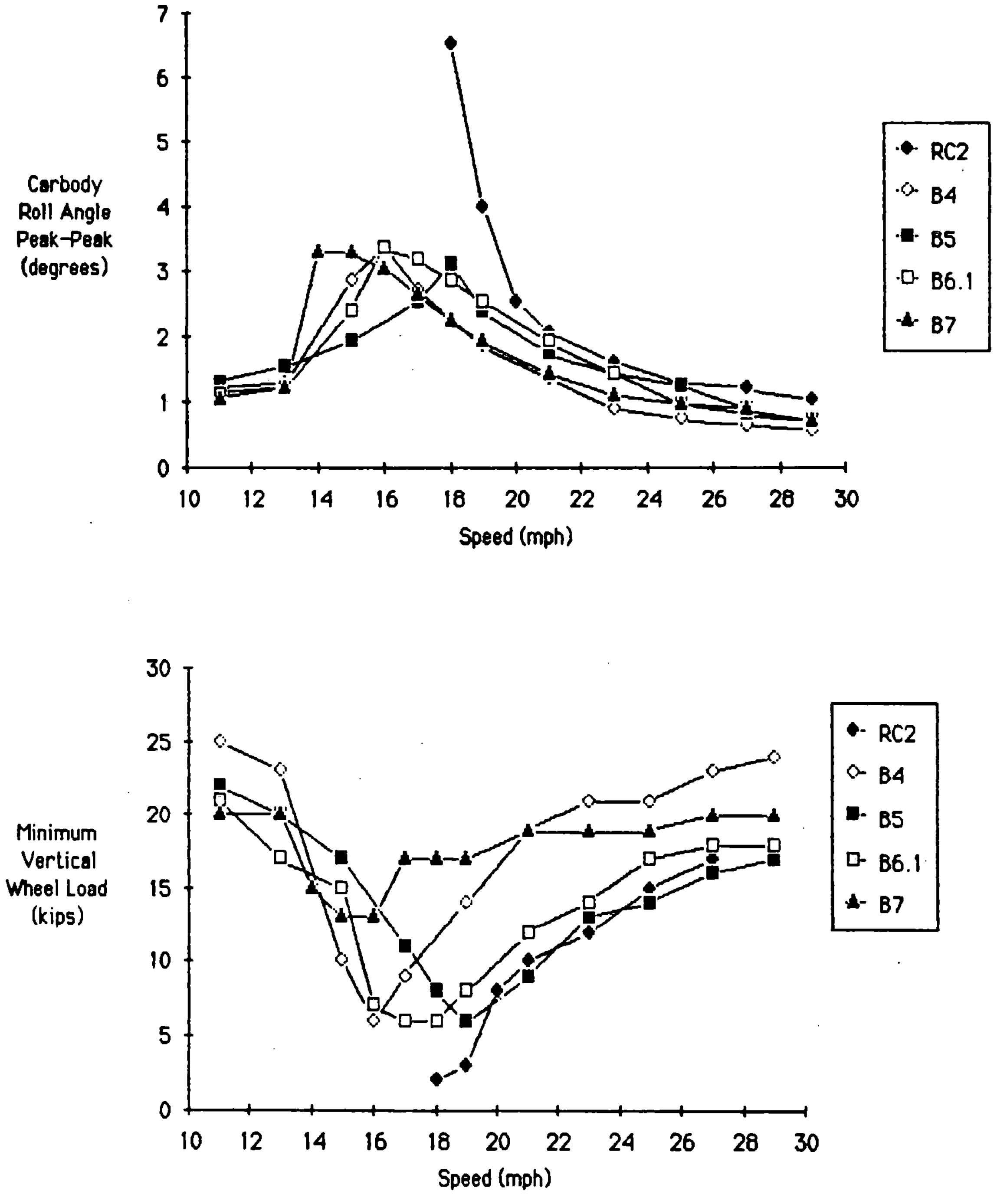
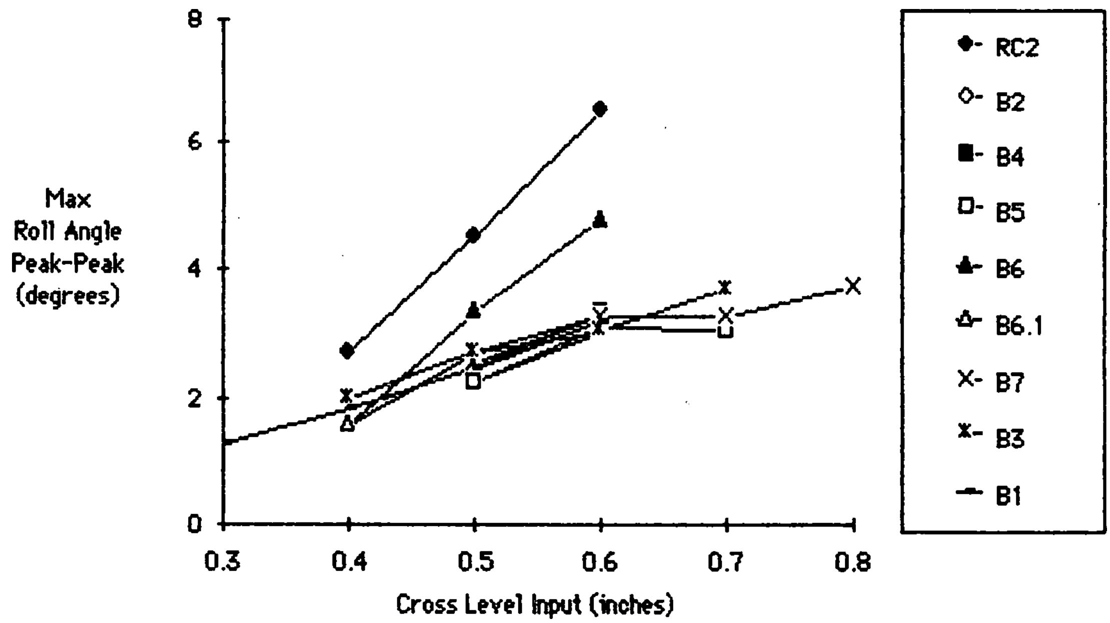


FIGURE 6-42
CONFIGURATION RESPONSE COMPARISON
0.60-INCH CROSS LEVEL RECTIFIED SINE STAGGERED RAIL
WITH 2.0-INCH SUPER ELEVATION



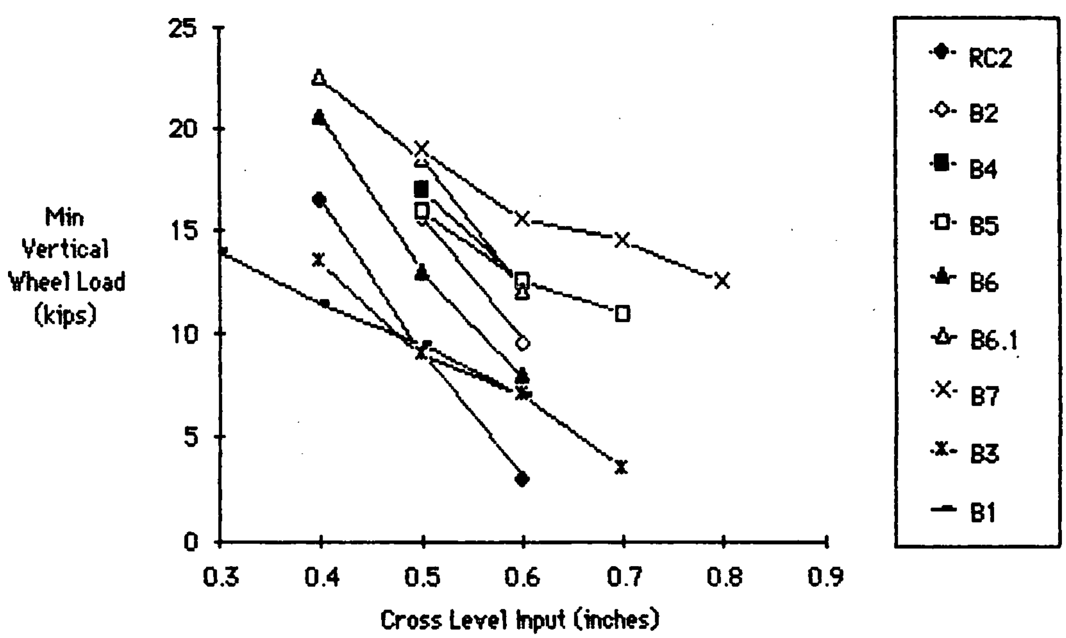


FIGURE 6-43
RESPONSE VS. CROSS LEVEL, ALL CONFIGURATIONS, RECTIFIED SINE STAGGERED RAIL WITH 2.0-INCH SUPER ELEVATION

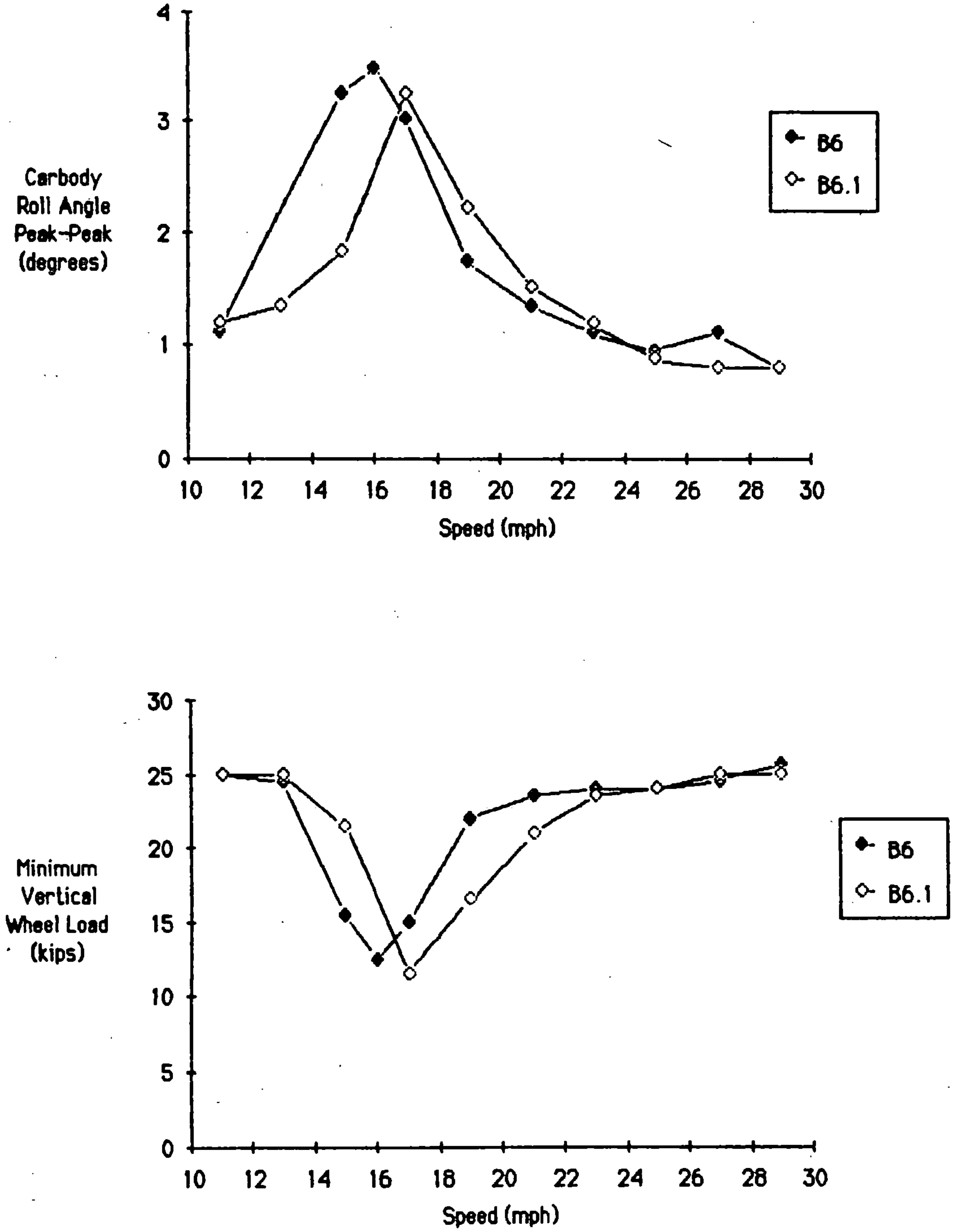
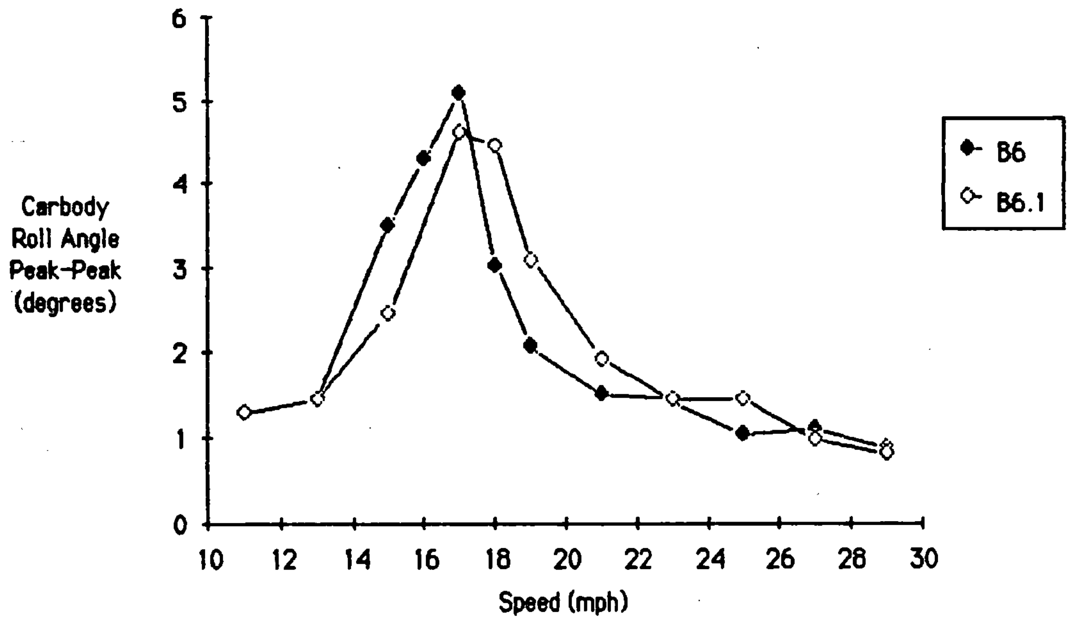


FIGURE 6-44
EFFECT OF SIDE BEARING CLEARANCE
0.60-INCH RECTIFIED SINE STAGGERED RAIL



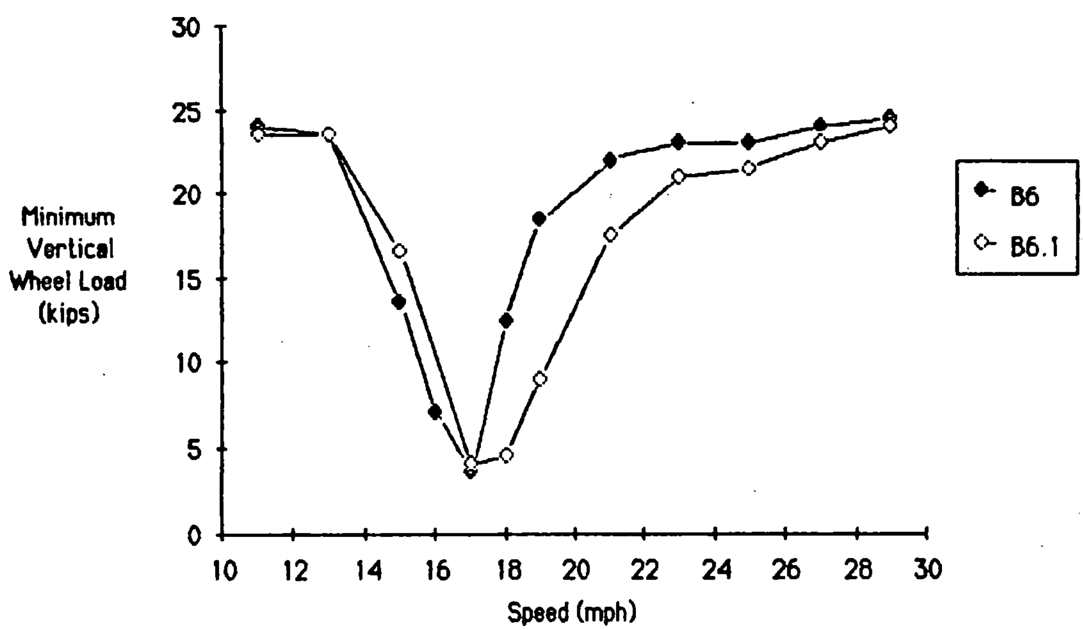


FIGURE 6-45
EFFECT OF SIDE BEARING CLEARANCE
0.60-INCH RECTIFIED SINE STAGGERED RAIL WITH LATERAL

- B5 - the S-2-C with single side springs and auxiliary hydraulic snubbers, and

- B7 - the S-2-C with constant contact resilient side bearings.

The following observations are made on the results:

- B5 is the best performer having the smallest roll angles and wheel lift above 0.90 inches.

- B7 ranks as the second best performer. It produces relatively small roll angles and even less wheel off-loading than B5. However, center plate lift occurs just at 0.80 inches cross level, apparently due to action of the constant contact resilient side bearings.

- B4 and B2 are next with B2 being better than B4 because B4 goes to slightly larger roll angles than B2 and is not able to go to as large input cross level amplitudes.

- RC2 results in the largest carbody roll angles and has wheel lift at the lowest cross level amplitude (0.70 inches). As a result, it ranks as the poorest performer.

Comparison of the effect of side bearing clearance is made in Figures 6-44 and 6-45 with Configuration B6 at 5/16 inch side bearing clearance and B6.1 at 1/4 inch side bearing clearance. This 1/16 inch change in clearance had very little effect except that the larger clearance resulted in slightly larger carbody roll angles.

<u>Conclusions Drawn from Staggered Rail Testing</u>

The following conclusions are based on the results of the VTU staggered rail testing with respect to the safety performance of the base 100-ton covered hopper car.

1. Light and partial load vehicles will reach wheel lift or center plate lift conditions at lower amplitudes of cross level and consequently have reduced safety margins relative to the fully loaded vehicle.

2. Increased friction snubber force enables vehicle to negotiate larger cross level, staggered rail without wheel lift. Optimum performance is achievable with a combination of friction and hydraulic snubbing.

3. Constant contact resilient side bearing results in improved performance, approaching the friction and hydraulic snubber case, with respect to reducing body roll angle and retarding the onset of wheel lift. However, these benefits are negated by an increased tendency for the center plate to lift off.

4. Increased side bearing clearance will yield slight loss in safety performance.

<u>Bounce Test Results</u>

There are three track conditions that may cause response of the freight car in its bounce mode:

- hard spots, such as at grade crossings,
- soft spots, such as may be caused by poor drainage, and
- non-staggered rail.

Non-staggered rail in the United States is, of course, very unusual and soft spots are avoided or corrected. However, grade crossings, and other track structures causing hard spots are existing conditions that must be dealt with. Consequently, tests were performed to measure the bounce response characteristics of the 100-ton covered hopper car. Three types of tests were performed:

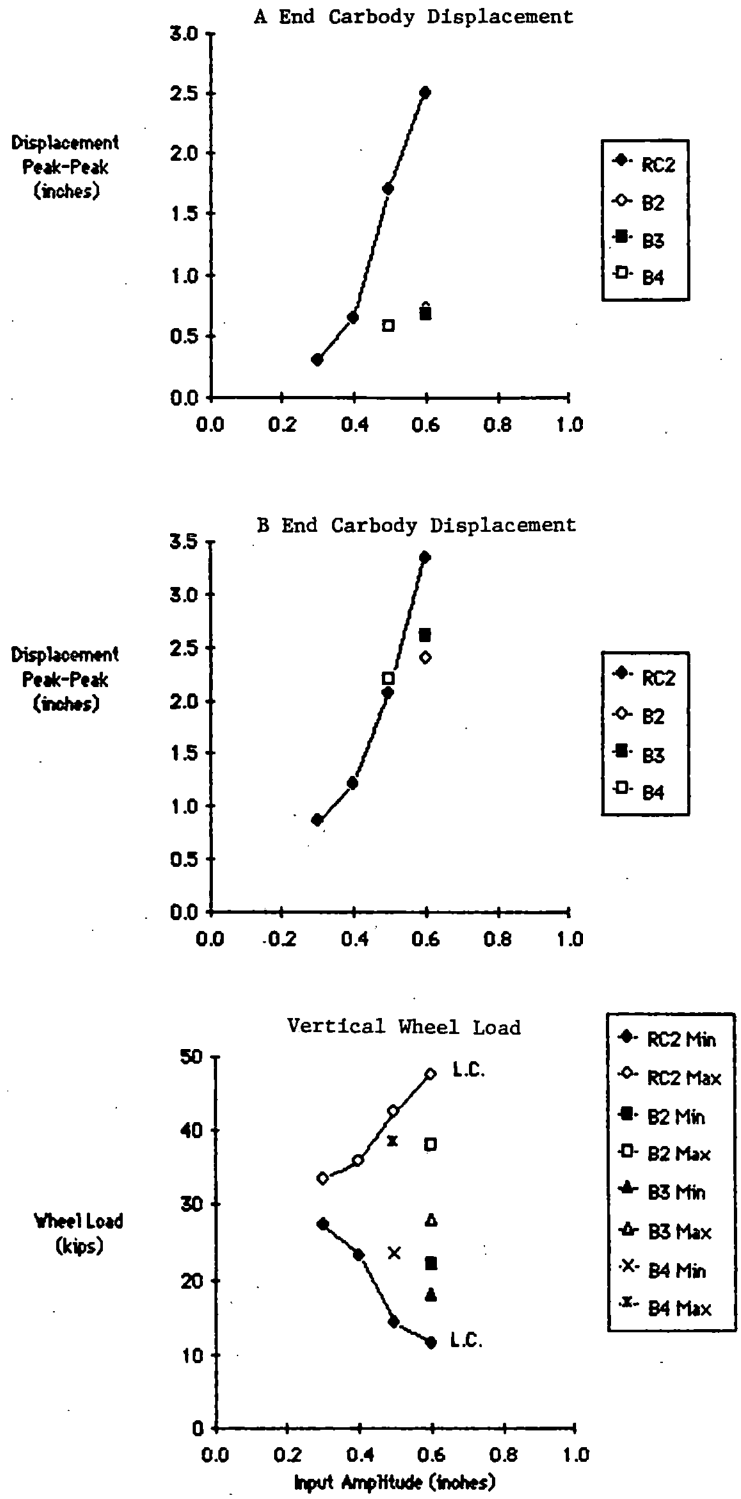
- speed sweeps, 40-100 mph, with 39-foot rail with rectified sine profile,

- constant speed runs over 10 39-foot rails with rectified sine profile, and

- constant speed runs over one bump, 1-cos shape and 17 and 25 feet long.

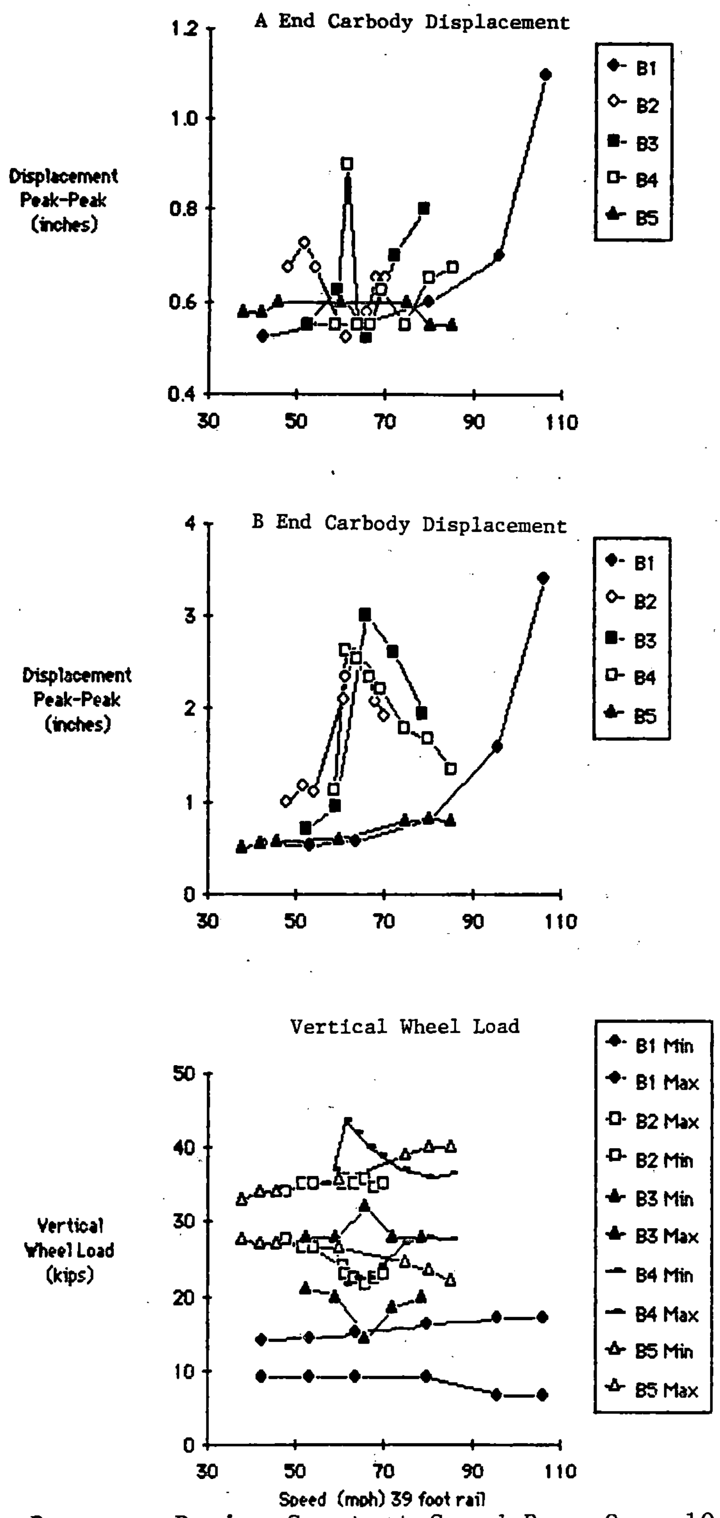
Results are summarized in Figures 6-46 through 6-49.

Figure 6-46 presents the results of the speed sweep tests which were performed in runs from 40 to 100 mph over 39-foot rail of constant amplitude profile variation, with a rectified sine shape. Maximum values of carbody displacement and vertical wheel loads were taken from the speed sweeps and plotted against input amplitude. The speed at which maximum response occurred varied from 81 to 55 mph as the input levels were varied from 0.30 to 0.60 inches. The RC2 configuration had the largest responses with a limit check condition reached at 55 mph with an 0.60-inch profile variation. For the RC2 configuration, the



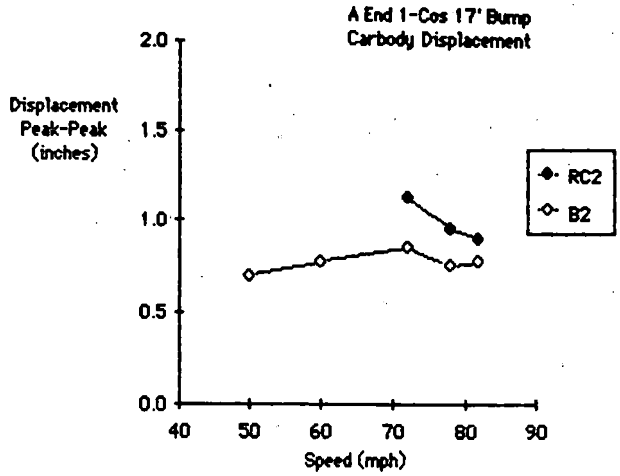
Maximum Response During 40-100 mph Acceleration Runs
Over 39-foot Parallel Rail with Rectified Sine Profile Shape

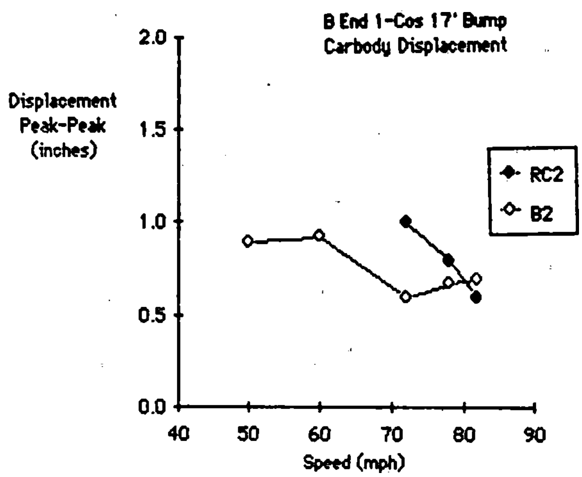
FIGURE 6-46
BOUNCE TEST RESULTS, ACCELERATION RUNS

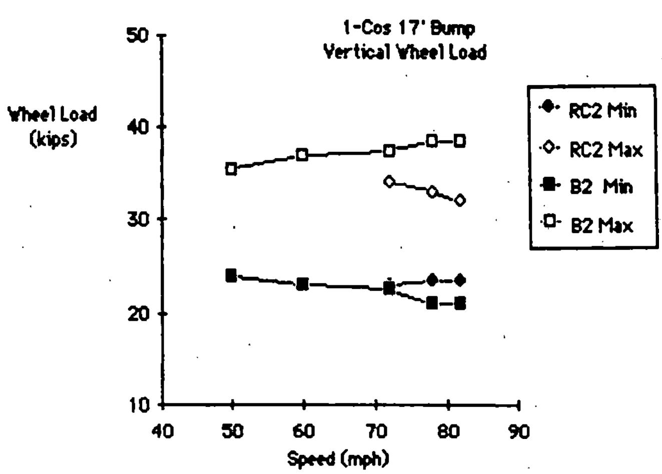


Maximum Response During Constant Speed Runs Over 10 Lengths of 39-foot Parallel Rail with Rectified Sine Profile Shape and 0.67 Inches in Height

FIGURE 6-47
BOUNCE TEST RESULTS, CONSTANT SPEED RUNS

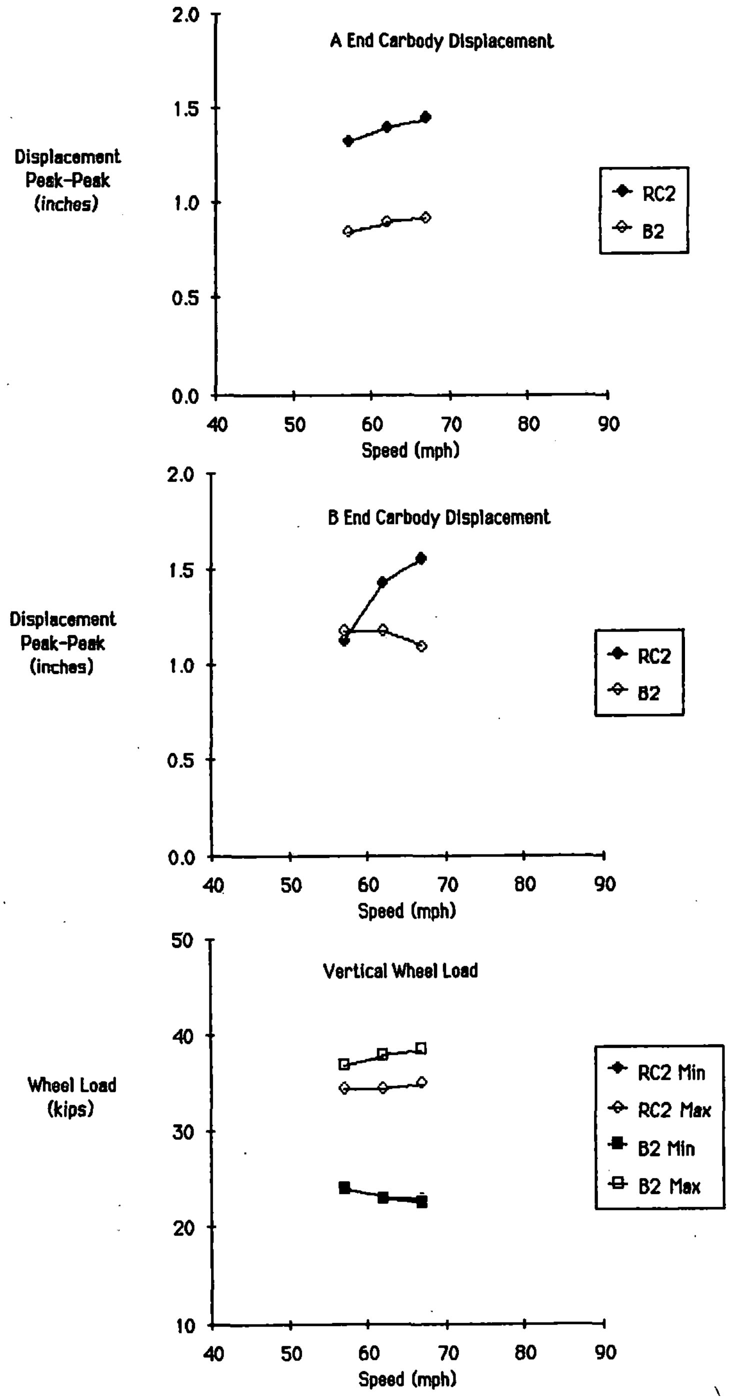






With a 1-cos Parallel Rail Bump, 17 Feet Long and 0.67 Inches in Height

FIGURE 6-48
BOUNCE TEST RESULTS, SINGLE 17-FOOT BUMP



With a 1-cos Parallel Rail Bump, 25 Feet Long and 0.67 Inches in Height

FIGURE 6-49
BOUNCE TEST RESULTS, SINGLE 25-FOOT BUMP

B end had larger responses than the A end. For the Barber configurations, the A end response was essentially a 1:1 ratio to the input.

Figure 6-47 presents the results of constant speed runs over ten lengths of 39-foot parallel (i.e., non-staggered) rail with rectified sine profile shape with a height of 0.67 inches. This test was not performed with the RC2 configuration. Five Barber configurations were tested.

The carbody displacements at the A end show the strong influence of the binding A snubbers, all A end displacements being relatively small. The empty vehicle, Configuration B1, has the largest displacements, at both ends, above 90 mph, but is very well behaved below 80 mph.

Configuration B5, the loaded vehicle with reduced friction snubbers and auxiliary hydraulic snubbers, was, by far, the best performer.

Figures 6-48 and 6-49 show vehicle responses to a single bump in the track with a 1-cos shape with a height of 0.67 inches, and lengths of 17 and 25 feet. The results show the RC2 responses to be consistently greater than for the B2 configuration.

In summary, the configuration which showed the best performance in the bounce testing was B5, the loaded vehicle with reduced friction snubbers and auxiliary hydraulic snubbers. The empty vehicle also performed very well below 80 mph. The poorest performance was given by the RC2 configuration, probably simply due to the relatively light friction damping forces.

<u>Yaw Test Results</u>

The yaw tests were an attempt to duplicate motions and loads that result from body hunting conditions. The results indicate at what frequency body hunting will occur and a general indication of the severity of the motions. The results from tests of the following five truck configurations are presented:

- RC2 - the Ride Control truck,
- B2 - the Barber truck in normal configuration,
- B3 - the Barber truck, with 27.6% off-load,
- B4 - reduced friction Barber truck, and
- B5 - reduced friction Barber truck with auxiliary hydraulic snubber.

The tests consisted of imposing yaw motions on the freight car in frequency sweeps from 1.5 to 4.0 Hertz, the typical hunting frequency range and corresponding to the speed range of approximately 30 to 80 miles per hour. Results from each configuration tested are shown in Figures 6-50 through 6-54, each showing plots of carbody roll angles, and wheel vertical and lateral loads. A summary is given in Figure 6-55.

The RC2 results (Figure 6-50) are typical in that the response-to-input ratio increased as the input increased. This is probably due to the friction snubber action which inhibits the truck bolster lateral movements more at small amplitudes because the snubbers have not started to slide.

Maximum response is also dependent on the friction forces provided by the snubbers: Configurations RC2, B4, and B5 have the smaller friction forces and result in the larger responses.

The auxiliary hydraulic snubbers in Configuration B5 are not effective with lateral motion of the truck and consequently do not help inhibit yaw motions.

6.6 CLI Evaluation Tests with Shape D Profile, Staggered Rail, Base Vehicle No. 2

The tests performed on Base Vehicle No. 1, described in Section 5, were performed for the primary purpose of providing experimental data for verification of a Cross Level Index. Having two 100-ton covered hopper cars also provided a broader base for evaluating the cross level index and for defining characteristics of typical in-service covered hopper cars.

This "broader base" was provided by the two cars having different trucks (Barber and Ride Control), having different truck spacing (40'5" and 45'7"), and having different c.g. heights (89.2' and 97" with test load conditions).

In order to compare performance between the two base vehicles, the staggered rail test with Shape D profile used in the Vehicle No. 1 tests was repeated with Vehicle No. 2. Shape D is a sequence of ten exponential low joints with the relative amplitude of each successive low joint varied to form the increasing-decreasing pattern shown in Figure 6-56. The left and right rail have the same shape but are shifted a half rail length to obtain the staggered rail condition.

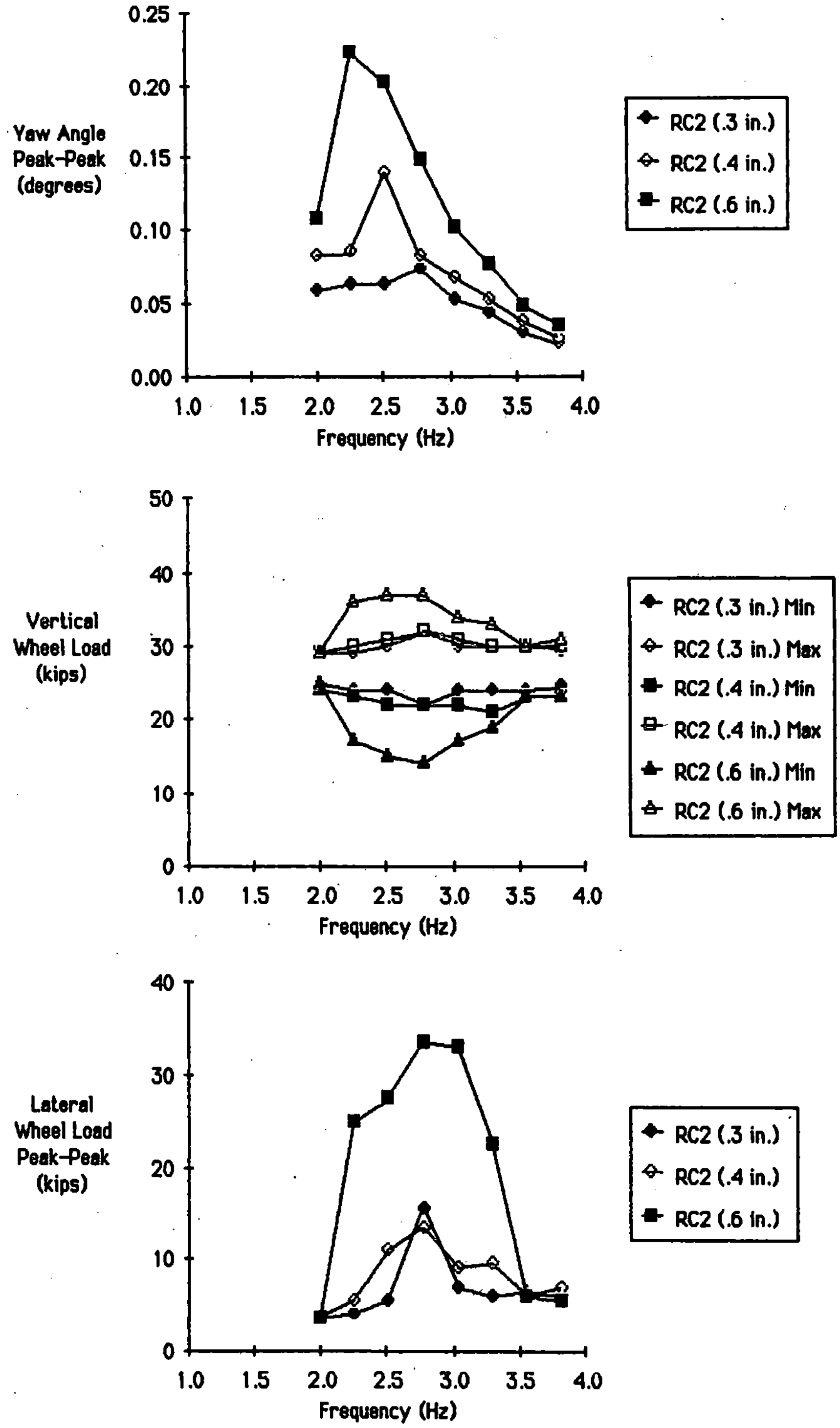


FIGURE 6-50
YAW TEST RESULTS, RC2 CONFIGURATION

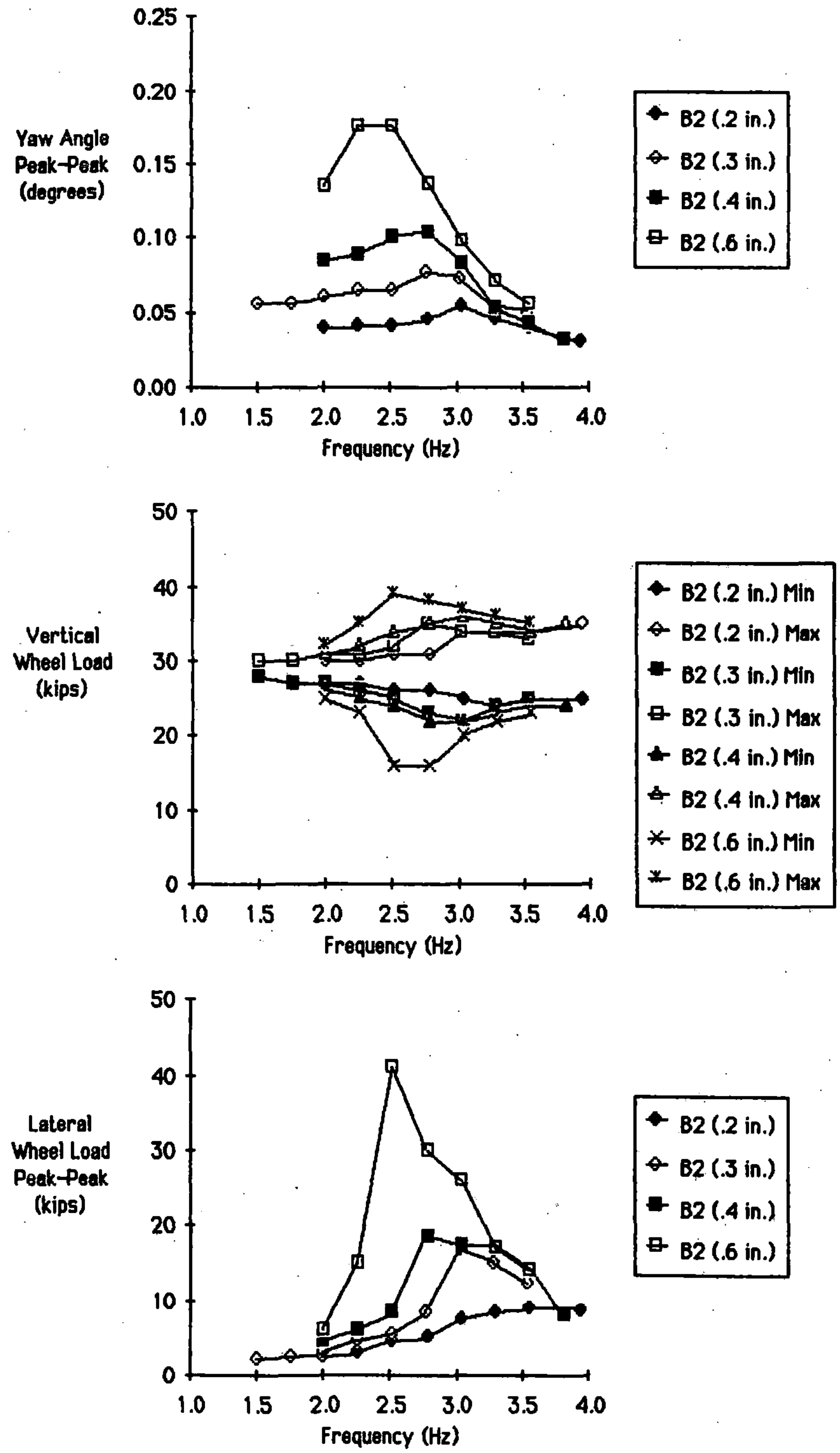


FIGURE 6-51
YAW TEST RESULTS, B2 CONFIGURATION

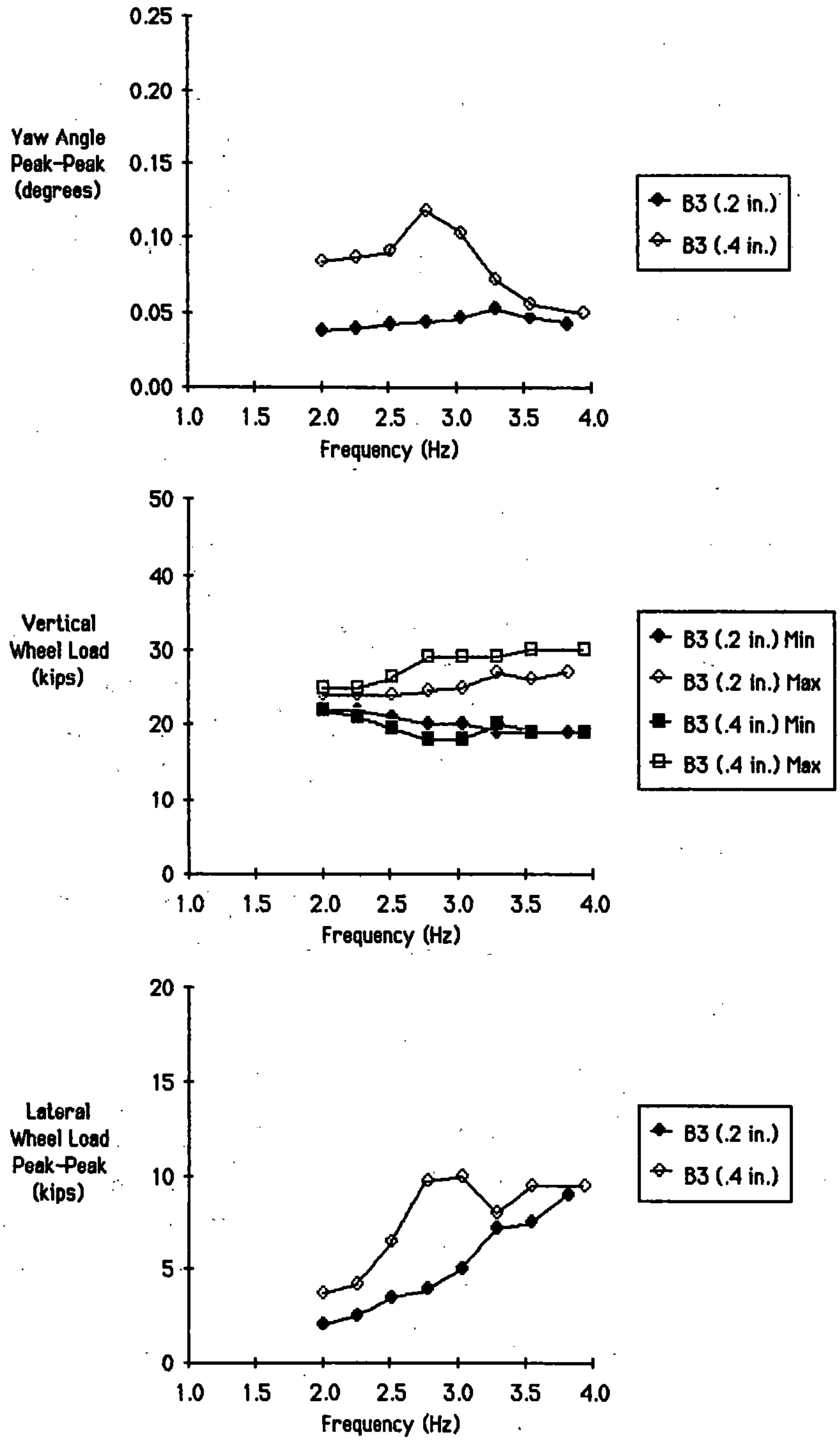


FIGURE 6-52
YAW TEST RESULTS, B3 CONFIGURATION

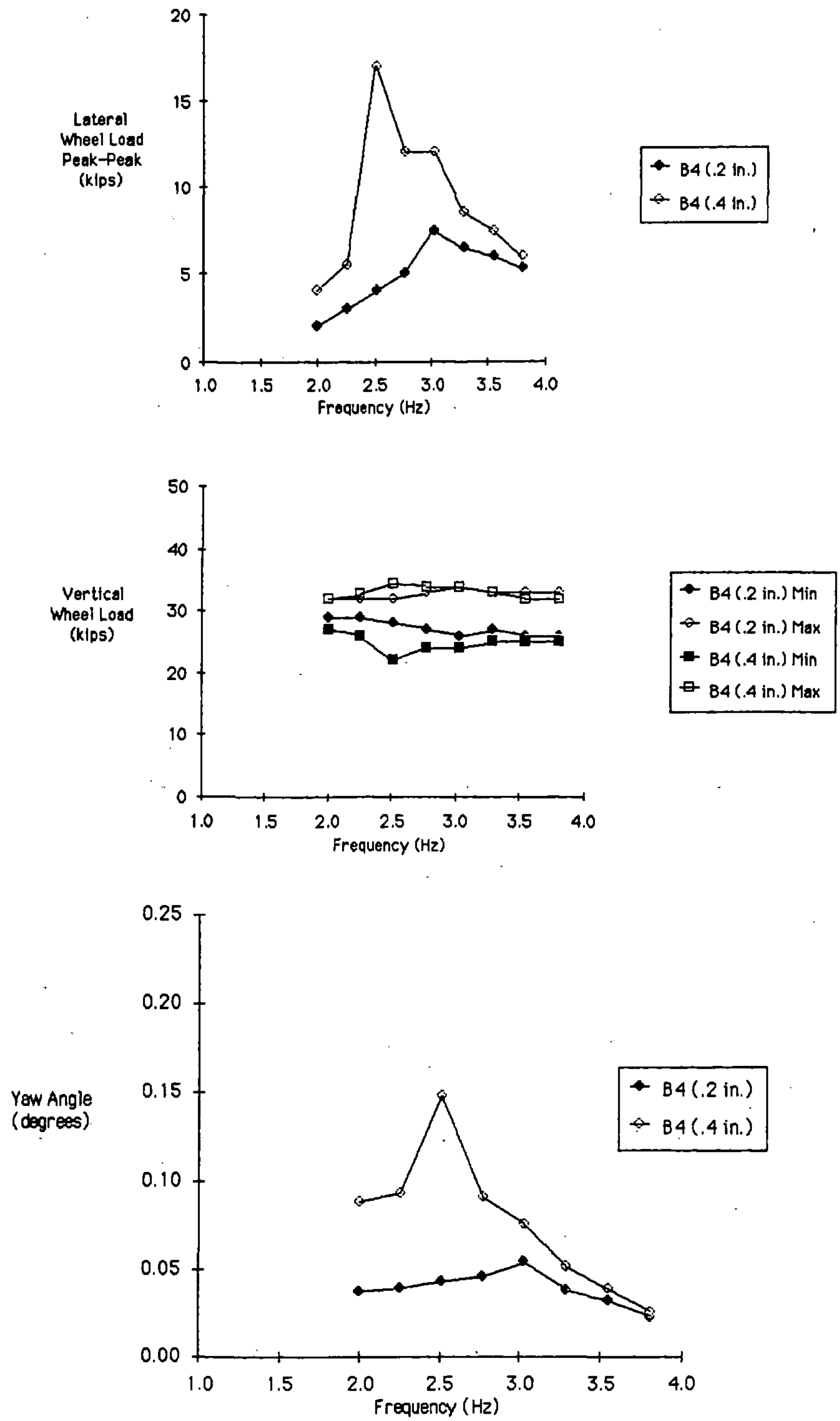


FIGURE 6-53
YAW TEST RESULTS, B4 CONFIGURATION

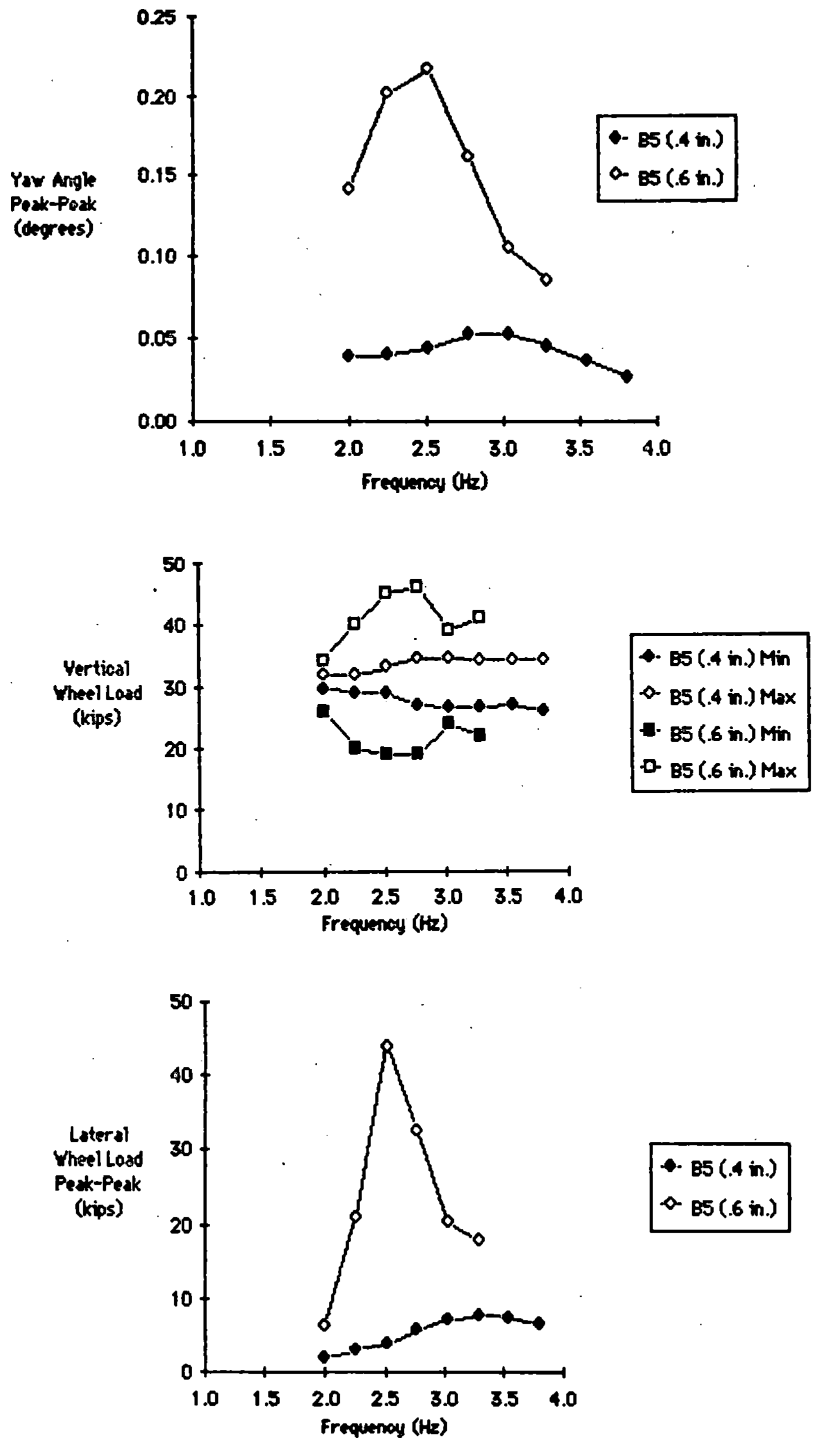


FIGURE 6-54
YAW TEST RESULTS, B5 CONFIGURATION

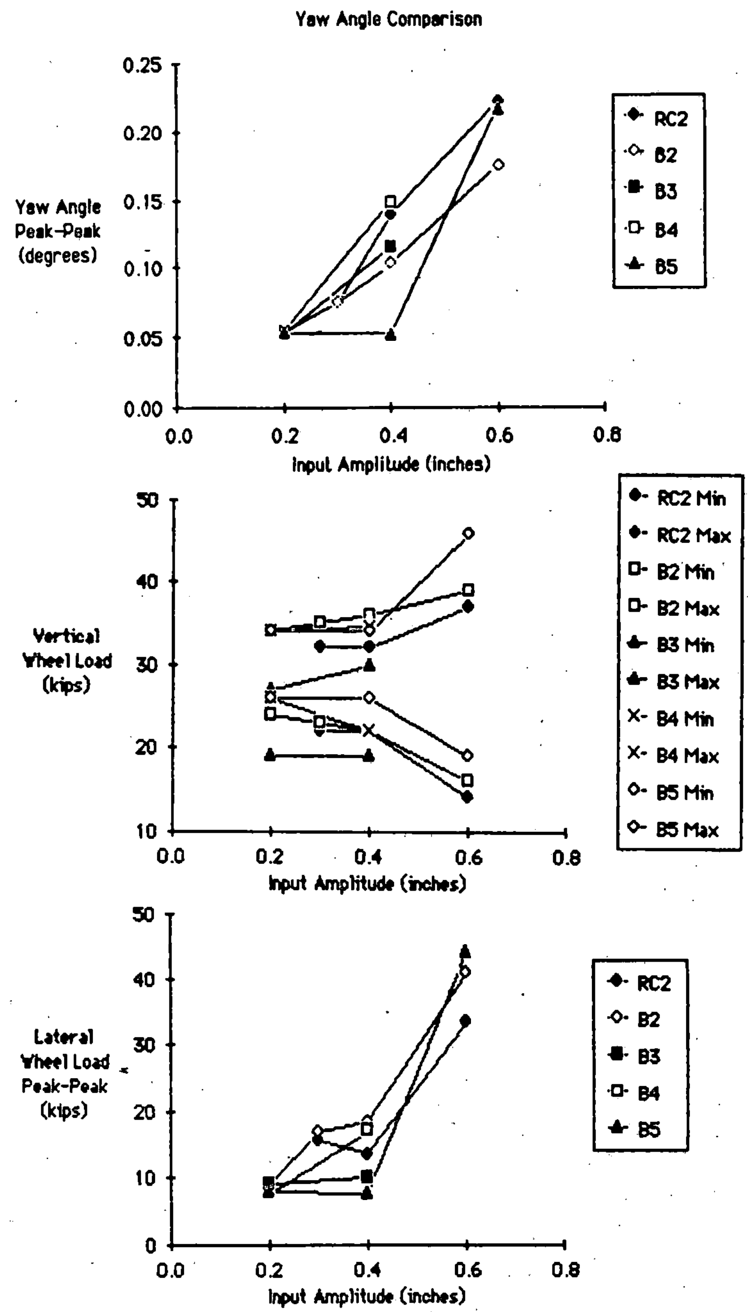


FIGURE 6-55
YAW TEST RESULTS, CONFIGURATION COMPARISON

6.6.1 Test Configurations, Shape D Staggered Rail Tests

There were four configurations of the Base Vehicle No. 2 tested with the Shape D staggered rail:

- RC2 - 99 tons lading, 89.2 in. c.g., Ride Control truck,
- B2 - 99 tons lading, 89.2 in. c.g., Barber S-2-C truck,
- B3 - 72.4 tons lading, 81.6 in. c.g., Barber S-2-C truck, and
- B1 - empty vehicle, 64.0 in. c.g., Barber S-2-C truck.

The Base Vehicle No. 1 configuration used the same Barber S-2-C trucks, had 101.3 tons of lading and a c.g. height of 97.0 inches.

6.6.2 Test Procedure, Shape D Staggered Rail Tests

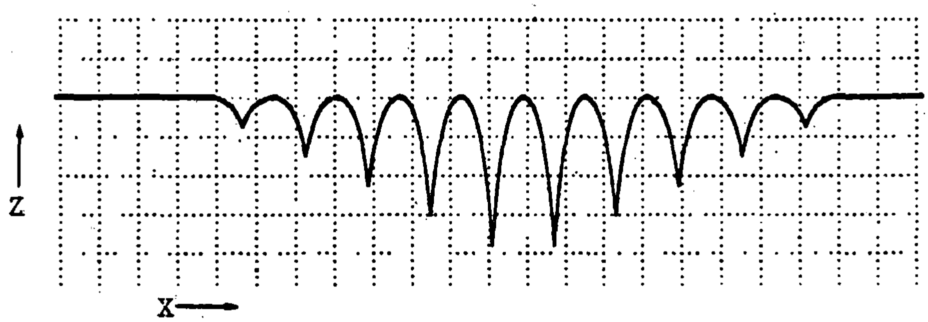
The test procedure followed was to apply the Shape D staggered rail profile to the vehicle over a range of speed and cross level amplitudes to find the critical speed and cross level that resulted in wheel lift.

6.6.3 Test Results, Shape D Staggered Rail Tests

Test results of carbody roll angle and vertical and lateral wheel loads are presented in Figures 6-57 through 6-60 for each of the four configurations tested. Stop-test conditions of wheel lift or center plate lift are also noted. Figure 6-61 presents a comparison of the four configurations for carbody roll angle and minimum vertical wheel load. The results from Base Vehicle No. 1 testing are also presented.

Performance of each configuration can be summarized in terms of the cross level where wheel lift or center plate lift is first encountered:

Configuration	Cross Level	Condition
B2 - Veh. No. 2	1.20 in.	near wheel lift
B3 - Veh. No. 2	1.20 in.	wheel lift
RC2 - Veh. No. 2	1.20 in.	wheel lift
B1 - Veh. No. 2	0.96 in.	center plate lift
B2 - Veh. No. 1	0.86 in.	wheel lift



Joint No.	1	2	3	4	5	6	7	8	9	10
Relative Amplitude	0.2	0.4	0.6	0.8	1.0	1.0	.8	.6	.4	.2

Left rail and right rail shifted a half rail length relative to each other for staggered rail condition.

FIGURE 6-56
SHAPE D EXPONENTIAL LOW JOINT PROFILE

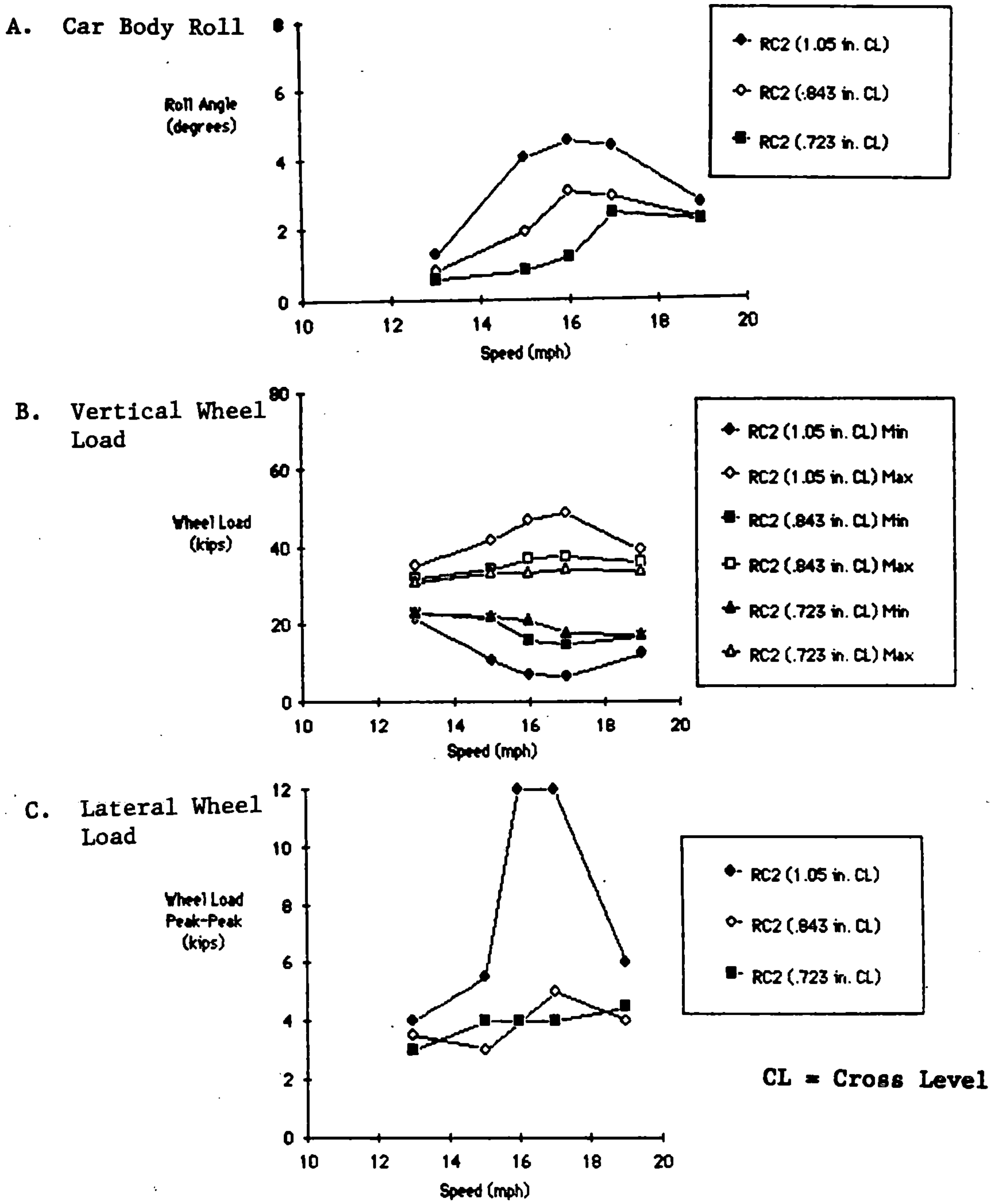


FIGURE 6-57
STAGGERED RAIL, SHAPE D TEST FOR CLI EVALUATION, CONFIGURATION RC2 RESULTS

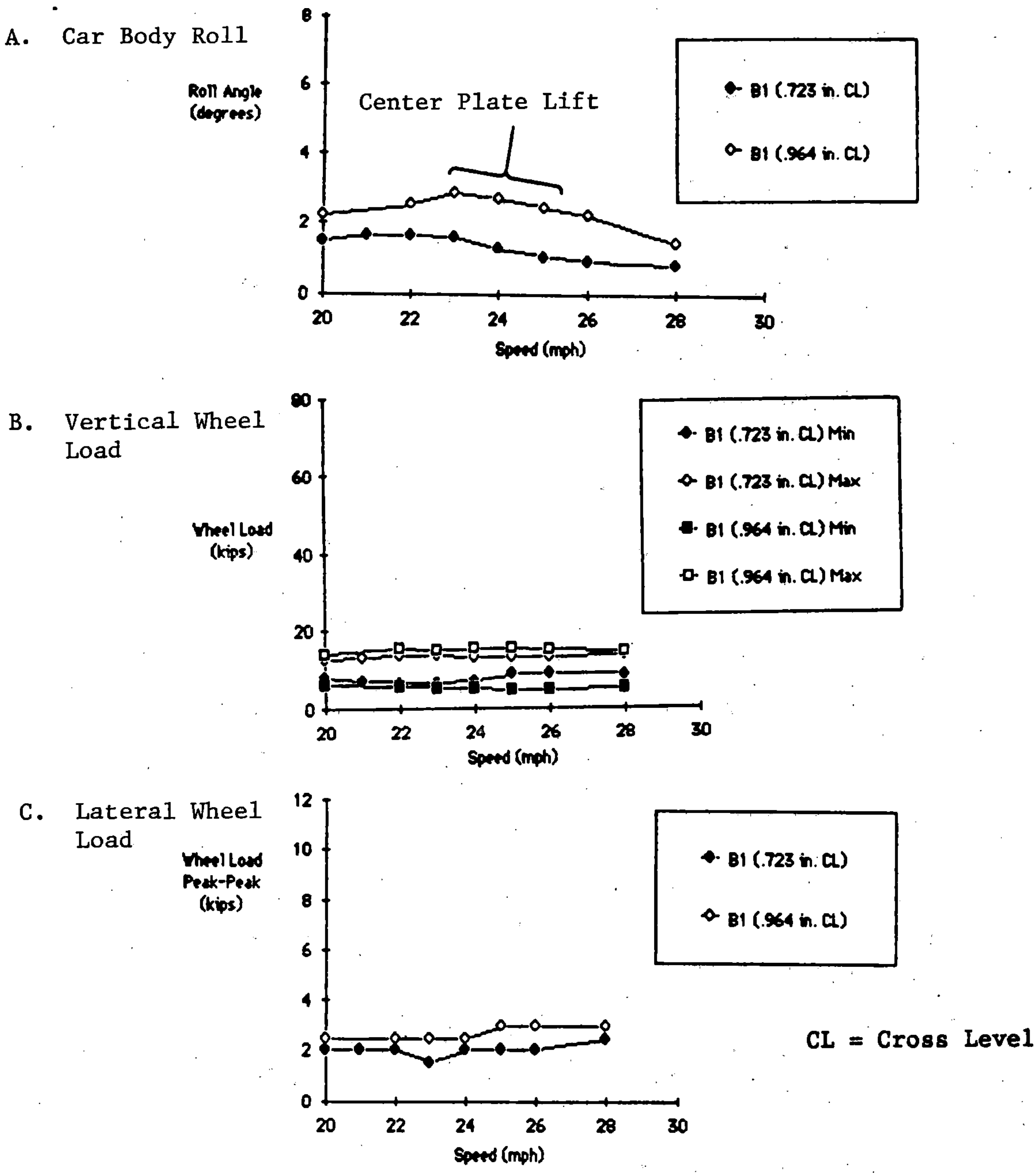


FIGURE 6-58
STAGGERED RAIL, SHAPE D TEST FOR CLI EVALUATION, CONFIGURATION B1 RESULTS

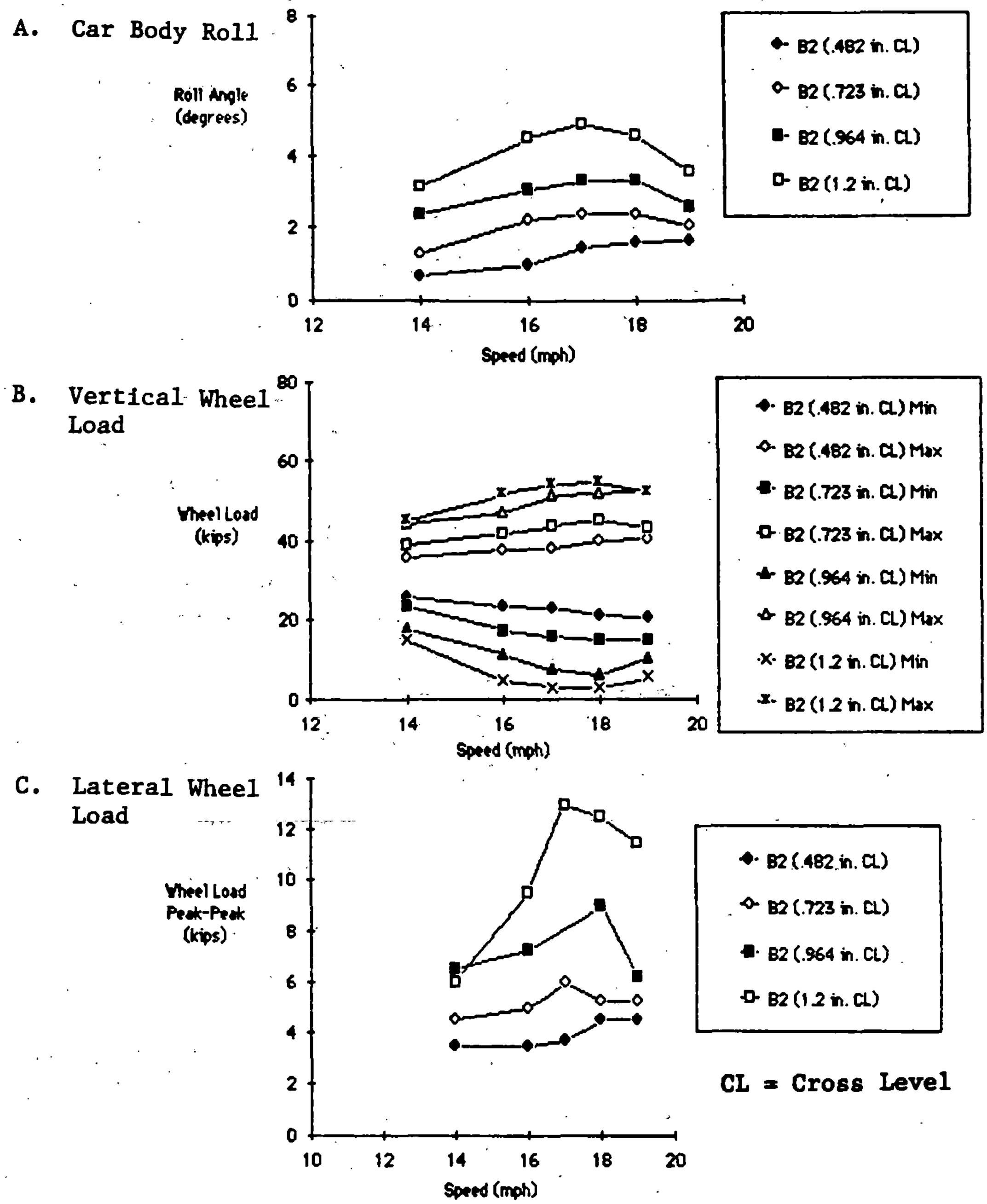


FIGURE 6-59
STAGGERED RAIL, SHAPE D TEST FOR CLI EVALUATION, CONFIGURATION B2 RESULTS

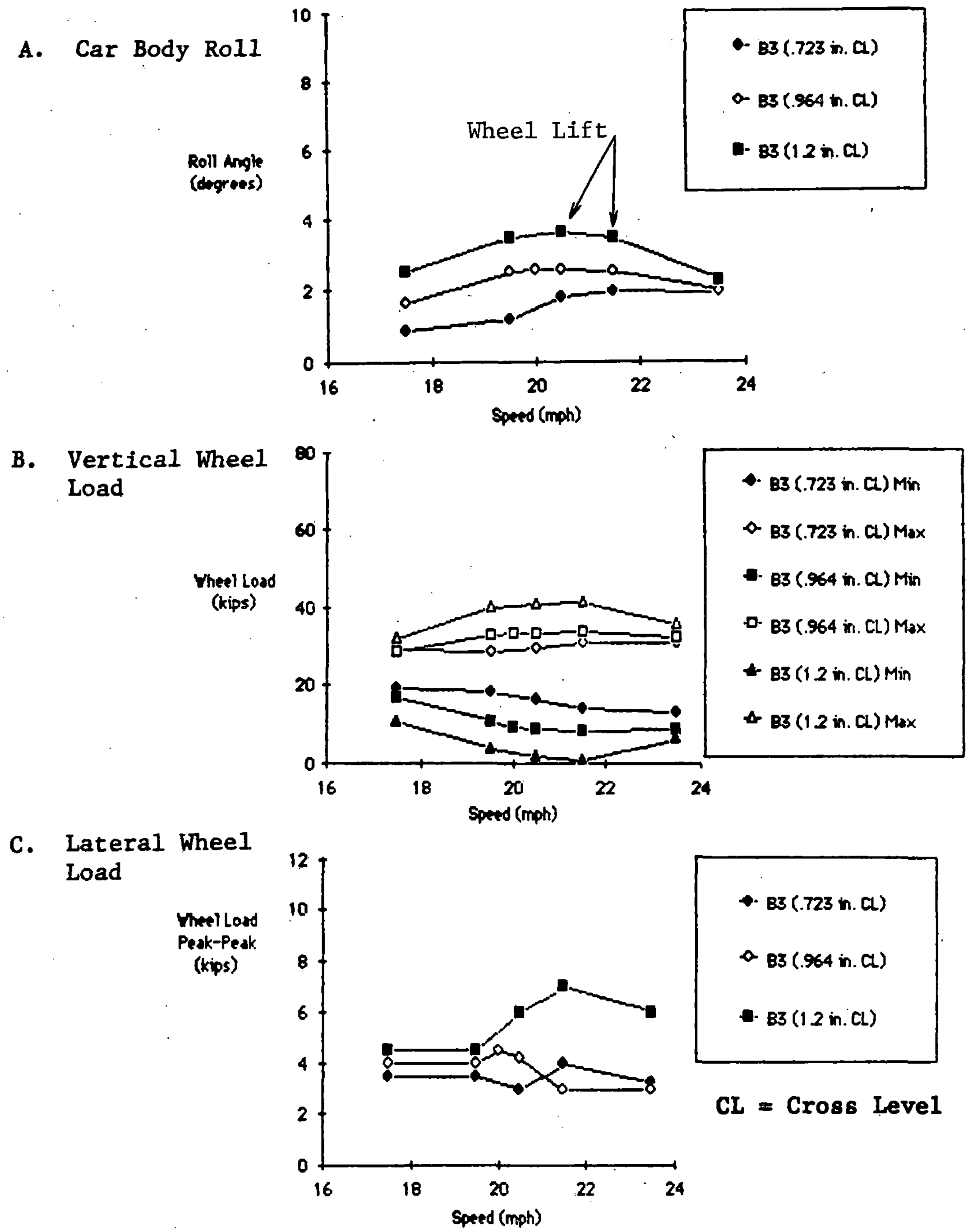
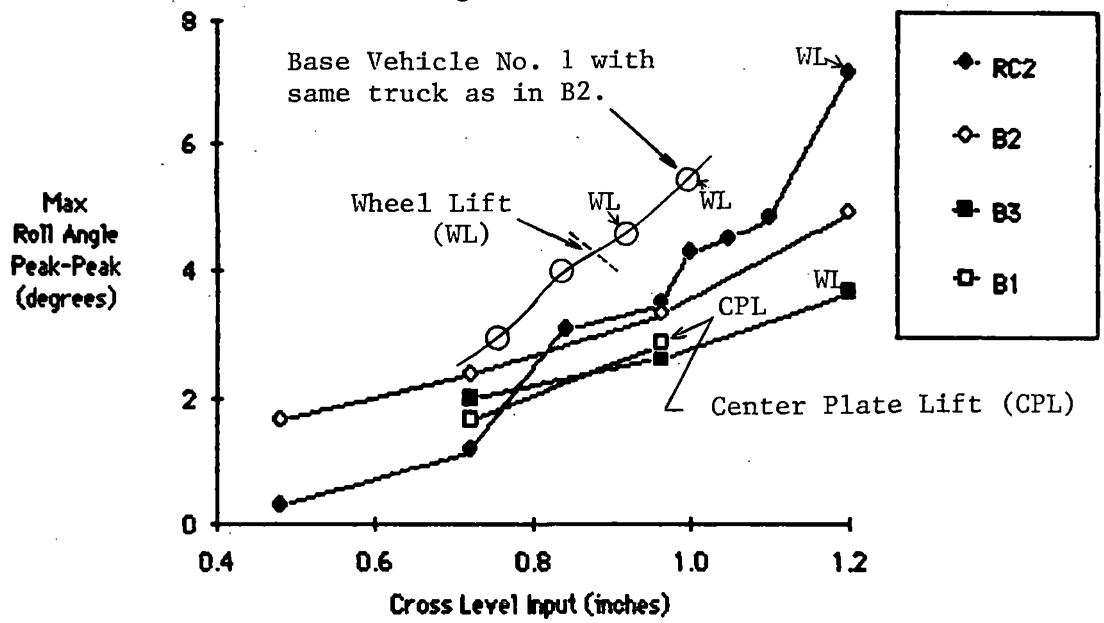


FIGURE 6-60
STAGGERED RAIL, SHAPE D TEST FOR CLI EVALUATION, CONFIGURATION B3 RESULTS

a. Maximum Car Body Roll Angles



b. Minimum Vertical Wheel Loads

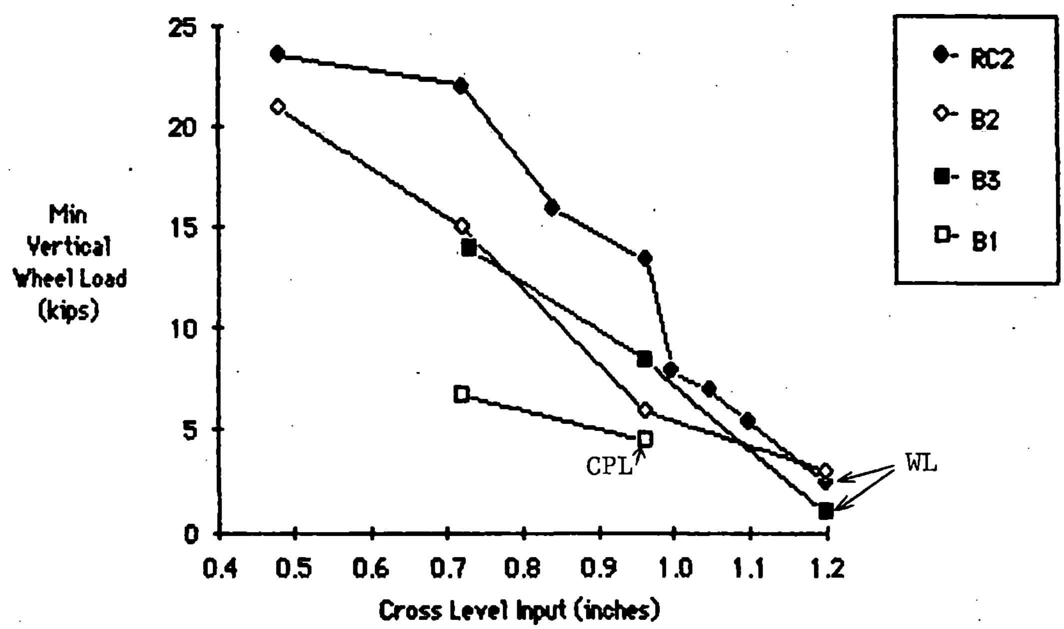


FIGURE 6-61
STAGGERED RAIL, SHAPE D TEST FOR CLI EVALUATION, COMPARISON OF CONFIGURATIONS

Configuration B2 performed the best in this particular staggered rail test. The cross level input of 1.20 inches was reached without encountering wheel lift. Extrapolation of wheel load data indicates that wheel lift would probably occur at about 1.3 inches cross level.

Configurations B3 and RC2 both have wheel lift at 1.2 inches cross level. However, the carbody roll angles for RC2 are at 7.1 degrees, peak-to-peak, compared to 3.6 degrees for B3. Configuration B3 has reduced lading weight and lowered c.g. (81.6 inches compared to 89.2 for B2) and was expected to perform better than the fully loaded configuration (B2). On the basis of carbody roll angle, it was the better performer. However, based on wheel lift, it was equal to the RC2 configuration and did not do as well as the B2 configuration.

Vehicle No. 1 was not expected to perform as well as Vehicle No. 2, as was the case, on the basis of two configuration differences: (1) the truck spacing of 40'5" is closer to the 39' rail length used, and (2) the c.g. height is greater. Vehicle No. 1 had larger carbody roll angle than any other configuration and reached wheel lift conditions before wheel lift or center plate lift by any other configuration.

7. SUMMARY AND CONCLUSIONS

The intent of this report has been to describe laboratory tests performed on two, 100-ton covered hopper cars and to present the test results. The laboratory tests performed generated three categories of results -- truck properties, vehicle modal characteristics, and dynamic response to selected track conditions simulated on the Vibration Test Unit. A brief summary of selected results divided into these three categories are presented below.

7.1 Truck Characterization Tests

Load-displacement tests were performed on each of the Ride Control and Barber trucks to determine their characteristic stiffness and friction snubber forces. Results of these tests are summarized in Tables 7-1 and 7-2, with the spring notation used shown in Figure 7-1. These results have two main applications: (1) to be used as a basis for comparison of different truck configurations and in the study of cause and effect of truck performance evaluations, and (2) for use in mathematic models.

Comparison of spring rates and friction forces between the Barber S-2-C and Ride Control trucks shows a number of differences. The friction forces are different, which is to be expected since the Barber truck has load variable and the Ride Control truck has constant column friction snubbers. The difference is made even larger by the binding snubbers in the Barber A end truck. At 90% load without binding, the load variable snubber friction force is about twice the constant column (7.5/3.5 kips), while with binding, it is about four times (13.3/3.5 kips).

The difference in friction force is also the probable cause for the suspension lateral stiffness being larger in the Barber truck (49.5 compared to 28 kips/inch) at the 90% load condition. A third difference between the two types of trucks is in the lateral stiffness of the side frames. However, because of the tendency for the side frames to roll, it must be assumed that this lateral stiffness is a variable property and can fall in a relatively broad range of values for any three-piece truck.

The spring constants shown in Tables 7-1 and 7-2 are for the mathematical model shown in Figure 7-1 which is based on the FRATE computer program [9]. These values have been calculated again in terms of the spring configuration used in the flexible

TABLE 7-1
SUMMARY OF SPRING RATE DATA

TRUCK ITEM	BARBER		RIDE CONTROL		UNITS
	EMPTY	90%	EMPTY	90%	
K(1), K(3)	527	647	650	650	K lb/in
K(2)	118	160	100	1025	K lb/in
K(4)	38.4	44.8	41.6	55.8	K lb/in
K(5)	17.9	49.5	20	28	K lb/in
K(6) a	623	890	850	1120	K in lb/deg
K(6) b	0	0	360	430	K in lb/deg
K(6) c	1067	1250	850	1120	K in lb/deg
K(6) d	0	0	0	0	K in lb/deg
Friction Force [1]	1.6	4.4	4.3	3.5	K lb
Friction Force [2]	3.0	7.5	---	---	K lb
Friction Force [3]	3.0	13.3	---	---	K lb

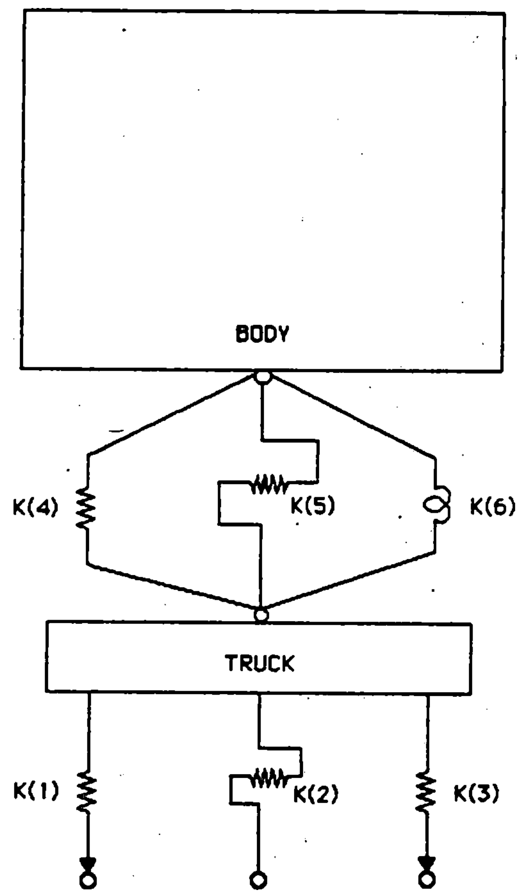
TABLE 7-2
ROLL SPRING RATES OF BARBER TRUCK WITH CONSTANT
CONTACT RESILIENT SIDE BEARINGS

TRUCK ITEM	EMPTY	90%	UNITS
K(6) a	645	960	K in lb/deg
K(6) b	732	1120	K in lb/deg
K(6) c	732	1120	K in lb/deg
K(6) d	0	0	K in lb/deg

[1] reduced friction snubbers (single side springs)
[2] normal snubbers (double side springs)
[3] binding snubbers (double side springs)

K(6) a: center plate seated
K(6) b: center plate rocking
K(6) c: side bearing contact
K(6) d: center plate lifted

Refer to Figure 7-1 for spring notation.



<u>Definition</u>

Body - The total mass of the vehicle body and contents.

Truck - The truck mass: actually a hypothetical value to represent the unsprung portion of the truck.

K(1), K(2), K(3) - Vertical and lateral stiffnesses combining side frame, journals, wheels, and rail.

K(4), K(5), K(6) - Vertical, lateral, and roll stiffnesses of the suspension system including truck bolster and center plate.

FIGURE 7-1
TRUCK SCHEMATIC SHOWING SPRING NOTATION

body freight car model used in the track-train Dynamic Harmonic Roll Series, [10] and [11]. The numbers are shown in Table 7-3. The center plate stiffness is low compared to [10] and [11] probably because the flexibility of the truck bolster and other local structures is included. The suspension vertical stiffnesses are within the variability of that property. The suspension lateral stiffnesses are also within the range of normal variability considering that the value for the Barber truck is high because of the high forces of the friction snubbers.

The differences seen in the "Side Frame and Track" spring constants are due to the test numbers being obtained with a relatively stiff track, the rails being situated on a relatively stiff laboratory floor. Consequently, the test, side frame and track stiffness, should be corrected with a proper value of track stiffness. However, also to be considered is the wide range of variability of this property as evidenced by the difference between values for the Barber and Ride Control trucks.

7.2 Modal Test Results

The modal test results provide a measure of the total vehicle dynamic characteristics in terms of the identification of significant modes of vibration and their characteristic frequencies. The bar chart in Figure 7-2 identifies the vehicle modes of dynamic response and the range of frequencies for each mode. Empty and loaded vehicle frequencies are shown separately.

The first roll mode is the mode with the lowest frequency, as expected. Its frequency, varying between 0.5 and 1.0 Hertz, is a broader range than expected.

It was not expected that there would be as much overlap of frequency as was found to exist in the next five modes. This may affect vehicle dynamic performance by causing a greater tendency for coupling between the modes. For example, the yaw and second roll modes overlap and may couple. Also, the bounce and pitch modes overlap and will tend to couple.

The torsion mode is lower than expected. This may be of benefit in curve entry and exit performance. It may, however, tend to be excited as a harmonic under hunting conditions.

Friction snubbing has two general effects on the modal frequencies: (1) they are increased and (2) they become amplitude dependent. The loaded car yaw mode illustrates these

TABLE 7-3
COMPARISON OF SPRING DATA TO AAR FLEXIBLE BODY MODELS

		BARBER	RIDE CONTROL	FRATE	LOVE&HUSSAIN REF [12]	TSE & MARTIN REF. [11]
Center Plate	Vertical	230	303	N/A	2120	2120
Suspension	Vertical	28.5	29.2	24.0	25.78	24.97
Suspension	Lateral	24.8	14.3	13.8	12.1	12.5
Side Frame (Track)	Vertical	647	650	91	250	208
Side Frame (Track)	Lateral	80	513	95	167	167

Note: All values are for half truck.

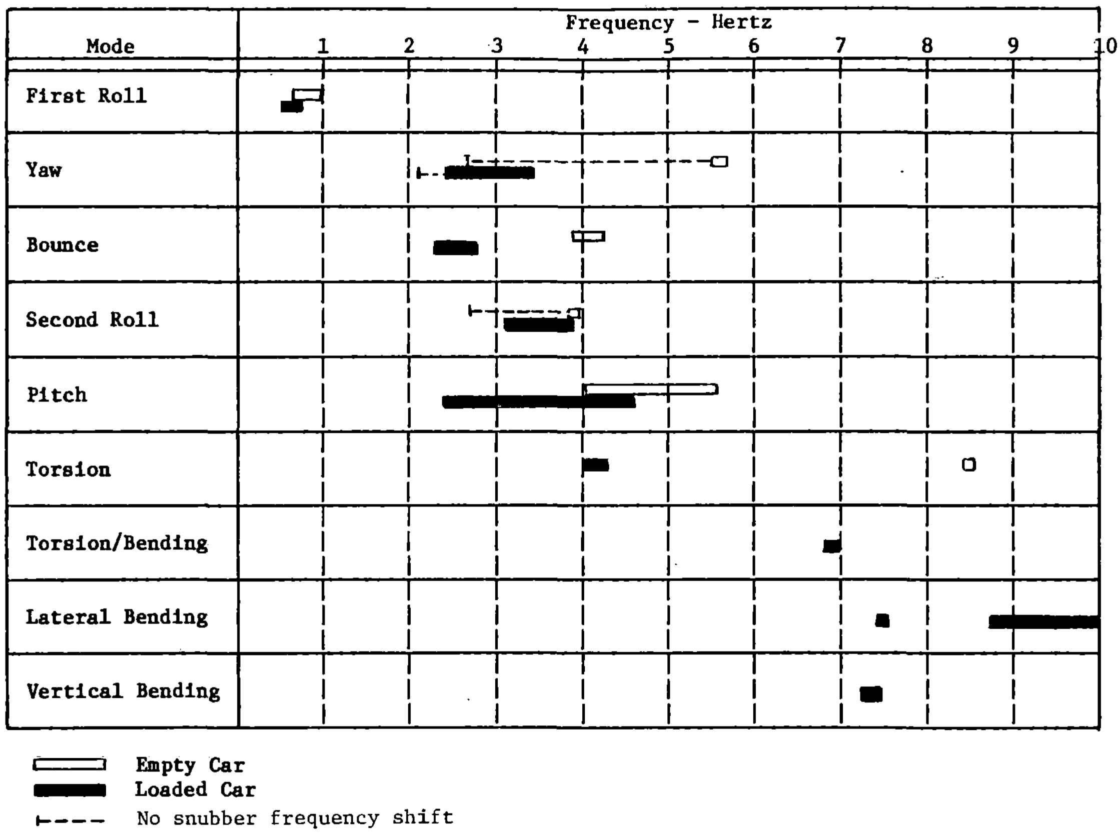


FIGURE 7-2
RANGE OF MODAL FREQUENCIES FOR BASE VEHICLE NO. 2

two effects as shown in Figure 7-2. The frequency shifts from 2.1 Hertz without snubbers to 3.5 Hertz with snubbers and small amplitudes and then will shift downward again toward the no-snubber frequency as the amplitude is increased.

A summary of vehicle response in the first roll mode from the modal testing (sinusoidal input) is presented in Figure 7-3 showing the maximum roll angle response of the vehicle body as a function of cross level input. Vehicle No. 1 is seen to have the largest response, the most likely cause being its c.g. height of 97.0 inches compared to 89.2 inches for the equivalent Vehicle No. 2 configuration (i.e., B2).

After Vehicle No. 1, the configuration order of decreasing response for Vehicle No. 2 is as follows:

- RC2 - Ride Control trucks with constant column friction force of about 3.5 kip
- B4 - Barber truck with single side springs and friction force of about 4.4 kip
- B7 - Barber truck with double side springs having friction 7.5 kip or greater but also with constant contact resilient side bearings
- B2 - Barber truck with double side springs
- B3 - Barber truck with double side springs but with 72.4% load and lower c.g. (81.6 inches)
- B5 - Barber truck with single side springs and auxiliary hydraulic snubbers

There are three general trends that can be deduced from this data:

- Increase in snubber force results in improved harmonic roll performance at large cross level inputs but poorer performance at small input level.
- Constant contact resilient side bearings causes poorer harmonic roll performance because of increased tendency for center plate lift.

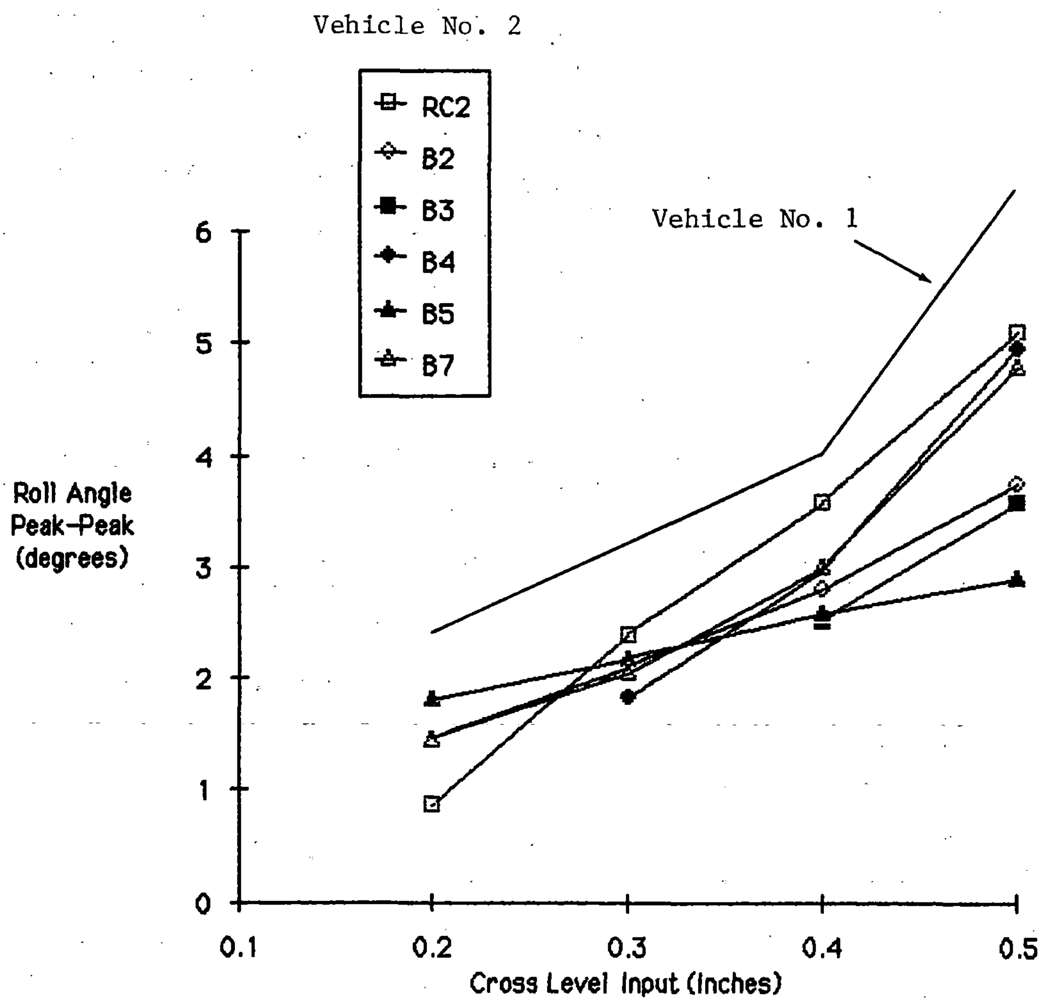


FIGURE 7-3
FIRST ROLL MODE MAXIMUM RESPONSE, COMPARISON BY CONFIGURATION

- Lower center of gravity improves harmonic roll performance with respect to car body roll angles. However, reduced load conditions results in earlier wheel lift or center plate lift.

7.3 Staggered Rail Performance

Staggered rail performance was measured in terms of the amplitude of cross level that could be used on the Vibration Test Unit before a limit condition, wheel lift or center plate lift, was reached. These limits are not derailment conditions but are, rather, first indicators that derailment conditions are being approached. They are for this reason considered as undesirable conditions but they are also undesirable in that they result in increased loads on both track and vehicle and will in the least case accelerate wear and in the extreme cause component failures. A summary of results is given in Table 7-4 with the staggered rail cross level of 0.75 inches used as the dividing line between acceptable and undesirable performance. That is, a configuration is acceptable if it can pass 0.75 inches without a limit condition being reached.

The following configurations are the best performers:

- B5 - Loaded Vehicle No. 2 with Barber S-2-C trucks, reduced friction and auxiliary hydraulic snubbers,

- B2 - Loaded Vehicle No. 2 with Barber S-2-C trucks with inner and outer side springs, and

- B4 - Loaded Vehicle No. 2 with Barber S-2-C trucks with reduced friction (outer side springs only).

The use of constant contact resilient side bearings, Configuration B7, resulted in degradation of performance to marginal conditions. The use of these side bearings actually resulted in better performance as to wheel lift with center plate lift being the limit condition experienced.

Off loading the vehicle resulted in degradation of performance. Both the empty and 73.4% load condition were shown to have undesirable performance.

The performance of Vehicle No. 1 was found to be undesirable in the configuration tested which is equivalent to B2 in Vehicle No. 2 (fully loaded with the Barber S-2-C truck with inner and

TABLE 7-4
SUMMARY OF STAGGERED RAIL HARMONIC ROLL
RESULTS FROM VTU TESTING

CONFIG. - VEHICLE	CROSS LEVEL	CROSS LEVEL & ALIGNMENT	CROSS LEVEL & SUPER ELEVATION
B1 - No. 2	0.60[1]	0.50[1]	0.60[1]
B2 - No. 1	0.70	____[2]	____[2]
RC2 - No. 2	0.70	0.65	0.60
B3 - No. 2	0.73	0.60	0.70
	///////		//////////////////
B7 - No. 2	0.80[1]	0.70[1]	0.80[1]
		/////////////	
B4 - No. 2	0.85	0.75	(0.75)[3]
B2 - No. 2	0.90	0.80	(0.75)[3]
B5 - No. 2	0.90	> 0.80	(0.80)[3]

[1] center plate lift
[2] not tested
[3] parenthesized cross levels are extrapolated estimates

///////// Performance divider: acceptable if 0.75-inch or greater cross level is reached; undesirable if limit condition is experienced at cross levels less than 0.75 inches.

outer side springs and roller side bearings). There are two differences between the two vehicles that would make Vehicle No. 1 the poorer performer in harmonic roll: (1) it has a higher center of gravity, 97.0 inches compared to 89.2 inches for the test configurations, and (2) its truck center spacing is closer to the 39-foot rail length used in the staggered rail testing, 40.167 feet for Vehicle No. 1 compared to 45.75 feet for Vehicle No. 2.

The combination of staggered rail cross level variation with super elevation and cross level with alignment variations both resulted in degradation of performance.

7.3.1 Conclusions Drawn from Staggered Rail Testing

The following conclusions are based on the results of the VTU staggered rail testing with respect to the safety performance of the base 100-ton covered hopper car.

1. Light and partial load vehicles will reach wheel lift or center plate lift conditions at lower amplitudes of cross level and consequently have reduced safety margins relative to the fully loaded vehicle.

2. Increased friction snubber force enables vehicle to negotiate larger cross level, staggered rail without wheel lift. Optimum performance is achievable with a combination of friction and hydraulic snubbing.

3. Constant contact resilient side bearing results in improved performance, approaching the friction and hydraulic snubber case, with respect to reducing body roll angle and retarding the onset of wheel lift. However, these benefits are negated by an increased tendency for the center plate to lift off.

4. Increased side bearing clearance will yield slight loss in safety performance.

7.3.2 Recommendations

- The poor performance of empty and partially-loaded 100-ton covered hopper cars suggests two areas of investigation: (1) review and reformulate train handling rules and procedures with respect to empty and off-loaded vehicles and (2) do a study to find explanation of performance degradation. With lower center of gravity, improved performance was anticipated.

- Recommend that the choice of truck snubbers be made on the basis of studies to find optimum properties. Avoid over-snubbing.

7.3.3 Track and VTU Test Comparisons for Staggered Rail

The track tests were performed on Vehicle No. 2 in the RC2 configurations and results are reported in Reference [2]. Summary comparisons of tangent, perturbed track tests to Vibration Test Unit test results are shown in Figures 7-4 and 7-5 and are discussed here.

In Figure 7-4, track test results are compared to VTU test results with four different rail profiles (refer to Section 6.5.2 and Figure 6-28 for description of these four profile shapes). The following observations are made in the track to VTU comparison:

- Critical speeds are within 0.5 miles per hour.

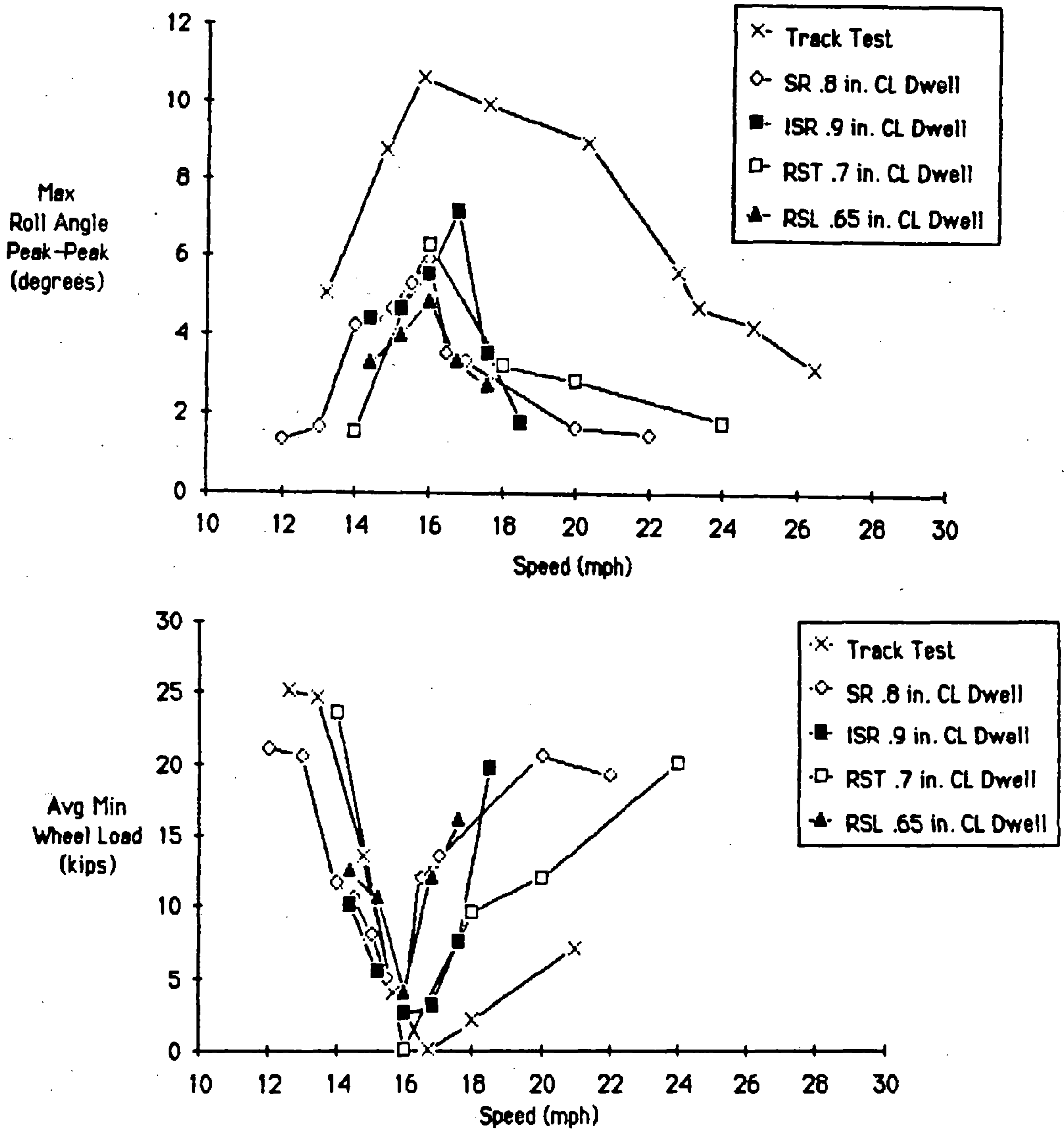
- Carbody roll angles are greater in the track test: i.e., 10.5 degrees compared to 5 to 7 degrees peak-to-peak.

- The critical cross level for the VTU tests were both greater and less than the 0.75-inch track perturbations. The "shimmed rail" profile, which was the shape closest to the perturbed track, went to 0.80 inches cross level input before reaching zero wheel load. The rectified sine with lateral, the shape believed to be most representative of actual track staggered rail profile, reached zero wheel load at 0.65 inches.

There are two causes that can be hypothesized as the source of difference in results between track and VTU results: drag loads on the test car, and track stiffness.

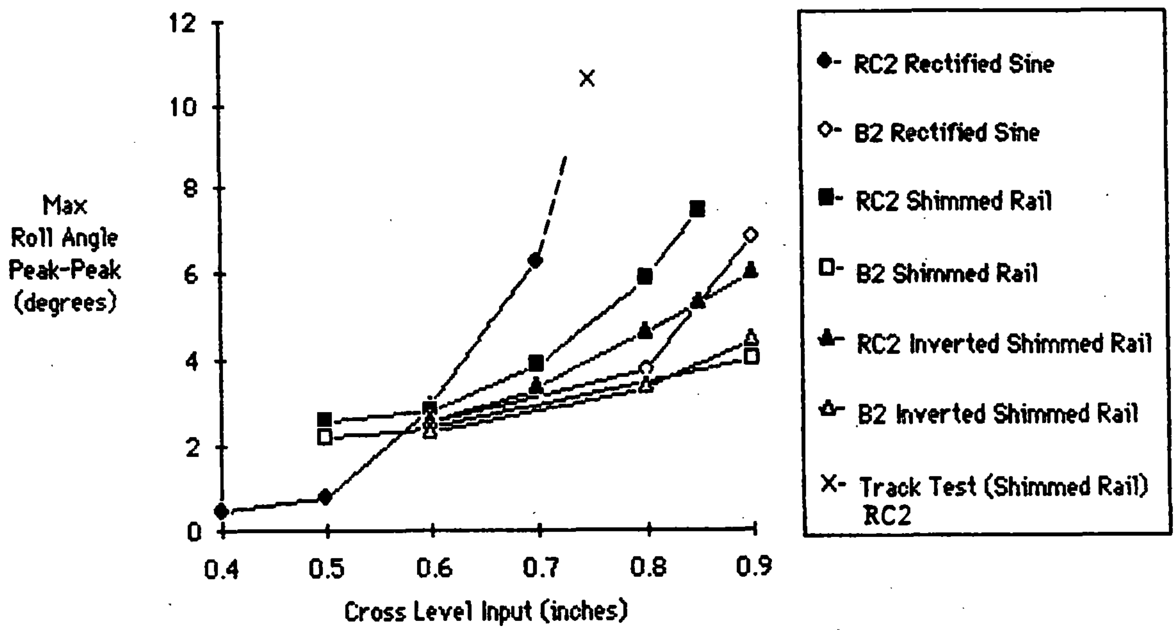
The track test procedure was to accelerate to the desired speed and enter the test section with essentially slack conditions. On the VTU there was a constant tension load of about 45,000 pounds on the couplers. It is expected that larger roll angles and earlier wheel lift would be experieced with the slack conditions.

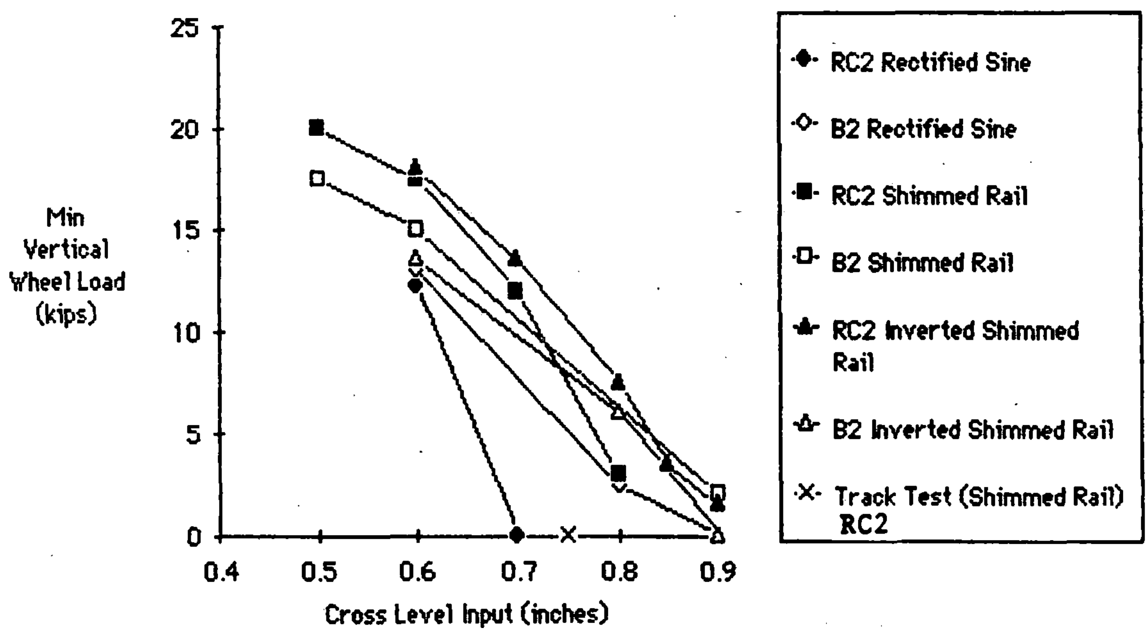
The VTU simulates track geometry variations at the wheel-rail interface and consequently does not account for any flexibility that exists in the track. The track flexibility has the effect



Track test data from Reference [2]
CL = Cross Level
SR = Shimmed rail, constant speed VTU test
ISR = Inverted shimmed rail, constant speed VTU test
RST = Rectified sine tangent track, constant speed VTU test
RSL = Rectified sine with lateral input, constant speed VTU test

FIGURE 7-4
RC2 RESPONSE TO STAGGERED RAIL, COMPARISON OF TRACK TEST TO SEVERAL PROFILES USED IN VTU





RC2 Rectified sine and Track test constant speed
All other speed slow-down tests

FIGURE 7-5
RC2 AND B2 RESPONSE VS. CROSS LEVEL
COMPARING TRACK TEST AND SEVERAL VTU PROFILES

of amplifying track perturbations under resonance conditions. That is, larger perturbations are needed in a rigid track to force this same harmonic roll response in a flexible track.

Several conclusions can be drawn:

- Track and VTU results are close and differences have been explained although further testing is needed to confirm the validity of these hypothesized explanations.

- VTU harmonic roll tests should be run with no load or buff load at the couplers since this represents worst case conditions.

- A staggered rail profile that has a downward cusp at the rail joints will probably result in more severe harmonic roll conditions than the profile shape used with staggered rail perturbed track. Based on VTU results, 0.60-inch cross level with downward cusp (rectified sine) staggered rail profile would result in the same responses as the 0.75-inch perturbed track.

7.4 Bounce Test Results

The purpose of the bounce testing was to obtain a measure of the 100-ton covered hopper car response to track hard spots such as grade crossings. The critical speed was found to be between 60 and 65 miles per hour. However, there were no responses measured that would indicate a safety problem.

In comparing the several configurations tested, the Ride Control truck, Configuration RC2, had the largest responses and the Barber truck with reduced friction and auxiliary hydraulic snubbers, Configuration B2, had the smallest responses. This excludes the empty condition where the critical speed was above 80 miles per hour.

7.5 Cross Level Index Evaluation

The CLI evaluation consisted of VTU simulation of several patterns of track cross level variation wherein the amplitude of each pattern was increased to the point of wheel lift. In the analysis of the data, the CLI of each critical condition would then be determined and added to the family of data from which the critical value of CLI would be extracted. This data analysis has been performed by the Transportation Systems Center and the results have been presented in [6].

For the purposes of this report, the results have been viewed from the perspective of how harmonic roll response is affected by different profile shapes but primarily as another measure of safety performance of the 100-ton covered hopper car. The findings have been presented in Sections 5 and 6 and are in agreement with the summary of staggered rail performance presented above.

7.6 Sine Wave Track Geometry Testing

The sine wave track geometry testing performed on three configurations of Vehicle No. 2 consisted of simulation of 39-foot staggered rail cross level variations in combination with 62.4-foot and 39-foot alignment variations, with sinusoidal wave forms used for both profile and alignment. Speed and amplitudes of cross level and alignment were varied to find the limit conditions of wheel lift or center plate lift. The purpose of the testing was to obtain another measure of the 100-ton covered hopper car for harmonic roll performance.

A summary of results is given in Table 7-5 in terms of cross level and alignment amplitudes of track geometry variations that result in limit conditions. The limit conditions encountered were center plate lift for the empty (B1) configurations and wheel lift for the loaded configurations, RC2 and B2.

Comparing these results with Table 7-4, where the rectified sine wave form was used for the staggered rail simulation, shows good agreement with slightly larger cross level inputs reached with the sine wave testing.

7.7 Yaw Test Results

The yaw tests were an attempt to duplicate motions and loads that result from a body hunting condition. The results were expected to indicate the frequencies at which body hunting will occur and a general indication of the severity of the motions.

The yaw frequency was found to fall between 2.5 and 3.0 Hertz. The maximum conditions tested, which were with a lateral sinusoidal input of 1.20 inches peak-to-peak, resulted in lateral carbody accelerations of about 0.28 g RMS which is representative of a full flange-to-flange hunting condition. Lateral wheel loads reached maximum values in excess of 40 kips with maximum L/V values of about 0.90.

TABLE 7-5
SUMMARY OF SINE STAGGERED RAIL TESTS, VEHICLE NO. 2

CONFIGURATION	CROSS LEVEL INPUT AT LIMIT CONDITION, INCHES				
	ALIGNMENT INPUT				
		62.4-FOOT			39-FOOT
	0	.5 IN.	1.0 IN.	1.5 IN.	.5 IN.
B1	.74	.74	.74	.56	.5
RC2	.74	.93	.93	.93	.56
B2	>.93	.93	.93	.74	.74

Limit conditions: B1 - center plate lift
RC2 - wheel lift
B2 - wheel lift

The only conclusion that can be drawn is that it is possible to create motions of the fright car that are representative of on-track hunting conditions and that the VTU can be a useful tool in studying loads and motions caused by body hunting.

7.8 Conclusions

The objectives of the laboratory tests performed on the two base 100-ton covered hopper cars have been to measure physical properties that control or influence dynamic response; to measure dynamic response characteristics; to measure response to certain track conditions; to determine the effects of configuration variations; to identify conditions and configurations that result in desirable and undesirable performance from a safety standpoint; and, finally, to present this data in a form that can be used for a basis for comparison to prototype covered hopper cars and for general reference. It is concluded that these objectives have been met with the results presented in detail in the body of this report and summarized within this section.

APPENDIX

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