U.S. Department of Transportation

Federal Railroad Administration

SAFETY MARGIN TESTING OF A 70-TON BOXCAR WITH SHIFTED PLYWOOD LADING

Office of Research and Development
Washington DC 20590

SUMMARY RESULTS

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DOT/FRA/ORD-84/15.01

September 1984
Final Report

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
DOT/FRA/ORD-84/15.01		
4. Title and Subtitle Safety Margin Testing of a 70-Ton Boxcar with Shifted Plywood Lading: Summary Results		5. Report Date September 1984
		6. Performing Organization Code
7. Author(s) George Kachadourian		8. Performing Organization Report No. 84W00003
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. P-83-C-00657
12. Sponsoring Agency Name and Address U.S. Department of Transportation Federal Railroad Administration Office of Research and Development 400 Seventh Street, S.W. Washington, D.C. 20590		13. Type of Report and Period Covered Final Report
		14. Sponsoring Agency Code

15. Supplementary Notes

*Under contract to: Association of American Railroads
Transportation Test Center
Pueblo, Colorado 81001

DTFR53-82-C-00282 TO #9

16. Abstract

Tests were performed at the Transportation Test Center, Pueblo, Colorado, in a joint Association of American Railroads (AAR) and Federal Railroad Administration (FRA) effort in the April-May-June period of 1982. AAR objectives were to perform a "pilot" test on the Vibration Test Unit (VTU) as a demonstration that the facility could meet their requirements for lading damage prevention testing. Pilot test results have been reported by the AAR.

The FRA performed "Safety Margin Tests" of a plywood lading that is suspected of being a derailment cause factor, with the objectives of determining conditions that cause the plywood to shift and the effect of the shifted plywood on derailment margins. Volume 1 presents a summary of the safety margin tests performed and test results. Volume 2 contains the data from which the summary was extracted.

17. Key Words	18. Distribution Statement
Boxcar Vibration Testing Boxcar Dynamics Lading Dynamics Freight Car Truck Data	Document is 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

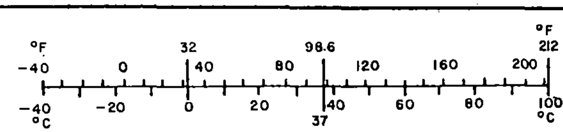

TABLE OF CONTENTS

		Page
1.	INTRODUCTION	1-1
2.	OBJECTIVES	2-1
3.	SAFETY MARGINS TEST DESCRIPTIONS	3-1
	3.1 Test Vehicle	3-1
	3.2 Test Procedures	3-8
	3.3 Instrumentation and Data Acquisition	3-11
	3.4 Data Output	3-11
4.	RESULTS SUMMARIZED	4-1
	4.1 Lading Shift Threshold	4-1
	4.2 Wheel Lift Thresholds	4-8
	4.2.1 Staggered Rail Test Results: Tangent Track	4-9
	4.2.2 Staggered Rail Test Results: Superelevated Track	4-22
	4.3 Track Geometry Response Data	4-35
5.	CONCLUSIONS	5-1
APPENDIX:	LIST OF REFERENCES	A-1

LIST OF ILLUSTRATIONS

		Page
TABLE 3-1:	DOTX 503 BOXCAR BASIC DATA	3-3
TABLE 3-2:	TRUCK CONFIGURATIONS, SAFETY MARGIN TESTING	3-6
TABLE 3-3:	TEST MATRIX	3-9
TABLE 3-4:	TEST MEASUREMENTS CHANNEL IDENTIFICATION AND DESCRIPTION	3-12
TABLE 3-5:	SAFETY MARGIN TESTING--RECORD FORMAT	3-13
TABLE 4-1:	PLYWOOD LADING LATERAL SHIFT DURING TRACK GEOMETRY TESTING	4-3
TABLE 4-2:	PLYWOOD LADING LATERAL SHIFT DURING HARMONIC ROLL TEST	4-4
TABLE 4-3:	TRUCK CONFIGURATIONS TESTED	4-10
TABLE 4-4:	RESULTS COMPARISON GROUPS - STAGGERED RAIL DATA	4-11
TABLE 4-5:	SUMMARY RESULTS OF SUPERELEVATED STAGGERED RAIL TEST	4-23
FIGURE 3-1:	DOTX 503, 70-TON BOXCAR ON THE VIBRATION TEST UNIT, SAFETY MARGIN TESTS	3-2
FIGURE 3-2:	PLYWOOD STACK ARRANGEMENT AND INSTRUMENT LOCATION, 70-TON BOXCAR SAFETY MARGIN TESTS	3-4
FIGURE 3-3:	TRUCK SPRING AND SNUBBER CONFIGURATIONS	3-5
FIGURE 3-4:	FORCE OUTPUT RATES OF RAILROAD DYNAMICS, INC. MDA CONTROL/MASTER HYDRAULIC SNUBBERS	3-7
FIGURE 4-1:	CONFIGURATION 1: CARBODY ROLL ANGLE AND LADING SHIFT IN STAGGERED RAIL TESTS	4-5
FIGURE 4-2:	CONFIGURATION 2: CARBODY ROLL ANGLE AND LADING SHIFT IN STAGGERED RAIL TESTS	4-6
FIGURE 4-3:	CONFIGURATION 2 WITH 3 INCH SUPERELEVATION, CARBODY ROLL ANGLE AND LADING SHIFT IN STAGGERED RAIL TEST	4-7
FIGURE 4-4:	CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1 AND 2	4-12
FIGURE 4-5:	WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1 AND 2	4-13
FIGURE 4-6:	CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 3, 2 AND 6	4-16
FIGURE 4-7:	WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS, 1, 3, 2 AND 6	4-17
FIGURE 4-8:	CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 4, 5 AND 7	4-18
FIGURE 4-9:	WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 4, 5 AND 7	4-19

LIST OF ILLUSTRATIONS
(Continued)

		Page
FIGURE 4-10:	CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 3, AND 4	4-20
FIGURE 4-11:	WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 3, AND 4	4-21
FIGURE 4-12:	CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 2, 5, AND 8	4-24
FIGURE 4-13:	WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 2, 5, AND 8	4-25
FIGURE 4-14:	CONFIGURATION 1, STAGGERED RAIL TESTS WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-26
FIGURE 4-15:	CONFIGURATION 2, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION	4-27
FIGURE 4-16:	CONFIGURATION 2, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-28
FIGURE 4-17:	CONFIGURATION 3, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-29
FIGURE 4-18:	CONFIGURATION 4, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-30
FIGURE 4-19:	CONFIGURATION 5, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, SHIFTED LADING	4-31
FIGURE 4-20:	CONFIGURATION 6, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-32
FIGURE 4-21:	CONFIGURATION 7, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-33
FIGURE 4-22:	CONFIGURATION 8, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED	4-34
FIGURE 4-23:	TYPICAL INPUT DISPLACEMENT PSD SPECTRA TRACK GEOMETRY RUNS	4-36
FIGURE 4-24:	TYPICAL RESPONSE ACCELERATION PSD SPECTRA TRACK GEOMETRY RUN 50	4-37
FIGURE 4-25:	SUMMARY PSD DATA, PEAK VERTICAL RESPONSE, 0.9 - 1.1 HERTZ	4-38
FIGURE 4-26:	SUMMARY PSD DATA, PEAK VERTICAL RESPONSE, 2.0 - 2.5 HERTZ	4-39
FIGURE 4-27:	SUMMARY PSD DATA, PEAK VERTICAL RESPONSE, 8.5 - 9.5 HERTZ	4-40
FIGURE 4-28:	SUMMARY PSD DATA, PEAK VERTICAL RESPONSE, 14 - 16 HERTZ	4-41
FIGURE 4-29:	SUMMARY PSD DATA, PEAK LATERAL RESPONSE, 0.9 - 1.1 HERTZ	4-42
FIGURE 4-30:	SUMMARY PSD DATA, PEAK LATERAL RESPONSE, 2.0 - 2.5 HERTZ	4-43

LIST OF ILLUSTRATIONS
(Concluded)

		Page
FIGURE 4-31:	SUMMARY PSD DATA, PEAK LATERAL RESPONSE, 8.5 - 9.5 HERTZ	4-44
FIGURE 4-32:	SUMMARY PSD DATA, PEAK LATERAL RESPONSE, 14 - 16 HERTZ	4-45
FIGURE 4-33:	TRACK GEOMETRY RESPONSE SUMMARY, RMS ACCELERATION ESTIMATED FROM PSD PEAKS	4-47

1. INTRODUCTION

A cooperative test effort between the Association of American Railroads (AAR) and the Federal Railroad Administration (FRA) was conducted at the Transportation Test Center (TTC), Pueblo, Colorado in the April-May-June period of 1982. The test vehicle used was the DOTX 503, which is a 50-foot 70-ton boxcar. The testing performed was primarily on the Vibration Test Unit (VTU) but also included longitudinal impacting and over-the-road measurements. The AAR test requirement, presented in Reference 1, and the FRA requirements, presented in Reference 2, were incorporated into the Implementation Plan, Reference 3.[1] The AAR objectives were primarily concerned with evaluation of the performance and operation of the VTU and to determine the feasibility of using the VTU as a rail vehicle simulator for damage prevention testing. Results of the AAR tests are contained in References 4 and 5.

The FRA objectives centered on the investigation of derailment cause and prevention using a boxcar and lading configuration that has been a suspected contributing cause to several derailments. The FRA tests were performed between June 8 and June 30, 1982; they have been identified as Safety Margin Testing, and are the subject of this report.

MITRE involvement in these tests has been in support of both the AAR and FRA phases of testing. This report, however, covers only the Safety Margin Testing conducted at the Transportation Test Center's Rail Dynamics Laboratory (RDL).

A review of freight car derailment statistics has shown that the combination of curved track with out-of-specification low joints and boxcars with plywood lading has been frequently involved in derailments.[2] It is suspected that in these cases the plywood lading had shifted laterally and was a major contributor to the cause of derailment.

The typical boxcar has an internal width of 114 inches; thus there is a total lateral clearance of 18 inches with 4 x 8 foot plywood. AAR loading specifications require that the plywood

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(1) The List of References can be found at the end of this report.

(2) Discussions with members of the Association of American Railroads (AAR) Subcommittee on Freight Claim and Damage Prevention; Tom Schoenleben, Chessie, Baltimore, Md., Harry Grosso, AAR, and Peter Kiliani, CONRAIL.

be placed in the center (laterally) and that longitudinal wedging be effected with wooden spacers between each adjacent stack of plywood. A longitudinal load between the plywood stacks is to be applied using air bags and held with wooden wedges. The objective of this AAR requirement is to minimize longitudinal dynamics and reduce the tendency for lateral shifting of the plywood.

The objectives of the Safety Margin Testing of this report are first to determine the threshold of track variations that will cause the plywood lading to shift and then to determine what track variations will result in wheel lift. The effect of various truck suspension systems, including hydraulic snubbers, on the response of the carbody and lading and the margin of derailment conditions were investigated.

This working paper is in two volumes: Volume 1 is the basic report with summarized test data; Volume 2 contains the base test data and plots used in developing the final summary data.

2. OBJECTIVES

A review of freight car derailment statistics has shown that a combination of curved track with out-of-specification low joints and boxcars with plywood lading frequently has been present when derailments have occurred. It is suspected that in these cases the plywood lading has shifted laterally and has been a major contributor to the cause of derailment. The objectives of the Safety Margin Testing described in this report were first to determine the threshold of track variations that will cause the plywood lading to shift, and then to determine what track variations will result in wheel lift. In order to accomplish these objectives the following four test objectives were defined:

1. Measure response of car and lading to various track input.

   - nominal track geometry (TG)
   - increased levels of TG variation
   - staggered joint bolted rail
   - superelevated rail with alignmemt wave lengths to excite roll and yaw

2. Determine track input types and magnitudes that cause lading shift.

3. Determine derailment margins for track conditions tested with centered and shifted lading.

4. Assess beneficial effects of hydraulic snubbers.

3. SAFETY MARGIN TEST DESCRIPTIONS

Safety margin tests required vibration testing of the DOTX 503 70-ton boxcar using the Vibration Test Unit (VTU) at the TTC RDL facility. The boxcar lading consisted of 48.25 tons of plywood with test configurations including both centered and laterally shifted lading. The VTU was used to simulate track conditions and to determine the threshold levels that result in lading shift and in wheel zero vertical load. This section describes the boxcar configurations tested, the test set up, data measurement and data playback.

3.1 Test Vehicle

The test configuration, shown in the photograph of Figure 3.1, consisted of the DOTX 503 70-ton boxcar loaded with 96,500 pounds of 1/2 inch CD grade fir plywood in 4 x 8 foot sheets. The vehicle is a 50-foot, high cube boxcar with dimensions as shown in Table 3-1.

The plywood was steel strapped in 33 inch high bundles with wooden 2 x 4 skids. The bundles were stacked three high in the arrangement shown in Figure 3-2 to a total height of approximately 102 inches in 12 stacks. Loading, centering and bracing was done according to the AAR specifications called out in Reference 5. This loading spec requires the plywood to be centered laterally in the boxcar, and braced so as to restrain fore and aft motion. The plywood could, and did, shift up to nine inches laterally from the centered position. Tests were conducted in both the centered and shifted conditions of the lading. The plywood was returned to its centered position through the use of air bags.

The test vehicle had Barber S-2-C trucks with load variable friction snubbers and 33 inch wheels. The basic spring nest makeup is shown in Figure 3-3. Eight spring nest configurations were tested, including the use of hydraulic snubbers and variations on the friction snubber force. These eight configurations are defined in Table 3-2. Two sets of hydraulic snubbers were used, both manufactured by Railroad Dynamics, Inc., and identified as MDA CONTROL/MASTER high force and low force units. The rated force output of these units is shown in Figure 3-4.

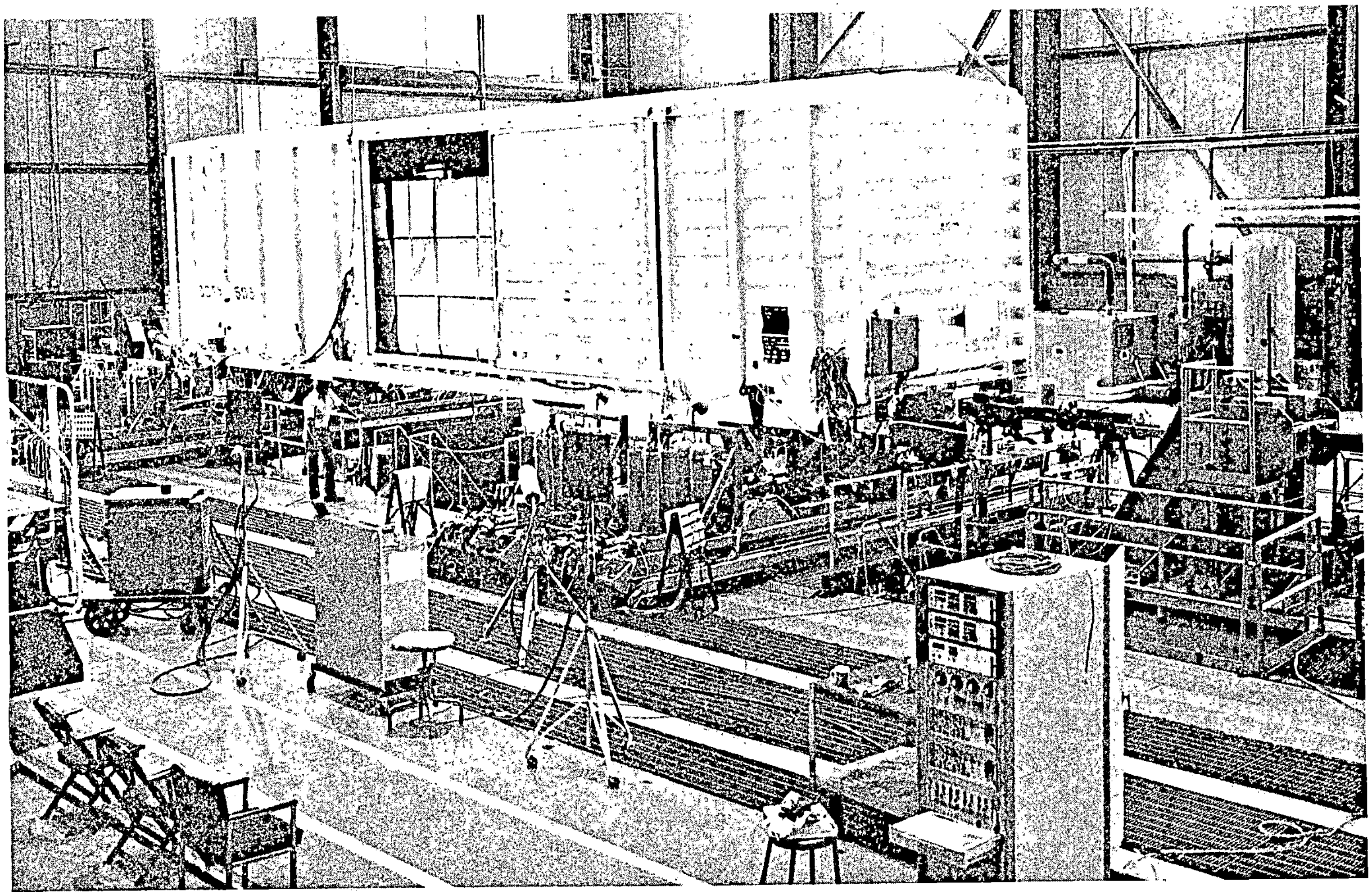


FIGURE 3-1
DOTX 503, 70-TON BOXCAR ON THE VIBRATION TEST UNIT, SAFETY MARGIN TESTS

TABLE 3-1
DOTX 503 BOXCAR BASIC DATA

BOXCAR WEIGHT C.G. DATA

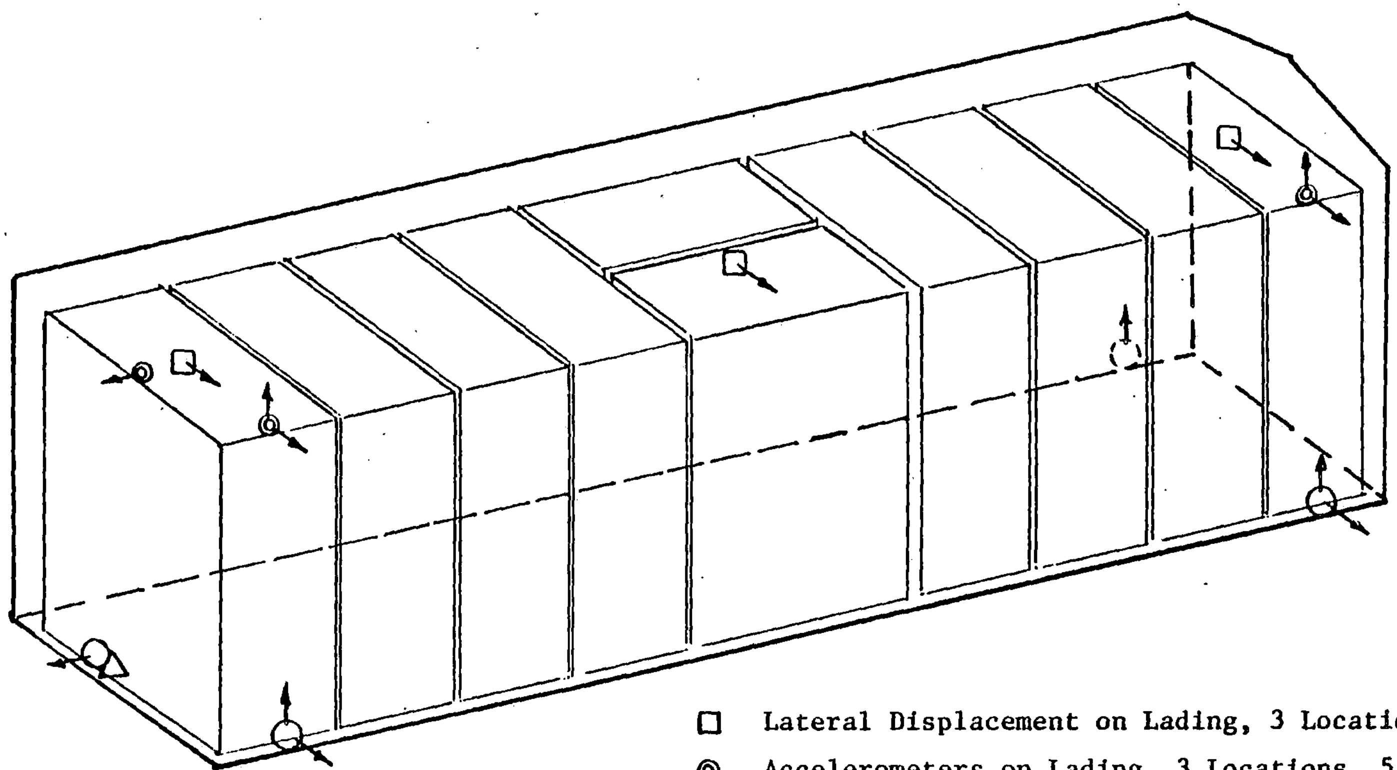
Item	Weight (lbs.)	C.G.[1] (in.)
Empty Car	61,600	53.00
Plywood	96,500	94.50
Total	158,100	78.33

[1]C. G. - center of gravity measured from top of rail

BOXCAR DIMENSIONAL DATA

Inside Dimensions: length = 50 feet
width = 9 feet 7 inches
height = 11 feet
volume = 5300 cu. ft.

Truck Spacing: 40.83 feet



□ Lateral Displacement on Lading, 3 Locations

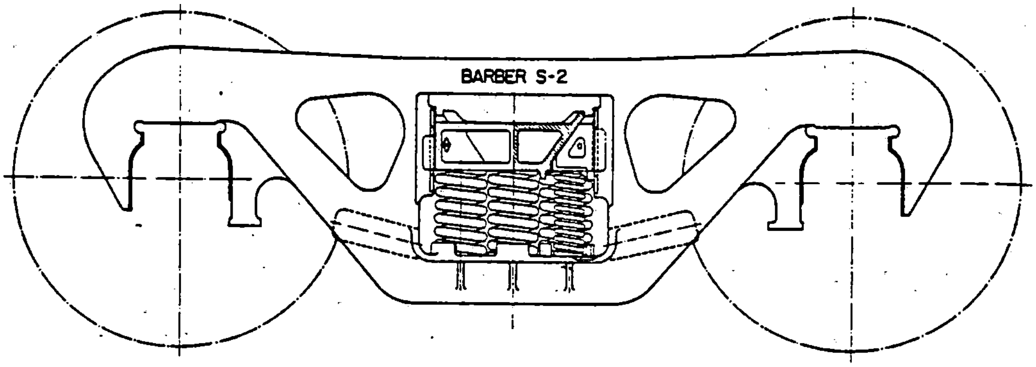
◎ Accelerometers on Lading, 3 Locations, 5 Accels.

○ Accelerometers on Carbody, 4 Locations, 6 Accels.

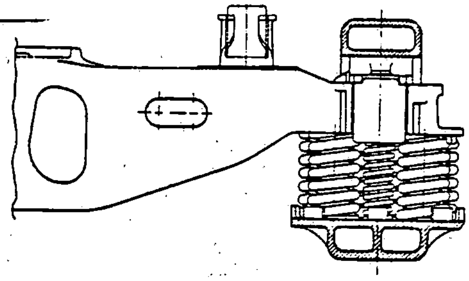
△ Roll Gyro on Carbody

FIGURE 3-2
PLYWOOD STACK ARRANGEMENT AND INSTRUMENT LOCATION,
70-TON BOXCAR SAFETY MARGIN TESTS

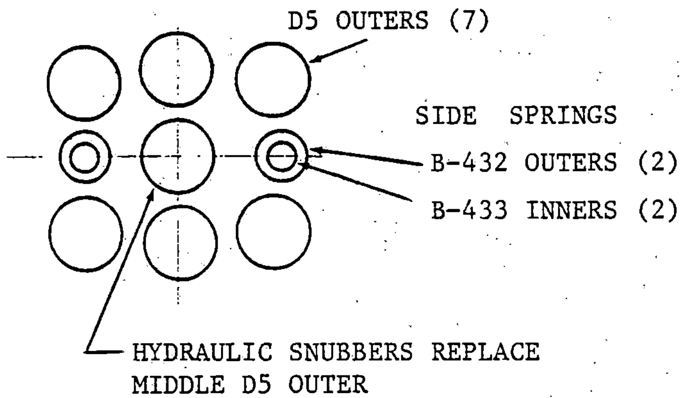
SIDE VIEW



END VIEW



SPRING NEST CONFIGURATION



NOTE: Snubber variations shown in Table 3-2.

FIGURE 3-3
TRUCK SPRING AND SNUBBER CONFIGURATIONS

TABLE 3-2
TRUCK CONFIGURATIONS, SAFETY MARGIN TESTING

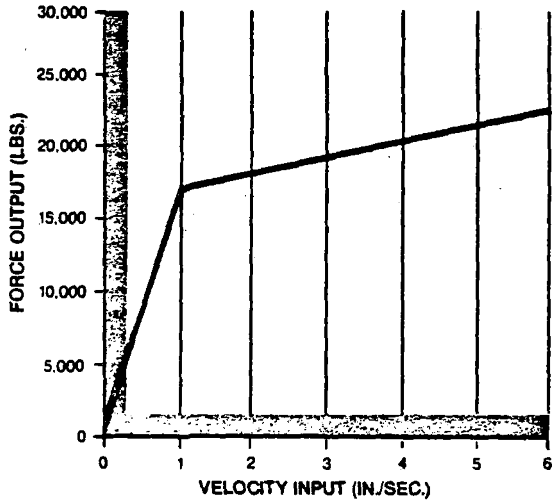
CONFIGURATION NUMBER	FRICTION(1) SNUBBERS	HYDRAULIC(2) SNUBBERS	SPRING RATES PER NEST(3) (lb./in.)
1	High	None	18,000
2	Low	Low	15,000
3	Low	High	15,000
4	High	Low	16,000
5	High	High	16,000
6	None	Low	13,000
7	None	High	13,000
8	Low	None	15,000

(1) Friction snubber condition:
High = with inner and outer side springs
(force equal about 5,000 lb./spring nest)
Low = with outer side spring only
(force equal about 2,800 lb./spring nest)

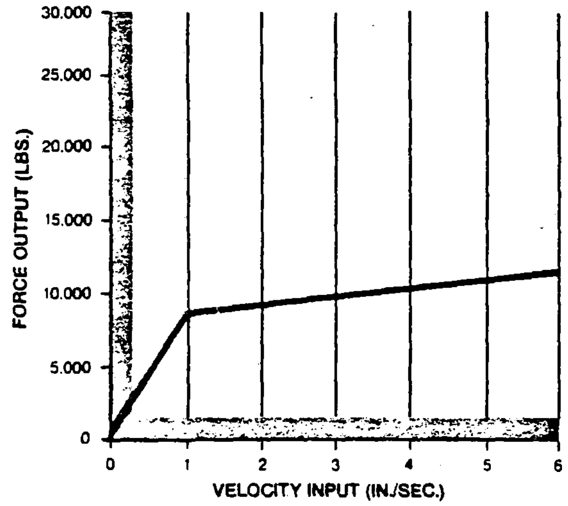
(2) Hydraulic snubbers. See Figure 3-4 for force rates.

(3) Spring rates per spring nest based on the following values:

D5 outers: 2,140 lb./in. each
B-432 Side outers: 984 lb./in. each
B-433 Side inners: 439 lb./in. each



HIGH FORCE
UNIT



LOW FORCE
UNIT

FIGURE 3-4
FORCE OUTPUT RATES OF RAILROAD DYNAMICS INC.
MDA CONTROL/MASTER HYDRAULIC SNUBBERS

3.2 Test Procedures

The safety margin testing involved duplicating two track conditions on the Vibration Test Unit. One was track geometry data measured on test track at the Transportation Test Center using the LTHD measurement system (Reference 4). The second condition was the simulation of a staggered joint, bolted rail profile.

A 5.5 minute section of the Track Geometry (TG) profile and alignment space curves from 30 mile per hour recordings of a 2.75 mile section of the TTC FAST loop was selected for the TG testing. This section of data was used for all TG tests at four levels, 1.25, 1.50, 1.75 and 2.00 times measured amplitude. However, it was found that one 1.75 run and all 2.00 runs were invalid due to numerical inconsistencies in the multiplication and analog to digital conversion process.

The test procedure followed was to perform each test, record response measurements, and monitor the real time strip chart recordings. The tests performed are listed in the Test Matrix of Table 3-3. The track geometry tests did result in lateral shifting of the plywood but not in wheel lift.

The low joint staggered rail condition that produces the harmonic roll phenomenon was simulated with a rectified sine wave. The right and left profile were phased so that the right rail was at a maximum at the time the left was at its low point and vice versa. The time phasing between axles was determined on the basis of axle and truck spacing, and the track speed that was being simulated. The procedure used was to perform constant cross level runs with the frequency of the generating sine wave varied from 0.65 to 0.20 Hertz. This corresponds to a speed slow down from 35 to 11 miles per hour using 39 foot rail. Successive runs were made, increasing the cross level with each new run until the lading started to shift from the centered position or until wheel lift occurred in the lading shifted condition. Table 3-3 is a matrix showing all successful runs made.

The track equivalent of this simulation is a slow down from 35 to 11 miles per hour over a length of track of 3.84 miles in 9.5 minutes, an average slow down rate of 0.0.42 mph/sec. An alternate test procedure would have been to perform constant speed runs in 1 mph steps with 10 rail lengths of the staggered rail low joints. The results would have been essentially the same for the one deceleration run and the 24 constant speed runs.

TABLE 3-3
TEST MATRIX

CONFIG.*	INCREASED TRACK GEOMETRY RUN/SCALE FACTOR		STAGGERED RAIL RUN/AMPLITUDE (INCHES)		SUPERELEVATED RAIL NORTH SIDE HIGH STAGGERED RAIL RUN/AMPLITUDE (INCHES)	
	CENTERED	SHIFTED	CENTERED	SHIFTED	CENTERED	SHIFTED
1	49/1.25	56/1/25	54/0.2	60/0.2		83/0.2
	50/1.50	57/1.50	55/0.4	61/0.4		84/0.3
	52/1.75#	58/1/75		62/0.6		85/0.4
	53/2.00#	59.2.00#				
2	122/1.25	118/1.25	126/0.2	98/0.2	128/0.2	101/0.2
	123/1.50	119/1.50	127/0.4	99/0.4	129/0.3	102/0.3
	124/1.75	120/1.75		100/0.6		103/0.4
	125/2.00#	121/2.00#				
3				104/0.2		109/0.2
				106/0.4		108/0.3
				107/0.6		
4				111/0.2		113/0.2
				110/0.4		112/0.3
5				115/0.3		117/0.2
				114/0.4		116/0.3
6		130/1.25		132/0.2		135/0.2
		131/1.75		133/0.4		136/0.3
				134/0.6		137/0.4
7		138/1.25		140/0.2		143/0.2
		139/1.75		141/0.4		144/0.3
				142/0.6		145/0.4
8		146/1.25		148/0.2		151/0.2
		147/1.75		149/0.4		152/0.3
				150/0.6		

*Suspension configurations are as shown in Table 3-2.

#These tests are invalid due to an error in the input track geometry introduced by the computerized process when increasing the amplitude.

Note: Total runs made from run #48 through run #153. Runs not listed were aborted or gave inconclusive results.

Low Joint Test Philosophy

Staggered joint, bolted rail will invariably have low joints. Continuous welded rail laid on rehabilitated bolted rail road bed will also retain some of the low joint properties because of residual effects in the road bed. There are three basic variables to consider when defining these low joints:

- cross level amplitude: the difference in elevation between the left and right rail at a given station, in inches. A stated amplitude of cross level for staggered rail defines the cross level at each low joint station with the low joint alternately at the left and right rail for each half rail length.

- shape: usually a cusped low joint--the sharpness of the cusp varies and is usually sharper than a rectified sine.

- rail length:- 39 foot rail is the standard length today. However, there is 33 foot rail (previous standard length) still in service, and there has been a limited amount of 45 foot and 60 foot rail used.

One additional variable is the critical speed (the speed at which the freight car has maximum amplitude of the harmonic roll motion). It varies between freight cars, varies depending on gross weight, and will vary with amplitude of roll motion generated.

The objective of the staggered rail test is to subject the freight car to a representative, worst-case, staggered rail profile. The test should indicate the maximum harmonic roll that the car would experience in service, and should be consistently repeatable so that variations in car and truck configurations can be accurately compared and evaluated. The AAR shimmed track test is designed to meet these objectives. The test track is stagger shimmed to 3/4 inch for 10 rail lengths. The test car is run over this shimmed track at several speeds so as to include a run at the critical speed.

The AAR shimmed track test can be duplicated on the Vibration Test Unit. However, in the interest of shortening the test time, the speed slow down test was devised so that the one run would accomplish the same results as a number of constant speed runs. For a linear system the envelope of response amplitude versus speed would be the same for both test methods since both have enough cycles at each speed to build up to maximum response. For nonlinear systems the responses generated by the two test methods may be different, and, in fact, it has been found that for freight cars a speed slow down through the critical harmonic roll speed will result in larger response than constant speed runs. Consequently, it was decided to use the speed slow down test on the VTU, since it would result in worst case conditions and would involve the least amount of test time.

Superelevation Tests

The VTU tests with superelevation were performed in a simulation of the freight car going through a curve at sub-balance speeds. For example, if the curve were of 4 degrees with 4 inches superelevation, it would have a balance speed of about 39 miles per hour. If then the car takes the curve at 20 miles per hour, the steady state wheel load condition would be the same as a stationary 3 inches of superelevation.

3.3 Instrumentation and Data Acquisition

The test vehicle was instrumented with accelerometers, string potentiometers, and a roll gyro. In addition, wheel load, and wheel lift measurements were provided by strain gauges mounted on the rail sections supporting each of the eight wheels. These rails were mounted on the Axle Support Systems Bearing Assembly (ASSBA) of the VTU with shims at each end to effect a simply supported beam and with the strain gauges located directly opposite the wheel locations for detecting maximum strain. The strain values for each wheel were independently monitored by the computer controlling the shaker and limit checked at extremely low values (15 micro inches/inch) representing the wheel lift condition.

The instrument description and channel identification are given in Table 3-4. Boxcar and lading instrument locations are also shown in Figure 3-2. Data from selected test channels were displayed in real time using three Brush Recorders and one electrostatic recorder. Table 3-5 gives the list of test channels acquired on the four recorders. Digital recording of all test and VTU system channels was also carried out as a backup for future data analysis.

3.4 Data Output

There were 28 channels of data recorded and displayed in real time on the four strip charts as listed in Table 3-5. In addition to this, there were a few input displacement and response accelerations from the Track Geometry Tests processed post-test into Power Spectral Density plots.

TABLE 3-4
TEST MEASUREMENTS CHANNEL IDENTIFICATION AND DESCRIPTION

NUMBER	DESCRIPTION
D1X	LATERAL DISPLACEMENT BETWEEN RIGHT SIDEWALL OF CARBODY AND TOP OF LADING; B END OF CAR
D2X	LATERAL DISPLACEMENT BETWEEN RIGHT SIDEWALL OF CARBODY AND TOP OF LADING; CENTER OF CAR
D3X	LATERAL DISPLACEMENT BETWEEN RIGHT SIDEWALL OF CARBODY AND TOP OF LADING; A END OF CAR
A4Y	LONGITUDINAL ACCELERATION OF LADING ON TOP ALONG CENTERLINE; B END OF CAR
A5Z	VERTICAL ACCELERATION OF LADING ON TOP RIGHT SIDE; B END OF CAR
A6X	LATERAL ACCELERATION OF LADING ON TOP RIGHT SIDE; B END OF CAR
A7Z	VERTICAL ACCELERATION OF LADING ON TOP RIGHT SIDE; CENTER OF CAR
A8Z	VERTICAL ACCELERATION OF LADING ON TOP RIGHT SIDE; A END OF CAR
A9X	LATERAL ACCELERATION OF LADING ON TOP RIGHT SIDE; A END OF CAR
A10Y	LONGITUDINAL ACCELERATION OF CARBODY ON BOTTOM ALONG CENTERLINE; B END OF CAR
A11Z	VERTICAL ACCELERATION OF CARBODY ON BOTTOM RIGHT SIDE; B END OF CAR
A12X	LATERAL ACCELERATION OF CARBODY ON BOTTOM RIGHT SIDE; B END OF CAR
A13Z	VERTICAL ACCELERATION OF CARBODY ON BOTTOM LEFT SIDE; B END OF CAR
A14Z	VERTICAL ACCELERATION OF CARBODY ON BOTTOM RIGHT SIDE; A END OF CAR
A15X	LATERAL ACCELERATION OF CARBODY ON BOTTOM RIGHT SIDE; A END OF CAR
A16Z	VERTICAL ACCELERATION OF CARBODY ON BOTTOM LEFT SIDE; A END OF CAR
A17X	LATERAL ACCELERATION OF CARBODY ON TOP RIGHT SIDE; B END OF CAR
D18Z	ROLL DISPLACEMENT ALONG CENTERLINE OF CARBODY FLOOR; B END OF CAR
XE1A	PISTON DISPLACEMENT ACTUATOR 1A (VERTICAL)
XE1B	PISTON DISPLACEMENT ACTUATOR 1B (VERTICAL)
XE1C	PISTON DISPLACEMENT ACTUATOR 1C (LATERAL)
SG1A	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 1A
SG1B	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 1B
SG2A	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 2A
SG2B	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 2B
SG3A	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 3A
SG3B	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 3B
SG4A	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 4A
SG4B	STRAIN GAUGE FOR WHEEL LIFT ON ACTUATOR 4B

TABLE 3-5
SAFETY MARGIN TESTING--RECORD FORMAT

RECORDER CHANNEL	BRUSH 1	BRUSH 2	BRUSH 3	ELECTROS
1	D18Z	D2X	XE1A	D18Z
2	XE1A	A7Z	XE1B	
3	D1X	D3X	XE1C	SG2A
4	45Z	A8Z	A12X	SG2B
5	A6X	A9X	A17X	SG3A
6	A11Z	A14Z	A11Z	SG3B
7	SG1A	A15X	A13Z	SG4A
8	SG1B	D18Z	D18Z	SG4B

4. RESULTS SUMMARIZED

The two primary objectives of the Safety Margin tests were to determine what conditions cause the plywood lading to shift laterally, and to determine the loss in derailment margin resulting from the lading shift. Consequently, the test data have been summarized in the two general categories of Lading Shift Threshold and Wheel Lift Threshold. Within each of these categories track geometry, harmonic roll, tangent track, and superelevated track conditions are discussed.

4.1 Lading Shift Threshold

Track Geometry Tests - VTU Endurance Testing

Prior to performance of the Safety Margin Tests, the test vehicle was subjected to a series of tests (for the AAR) with the objective of demonstrating capability of the Vibration Test Unit to operate for sustained periods of time using track geometry as input excitation. These "Endurance Tests," reported in Reference 5, consisted of three, three-hour periods of continuous excitation on the VTU, using track geometry data reformatted from LTHD data acquired on track at the TTC. Each three-hour endurance test contained measurements made on selected sections of the Facility for Accelerated Service Testing (FAST), the Train Dynamics Track (TDT), and the Railroad Test Track (RTT) at speeds of 15 and 30 miles per hour. The track geometries measured included track sections that were perturbed for bounce and harmonic roll.

As reported in Reference 5, the VTU endurance testing was conducted with the plywood lading centered and shifted. However, in the course of the centered tests, the lading shifted laterally to the point of making contact to the side wall. Thus, for the purposes of the Safety Margin evaluation, track geometry tests at the measured level (1.00) will cause a complete shift of the plywood lading in less than three hours.

Track Geometry Tests - Safety Margin Testing

Even though the TG Safety Margins Tests were performed at increased amplitudes, they were not as severe as the endurance testing for two reasons. First the sections of perturbed track were not included, and second the test time was short (5 1/2 minutes against 3 hours).

There were five TG runs made with centered lading: two in Configuration 1, and three in Configuration 2. (The runs considered invalid because of input wave form problems are excluded; refer to Table 3-3). (Configurations 1 and 2 are defined in Table 3-2). The results of these five runs in terms of plywood lateral shift are given in Table 4-1. There was lateral shifting of the plywood lading in each TG test with approximate averages of 0.25 inches in Configuration 1, and 0.71 inches in Configuration 2 for each 5 1/2 minute run.

Staggered Rail Tests

A summary of results from the staggered rail tests with centered lading is presented in Figures 4-1, 4-2, and 4-3, and in Table 4-2. The data show that the plywood will shift with staggered rail low joint cross levels of 0.40 inches with tangent track, and 0.20 inches with 3.0 inches superelevation.

Summary

The summary conclusion to be drawn, relative to the threshold of rail profile variations that will cause lading to shift, is as follows: The plywood lading, when centered and braced according to AAR specifications, will shift to one side, to the point of pressing against the side wall, under the following conditions:

1. With measured track geometry, the lading will shift completely within three hours of service.

2. With simulated staggered rail, the threshold cross level is less than 0.40 inches on tangent track, and less than 0.2 inches at sub-balance speeds on curves. However, since the staggered rail low joints would appear in service in conjunction with measured track geometry, they do not in actuality have a threshold but will act to accelerate the lateral shift movement of the plywood.

TABLE 4-1
PLYWOOD LADING LATERAL SHIFT DURING TRACK GEOMETRY TESTING

Run No.	Input Level	Lading Lateral Shift A End (inches)	Lading Lateral Shift B End (inches)
Configuration 1			
49	1.25	.21	.20
50	1.50	.24	.26
Configuration 2			
122	1.25	.91	.73
123	1.50	.64	.56
124	1.75	.47	.95

Note: Duration of each run was 5.5 minutes, representing 2.75 miles of FAST track, at 30 miles per hour.

TABLE 4-2
PLYWOOD LADING LATERAL SHIFT DURING HARMONIC ROLL TEST

Run No.	Cross Level Input (inches)	Lading[1] Shift (inches)	Rail[3] Lengths	Critical Speed (mph)
Configuration 1, Tangent Track				
54	0.20	0.0	---	21.0
55	0.40	6.0	100	19.5
Configuration 2, Tangent Track				
126	0.20	0.0	---	21.2
127	0.40	2.5	70	20.2
Configuration 2, 3.0 Inches Superelevation				
128	0.20	3.0	60	22.0
129	0.30	6.0[2]	300[4]	23.5

[1] Average of three measurements: A end, center, and B end

[2] Lateral shift at A end; B end and center had shifted to wall at beginning of test

[3] Rail lengths over which shifting of lading occurs

[4] Lading shifted in this test before reaching critical speed

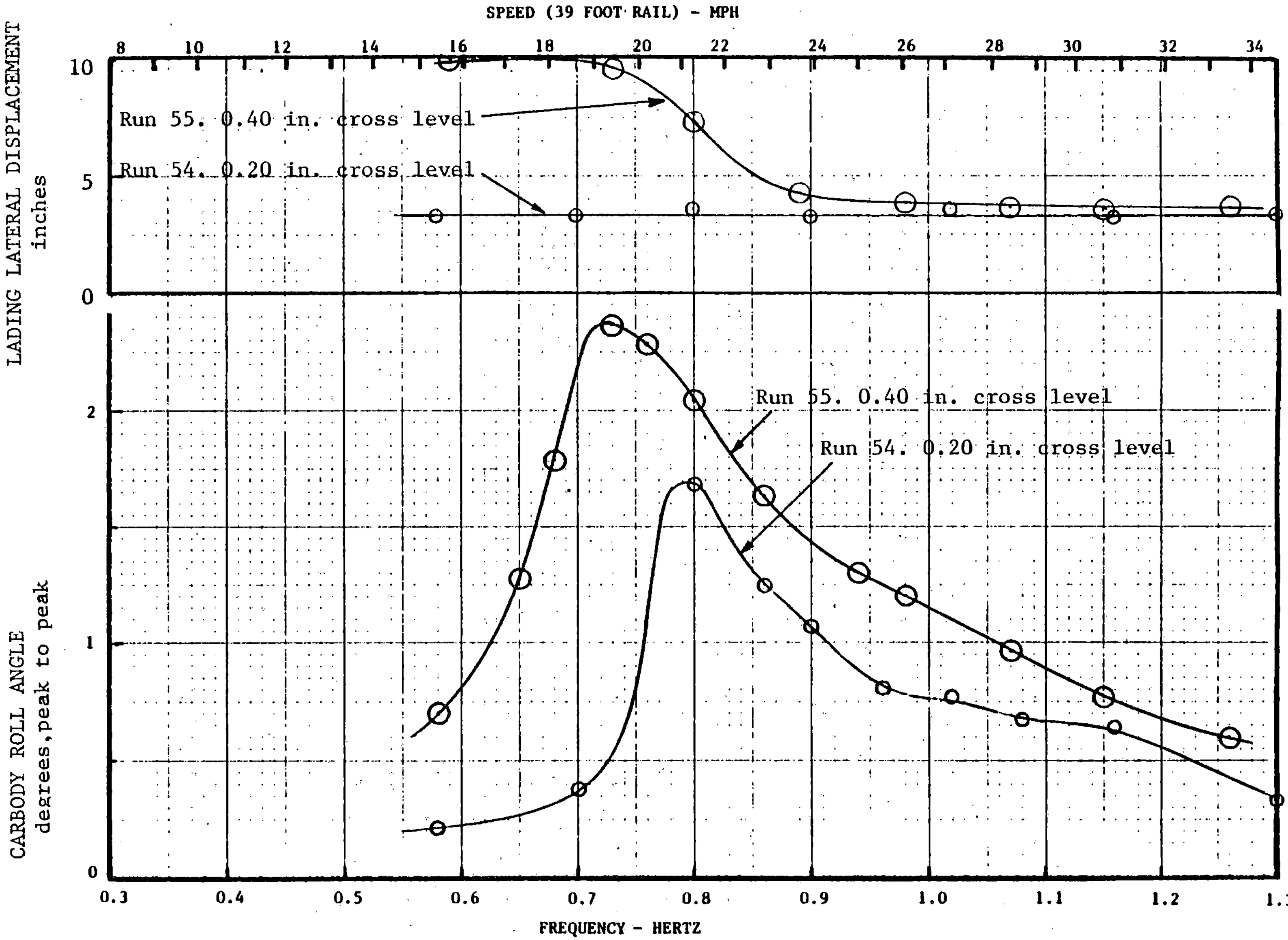


FIGURE 4-1
CONFIGURATION 1: CARBODY ROLL ANGLE AND LADING SHIFT
IN STAGGERED RAIL TESTS

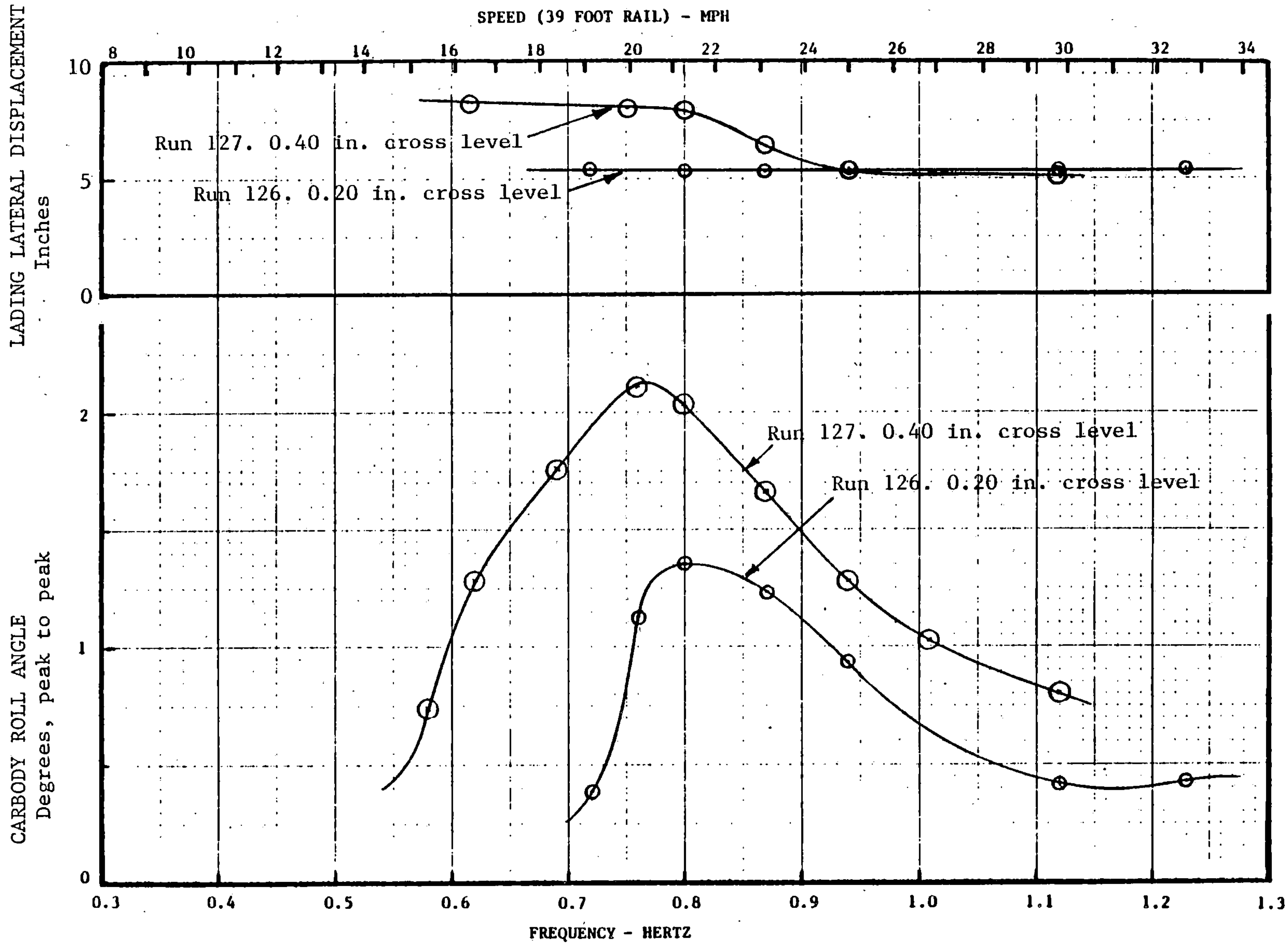


FIGURE 4-2
CONFIGURATION 2: CARBODY ROLL ANGLE AND LADING SHIFT
IN STAGGERED RAIL TESTS

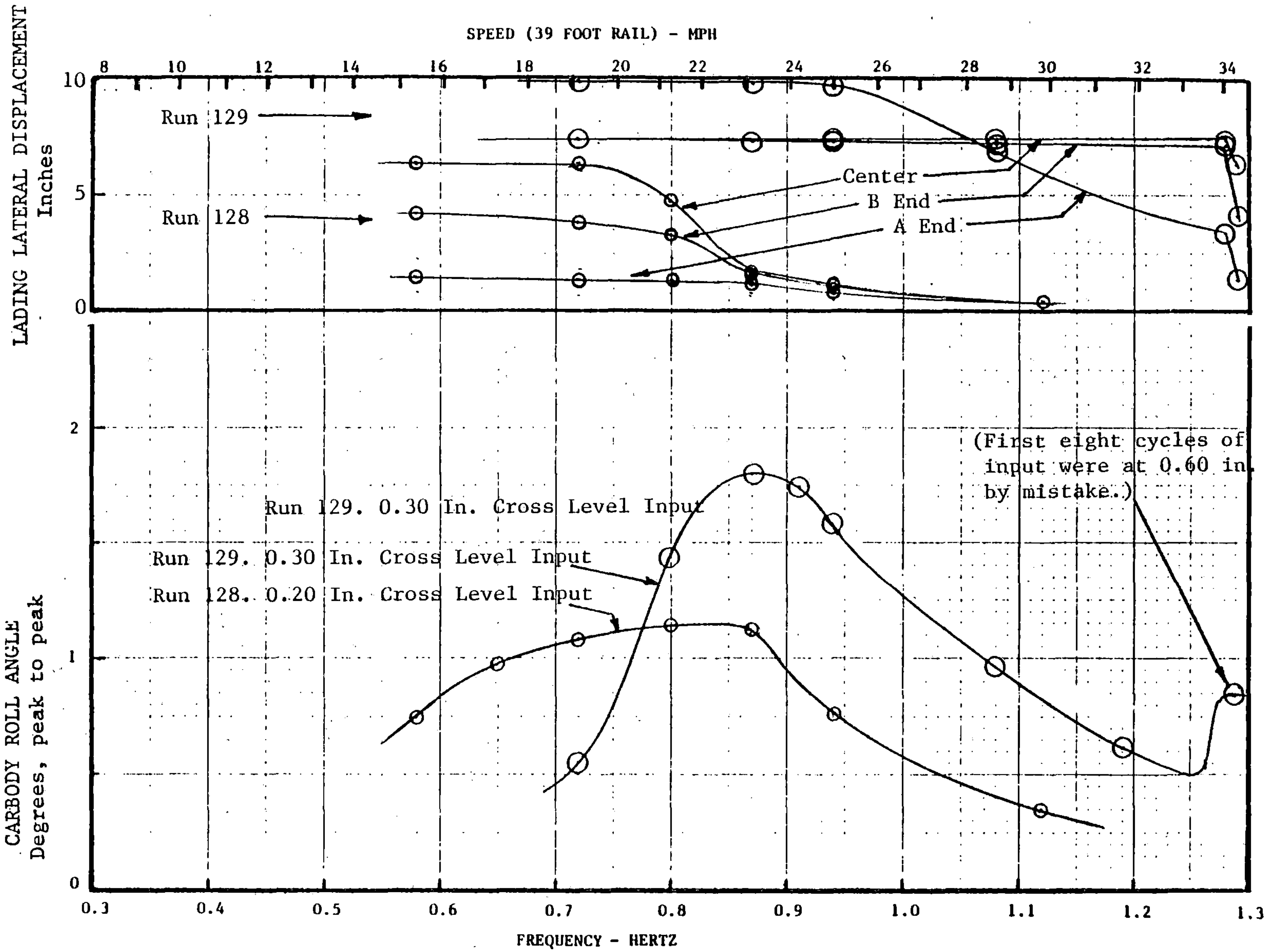


FIGURE 4-3
CONFIGURATION 2 WITH 3 INCH SUPERELEVATION,
CARBODY ROLL ANGLE AND LADING SHIFT
IN STAGGERED RAIL TEST

4.2 Wheel Lift Thresholds

Development of Criteria

There were several vehicle response parameters considered in search for a derailment margin measurement criterion. They were:

- o Wheel lift (near zero load)
- o Carbody roll angle
- o Amplitude of wheel lift
- o Maximum wheel loads
- o L/V
- o Center plate lift

Center plate lift was rejected for use as a derailment margin criterion since it has been found to occur only when the vehicle is lightly loaded or empty.

The L/V criterion was rejected since, for harmonic roll, the lateral and vertical wheel loads reach maximum values together so that the L/V ratio stays relatively small.

While monitoring wheel loads, it was found that wheel lift occurred before significantly large wheel loads were reached on the opposite wheel. The maximum vertical wheel load measured prior to wheel lift occurred on the low rail in the superelevation testing and was about 35,000 pounds. This is about 1.75 times tangent track static load.

The use of amplitude of wheel lift as a derailment criterion was rejected primarily because of the risk of damage to the test facility. It was also felt that to include wheel lift as a permissible service condition would be very undesirable because of the increased loads due to wheel rail impact.

In the harmonic roll condition, other things being constant, the relationship between carbody roll and wheel lift varies with center of gravity (c.g.) height. That is, if the c.g. height is increased, wheel lift will occur with a smaller carbody roll angle. Wheel lift would then seem to be the more meaningful criterion to use. Consequently, in the staggered rail harmonic roll tests, carbody roll angle and vertical wheel loads were the main parameters monitored for measuring performance, but zero wheel load was identified as the limit criterion.

4.2.1 Staggered Rail Test Results: Tangent Track

The two most significant measurements in the staggered rail tests were the carbody roll angle, as measured by the roll gyro, and wheel unloading, as measured by the rail section strain gages. The eight configurations tested, listed in Table 3-2 and again in Table 4-3, are variations of friction and hydraulic snubbing. The objective of the testing was to show what staggered rail cross levels would cause wheel lift, and to show what snubbing conditions improved performance.

The data were plotted in the groups shown in Table 4-4, in order to show better the relative performance for the several config- urations. Table 4-4 also states the comparison objective for each group, and the two figures presenting results; one shows roll angle, and the other shows wheel unloading for each group.

Comparison of Centered and Shifted Lading in Staggered Rail Test Results

Configuration 1 is the basic configuration, the Barber S-2-C truck as received with the test boxcar. It has both inner and outer side springs and has, as identified in this report, high friction force. Configuration 2 has reduced the friction snubbing force by removal of the inner side springs, and has the low force hydraulic snubber in the center spring position.

The test results shown in Figures 4-4 and 4-5 are for both centered and shifted conditions of the lading from tests with Configurations 1 and 2. There does not appear to be any significant difference in response in these four configura- tions, for the two cross levels tested (0.20 and 0.40 inches). The carbody roll angles in Configuration 2 are smaller by 10-20 percent, but there also appears to be more wheel unloading.

Low Force Hydraulic Snubber with Changes in Friction Snubber Force--Figures 4-6 and 4-7

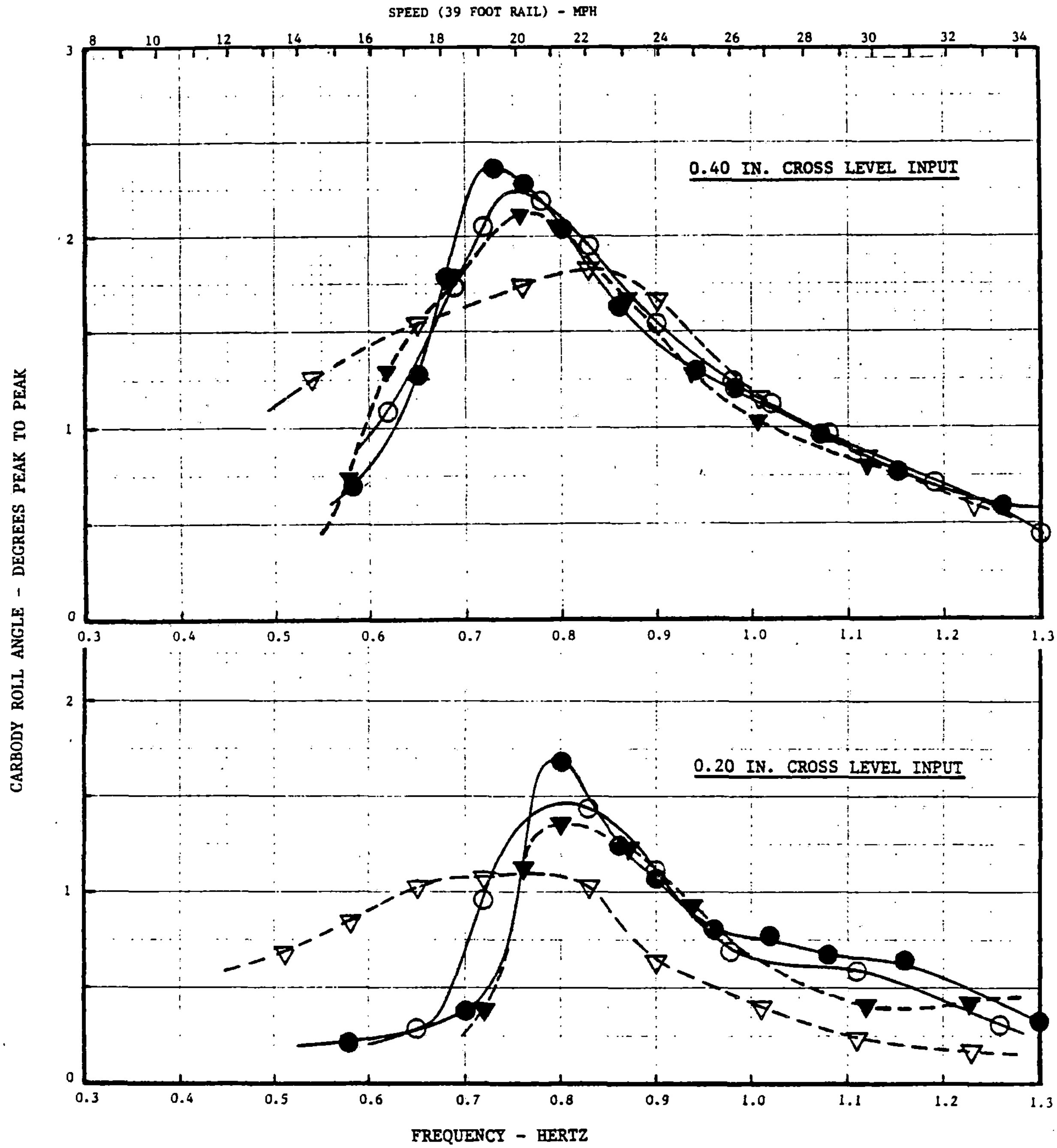
The addition of the low force hydraulic snubber resulted in a smaller carbody roll angle at the critical speed and larger roll angles above the critical speed; this is as expected in that more damping reduces response at resonance and increases response above resonance. However, when the friction snubbing is reduced, with low force hydraulic snubber, the response is reduced at and above the critical speed. This is not as one would expect, but can be explained on the basis that the system is nonlinear and its behavior will vary from linear theory. It appears that there is an optimum damping to minimize carbody roll angle.

TABLE 4-3
TRUCK CONFIGURATIONS TESTED

CONFIGURATION NO.	PLOT SYMBOL	FRICTION SNUBBER	HYDRAULIC SNUBBER
1	○	High	None
2	▿	Low	Low
3	◇	Low	High
4	◇	High	Low
5	▽	High	High
6	◿	None	Low
7	◿	None	High
8	○	Low	None

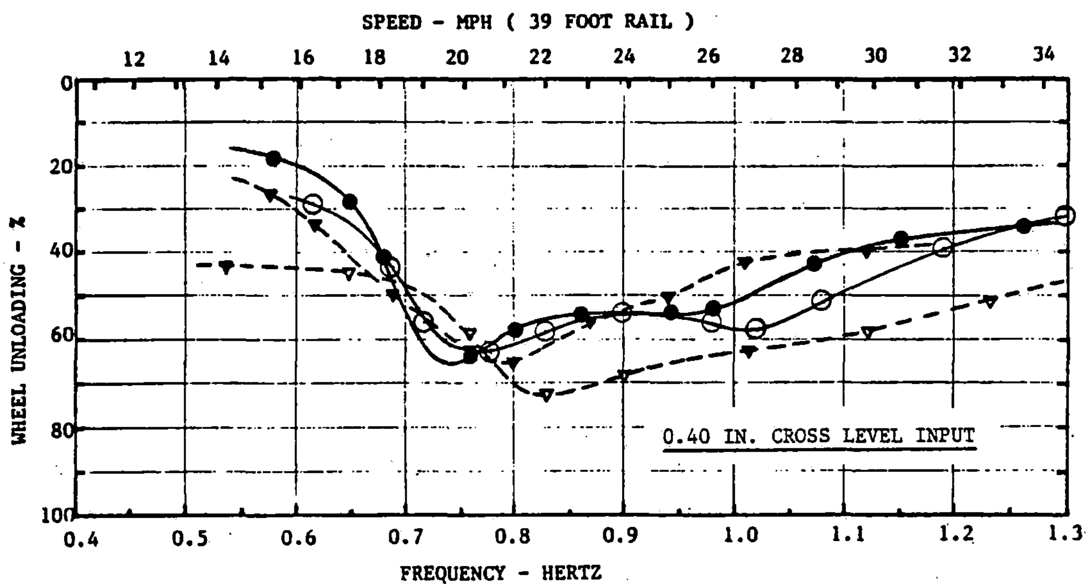
TABLE 4-4
RESULTS COMPARISON GROUPS - STAGGERED RAIL DATA

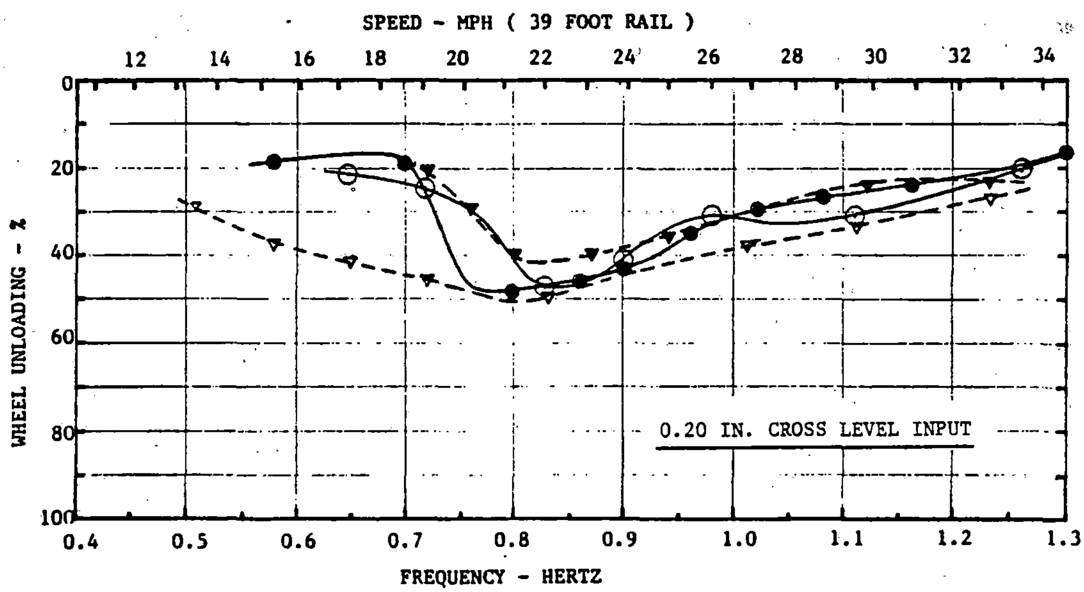
CONFIGURATIONS COMPARED	COMPARISON OBJECTIVE	FIGURE NUMBERS
1, 2	Centered vs. shifted lading in Configurations 1 and 2	4-4, 4-5
1, 2, 3, 6	Effect of friction snubber with low force hydraulic	4-6, 4-7
1, 4, 5, 7	Effect of friction snubber with high force hydraulic	4-8, 4-9
1, 3, 4	Effect of hydraulic snubber with high force friction	4-10, 4-11
1, 2, 5, 8	Effect of hydraulic snubber with low force friction	4-12, 4-13



CONFIGURATION	SYMBOL	FRICTION SNUBBER	HYDRAULIC SNUBBER	LADING
1	——●——	HIGH	NONE	CENTERED
1	——○——			SHIFTED
2	– – ▼ – –	LOW	LOW	CENTERED
2	– – ▽ – –			SHIFTED

FIGURE 4-4
CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING
CONFIGURATIONS 1 AND 2





CONFIGURATION	SYMBOL	FRICTION SNUBBER	HYDRAULIC SNUBBER	LADING
1	——●——	HIGH	NONE	CENTERED
1	——○——			SHIFTED
2	– – ▼ – –	LOW	LOW	CENTERED
2	– – ▽ – –			SHIFTED

FIGURE 4-5
WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1 AND 2

The results of Configuration 6 (no friction snubbers) should be noted. Above 20 mph, the roll angles are greatly reduced. At critical speeds, the roll angle is less at small cross level input (0.20 inches) but increases, with increased cross level, at a higher rate than configurations with more damping.

The least amount of wheel unloading occurred with the least damping, Configuration 6, with the general trend for wheel unloading to increase as damping was increased. However, in all cases wheel zero load was reached with 0.60 inches cross level input.

<u>High Force Hydraulic Snubber with Changes in Friction Snubber--Figures 4-8 and 4-9</u>

Comparing Configurations 1 and 7, high friction only and high hydraulic only, the performances are very similar. Zero wheel load is reached in both cases at 0.60 inches cross level input.

When the high force hydraulic snubber is used in combination with either the high or low force friction snubber, performance is worsened: roll angles are larger and there is greater wheel unloading. Zero wheel load was reached with 0.40 inches cross level input for both Configurations 4 and 5.

<u>High Force Friction Snubber with Changes in Hydraulic Snubber--Figures 4-10 and 4-11</u>

The hydraulic snubbers in combination with high force friction snubber resulted in poorer performance. The roll angles did not show this difference so much as wheel load data. Greater wheel unloading resulted with both high and low force hydraulic snubber. Zero wheel load was reached at 0.40 inches cross level with the high force friction--high force hydraulic combinations.

<u>Low Force Friction Snubber with Changes in Hydraulic Snubber--Figures 4-12 and 4-13</u>

The low force hydraulic, low force friction (Configuration 2) did not show any significant change over Configuration 1. The Configuration 2 roll angle was generally less while wheel unloading was generally more than Configuration 1.

The high force hydraulic snubber resulted in poorer performance; roll angles were greater and wheel unloading was greater. Zero wheel load was reached with 0.40 inches input.

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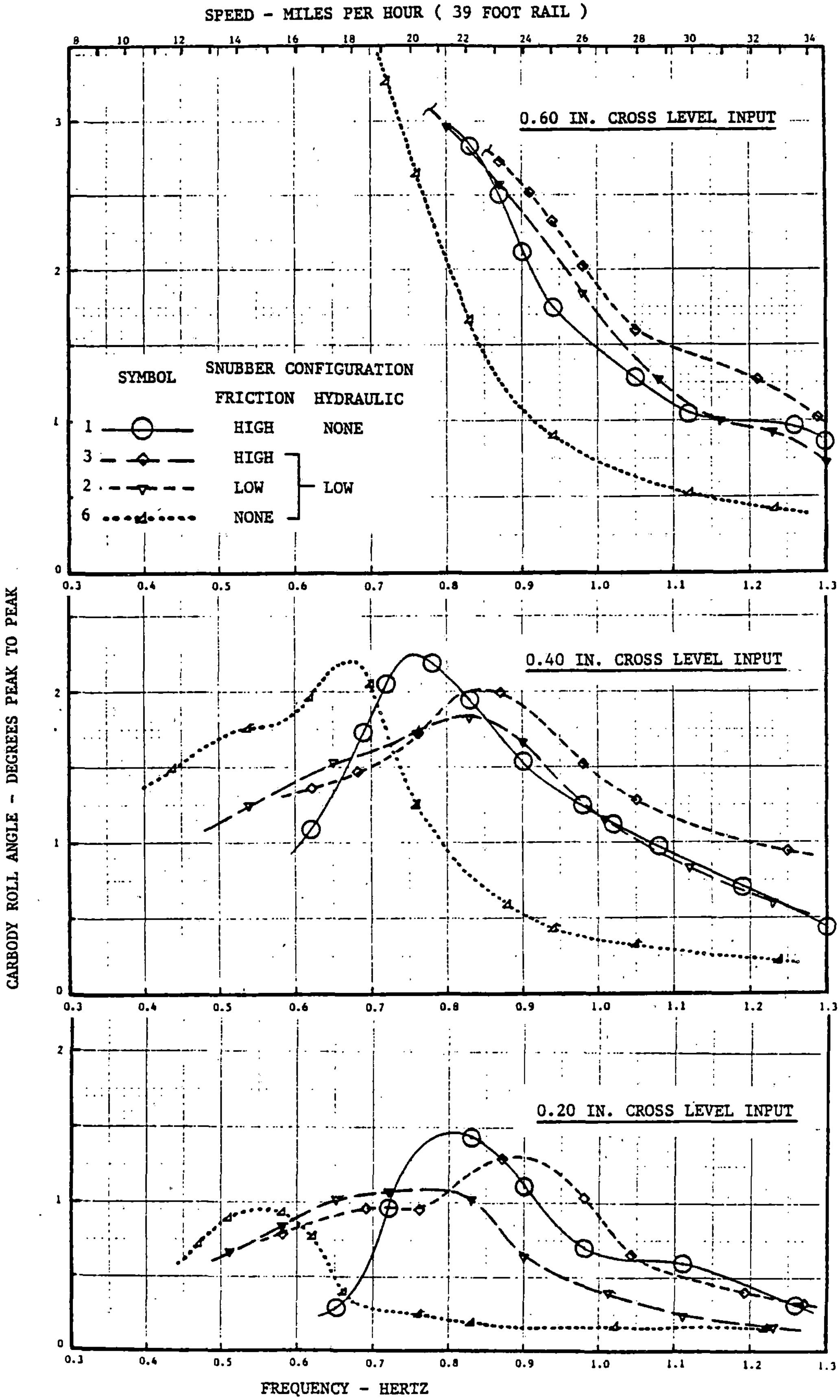


FIGURE 4-6
CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING
CONFIGURATIONS 1, 3, 2 AND 6

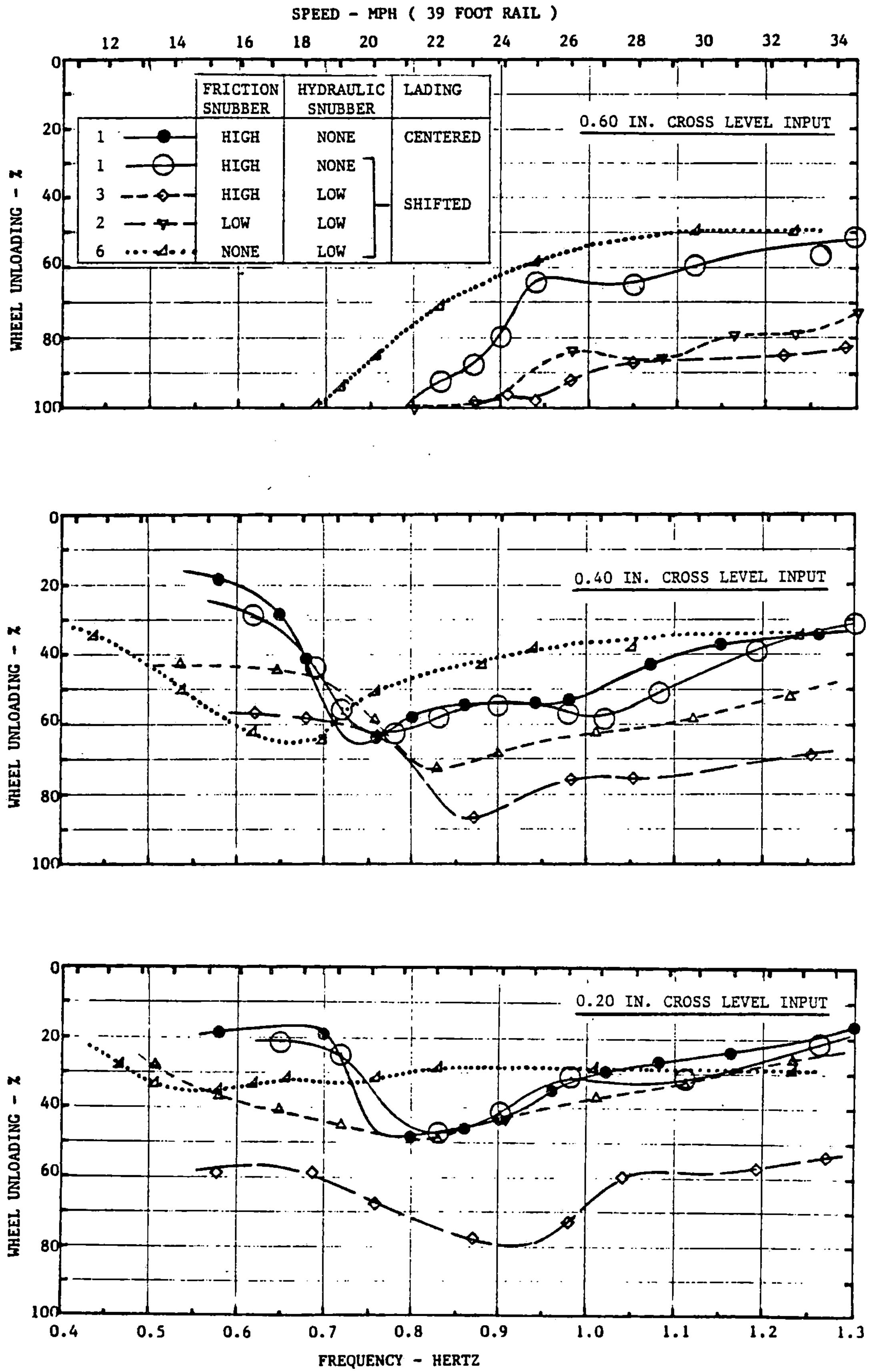


FIGURE 4-7
WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 3, 2 AND 6

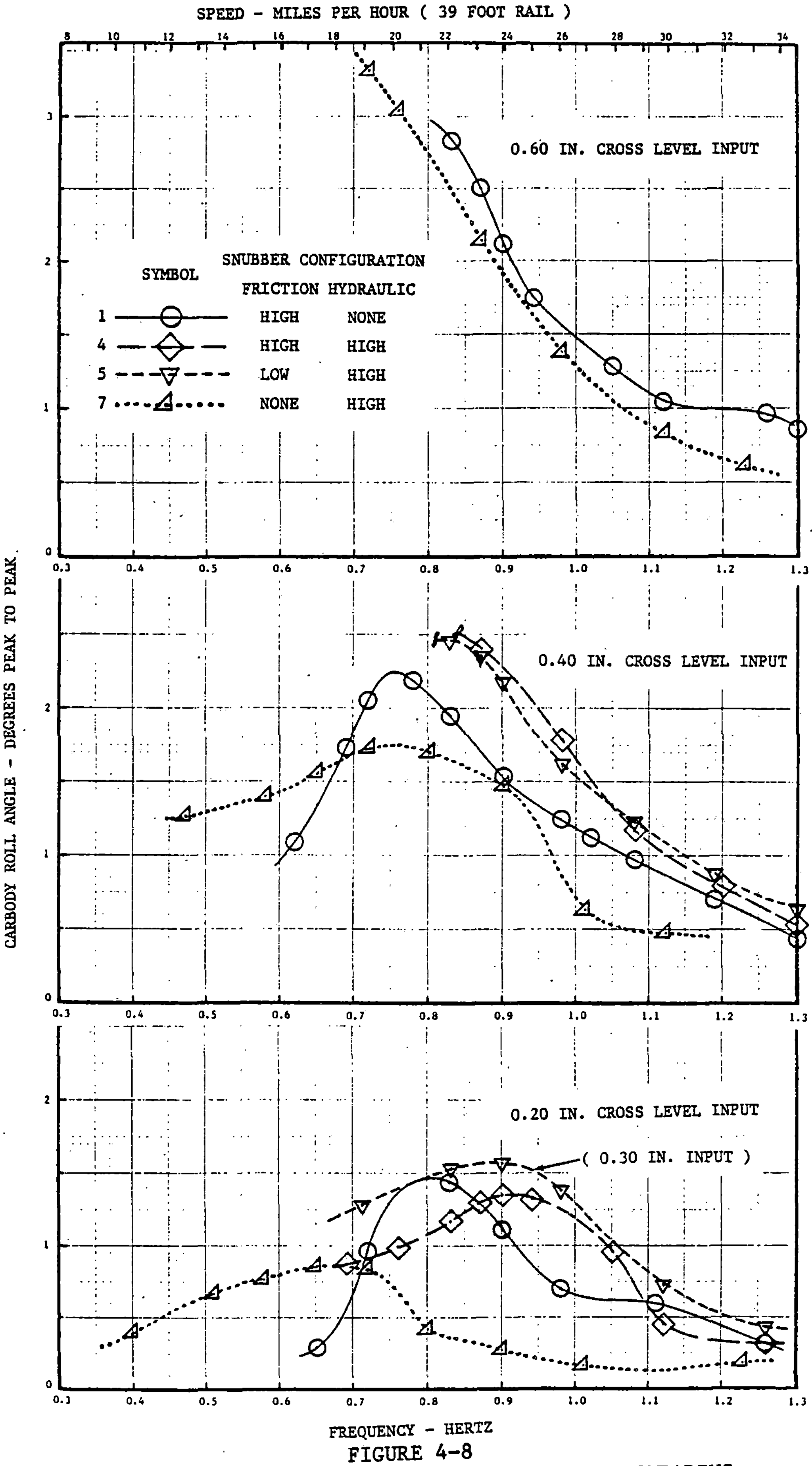


FIGURE 4-8
CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 4, 5 and 7

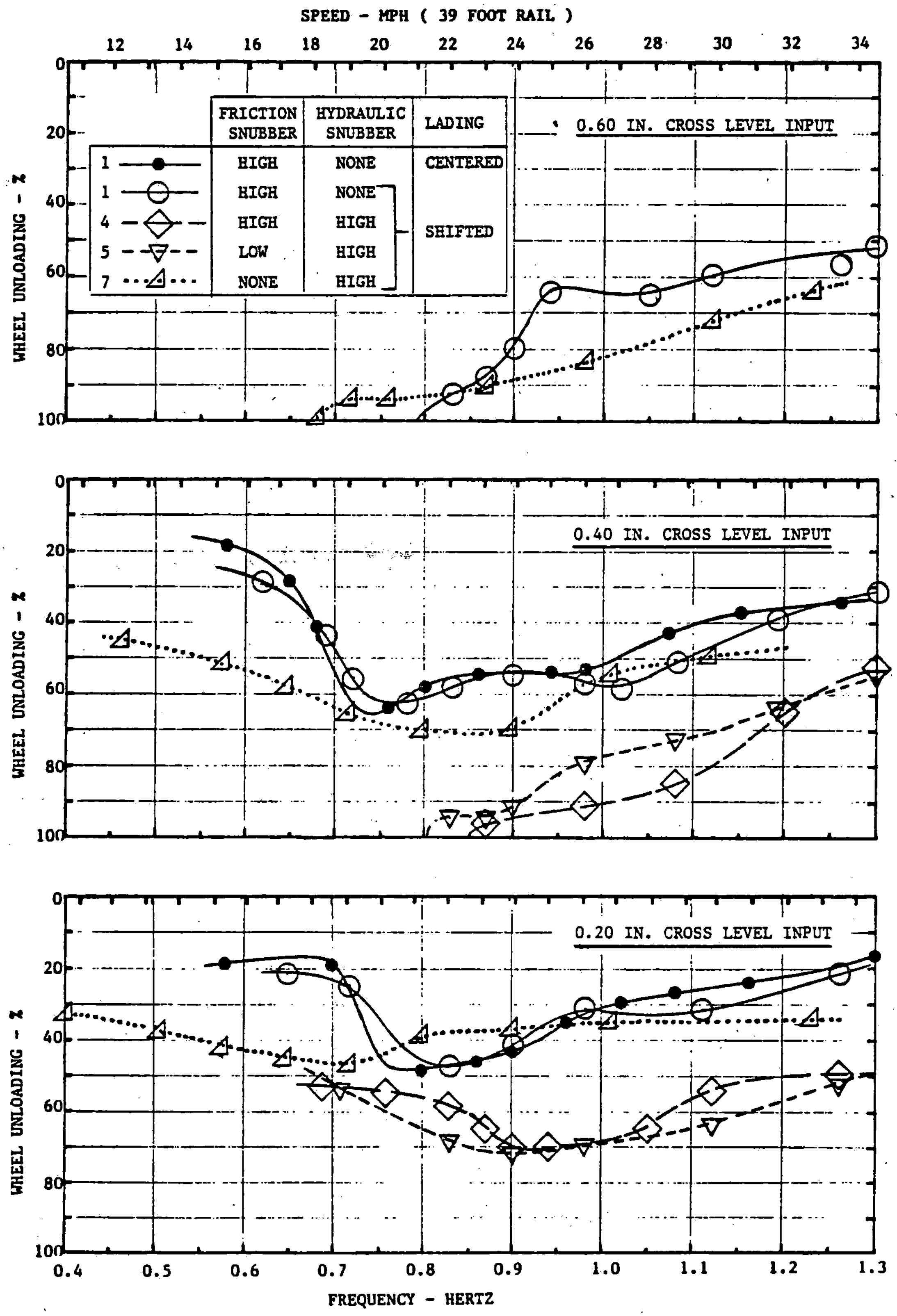


FIGURE 4-9
WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 4, 5 AND 7

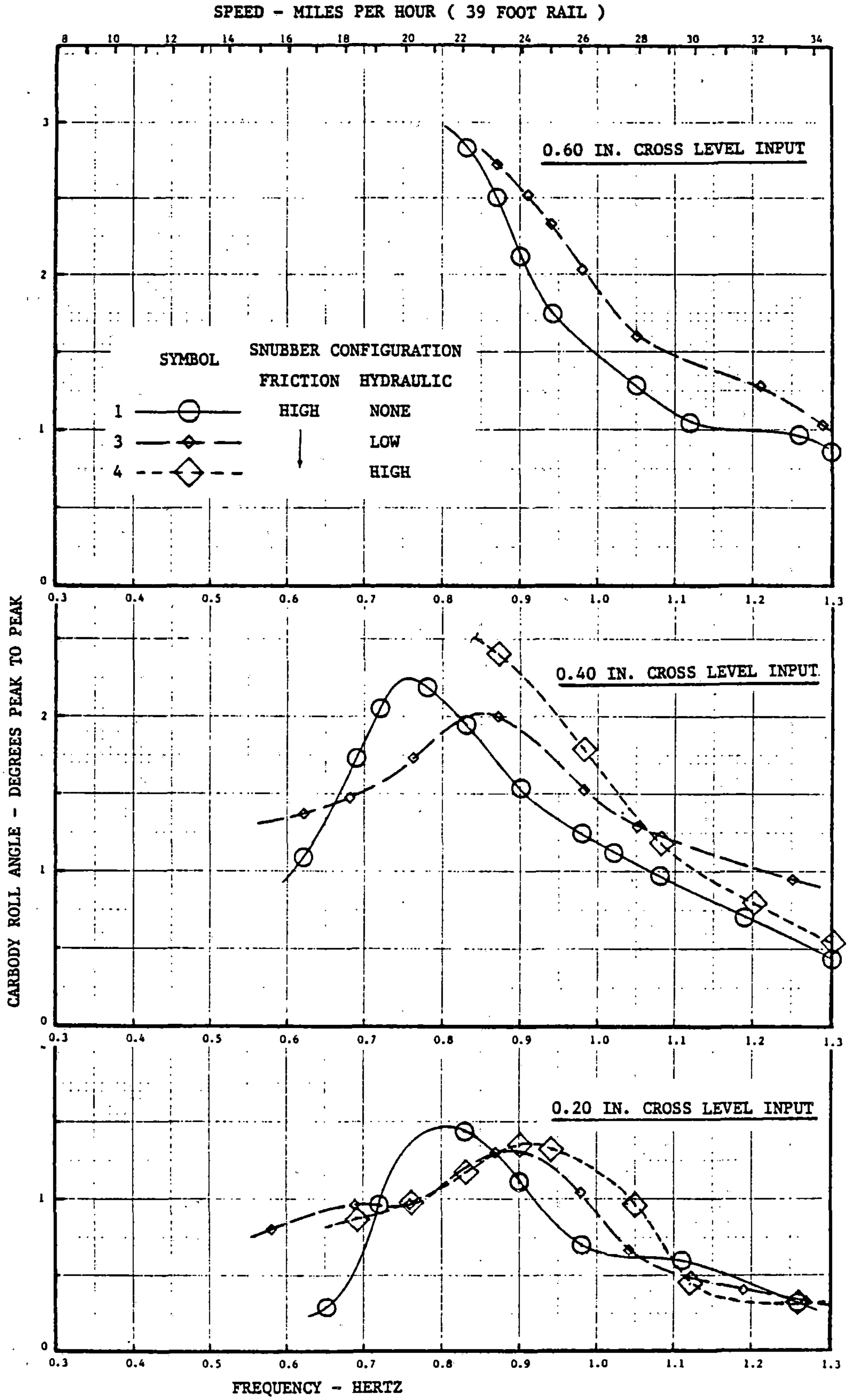


FIGURE 4-10
CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 3, AND 4

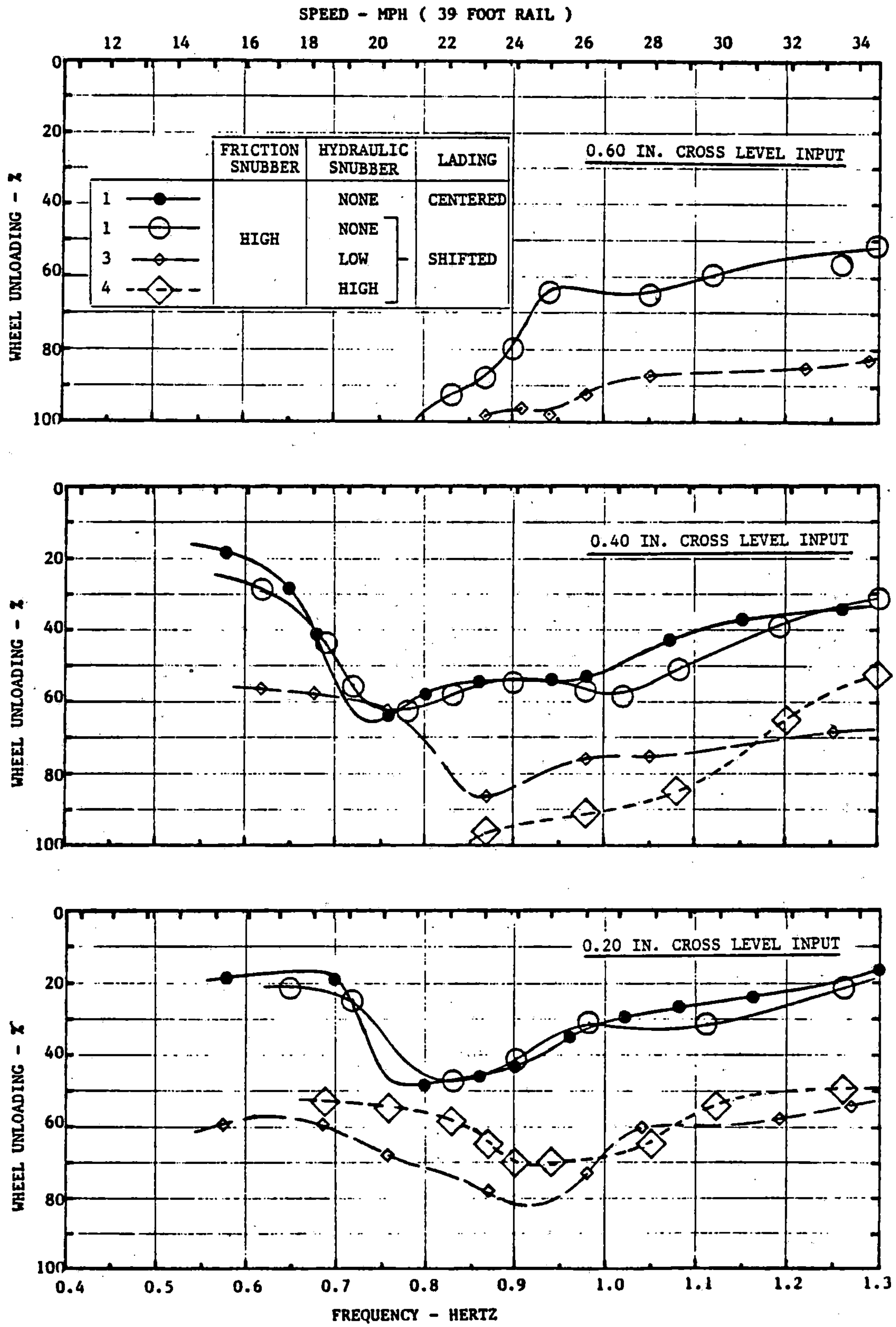


FIGURE 4-11
WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 3, AND 4

Summary

The results of the staggered rail tests with variation in friction and hydraulic snubber leads one to the conclusion, which seems to be universal when dealing with freight car dynamics, that we are dealing with a complex system and it is very difficult to quantify its behavior. One definitive statement can be made; there is an optimum damping force--friction, hydraulic or combined--beyond which the harmonic roll performance will degrade. For the test vehicle the high force friction snubber (inner and outer side springs), and the high force hydraulic snubber (used separately) are both beyond that optimum point. Harmonic roll performance would be improved with smaller damping forces.

Although the results indicate that optimum damping would comprise a combination of both friction and hydraulic damping, probably at force levels lower than those used in these tests, there was not enough testing done to reach a quantitative conclusion.

4.2.2 Staggered Rail Test Results: Superelevated Track

The rectified sine, staggered rail tests were performed with 3.0 inch superelevation, and with the plywood lading shifted for all eight truck configurations listed in Table 4-5. The test results are presented in Figures 4-14 through 4-22. These show the minimum vertical wheel loads for both the low rail and high rail sides, and the amplitude of the carbody roll angles. Table 4-5 also lists the maximum cross level of staggered rail in each case before reaching zero wheel load.

Zero wheel load was reached in every case near 0.30 inches cross level. With Configurations 1, 2, 6, and 7 zero wheel load occurred between 0.30 and 0.40 inches. With Configurations 3, 4, 5, and 8 zero wheel load occurred between 0.20 and 0.30 inches cross level.

The conclusion drawn was that wheel lift will very probably occur if a 70 ton boxcar loaded with plywood negotiates curved track with 0.40 inch low joints at sub-balance speeds. This assumes the plywood is shifted, which is a certainty based on tests performed here, and that the shift is to the same side as the low rail in the curve, which is a 50/50 chance.

Zero wheel load was reached in every case near 0.30 inches cross level. With Configurations 1, 2, 6, and 7 zero wheel

TABLE 4-5
SUMMARY RESULTS OF SUPERELEVATED STAGGERED RAIL TEST

Configuration Number	Snubber Conditions		Max Cross Level before Zero Wheel Load (inches)
	Friction	Hydraulic	
1	High	None	0.30
2	Low	Low	0.30
3	High	Low	0.20
4	High	High	0.20
5	Low	High	0.20
6	None	Low	0.30
7	None	High	0.30
8	Low	High	0.20

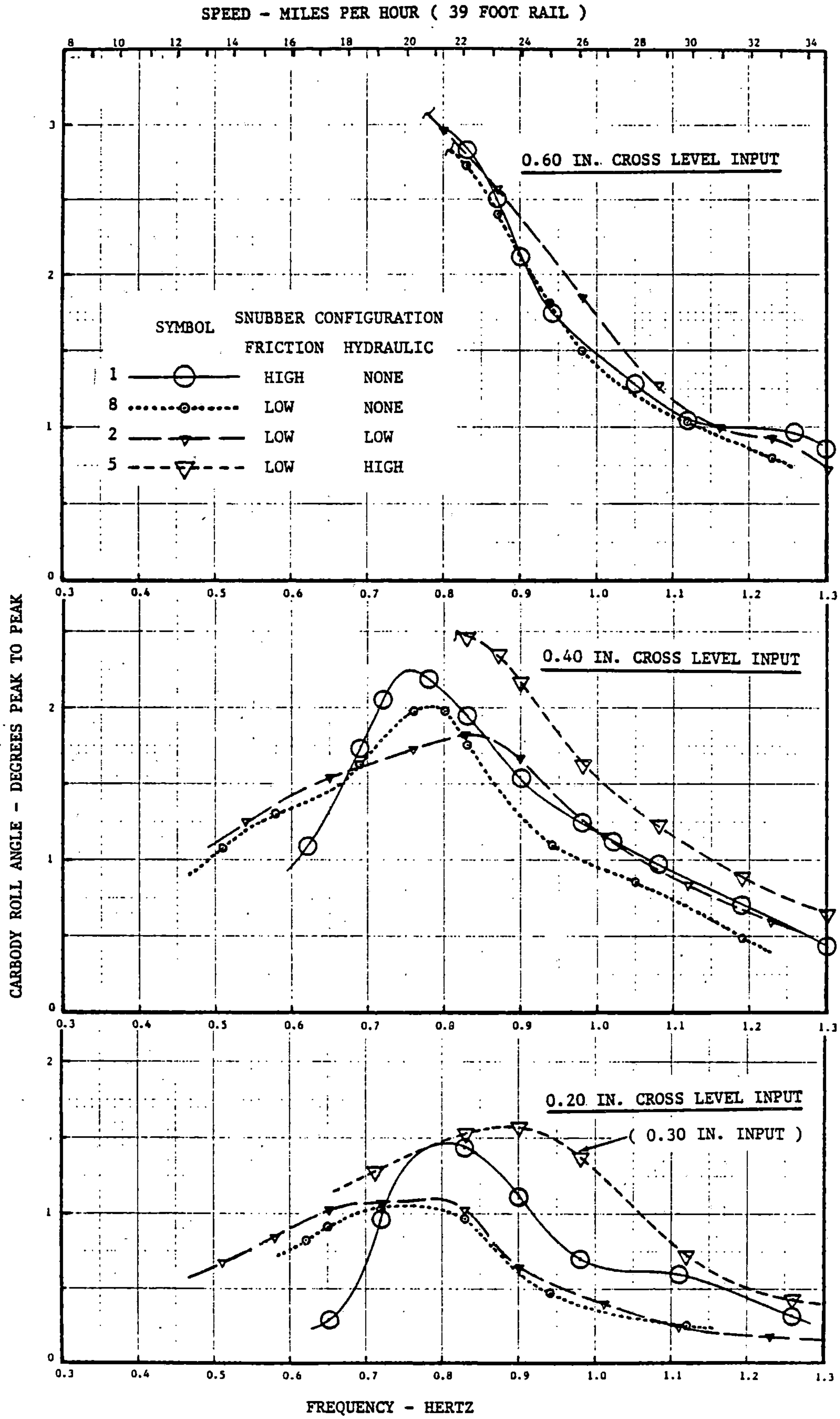


FIGURE 4-12
CARBODY ROLL ANGLES, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 2, 5, AND 8

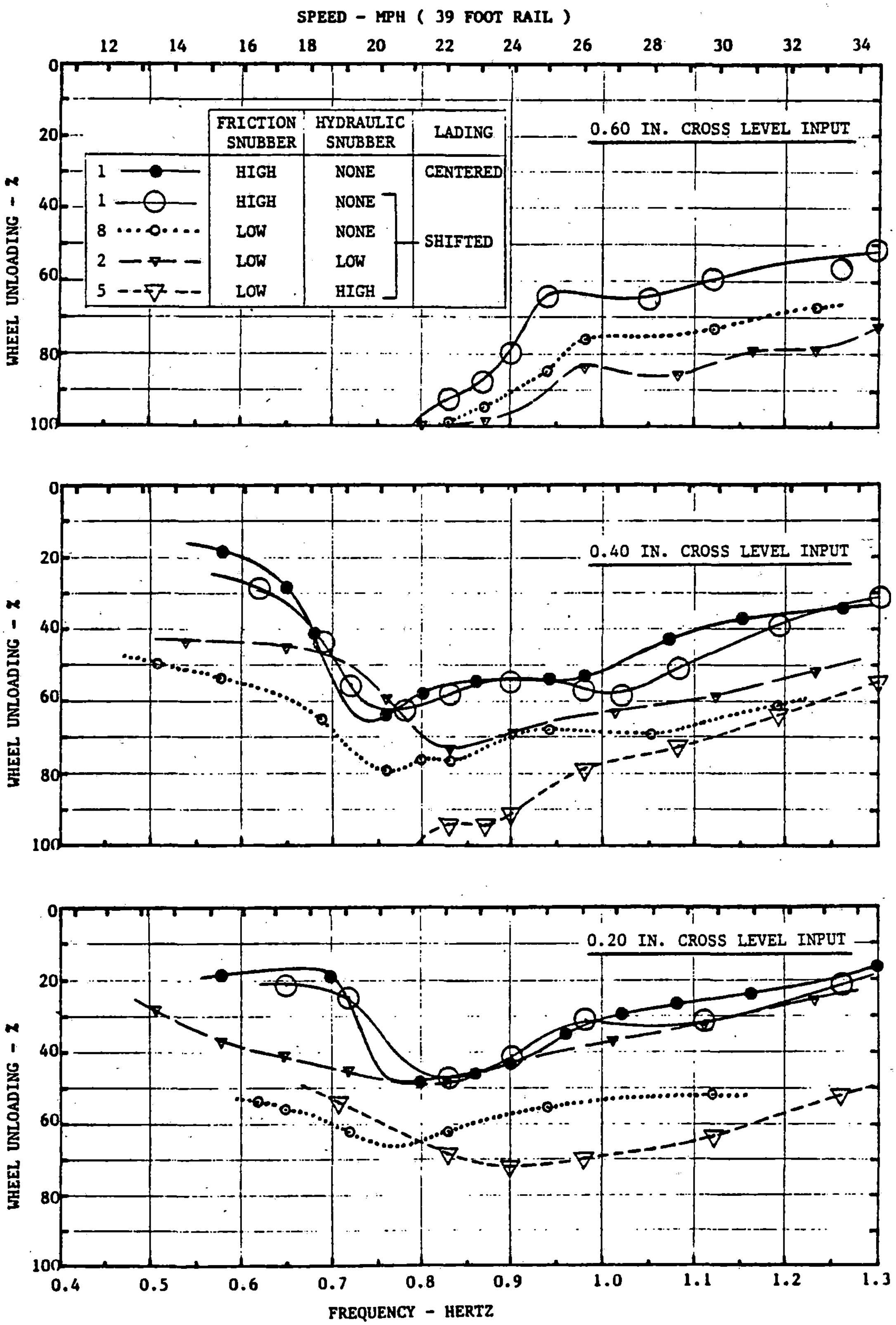


FIGURE 4-13
WHEEL % UNLOADING, STAGGERED RAIL TESTS, COMPARING CONFIGURATIONS 1, 2, 5, AND 8

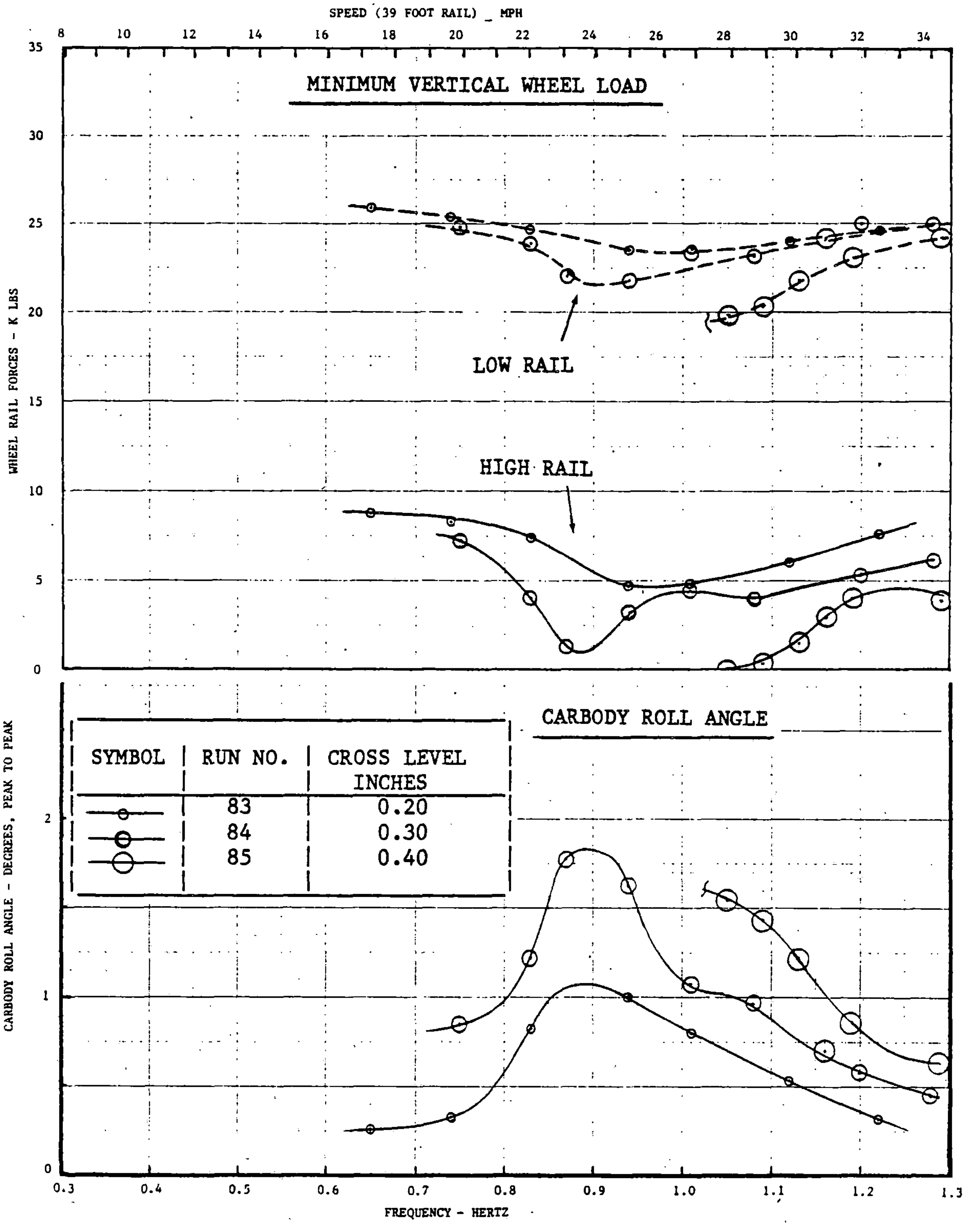


FIGURE 4-14
CONFIGURATION 1, STAGGERED RAIL TESTS WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED

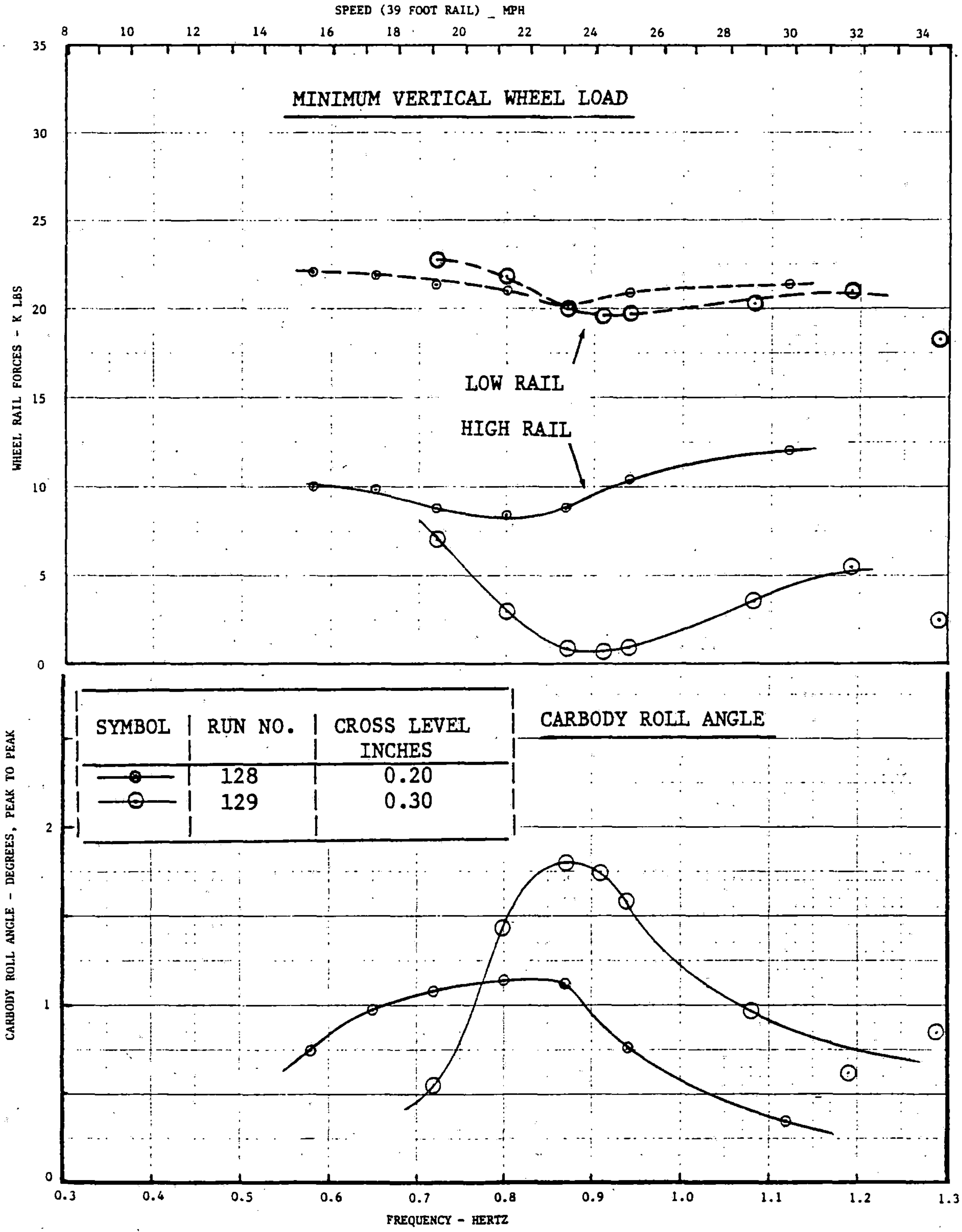


FIGURE 4-15

CONFIGURATION 2, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION (Lading was centered at start of test and shifted during test. See Figure 4.3)

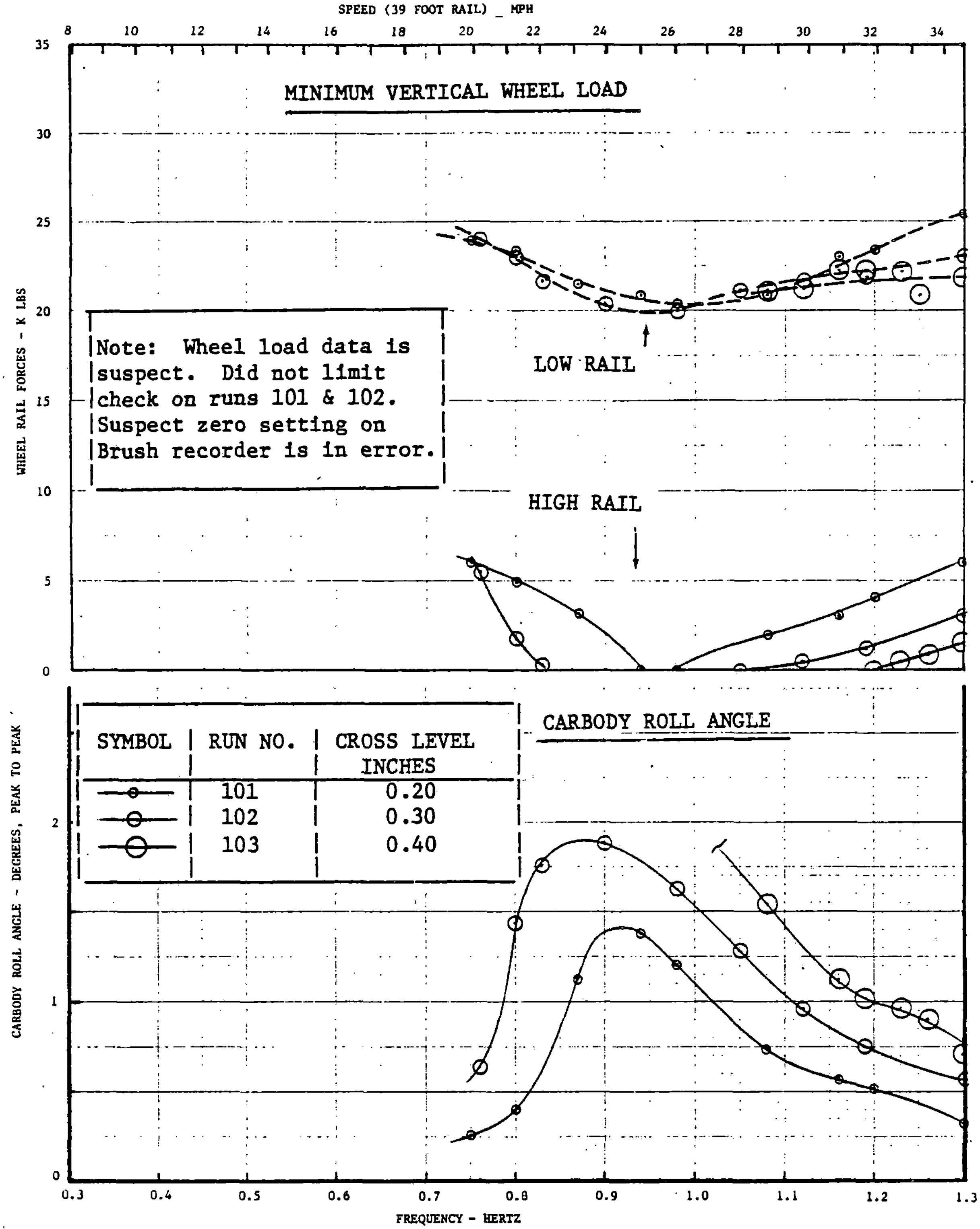


FIGURE 4-16
CONFIGURATION 2, STAGGERED RAIL TEST WITH 3.0 INCHES
SUPERELEVATION, LADING SHIFTED

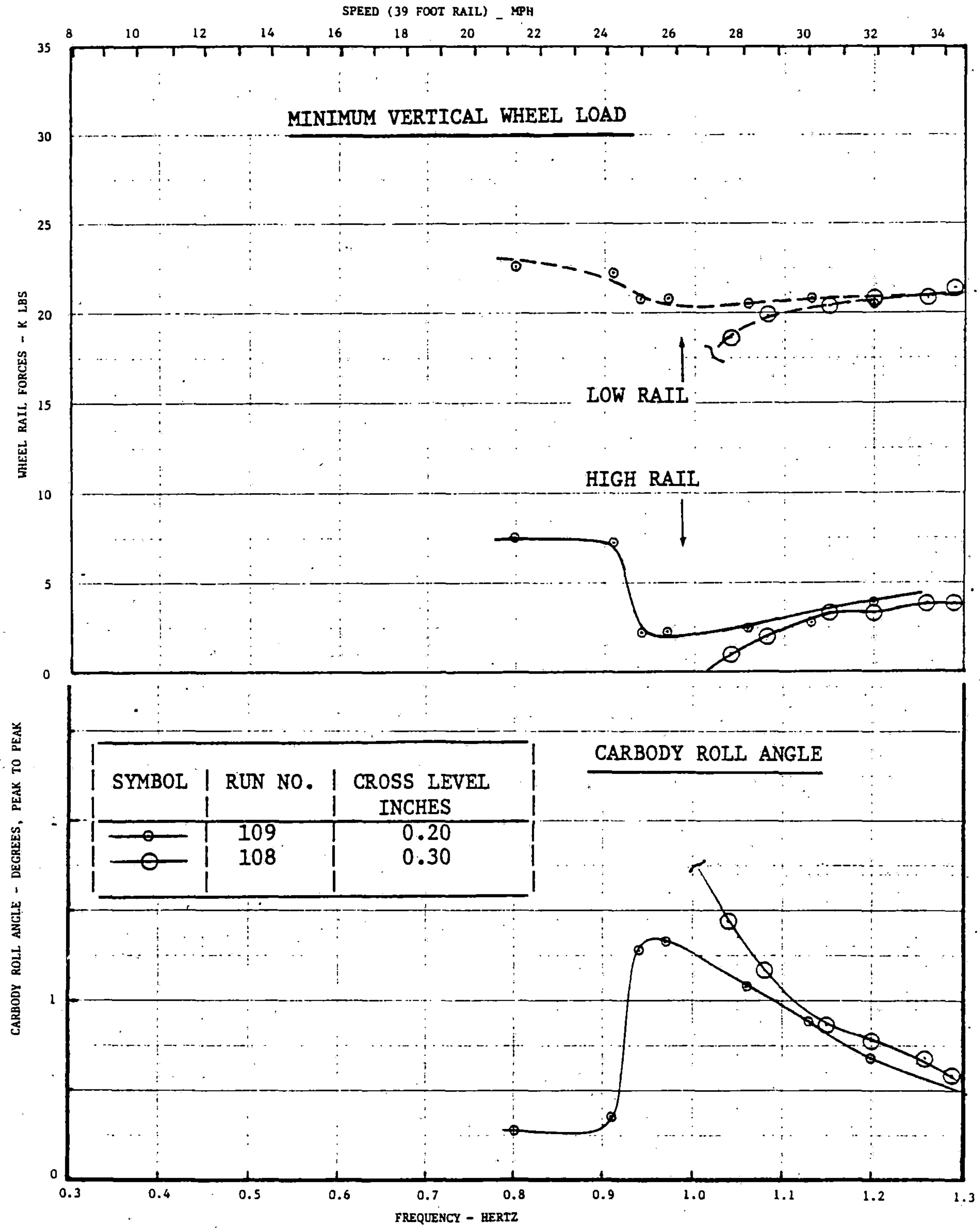


FIGURE 4-17
CONFIGURATION 3, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED

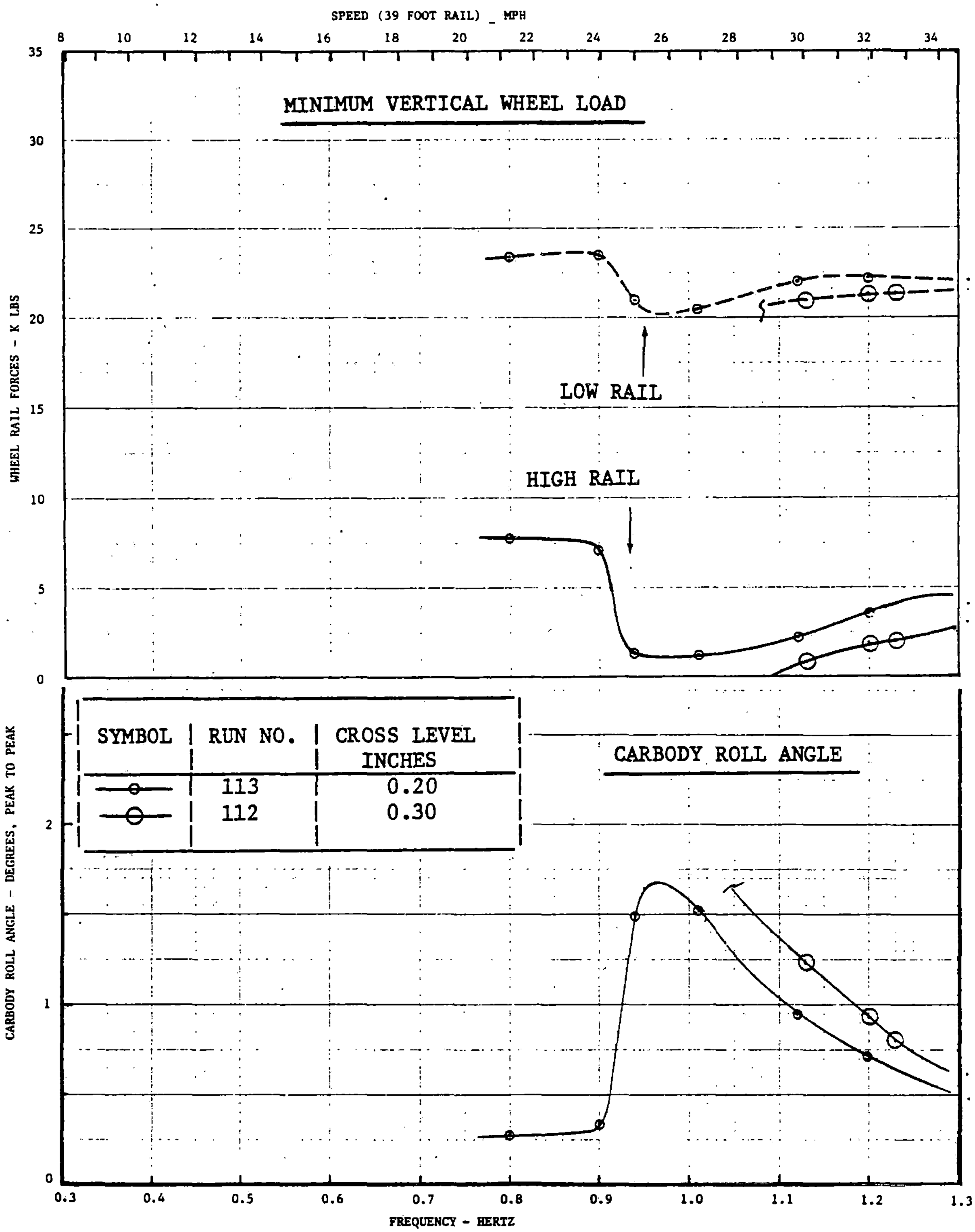


FIGURE 4-18
CONFIGURATION 4, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED

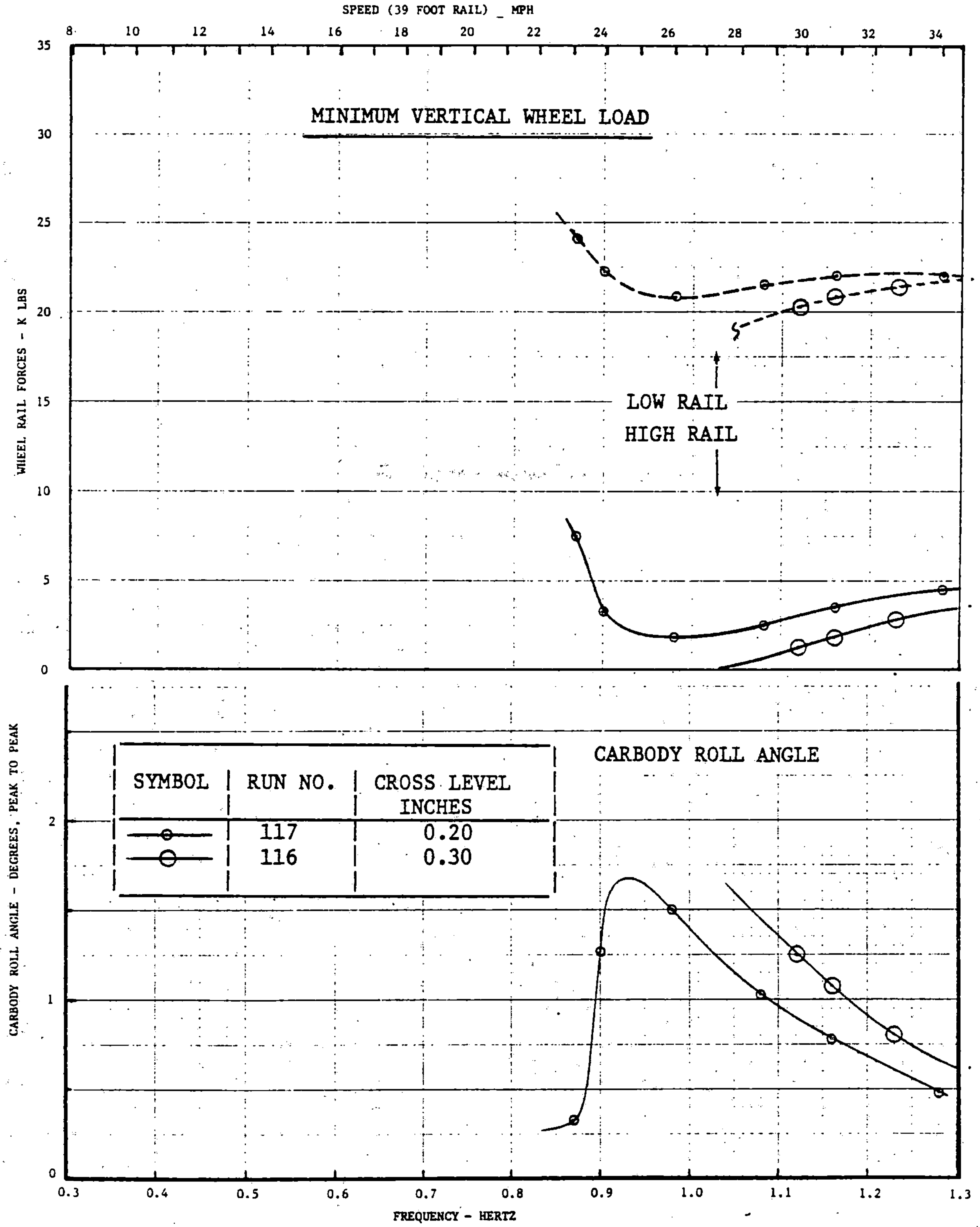


FIGURE 4-19
CONFIGURATION 5, STAGGERED RAIL TEST WITH 3.0 INCHES
SUPERELEVATION, SHIFTED LADING

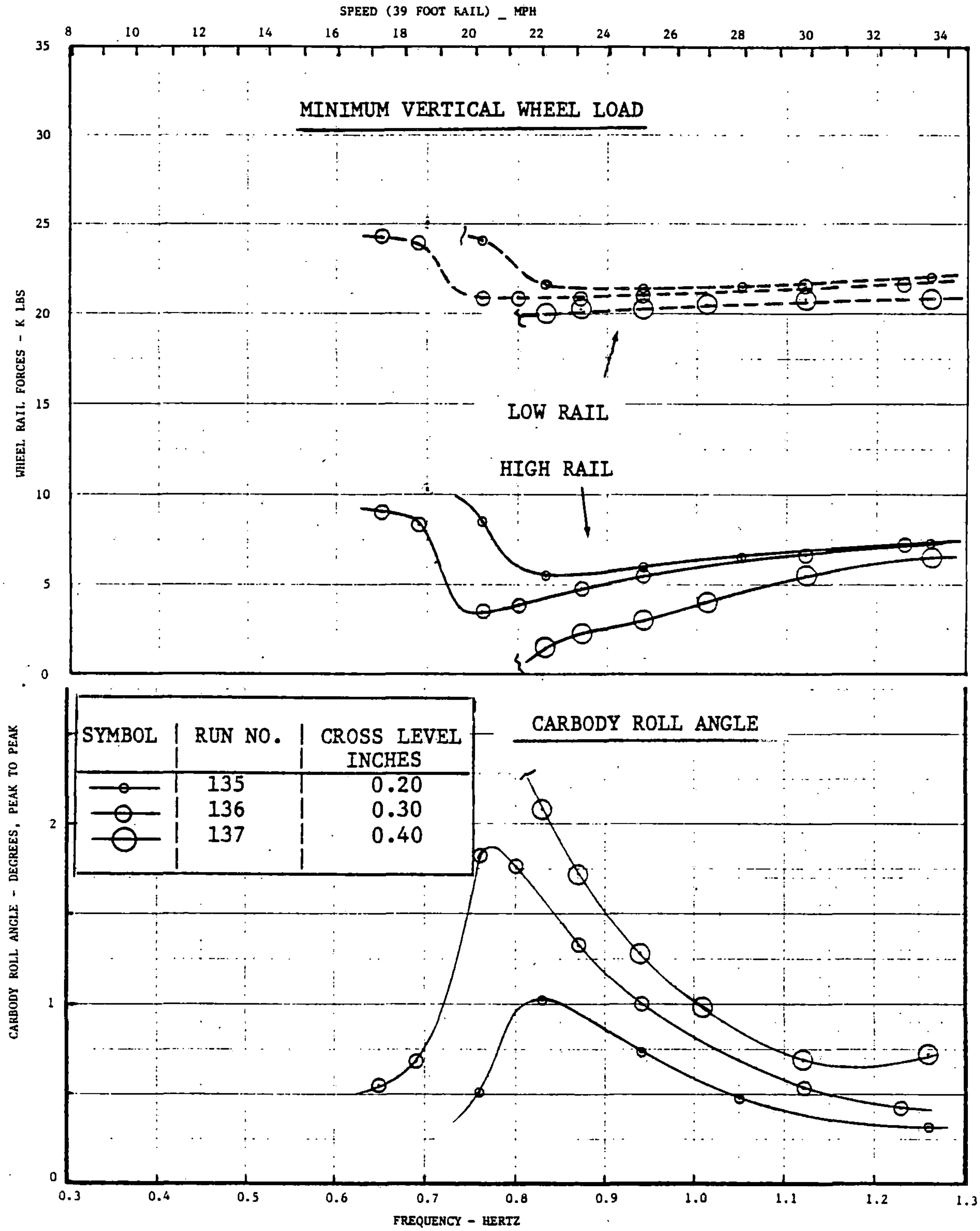


FIGURE 4-20
CONFIGURATION 6, STAGGERED RAIL TEST WITH 3.0 INCHES
SUPERELEVATION, LADING SHIFTED

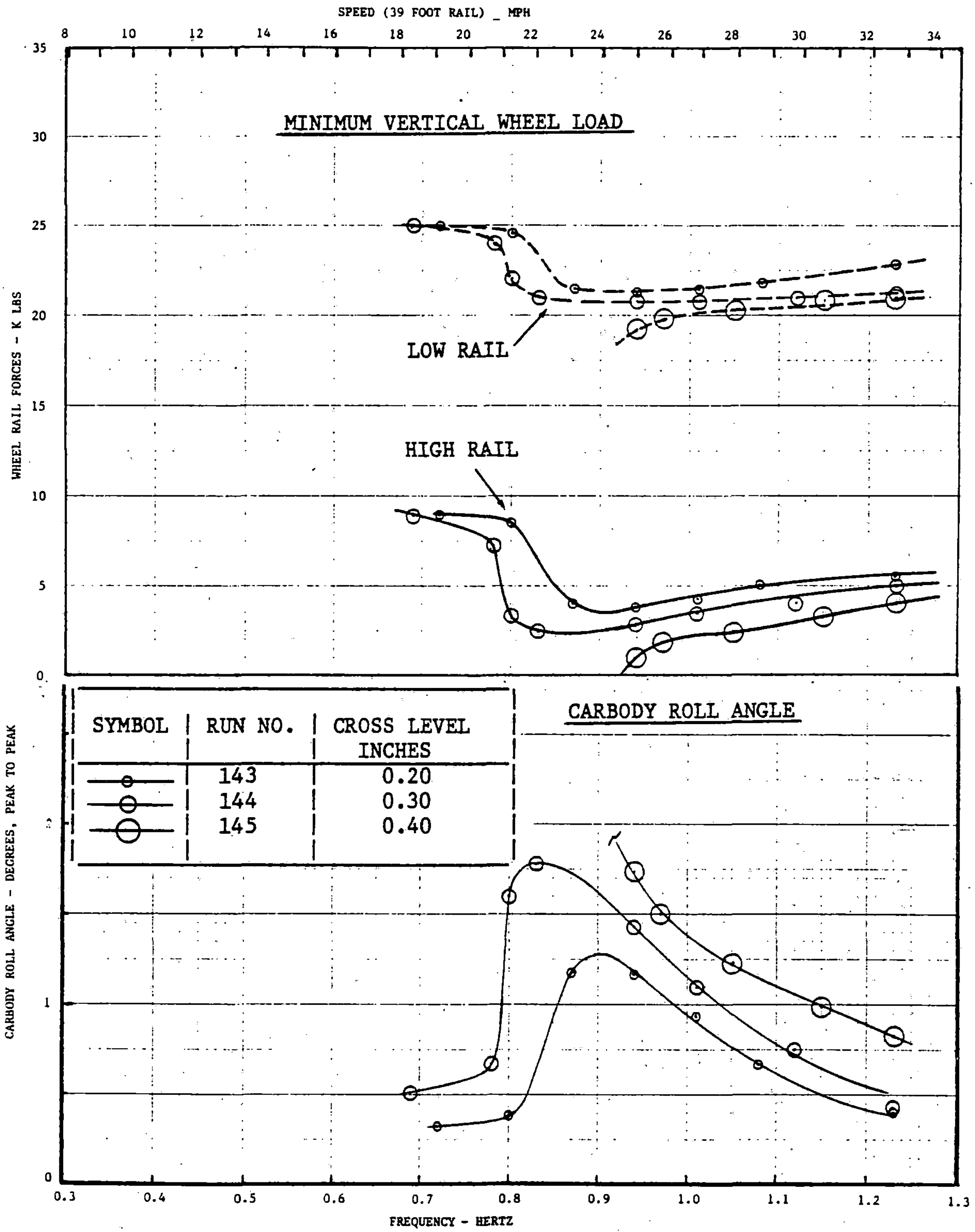


FIGURE 4-21
CONFIGURATION 7, STAGGERED RAIL TEST WITH 3.0 INCHES
SUPERELEVATION, LADING SHIFTED

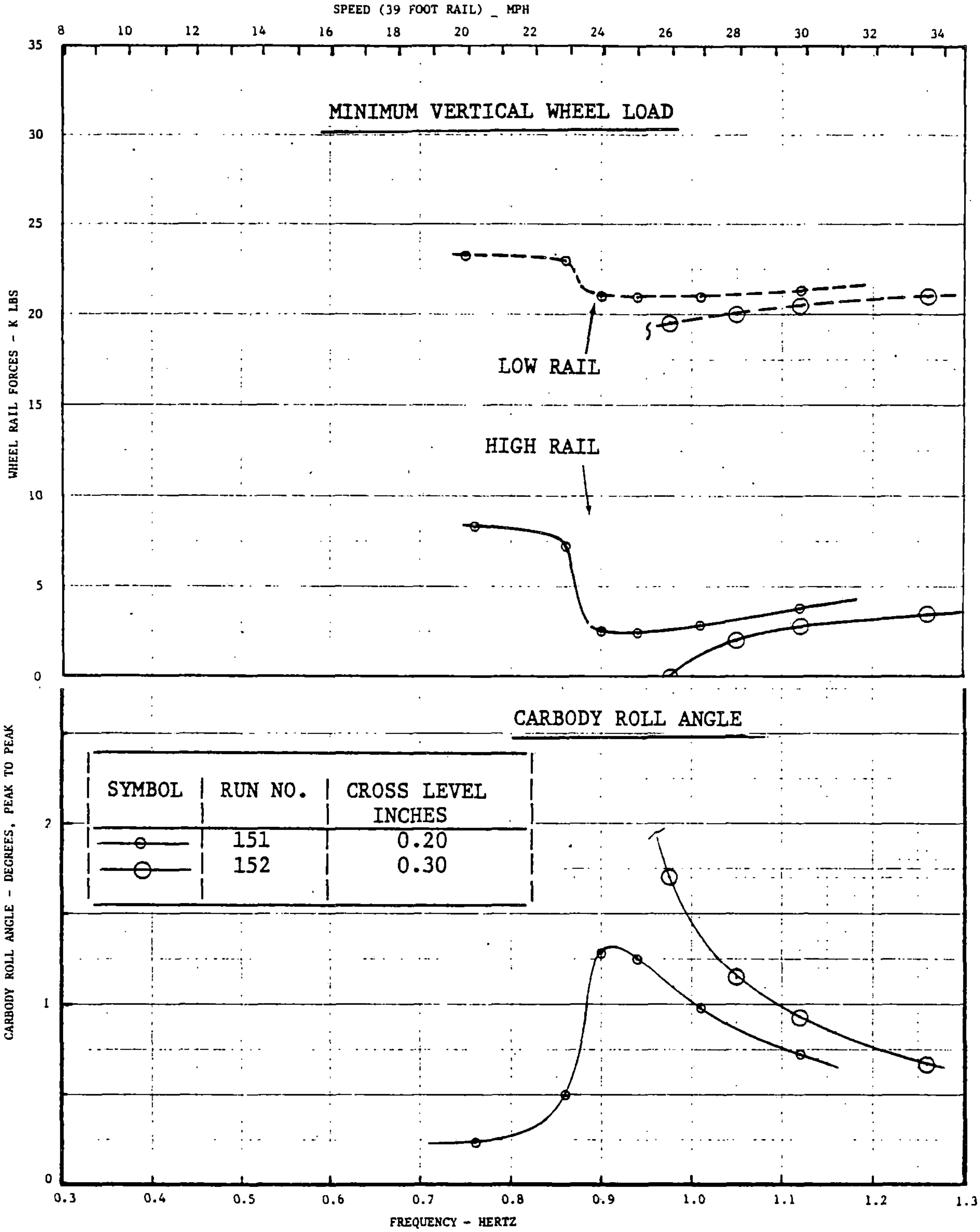


FIGURE 4-22
CONFIGURATION 8, STAGGERED RAIL TEST WITH 3.0 INCHES SUPERELEVATION, LADING SHIFTED

load occurred between 0.30 and 0.40 inches. With Configurations 3, 4, 5, and 8 zero wheel load occurred between 0.20 and 0.30 inches cross level. Configurations with greater damping generally had poorer performance.

The conclusion drawn was that wheel lift will very probably occur if a 70 ton boxcar loaded with plywood negotiates curved track with 0.40 inch low joints at sub-balance speeds. This assumes the plywood is shifted, which is a certainty based on tests performed here, and that the shift is to the same side as the low rail in the curve, which is a 50/50 chance.

There was no friction/hydraulic damping configuration that showed significant improvement in performance. The results did indicate that there is an optimum amount of damping and that most of the configurations tested may be beyond that optimum.

4.3 Track Geometry Response Data

Power Spectral Density (PSD) plots of in.^{2}/Hz and g^2/Hz input displacements and response acceleration from the TG tests were processed at the RDL. Figure 4-23 shows typical PSD spectra of input displacement for the four runs 56 through 59 which are with Configuration 1, shifted plywood, for the input levels of 1.25, 1.50, 1.75, and 2.00. Runs 56, 57, and 58 are seen to have the same spectra with expected increased levels while run 59 is seen to have a change in spectrum which is typical of the problem noted earlier (see the footnote in Table 3-3). Figure 4-24, which is presented as typical accelerations response PSD spectra, is from run 50, Configuration 1 with centered lading at the 1.50 level. The top plot is A8Z which is a vertical accelerometer on top of the plywood at the A end. The bottom plot is A9X which is a lateral accelerometer at the same location. The spectra of both PSD plots appear to have four frequency bands of increased response: 0.9-1.1, 2.0-2.5, 8.5-9.5, and 14-16 Hertz.

In an attempt to show the relative responses between Configuration 1 and Configuration 2, and between centered and shifted lading, the summary plots of Figures 4-25 through 4-32 were developed: a separate figure for vertical and lateral response for each of the four frequency bands mentioned above is shown. However, a review of this data showed no significant nor consistent differences between centered and shifted plywood, insofar as accelerations response was concerned.

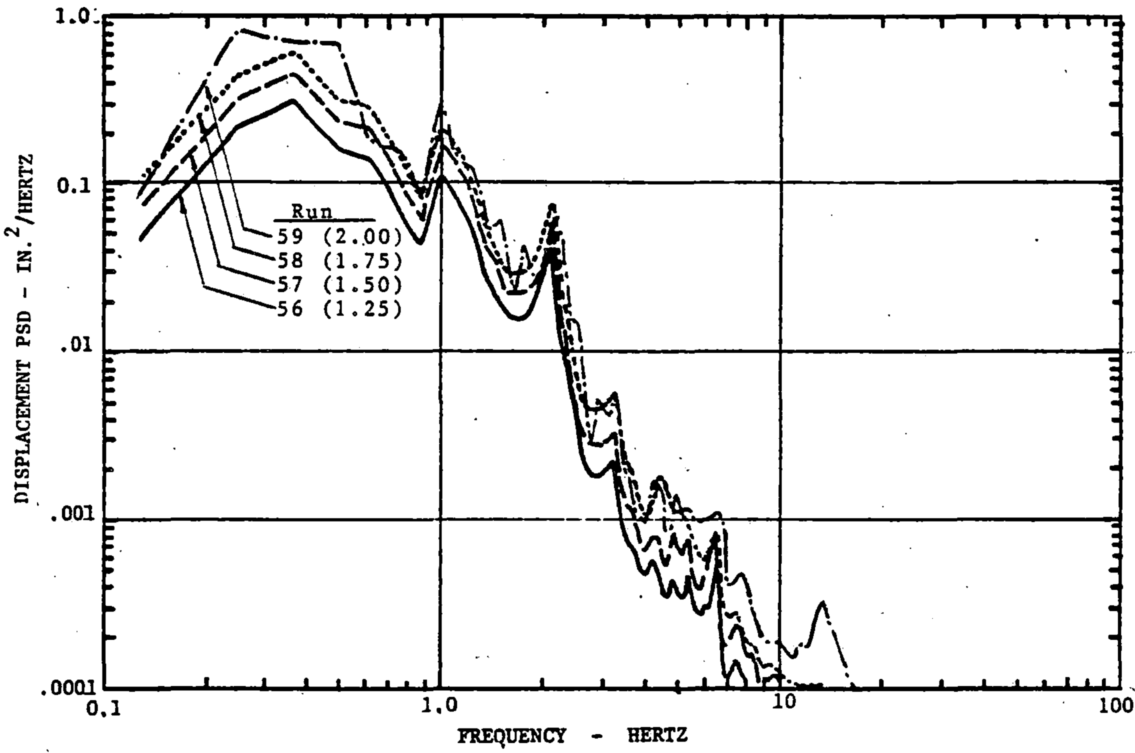
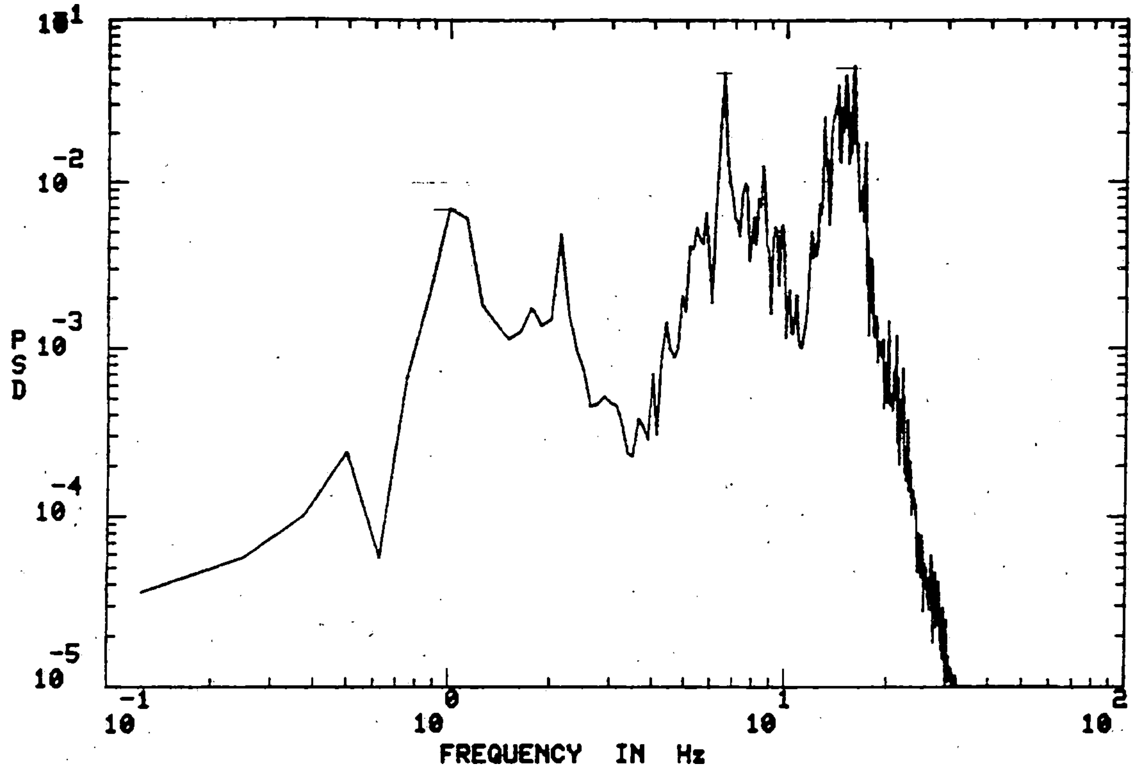
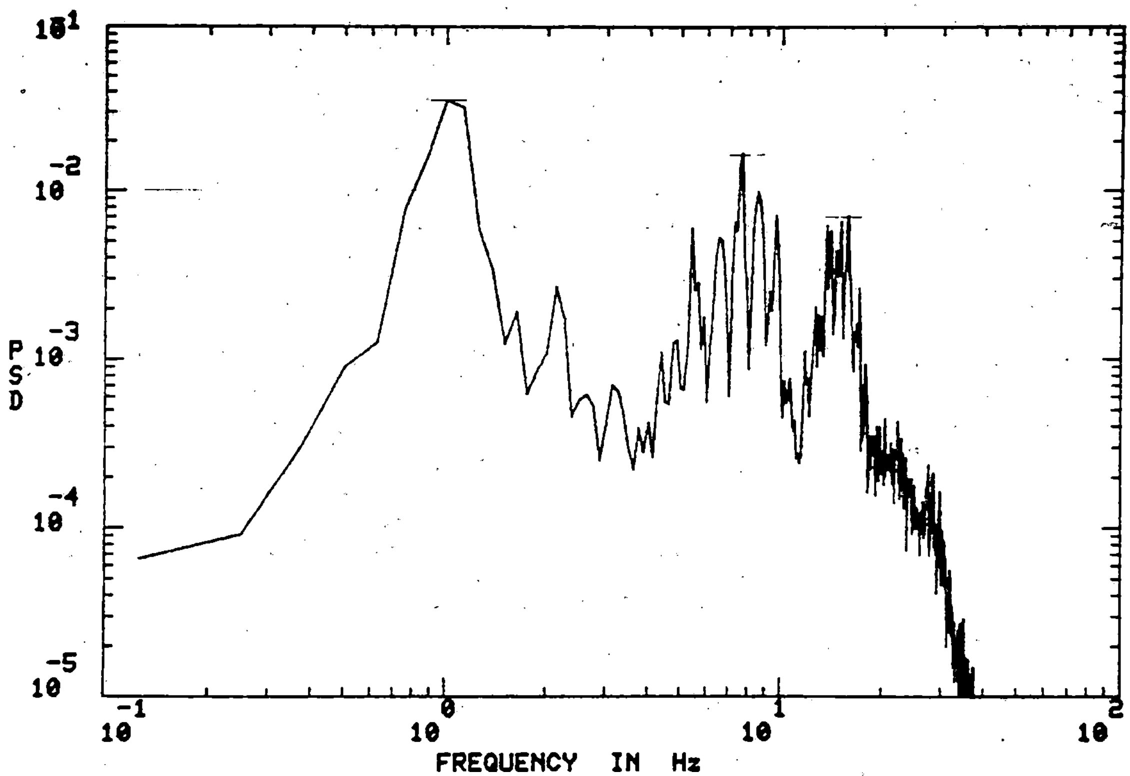


FIGURE 4-23
TYPICAL INPUT DISPLACEMENT PSD SPECTRA TRACK GEOMETRY RUNS



VTU PLT SAFTY MAR A8Z(C057) BLKS (27-37)[SECS 220-300]RUN 50



VTU PLT SAFTY MAR A9X(C058) BLKS (27-37)[SECS 220-300]RUN 50

DATE: 6/17/82

FIGURE 4-24
TYPICAL RESPONSE ACCELERATION PSD SPECTRA
TRACK GEOMETRY RUN 50

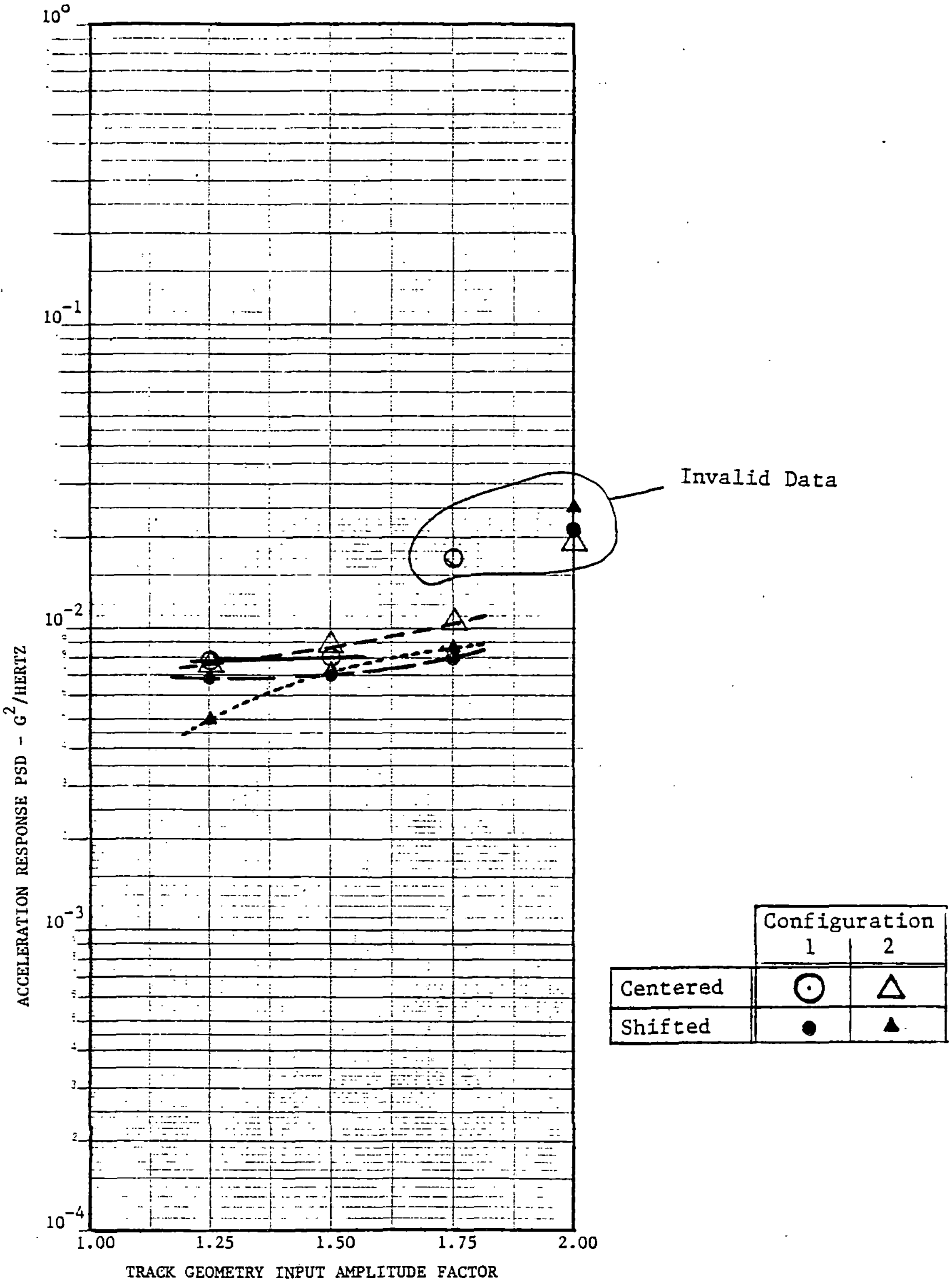


FIGURE 4-25
SUMMARY PSD DATA, PEAK VERTICAL RESPONSE,
0.9 - 1.1 HERTZ

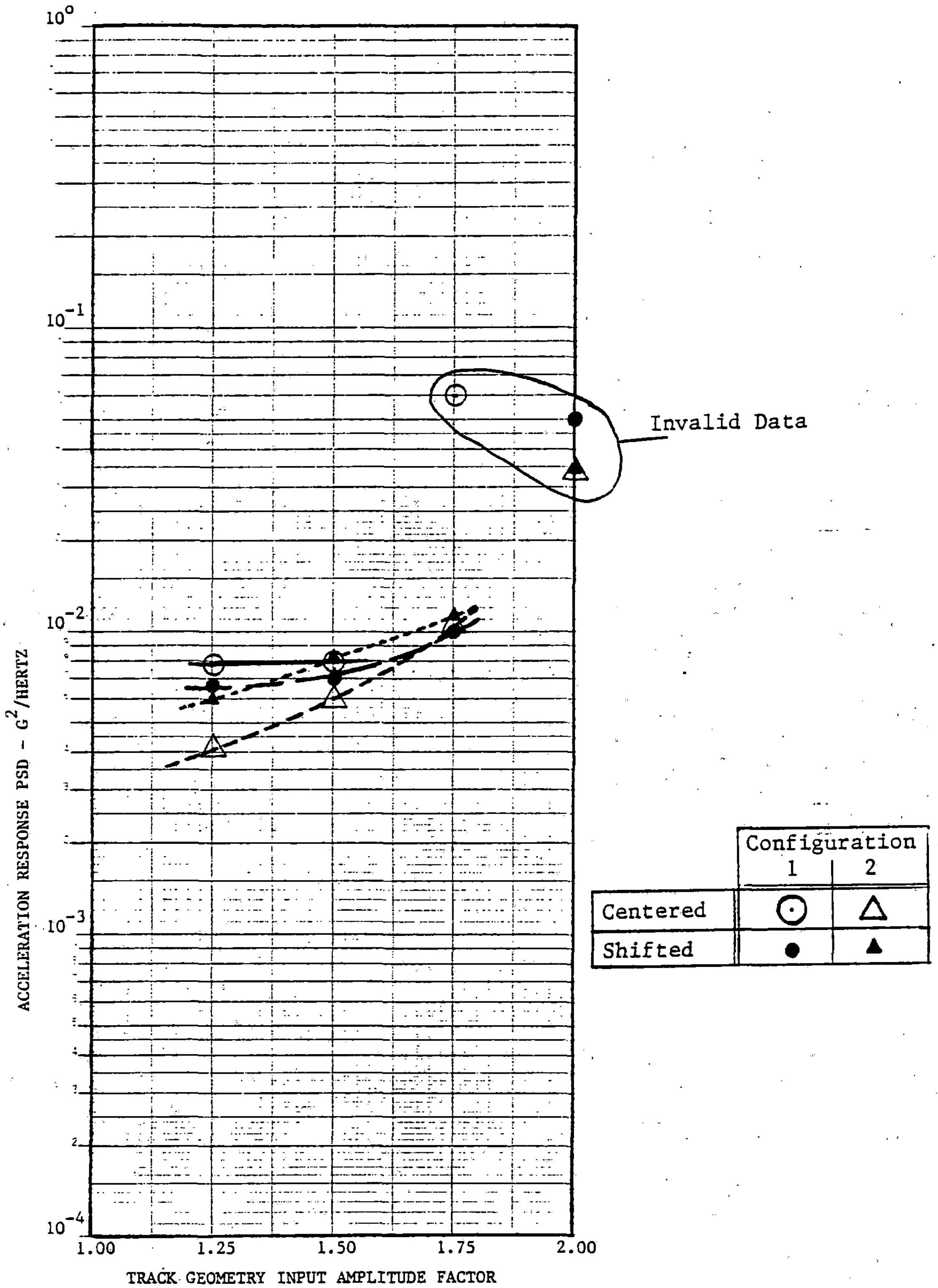


FIGURE 4-26
SUMMARY PSD DATA, PEAK VERTICAL RESPONSE,
2.0 - 2.5 HERTZ

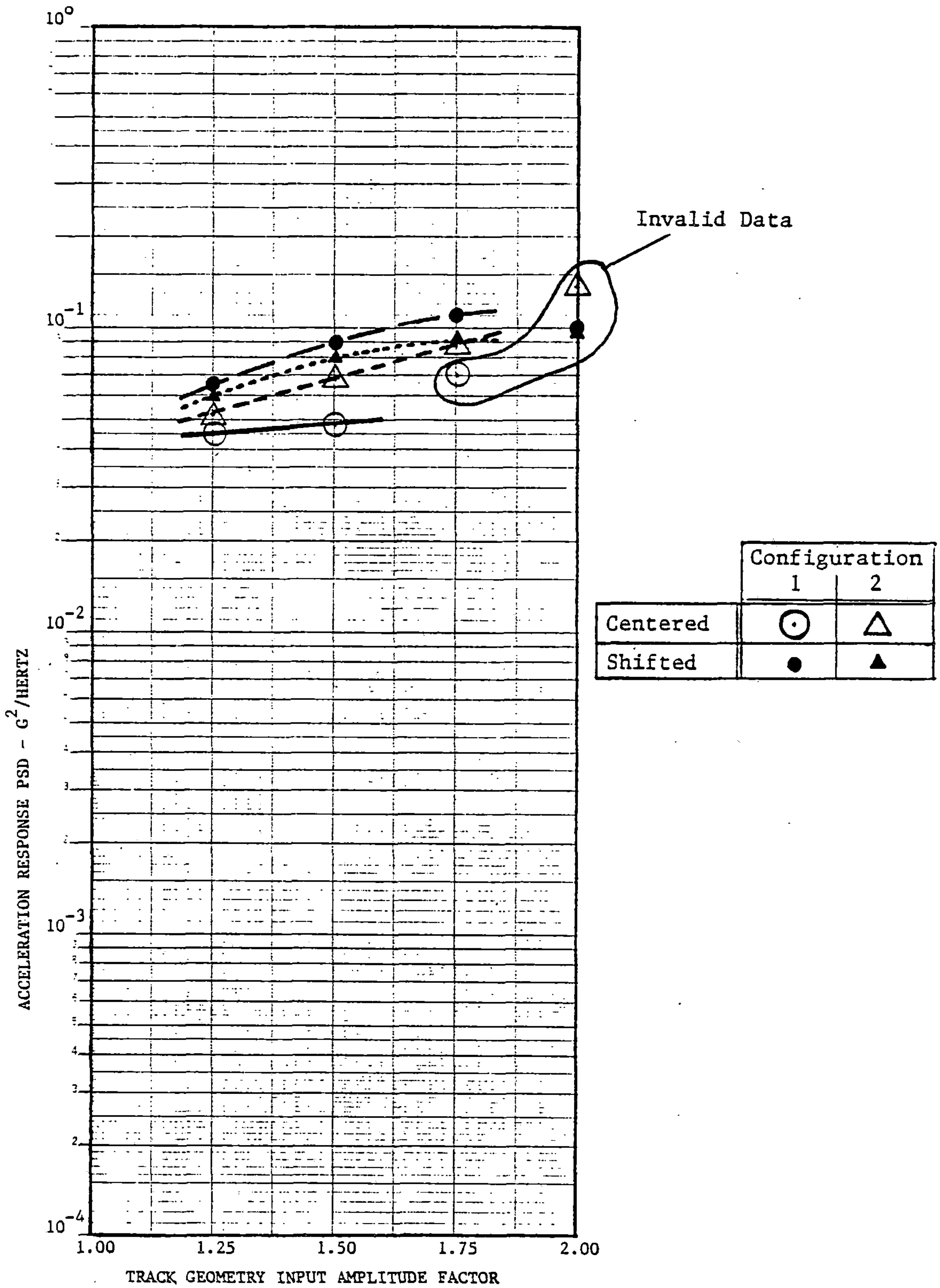


FIGURE 4-27

SUMMARY PSD DATA, PEAK VERTICAL RESPONSE, 8.5 - 9.5 HERTZ

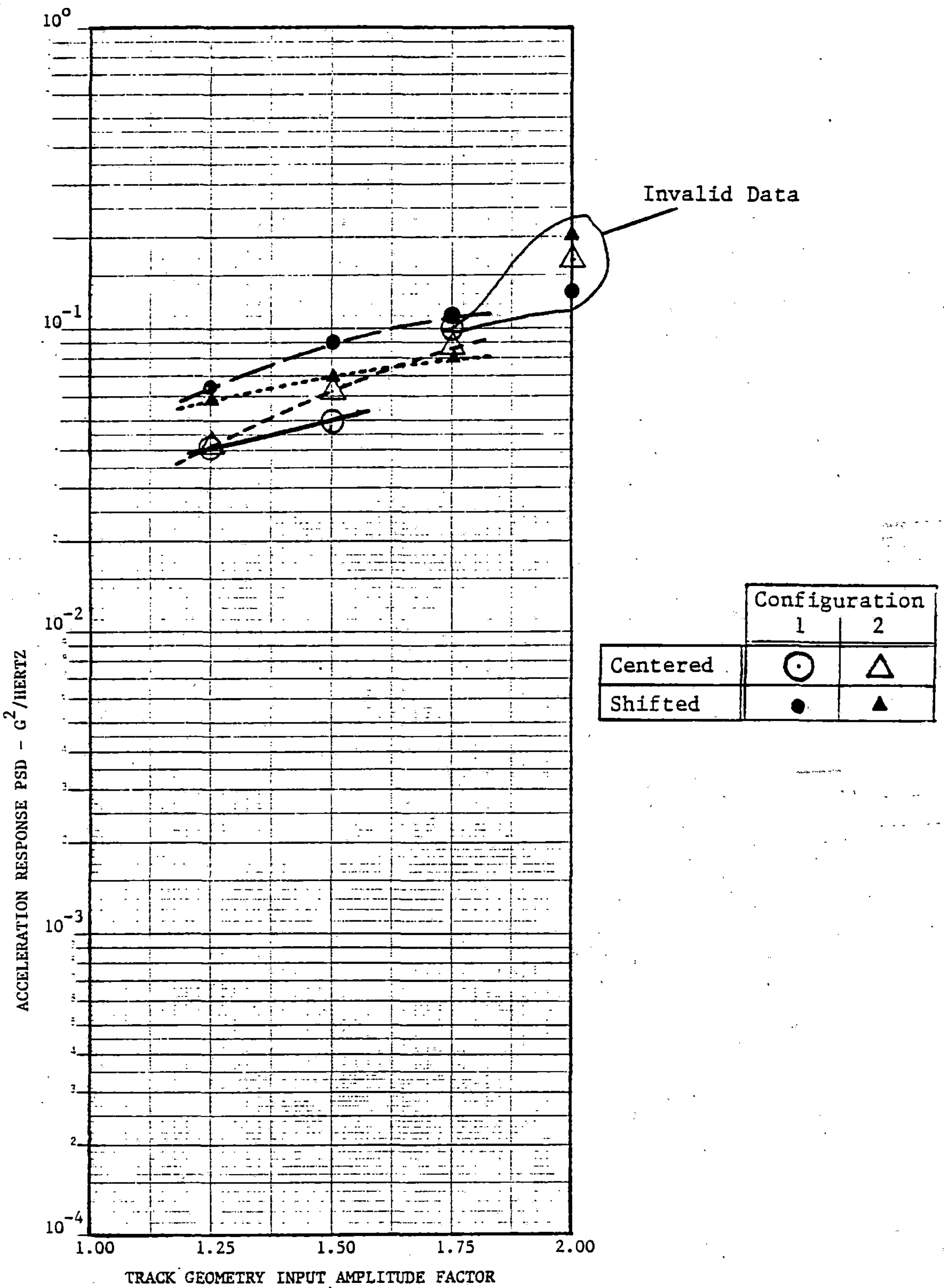


FIGURE 4-28
SUMMARY PSD DATA, PEAK VERTICAL RESPONSE,
14 - 16 HERTZ

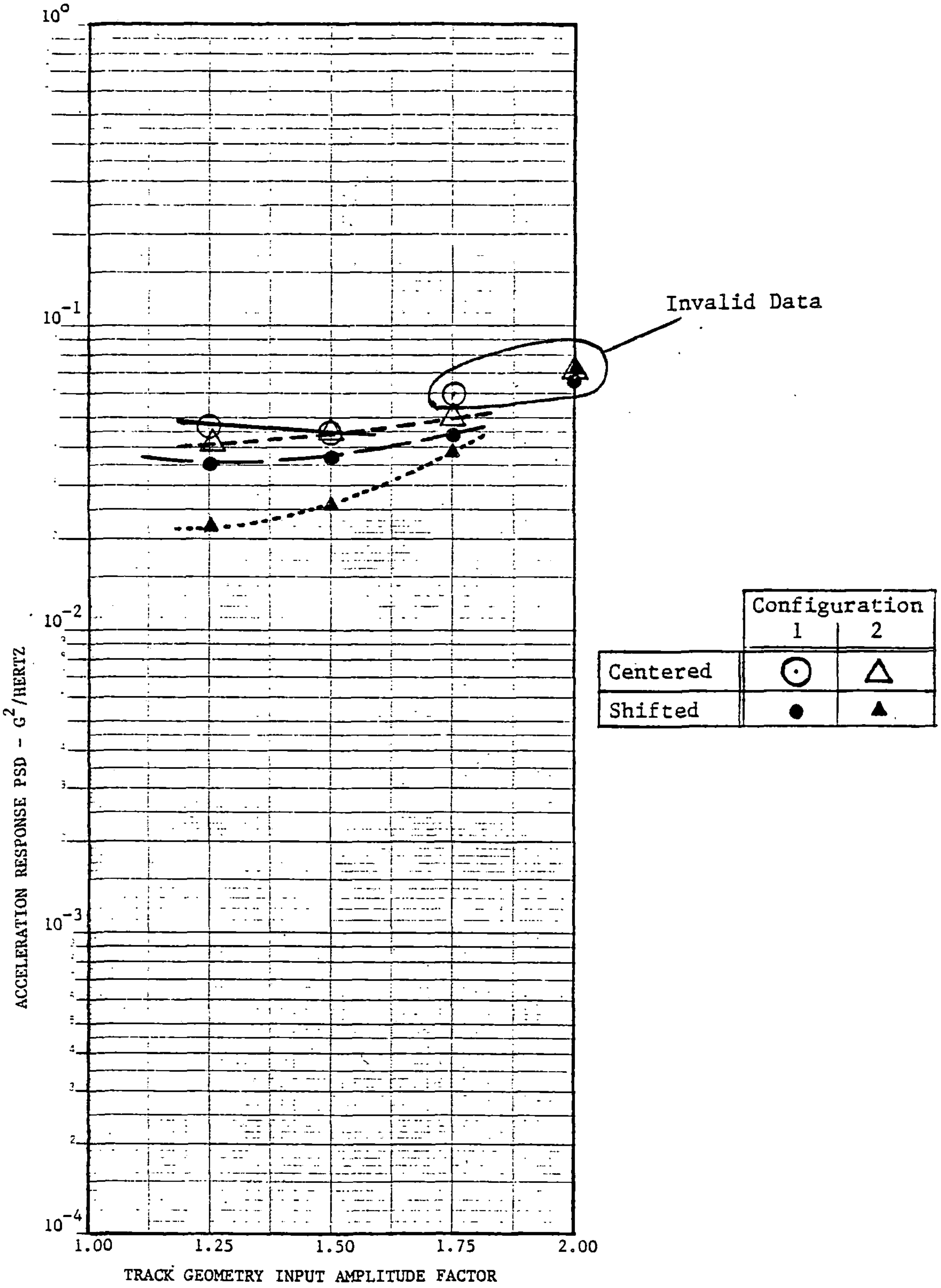


FIGURE 4-29
SUMMARY PSD DATA, PEAK LATERAL RESPONSE,
0.9 - 1.1 HERTZ

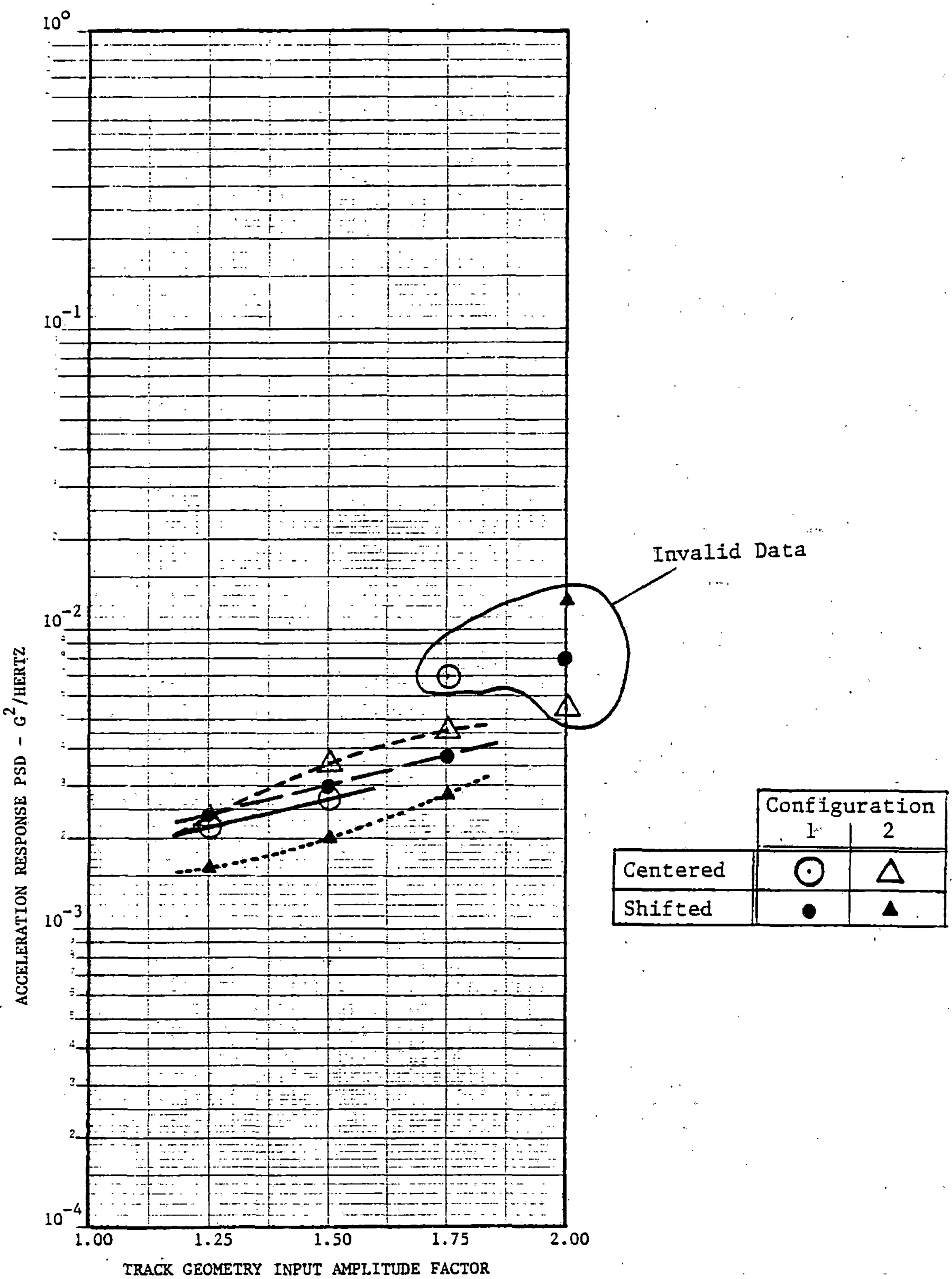


FIGURE 4-30
SUMMARY PSD DATA, PEAK LATERAL RESPONSE,
2.0 - 2.5 HERTZ

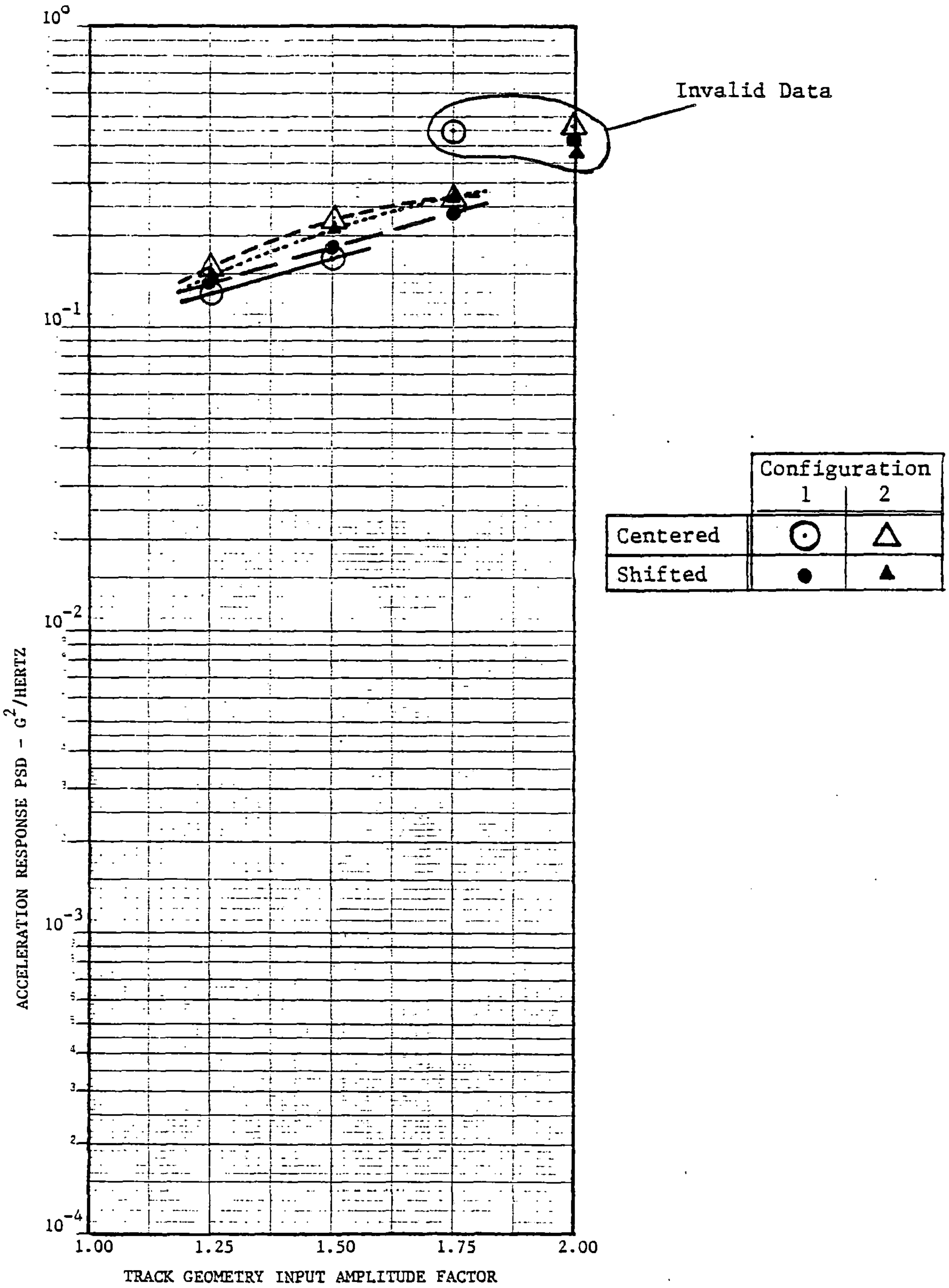


FIGURE 4-31
SUMMARY PSD DATA, PEAK LATERAL RESPONSE,
8.5 - 9.5 HERTZ

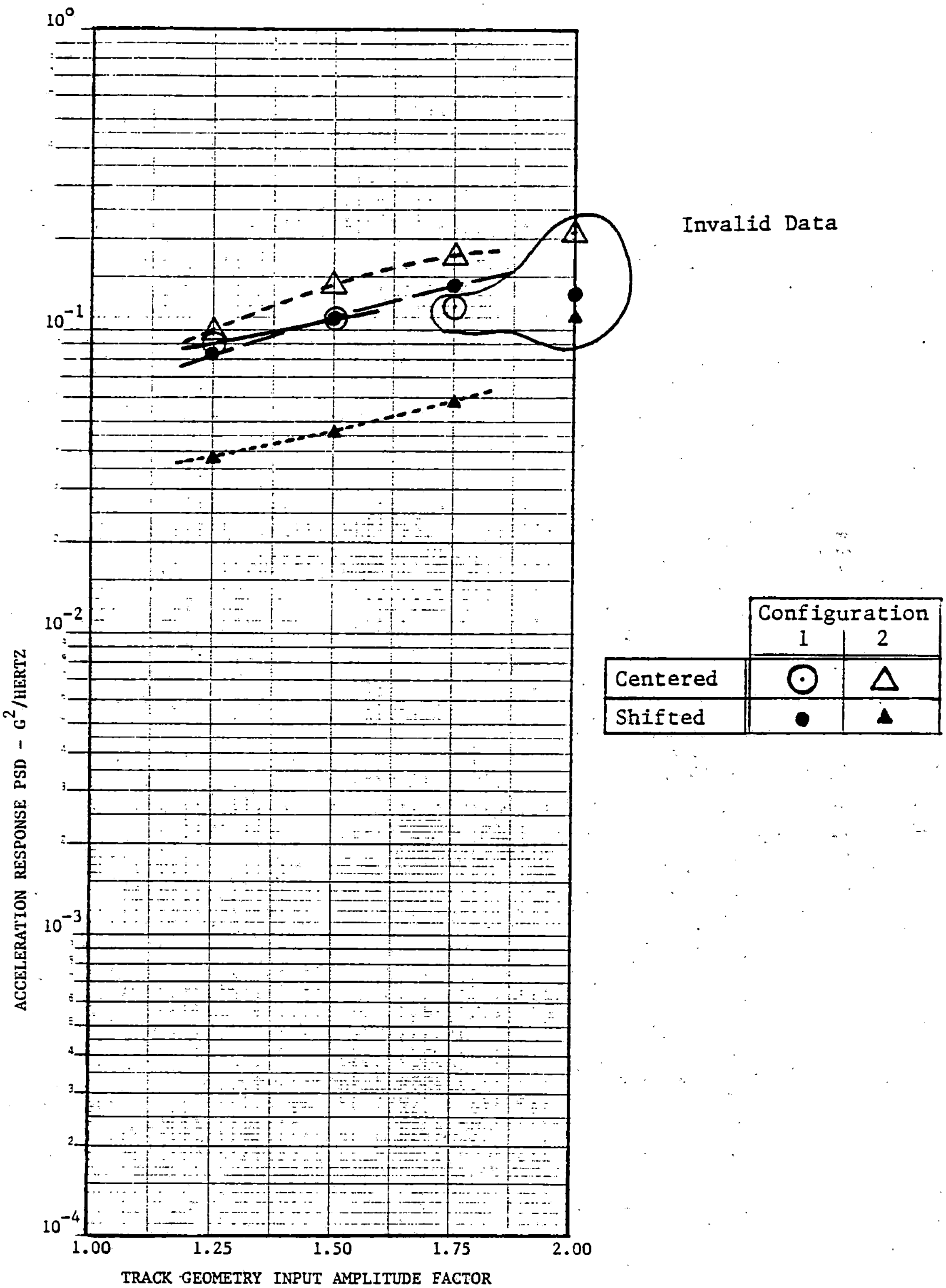


FIGURE 4-32
SUMMARY PSD DATA, PEAK LATERAL RESPONSE
14 - 16 HERTZ

One final view of the TG, PSD data was taken by an approximate conversion of PSD to g's rms. This was done for the four frequency bands discussed above (0.9-1.1, 2.0-2.5, 8.5-9.5, and 14-16 Hertz) using the relationship:

$$g_{rms} = (PSD \times \Delta f)^{1/2}$$

where Δf = frequency band between the half power points of a PSD peak, Hertz

PSD = the peak PSD, g^2/Hertz

g_{rms} = root mean square acceleration.

The results, shown in Figure 4-33, are the maxima from all the TG runs with Configurations 1, and 2, and with lading centered and shifted. There are two observations that can be made. First, the acceleration response increase from the 1.25 to the 1.75 level is about 40 percent for three of the four frequency bands. In the fourth band, 0.9-1.1 Hertz, accelerations responses change very little between the 1.25 and 1.75 levels possibly because this is the roll mode response and is in a nonlinear condition.

The second observation is that the significant acceleration levels are in the 14-16 Hertz band. These levels run from 0.55 to 0.75 g rms or occasional peak values of about 2.0 g. This is significant in that these data represent track speeds of 30 mph and although out of the speed range of harmonic roll, there is enough bounce and jiggle of the plywood stacks that small amounts of asymmetry and lateral forces will result in lading shift.

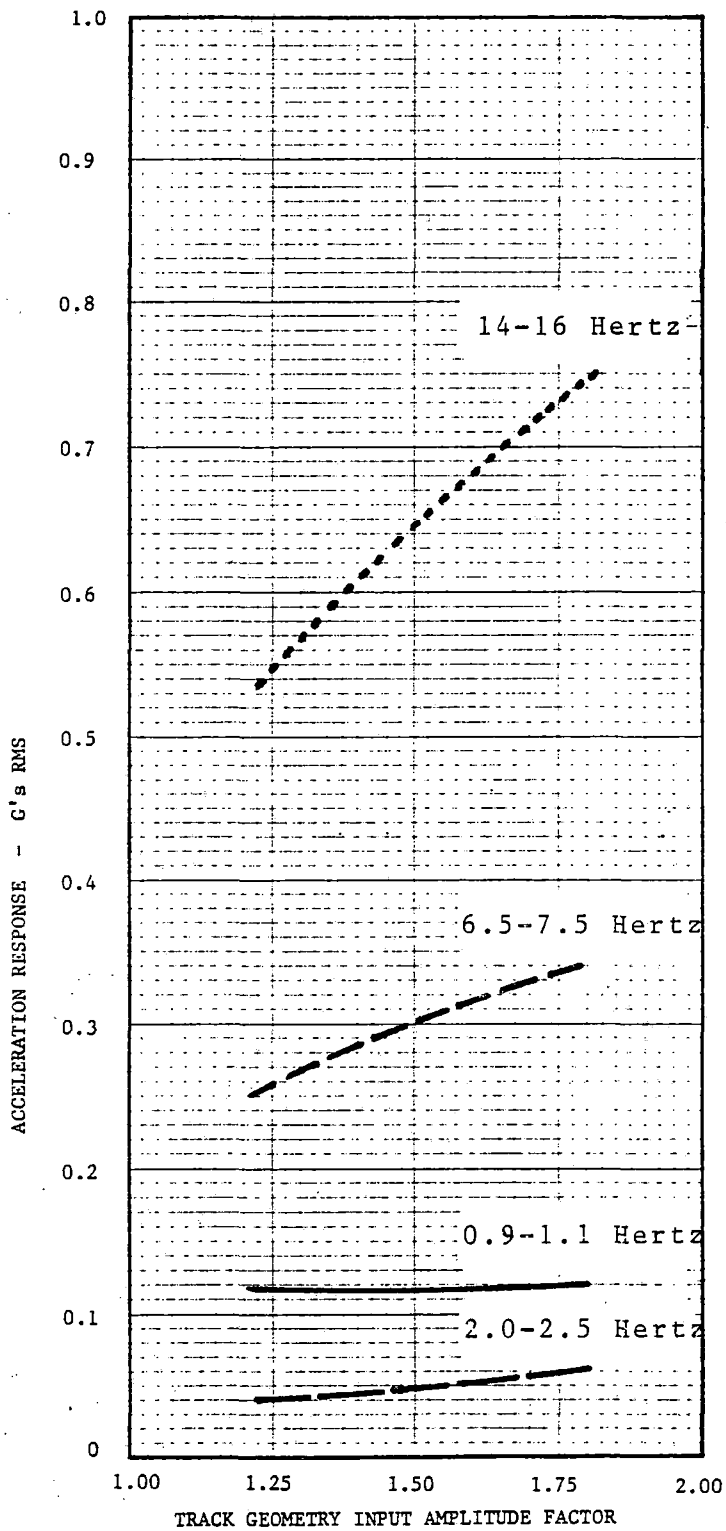


FIGURE 4-33
TRACK GEOMETRY RESPONSE SUMMARY, RMS ACCELERATION
ESTIMATED FROM PSD PEAKS

5. CONCLUSIONS

Threshold Levels for Plywood Lading Shift

The test results showed, that with normal over the road conditions, plywood lading will shift laterally to the point of contacting the side wall. The changes in snubber configuration did not show significant improvement in the tendency to shift.

Derailment Margin

Zero wheel load was adopted as the point of derailment zero margin. A quantitative assessment of the loss in margins due to lading shift could not be made because the lading shifted from the centered position in the test performance before the zero wheel load level could be determined.

With the plywood shifted and tangent track conditions, zero wheel load conditions were reached with 0.60 inch cross level using rectified sine profile shapes. Therefore, it is very likely that this configuration would also not pass the AAR's 3/4 inch shimmed track test for special devices.

On superelevated track, with the assumptions that the train speed is below the balance speed of the curve and that the plywood is shifted to the same side as the low rail, the zero margin cross level is 0.30 inches.

The conclusion can consequently be made that curved track with staggered rail low joints of 0.30 inches or more will cause wheel lift in plywood ladened boxcars.

There are other factors that could not be included in this study which will make conditions worse. Train action and drag loads will have load components that will tend to roll the car towards the low rail. The dynamic motions induced by curve entry transients will start the roll motion so that only a few low joints will result in maximum roll response.

Snubber Variation

The better performances resulted with configurations having low force snubbers. However, the testing was not extensive enough to quantify the optimum range or type of damping.

APPENDIX A

LIST OF REFERENCES

1. Harry A. Grosso, Vibrations Test Unit-Pilot Test, AAR Test Requirements, Association of American Railroads letter, January 27, 1982, File: G948.

2. George Kachadourian, FRA Test Requirements, VTU Pilot Test, MITRE Corporation Working Paper No. WP-82W00001, January 1982.

3. Wade D. Dorland, et al, Implementation Plan, Rail Dynamics Laboratory, Vibration Test Unit Pilot Test, Report No. VTU-IP-82-03-03, March 1982.

4. Britto Rajkumar and Firdausi Irani, VTU Pilot Program, Validation of the Track Geometry Input to the Vibration Test Unit (VTU). Boeing Services International, Transportation Test Center, Pueblo, Colorado, July 26, 1982.

5. Britto Rajkumar and Firdausi Irani, Validation of the Vibration Test Unit Endurance Capability, Vibration Test Unit, Boeing Services International Transportation Test Center, Pueblo, Colorado, August 18, 1982.