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16. Abstract

In an effort to better understand the practical aspects of the truck hunting response of two typical freight vehicles, a Federal Railroad Administration (FRA) test program was conducted at the Transportation Test Center (TTC). Axle and car-body accelerations were measured on a conventional design coal hopper car and an 89-foot intermodal flat car. Variables introduced into the test design included loaded weight on rail, truck design, side bearing configuration, and wheel profile. All test runs were conducted on the Railroad Test Track (RTT) under completely dry conditions.

On those runs calling for the CN Heumann wheel profile, instrumented wheelsets were used. This data was used to define the wheel/rail loads produced during truck hunting. A further effort was made to determine the force/acceleration relationship for a limited number of test configurations.

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EXECUTIVE SUMMARY

The objective of this report is to document the results of vehicle stability tests at the Transportation Test Center (TTC). This was carried out in order to evaluate vehicle behavior on good tangent track at relatively high speeds (up to 70 mph). By performing the truck hunting tests for lateral stability, a further understanding of the typical force-acceleration environment at the wheel/rail interface was possible. These tests also required the measurement of axle and carbody acceleration as well as the use of instrumented wheelsets to quantify wheel/rail forces during truck hunting.

It was recognized at the time these tests were being planned that a similar investigation was being requested by the Technical Advisory Committee for the Facility for Accelerated Service Testing (FAST) Program. By combining the technical objectives of both the Vehicle Track Interaction (VTI) and FAST Projects, significant economies were realized.

The purpose of the work contained in this document is to report on a series of tests on a coal hopper car and an 89-foot intermodal flat car which were carried out to evaluate factors which affect hunting.

Variables which were investigated include:

- o Wheel Profile
- o Axle Load
- o Side Bearing Configuration
- o Truck Design

The major emphasis was to identify the critical speeds at which hunting was initiated for the various categories and any important interaction within these categories. Also, the relative accelerations and wheel/rail forces at a range of test speeds were studied. An auxiliary set of instrumented coupler measurements were taken to quantify the relationship between rolling resistance and hunting. This effort was sponsored by the AAR Energy Research Program in cooperation with the FRA.

A second objective was to identify a general hunting (vehicle dynamics) model and compare measured and predicted response for the test conditions.

The overall utility of the data base resulting from the testing is twofold. One is to provide sound statistical results from the on-track testing. This provides comparative data that may be of use to the railroad community. The second is to have available a data base for the validation of vehicle dynamics models. Ultimately, the data may then play a role in car design, safety, maintenance, and energy studies.

The intermodal and coal hopper basically reacted in a similar fashion to changes in the test variables. A reasonable relationship between axle load and hunting threshold speed was identified. It indicated that by increasing the weight on rail by one-third of the test cars' capacity, hunting on-set speed was generally outside of the range of normal operating speed even for the most hunting-prone configurations. The on-set speed was increased by approximately 15 to 20 miles per hour with this additional lading.

The influence of using properly installed constant contact side bearings was to increase the on-set speed by 10 to 15 miles per hour. This is similar to what has been experienced for past tests. The use of instrumented wheelsets also demonstrated an interesting feature of the constant contact side bearing. This was the reduction of the peak lateral force for a given acceleration due to an apparent decoupling of the masses within the vehicle.

Promising results were generated by a relatively simple stability model. Although the trucks were not characterized for this test program, widely used truck characteristics generated good agreement between the test and model data for some runs. Particularly good correlation was found for those cases using roller-type side bearings. Those with constant contact side bearings were difficult to evaluate since tests were run only to 70 miles per hour and, at that point, there was generally no hunting activity.

The use of a retrofit radial truck eliminated any tendency to hunt up to the 70 mile per hour maximum test speed.

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1.0 INTRODUCTION

1.1 BACKGROUND

The objective of the Federal Railroad Administration (FRA) sponsored Vehicle/Track Interaction (VTI) Research Program was to deal with locomotive and freight car response to large lateral and vertical variations in track geometry[1] and to evaluate truck hunting on specific freight cars. It was part of an ongoing effort to establish performance-based safety specifications for track. Testing was carried out in conjunction with the Transportation Systems Center (TSC) to evaluate the response to cross-level and alignment and various rail restraint anomalies.

Measurements of wheel/rail forces, carbody accelerations, and carbody component motions were recorded. Predictions of vehicle response were carried out at TSC. Results of this investigation are covered in detail in the final FRA report.

The objective of this portion of the project, which is the subject of this report, was for the Association of American Railroads (AAR) to perform additional vehicle stability tests on the VTI test cars at the Transportation Test Center (TTC). This was to be carried out in order to evaluate vehicle behavior on good tangent track at relatively high speeds (up to 70 mph). By performing the hunting tests for lateral stability, a further understanding of the typical force-acceleration environment at the wheel/rail interface was possible. These tests also required the measurement of axle and carbody acceleration as well as the use of instrumented wheelsets to quantify wheel/rail forces during hunting.

It was recognized at the time that these tests were being planned that a similar investigation was being requested by the Technical Advisory Committee for the Facility for Accelerated Service Testing (FAST) Program. By combining the technical objectives of both the VTI and FAST Projects, significant economies were realized.

The VTI test vehicle used in the hunting tests was the 89-foot intermodal flat car. It was tested empty and loaded with 40-foot highway trailers. Also used in the hunting tests was a 100-ton capacity coal hopper from the FAST consist. Both vehicles were also tested under various loading conditions and with two different types of freight trucks. Instrumented wheels were used to measure wheel/rail forces.

1.2 APPROACH

Several past studies have focused on rail vehicle lateral instability on good tangent track.[2,3,4,5] The tests often focused on specific solutions to hunting, e.g., how side bearing modification might increase onset speed. A variety of organizations including railroads, car manufacturers, etc., sponsored these tests. As such, detailed results were not always available to the general public.

Other research has taken a more general approach by mathematically modelling the hunting stability of railway vehicles.[6,7,8,9] Although detailed in nature, the applicability of the models was sometimes left in question, since verification with controlled testing was not necessarily carried out. Another weakness was that past tests were done on a variety of tracks, with a range of vehicles, which made generalization for modelling extremely difficult. The work sponsored under the VTI task order took the approach of creating a large data base from several configurations and speeds. Control was exercised over these variables that have caused ambiguities in past tests. Using a single test track and vehicles with components in common from configuration-to-configuration, differences in behavior could be attributed to the variable of interest. Over a range of speeds, careful measurement was made of vehicle and axle accelerations and wheel/rail forces in order to ascertain overall performance.

Forces between wheels and rails have an effect which results in changes in track geometry leading to periodic and costly track maintenance. An additional concern is derailment potential. The degree to which hunting vehicles may be responsible for a disproportionate amount of maintenance and expense is unquantified.

The purpose of the work contained in this document is to report on a series of tests which were carried out to evaluate a range of typical car types with known characteristics which affect hunting.

Variables which were investigated include:

- Wheel Profile
- Axle Load
- Side Bearing Configuration
- Truck Design

The major emphasis was to identify the critical speeds at which hunting was initiated for the various categories and any important interaction within these categories. Also, the relative accelerations and wheel/rail forces at the range of test speeds were studied. An auxiliary set of instrumented coupler measurements were taken to quantify the relationship between rolling resistance and hunting. This effort was sponsored by the AAR Energy Research Program in cooperation with the FRA.

The VTI project was one of two vehicle performance tests carried out in the same general time frame in which the instrumented coupler was used. The intention was to verify its measurement integrity in a controlled test environment. Since it was basically a successful effort, the results proved interesting and have been included in this report as a contribution to the understanding of resistance of unstable vehicles.

A second objective was to identify a general hunting (vehicle dynamics) model and compare measured and predicted response for the test conditions.

The overall utility of the data base resulting from the testing is twofold. One is to provide sound statistical results from the on-track testing. This provides comparative data that may be of use to the railroad community. The second is to have available a data base for the validation of vehicle dynamics models. Ultimately, the data may then play a role in car design, safety, maintenance, and energy studies.

2.0 OBJECTIVES

2.1 HUNTING CHARACTERISTICS

The basic objective was to record, over a range of operating speeds, the response of the axles and carbodies in terms of accelerations and wheel/rail forces.

A major reason for doing a large scale hunting test, under varying, yet controlled vehicle conditions, was to generate a single data base which would accurately show the influence of a number of independent variables. Previous experience has shown that tests performed under seemingly similar sets of circumstances do not necessarily generate compatible data. This is presumably due to the subtle, yet significant differences, in truck suspensions, track geometry, and levels of wheel/rail adhesion, which may act individually or in combination with each other.

Examples of differences which might not be readily apparent can be attributable to the following factors:

- Varying states of wheel and/or rail wear generating differing contact geometry conditions.
- Differing track lateral stiffness and gage.
- Contact conditions which may include inconsistent friction levels due to sporadic lubrication or rail contamination due to dust or oxidation.
- Truck characteristics which can be altered by relatively small differences in spring stiffness and component wear.

Documenting hunting characteristics involves, among other things, identifying the onset speeds for hunting instability, which generally varies due to a number of major factors. Typically, lateral axle oscillations occur in the 2.0 Hz to 4.0 Hz range. Speeds beyond the onset produce proportionately higher amplitudes of sustained hunting at the same basic frequency.

Therefore, the basic objective was to record, over a range of operating speeds, the response of the axles and carbodies in terms of acceleration and wheel/rail forces.

2.2 CORRELATION OF WHEEL/RAIL FORCES WITH AXLE ACCELERATION

Several tests in the past have involved measuring the change in axle acceleration with increasing speed as a vehicle approached and exceeded the threshold hunting speed. However, the impact force at the wheel/rail interface was still unknown. A smaller number of tests have used load measuring wheelsets to record the lateral forces and L/V values during hunting. The basic relationship between accelerations and the forces at the wheel rail interface is still somewhat of an unknown.

The VTI hunting test presented an opportunity to simultaneously measure the accelerations and forces as the vehicles were tested through several speeds. This was due to the fact that both 33- and 36-inch instrumented wheelsets were readily available at the time that the tests were run.

It was of interest to explore the consistency and magnitude of the relationship between forces and acceleration. The objective was to deduce forces for uninstrumented wheels from acceleration measurement. A potential application of this technique would be in the area of profile testing. It is well recognized that the wheel profile plays a critical part in hunting behavior. Since it is impractical to cut a number of different wheel profiles on an instrumented axleset, estimates of hunting forces could be derived from the lateral accelerations. If a basic understanding could be developed between force and acceleration for a typical vehicle, the changes in wheel/rail force as a result of a variety of modifications could be calculated.

2.3 TRAIN RESISTANCE MEASUREMENTS

The resistance of a vehicle to forward motion increases at some proportion with train speed. This is under steady state conditions where no significant dynamics are present. These resistance forces include truck suspension, bearing, wheel/rail interface, and aerodynamic losses.

The speed at which the lateral instability, or hunting occurs contributes a source of resistance generally not considered. It is also a wheel/rail interface process; but, it is unknown to what extent the rolling resistance of a hunting vehicle changes. The review of measurements taken during the VTI tests provide some indication of the magnitude of this effect.

2.4 STABILITY MODEL COMPARISONS

The last objective of the VTI study was to produce predictions of hunting characteristics with analytical vehicle models. The intention was to evaluate the models in terms of accuracy, speed, and ease of use. Test cases would be as similar as possible to those tested on track.

The stability model used is a linear vehicle stability model.[10] It is capable of predicting the presence of instability in the truck/carbody system. This situation is analogous to hunting. The lowest speed at which instability is mathematically detected represents the prediction of hunting onset. This is achieved using an eigenvalue/eigenvector approach.[11] Input to the model can involve (depending on the number of degrees of freedom) a fairly simple description of the truck and car parameters. This includes dimension, weights, and suspension parameters. Wheel/Rail geometry is given in terms of a simple set of linearized contact conditions.

The second and more complex model can predict the vehicle behavior in terms of the lateral, vertical, and longitudinal motion of a rolling car and truck over a variety of track conditions. Input can be as complex as required by the end user. For this application, the input parameters were set by the need to predict accelerations and forces measured in test. Full wheel/rail contact geometry is also used in this model.

Predicted and measured results were compared for a limited number of cases with both models.

3.0 METHODOLOGY

Two major designs were involved. The base case was a standard three-piece truck. The other design was a retrofit arrangement, primarily aimed at providing radial steering of the axles. Cross-linking the axles, therefore, effectively cross-braces the sideframes. This locks (or squares) the sideframes and inhibits the Parlett-Martin or lozenging which is inherent in severe truck hunting. Although there are other modifications available to reduce hunting tendencies, this was the only major alternate truck tested.

3.1 SIDE BEARINGS

Two basic forms of side bearings were tested on both car designs. The base case was roller-type side bearings and the alternate side bearings were of a constant contact design. Previous theoretical and experimental studies have shown the influence of side bearing alteration.[12]

3.2 WHEEL PROFILE

Prior tests and analysis have shown the importance of wheel profile.[13,14] In order to demonstrate the effect of wheel shape on hunting, three different profiles were tested. These were a new 1:20, a modified CN Heumann, and a profile based on a typical FAST worn profile. This provided a large range of effective conicities; from .05 for the 1:20, to .35 for the Heumann.

3.3 CAR TYPE

A 51-foot, 100-ton capacity coal hopper, and an 89-foot intermodal flat car were both used in the test. These car designs represent two of the most common type found in general railroad service. Vehicle characteristics are listed below.

<u>INTERMODAL FLAT CAR</u>
- Length - 89.4 ft.
- Unloaded Weight - 61,850 pounds
- Loaded Weight - 170,000 pounds
- Truck Centers - 66 ft.

<u>COAL HOPPER CAR</u>
- Length - 51.25 ft.
- Unloaded Weight - 63,500 pounds
- Loaded Weight - 263,000 pounds
- Truck Centers - 42.25 ft.

A general view of the test cars are shown in Figures 3-1 and 3-2.

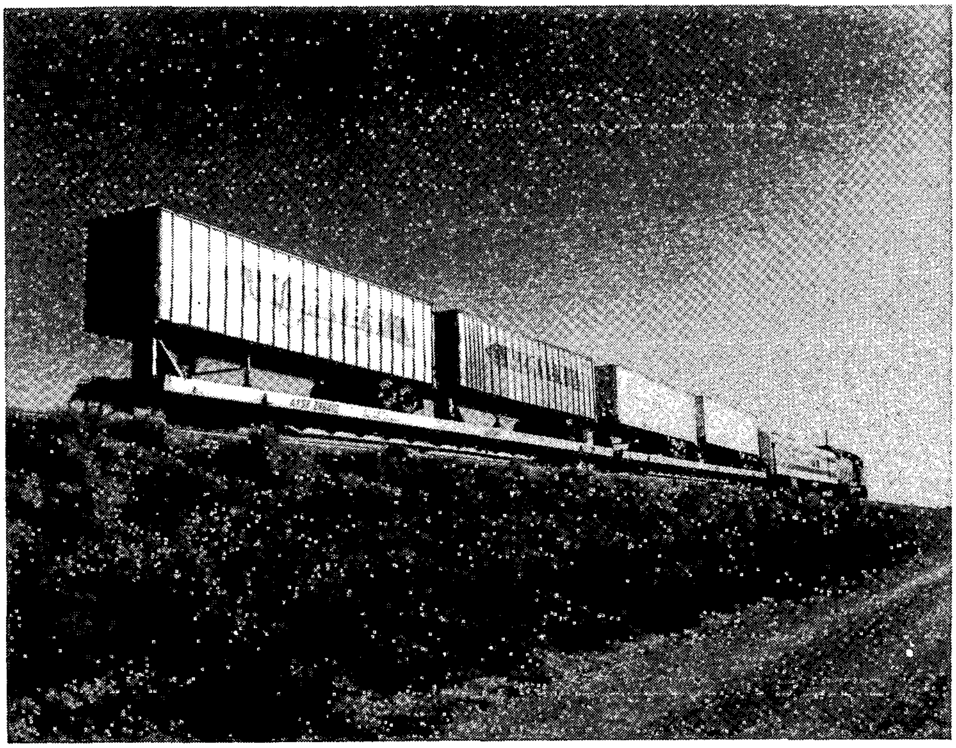


FIGURE 3-1. GENERAL VIEW OF TTX 89' FLAT CAR TEST CONSIST.

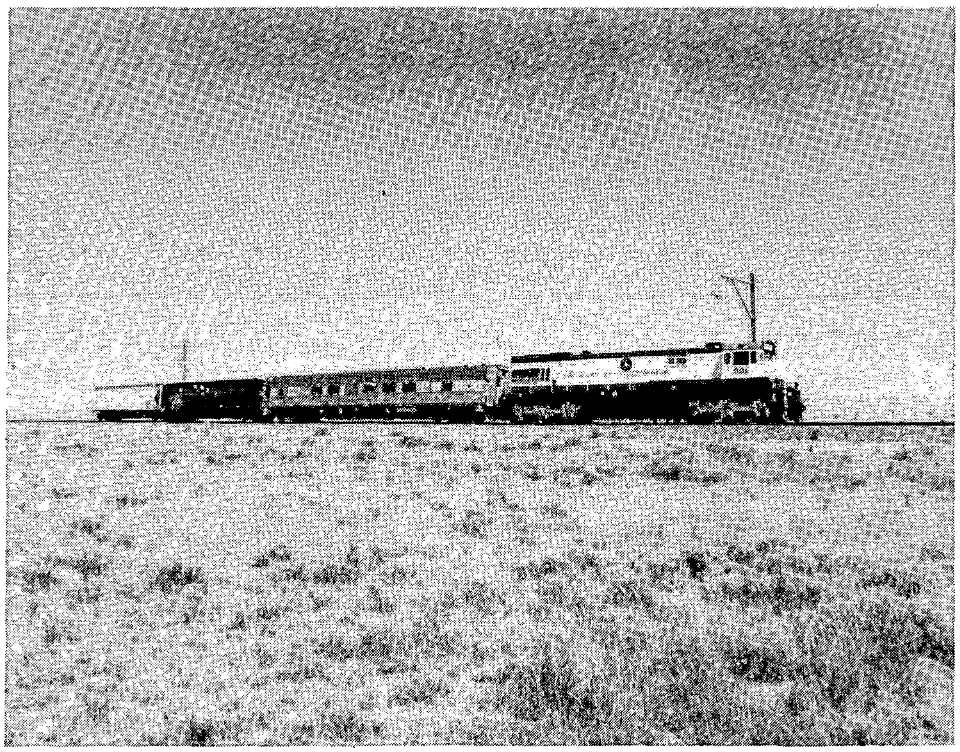


FIGURE 3-2. GENERAL VIEW OF 100-TON COAL CAR TEST CONSIST.

3.4 TEST CONSIDERATIONS

3.5 RAIL CONDITIONS

The test track used was a tangent portion of the Railroad Test Track (RTT). In total, there were 5,000 feet of tangent track along which data was collected. An overview of the RTT is shown in Figure 3-3. The RTT is constructed of 136 RE rail on hardwood cross ties. The test zone is comprised of continuous welded rail and jointed rail in 39-foot lengths.

Testing was carried out on dry rail only. Measurements of both the wheel and rail profiles were taken at the time of testing. Wheel/rail contact geometry was then calculated for all combinations.

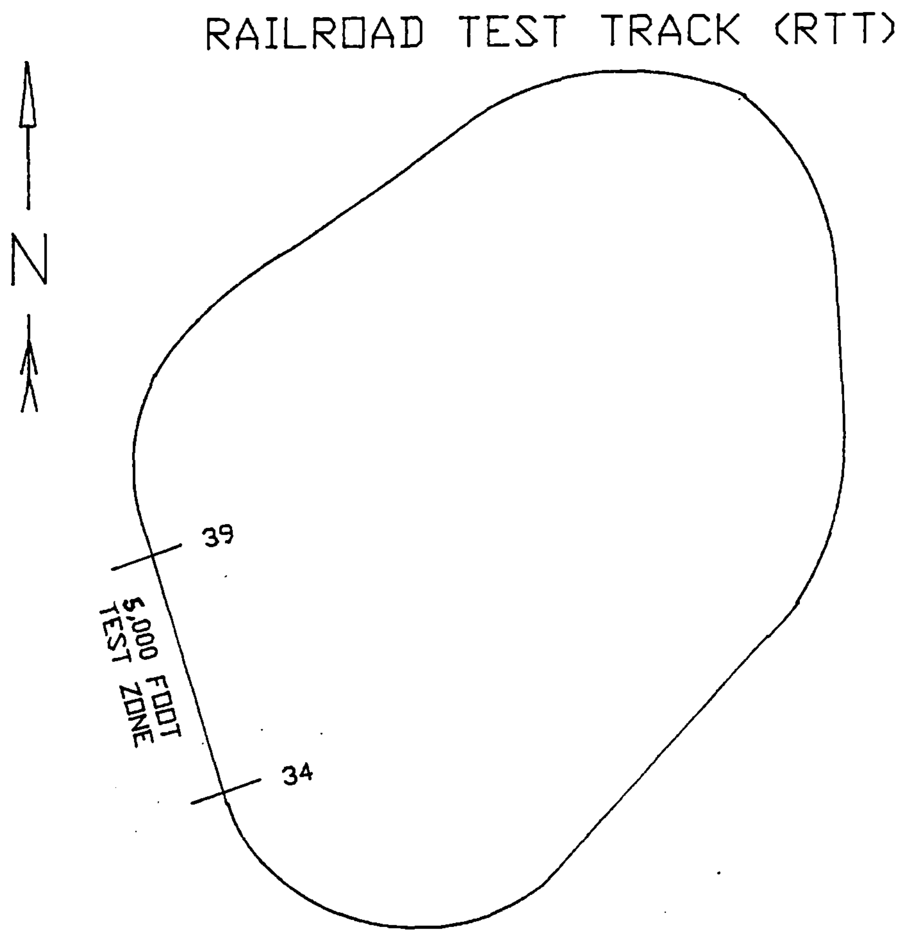


FIGURE 3-3. OVERVIEW OF TTC RAILROAD TEST TRACK (RTT).

4.0 MEASUREMENT TECHNIQUES

4.1 INSTRUMENTATION

4.1.1 Carbody Accelerations

Three laterally oriented carbody accelerometers were used per test car. They were located at each end and at the center of the carbody. The purpose was to measure carbody response in terms of magnitude and phase. End-to-end carbody activity and accelerations relative to axle acceleration was monitored. The signals were digitized at a rate of 64 samples per second. Analog filtering was set at 20 Hz. Fifteen Hz filtering of the digital data was carried out at the time of analysis. This cutoff frequency is used as a standard in AAR car certification.

4.1.2 Axle Accelerations

Lateral acceleration of the test axles was measured by accelerometers mounted on the bearing adapters along the right side of the test cars. Figure 4-1 is an example of the setup. A schematic representation of the entire car's instrumentation is given in Figure 4-2.

4.1.3 Wheel and Rail Profiles

Documentation of the wheel and rail profiles was accomplished using the British Rail wheel and rail profilometers. The wheel profilometer is shown in Figure 4-3. It consists of a reference frame with a measuring head for each wheel. Forty positions per wheel are measured with the heads. Centering of the gage is accomplished by the small dial gages contacting the inner rim of the wheels. Figure 4-4 is a typical example of the resulting x,y coordinates from this device.

The rail profilometer is shown in Figure 4-5, having only a single measuring head. The dial gage measures radially about the rail and then is turned end-for-end to measure the other rail. Gage measurement is derived from the small dial gage which contacts the gage face of the rail. Figure 4-6 shows a rail profile taken with this device.

The fundamental advantage to measuring pairs of wheels and rails at the same time is to obtain the appropriate surface geometry and orientation of the surfaces in rotation and translation. This then allows for the mathematical superposition of the wheels on rails. Contact geometry calculations can then be made for a range of potential contact positions of the wheel-on-rail.

4.1.4 Force Measurement

A total of three different instrumented wheelsets were used in this test. One was a 33-inch wheelset, used under the

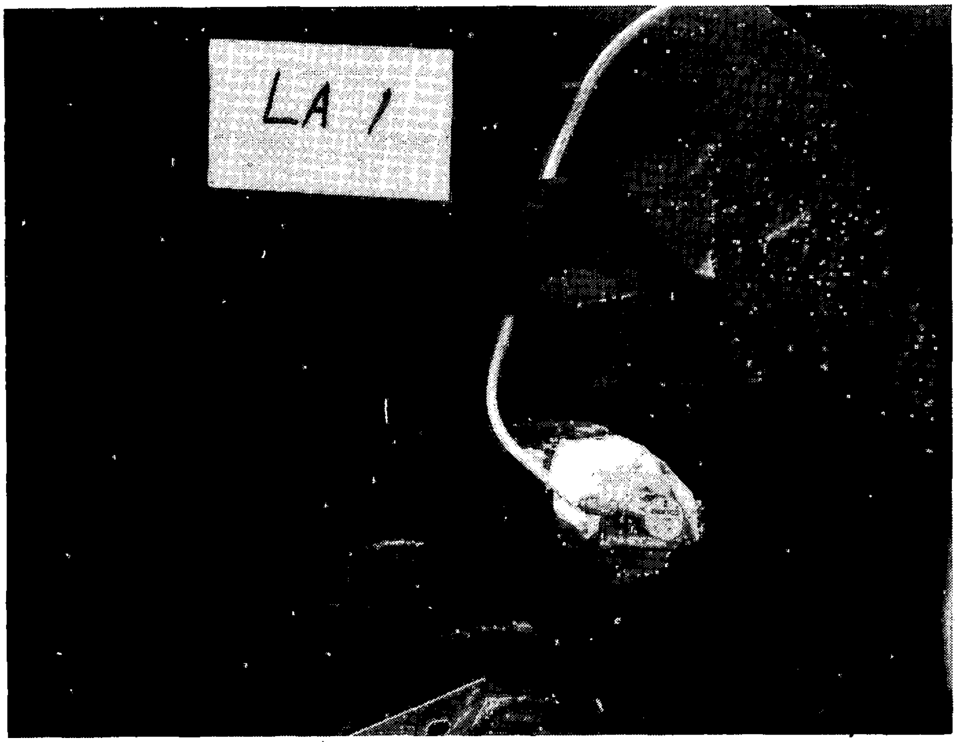


FIGURE 4-1. CLOSE-UP OF AXLE MOUNTED ACCELEROMETER.

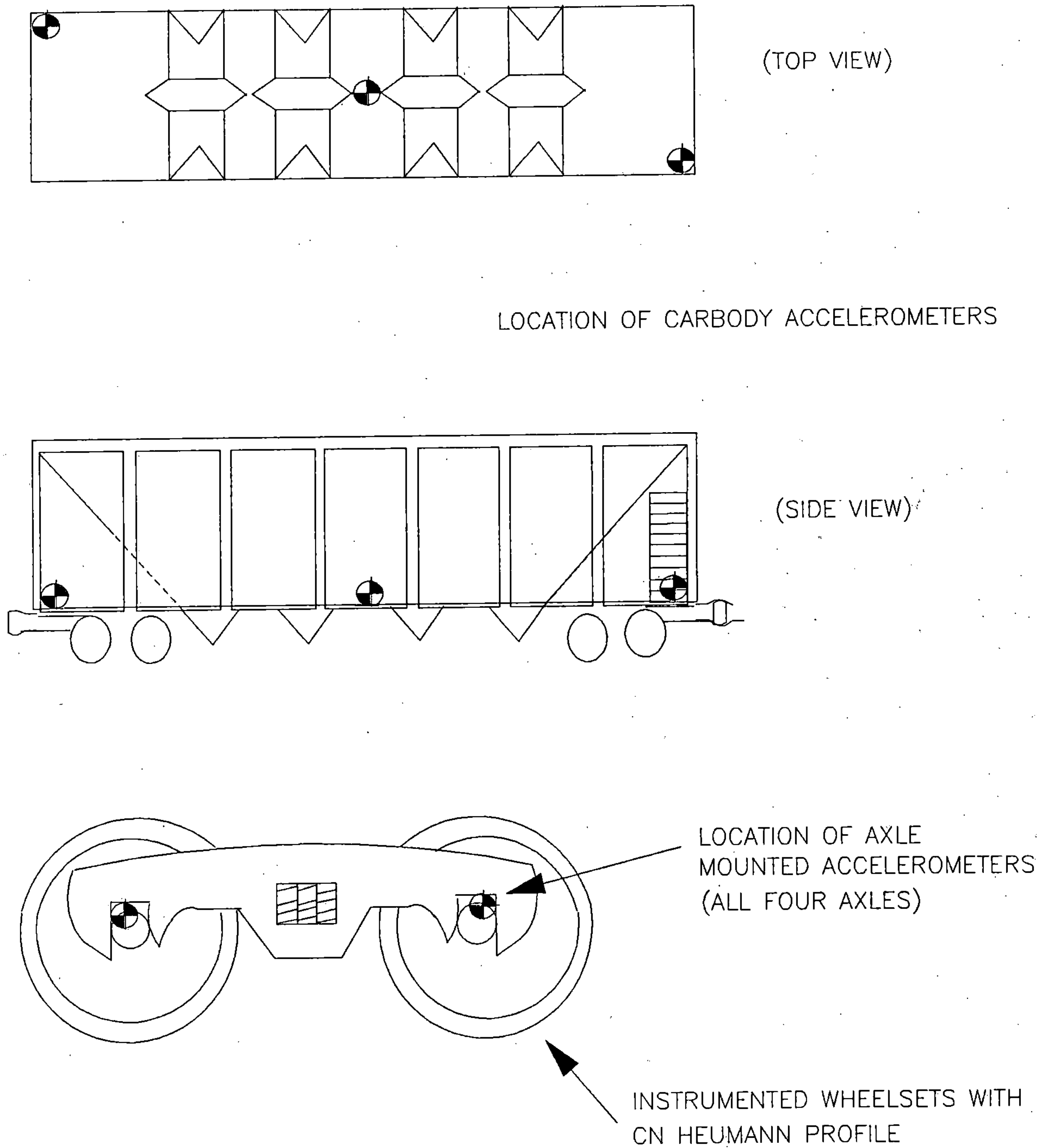


FIGURE 4-2. SCHEMATIC OF TEST CAR'S INSTRUMENTATION.

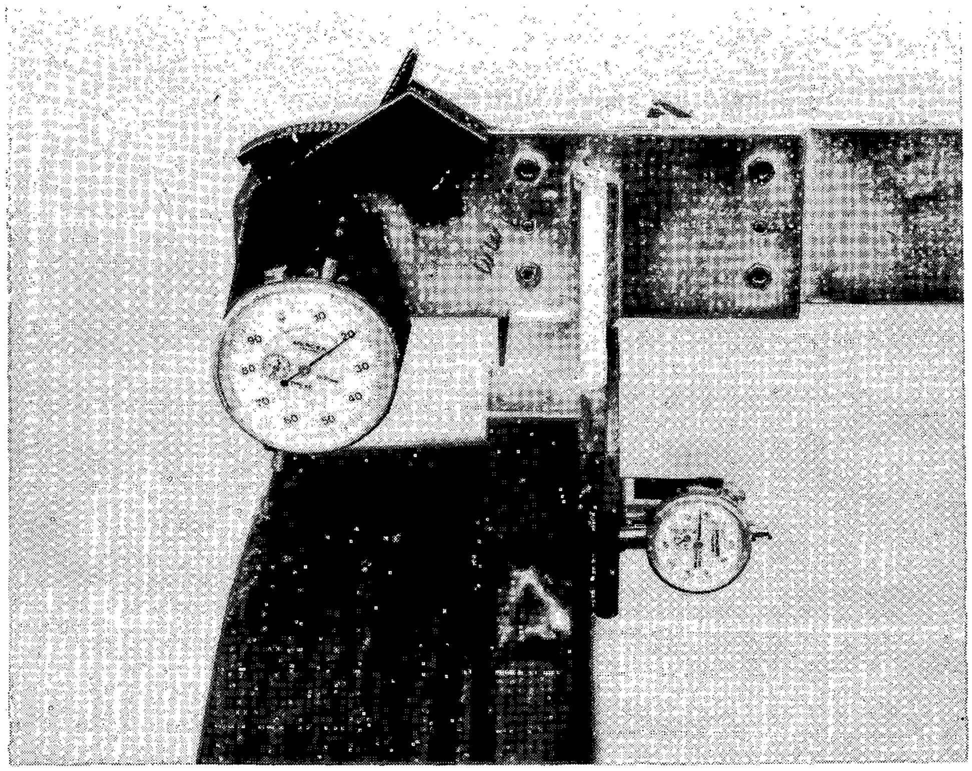


FIGURE 4-3. BR WHEEL PROFILOMETER.

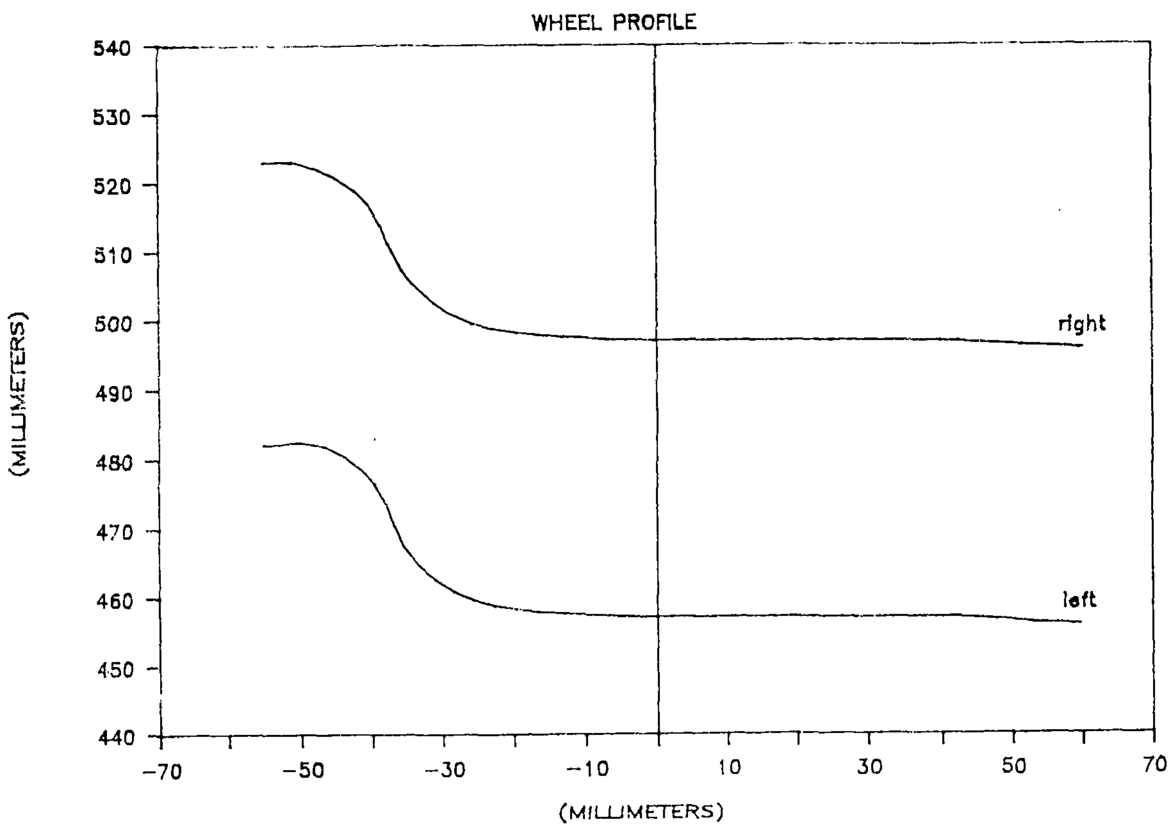


FIGURE 4-4. *GRAPHICAL RESULT OF WHEEL PROFILE MEASUREMENT.*

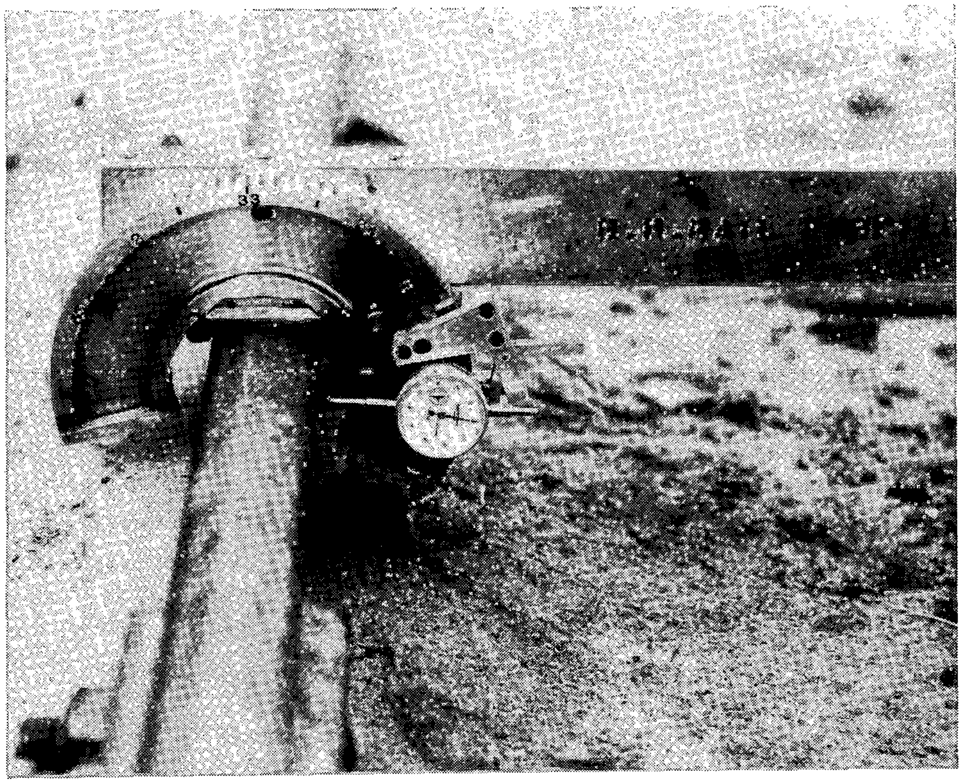


FIGURE 4-5. BR RAIL PROFILOMETER.

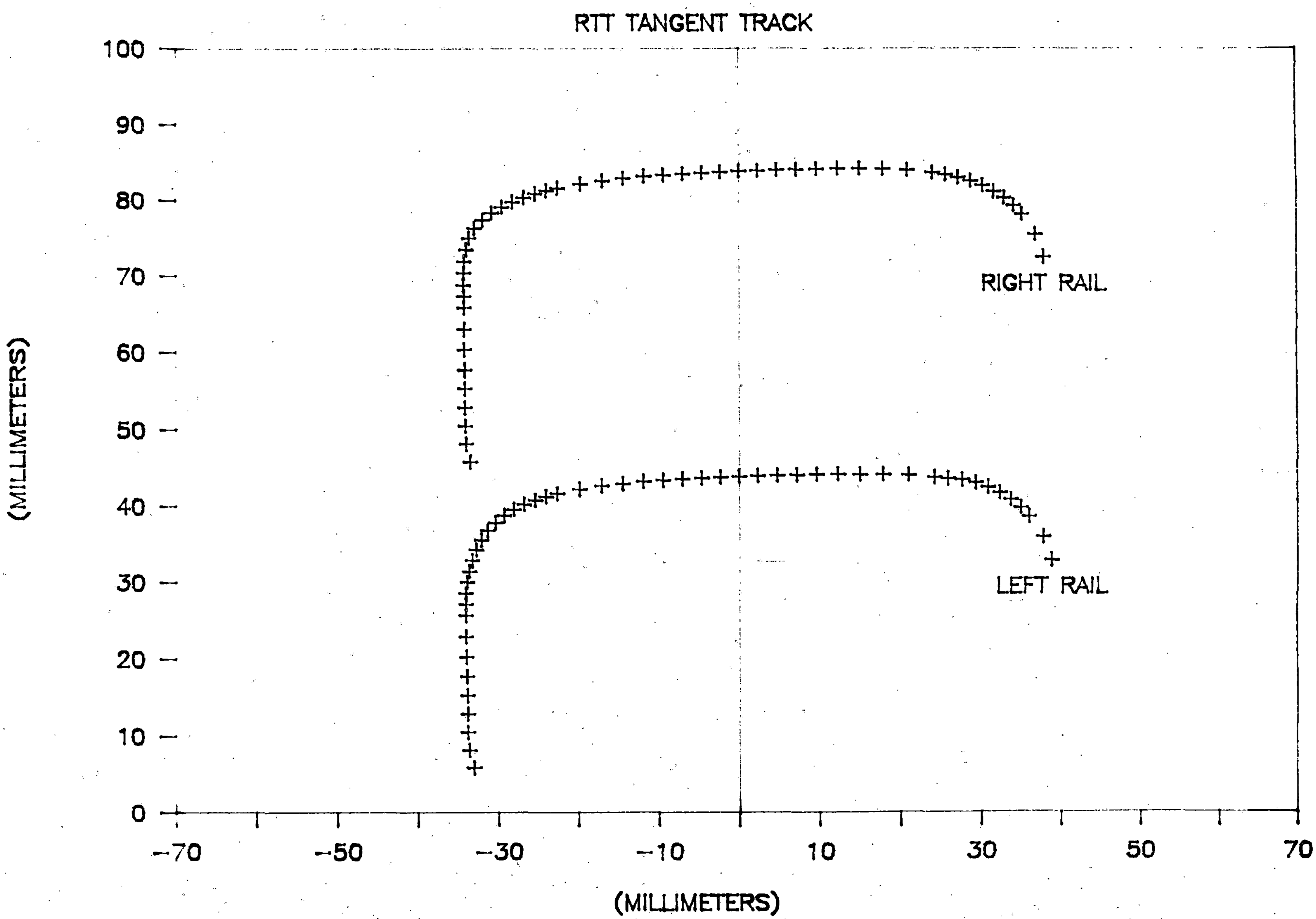


FIGURE 4-6. GRAPHICAL RESULT OF RAIL PROFILE MEASUREMENT.

intermodal flat car, two were 36-inch, 100-ton capacity axlesets used for testing the hopper car. All axlesets measured vertical and lateral forces.

Depending upon speed, the wheelset data is sampled at 400 or 500 samples per second. A typical time history of lateral force during truck hunting is presented in Figure 4-7.

4.1.5 Train Resistance

For all test runs, the resistance of the test vehicle was measured using a load cell coupler. The purpose of the design was to accurately measure the absolute and incremental effects which contribute to rolling resistance. It was specifically intended for use with a small test consist. The accuracy and resolving power of the system was in part due to the fact that a 50,000 pound load cell was used rather than scaling over the full range of a higher capacity load cell. It also has proven less troublesome and more sensitive than a strain gage coupler that has been used in the past.

The single car coupler with fitted yokes and associated hardware is shown in Figure 4-8. The horizontal bars above and below the load cell are a safety feature. They generally bear no load except when the load cell is removed or if there is a failure in the basic system.

4.1.6 Video Recording

Video tapes were made during all test runs. The video camera recorded the motion of one side of the lead truck. This allowed for real-time monitoring of hunting activity and provided a visual backup during the review of the test data.

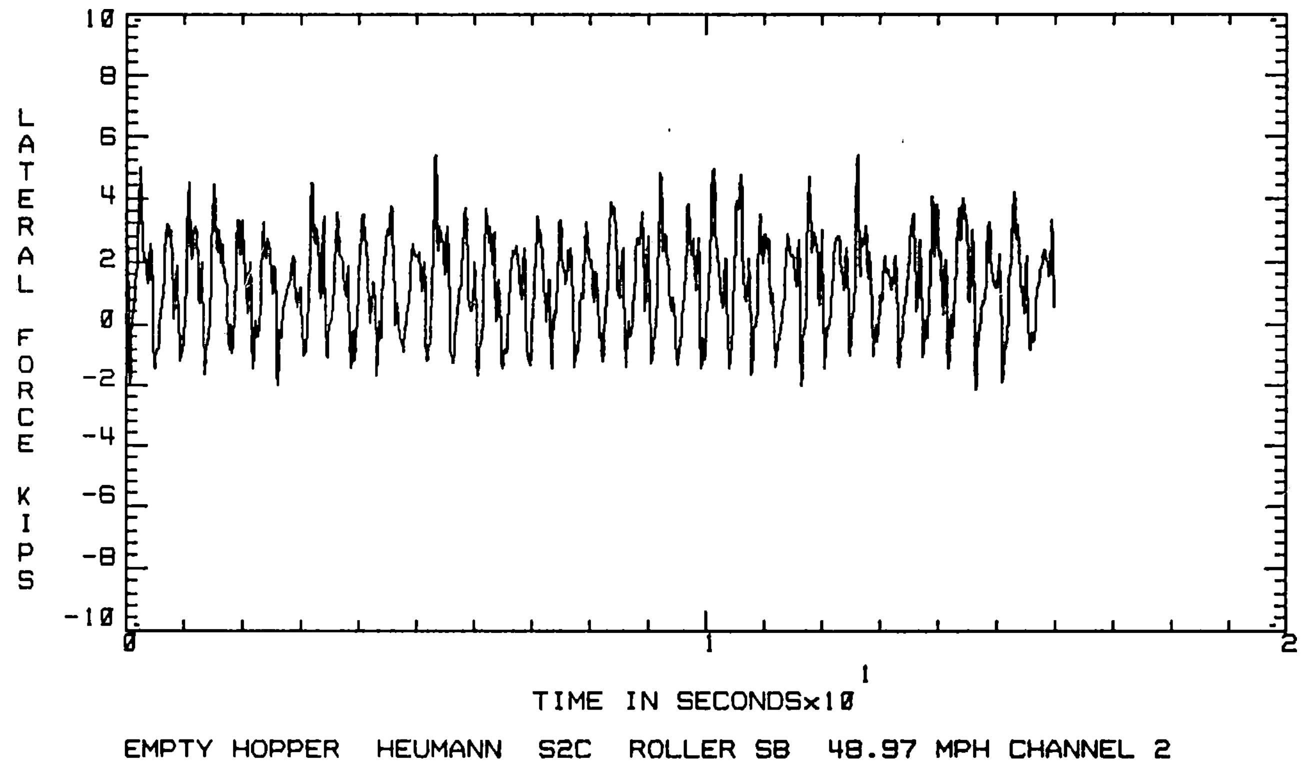


FIGURE 4-7. TIME HISTORY OF WHEEL FORCES DURING HUNTING.

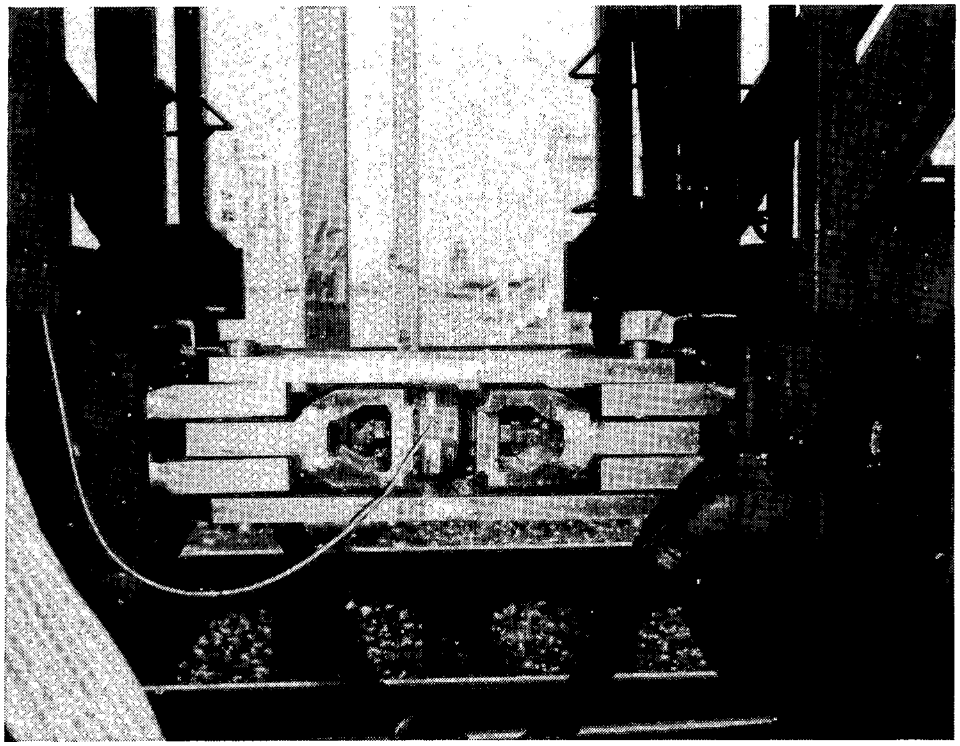


FIGURE 4-8. LOAD CELL COUPLER ARRANGEMENT.

5.0 TEST IMPLEMENTATION

5.1 TEST MATRIX

Table 5-1 contains the layout of test runs for the intermodal flat car portion of the test. Most configurations were tested at an initial speed of 30 mph and at increasing 5 mph increments, up to the exceeding of safety criteria or a maximum speed of 70 mph. Testing was terminated if a repeated peak-to-peak lateral acceleration level of 1.0 g or more occurred.

Coal car tests were conducted on a 51-foot coal hopper. Test conditions in terms of speeds were similar to those for the intermodal flat cars. The coal car test matrix is outlined in Table 5-2.

5.2 TEST ZONE

As described in Section 3.2, the RTT was used for the hunting test. For the analysis, a specific 2,500-foot section of track or test zone was identified. This subsection was selected based on the observed response for a range of test cars. If onset was experienced anywhere along the extent of the RTT, it would usually occur most consistently within this test zone. By editing the data files to include only this test segment, there was a significant reduction in processing time and expense.

Since the geometric relationship between the wheel and rail has an important influence on the stability of railway cars, the shape of the running surface was measured at several sites. This was accomplished with the BR design rail profilometer. The cross-section of the RTT test rail at one location is shown in Figure 5-1. A new AREA 136 RE rail is also shown relative to the measured rail. The measured rail is very similar to a new rail cross-section. It was also found that there was very little variation in the basic shape of the rail along the test zone. Measurements revealed that track gauge averaged 56.4 inches overall.

5.3 TEST WHEELS

Accurate profile measurement of each test wheel was performed as the wheelsets were removed from the TTC Hegenscheidt wheel lathe. This was done to verify that the profile met the basic test requirements in terms of surface shape. The three profiles tested were the standard AAR 1:20 profile, an average "worn" FAST profile, and the CN Heumann. A variation on the CN profile was used to provide the effect of wider track gage. This was done by machining a narrow flange CN profile onto the wheelset, therefore, producing increased flangeway clearance.

TABLE 5-1

TEST CONFIGURATIONS FOR TOFC TESTING

CONFIGURATION	LADING	WHEEL PROFILE	TRUCK
1	EMPTY	HEUMANN	S2C
2	LOADED	HEUMANN	S2C
3	LOADED	HEUMANN	S2C
4	EMPTY	HEUMANN	S2C
5	EMPTY	FAST WRN	S2C
6	EMPTY	FAST WRN	S2C
7	LOADED	FAST WRN	S2C
8	LOADED	FAST WRN	S2C
9	LOADED	1:20	S2C
10	LOADED	1:20	S2C
11	EMPTY	1:20	S2C
12	EMPTY	1:20	S2C

TABLE 5-2

TEST CONFIGURATIONS FOR HOPPER TESTING

CONFIGURATION	LADING	WHEEL PROFILE	TRUCK
13	EMPTY	HEUMANN	S2C
14	EMPTY	HEUMANN	S2C
15	EMPTY	HEUMANN	DR1
16	1/3 LOADED	HEUMANN	DR1
17	1/3 LOADED	HEUMANN	S2C
18	1/3 LOADED	HEUMANN	S2C
19	2/3 LOADED	HEUMANN	S2C
20	2/3 LOADED	HEUMANN	S2C
21	EMPTY	N FLANGE HEU.	S2C
22	EMPTY	FAST WORN	S2C

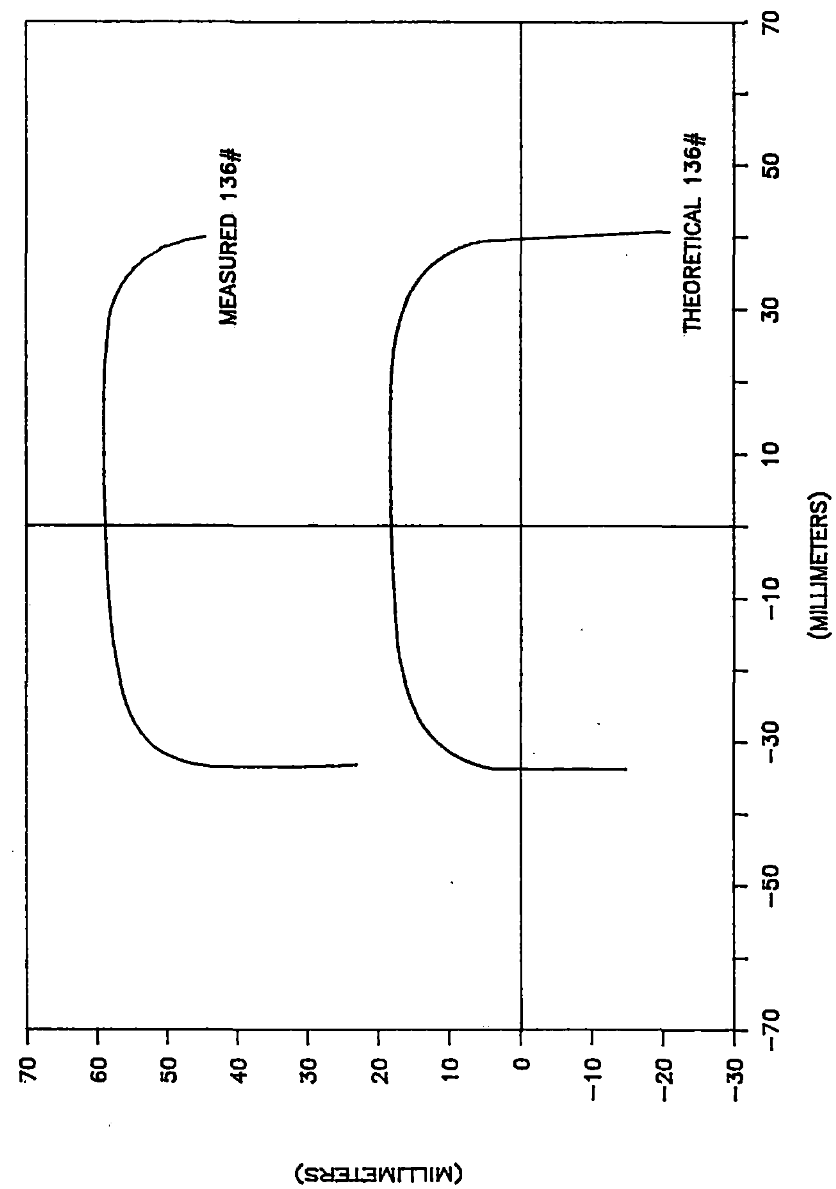


FIGURE 5-1. OVERLAY OF MEASURED TEST RAIL AND NEW 136# RAIL.

The standard 1:20 is shown in Figure 5-2. Because this is a straight line profile, it generates a 1:20 or 0.05 effective conicity on the rail. The other test profiles contain a series of radii which produce a more complex wheel/rail contact condition. The standard method of analyzing the conicity is to plot the change in rolling radius difference against the lateral position of the wheel on a specific rail.[15] Figures 5-3 and 5-4 present the basic geometry for the FAST worn profile and the CN Heumann. Figure 5-5 shows the rolling radius change with lateral shift across the 136 RE rail for the FAST profile. It results in a greater rate of rolling radius change per unit of lateral movement than the 1:20 profile presented in Figure 5-6. Thus, it provides higher conicity, producing a lower onset speed than the 1:20 for the same vehicle.[16]

A linear estimate of the conicity can be made which is comparable to the straight line used in the 1:20 case. This involves the use of a describing function which linearizes the rate of rolling radius change with lateral shift. The magnitude of the conicity is also dependent on the amplitude of wheelset excursion across the rail. With wheels which have a compound shape, the conicity increases with increased amplitude. For these wheels, the flange-to-flange amplitude is used since this is the operative range of the wheel when hunting.

The CN rolling radius plot is given in Figure 5-7. This is for a standard Heumann on the 136 RE rail. The conicity is significantly greater than the two preceding examples. The Heumann is based on a lateral expansion of the AREA 115 RE rail shape which is similar to the AREA 132 RE rail. This resulted in a wheel profile that is more compatible with either the 119 RE or 132 RE rail. On these rails, the wheel conicity is somewhat lower. It, therefore, produces a less than optimum situation on 136 RE rail. This condition causes hunting at a relatively low speed for standard vehicles.

The narrow flange Heumann was produced to look at increased flangeway clearance, a situation which occurs with wide gage or wheels with worn flanges. Included in Table 5-3 is the profile type and the conicity which result from the wheel/rail contact.

5.4 VEHICLE DESCRIPTION

The intermodal flat car used for the flat car test series was a standard Trailer Train 89-foot "piggyback" flat, capable of carrying two 45-foot trailers. This resulted in two distinct on-rail weights of 61,850 pounds and 159,450 pounds, both of which were tested. The trucks were standard three-piece Barber S-2-C design. The standard setup had single roller side bearings. Constant contact side bearings were also tested. This configuration is of interest since the yaw stiffness of the truck is generally increased by the use of contact side bearings. By inhibiting the yaw, the on-set speed of a vehicle normally increases. Basic features are shown in Figure 5-8.

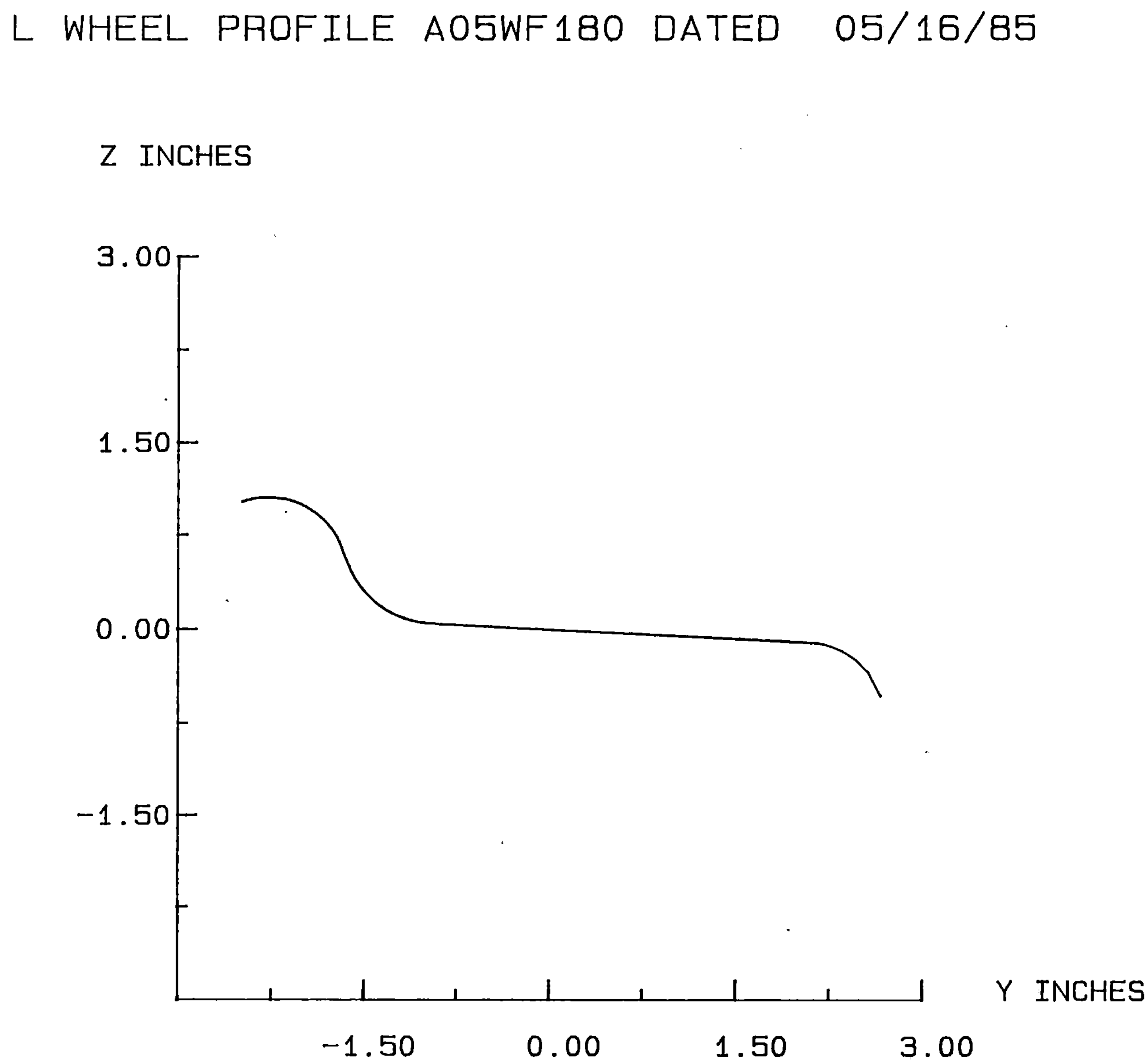


FIGURE 5-2. STANDARD AAR 1:20 PROFILE.

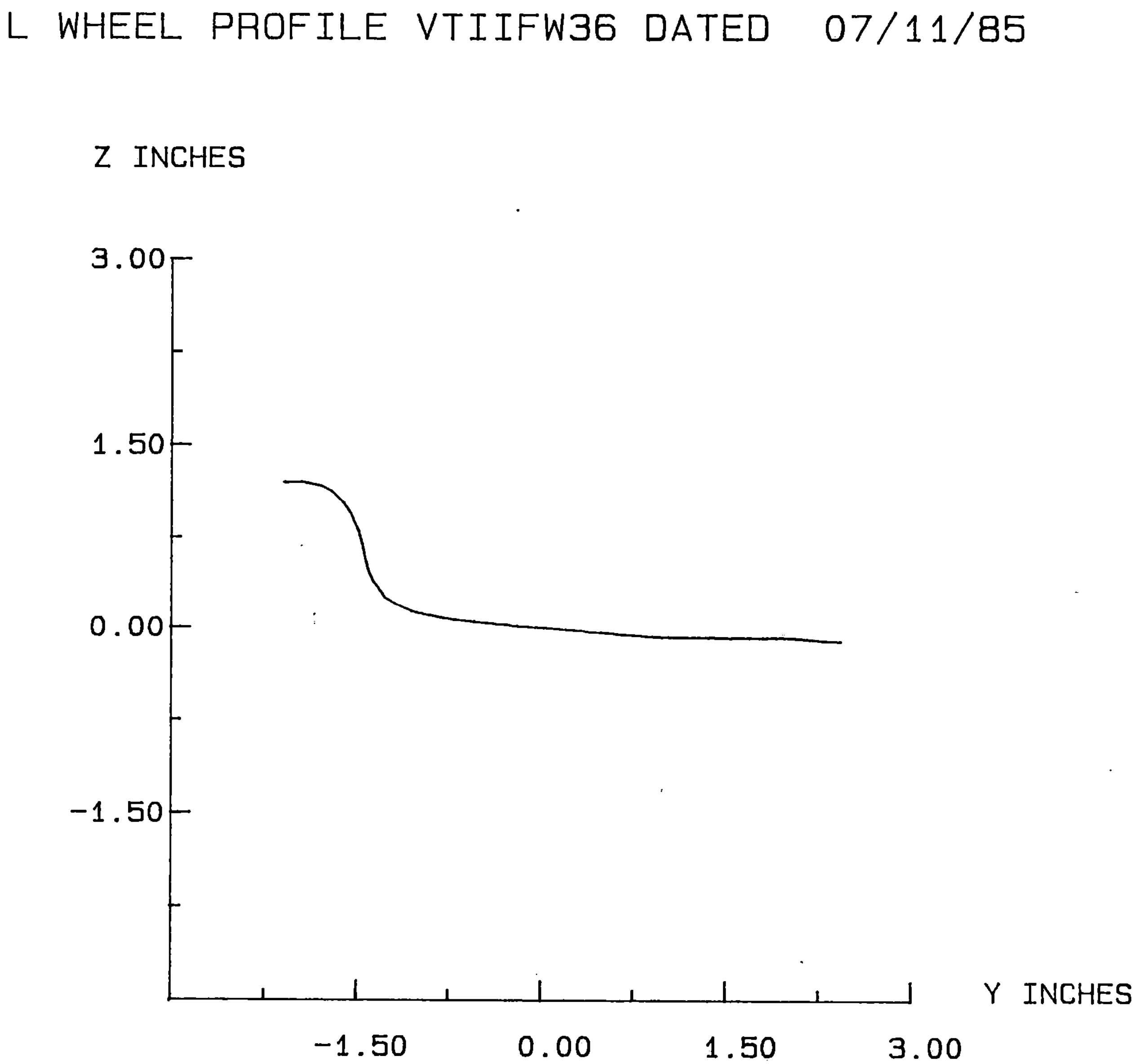


FIGURE 5-3. AVERAGE FAST WORN WHEEL PROFILE.

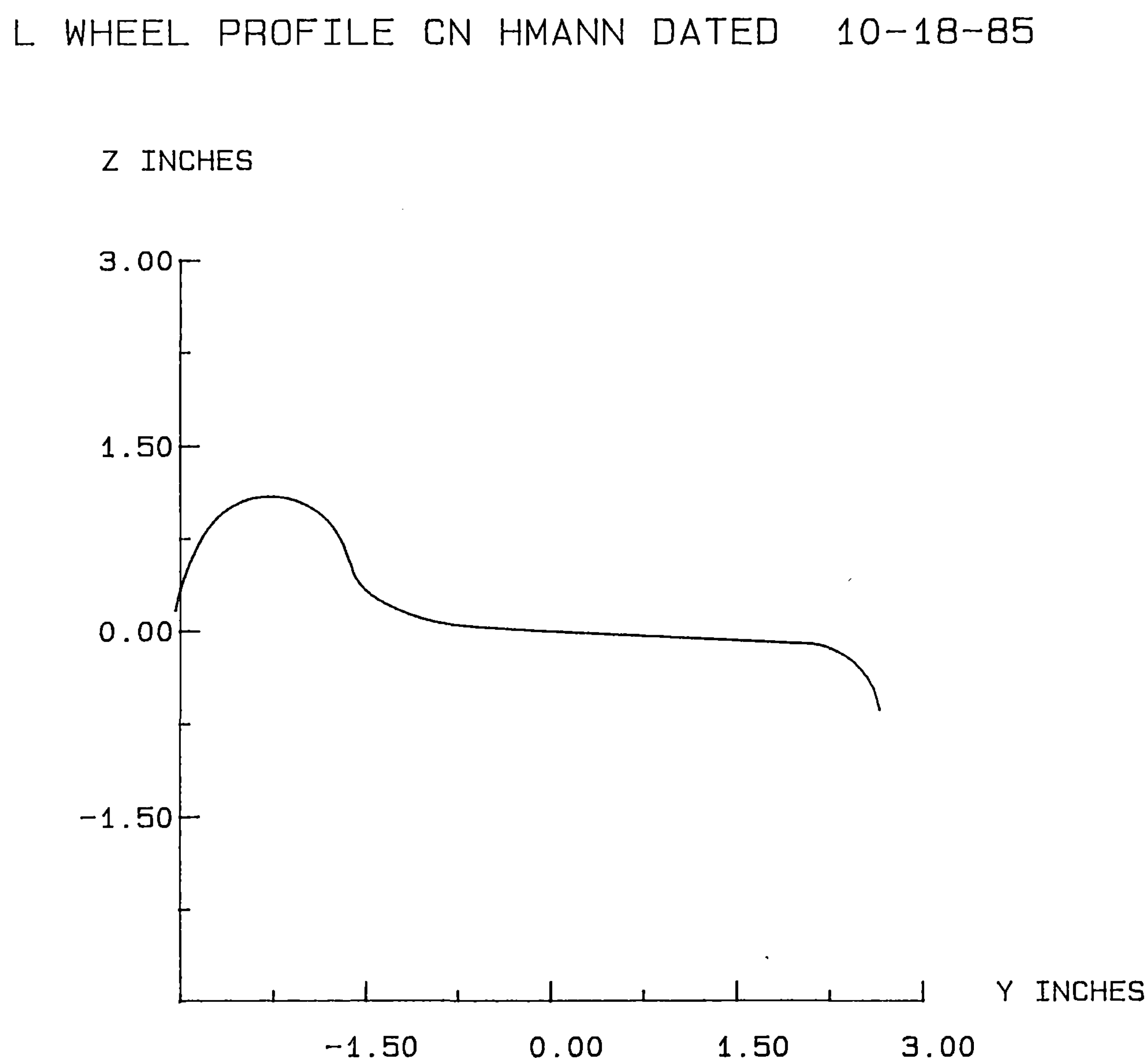


FIGURE 5-4. CN HEUMANN WHEEL PROFILE.

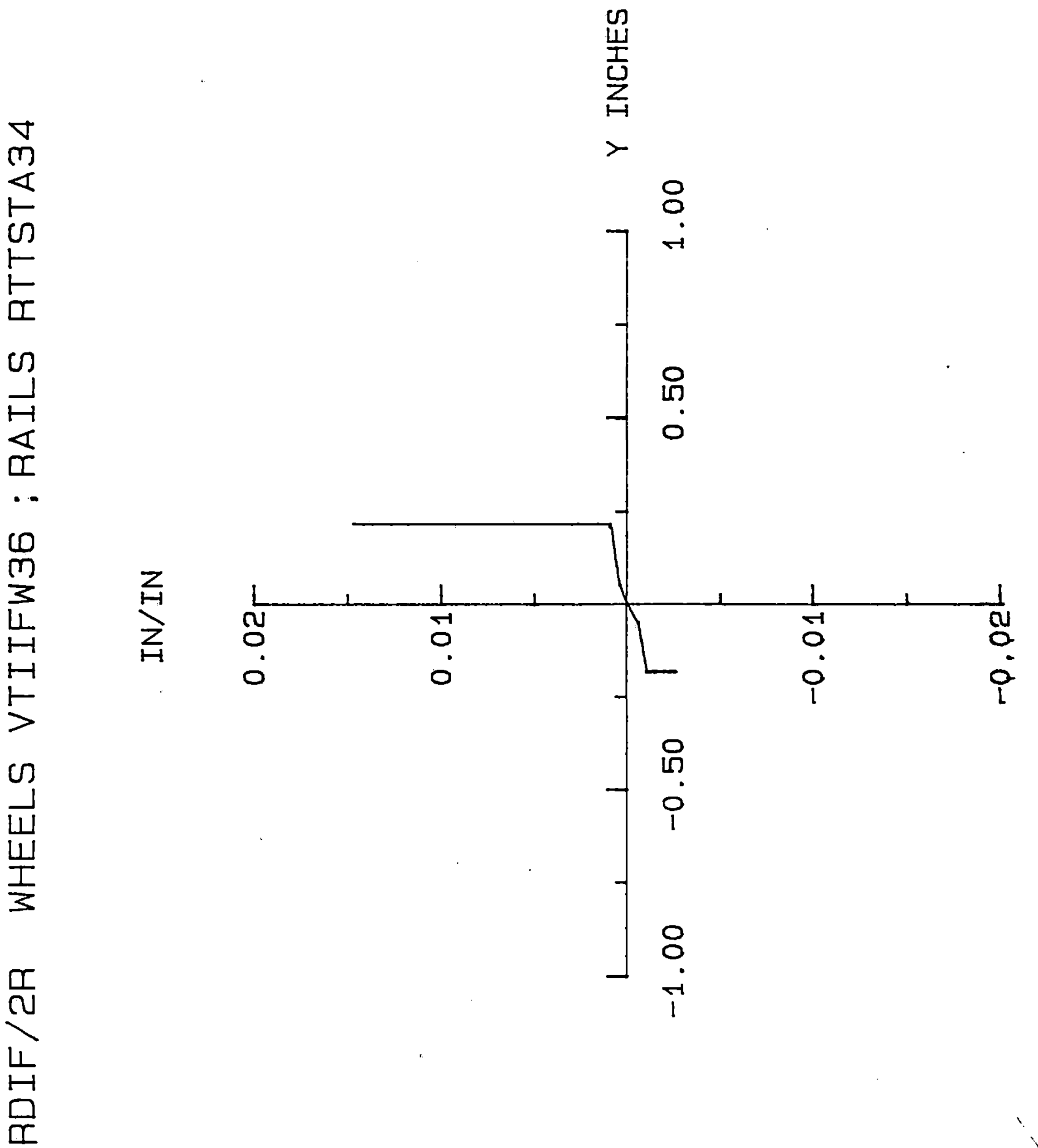


FIGURE 5-5. ROLLING RADIUS CHANGE PLOT FOR FAST WORN WHEEL.

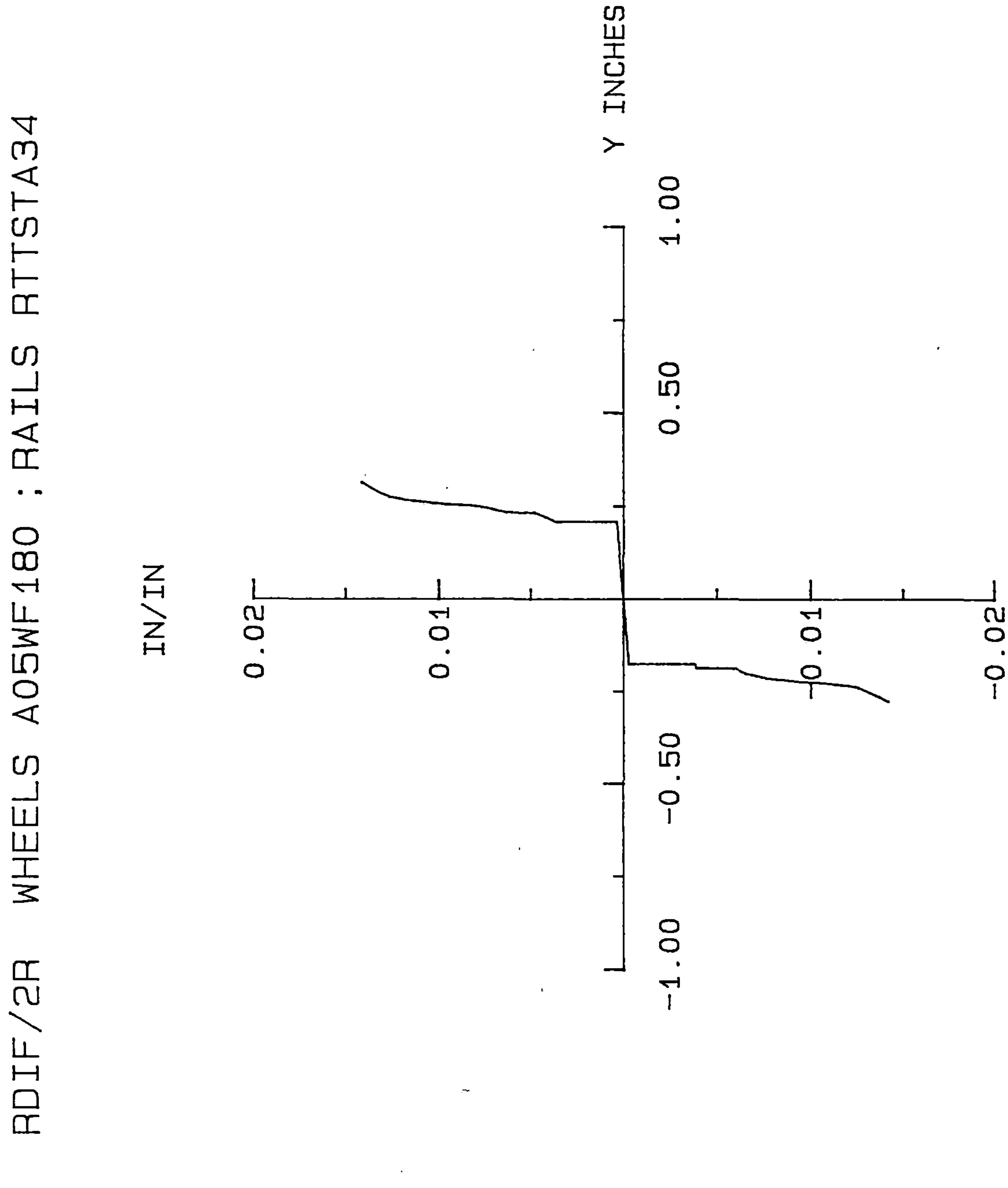


FIGURE 5-6. ROLLING RADIUS CHANGE PLOT FOR AAR 1:20 WHEEL.

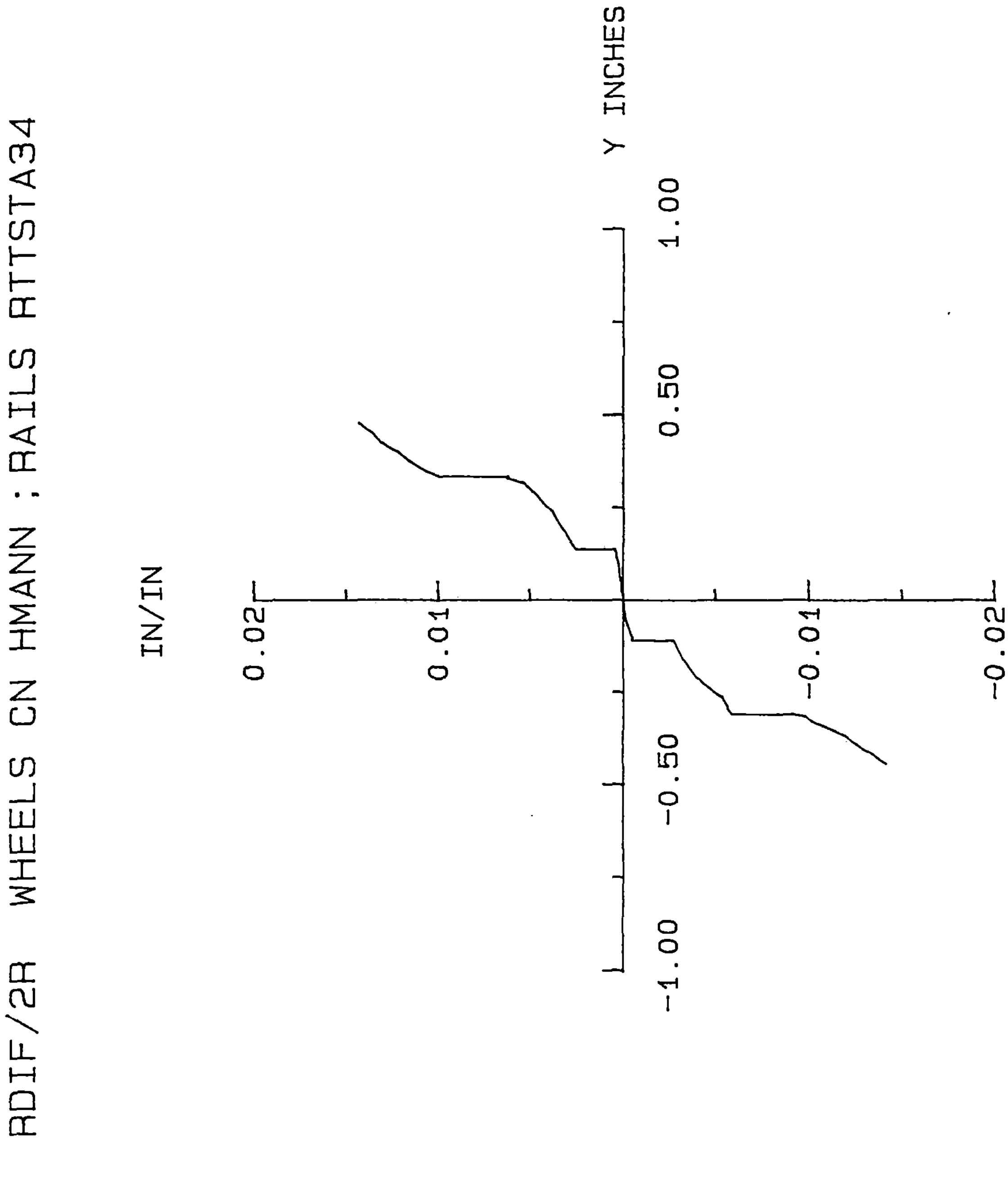


FIGURE 5-7. ROLLING RADIUS CHANGE PLOT FOR CN HEUMANN WHEEL.

TABLE 5-3

WHEEL PROFILE TYPE AND EFFECTIVE CONICITIES

WHEEL PROFILES	CONICITIES
AAR 1:20	0.05
AVG. FAST WORN	0.15
CN HEUMANN	0.35

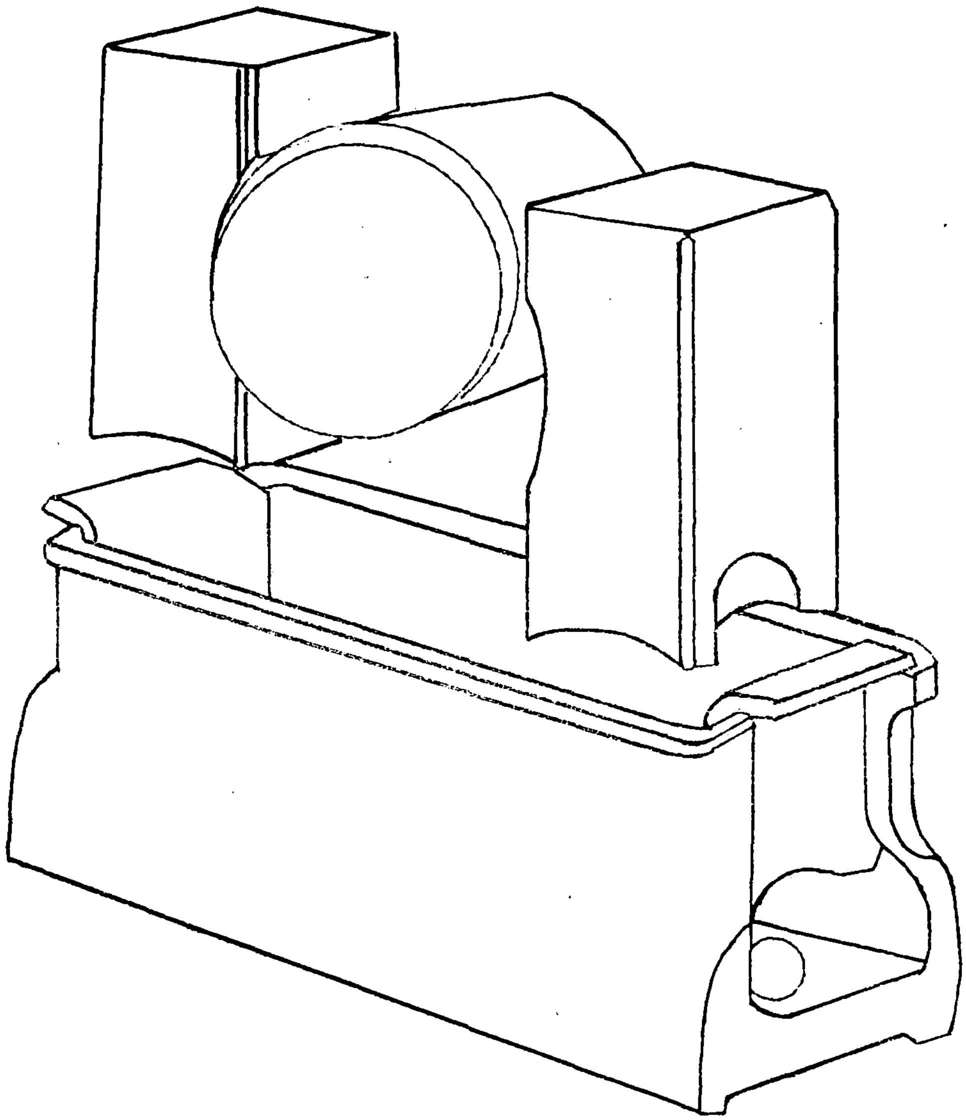


FIGURE 5-8. DRAWING OF 70-TON STUCKI CONSTANT CONTACT SIDE BEARINGS.

The hopper car test series used a surplus Union Pacific car. It was equipped with Barber S-2-C trucks and double roller side bearings. It was tested at three different axle loads; empty, one third capacity, and two thirds capacity. Since hunting threshold increases with weight, only limited runs were made at the higher loads.

Constant contact side bearings were also investigated on the hopper car. These are shown in Figure 5-9. They are a Stucki 100-ton standard design.

All running gear was thoroughly inspected for defects and wear. All test equipment was observed to be in average worn-in condition with no unusual characteristics. Components had typical surface properties and indicated fairly symmetric wear patterns.

5.5 TEST CONDUCT

A nominal operating speed range of 30 to 70 mph was pre-established for all test cars. Intervals of 5 mph were used if the vehicle was likely to hunt at an intermediate test speed. Ten mile per hour intervals were utilized if the test configuration was likely to reach threshold levels at only the higher speeds. The maximum test speed was dictated by the observance of axle acceleration in excess of 1.0 g's peak-to-peak up to maximum of 70 mph.

Whenever possible, the tests involving the load cell coupler were carried out in light wind conditions. This was done to minimize extraneous aerodynamic drag on the test car. The effect of even moderate winds can contribute significant resistance at the higher test speeds.

For this reason, the data for resistance has been screened and only used if the wind was less than five miles per hour. The additional factors of train acceleration and grade were also considered into the data analysis.

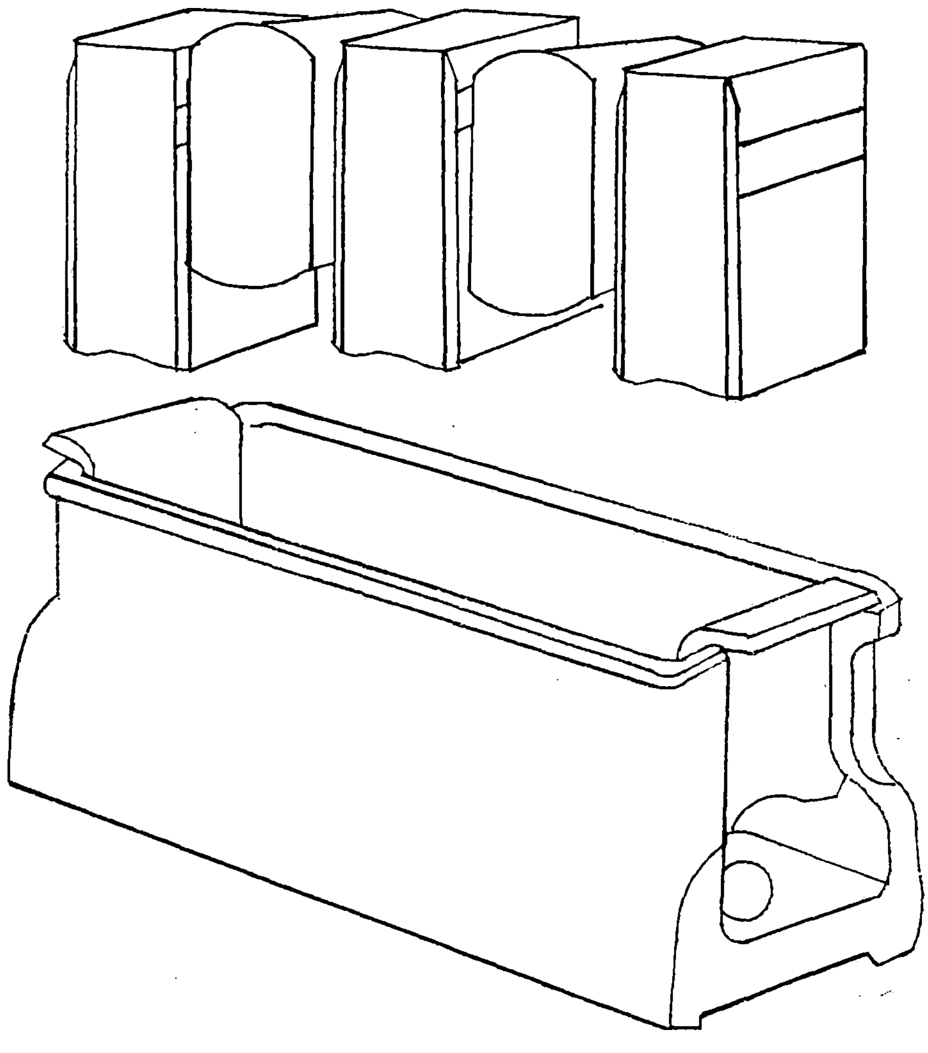


FIGURE 5-9. ILLUSTRATION OF 100-TON CAPACITY STUCKI CONSTANT CONTACT SIDE BEARINGS.

6.0 DATA ACQUISITION AND REDUCTION

A different instrumentation test car was used for each of the two major test series. The FRA T-7 test car was utilized for the flat car testing. The AAR-100 test car was used in the hopper car series. The reason for the change was a result of scheduling of the T-7 test car on other FRA sponsored tests. Although the acquisition and computer equipment was different, signal conditioning and sampling was consistent on both cars.

The T-7 test car contained 64 channels of signal conditioning equipment and a Hewlett Packard 1000 computer. Digital data was recorded on tape drives with data reduction carried out at the end of each major test sequence. Because the hunting tests are conducted on tangent track, the standard deviation was used, as they represent the true Root Mean Square (RMS) level of acceleration. This eliminates any influence of a DC signal offset in the raw data.

Strip chart recorders were used to monitor real-time data. This insured that test conditions were within safety limits and that the instrumentation was operating properly.

The AAR 100 car had a capacity of 64 channels of signal conditioning and was, at the time of the test, equipped with a Digital Equipment Corporation (DEC) PDP 11/34 computer. The same basic near real-time statistics were also generated on the test car. Strip chart recorders were also placed on the AAR 100 car for the monitoring of real-time data.

In addition to the level of accelerations and forces, constant recording of train speed was maintained. This allowed for a very accurate correlation between vehicle response and speed.

6.1 DATA REDUCTION AND ANALYSIS

Data tapes were transported to the TTC VAX DEC 11/780 and downloaded to disc. Time histories of selected channels were produced and edited to contain data for the 2,500-foot test zone. Due to the similarity of response for certain channels and the time and expense of reducing all recorded data, a final screening process reduced the number of files by approximately 50 percent. An example of this was to use one axle of wheel force data rather than both axles in the process of wheel/rail force analysis.

The acceleration data was all digitally filtered down to a maximum level of 15 Hz. This is the standard used by the AAR in certification testing since it eliminates accelerations not associated with hunting. At this point, statistics were generated for the remaining channels over the selected test zone. This included the descriptive measures shown in Table 6-1.

TABLE 6-1

EXAMPLE STATISTICS FOR HUNTING ANALYSIS

FILENAME: 08300233423.UD3		
NO. OF DATA BLOCKS	=	3.5000
NO. OF DATA PTS.	=	3584.0000
MIN. VALUE	=	-0.964917E-01
MAX. VALUE	=	0.371152E+00
MEAN VALUE	=	0.214607E+00
RMS VALUE	=	0.222789E+00
STD DEV	=	0.598314E-01
MAX - MIN	=	0.467644E+00

Frequency and power level is identified using Power Spectral Density Plots (PSD), as presented in Figure 61, for a 50 mph test run with the standard coal hopper. The Data Reduction System (DRS) program was used for all of the post-test analysis. It is the primary dynamic data reduction package at the TTC.

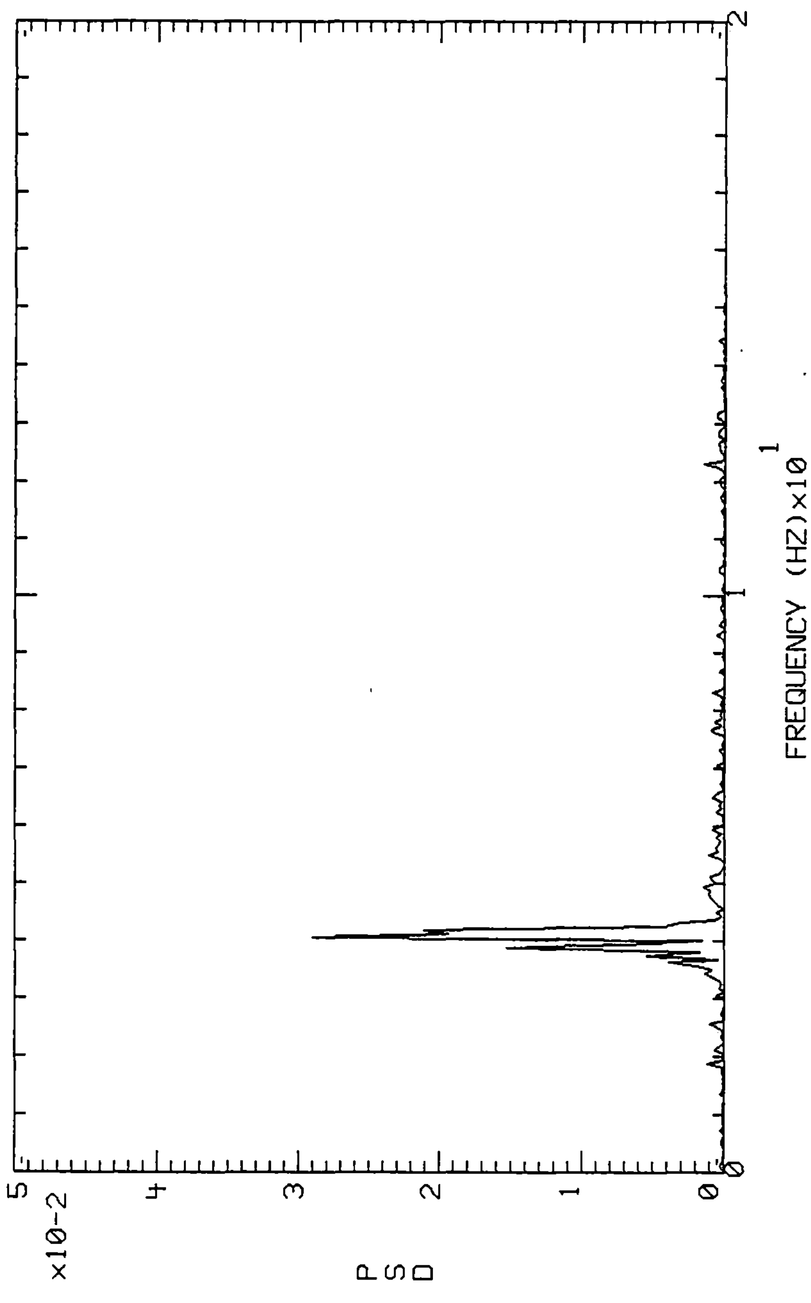


FIGURE 6-1. TYPICAL POWER SPECTRAL DENSITY PLOT (PSD) SYSTEM.

7.0 DISCUSSION OF RESULTS - FLAT CAR TESTS

A presentation of the following results identifies the major trends within test variables.

7.1 TRAILER ON FLAT CAR TEST SERIES

The baseline case for this test category is the 89-foot flat car outfitted as follows:

- o Unloaded Condition
- o Single Roller Side Bearing
- o S-2-C Trucks
- o Heumann Wheel Profile

The discussion of test results is generally covered in the following sequence:

- o Axle Accelerations
- o Carbody Accelerations
  - -Where Appropriate:
    - Wheel/Rail Forces
    - Correlation of Acceleration and Forces
    - Rolling Resistance and Speed Relationship

7.2 AXLE LATERAL ACCELERATION

A standard method of characterizing the hunting behavior of rail vehicles involves the measurement of axle acceleration. This was accomplished by mounting lateral low frequency response accelerometers on the four bearing adapters for each of the test cars. It is assumed that for the normal band of hunting frequency, that the adapter movement is equal to that of the axle. It was placed there due to ease of access and to avoid the difficulty of instrumenting a rotating axle. Time histories within the designated 2,500-foot test zone typically show that all four axles behave similarly for a given test configuration.

Since this was the first configuration, it forms the baseline for comparison in this analysis. Figures 7-1 through 7-5, show the increase in lateral acceleration for the lead truck, lead axle, from the initial test speed of 30 mph, up to the terminal speed of 45 mph. These plots are for only 16 seconds of testing, but reflect the overall trends in performance.

Subsequently, the RMS has been calculated for all axles and speeds for the entire 2,500-foot test zone. As an index of hunting, the RMS for the entire test run is calculated for each axle and the average is plotted against speed for each configuration. Included in Figure 7-6 is the plotted relationship between speed and RMS acceleration for the base case flat car. From the trend it is clear that the onset of hunting occurred in the 40 to 42 mph range. This is apparent from the change in the

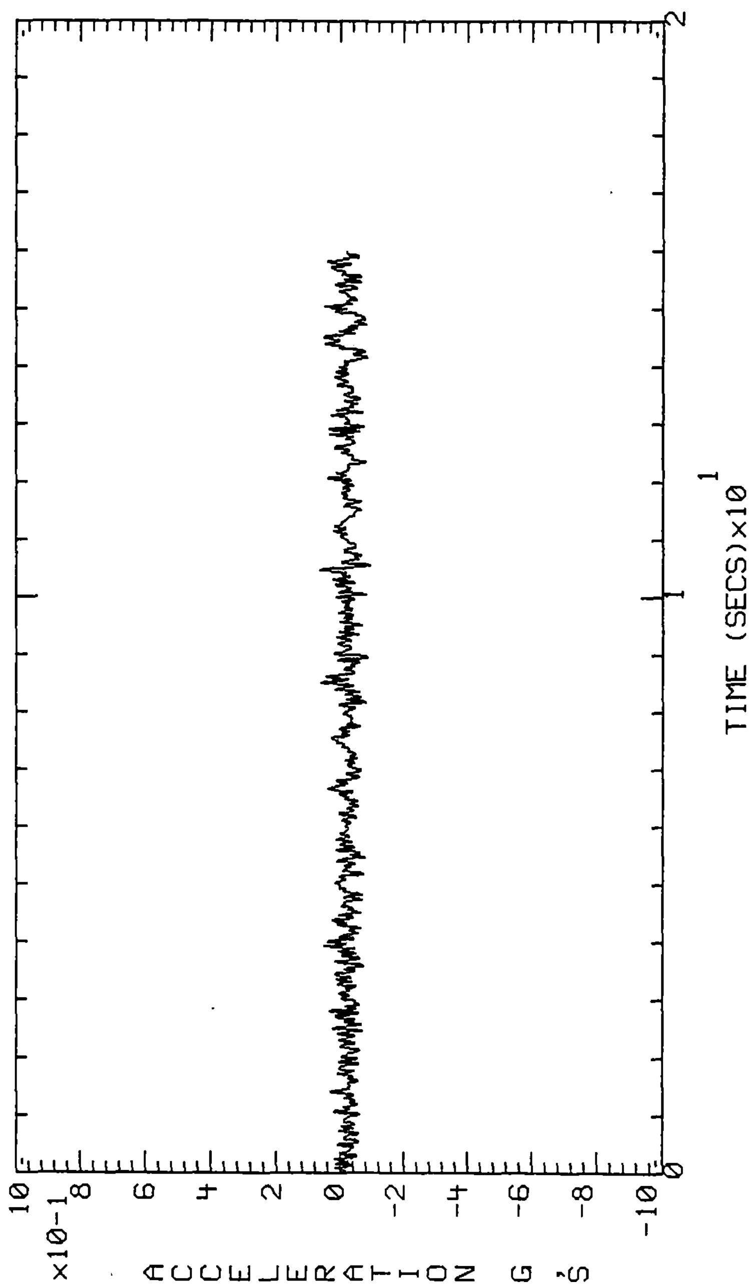


FIGURE 7-1. AXLE ACCEL. TIME HISTORY OF 30 MPH TOFC RUN.

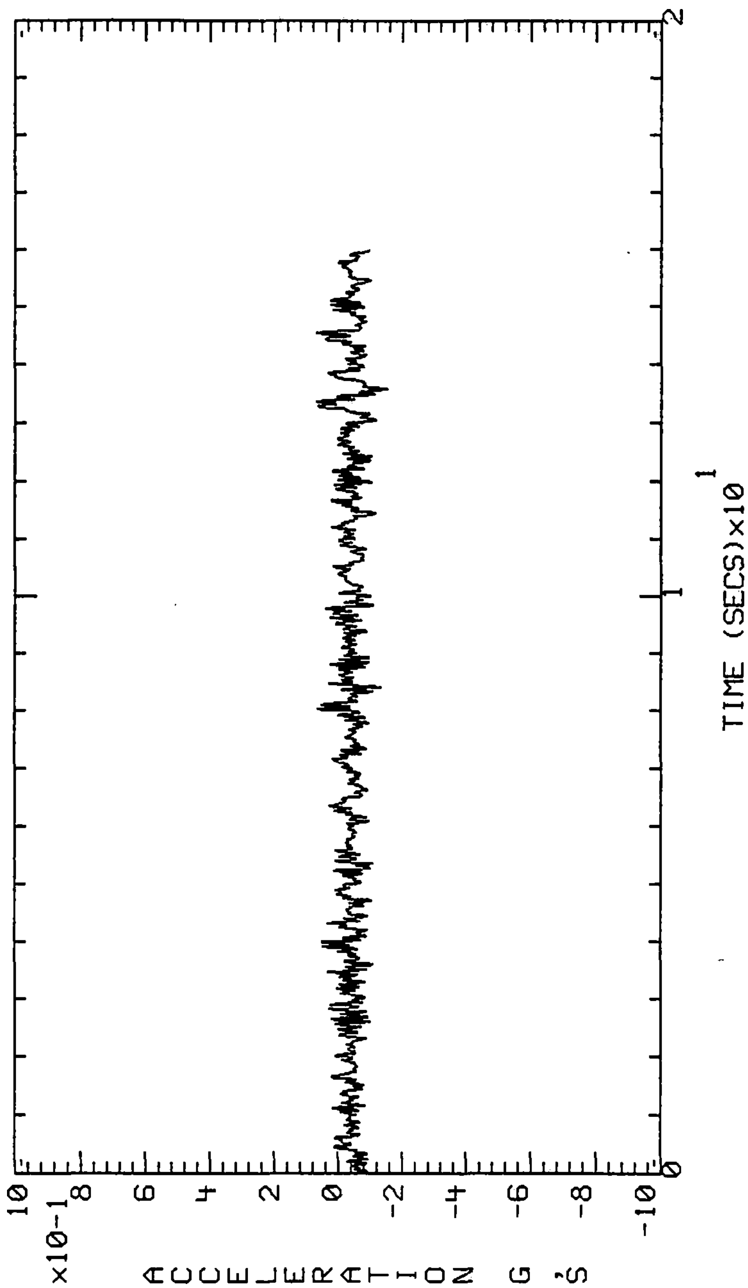


FIGURE 7-2. AXLE ACCEL. TIME HISTORY OF 35 MPH TOFC RUN.

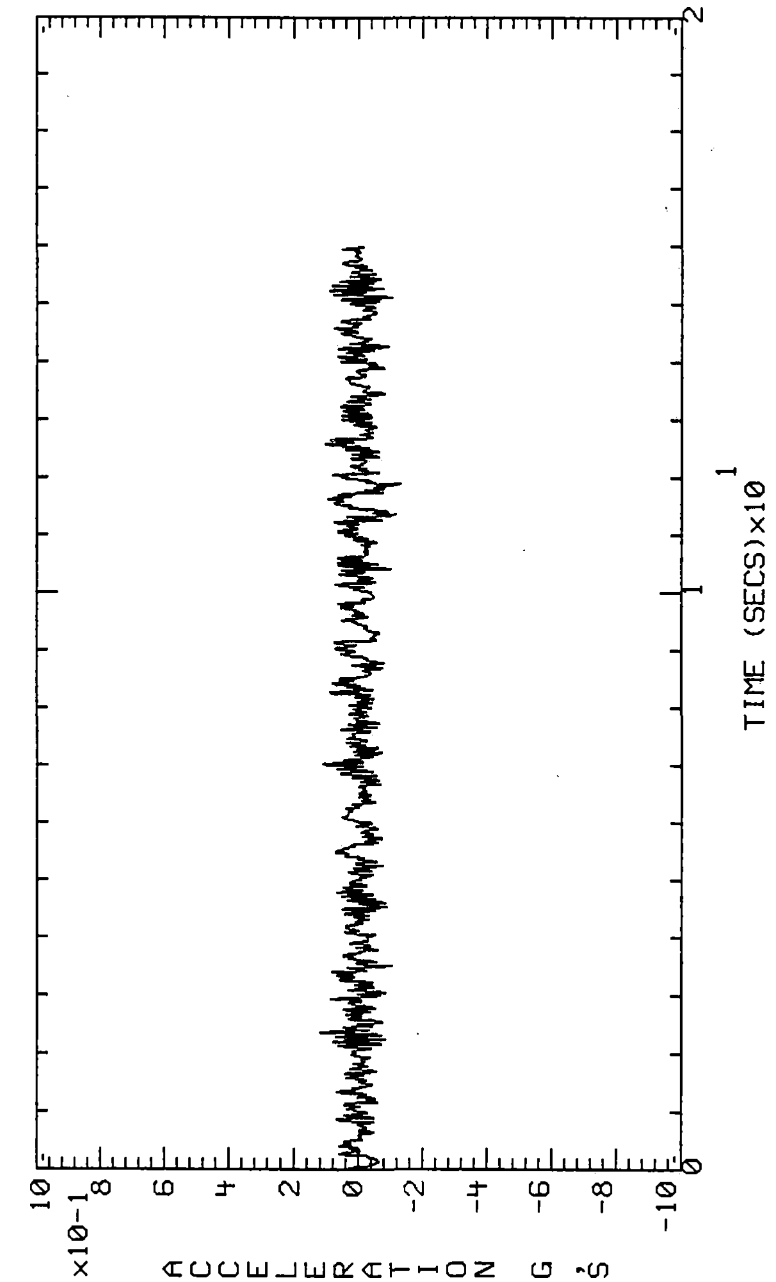


FIGURE 7-3. AXLE ACCEL. TIME HISTORY OF 40 MPH TOFC RUN.

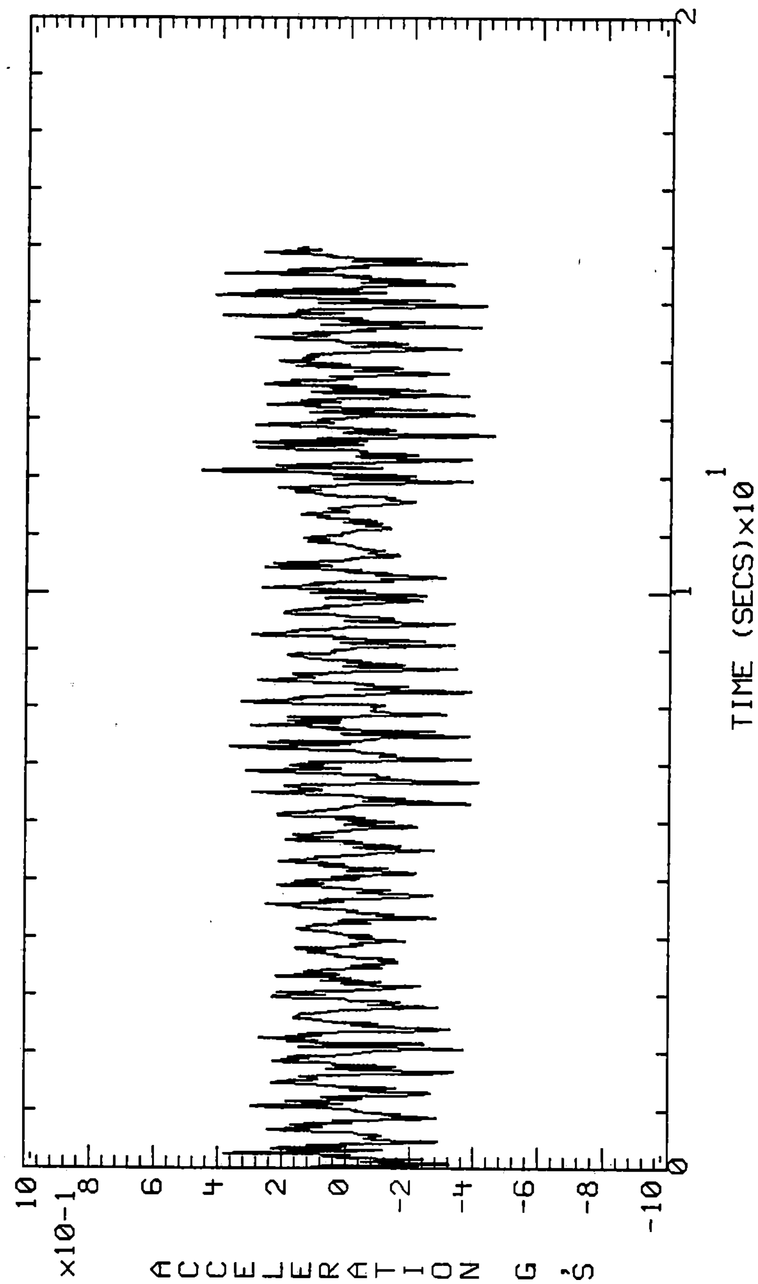


FIGURE 7-4. AXLE ACCEL. TIME HISTORY OF 42 MPH TOFC RUN.

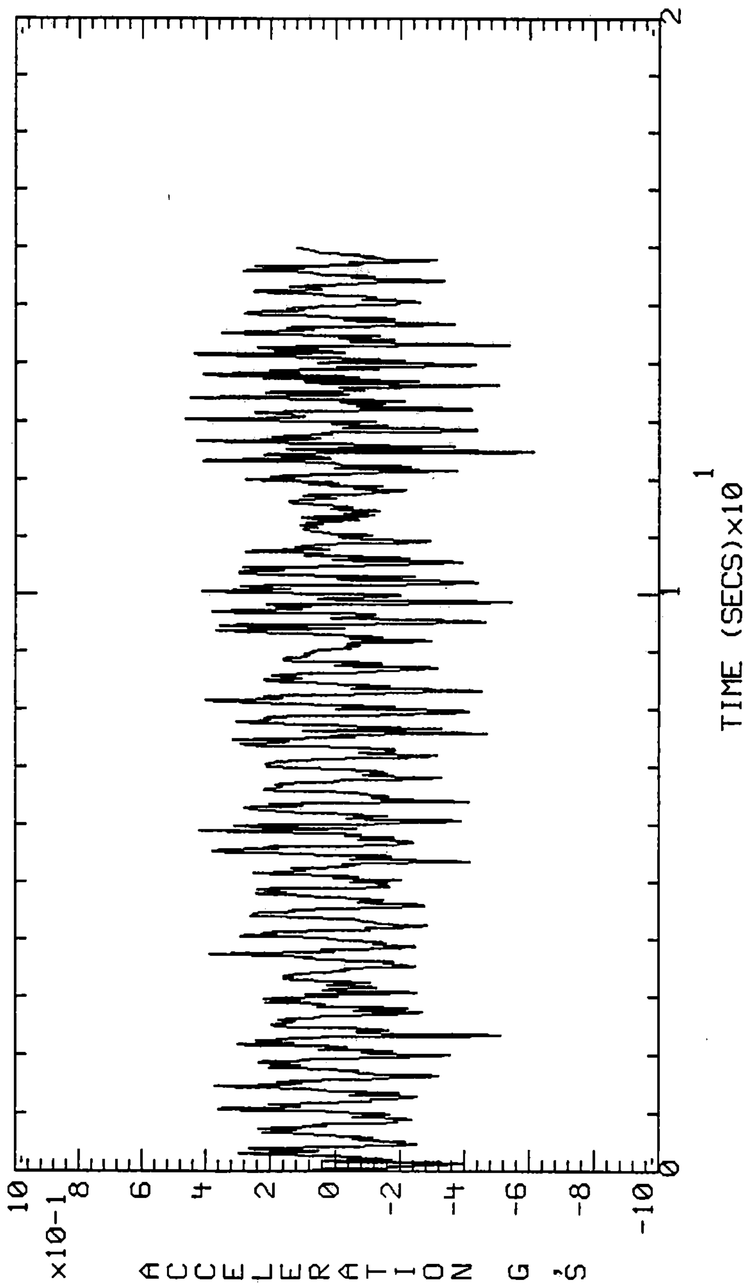


FIGURE 7-5. AXLE ACCEL. TIME HISTORY OF 45 MPH TOFC RUN.

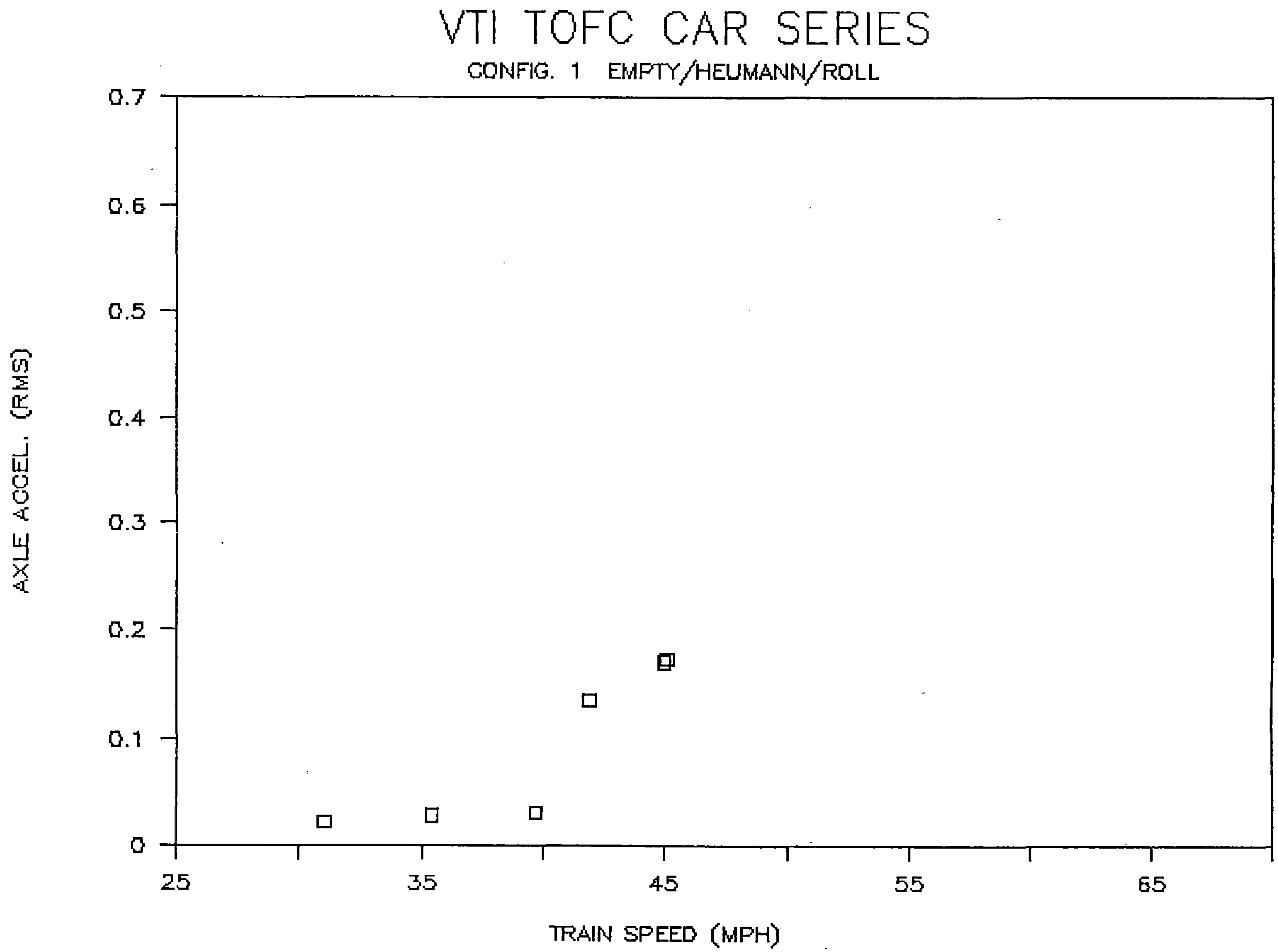


FIGURE 7-6. RMS AXLE ACCEL. WITH SPEED FOR BASE TOFC CAR.

rate of acceleration that occurs with speed. At this point the random vibration at the axle, as well as that due to the roll frequency of the wheel, becomes insignificant as the hunting frequency dominates. This effect can be seen in the example time histories for the 40 and 42 mph runs. The amplitude has increased significantly and the fundamental 2.7 Hz frequency is becoming apparent. The speed of 45 mph produced sustained flange-to-flange hunting.

The frequency content for these three examples is evident using the PSD plot for the the acceleration measurements. Figure 7-7 contains the PSD for the 40 mph run which has no dominant frequency and very low amplitude. The onset observed at 42 mph shown in Figure 7-8 indicates the development of hunting with increased amplitude and the emergence of a fundamental frequency at 2.7 Hz. The highest test speed run for this configuration is shown in Figure 7-9, the driving frequency is clearly shown and the amplitude has increased by over 35 percent.

The phase relationship between the two axles on the lead A-end test truck is shown in Figure 7-10. The presence of zero angle in the dominant hunting frequency range of 2.5 to 3.5 Hz reinforces the observation that the axles operated in unison during full flange-to-flange hunting.

7.3 CARBODY ACCELERATION

Carbody RMS acceleration for the base car over the range of speed conditions is presented in Figure 7-11. The location for this accelerometer is at the extreme end of the flat car, in the corner area of the deck. Compared to the equivalent plots for the axle acceleration, the RMS values are approximately 50 percent higher at the hunting test speeds. The input of energy is then exciting the carbody and the lateral acceleration is greater than that experienced by the axle, with yet the same basic frequency.

The phase angle between the carbody and truck is represented in Figure 7-12.

The phase angle in this plot shows less coherent motion between the car and truck in comparison to the case between axles.

The motion of the A-end of the car relative to the B-end is presented in Figure 7-13. The phasing of the two reflects this moderate relationship at 2.5 to 3.0 Hz.

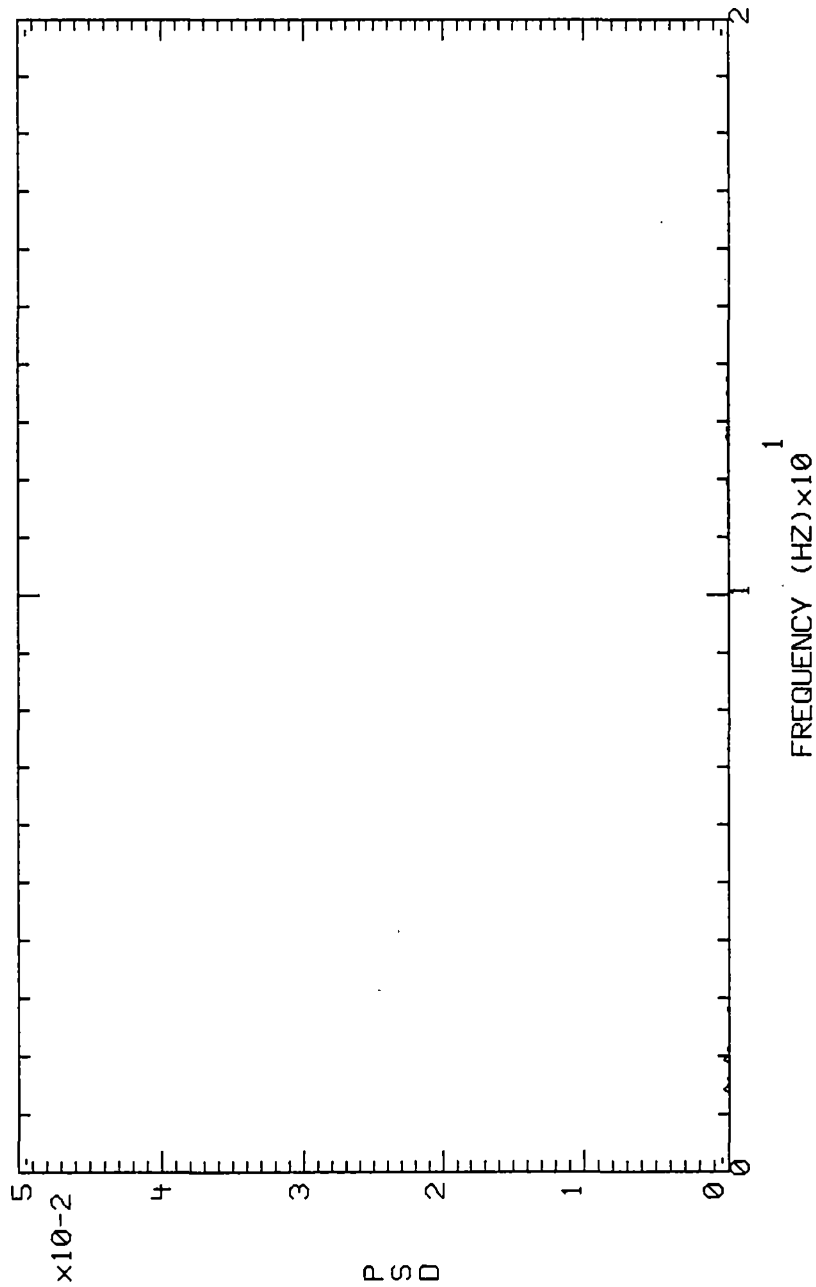


FIGURE 7-7. PSD FOR 40 MPH RUN OF BASE TOFC CAR.

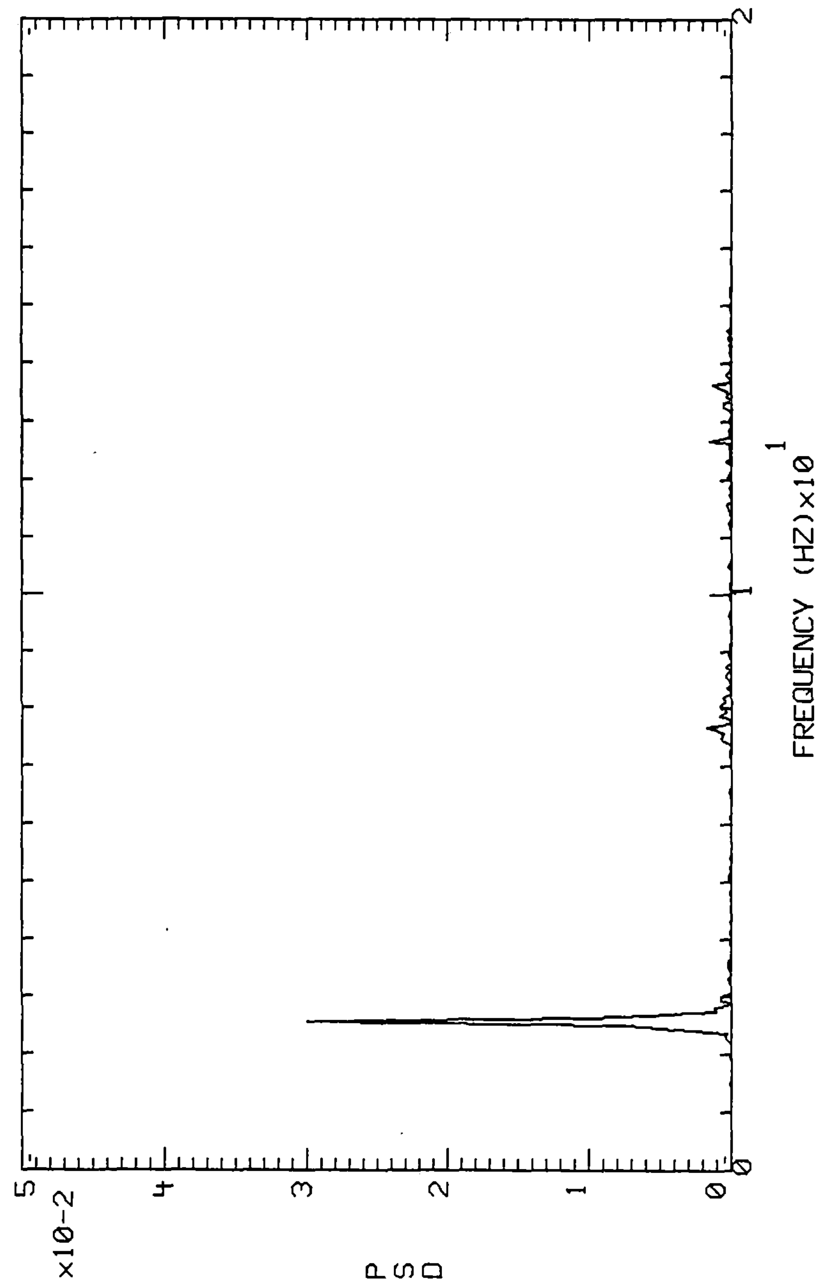


FIGURE 7-8. PSD FOR 42 MPH RUN OF BASE TOFC CAR.

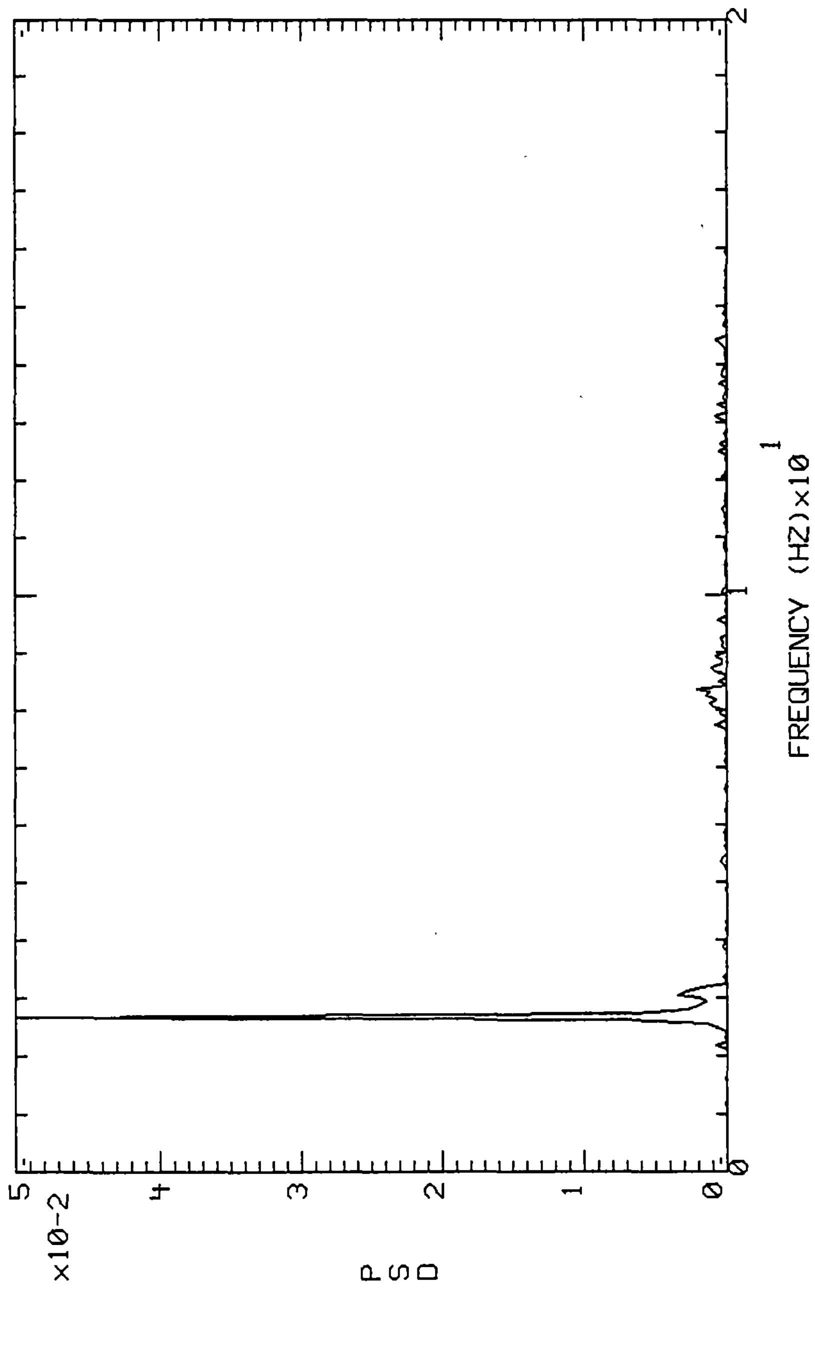


FIGURE 7-9. PSD FOR 45 MPH RUN OF BASE TOFC CAR.

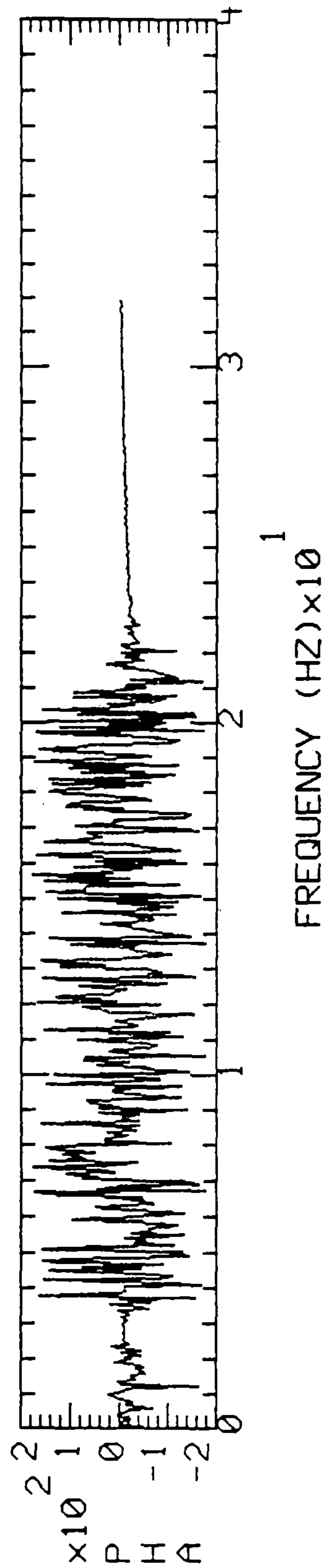


FIGURE 7-10. CROSS POWER SPECTRUM, AXLE-TO AXLE PHASE ANGLE.

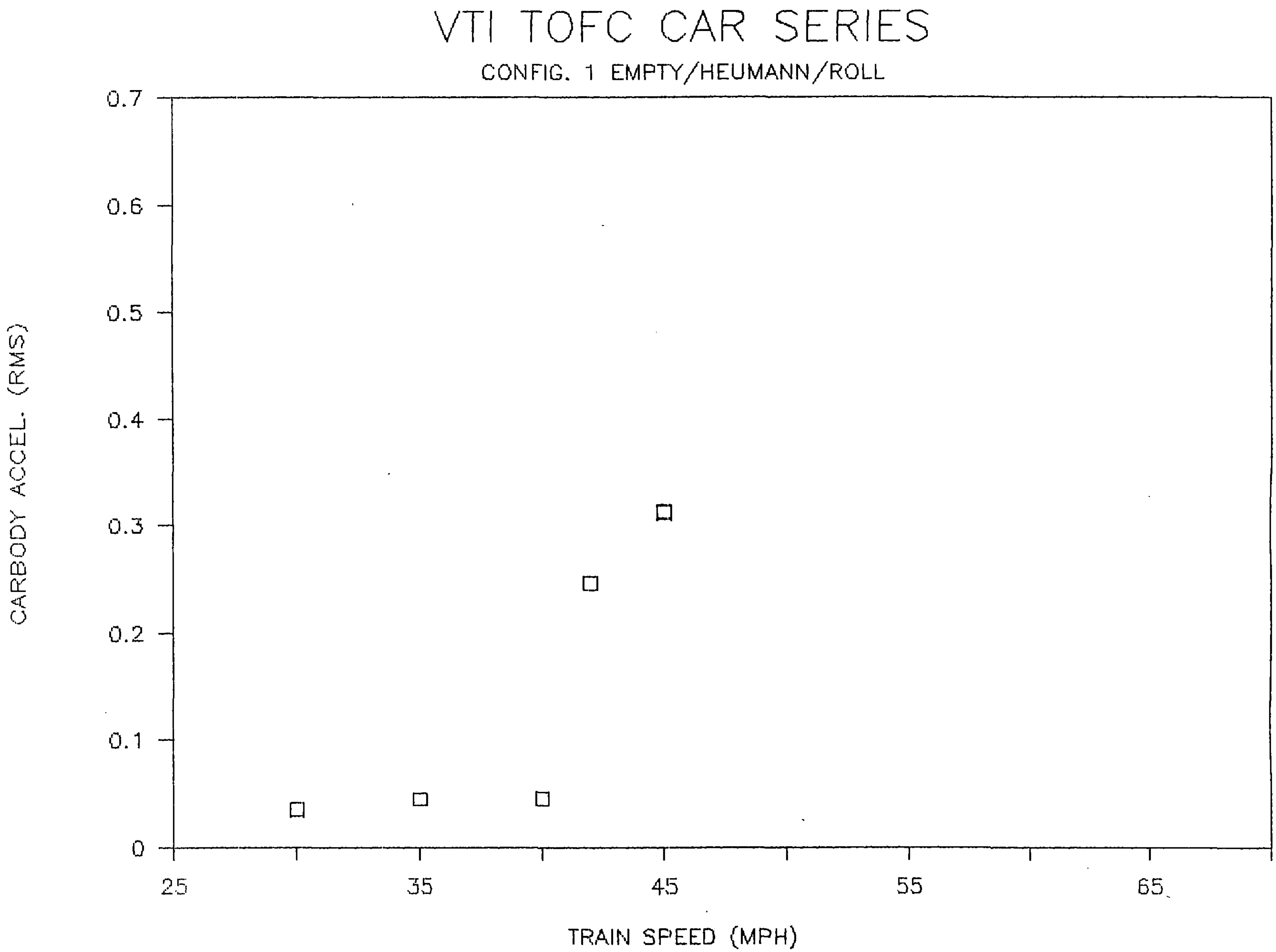


FIGURE 7-11. RMS CARBODY ACCEL. WITH SPEED FOR BASE TOFC CAR.

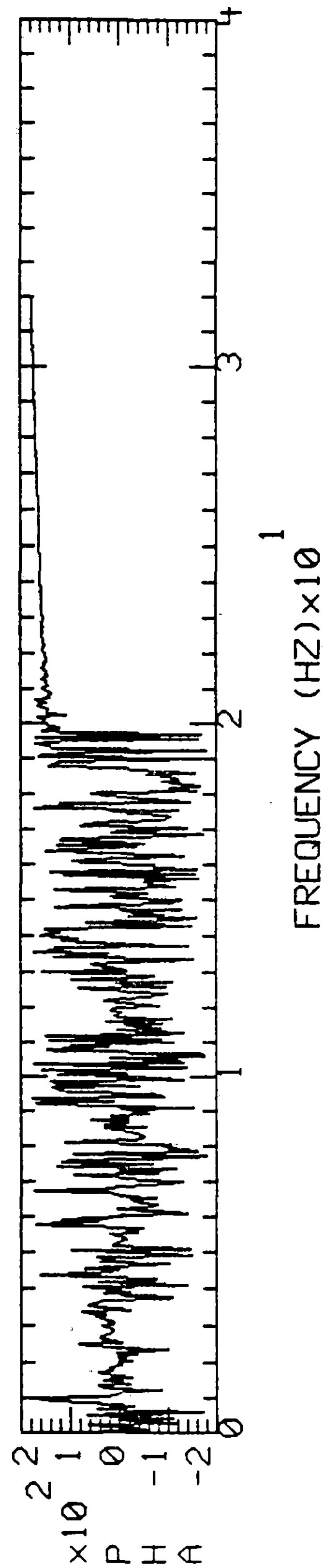


FIGURE 7-12. CROSS POWER SPECTRUM, CAR-TO-TRUCK PHASE ANGLE.

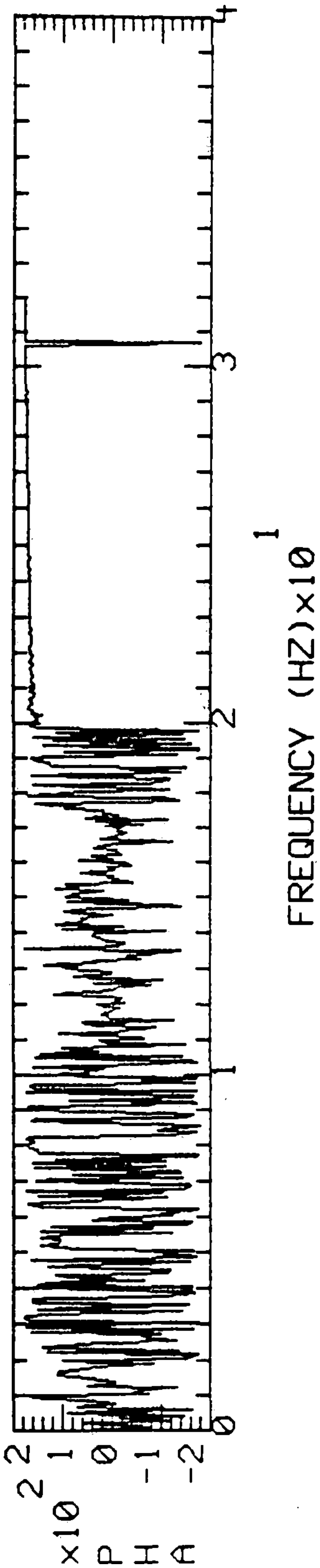


FIGURE 7-13. CROSS POWER SPECTRUM, A-END RELATIVE TO B-END OF CAR.

7.4 L/V RATIOS

Having measured both vertical and lateral loads simultaneously during the hunting runs, an analysis of L/V ratio was carried out. The study of this measurement indicates the derailment tendencies for the various configurations and speeds. The duration of a high L/V event was also examined.

The minimum amount of time that a critical L/V ratio need be present in order to result in a flange climb derailment is unknown. As an illustration of how the VTI instrumented wheelset data could be applied to derailment analysis, instantaneous values will be used. This results in a very conservative analysis, since the L/V drops off very quickly from the peak.

The worst configurations from a stability viewpoint were the baseline cars. The calculated L/V ratio at the 45 mph test speed for the baseline intermodal flat is given in Figure 7-14. The data is a result of the simultaneous collection of lateral and vertical forces. The lateral and vertical signals were low band pass filtered at 5.0 Hz prior to the L/V calculation. This eliminates all but the hunting related response.

The peak values fluctuate at this test speed and the maximum L/V is used to determine the risk of derailment. In order to evaluate the derailment potential, Nadal's criteria is applied.[17] Although by today's standards it is somewhat simple in nature, it is considered a conservative estimate of derailment proneness for steady state conditions and has been widely used in the railroad community. The force systems are based on the following equations. The Nadal criterion is based on the assumption of simple equilibrium of the forces between wheel and rail at the single point of flange contact.

L = N sin(theta)-mu N cos(theta)
V = mu N sin(theta)+N cos(theta)

In its simplest form the L/V criterion is calculated as:

L/V = tan (theta-arctan mu)

where:
L = lateral force
V = vertical force
theta = maximum contact angle between wheel and rail
mu = coefficient of friction

As can be seen, the criterion is influenced by the maximum contact angle and the coefficient of friction at that point of contact. The three wheel profiles used in the VTI investigation yield the following maximum contact angles:

- AAR 1:20 - 70°
- Average FAST Worn - 70°
- CN Heumann - 65°

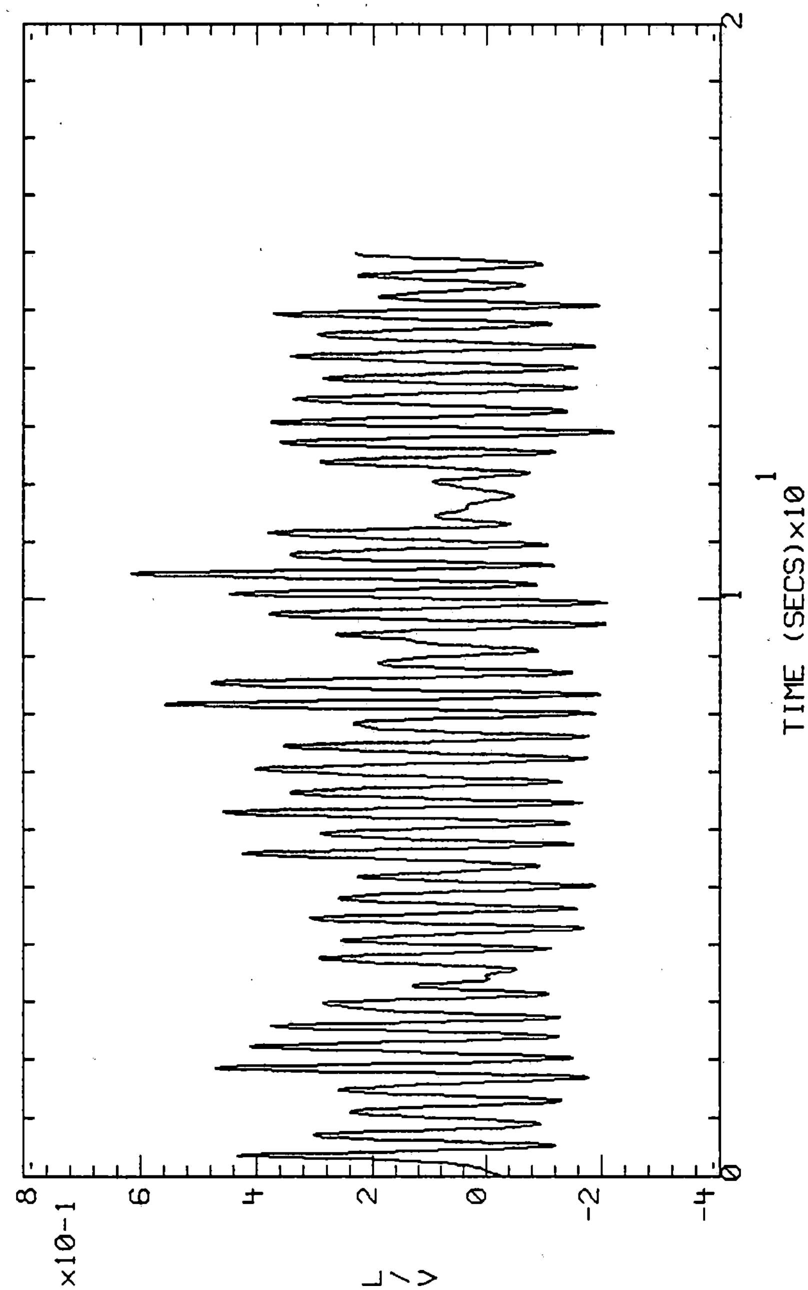


FIGURE 7-14. L/V AT 45 MPH FOR BASELINE TOFC.

Taken from the data in the intermodal flat time history, a peak value of .62 occurs at about 10.5 seconds into the test run. This is with a CN wheel on AREA 136 RE rail. An analysis of the contact geometry for this combination yields a maximum contact angle of 64 degrees. Assuming a coefficient of friction of 0.45, a critical L/V of 0.83 results from Nadal's theory. This is nearly 30 percent higher than the maximum instantaneous observed L/V. An analysis of the remaining configurations indicated no greater tendency toward derailment. This would indicate that potential flange climb derailment was never a problem during the test.

It is important to remember that the track conditions were ideal and only a limited range of contact conditions were experienced. An exceptional combination of wheel/rail profile, or excessive speed resulting in high dynamic input could result in a "hunting" flange climb derailment.

Since peak L/V values are speed dependent, the ratio increases steadily with speed once hunting has been achieved. The trend in Figure 7-15 indicates that a critical level could be experienced within the range of normal train operation. This is still for an instantaneous measurement, the significance of time extent has been ignored.

7.5 CORRELATION OF AXLE ACCELERATION AND WHEEL/RAIL FORCES

As stated in the objectives, an understanding of the force environment between the wheel and rail during hunting is relatively undocumented. The VTI hunting test presented an excellent opportunity to explore this area. At the time this test was executed, the checkout and validation of the 33-inch and 36-inch instrumented wheelsets had just been completed. The profile applied to the running surface was the modified CN Heumann, which provided hunting at a relatively low operating speed. The standard interchange 1:20 wheel profile had been applied in the past to load measuring wheels. This may have been appropriate for the study of curving behavior for some vehicles but provided little information for hunting investigations. This is due to the low effective conicity provided by the 1:20 profile.

Having seen from the previous section that flange-to-flange hunting occurred at 45 mph with the Heumann profile, this test run was used in the analysis. Shown in Figure 7-16 is a time history for the lateral wheel/rail loads measured by the instrumented CN Heumann wheelset during hunting. It contains the same basic characteristics as acceleration in terms of frequency as shown in the power spectral density plot in Figure 7-17. The zero degree phase at the 2.5 to 3.0 Hz range verifies this. The force time history shows a positive bias. This is the result of the lateral component of the vertical load on a tapered wheel. Considering the contact angle of the wheel and the static vertical load, this amount of lateral force offset is reasonable.

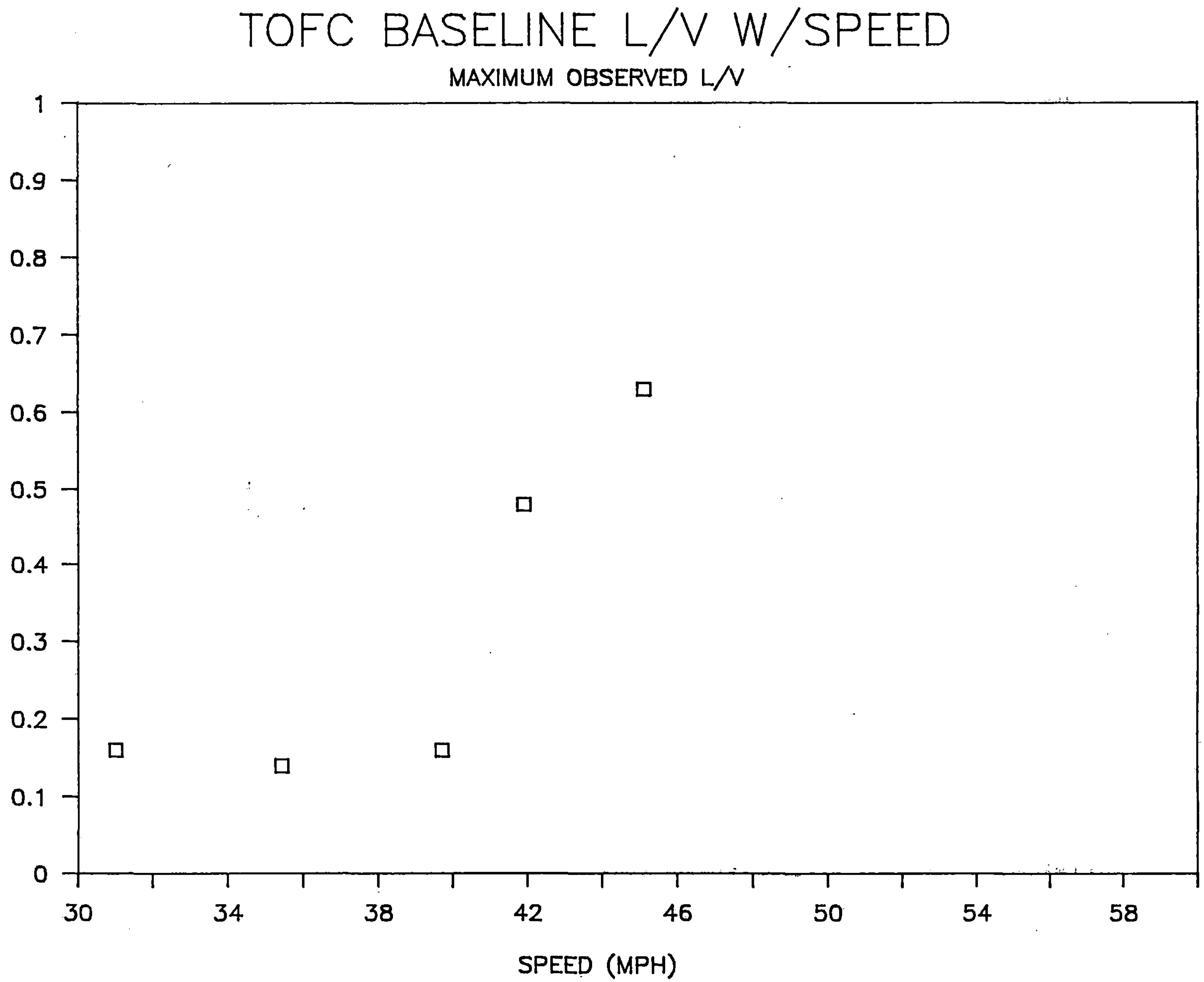


FIGURE 7-15. TOFC BASELINE L/V VS. SPEED.

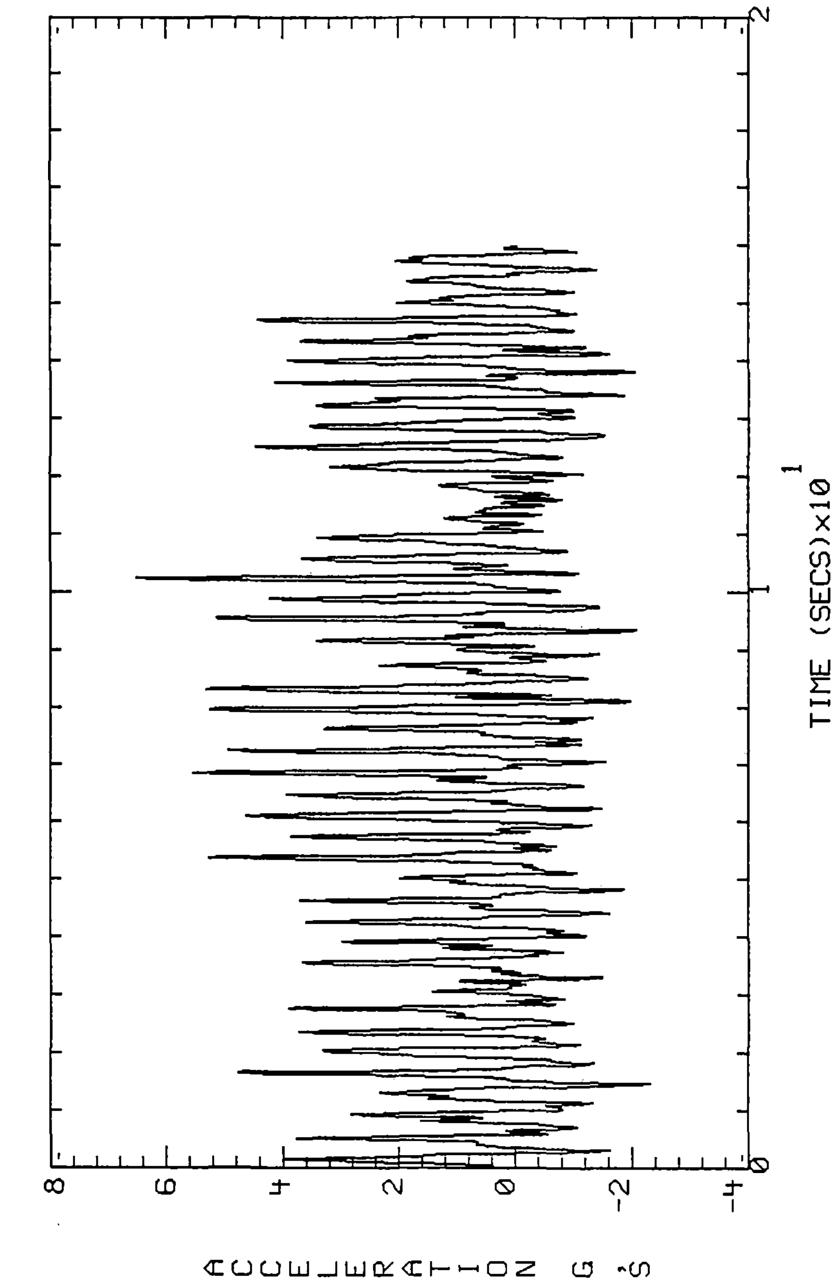


FIGURE 7-16. TIME HISTORY FOR LATERAL WHEEL/RAIL LOADS.

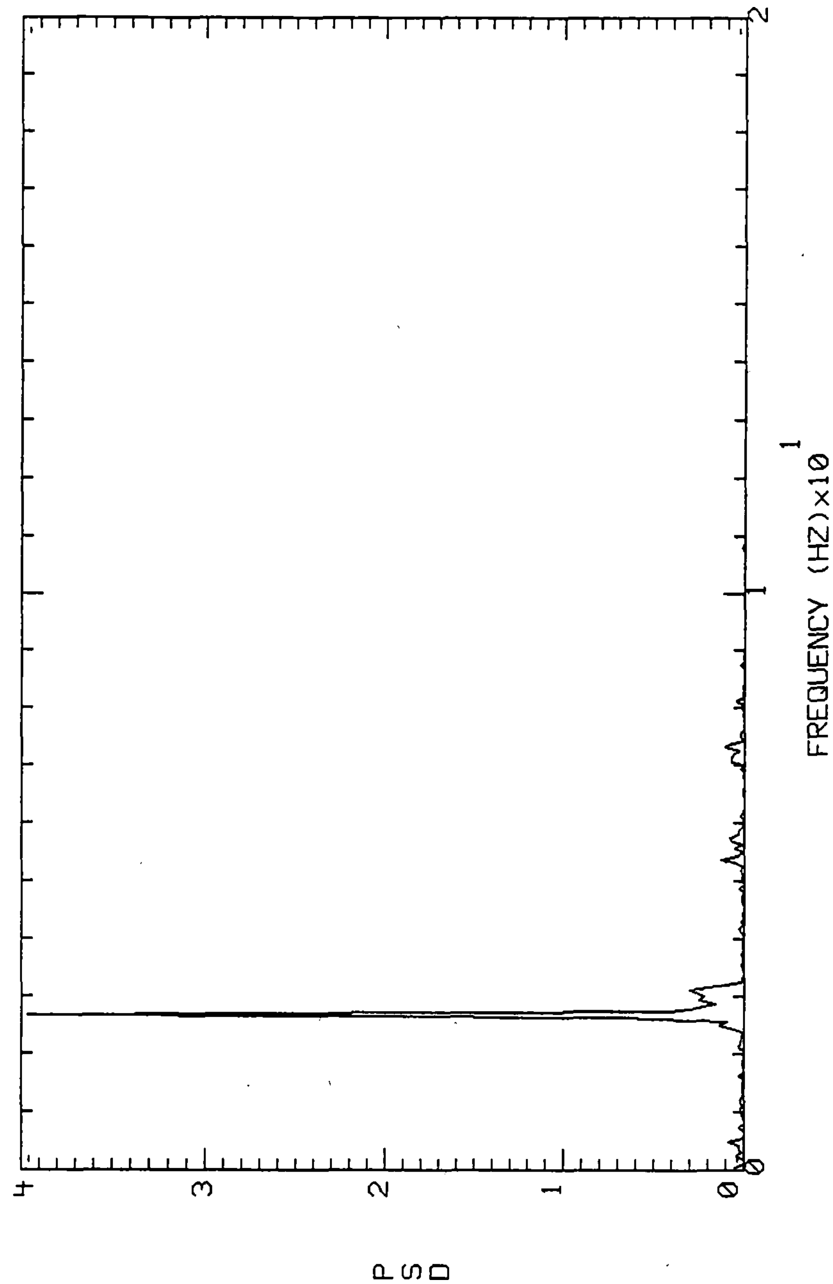


FIGURE 7-17. PSD OF LATERAL WHEEL/RAIL FORCE.

The correlation of the lateral load data and the acceleration data initially involved a simple x,y plot of discrete data points as in Figure 7-18. Although a convincing relationship is present, there is still a large scatter in the data. Having established the hunting frequency as being less than 5.0 Hz, all frequency content above this cutoff was filtered out. The result of this is given in Figure 7-19. Here greater correlation has resulted, as well as an increase in the slope from 14.02 to 17.16. This relates to a 17,160 pound lateral force at 1.0 g acceleration. The intercept of 0.773 indicates a lateral force of 773 pounds at zero acceleration which is due to the net effect of the vertical load on the tapered running surface. The slope coefficient of 17.16 relates to a 17,160 pound change in lateral force for each 1.0 g of acceleration. Taken into account with the 773-pound intercept, this equates to an effective force for the single axle of 17,932 pounds at 1.0 g acceleration. This compares very favorably with a total axle load of 15,375 pounds.

This relationship then suggests that the carbody, trucks and axles are operating, for the most part, as a single entity. From this relationship, it is possible to deduce the wheel/rail loads from lateral axle accelerations produced under the same axle load but with different wheel or rail profiles.

Figure 7-20 contains the x,y plot of force and axle acceleration data for the intermodal flat cars with constant contact side bearings. This is the result of the final test run at 64 mph. Full flange-to-flange hunting was nearly continuous. The slope of the data indicates 8,999 pounds per 1.0 g's of acceleration. This is half the value for the roller side bearing example. Examination of carbody acceleration data indicates, that with constant contact side bearings, decoupling between the carbody and bolster occurs. The difference in carbody acceleration of the baseline vehicle and constant contact side bearing equipped vehicle is given in Figure 7-21. The later test case produced carbody excitation of about 50 percent of the roller side bearing equipped intermodal flat car. It is proposed that the change in carbody mode explains the differences in the wheelset force/acceleration relationship. These sets of data provided the only sustained hunting cases for the intermodal flat car and further investigation of this should be carried out in the future.

7.6 AXLE ACCELERATION INFLUENCED BY AXLE LOAD

Having established the basic relationships in the prior section, the effect of increased lading or axle load was investigated. The same base test car was run with two empty 40-foot trailers which increased the weight on rail from 68,000 pounds to 159,450 pounds.

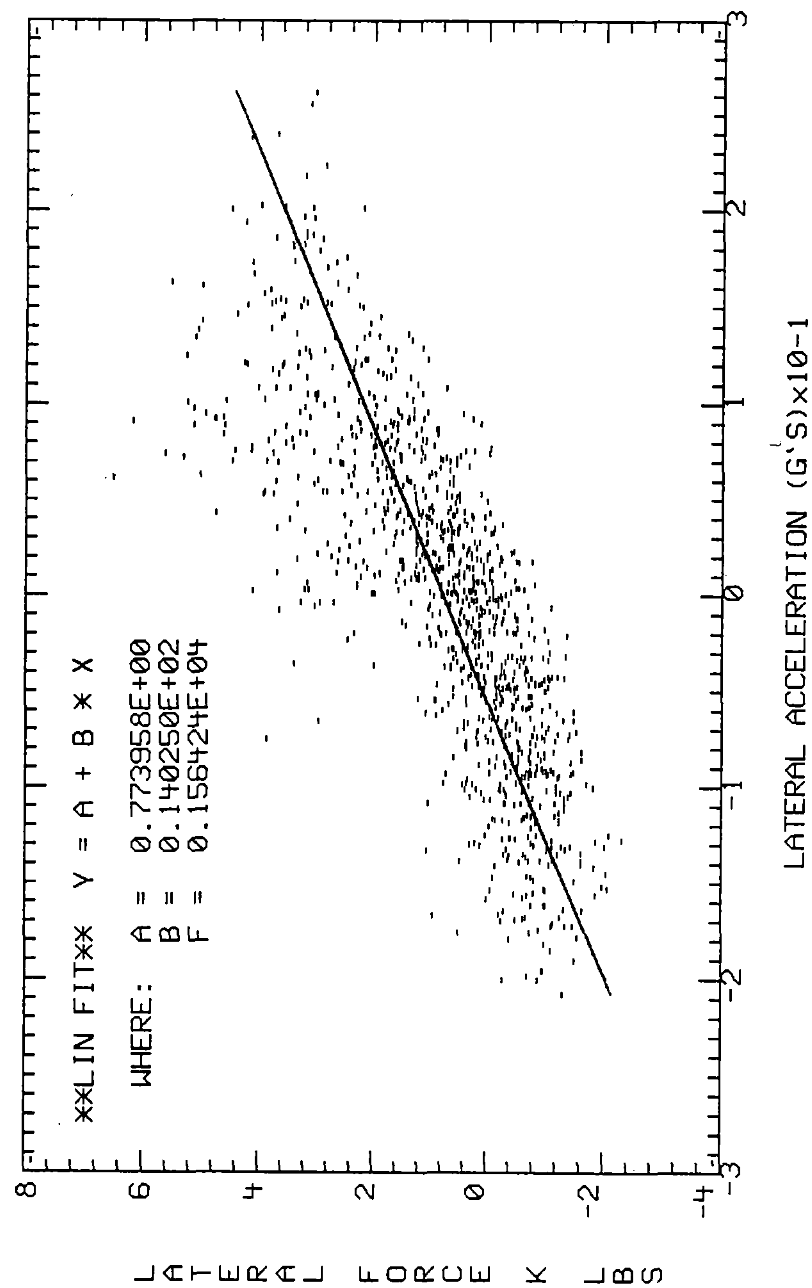


FIGURE 7-18. X,Y PLOT OF DISCRETE AXLE ACCELERATION AND LATERAL WHEEL FORCE-FILTERED AT 15 HZ.

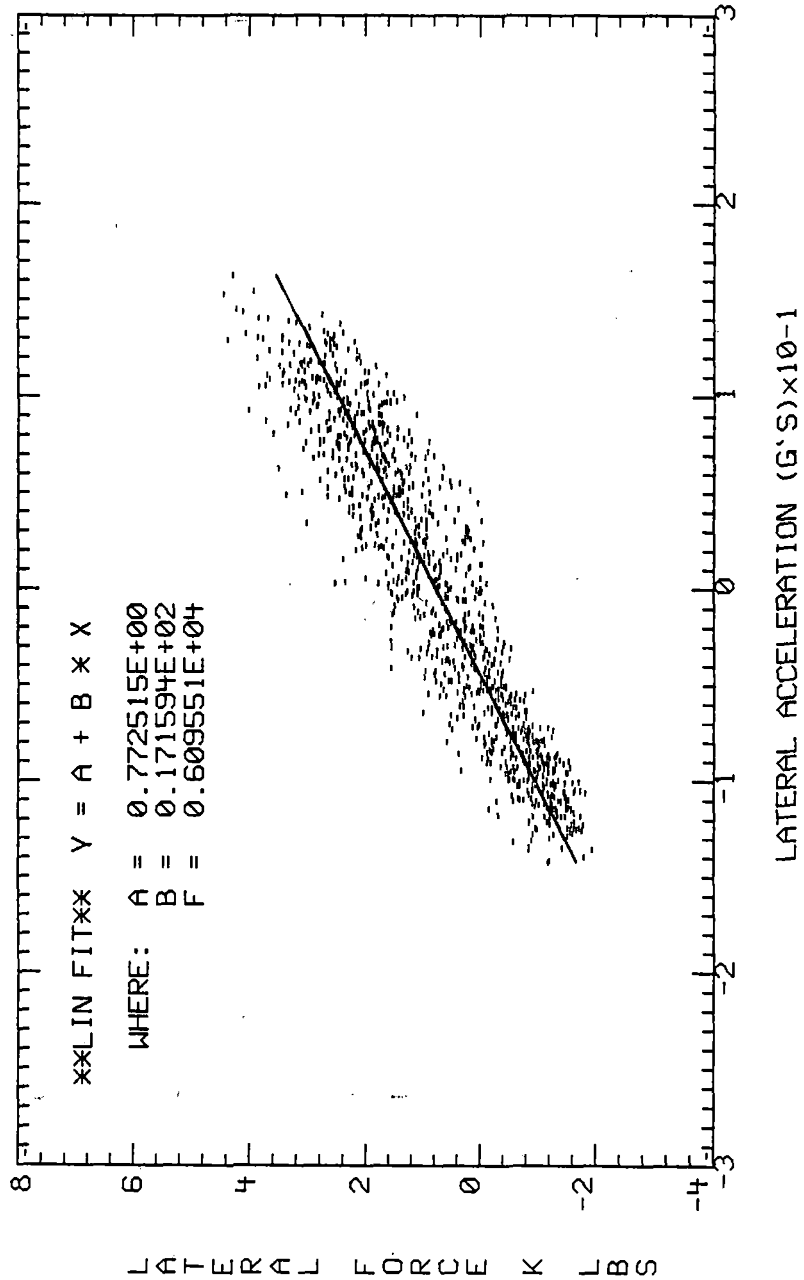


FIGURE 7-19. X,Y PLOT OF DISCRETE AXLE ACCELERATION AND LATERAL WHEEL FORCE-FILTERED AT 5 HZ.

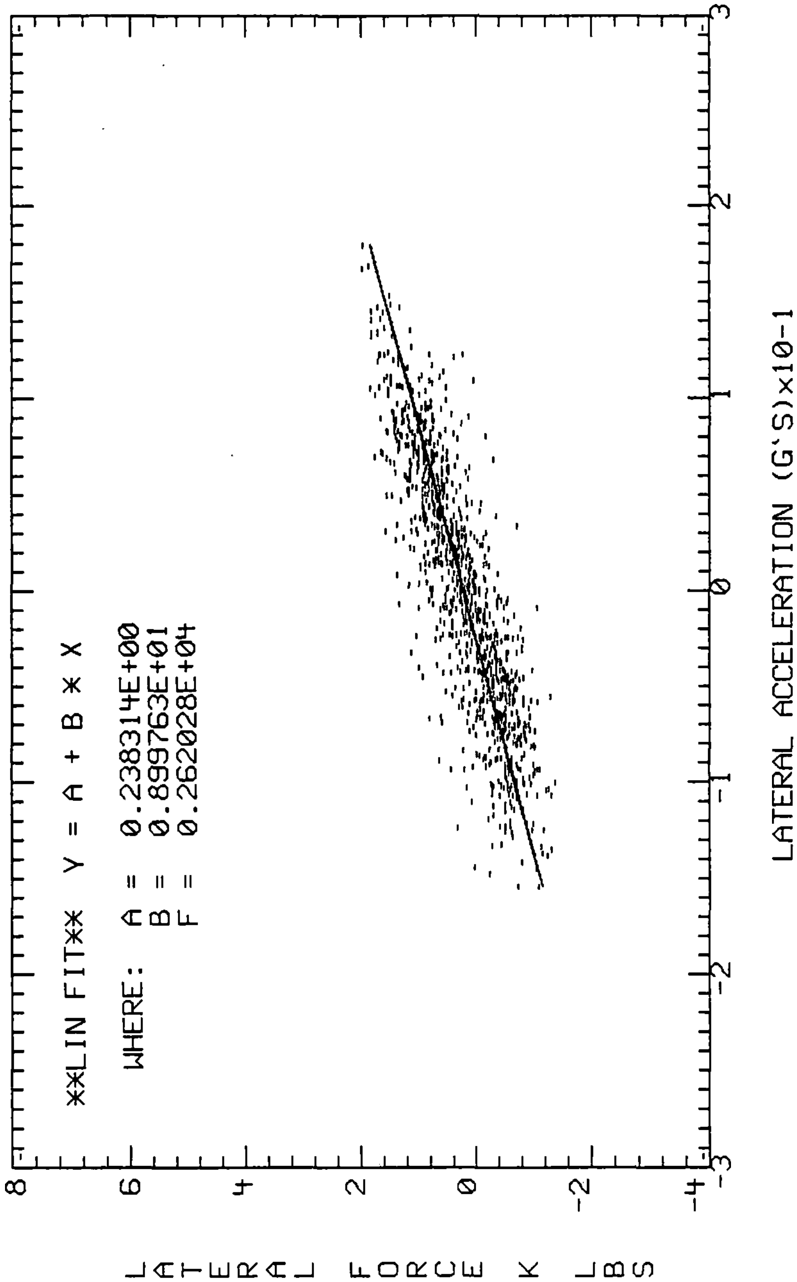


FIGURE 7-20. X,Y PLOT OF FORCE AND ACCELERATION WITH CONSTANT CONTACT SIDE BEARING.

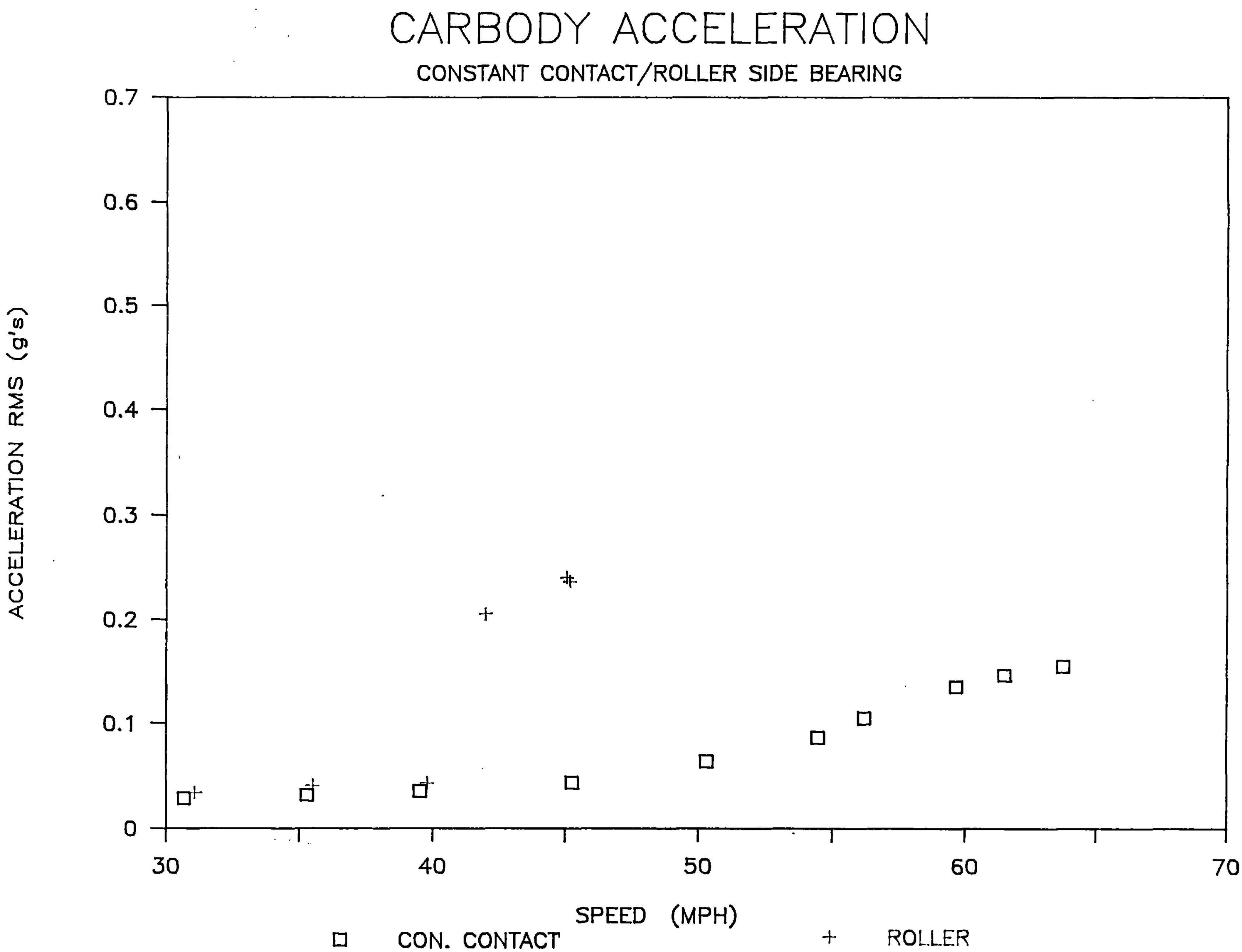


FIGURE 7-21. CARBODY ACCELERATION - CONSTANT CONTACT VS. ROLLER.

The effect on axle acceleration is presented in Figure 7-22. In contrast to the unladen flat, in which the maximum test speed was 45 miles per hour, the loaded case ran up to 70 mph. The axle acceleration was significantly less than the unloaded configuration at all test speeds. The loaded car produced a maximum RMS level of .12 g. This is approximately half of the measured RMS acceleration at the 45 mph test speed for the unloaded car. At the 45 mph speed, the loaded car yielded only a 0.02 to 0.05 RMS. The carbody response in Figure 7-23 is correspondingly lower in comparison to the base case car. The onset of hunting was first perceptible at the last test speed of 70 mph, indicating that the axle load had a significant influence on the hunting characteristic of a flat car.

The lateral forces were measured during these runs, but showed no incremental hunting influence since the threshold was never achieved.

7.7 SIDE BEARING EFFECT

A well-known modification to reduce the hunting tendency and, therefore, raise the hunting threshold speed is through the use of constant contact side bearings. One type of side bearing was investigated as part of the VTI program. By providing a higher rotational stiffness and damping, the onset speed is raised since it requires more force and, therefore, a higher speed in order to exceed the breakaway torque. At this point sustained hunting will occur. The data in Figure 7-24 shows the influence of the side bearing modification on axle acceleration. The test car was unloaded and in all respects the same as the base car with the exception of side bearings. Here an onset speed of approximately 52 mph was detected. The transition from the onset continues on up to a speed of 65 mph. At this point, isolated accelerations in excess of the established safety limits were detected. Overall acceleration of the axles was still less than those for the maximum test speed for the base car. Likewise, the carbody acceleration shown in Figure 7-25 was also relatively lower.

A loaded car application of constant contact side bearings was made, and this is shown as it affected axle acceleration in Figure 7-26. As was expected, the vehicle ran stably over the entire range of test speeds.

7.8 WHEEL PROFILE INFLUENCE

Effective conicity results from the shape of the rail, the shape of the wheel, and the relative gage. Because all test runs were conducted on the same track, the conicity was controlled by the wheel shape. The running surface of wheels under the base car was based on the CN Heumann. Alternative wide flange Heumann, FAST worn wheel, and standard 1:20 wheel profiles were also evaluated.

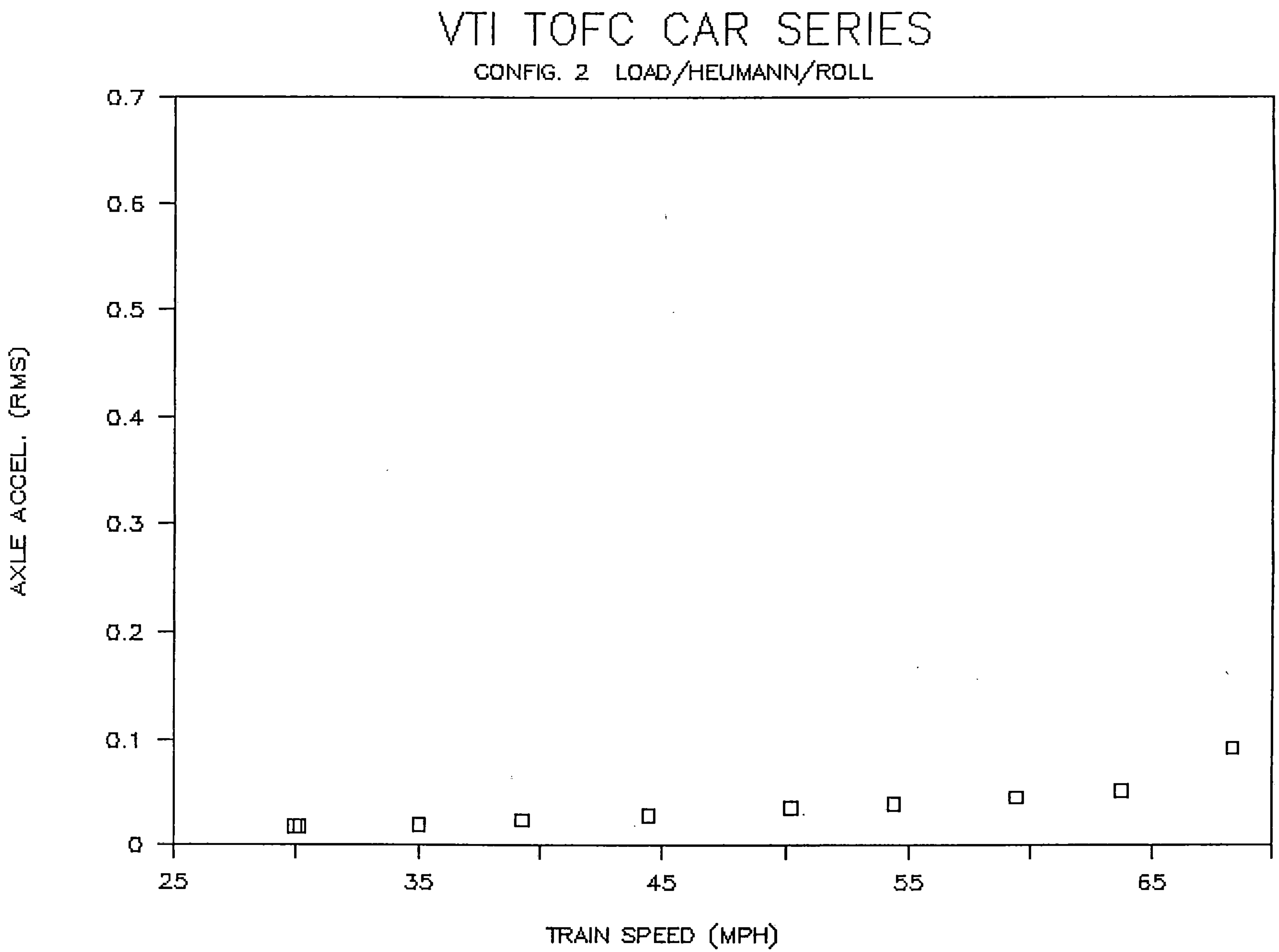


FIGURE 7-22. RMS AXLE ACCEL. WITH SPEED-LOADED FLAT.

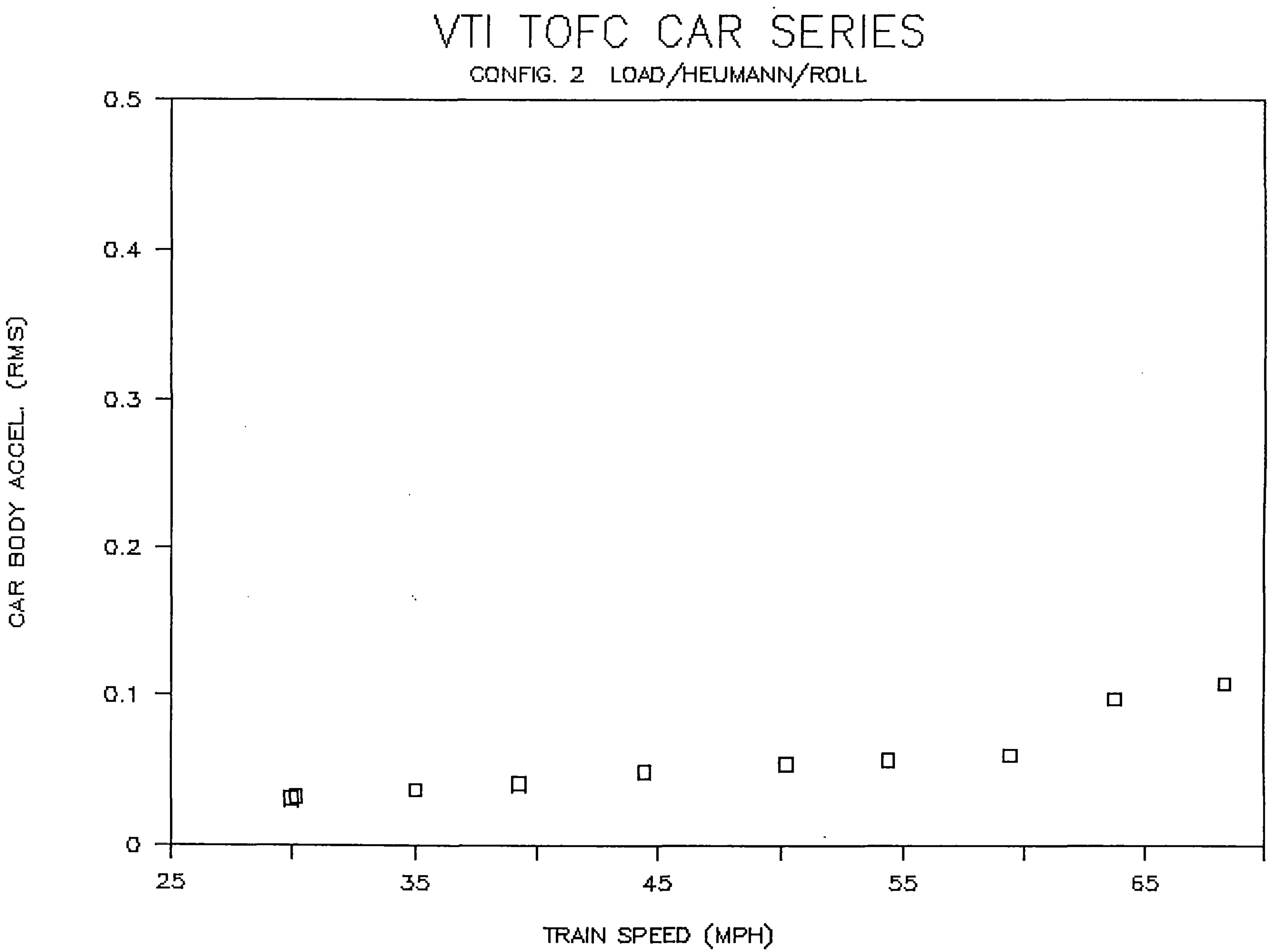


FIGURE 7-23. RMS CARBODY ACCEL. WITH SPEED-LOADED FLAT.

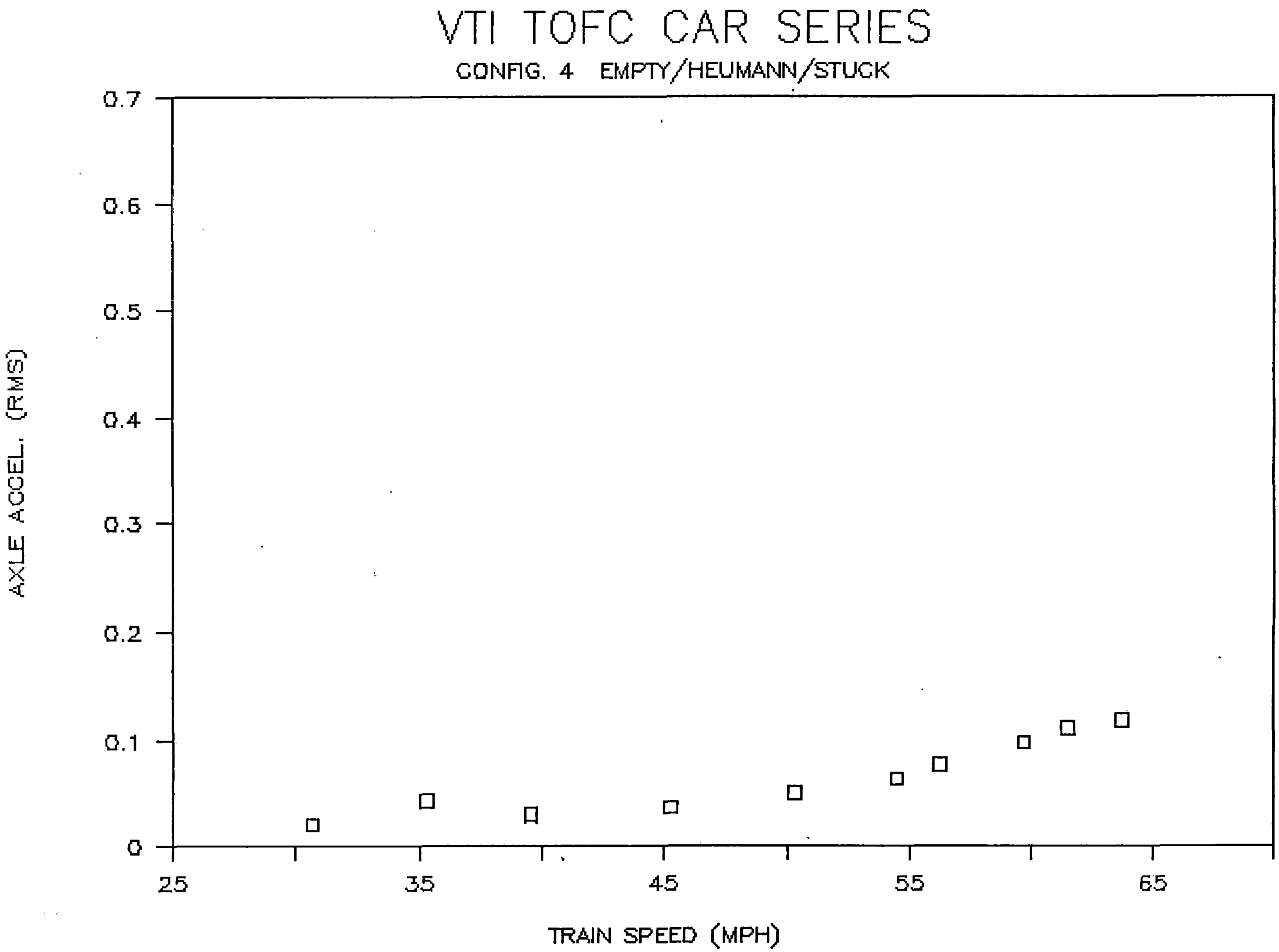


FIGURE 7-24. RMS AXLE ACCEL. WITH SPEED-LOADED FLAT W/ CONSTANT CONTACT SIDE BEARINGS.

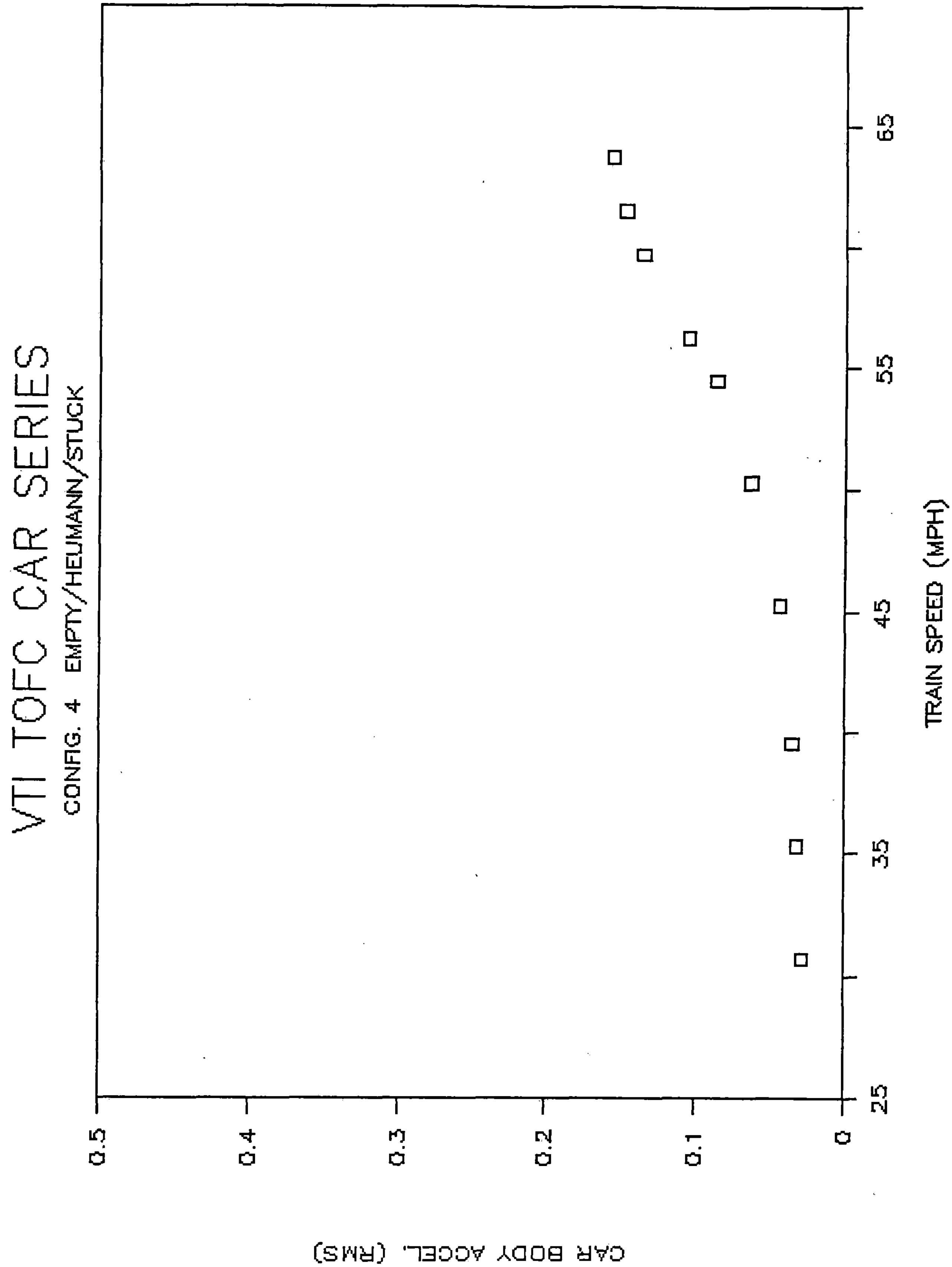


FIGURE 7-25. RMS CARBODY ACCEL. WITH SPEED-LOADED FLAT W/ CONSTANT CONTACT SIDE BEARINGS.

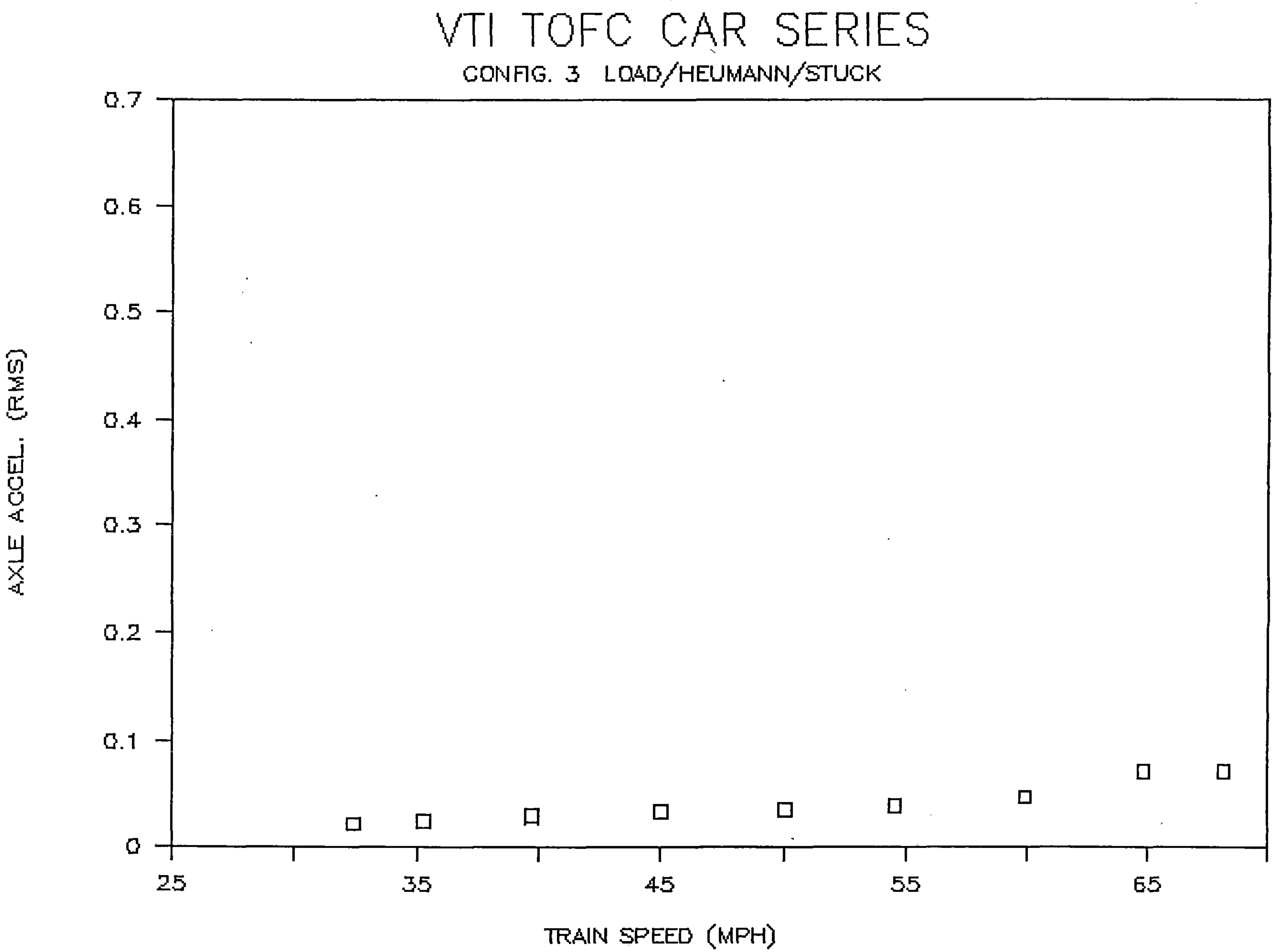


FIGURE 7-26. RMS AXLE ACCEL. WITH SPEED-LOADED FLAT W/ CONSTANT CONTACT SIDE BEARINGS.

Average FAST worn wheel profile on 136 RE rail produces an average effective conicity of 0.15. This wheel was used to see what effect this change would have on onset speed and the associated accelerations.

Figure 7-27 plots the average RMS acceleration for the four test axles with speed. The test car was in the base case condition for this series of runs. The hunting onset was approximately 55 mph, an increase of over 20 mph from the Heumann.

Instrumented wheelsets with 1:20 wheel profiles were part of the initial test design. During the test program they were only used for a limited period. The data obtained showed no hunting for a new AAR 1:20 profile.

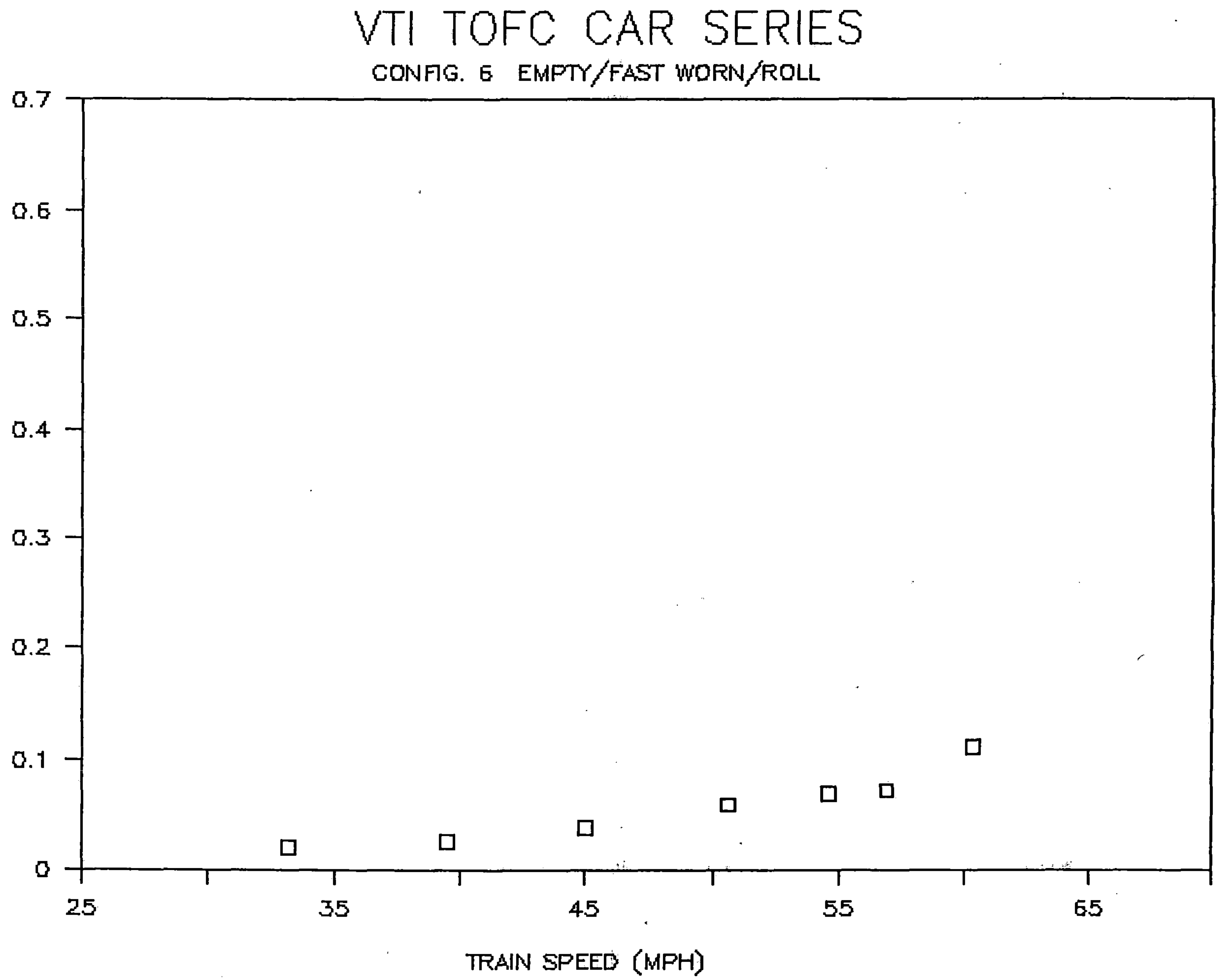


FIGURE 7-27. RMS AXLE ACCEL. VS. SPEED FOR AVERAGE FAST WORN WHEEL PROFILE ON BASE TOFC CAR.

8.0 DISCUSSION OF RESULTS - HOPPER CAR TESTS

Similar to the flat car test, the following sections cover the result of the 100-ton capacity coal hopper tests.

8.1 COAL HOPPER TEST SERIES

The base car configuration for the hopper car testing included:

- Unloaded Condition
- Double Roller Side Bearings
- S-2-C Trucks
- CN Heumann Wheel Profile
- Truck Design

The discussion of test results follows the same sequence as the flat car tests. This includes:

- Axle Acceleration
- Carbody Acceleration
  - Where Appropriate:
    - Wheel/Rail Forces
    - Correlation of Accelerations and Forces
    - Rolling Resistance and Speed Relationship

8.2 AXLE ACCELERATIONS

Four lateral accelerometers on each of the four bearing adapters were used to quantify hunting. This is the same as that used on the flat car. The time histories for the range of test speeds for the lead axle are given in Figures 8-1 through 8-5. An offset of approximately .25 g's exists in the data for this axle. This was due to a signal conditioning error in the instrumentation setup. Knowing that the acceleration varies about zero, the zero was reestablished by calculating the average and subtracting it from each discrete data point. The subsequent statistical calculations have used this technique to remove the bias from the data.

In general, the level of acceleration at the lower speeds was somewhat greater than that found for the baseline flat car test. However, sustained hunting was not achieved until nearly the same test speed as the flat car test.

The baseline axle acceleration data for the unloaded hopper is shown in Figure 8-6. The threshold is around 42 mph. Testing was terminated at 50 mph. This is similar to the speed range for the flat car testing. Figure 8-7 shows the average RMS acceleration for the four axles of the baseline flat car configuration and the baseline hopper configuration. Although neither vehicle suspension was characterized, the hunting response is generally the same.

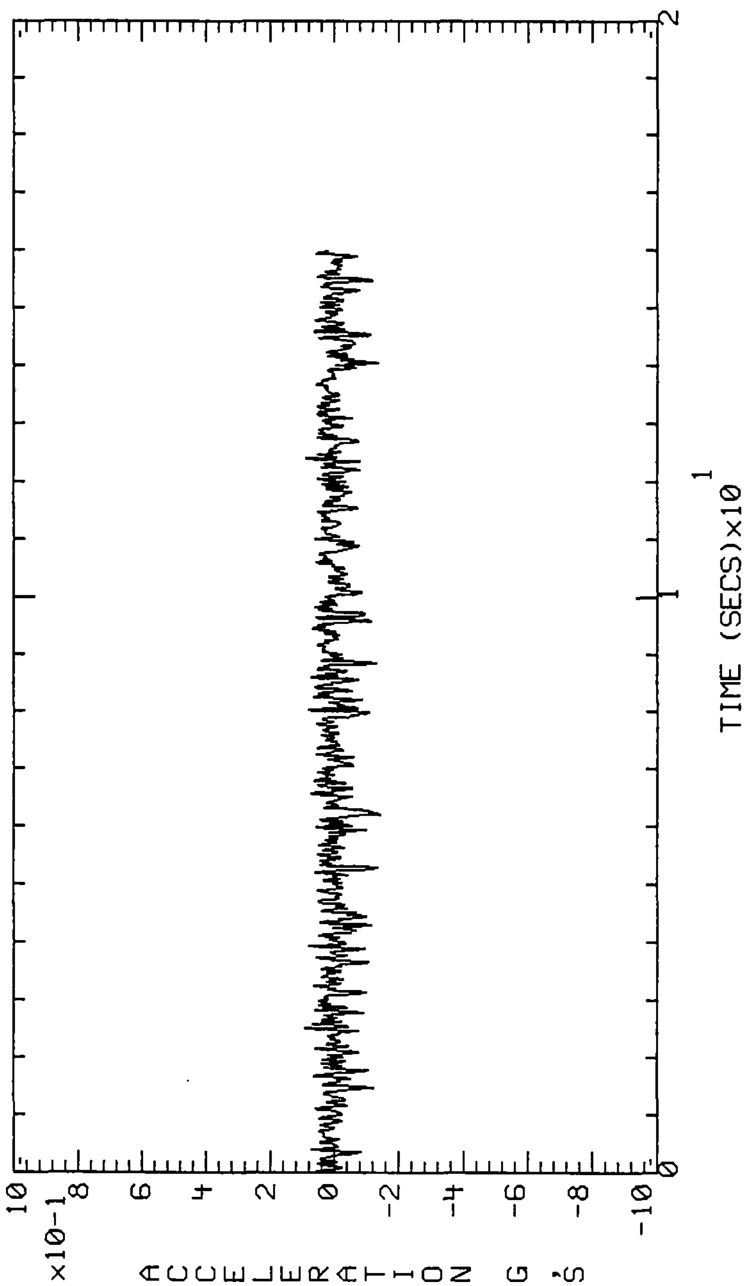


FIGURE 8-1. TIME HISTORY OF AXLE ACCELERATION FOR BASELINE COAL CAR AT 30 MPH.

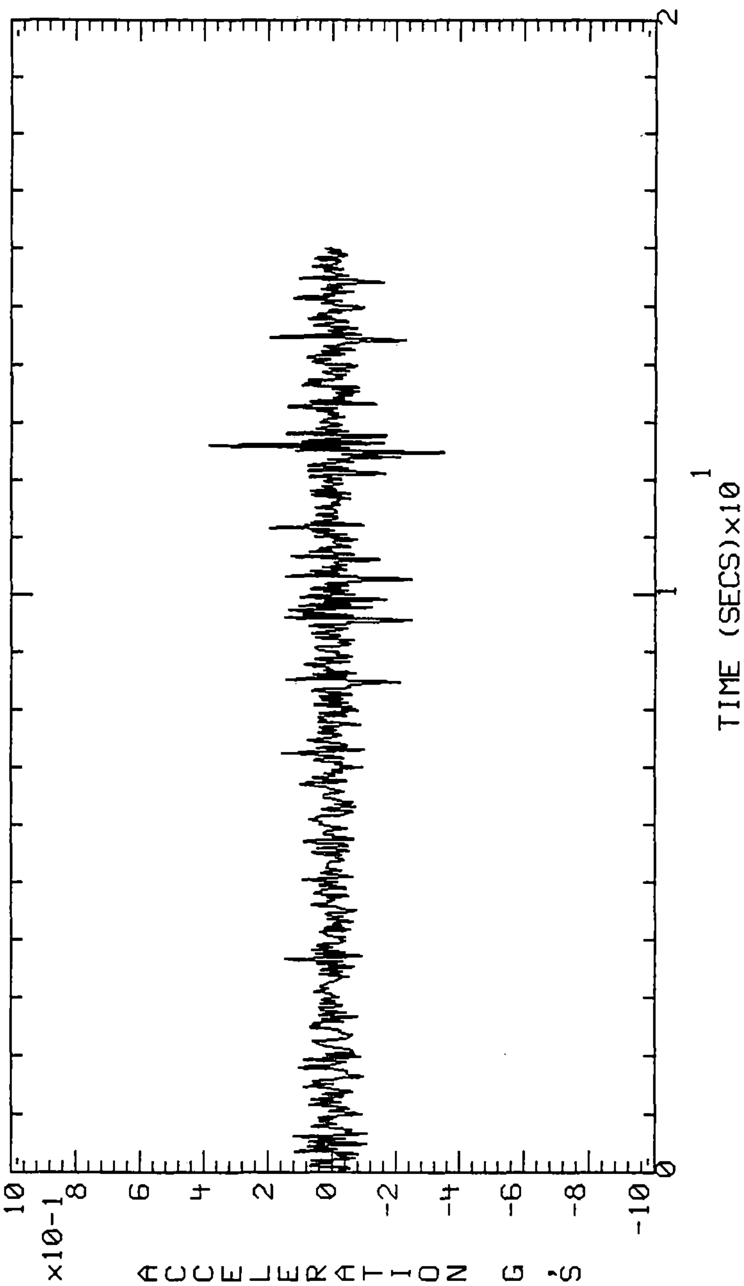


FIGURE 8-2. TIME HISTORY OF AXLE ACCELERATION FOR BASELINE COAL CAR AT 35 MPH.

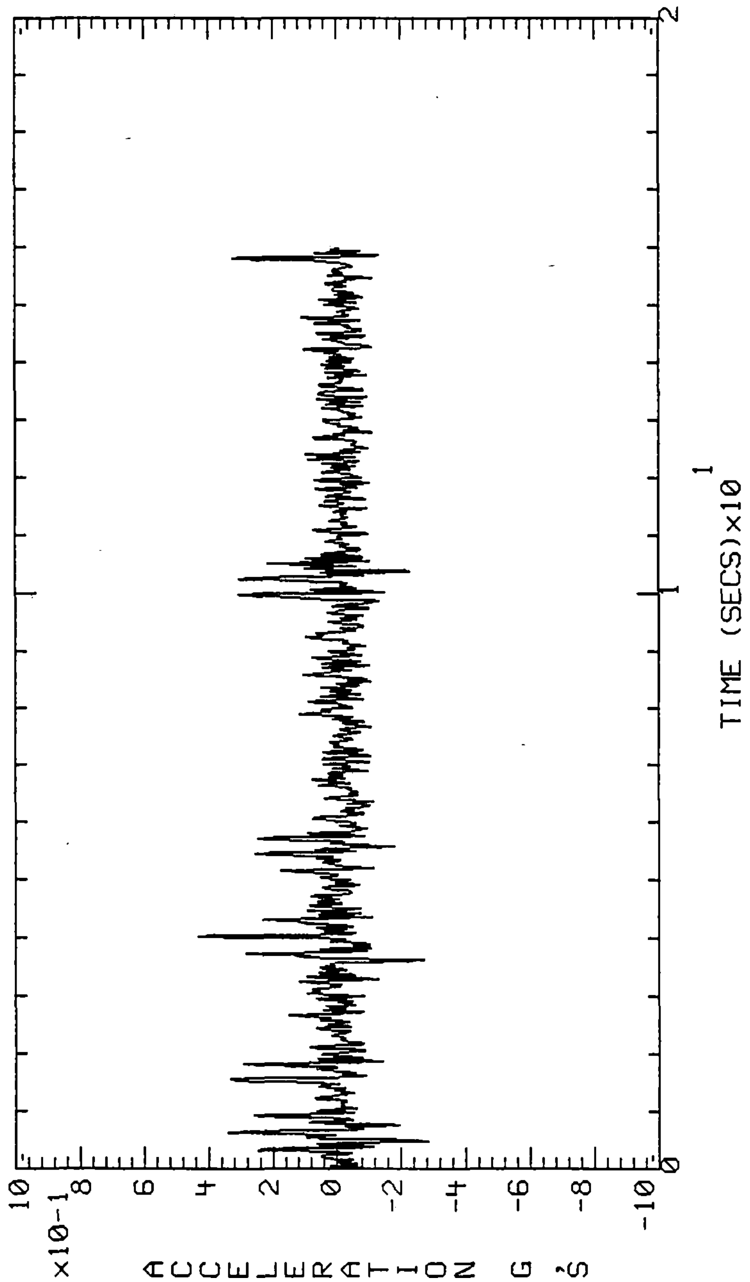


FIGURE 8-3. TIME HISTORY OF AXLE ACCELERATION FOR BASELINE COAL CAR AT 40 MPH.

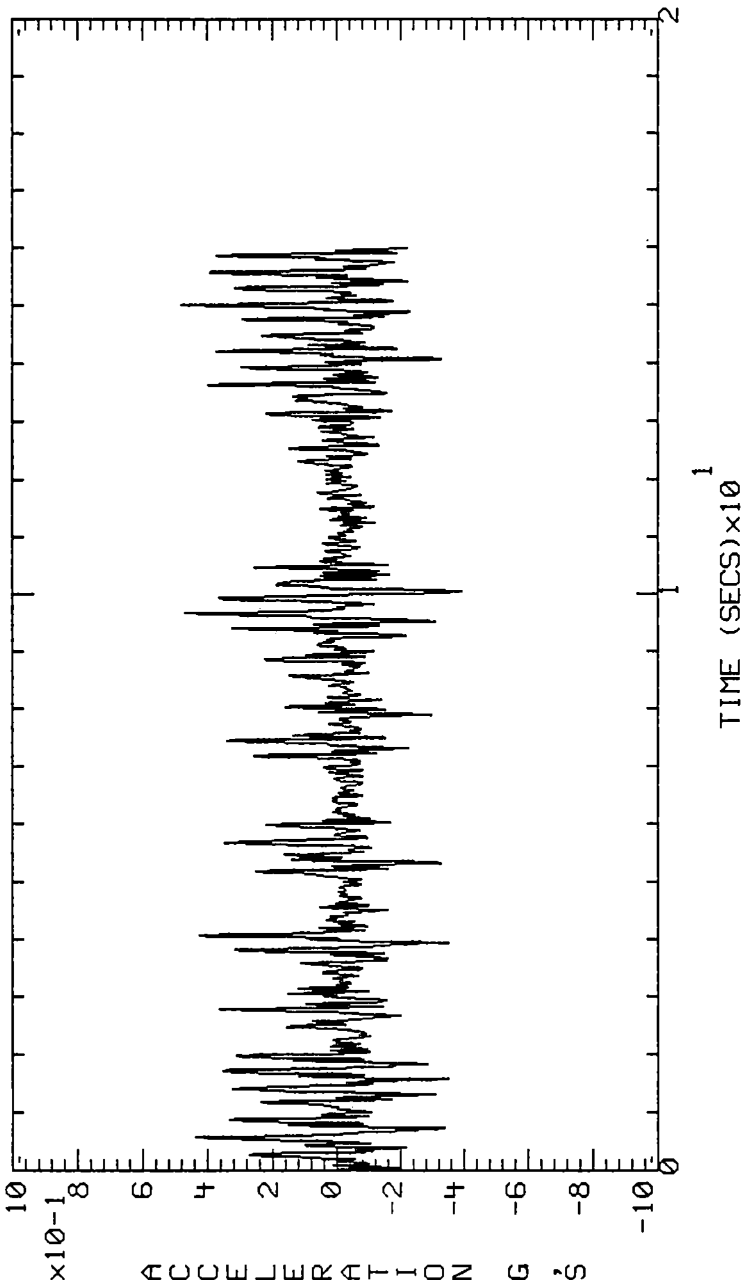


FIGURE 8-4. TIME HISTORY OF AXLE ACCELERATION FOR BASELINE COAL CAR AT 45 MPH.

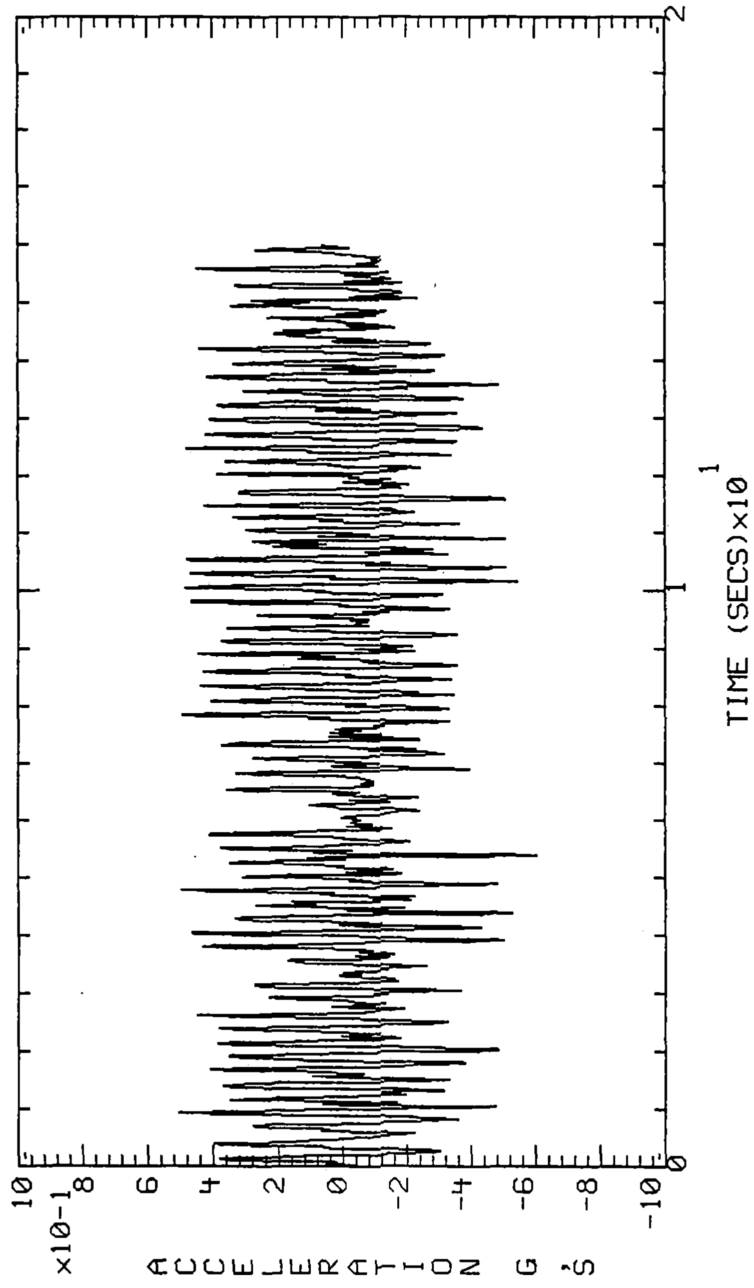


FIGURE 8-5. TIME HISTORY OF AXLE ACCELERATION FOR BASELINE COAL CAR AT 50 MPH.

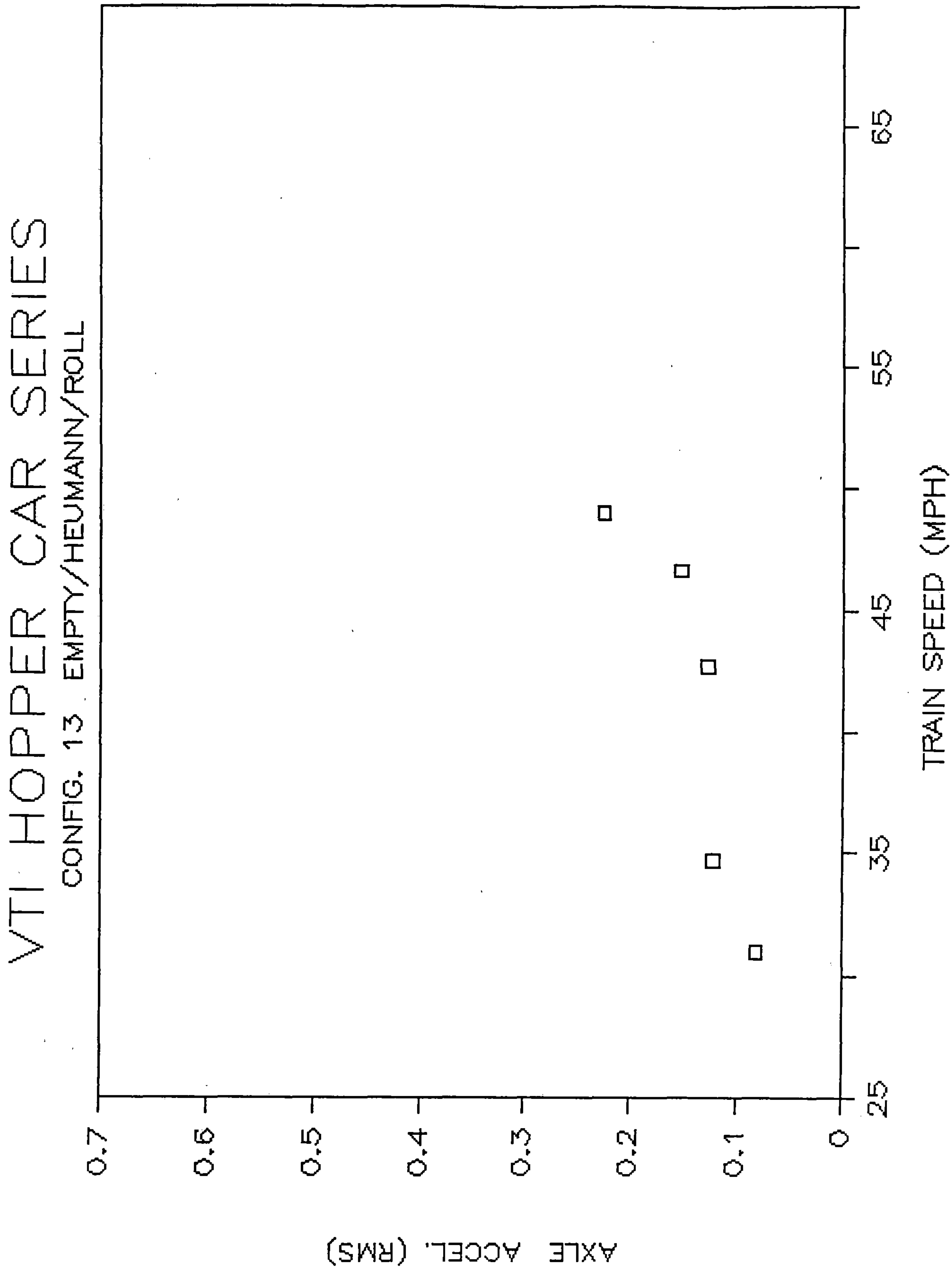


FIGURE 8-6. RMS AXLE ACCELERATION WITH SPEED BASE-LINE COAL CAR.

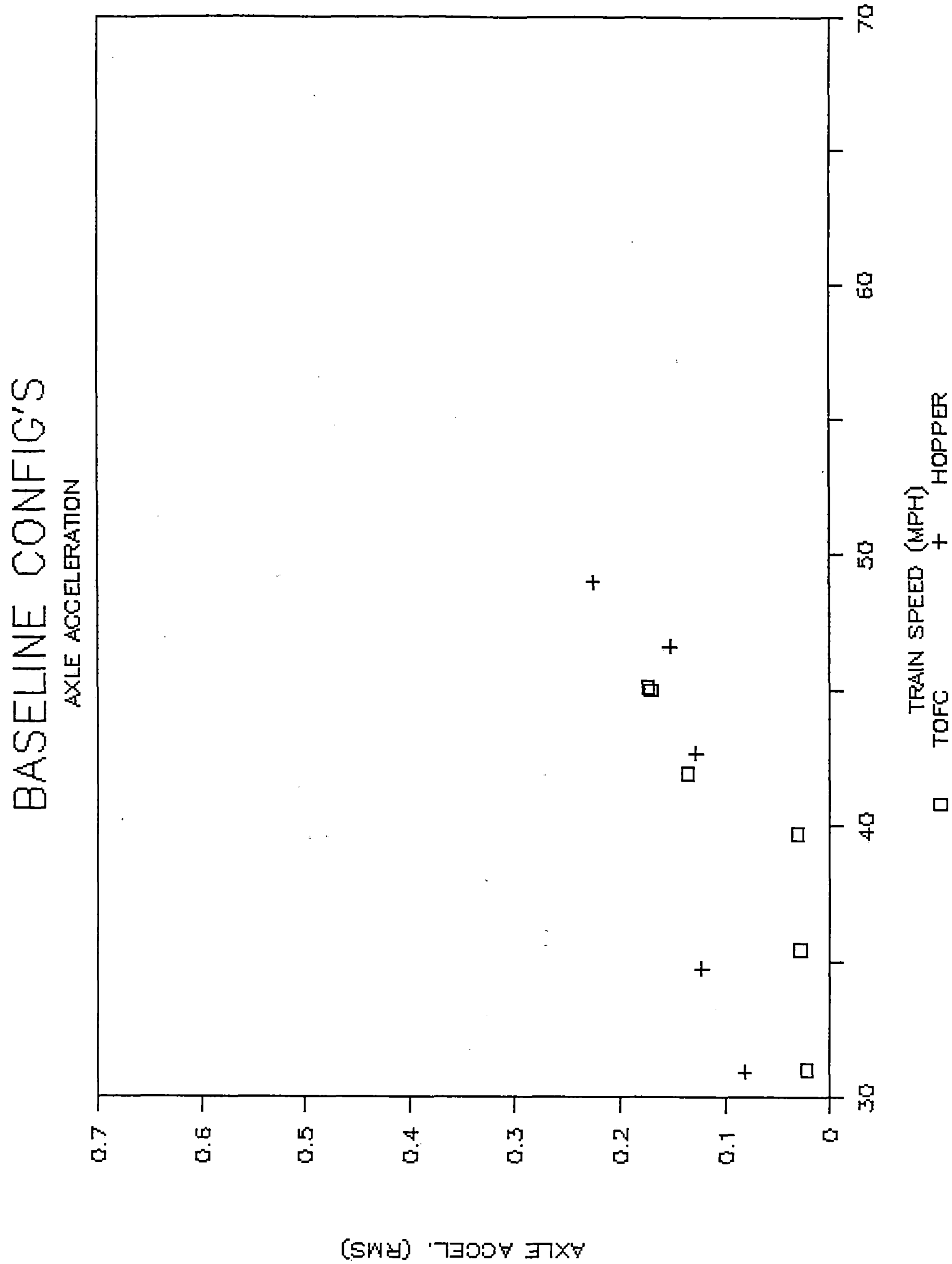


FIGURE 8-7. OVERLAY OF RMS AXLE ACCELERATION WITH SPEED FOR BASELINE COAL CAR AND BASELINE TOFC CAR.

Frequency content of the axle acceleration for the highest speed is given in the PSD, Figure 8-8. The amplitude of the PSD is twice the intermodal flat car results. The peak amplitude for the axle acceleration was approximately 0.045 for baseline flat car at the terminal test speed of 45 mph. The hopper data yields a 0.085 peak amplitude at 50 mph.

There was also a difference in the frequency at peak amplitude probably reflecting the dissimilarity in vehicle condition and design. Although similar in some respects, the 70-ton truck under the longer intermodal flat car resulted in a different frequency response in comparison to the shorter 100 ton coal car. The dominant frequency for the flat car was roughly 2.7 Hz. For the hopper the prevailing frequency was approximately 2.3 Hz.

The relationship of the axle-to-axle excitation is presented in Figure 8-9. The phase angle at the fundamental hunting frequency is essentially zero which relates to an in-phase situation. Due to this relationship in axle motion, the truck can be considered a single body in terms of lateral acceleration.

8.3 AXLE ACCELERATION AS INFLUENCED BY AXLE LOAD

Using the baseline hopper as the standard or control case, comparisons of axle acceleration were made for loaded cars. The same equipment was used and tested with an additional 33,000 pounds and 66,000 pounds of lading. The objective was to see how axle load would affect the hunting onset speed.

Figure 8-10 is a plot of RMS acceleration averaged for the four axles at the three weight levels. The one-third loaded case was tested up to a maximum speed of 65 mph. The peak-to-peak criteria of 1.5 g's was exceeded at this speed. The two-thirds loaded car ran up to the maximum test speed of 70 mph. At this point the car was still running stably. Based on the increase in hunting threshold speed for the empty to one-third loaded configuration, it is somewhat probable that the onset for the two-third load was approximately 80 mph. The net change in threshold for the 33-ton difference was approximately 15 mph. This represents the change from 47 to 62 miles mph. The threshold for the two-thirds load was never measured since it was beyond the 70 mile per hour maximum test speed.

The purpose of collecting this data was to experimentally determine the rate of change in terms of hunting performance with respect to axle load which will ultimately be used in stability model verification.

The last section of this report presents measured and analytically derived hunting characteristics. This will compare the influence of a variety of tested variables relative to the theoretical results.

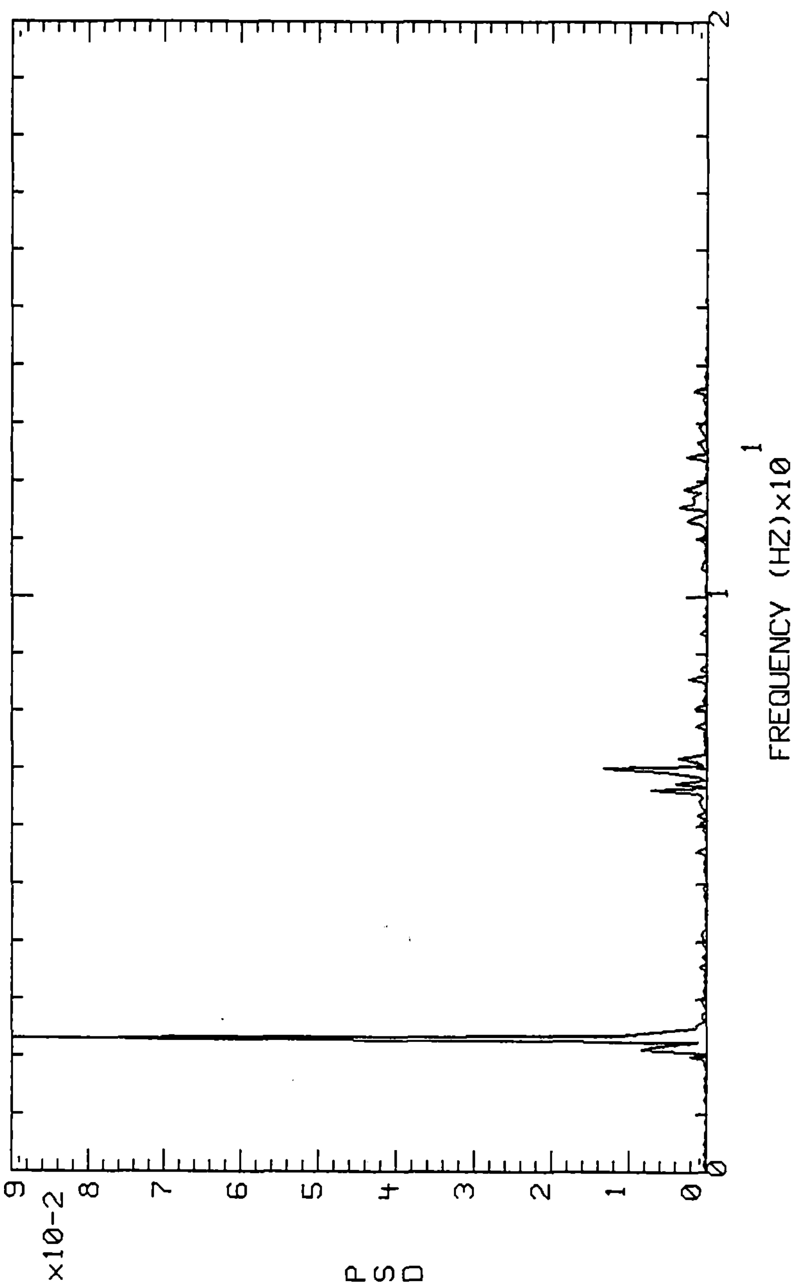


FIGURE 8-8. PSD OF AXLE ACCELERATION FOR THE MAXIMUM TEST SPEED OF THE BASELINE COAL CAR.

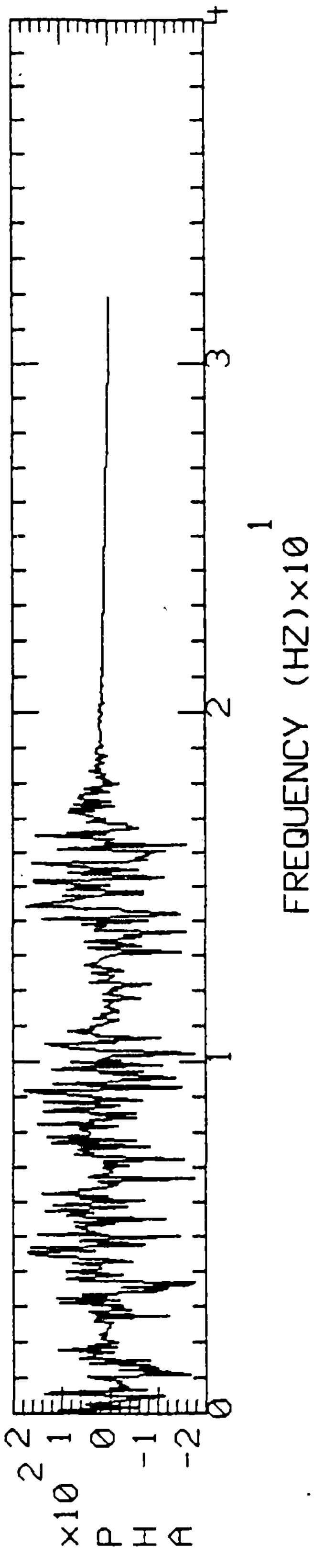


FIGURE 8-9. CROSS POWER SPECTRUM FOR AXLE-TO-AXLE ACCELERATION.

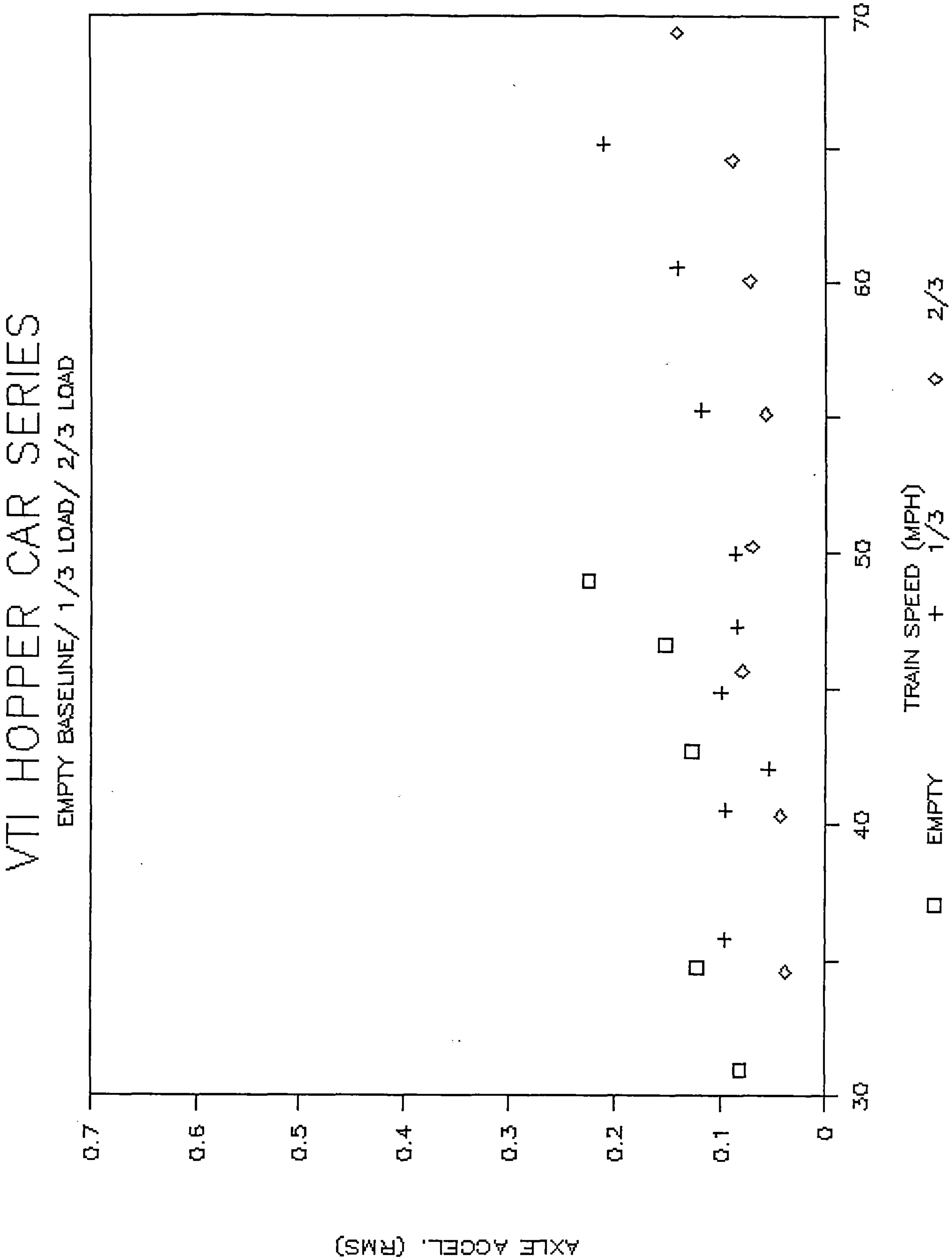


FIGURE 8-10. RMS AXLE ACCELERATION AVERAGE OF THREE WEIGHT CLASSES.

8.4 CARBODY ACCELERATION

Figure 8-11 contains the data of average RMS carbody acceleration for the A-end, center, and B-end of the car with speed. As was the case with the flat car, the magnitude of the RMS response is approximately 50 percent higher than the axle acceleration at a given vehicle speed where hunting is present. Also, the magnitude of carbody accelerations are quite similar to that for the flat car series. Shown in Figure 8-12 is the average carbody acceleration RMS with speed for the two baseline vehicles.

Movement of the A-end of the car, to the truck on that end of the test car, was not clearly in phase. The plot of the phase relationship in Figure 8-13 shows wide range in phase angle between the two bodies within the band of truck hunting frequency. Investigating the carbody acceleration revealed significantly different frequency of excitation. The PSD in Figure 8-14 indicates a major response in the 2.0 to 2.2 Hz range, equal to the truck for the same car end.

8.5 SIDE BEARING EFFECT ON HUNTING

As observed with intermodal flat car, the change in side bearing configuration also affected the hunting response. Given in Figure 8-15 is the RMS axle acceleration for the baseline car outfitted with roller bearings and constant contact side bearings. It provides for continuous contact and is designed to increase the pivot torque about the truck center pin.

The observed onset for the constant contact side bearing configuration was in the 65 to 67 mph range. This increase of about 18 mph is similar to the change observed on other hunting tests with like equipment, as well as the change noted for the intermodal flat car in Section 7.0.

8.6 HUNTING RESPONSE TO WHEEL PROFILE CHANGE

The dynamics of hunting are initiated at the wheel/rail interface. The resulting effective conicity discussed earlier in the report has a critical effect on the hunting behavior of a vehicle.

The coal hopper used in test was equipped with CN Heumann, FAST worn intermediate conicity, and narrow flange CN Heumann wheels. This provided a wide range of conicity and allowed for a parametric study of profile effect on hopper stability.

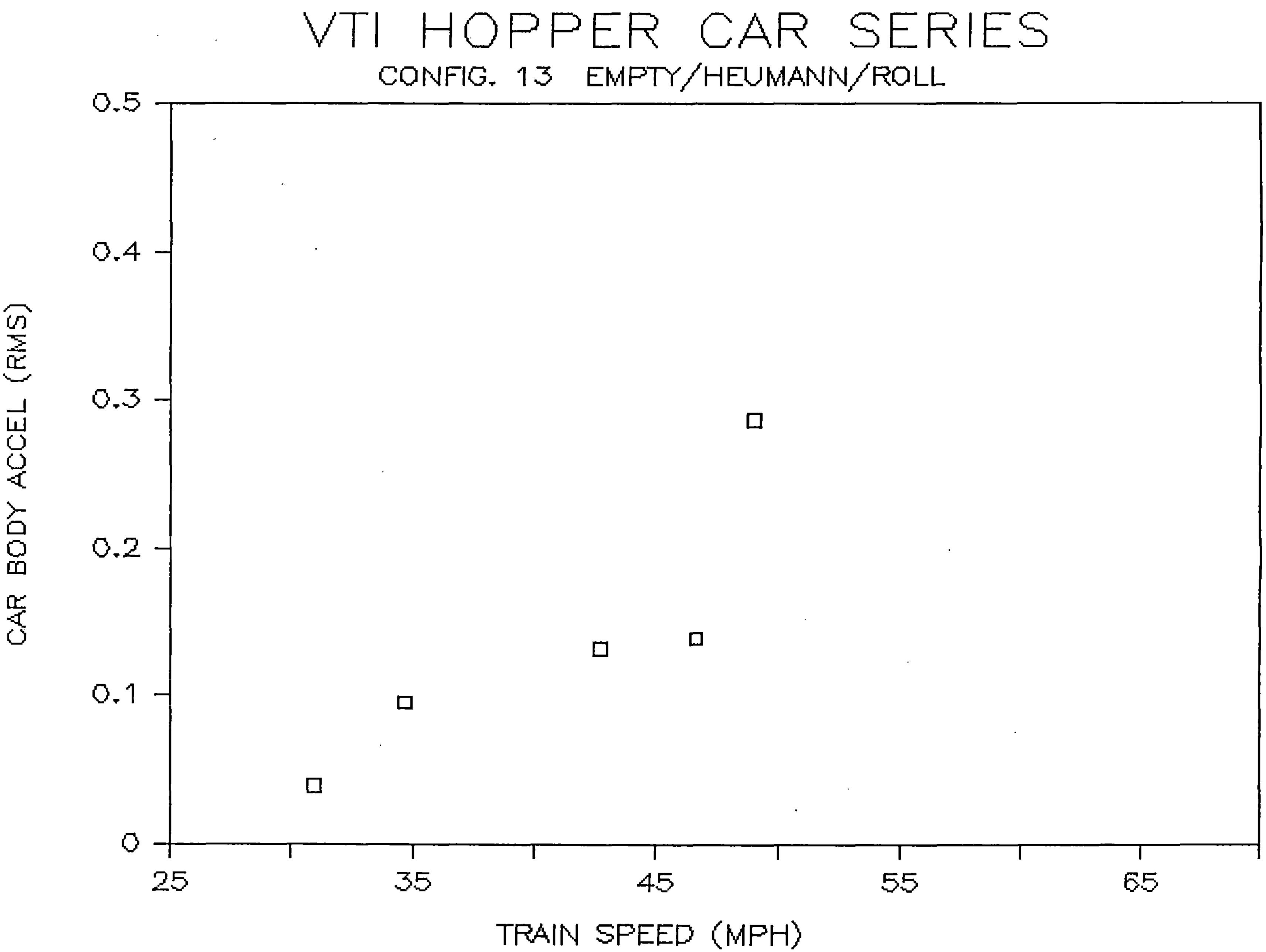


FIGURE 8-11. RMS CARBODY ACCELERATION WITH SPEED.

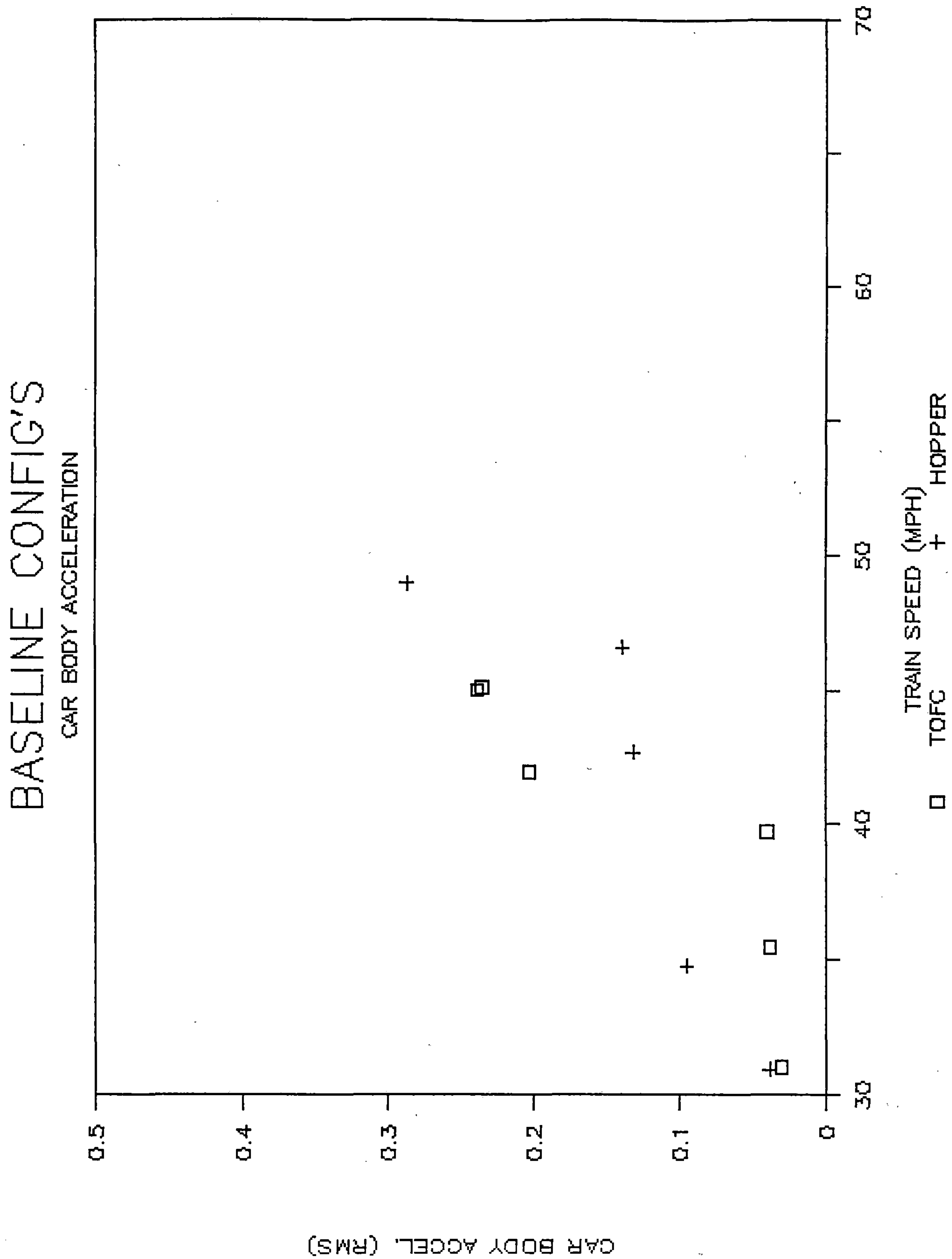


FIGURE 8-12. OVERLAY OF CARBODY ACCELERATION FOR THE BASELINE TOFC CAR AND BASELINE COAL CAR.

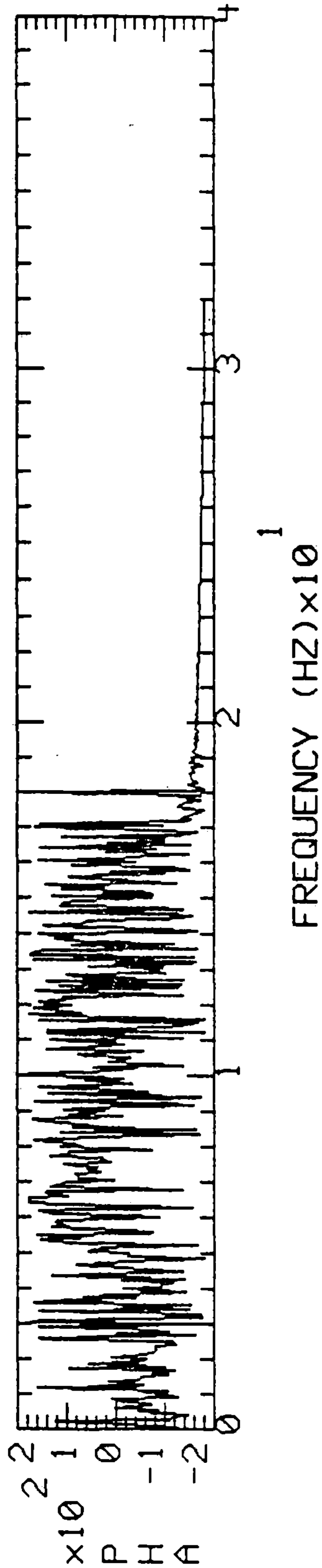


FIGURE 8-13. CROSS POWER SPECTRUM OF CARBODY-TO-TRUCK ACCELERATION.

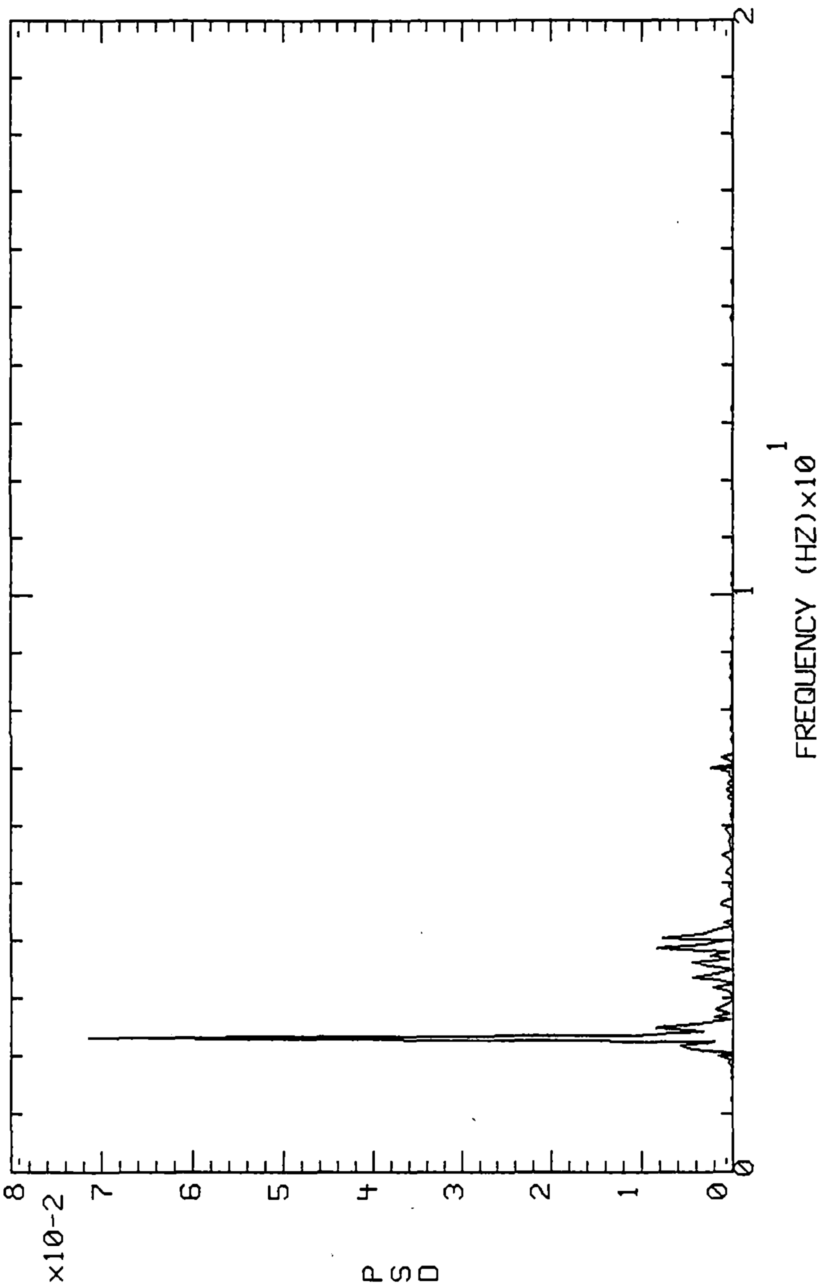


FIGURE 8-14. PSD OF CARBODY ACCELERATION.

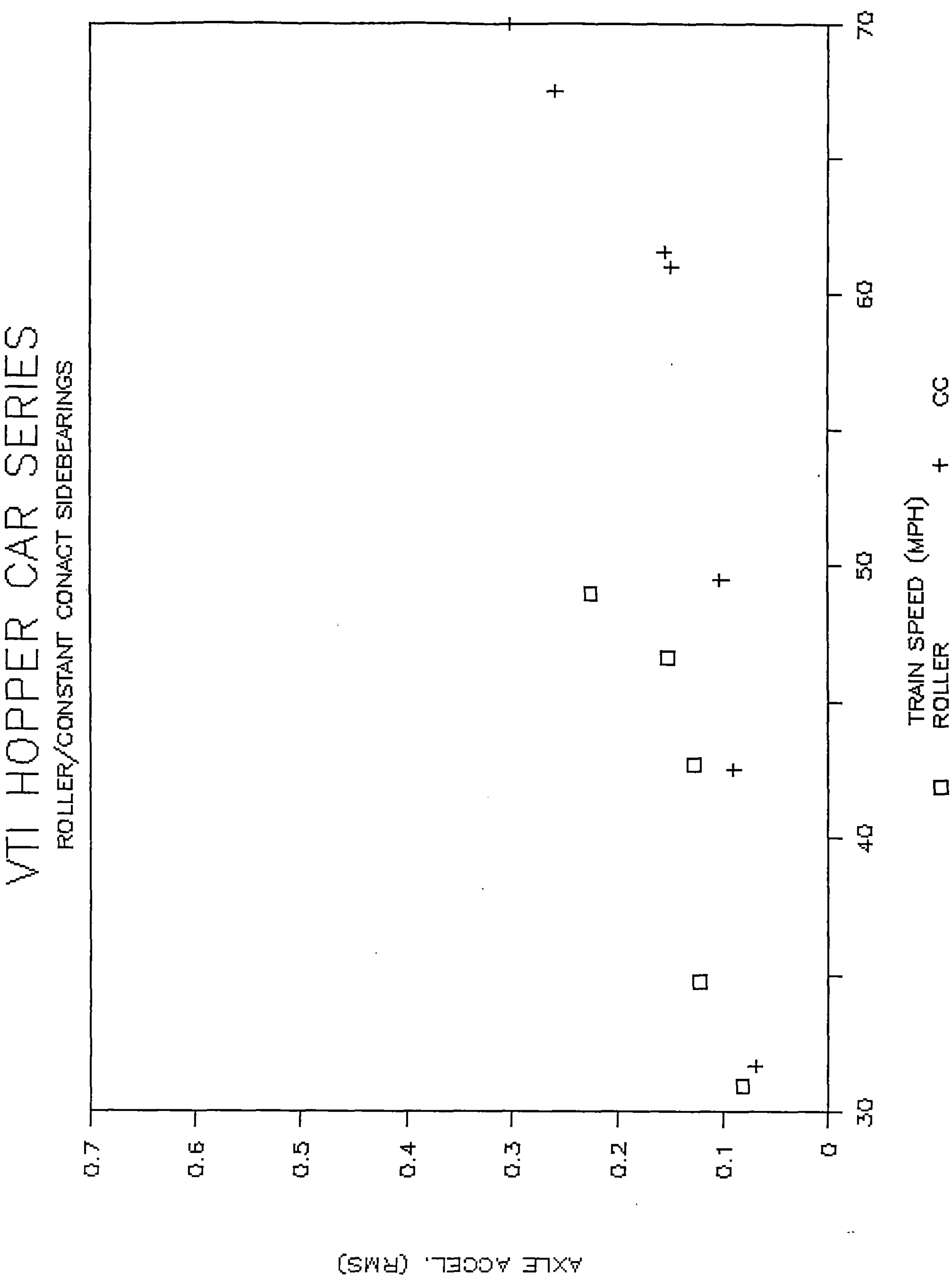


FIGURE 8-15. COMPARISON OF SIDE BEARING CONFIGURATIONS - ROLLER/CC.

Figure 8-16 is an overlay of the results for all three wheel profiles. The 0.15 conicity or FAST profile wheel never provided axle acceleration in the hunting frequency range. The increase seen up to the 70 mph speed is at higher frequency, it is predominately driven by the rolling of the wheelset. The narrow flange Heumann, however, responded similarly to the standard or wide flange Heumann. This similarity is in terms of the range of allowable operating test speed. The overall characteristics are dissimilar in other respects though. The average RMS level at the low end of the speed spectrum from 30 to 45 mph is higher for the standard Heumann. It appears there is a regime of oscillatory motion from 35 to 45 mph which is less than flange-to-flange hunting. For the same speed interval the narrow flange version is running stably.

The threshold is distinct and is approximately 46 mph. The threshold for the wide flange Heumann estimated at 47 mph, is basically the same. The conclusion drawn from this data is that the flangeway clearance increase provided by varying the wheel profile is negligible.

8.7 TRUCK DESIGN INFLUENCE

The change in a freight trucks characteristics is known to have a strong effect on the hunting response of the vehicle. A significantly different truck was used to demonstrate this influence. A version of the Dresser DR-1 retrofit truck was tested under the 100-ton capacity carbody used in the other parts of the hopper car study. By effectively crossbracing the truck via the interconnecting steering arms shown in Figure 8-17, the lozenging of the sideframes is greatly inhibited. With this in mind, the test was run to verify the expectation that the truck would provide a higher hunting speed.

The plot of axle acceleration in Figure 8-18 reveals no indication of continuous hunting up to the threshold of about 67 mph. At this point, there is a fairly slight increase in the slope of RMS acceleration and speed. This results in a 20 mph change in onset for the DR-1 truck in comparison to the conventional truck.

8.8 L/V RATIO

Given in Figure 8-19 is the time history of the L/V ratio for the baseline hopper. The overall response is quite similar to that of the baseline intermodal flat car. Since this is a result of the same CN Heumann surface geometry, the derailment criteria is the same 0.83 L/V. Here the maximum L/V of .73 is approaching the critical value for the single peak value.

Given additional car speed the critical value would likely be exceeded. The remaining question is the duration of the L/V event necessary for flange climb. Figure 8-20 is a plot of peak L/V against speed for this vehicle configuration.

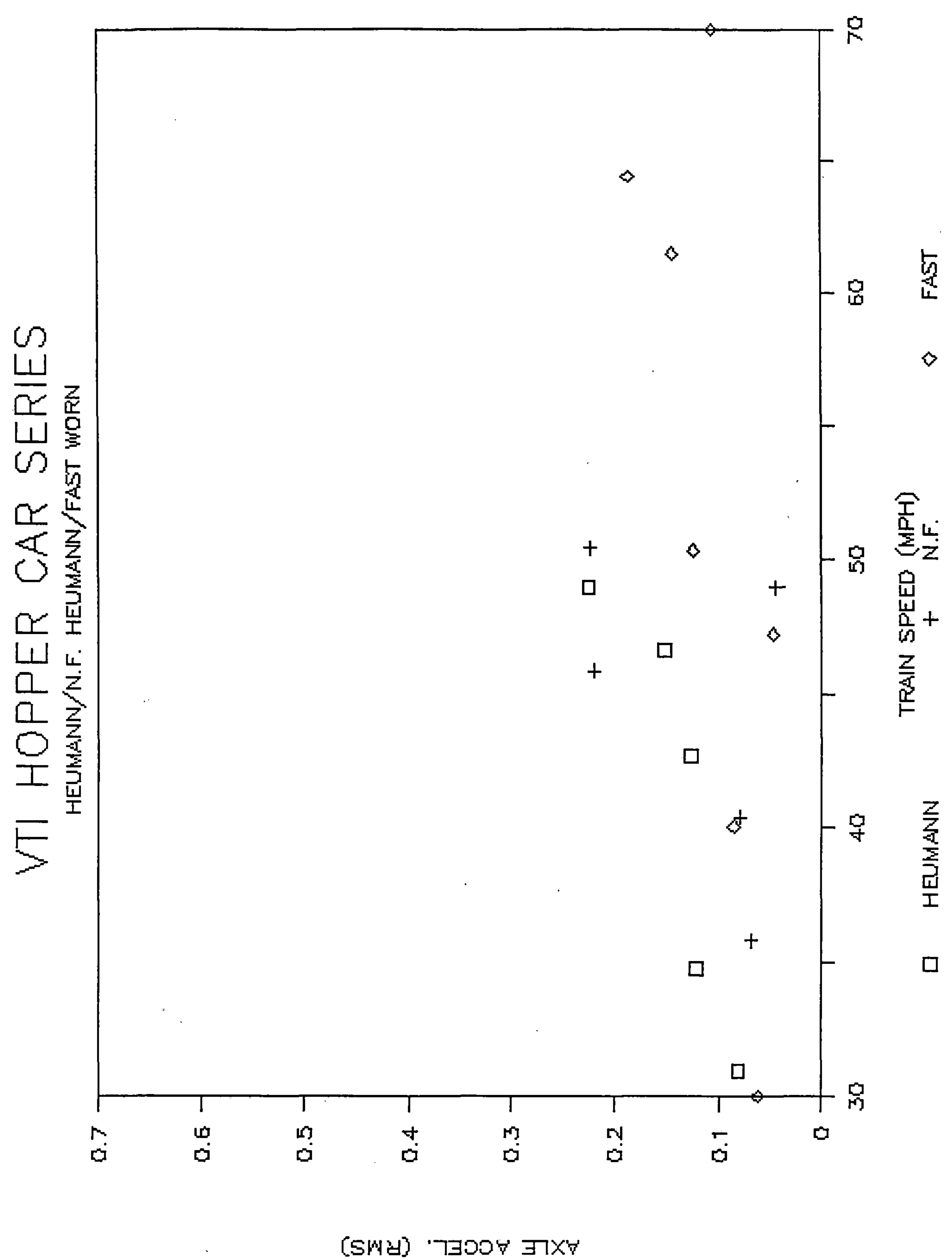


FIGURE 8-16. OVERLAY OF THE HEUMANN, N.F. HEUMANN, AND FAST PROFILE AXLE ACCELERATION.

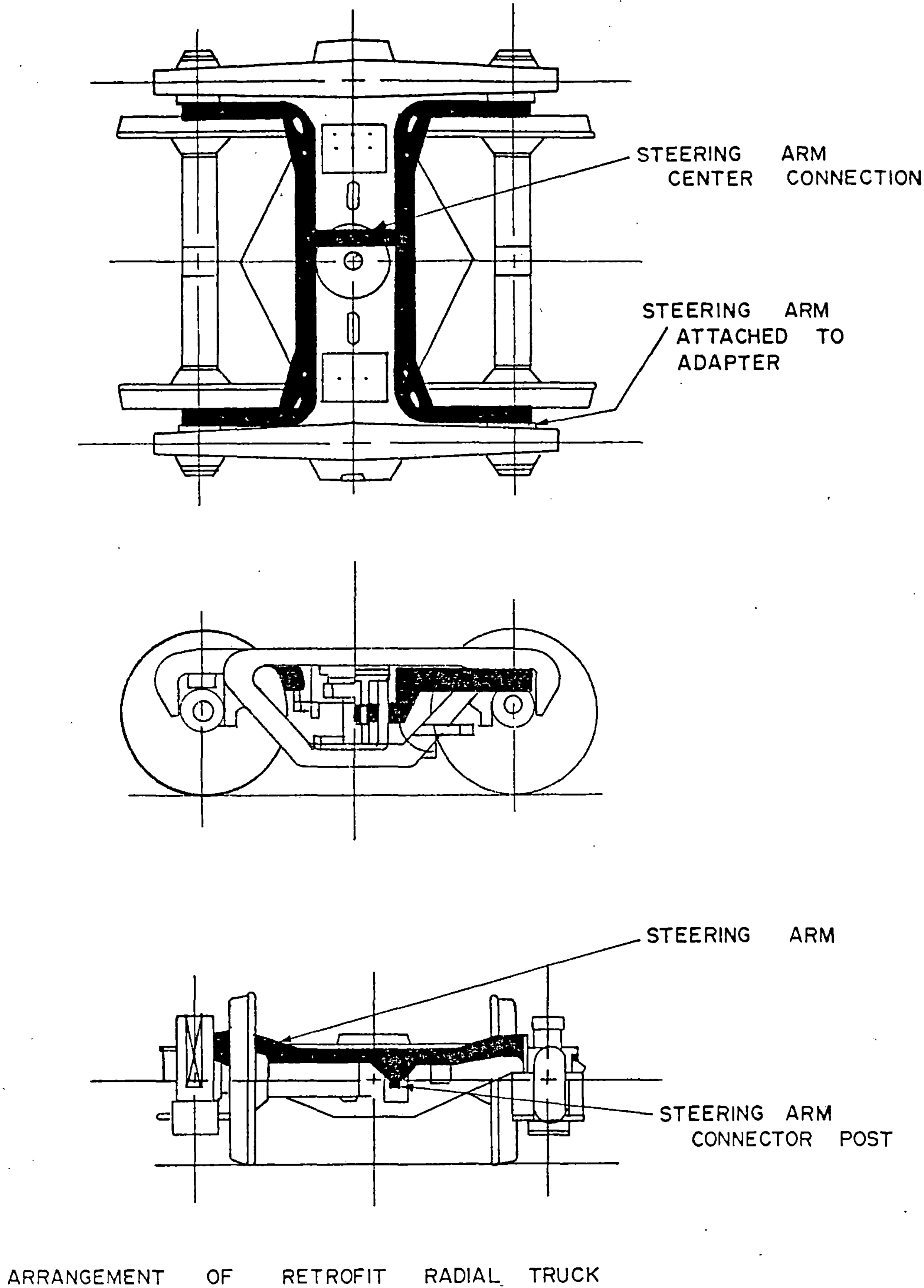


FIGURE 8-17. DR-1 STEERING ARMS.

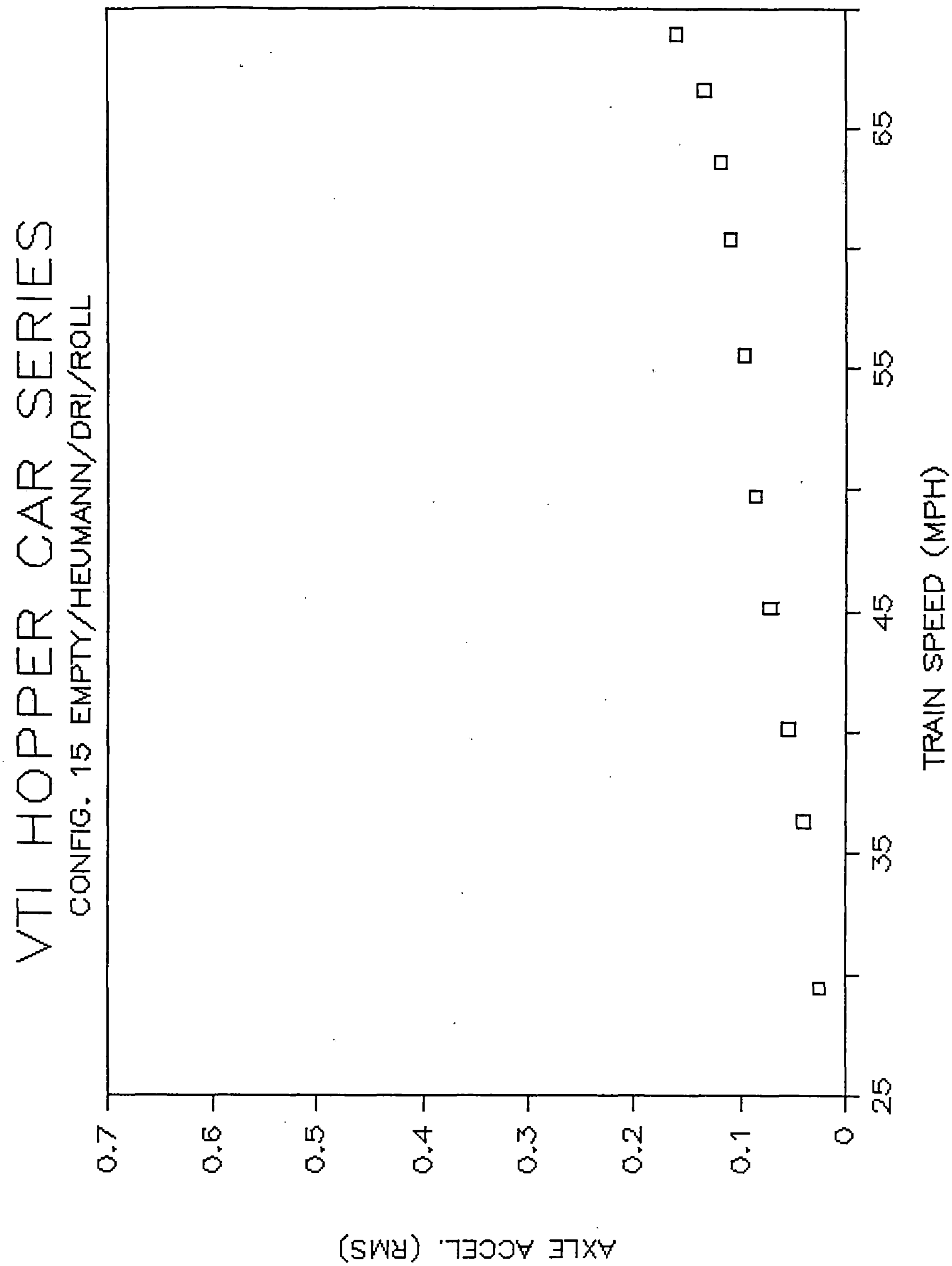


FIGURE 8-18. RMS AXLE ACCELERATION OF DR-1 BASELINE HOPPER.

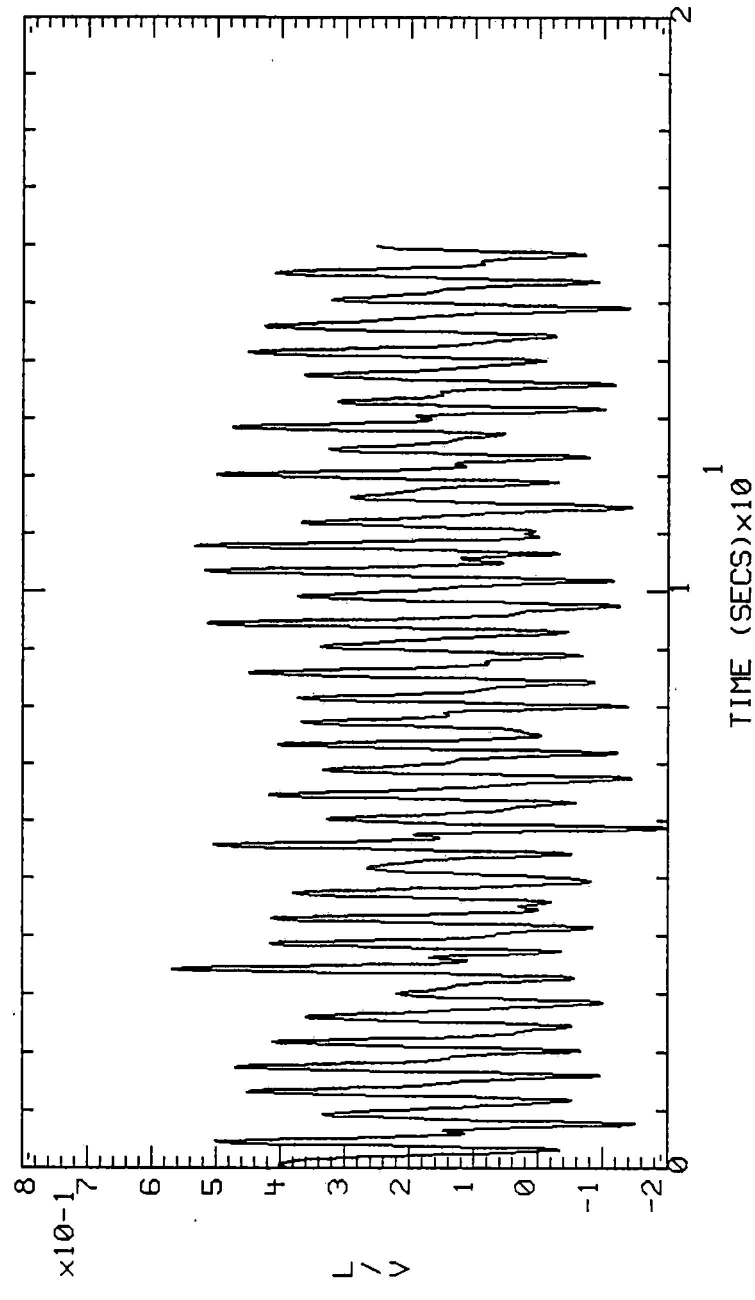


FIGURE 8-19. TIME HISTORY OF THE L/V RATIO - BASELINE HOPPER.

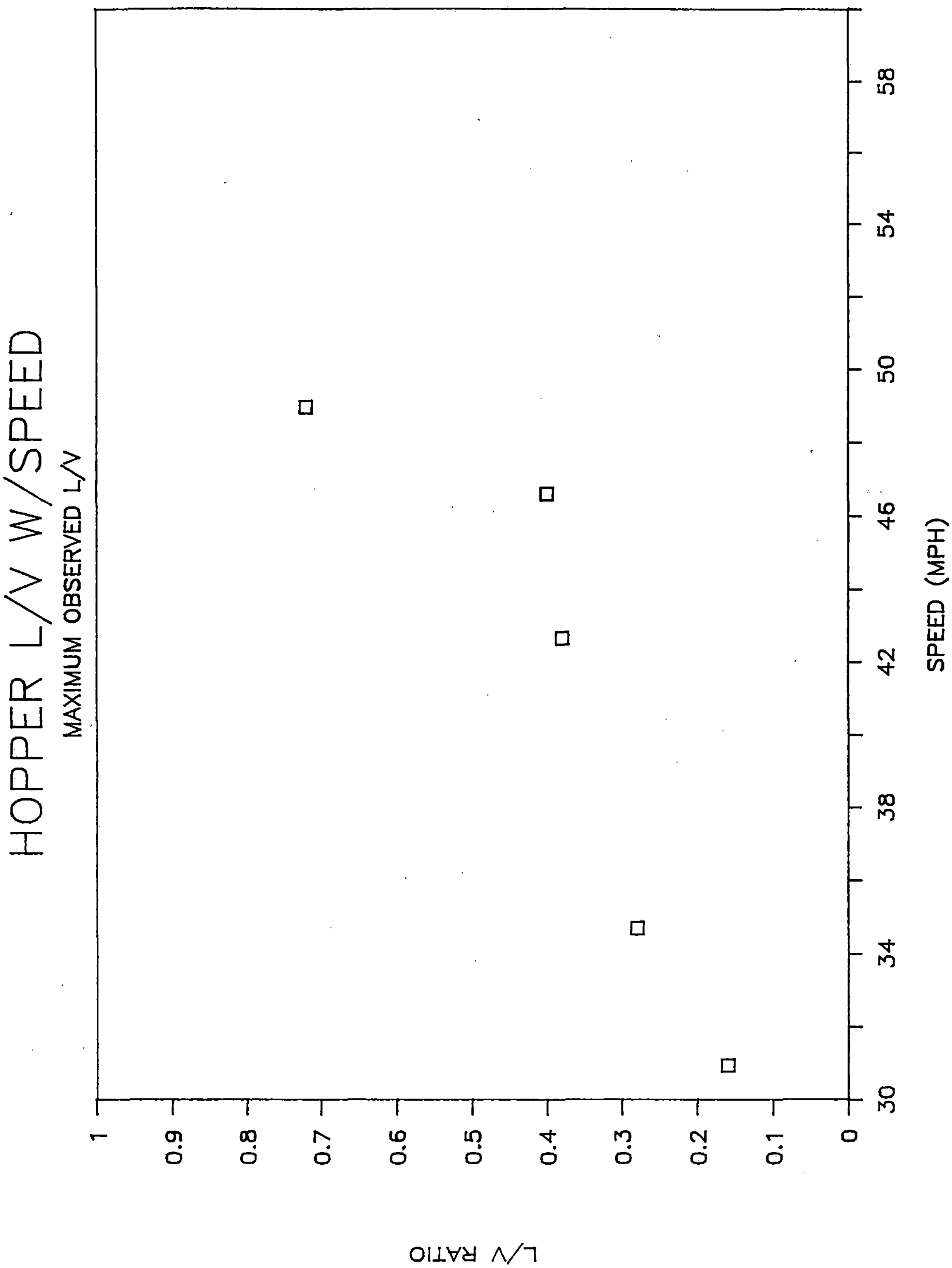


FIGURE 8-20. PEAK L/V VS. SPEED.

8.9 CORRELATION OF AXLE ACCELERATION AND WHEEL/RAIL FORCES

A study of the wheel/rail force environment as it relates to axle acceleration is covered in this discussion. The maximum flange-to-flange hunting speed of 50 mph for the base hopper car configuration is covered. The time history of wheel forces measured by 36-inch instrumented wheelsets is given in Figure 8-21. The positive offset is attributed to the lateral component resulting from the weight of the car on an effectively tapered wheel profile. Examination of time history overlays between force and acceleration revealed good agreement in response. The point-for-point correlation of acceleration and lateral force is shown in Figure 8-22. The somewhat scattered pattern indicates a weaker force-acceleration relationship than that with the intermodal flat car testing. In the hopper car case, the trend is apparent between the two measured channels, but the force response to the acceleration is not truly linear and has a slope term which is far less than the car mass should provide as experienced from the baseline intermodal flat car analysis. In this case, the constant term was 916 pounds and the slope was 5,021 pounds per 1.0 g's of acceleration.

An examination of the shape characteristics reveals a non-linear response for negative acceleration. This has the net effect of reducing the slope term. In general, this indicates that a simple acceleration/force correlation does not always exist.

8.10 ROLLING RESISTANCE CHANGES WITH A HUNTING VEHICLE

As described earlier in the report, a load cell coupler developed by the AAR, at the TTC, was introduced into the test consist to measure rolling resistance. The intention was to measure rolling resistance on a non-interfering basis, such that the scheduling and cost of the VTI test was not impacted. The test runs were used to initially see if the resolution of the load cell system was sufficient to detect rolling drag for an unloaded car at different speeds. In the process of the shakedown, additional considerations in using the system were discovered. All costs associated with the analysis of the coupler data, as well as the installation of associated hardware was funded by the AAR Energy Research Program. The resistance data presented in this report illustrates the potential value in understanding the additional drag hunting cars may generate. This will be useful in stability studies when energy consumption is a factor.

Traditional approaches to train resistance such as that addressed in the original Davis equation and later versions of it include no terms to account for unstable vehicle behavior. The method utilized in this study was to generally estimate the difference in resistance between the baseline car and other unloaded test configurations.

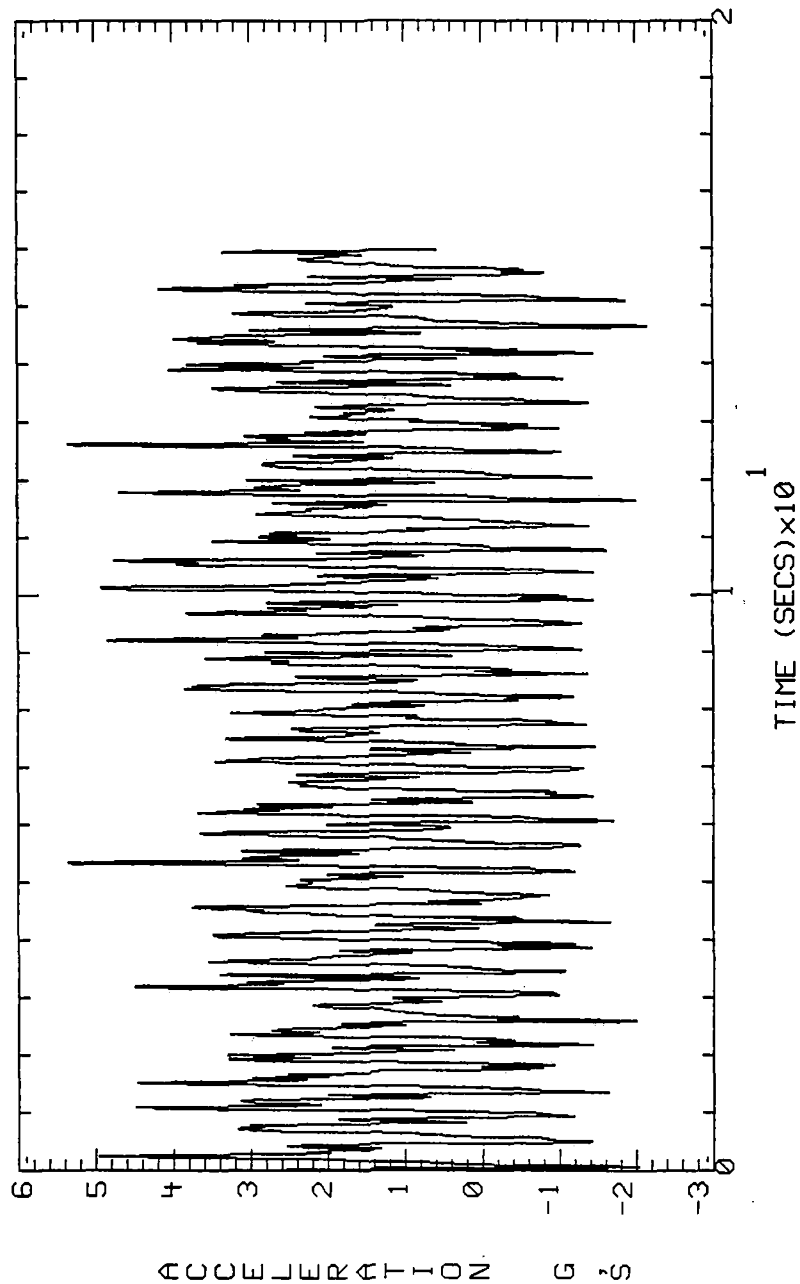


FIGURE 8-21. TIME HISTORY OF LATERAL WHEEL FORCE FOR HUNTING COAL CAR.

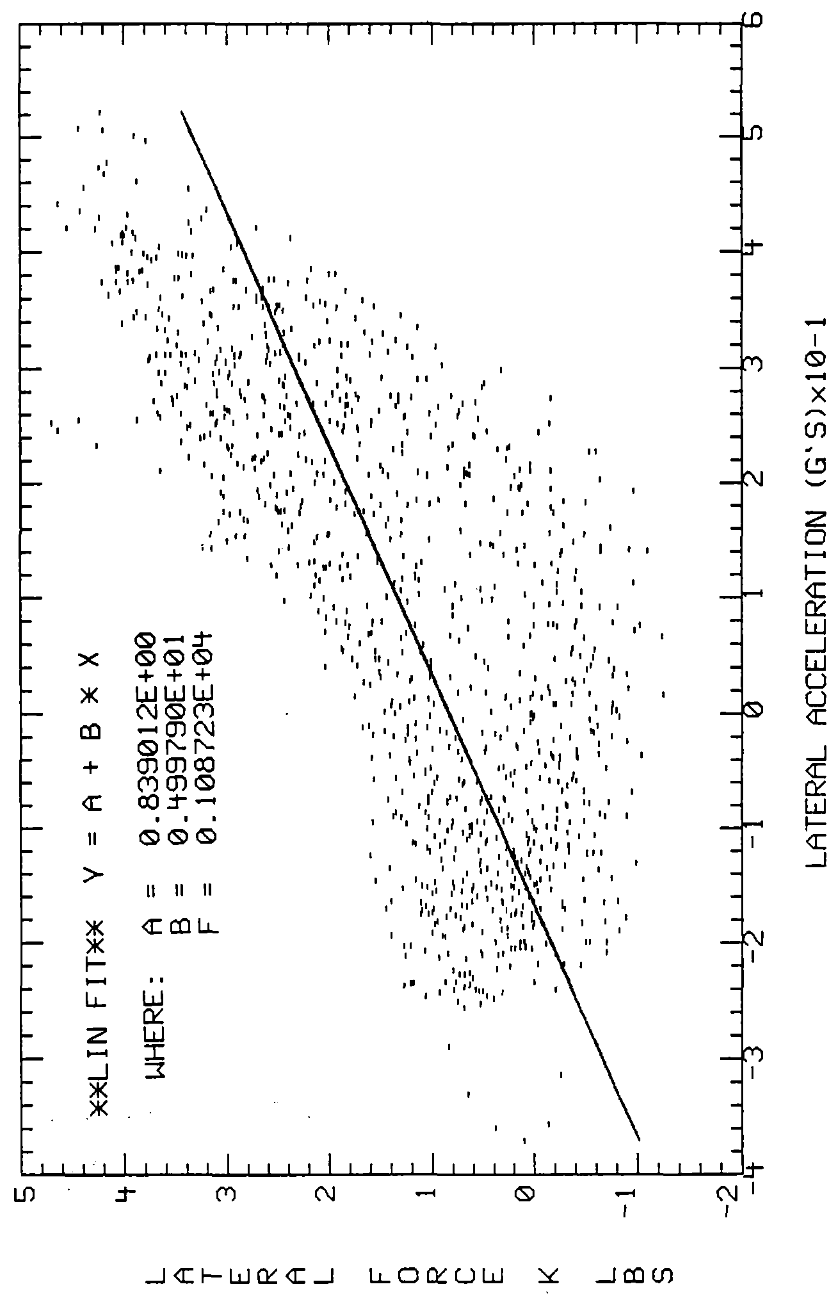


FIGURE 8-22. X,Y PLOT OF DISCRETE LATERAL FORCE AND ACCELERATION ON COAL CAR.

Figure 8-23 presents the incremental change in total resistance for the baseline car over the range of test speeds. As previously illustrated, this configuration started hunting at a speed of approximately 47 mph. At this point, an incremental increase in rolling resistance results due to the energy dissipated in moving the axles and carbody laterally.

Shown in Figure 8-24 is a comparative plot of five test configurations. Assuming that the aerodynamic drag in all cases is equal, the resistance difference is a result of the energy dissipated at the wheel/rail interface. This would include steady state behavior and dynamic instability associated with hunting.

The five configurations include the baseline car, the baseline car with constant contact side bearings, the DR-1 truck equipped car, the baseline car with narrow-flange Heumann wheels, and the baseline car with the FAST worn wheel profile. In order to compare the predicted resistance from the second order curve fit at 50 mph, Table 8-1 has been produced. Recognizing that the wind conditions were not continuously recorded during test and the test runs could not be postponed due to undesirable wind conditions, there is most likely some additional aerodynamic influence on the data. Similar wind conditions were observed for all five cases. The differences listed in the preceding table may be indicative of the penalty in energy consumed as a result of hunting freight vehicles. Far more controlled testing is needed if accurate resistance equations are to be developed.

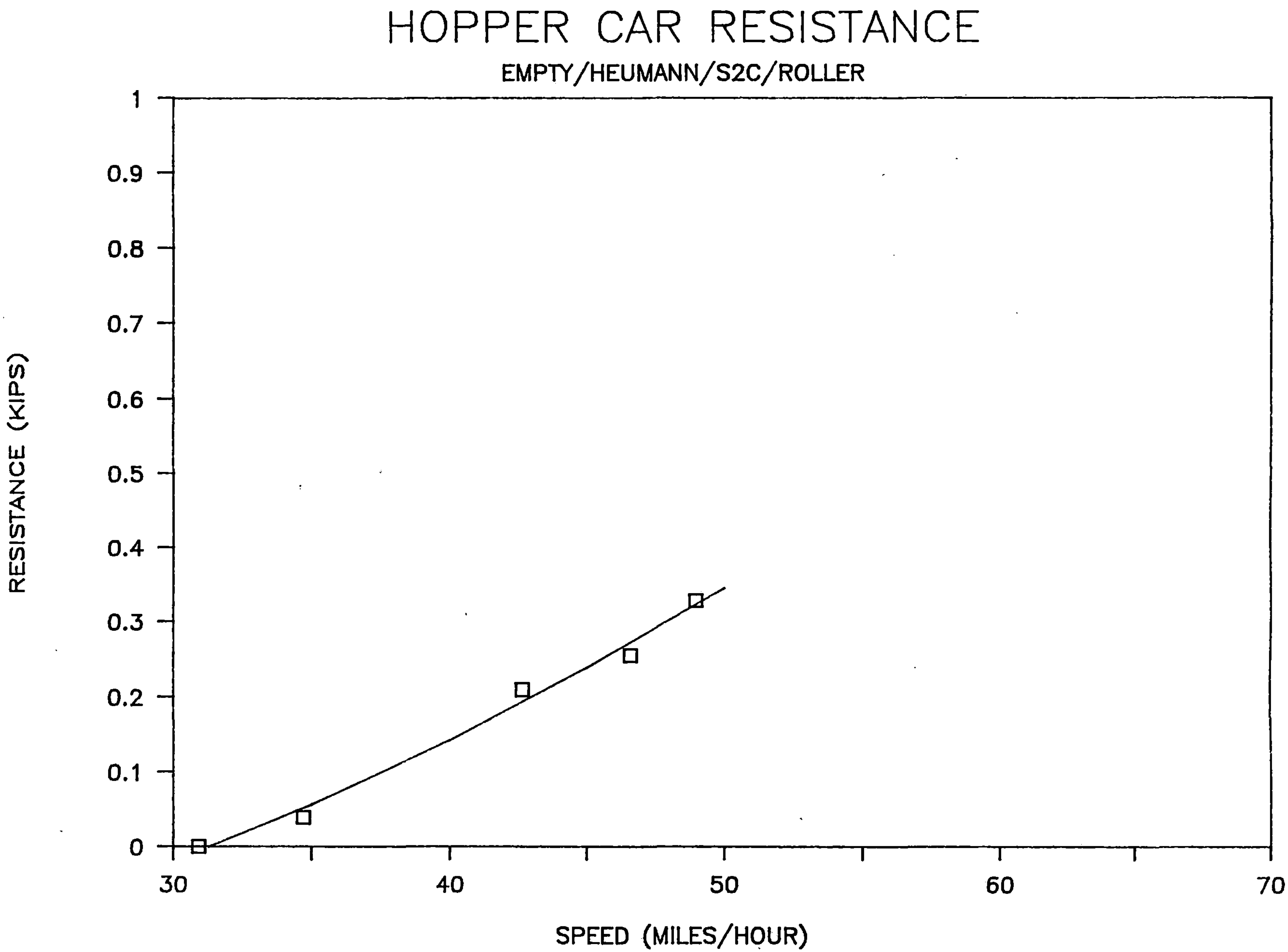


FIGURE 8-23. HOPPER CAR RESISTANCE/SPEED CHANGE.

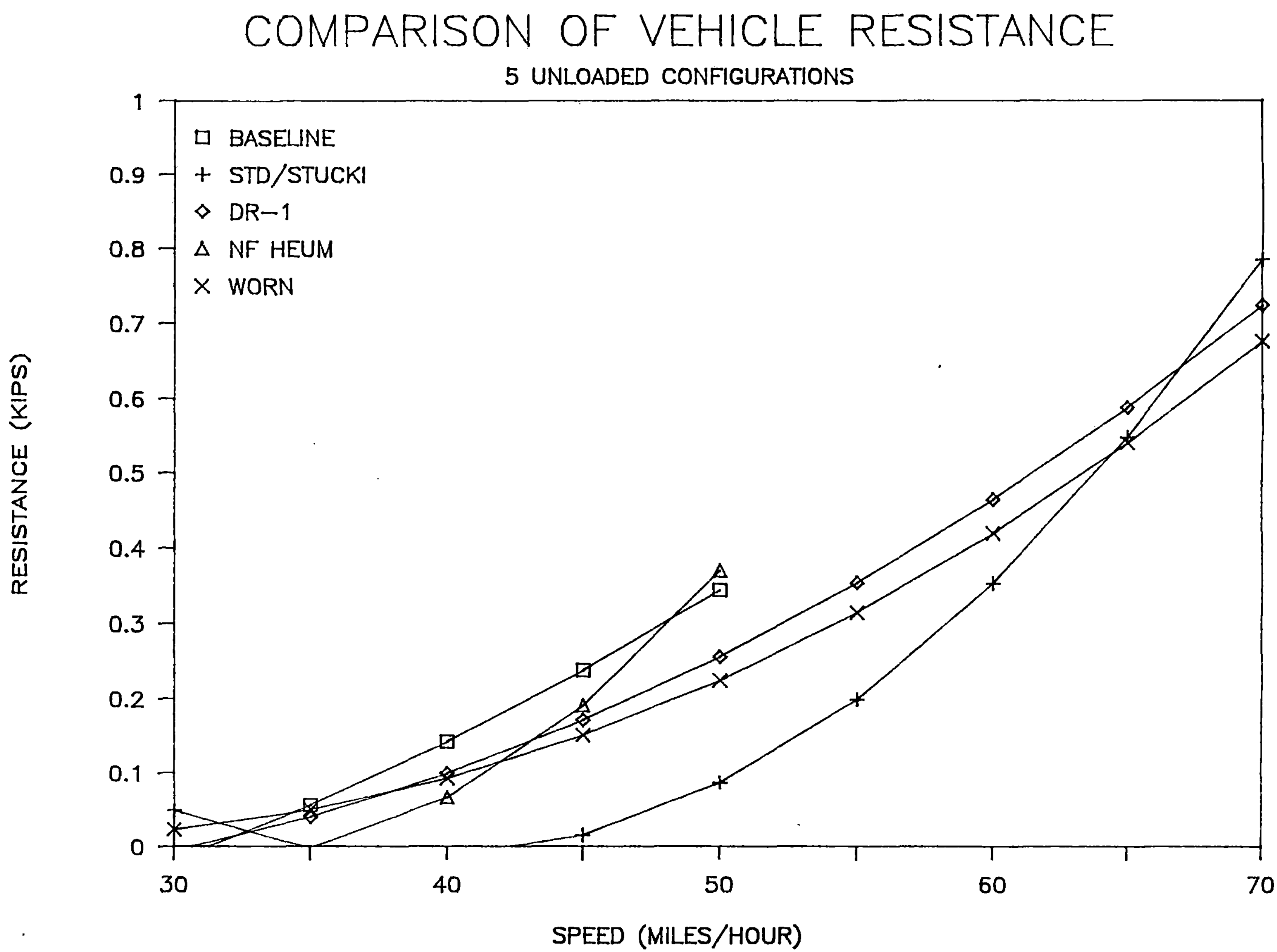


FIGURE 8-24. COMPARATIVE PLOT FOR FIVE TEST CONFIGURATIONS RESIST./SPEED.

TABLE 8-1

COMPARISON OF PREDICTED RESISTANCE
ESTIMATED AT 50 MPH

CONFIGURATION	RESISTANCE (LBS.)
BASELINE	350
CONV. TRK/STUCKI	90
DR-1	260
N. FLANGE HEUM.	370
AVG. FAST WORN	220

9.0 COMPARISON OF TEST DATA WITH GENERAL VEHICLE MODELS

Since hunting of freight vehicles in North America has been a recognized problem for a number of years, development of analytical models that predict this behavior has been undertaken by several parties. The results previously discussed will be compared to predictions for two such models.

9.1 LINEAR STABILITY MODEL

The first model covered is a linear model in which a variety of trucks can be described. The model has a maximum of 17 degrees of freedom consisting of:

- o lateral and yaw motion of each wheelset;
- o lateral, yaw, and roll rigid body motions of the car body; and
- o lateral, yaw, and warping motion for each truck.

More elaborate versions of this model exist which account for torsional motions of the wheelset, as well as lateral bending and carbody torsional modes. For the purpose of this exercise, the simpler version is sufficient. This is because the comparisons involved are not generally related to these additional parameters. In this sense, they can be considered constant and fixed for all test cases.

The model uses an eigenvalue/eigenvector method for analyzing the stability of rail vehicles using linear approximations. The model allows for the description of a variety of truck types, load conditions, and wheel/rail contact geometry. The products of the model as they relate to this particular investigation include the speed at which the vehicle becomes unstable and the oscillatory frequency of the major vehicle components. The onset speed prediction is used in the following section to compare with the experimental findings.

9.2 NUCARS MODEL PREDICTIONS

A general dynamic vehicle model recently being developed by the AAR has been entitled NUCARS. This is an acronym for New and Untried Car Analytical Regime Simulation. Unlike the linear stability model, NUCARS is capable of simulating severely non-linear systems. Suspension characteristics can be expressed by straight line segments or by complex hysteresis loops. A lumped mass vehicle description technique allows for complicated vehicle analysis where appropriate. It is initiated by providing run parameters such as initial vehicle conditions, velocity, and track description. It also allows for complete description of possible wheel/rail contact conditions.

The results include time-histories from simulated "transducers" which record the predicted motion of specific vehicle elements. Typical time histories would involve lateral axle and carbody motion, vertical motion, roll response, and derived values such as velocity, and acceleration. Subsequent analysis can be carried out to determine frequency domain characteristics and phase angle relationships for particular vehicle components.

9.3 TEST AND MODEL RESULT COMPARISONS

At the time that this report was prepared, the NUCARS Model was still under development. Comparison of NUCARS predictions with test results is part of the FRA Safety Aspects of New Trucks and Lightweight Cars Test Program. The intention is to also use limited runs from the VTI data base as further validation.

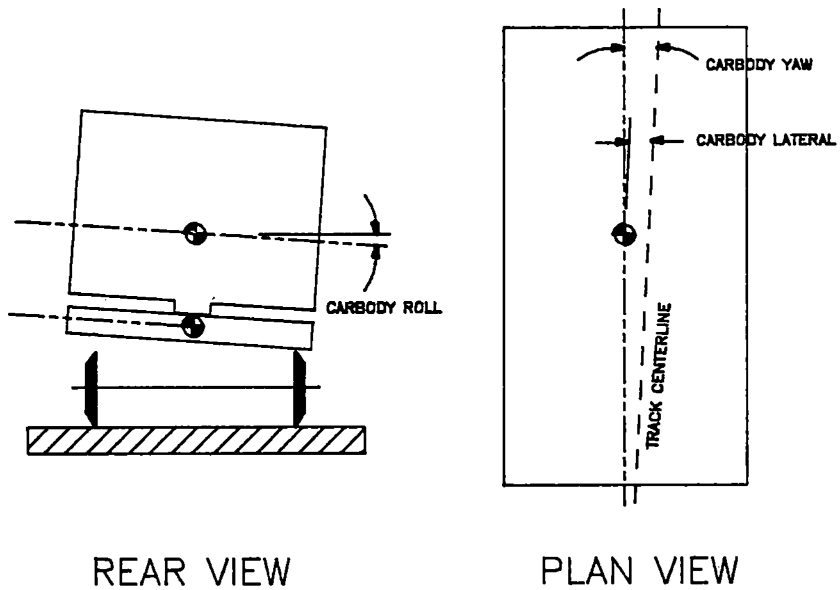
Direct comparison of theoretical results have been made using a Nine Degree of Freedom version of the previously described Linear Stability Model. This model determines the hunting onset speed for a rail vehicle with the following degrees of freedom:

DEGREE OF FREEDOM	DESCRIPTION
1	Car Body Roll
2	Car Body Yaw
3	Car Body Lateral
4	Front Truck Yaw
5	Front Truck Warp
6	Front Truck Lateral
7	Rear Truck Yaw
8	Rear Truck Warp
9	Rear Truck Lateral

Shown in Figure 9-1 is a representation of the motion related with each degree of freedom. The nine degree of freedom model is the simplest full vehicle model which is capable of modeling the lateral stability of a rail vehicle. For this model, the vehicle is described by point masses connected by the appropriate linearized suspensions. The wheel/rail contact geometry was obtained by use of a computer program which produces a contact geometry table, and more important to the nine degree of freedom model, a table of linearized contact geometry characteristics. In order to make predictions for VTI, intermodal flat and hopper car vehicle descriptions were obtained from available literature, and linearized wheel/rail contact geometry was generated using measured wheel and rail profiles from the actual VTI test.

Predictions were first made for the intermodal flat car runs. There were four configurations of the intermodal flat car modelled: empty intermodal flat with roller side bearings; loaded intermodal flat with roller side bearings; empty intermodal flat with constant contact side bearings; and loaded intermodal flat with constant contact side bearings. These four

CARBODY DEGREES OF FREEDOM



BOGIE DEGREES OF FREEDOM

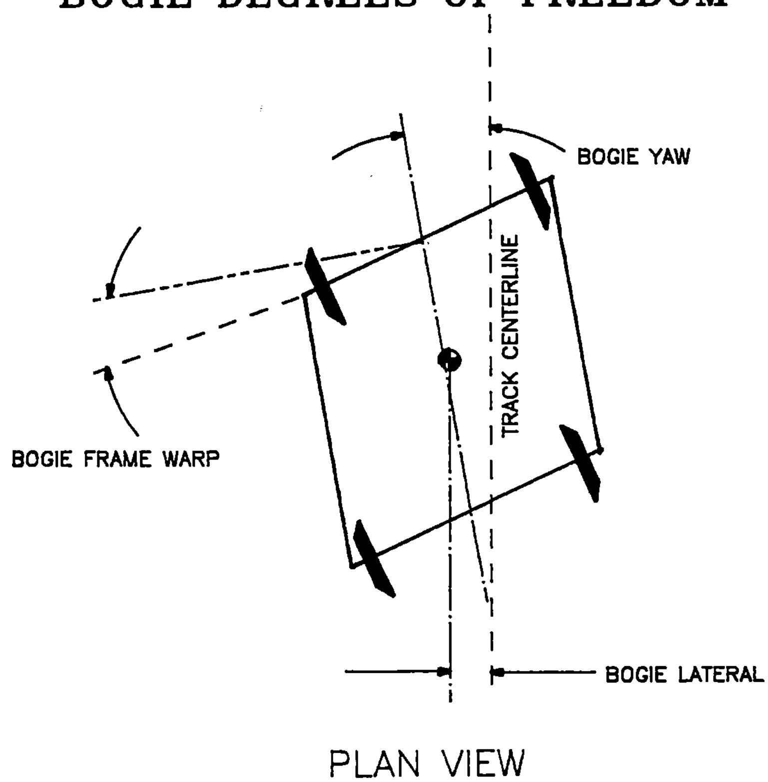


FIGURE 9-1. MOTION RELATED TO VARIOUS DEGREES OF FREEDOM.

configurations were each run with CN Heumann, average FAST worn, and new AAR 1:20 wheel profiles. The test and modelling results are compared in the following discussion. For the empty intermodal flat with roller side bearings, Figure 9-2, the model predictions clearly show good agreement with the test results. In Figure 9-3, the model predicted more modest increases in hunting onset speed than observed with the loaded intermodal flat with roller side bearings. When constant contact side bearings were added to the vehicle model, the 9-degree of freedom results for the empty intermodal flat again showed good agreement with the test results for all wheel profiles as shown in Figure 9-4. When the loaded intermodal flat with constant contact side bearings was modelled, the results showed good agreement with the test for the average FAST worn, and the AAR 1:20 (since both showed no instability over the allowable range of operating speeds). However, the model predicted a considerably greater increase in hunting stability for the loaded case with the CN Heumann profile than was observed in the actual test. These results are given in Figure 9-5.

The next set of predictions were made for a hopper car with conventional three-piece trucks, various loading conditions, roller and constant contact side bearings, and the CN Heumann wheel profile. For the hopper car with roller side bearings in Figure 9-6, reasonably good agreement can be seen between test and model results for the empty car with a Heumann profile on wide gage, empty car with the Heumann, and the two-thirds loaded hopper with the Heumann. When constant contact side bearings were added, the predictions in Figure 9-7 were close to tested results for the empty and a one-third loaded intermodal flat, especially when looking at the change in hunting onset speed from the roller bearing to the constant contact side bearing cases. For the two-thirds full hopper case with the constant contact side bearings, the model results are more conservative than the test results.

In general, the model was able to establish similar trends in hunting onset characteristics as shown in the actual tests. Where discrepancies arose, the cause was probably due to the inaccuracies in the vehicle descriptions. Since the test vehicles were not characterized in the course of testing, "nominal" intermodal flat car and hopper car characterizations had to be used. The inherent assumption here is that the nominal parameters obtained are close to the actual parameters, which may not be the case. For more accurate results, specific vehicles must be characterized or a comprehensive data base of vehicle and truck characterizations needs to be developed. The latter would allow more accurate vehicle descriptions to be assembled for better accuracy when doing general cases, and the former is the only way to ensure accurate results for specific vehicles. Consequently, for this set of modelling runs, the trends runs, the trends represented by the results are more important than the actual numerical results.

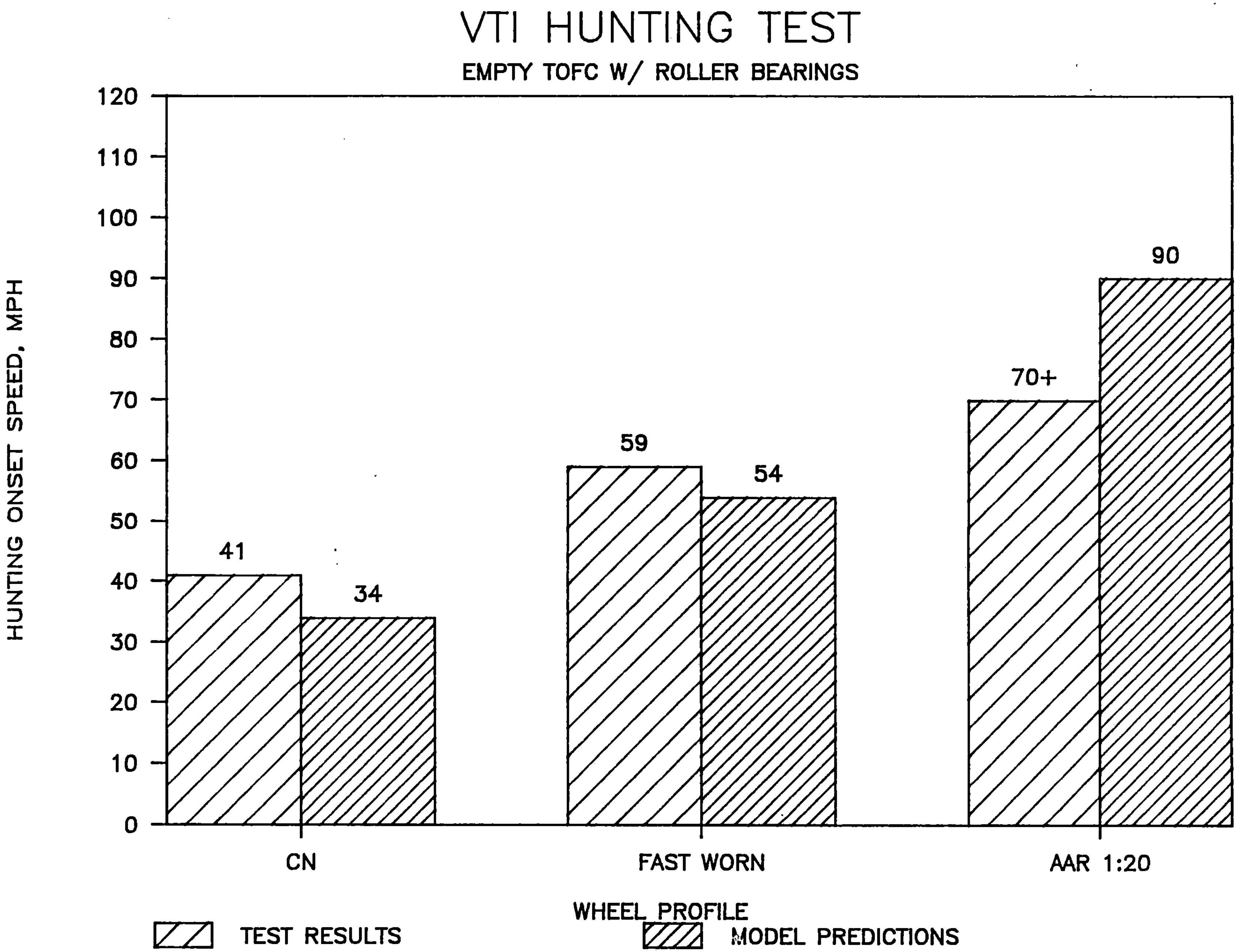


FIGURE 9-2. WHEEL PROFILE PREDICTIONS - EMPTY TOFC W/ROLLER BEARING.

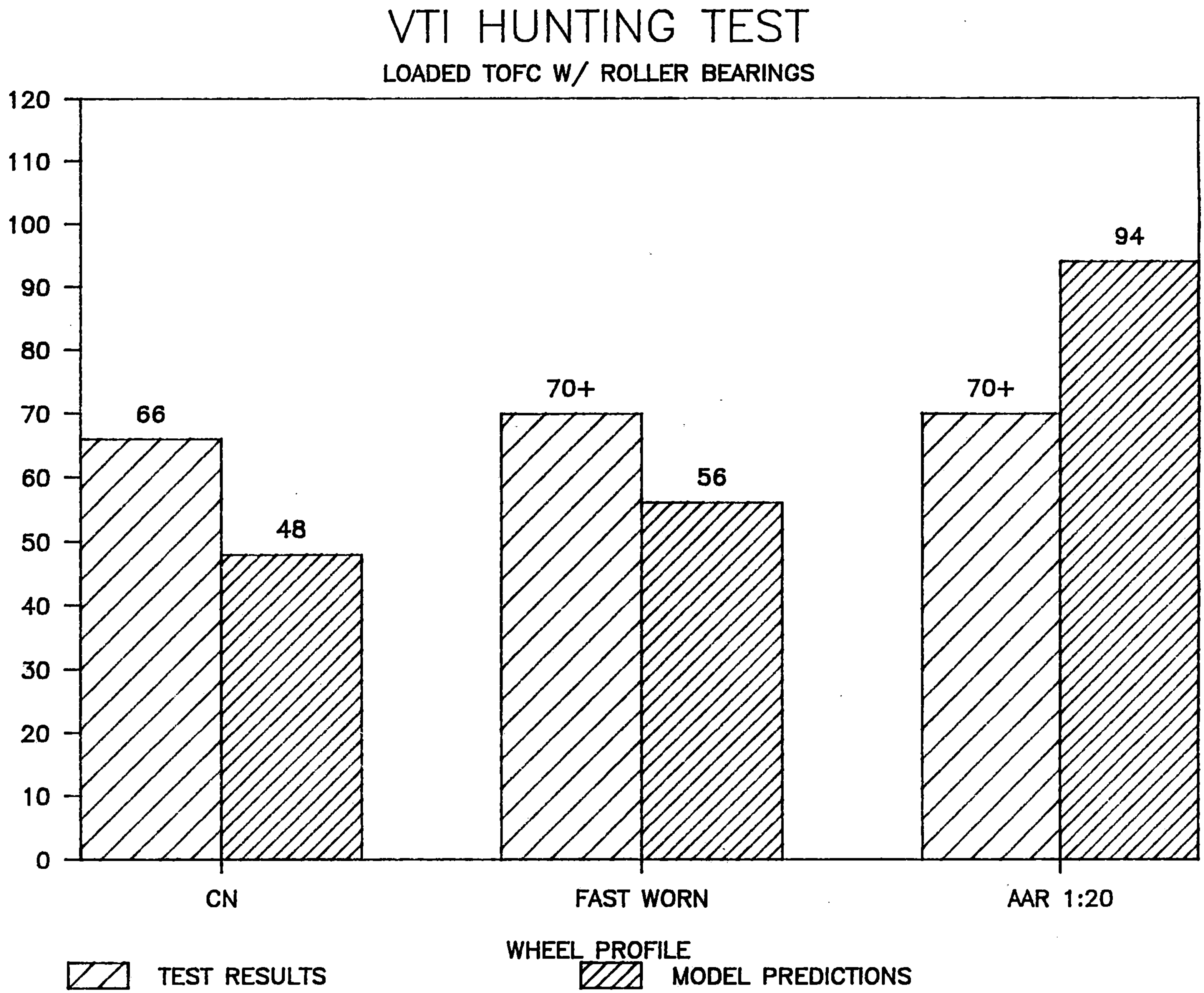


FIGURE 9-3. WHEEL PROFILE PREDICTIONS - LOADED TOFC W/ROLLER BEARING.

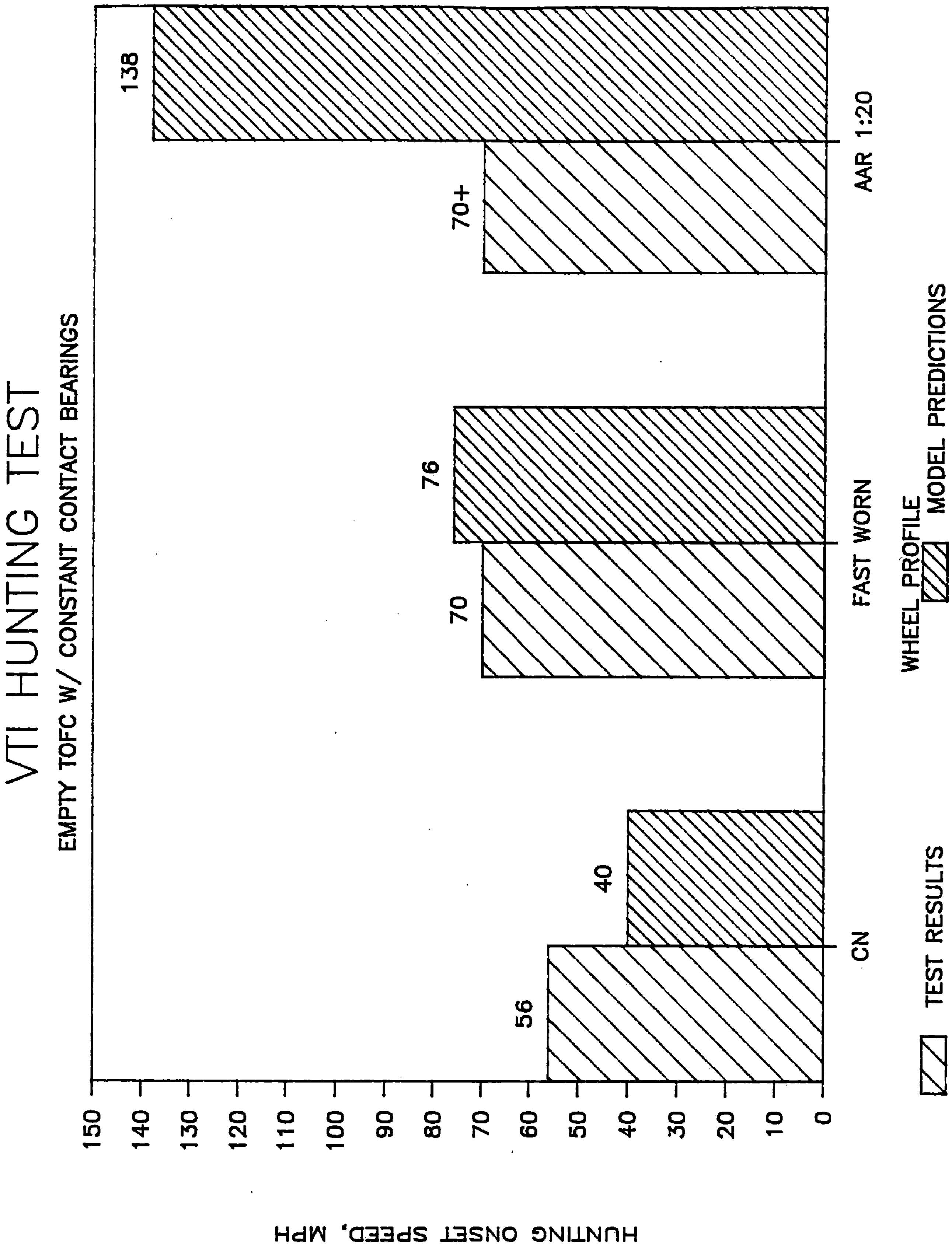


FIGURE 9-4. WHEEL PROFILE PREDICTIONS - EMPTY TOFC W/CONSTANT CONTACT BEARINGS.

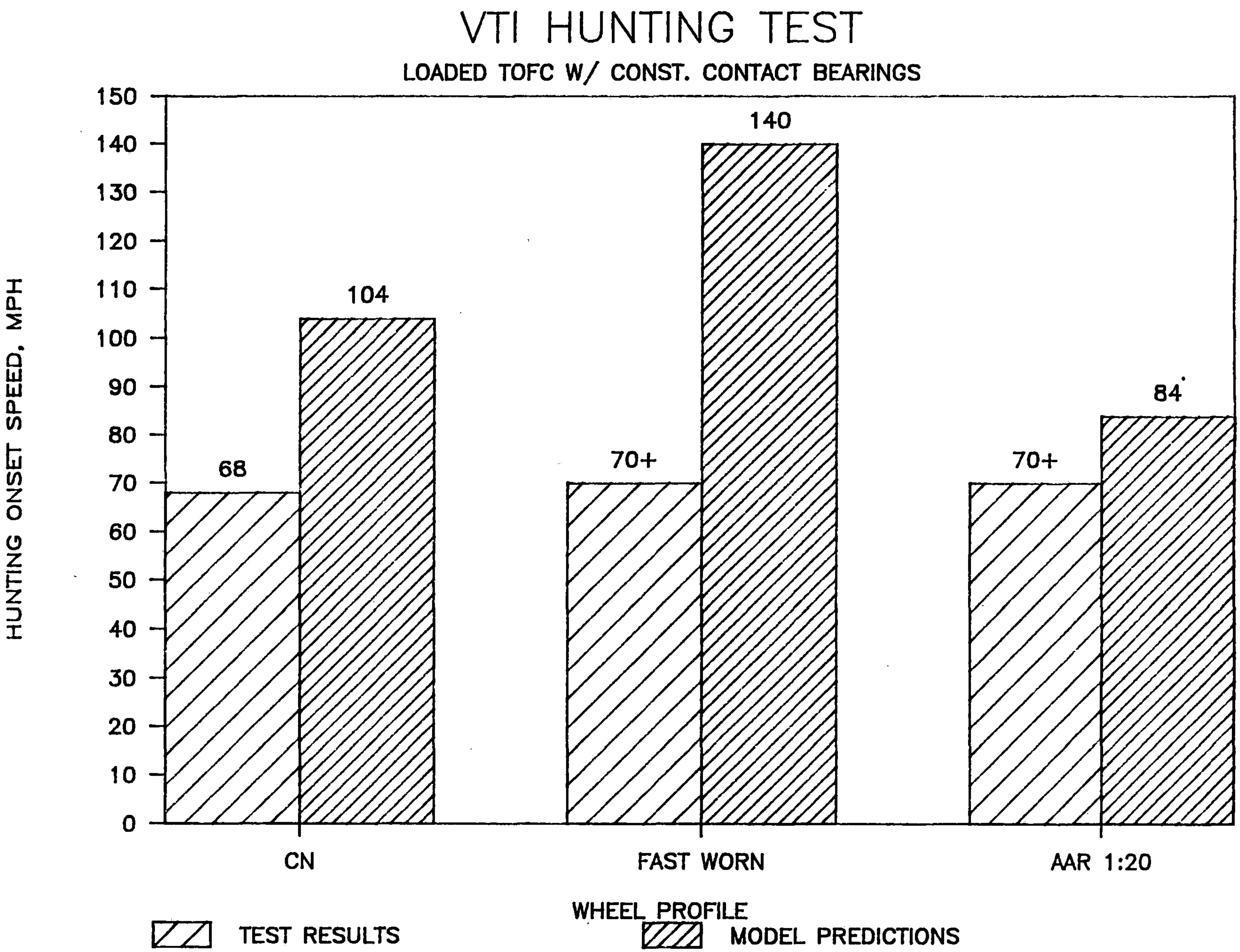


FIGURE 9-5. WHEEL PROFILE PREDICTIONS - LOADED TOFC W/CONSTANT CONTACT BEARINGS.

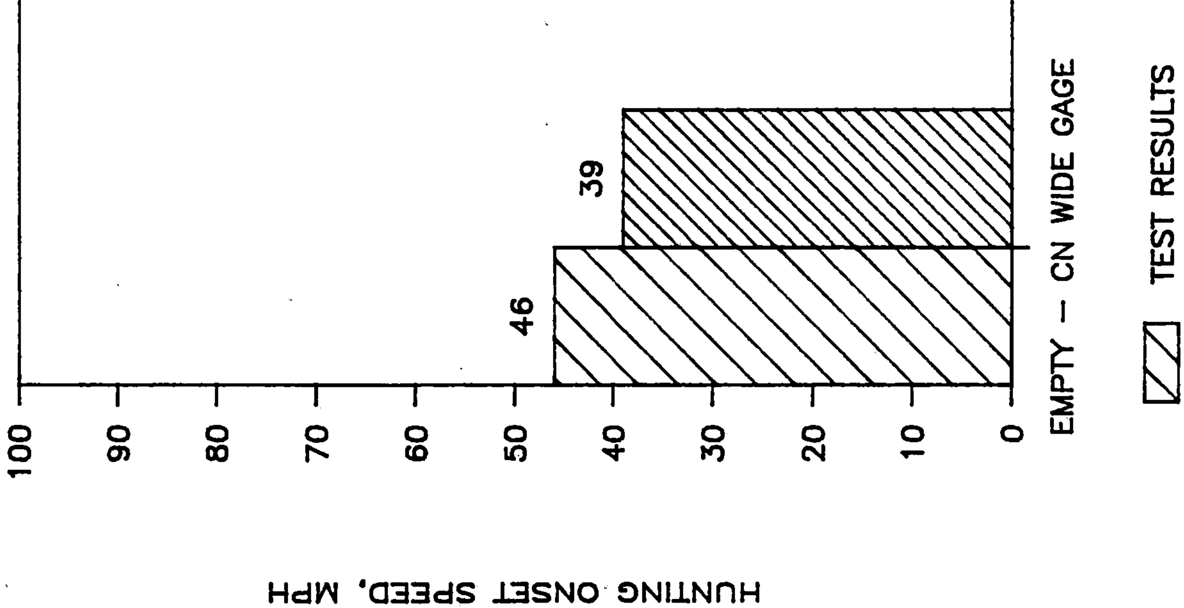
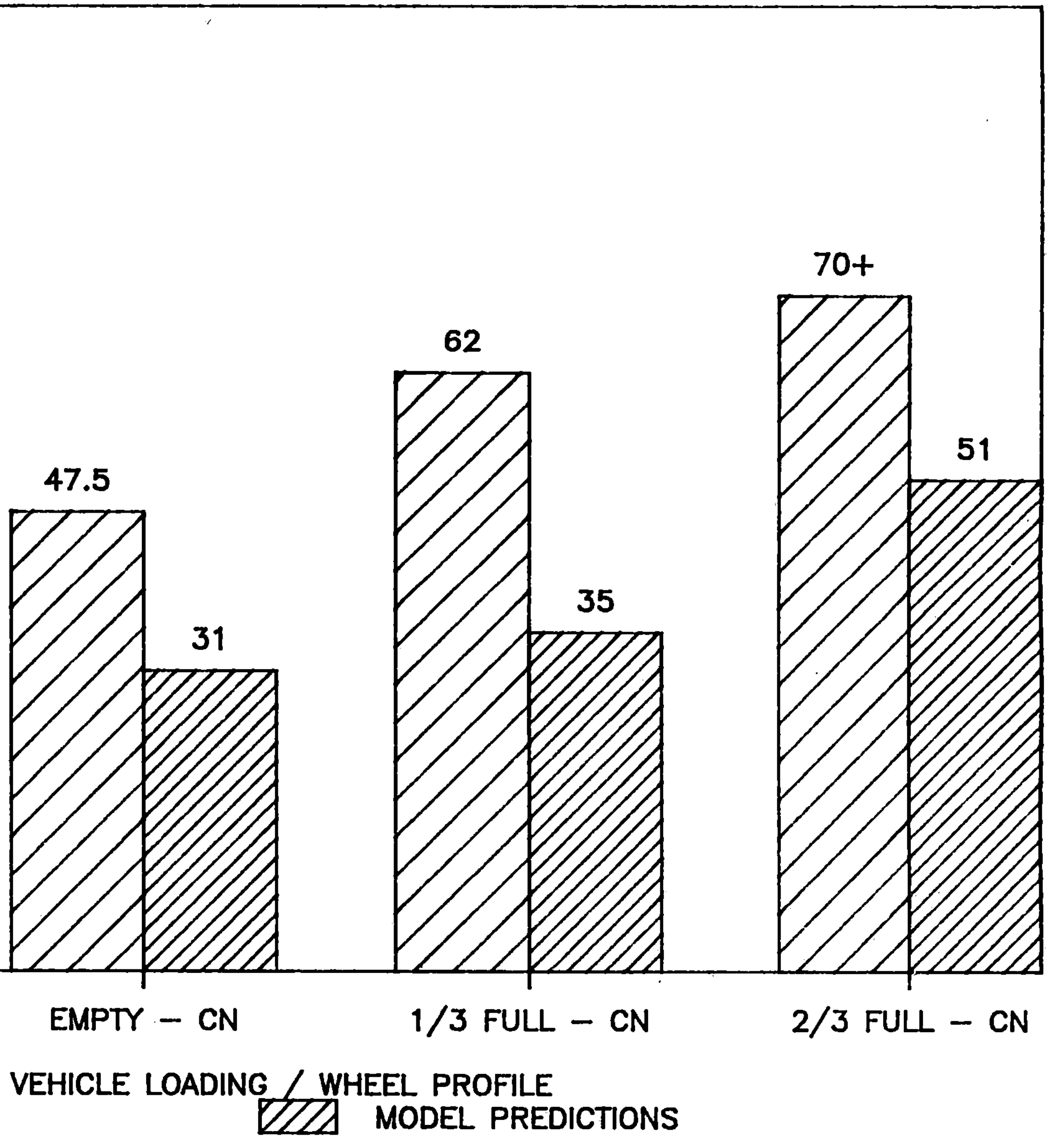


FIGURE 9-6. VEHICLE LOADING/WHEEL PROFILE PREDICTIONS – HOPPER W/ROLLER BEARING.

VTI HUNTING TEST

HOPPER W/ ROLLER BEARINGS



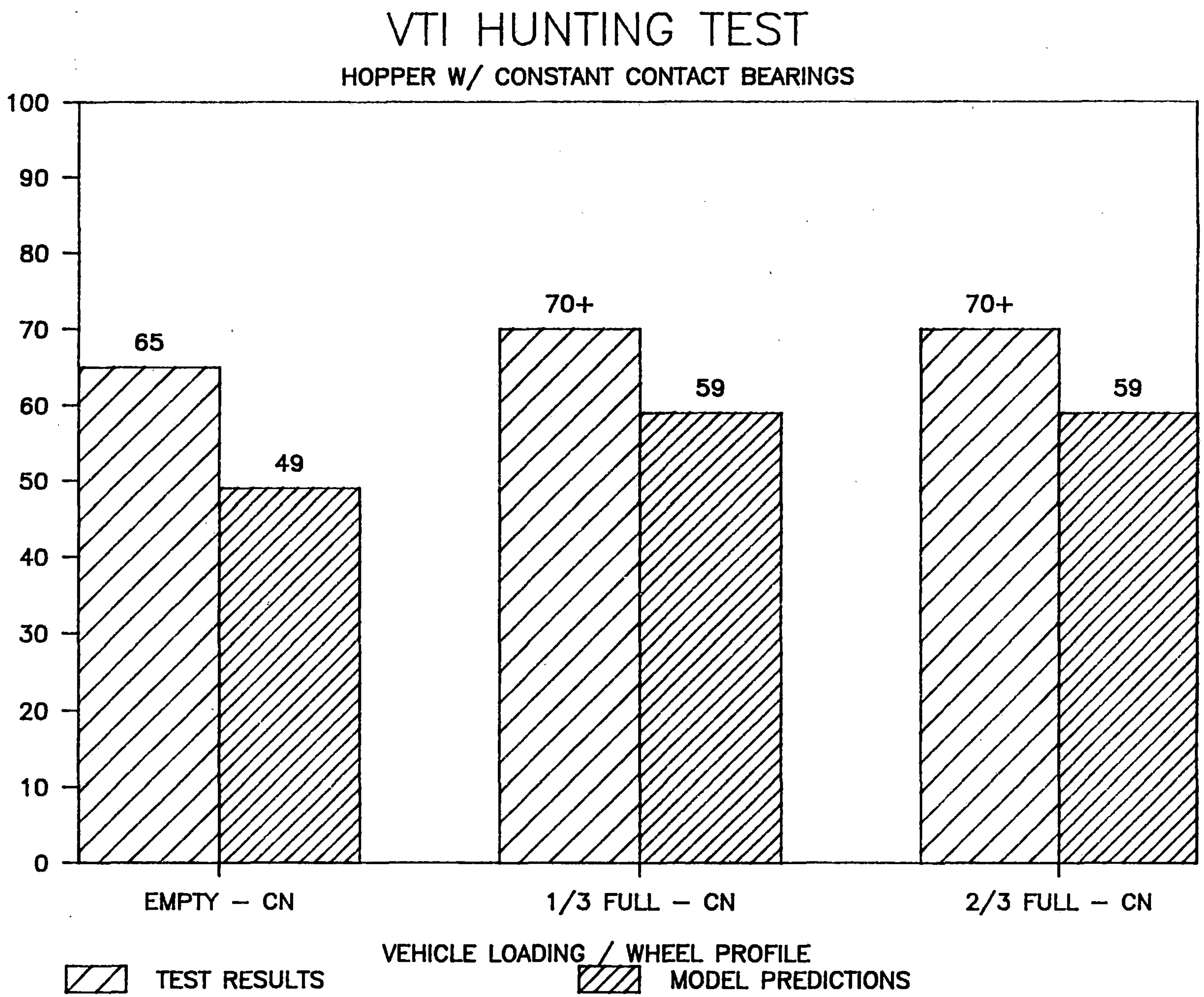


FIGURE 9-7. VEHICLE LOADING/WHEEL PROFILE PREDICTIONS - HOPPER W/CONSTANT CONTACT BEARINGS.

10.0 CONCLUSIONS

The VTI hunting test has met the ultimate objective of generating a significant and general data base. The overall trends between hunting onset and major variables have been established. The most important was the observation and measurement of the general reaction of two major classes of rail vehicles to several variables known to affect hunting. This was studied in the context of axle and carbody acceleration. Next in importance was the observation of force/acceleration changes for the two cars tested. The following sections summarize these findings.

10.1 INTERMODAL FLAT CAR TESTS

The controlled test variables are treated separately with an emphasis on the most important observations. The basic intermodal flat car response to changes in equipment are covered here.

10.1.1 Axle Load

For the 89-foot flat car baseline configuration, the threshold speed increased by 25 mph when retested with two loaded trailers. This put the loaded onset at about 66 mph with the Heumann profile. This change explains why the observation of severe hunting is generally associated with unloaded intermodal flat car equipment. In general, the loaded case has not gained the reputation for serious hunting.

10.1.2 Side Bearing Influence

In response to the problems associated with hunting of empty or unloaded freight equipment, a number of constant contact side bearing designs have evolved. One such product tested increased the onset speed from 41 to 56 mph. Although this is a significant increase, the instability still occurs within the range of typical operating speeds.

The resulting forces, however, are significantly less. This additional benefit, if generally true, has not been raised. A 30 percent to 40 percent reduction in peak lateral force may reduce track damage and the likelihood of derailment.

10.1.3 Wheel Profiles

Considerable effort has gone into designing profiles which will delay the onset of hunting. The new AAR 1:20 wheel profile has been shown to provide a stable operation beyond the 70 to 80 mph range. It is also well documented that the new 1:20 has a rather short life in that form, due to the relatively poor curving response with three-piece equipment. As the shape changes, so does its hunting response. The design goal has been to adopt a profile which yields acceptable hunting and curving performance.

The sole objective of the profile investigation within the VTI program was to see if a detectable trend (with profile change) existed. The intermodal flat car results demonstrated that the 1:20 is a superior wheel profile from a hunting viewpoint. It showed little signs of instability. The FAST worn wheel, which provides an effective conicity somewhere in between that of typical service worn wheel and a new wheel, brought the hunting speed down to 59 mph. The CN Heumann dropped the critical speed further to just over 40 mph.

Testing to develop a reasonable trade-off between curving performance and hunting is presently underway within the AAR. Major emphasis is now being placed on track/rail profile modifications to increase hunting onset.

10.2 HOPPER CAR TESTS

The 100-ton capacity coal hopper testing was intended to investigate another typical North American rail car. The differences in the designs are most notably the reduced length, increased capacity, and a generally captive operating service environment.

10.2.1 Axle Load

The coal car was tested empty, at one-third capacity, and at two-thirds capacity. The intention was to develop sizeable changes in axle load which would produce noticeable variation in stability. The baseline unloaded car went unstable at roughly 47.5 mph. By loading 66,000 pounds of ballast into the car, the threshold elevated to 62 mph. With the highest axle load, which included 132,000 pounds of ballast, the onset speed was greater than 70 mph.

10.2.2 Side Bearings

Similar to the 70-ton intermodal flat car arrangement, the hopper car was outfitted with side bearings intended to increase the initiation of hunting. The addition of the constant contact side bearings increased the speed by 17.5 mph. This is very similar to the changes seen in the intermodal flat car test case.

An observation also was made of the change in lateral wheel force with respect to acceleration for the side bearing equipped car. The peak forces during flange-to-flange hunting with maximum acceptable accelerations produced force reductions of 40 percent.

10.2.3 Wheel Profiles

The role of the wheel profile in determining the stability of the freight car has been discussed. The reaction of the hopper car to the change in profile was very noticeable.

The 1:20 was not tested since the wheel has proven stable in past tests. The intermediate conicity generated by the average FAST worn wheel increased the hunting speed by approximately 20 mph. There was a minute difference in the narrow flange version of the CN Heumann and the wide flange.

10.2.4 Truck Design

The interest in reducing wheel and rail wear, as well as train resistance, has brought about the advent of steering freight car trucks. There were several reliable designs available in the late 1970s. Among these, three were tested in the FAST program for wheel and truck wear, and overall reliability.

By interlocking the wheelsets, an improvement (in hunting performance compared to a three-piece truck) has also been detected.

The DR-1 equipped trucks increased the threshold speed from 47.5 mph to over 70 mph. Thus, the radial or steering design yields a significant benefit in terms of stability as well as the previously documented reduction in wheel and rail wear.

11.0 RECOMMENDATIONS

The overall objective of the VTI hunting project was to provide reliable results for a series of test runs under the most uniform conditions possible. In this respect, the test was very successful. Great care was taken to insure the validity and accuracy of the data. Consistency was probably the greatest virtue of the test; without this, the results could not be used in either an absolute or comparative sense.

The results have put some unknowns, such as the hunting wheel/rail force environment, into perspective. For the two cars tested, the measured force against acceleration characteristics are distinctly different. This may have to do with the relative amplitude and phasing of the body relative to the truck and axle motions. A difference in the fundamental hunting frequency for the two cars was identified. There also seems to be a change in force for equal accelerations when the side bearings are changed. The measurement of L/V ratio and the significance in terms of derailment potential can now be further explored with the data acquired in the tests.

It appears that the resistance of hunting vehicles may contribute significantly to the energy required of locomotive power.

Lastly, the analytical results indicate some promise in terms of general reliability.

As with any large test program, certain generalities must be made at the expense of some of the more subtle indications in the data. These indications may warrant further scrutiny and perhaps additional research. The one area that falls into this category is an understanding of the truck that is being studied. For this test, it was assumed that the test specimens were typical, and the results seemed average in many ways. General observations in revenue service and within other vehicle tests make the definition of "typical" difficult. This is due to the wide range of response characteristics for a group of vehicles under identical conditions. To truly explain this variation, a means of quantifying the critical characteristics is needed for a relatively large sample of cars.

Truck characterization has been done in the past for curving analysis and has yielded valuable information. The problem has been that the time and cost has been substantial. Efforts are now underway at the TTC to allow for more rapid truck characterization. This will result in characterization of test vehicles at a cost lower than what was possible in the past.

This would allow for the routine characterization of vehicles involved in any test. Having this information available would increase the understanding of the stability response in freight cars.

The most obvious use of the data would be to incorporate accurate and specific values into dynamic vehicle models. By having data, which is taken with a high degree of confidence, more accurate models may evolve. The benefits of this development will ultimately be gained by the railroad industry in terms of safety. It could also help the railroads in their decision making process regarding car design and approaches to dealing with existing problems.

REFERENCES

1. Task Proposal - Test Plan, Vehicle/Track Interaction, Task Request No. 14, Contract DTFR-53-82-C-00282, April 1984.

2. Southern Pacific Transportation Company Technical Research and Development Group, "Freight Car Truck Design Optimization, Vol. VIII - Results Report for Test Series 2 and 5," Final Report FRA-OR & D-78-12, VIII, May 1978.

3. Hengel, M.F., Montgomery, C., and Billingsley, R.H., "The Influence of Truck Hunting Severity on Freight Car Structures," Paper No. 79-WA/RT-1, American Society of Mechanical Engineers, New York, NY, August 1979.

4. ElMadany, M.M. and RamaChandran, P.V., "Lateral Stability of Flat Rail Cars - An Over-the-Road Investigation," Int. J. of Vehicle Design, Vol. 2, No. 2, 1981.

5. ElMadany, M.M. and RamaChandran, P.V., "Lateral Dynamics of 70-Ton Freight Car Trucks," Paper No. 81-WA/RT-2, American Society of Mechanical Engineers, New York, NY, November 1981.

6. Hannerbrink, D.N., Lee, H.S.H., Weinstock, H., and Hedrick, J.K., "Influence of Axle Load, Track Gauge, and Wheel Profile on Rail-Vehicle Hunting," Paper No. 76-WA/RT-3, ASME Journal of Engineering for Industry, August 1976.

7. D'Souza, A.F. and Caravavatna, P., "Analysis of Nonlinear Hunting Vibrations of Rail Vehicle Trucks," Paper No. 79-DET-25, American Society of Mechanical Engineers, New York, NY, 1979.

8. Anderson, J.A. and Byrne R., "High-Speed Lateral Stability of a Freight Car Related to Modifications of Conventional Trucks," ASME Journal of Engineering for Industry, Vol. 100, February 1978.

9. Yokose, K., Takahashi, S., and Yamaguchi, J., "On Running Stability of Three-Piece Bogie and Its Prevention of Truck Hunting," Railway Technical Research Institute, Quarterly Reports, Vol. 19, No. 4, 1978.

10. Hadden, J.A., and Law, E.H., "Effects of Truck Design on Hunting Stability of Railway Vehicles," Paper No. 76-WA/RT-1, American Society of Mechanical Engineers, December 1976.

11. Law, E.H., Hadden, J.A., and Cooperrider, N.K., "General Models for Lateral Stability Analyses of Railway Freight Vehicles," Interim Report No. FRA-OR & D-77-36, June 1977.

12. Hawthorne, V.T., "Truck Hunting in the Three-Piece Freight Car Truck," Paper No. 79-WA/RT-14, American Society of Mechanical Engineers, New York, NY, December 1979.

13. Garin, P.V. and Cappel, K.L., "Some Relationships Between Dynamic Performance of Freight Car Trucks and Worn Wheel Tread - Rail Geometry," ASME/IEEE Meeting, Chicago, IL, April 1976.

14. Elkins, J.A., Blader, F.B., and Leary, J.F., "Development of Improved Freight-Car Wheel Profile," ASME Winter Annual Meeting.

15. Heller, R. and Cooperrider, N.K., "Users' Manual For Asymmetric Wheel/Rail Contact Characterization Program," Interim Report No. FRA/RRD-11, December 1977.

16. Cooperrider, N.K., Law, E.H., Hull, R., Kadala, P.S., and Tuten, J.M., "Analytical and Experimental Determination of Wheel-Rail Constraint Relationships," Interim Report No. FRA-OR & D-76/244, December 1975.

17. Gilchrist, A.O. and Brickle, B.V., "A Re-examination of The Proneness To Derailment of a Railway Wheelset," Journal Mechanical Engineering Science, Vol. 18, November 3, 1976.

Vehicle Track Interaction Truck Hunting, 1989
US DOT, FRA, John F Leary