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SAFETY ASPECTS OF NEW TRUCKS AND LIGHTWEIGHT CARS, CAR 2

Office of Research and Development
Washington D.C. 20590

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METRIC CONVERSION FACTORS

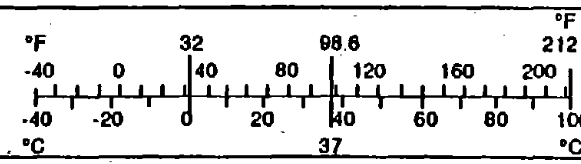
Approximate Conversions to Metric Measures

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

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

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hecteres (10,000 m^2)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32	Fahrenheit temperature	°F



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16. Abstract

The Federal Railroad Administration and Association of American Railroads (AAR) sponsored a program to validate new techniques for the analysis and testing of safety and track worthiness aspects of new freight cars and suspension designs. The program involved subjecting a lightweight aluminum coal gondola to laboratory tests, on-track tests, and simulation of tests using a computer model as required by AAR Specification M-1001, Chapter XI.

Results of the track tests and computer model predictions showed areas where specifications in Chapter XI could be improved. Also, increased accuracy is required from the instrumented wheel sets used to measure wheel/rail forces.

A new facility was successfully developed to measure the suspension characteristic data required for the computer model. The computer model was shown to be a necessary tool for interpretating track test results. Good correlation was shown between test data and model predictions especially when input data was improved to more closely represent actual test conditions.

Test results and model predictions were used to evaluate the test vehicle's safety performance, which exceeded Chapter XI safety criteria in several test regimes.

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EXECUTIVE SUMMARY

As a part of a joint Federal Railroad Administration (FRA) and Association of American Railroads (AAR) research program, a series of tests and analyses were conducted on a light weight aluminum coal gondola. The tests and computer analyses were based on those required by Chapter XI of the AAR's *Manual of Standards and Recommended Practices*, M-1001. The primary objective was to evaluate the effectiveness of Chapter XI type tests in determining the track worthiness safety aspects of new light weight freight car and truck designs. The general methodology for this project was to:

- Conduct laboratory tests to measure the test vehicle suspension and car body characteristics
- Use the measured characteristics in a mathematical model to predict the dynamic behavior of the test vehicle
- Perform Chapter XI type tests of the vehicle to measure vehicle performance
- Compare predicted performance with test results, and use this comparison to adjust the model parameters to increase the accuracy of the mathematical simulation

Results of the tests and computer model predictions showed several areas where the Chapter XI tests and analyses could be improved. Alterations to the wording of Chapter XI combined with these improvements would clarify the application of Chapter XI to non-standard cars. Specific suggestions include:

- Changing the minimum wheel load criterion to include an absolute minimum acceptable wheel load
- Requiring empty and loaded car tests and analyses for all test regimes
- Changing the lateral car body acceleration criterion to reflect vehicle safety parameters and not car body ride quality
- Changing and/or supplementing the car body roll angle criterion to accurately detect center plate separation

The New and Untried Car Analytic Regime Simulation (NUCARS) computer model predictions were generally successful when compared to test results, provided the model was used with accurate input data. Comparisons were least successful when required to simulate the effects of friction connections. Making small changes in the simulated friction level was shown to have an enormous effect, ultimately producing very good correlation with the test data especially in the pitch and bounce test section. Modifications to the NUCARS model are underway to improve modeling of these friction connections.

The large variability in predicted car performance due to small changes in friction implies that actual test vehicle performance will be heavily dependent on the actual friction level. This could present problems of repeatability between similar vehicles whose friction levels are different due to manufacturing tolerance or other small design differences.

To provide better quality input data for the NUCARS model, the AAR successfully developed a new laboratory facility known as the Mini-Shaker Unit (MSU). The MSU is capable of measuring suspension stiffnesses and damping, and rigid and flexible body modal characteristics, such as resonant frequencies and structural damping. The MSU was used in place of the Vibration Test Unit (VTU) that had been used in previous projects. The MSU is much less complicated, less expensive to operate and produces more accurate results than the VTU. The MSU has subsequently been used in many other major test programs at TTC with great success.

The post-test predictions were also improved by using revised lateral and yaw suspension characteristics derived from dynamic measurements made during the track tests. Measured axle misalignments were also included. Most of the derived characteristics compared favorably to the MSU characterization test results and to results being measured in other AAR research programs. This method of deriving suspension characteristics from track test data shows promise for future projects. More accurate instrumented wheel sets will be required to provide reliable results by these means, as well as more extensive instrumentation to measure the accelerations of all the trucks' component parts.

The test car was equipped with premium quality modified three-piece trucks which included primary shear pads at the bearing adaptors, and redesigned friction snubbers to increase the truck warp stiffness. It appears that the shear pad characteristic data from the MSU tests used in the modeling was incorrect. The stiffness value used in the simulations was probably too high, preventing a good correlation with the test data in curves and possibly the yaw and sway test zone. It appears to have had more effect on the empty car predictions than the loaded car. It is likely that estimates of the shear pad stiffness made from the track test data could have clarified this problem.

Pre-test predictions were made using track inputs defined mathematically. Post-test NUCARS predictions were made after completion of the track tests using actual measured track geometry as input. These measurements were made by ENSCO using their inertial track geometry measurement system. The measured track data included the surface roughness actually present in the track. This in combination with the accurate measurement of the actual perturbation amplitudes caused the post-test predictions to match the test results more closely than the pre-test predictions. For future modeling efforts it would be useful to impose a surface roughness on top of the simulations of perfect perturbations. These results also imply that it is necessary to impose construction tolerances on the test track perturbations to ensure repeatability of the tests.

Using the NUCARS computer model was essential to support the track test results to determine compliance with the established safety criteria. In the case of the empty car tests the instrumented wheel sets were so unreliable that the NUCARS predictions were required to validate the wheel set force measurements. It may be difficult to ever build an instrumented wheel set capable of measuring empty car forces as accurately as required by Chapter XI. Therefore computer simulations should be carried out to assist in interpreting test results. In order to make valid comparisons between test and model, a small amount of additional instrumentation to measure suspension deflections will be required. In the case of a traditional vehicle with three-piece trucks this could be as few as four string potentiometers measuring spring deflections, with two more measuring bolster rotation angles.

Instrumented wheel sets are necessary for measuring the safety criteria required by Chapter XI type tests. During the tests it was found that the currently available wheel sets are not accurate enough for testing empty or light weight cars. The measured wheel/rail force test data showed extreme variability especially when compared to model predictions made with AAR's NUCARS computer model. The analyses show that the empty car test results should be viewed with caution. Most of the force and L/V data from the track tests are subject to a very wide margin of error.

A review of currently available instrumented wheel sets indicates that none is likely to be able to resolve the very low loads imparted by a light car such as the one tested. It is doubtful that any are now capable of better than +/- 1000 pound accuracy. This is inadequate for testing a car with a 5000 pound wheel load. The measurement accuracy of the L/V ratios is even worse because these are taken as a ratio of two measured forces.

A secondary project objective was to evaluate the safety aspects of the aluminum coal gondola using the Chapter XI type tests and NUCARS analyses. Chapter XI criteria were measured and/or predicted to be exceeded for the following test regimes. Note that not all of these tests are currently part of Chapter XI. Due to the previously mentioned inaccuracies in the empty car wheel force data, some test results could be erroneous.

1. Empty Car Tangent Hunting (model and test)
2. Empty Car Curved Hunting (model and test)
3. Loaded Car Tangent Hunting (model and test)
4. Loaded Car Curved Hunting (model and test)
5. Empty Car Single Bounce (model and test)
6. Loaded Car Single Bounce (test)
7. Empty Car Twist and Roll (model and test)
8. Empty Car Yaw and Sway (model)
9. Loaded Car Yaw and Sway (model and test)
10. Loaded Car Dynamic Curve (test)

The poor behavior was largely attributed to a tendency for the truck to hunt at high speeds when the car was both loaded and empty. The truck design features that allowed the hunting did however permit good curving behavior. Large amounts of friction in the truck suspension were also identified as possible causes of poor vertical response in the empty condition.

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1.0 INTRODUCTION

Recently there has been a significant increase in the rate of introduction of new freight car and truck designs. This has occurred for a number of reasons including the railroad industry's desire to carry greater loads and the increase in demand for intermodal traffic. This latter requirement gives relatively low vehicle loads and as a result has led to the development of a number of single axle suspension designs and articulated vehicles with trucks under each articulation joint. In order to minimize energy consumption and maximize load-to-tare ratios, vehicle bodies are being designed with lightweight structures, which can include new materials untested in the railroad environment.

For many years, the vast majority of freight cars running on the North American freight railroads have been equipped with two conventional three-piece trucks. Accordingly there is considerable experience with regard to the performance of these cars. With the introduction of new intermodal car designs, the Mechanical Division of the Association of American Railroads (AAR) recently introduced Chapter VIII of its *Manual of Standards and Recommended Practices*,[1] containing requirements for testing these vehicles. In 1984 the AAR took the initiative to form an ad-hoc committee with industry wide support for the purpose of applying recent technology advances to the approval process for all new freight car designs. This committee made recommendations to the AAR Mechanical Division's Car Construction Committee for a series of tests and analyses to be performed as a part of a new certification process for new vehicle designs. These requirements, approved in 1987 by the Car Construction Committee, are part of the AAR's *Manual of Standards and Recommended Practices*, Chapter XI.[1] A copy of the Chapter XI requirements is included in Appendix A.

Prior to the acceptance of the new Chapter XI certification test process by the Car Construction Committee, the Federal Railroad Administration (FRA) and the AAR sponsored a research program that developed new safety criteria for identifying critical response parameters of rail vehicles, with tentative limits and test requirements, to evaluate the dynamic performance of new vehicles. The tentative limits and test requirements used for the evaluation were the guidelines proposed under Chapter XI.

Phase 1 of this research program, "Safety Aspects of New and Untried Freight Cars," tested a newly designed vehicle selected by the project steering committee. The vehicle was the TTX Company (previously known as Trailer Train Company) skeletal platform car, with single axle, leaf spring suspension, known as the Frontrunner(Tm). These tests are referred to as the Lightweight Car 1 tests.

Chapter XI guidelines suggest that a new vehicle be mathematically modeled to predict its dynamic response to the track irregularities defined for the on-track tests. The newly developed New and Untried Cars Analytic Regime Simulation (NUCARS) computer model was used and partially validated as a part of this project (Phase 1). This model was used to predict response to all the proposed Chapter XI test tracks.

To provide initial input parameters to the NUCARS mathematical models, such as suspension stiffnesses and body structural flexibility parameters, Chapter XI recommends performing vehicle characterization tests to measure this data. The Lightweight Car 1 test program investigated various methods for determining vehicle parameters for input to NUCARS and was completed by conducting all the recommended Chapter XI track tests on the test vehicle. Test results were compared to NUCARS predictions to partially validate the model's ability to predict a vehicle's dynamic response to known track irregularities.[2]

Completion of the Car 1 test program has assisted in the development of new safety criteria for critical response parameters and test requirements to evaluate the dynamic performance of new vehicles. To continue this process, the FRA has begun Phase 2 of this program to evaluate the dynamic performance of a second lightweight vehicle and to further develop safety performance criteria and test requirements.

The scope of the Car 2 test is similar to Car 1, which is essentially applying Chapter XI to a new vehicle design, measuring its suspension and other parameters, modeling it with NUCARS, performing a series of track tests, and comparing model predictions with track test results. In this case, the project is jointly funded by the AAR and the FRA, with the AAR funding the development of new facilities and tests for performing the vehicle characterizations, and the FRA funding the remainder.

2.0 OBJECTIVES

This project has two basic objectives:

1. To determine whether relatively inexpensive procedures can be devised which could be used for the analysis and testing of the safety aspects of new designs of lightweight cars and trucks.
2. To evaluate the safety aspects of a new design of lightweight car and truck using these procedures.

It is hoped that if successful procedures are developed, these could become part of a revised set of testing requirements for new car designs to be used voluntarily by the industry.

3.0 PROJECT METHOD

3.1 GENERAL PHILOSOPHY

The project was organized to evaluate the safety aspects of a new vehicle using a predetermined set of analyses and test procedures. The overall flow of the project was envisioned as follows:

1. Measure the vehicle's suspension and car body resonance characteristics.
2. Perform a pre-test analysis of the vehicle by mathematically modeling with NUCARS.
3. Subject the vehicle to a predetermined track test sequence similar to Chapter XI.
4. Perform post-test analysis using a specially modified version of NUCARS that reads actual track geometry data for input. Make use of test results to refine the input to the NUCARS model.
5. Compare track test results with model predictions and determine the safety performance of the test vehicle. From these results, evaluate the analysis and test methods used for their effectiveness in measuring vehicle safety performance.

Analysis of the overall results of the Lightweight Car 1 test program indicated several areas for improvement of test and analysis techniques. These were integrated into the test method for this project. The following subsections outline the various phases of this project and how the results of the Car 1 project affected their implementation in this effort.

3.2 VEHICLE CHARACTERIZATION

Vehicle characterization is the process of determining the various vehicle suspension and structural characteristics, such as spring stiffnesses and damping, and car body natural bending modes. This data is then used as input to the NUCARS computer model.

The Lightweight Car 1 suspension characterization measurement tests were performed on the Vibration Test Unit (VTU) at the Transportation Test Center (TTC), Pueblo, Colorado. The VTU proved to be cumbersome for performing these tests, which resulted in test procedures that cannot be regarded as "simple." The VTU is also expensive to operate. For the Lightweight Car 2, it was decided that simpler facilities and tests should be tried. It was also decided that the AAR would fund the development of a new test facility and demonstrate its use during this project.

Some of the tests for characterizing the yaw suspensions involved lifting the vehicle on air bearing tables. These test procedures proved satisfactory for Car 1 and were therefore used for testing Car 2.

3.3 PRE-TEST ANALYSIS

As with Car 1, the pre-test analysis involved modeling the vehicle negotiating appropriate Chapter XI test zones, using the NUCARS computer model.[3,4] NUCARS has been in a continuous state of development since completion of the Car 1 project and has had many improvements in speed and accuracy. Input data for NUCARS was obtained from the vehicle characterization tests and supplemented where necessary by manufacturers' specifications.

A major problem encountered when doing the NUCARS modeling of Car 1 was determining certain suspension characteristics from the characterization tests. The AAR has been developing a computer program for assisting in vehicle parameter identification. This program was used for this project to evaluate its effectiveness in identifying vehicle parameters.

3.4 TRACK TESTS

Similar to the Car 1 project, the Chapter XI test sequences were the basis for the track tests. Alterations included performing all test regimes with both an empty and loaded vehicle. In addition, a wide range of curves was tested to better evaluate vehicle curving behavior.

3.5 POST-TEST ANALYSIS

A version of NUCARS, which reads actual test track geometry as input, was used for post-test modeling for Car 1. An updated version of this program was developed to include all current improvements in speed and accuracy. Actual geometry of the track test zones was measured just prior to the track tests using an inertial track geometry measurement system developed by ENSCO Corporation.

Results of the Car 1 project indicated that predictions of vehicle yaw and lateral suspension dynamics may have been hampered by inaccurate measurement of the yaw and lateral suspension characteristics. For the Car 2 project, an attempt was made to refine the lateral and yaw suspension characteristics by making use of dynamic measurements of these suspensions during the track tests. These refined values were used in the post-test model predictions.

Test results were compared with the post-test model predictions. An evaluation of the performance of NUCARS at predicting test results was made from this comparison. All test results and model predictions were compared to the Chapter XI performance criteria to evaluate the test vehicle's safety performance.

4.0 TEST VEHICLE

The project steering committee set several guidelines in choosing the test vehicle. The vehicle had to be of a new design that had not been subjected to the AAR Chapter XI process and was not in regular service.

These guidelines were chosen to ensure that the vehicle would be of general interest to the railroad community, would represent a significant attempt to improve vehicle performance technology, and would be significantly different than Car 1 (TTUX Frontrunner). At the same time the steering committee wanted a vehicle that was not too radical a departure from current technology so as to be representative of vehicles likely to be designed in the near future.

The guidelines chosen for the test vehicle are as follows:

1. Standard configuration (single car body on two trucks)
2. Bulk or container load (no trailers)
3. Designed for general interchange use
4. Improved or modern truck design
5. Car body design for light weight or extra payload

The chosen test vehicle was a Trinity Industries 100-ton aerodynamic aluminum coal gondola car, known as PSMX 111 (Figure 1). The car is constructed with an aluminum semi-monocoque body with steel stub sills. The light weight (including trucks) is 41,400 pounds, the load limit is 221,600 pounds, and the gross weight is 263,000 pounds. This lightweight construction allows for carrying a load of 11 tons more coal than a normal 100-ton gondola, while maintaining a nominal 33,000 pound wheel load.

For the purposes of this test, the vehicle was equipped with two American Steel Foundries (ASF) Roadmaster trucks (Figure 2). These are a modified three-piece design, having a primary suspension consisting of rubber shear pads at the axle bearing adaptors. The rubber shear pads are designed to center the axles within the pedestal jaws to attempt to maintain the axles square relative to each other. While maintaining nominal alignment, the shear pads have longitudinal and lateral flexibility allowing the axles to "steer." These trucks are equipped with variable rate friction snubbers (dependent on vertical load). The design of the friction snubber castings is also modified to attempt to provide greater resistance to truck lozenging (truck warping).

It must be noted that these trucks were different than the trucks originally installed under this car. The original trucks were a normal three piece truck, with a special two stage spring arrangement. This would have provided a much softer spring rate for the empty car while providing a similar stiffness for the loaded car. To fulfill the project objective of evaluating a new truck design the ASF Roadmaster truck was chosen, because it represented a major modification of the normal three piece truck design.

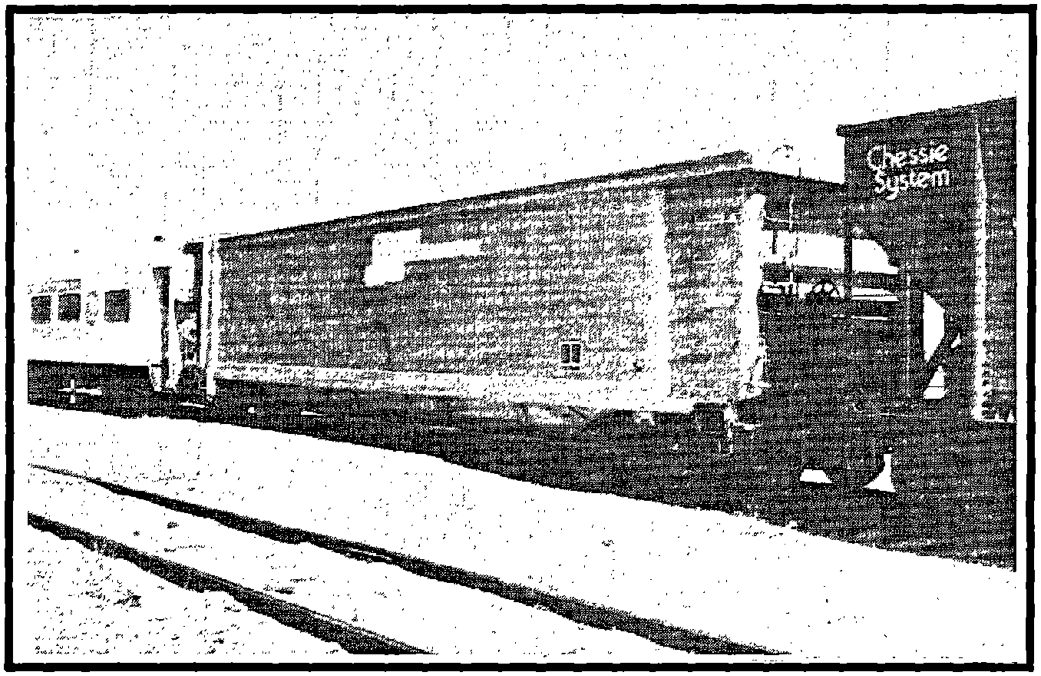

Figure 1. Test Vehicle: PSMX III Aluminum Coal Gondola

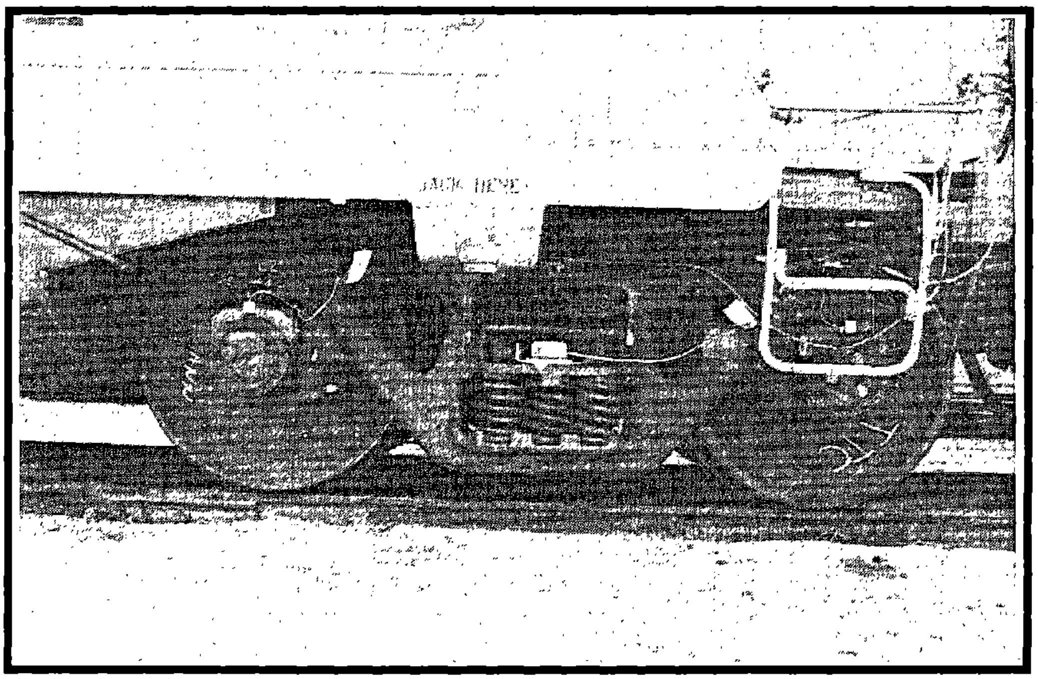

Figure 2. Test Truck: ASF Roadmaster Truck with Test Transducers Installed

The trucks are also equipped with constant contact side bearings to control body roll and truck hunting.

The vehicle suits the criteria selected. Its design is very different from the Frontrunner Car of Phase 1, which will allow for greater confidence in the wide applicability of the testing and analysis methods being evaluated.

5.0 VEHICLE CHARACTERIZATION TESTS

5.1 INTRODUCTION

The research program required analytical predictions of test vehicle performance before commencement of the on-track tests. There are two objectives for these predictions: (1) to identify critical areas of performance, so that testing could concentrate on those areas, and (2) to provide further validation of the NUCARS computer model being used to make the predictions.

In order to obtain good predictions, it is essential to have accurate knowledge of the test vehicle's suspension characteristics and modal parameters. This information is best obtained by performing suspension characterization and modal analysis tests.

In previous test programs, the AAR has performed these tests on the VTU and on various jury-rigged, quasi-static test stands. The results from these tests have often been less accurate than desired. In addition, the cost of using the VTU is usually high, making it an undesirable means for routinely obtaining suspension characteristic data.

In order to address the problems of accuracy and cost, the AAR, as part of this research program, developed and tested a new vehicle characterization facility for obtaining suspension and modal data. This facility has become known as the Mini-Shaker Unit (MSU) and was used for performing most of the characterization tests for this project.

One of the difficulties encountered during the Car 1 project was in determining certain suspension characteristics, such as the height of action, from the suspension characterization test data. The greatest difficulties were encountered when trying to analyze the suspension of the trailer load and its interactions with the test vehicle. In many instances these parameters have to be determined by trial and error, until the test results match predictions.

To address these difficulties, the AAR has been developing new "Parameter Identification" computer software to assist in the analysis of vehicle characterization test data. This software is intended to formalize the often haphazard process of converting test results into suspension characteristics that accurately represent the test vehicle. To evaluate the new software's practicality, some of the vehicle characterization test data was analyzed using these new techniques.

5.2 VEHICLE CHARACTERIZATION TEST FACILITY

5.2.1 Vehicle Characterization Tests With The VTU

The VTU was originally designed to perform long term vibration tests on a complete vehicle to simulate the running of a car along actual track. This system has performed well for evaluating lading damage, structural stress levels, and for researching the effects of a variety of track perturbations on vehicle dynamic response. The VTU is ideally suited for performing these tests.

To accomplish these tests, the VTU shakes the whole vehicle including the wheels and trucks. The VTU consists of moving platforms with short sections of "rail" mounted onto them. The vehicle rests on these rails, one axle to each platform. The vehicle is excited by lateral and vertical actuators that move each platform independently. Because the actuators support the entire weight of the vehicle, the actuators need to be large and powerful, with high hydraulic flow rates. These are therefore expensive to operate and maintain, when compared with conventional actuators.

During many previous test programs, including the Car 1 program, vehicle characterization tests were also performed on the VTU. This was accomplished by jury rigging fixtures that would hold the car body stationary while moving the suspension beneath it. Measurements of wheel/rail forces were also required. These were made using the VTU "rails" which had been strain gaged to detect incipient wheel lift during the vibration tests. Subsequent analysis has shown that these are not accurate enough for good characterization results. This jury rigged system, combined with the operational and maintenance expense of the VTU and the less than desired accuracy in the force measurements, led to the conclusion that the VTU is not ideal for performing vehicle characterization tests. The VTU is nonetheless still well suited for the tasks for which it was designed: whole vehicle vibration, track perturbation, and vehicle dynamics tests.

One of the main goals of the Car 2 program was therefore to develop a new vehicle characterization testing facility that would be cheaper to operate and produce better results.

5.2.2 Design of the MSU

To address the problems encountered with the VTU, a newly designed facility was recommended. This new facility was to have the following features:

1. Excitation to the car body, to reduce expense of actuators. Excitation at only one end of the car.
2. Instrumented rails under the wheels to measure vertical and lateral forces. These must be more accurate than the ones used on the VTU. Instrumented rails only at one end of the car.
3. Portability, to allow installation of the test rig at other sites.
4. Simple desk top computer based control system and data acquisition system.

5.2.3 Instrumented Rails

The first items addressed in the design process were the instrumented rails. Several different designs were studied and two existing designs were tested. The first design tested was based on the strain gage arrangement frequently used in measuring wheel/rail forces in the field. This involves mounting strain gages on the base and the web of a standard rail section. In the past, problems have been encountered with the linearity of this arrangement[5,6] and with cross talk between vertical and lateral signals. These problems were confirmed by simple tests in the laboratory.

The second design tested was strain gaged rail sections that had been developed by ENSCO under contract to the FRA. These rail sections were originally designed to be used as part of a sticking brake detector. These also proved to have considerably more cross talk than was desirable; therefore, a new design was developed that was based on a specially machined bar of steel, with pockets machined for mounting strain gages in locations where cross talk between vertical and lateral strains would be minimized. Strain gages were mounted on the top surfaces of the rails, as shown in Figure 3, and on the bottom surfaces of holes machined through the sides of the bars, as shown in Figure 4. Tests on these bars showed greater linearity and less cross talk than any previous design.

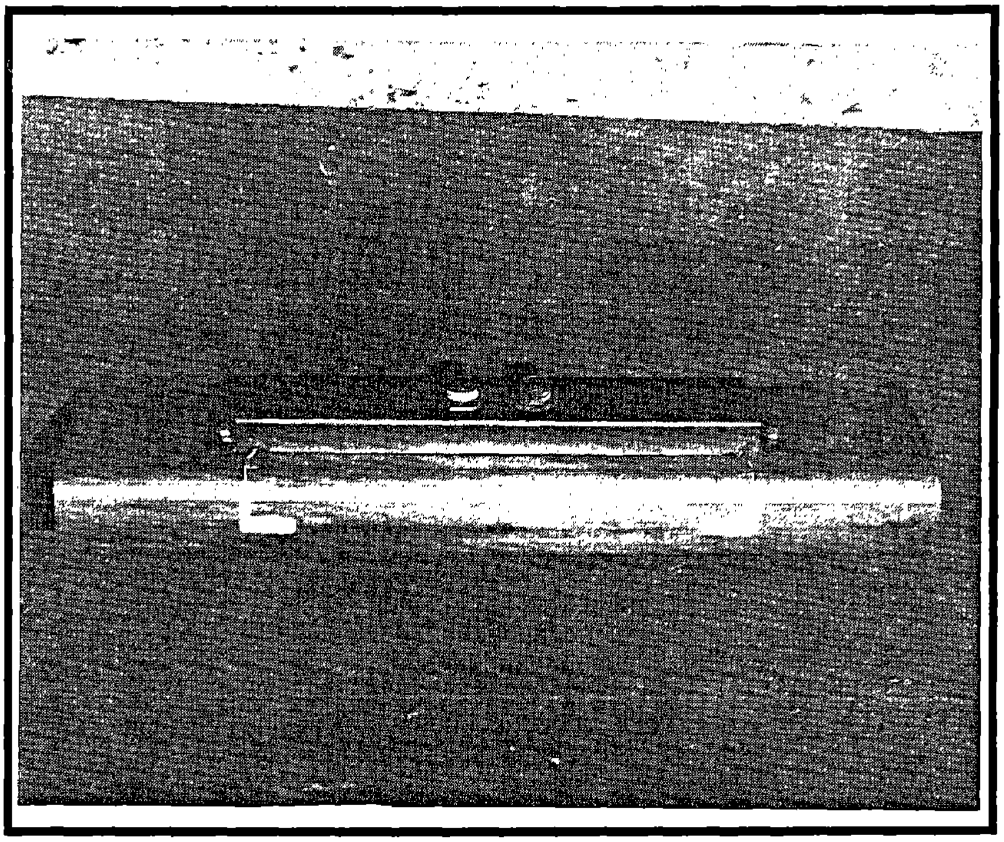

Figure 3. MSU Instrumented Rail, Showing Position of Strain Gages in Pockets on Top of Rail

The rails are mounted to a platform which can be bolted to the concrete floor. Ramp rails are mounted on the ends of the platform which can be aligned with railway tracks. This makes it possible to push the test vehicle into position. The VTU requires lifting the vehicle with cranes.

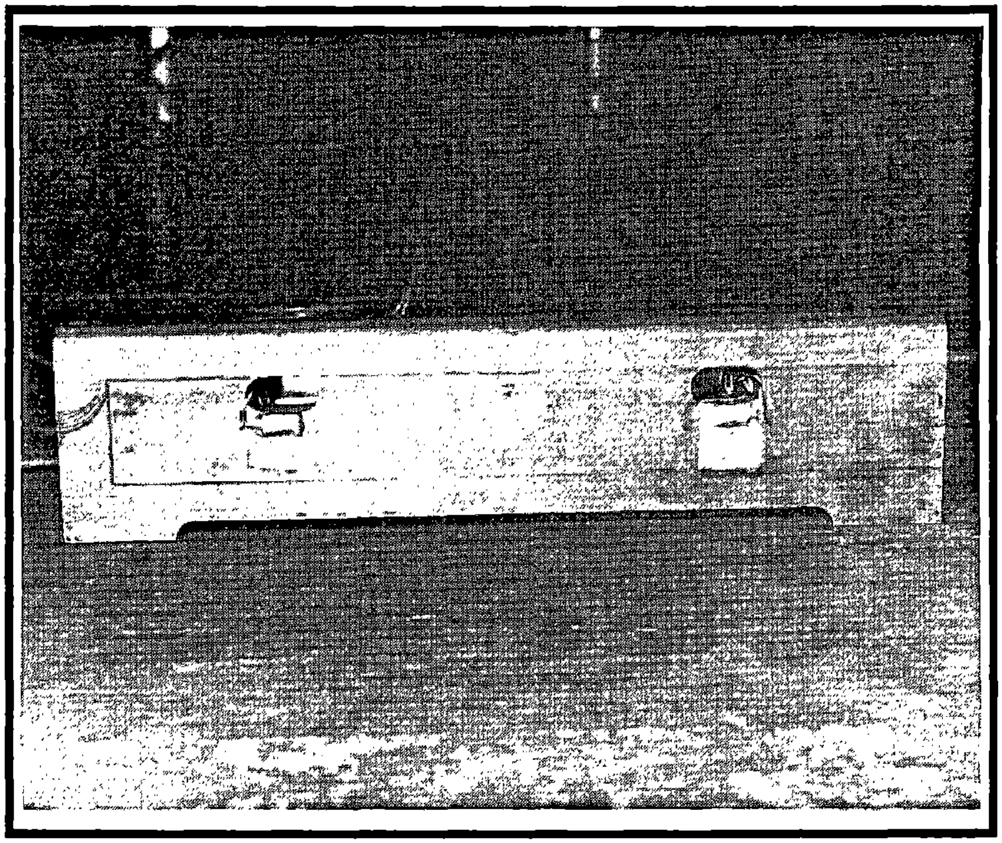

Figure 4. MSU Instrumented Rail Showing Holes for Strain Gages Machined in the Side

5.2.4 Hydraulic System

The basic plan for exciting the vehicle was to connect the vehicle car body to the ground by means of vertical and lateral hydraulic actuators. Because they would not have to support the vehicle, the actuators could be considerably smaller than those used in the VTU.

The ideal mounting position for the vertical actuators would have been to fasten them to the floor and have them run upward to the car body. This could not be done for two reasons:

1. No actuators were available that were short enough to fit between the car and the floor.

2. It was not feasible to excavate pits beside the test location in which to fit longer actuators. These pits would have also required reaction masses in the bottom to react the forces.

The actuators were therefore attached to large concrete blocks, as shown in Figure 5. These blocks had been made as flatcar loads for another research project and were now available to be used as reaction masses. Two 55 kip actuators were used.

The lateral actuator was also mounted to one of the reaction masses, as shown in Figure 6. If no reaction masses were available, a load reaction frame would be constructed and fastened to the floor. A 20 kip lateral actuator was used.

Except for the two 125,000 pound reaction masses, this system is reasonably portable. If taken to other locations, an alternate reaction arrangement, such as mounting the actuators to the floor, would be used.

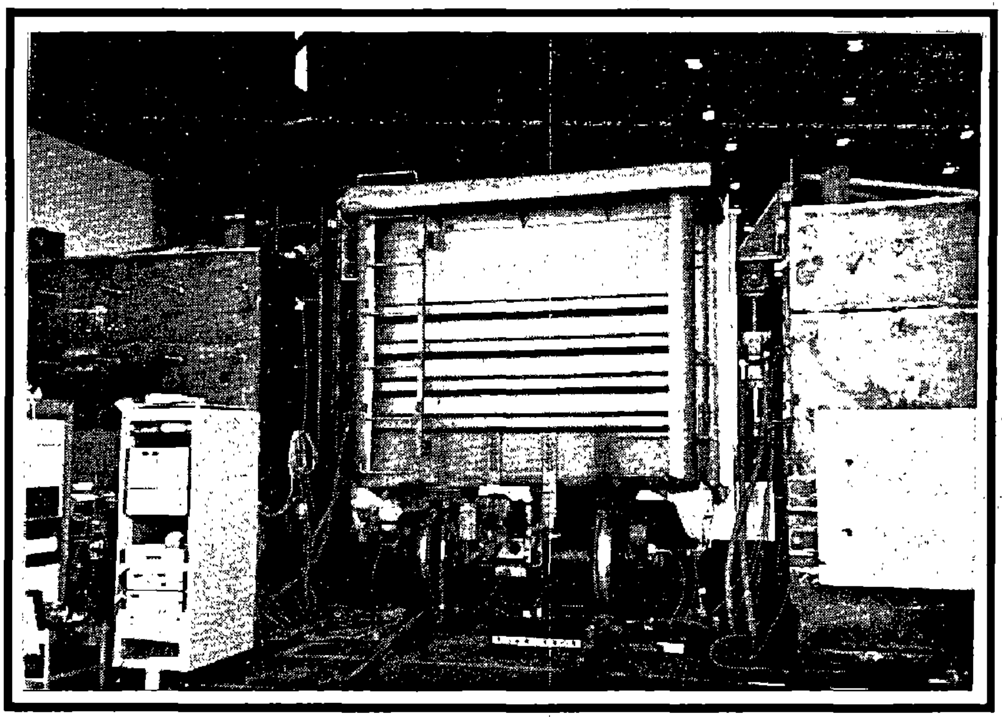

Figure 5. Test Vehicle Installed in MSU Showing Attachment of Hydraulic Actuators

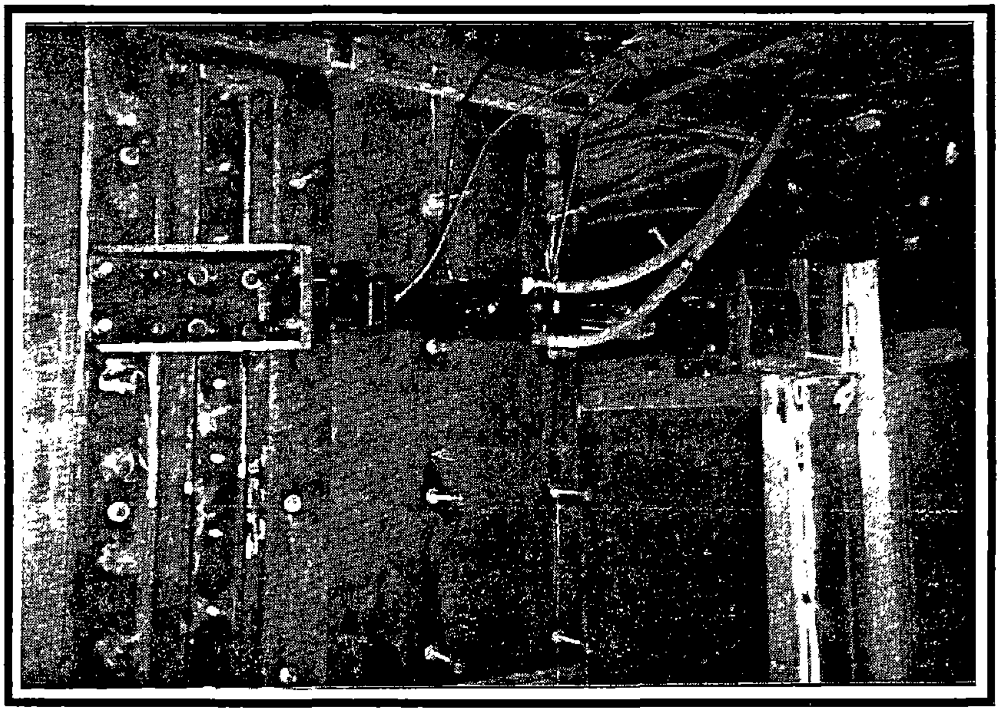

Figure 6. Attachment of MSU Lateral Actuator to Concrete Block and Test Vehicle

5.2.5 Control System and Data Acquisition

Control of the actuators was accomplished using a Hewlett-Packard (HP) 330 desk top computer linked to the hydraulic controllers by a function generator. The same computer was used to acquire the test data during the tests. Figures 7 and 8 show the control and data acquisition system.

A single computer program acted to generate the control signals and collect the test data. A wide range of control functions were possible, including frequency sweeps and constant frequency dwells. Control was by either constant displacement amplitude or constant force amplitude input from the hydraulic actuators.

The data acquisition system consisted of the HP 330 computer linked to a HP 6942A multi-programmer analog-to-digital converter. Digital test data was stored for future analysis on 20 megabyte Bernoulli type removable hard disk cartridges.

Immediate post-test "quick look" data analysis is also possible using the same data acquisition and control software. Time history plots and cross plots of one data channel against another are available to allow quick verification of test results. Frequency domain analysis is also possible. Fast Fourier Transforms (FFTs) can be generated for any data channel. Transfer functions can be calculated between different channels of data.

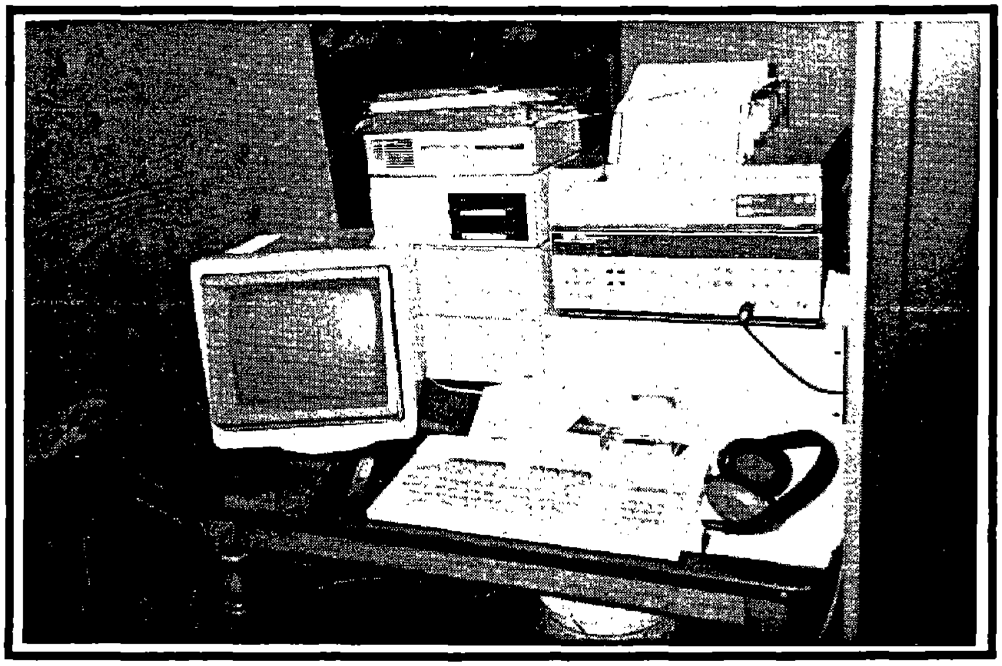

Figure 7. Hewlett-Packard Desk Top Computer System used for MSU Control and Data Acquisition

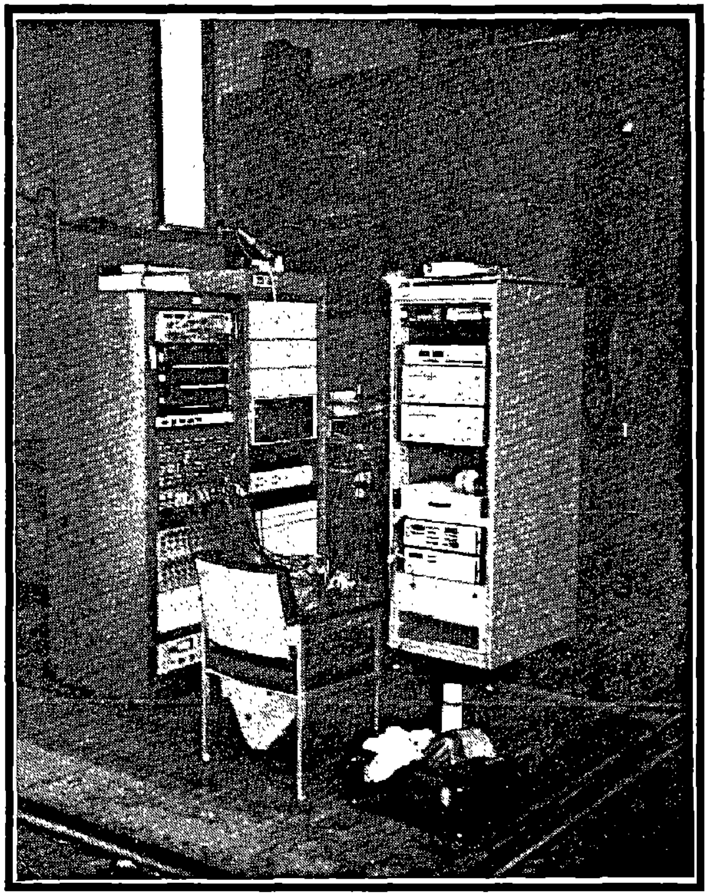

Figure 8. Signal Conditioning and MTS Hydraulic Control System for MSU

5.3 VEHICLE CHARACTERIZATION TESTS

5.3.1 Test Objectives

The objectives of performing vehicle characterization tests were:

1. To measure the suspension characteristics of the test vehicle using the Chapter XI guidelines. The data obtained is to be used in the NUCARS model.
2. To measure modal parameters of the test vehicle for use in the NUCARS model.
3. To validate the new parameter identification software.
4. To evaluate the MSU for its practicality and cost effectiveness in obtaining suspension characteristics and modal test data.

5.3.2 Test Measurements

Test measurements consisted of input forces and displacements of the hydraulic actuators, response displacements measured across the various suspension elements, car body accelerations, and vertical and lateral rail forces. A complete list of instrumentation used is contained in the Test Implementation Plan.[7]

A sign convention consistent with the NUCARS model was chosen for all data. The test vehicle was placed with its A-end over the instrumented rails. When standing facing the B-end of the car, the x-axis was chosen to be longitudinal with positive motion forward. The y-axis was laterally to the left. The z-axis was vertical with positive motion up. Clockwise rotation about the named axes was taken as positive.

Deflections across springs were positive for extension; negative for compression.

5.3.3 Test Procedures

There were four different basic test procedures:

1. Vertical characterization tests
2. Roll characterization tests
3. Lateral characterization tests
4. Body bending mode (modal) tests

5.3.3.1 Vertical Characterizations

The vertical characterization tests were performed two different ways to compare the effectiveness of the different methods. The first method was quasi-static characterization. Both vertical actuators were connected to the car body. The actuators were stroked in-phase at a constant frequency of 0.1 Hz. A variety of runs were made at different amplitudes up to the point at which the truck springs were fully compressed.

These tests were then repeated with the frequency being swept from 0.1 Hz, increasing until the suspension passed through a vertical resonance. At resonance, it was expected that only low force inputs would be required to achieve maximum suspension deflections. Unfortunately, this was not the case. The hydraulic actuators appeared to be flow limited at resonance, causing the input displacements to drop. This prevented achieving full suspension travel at resonance.

5.3.3.2 Roll Characterizations

These tests were similar to the vertical tests except that the vertical actuators were operated out-of-phase. Because of tight clearances between the car body and the two reaction masses, only quasi-static inputs were used. This was to avoid the possibility of the car body striking the actuator support brackets on the reaction masses. During the quasi-static tests, it was possible to monitor the car body roll by eye and keep roll motions under control.

5.3.3.3 Lateral Characterizations

These tests required removal of the vertical actuators. The left side reaction mass was repositioned and a lateral actuator connected between it and the car.

Both quasi-static and resonance type tests were performed in a manner similar to the vertical tests. It was hoped that the lateral tests would excite both lateral and roll suspension resonances. Unfortunately, only a small amount of roll motion occurred during the lateral tests near resonance. This is probably due to the low center of gravity of the loaded car, combined with the lateral actuator being positioned close to the roll center height of the vehicle.

5.3.3.4 Body Bending Mode (Modal) Tests

These tests are performed to identify the following three primary body structural bending modes, and are basically extensions to the other tests.

1. Lateral Bending
2. Vertical Bending
3. Torsion

The vertical bending mode test is performed with the two vertical actuators operating in-phase, as in the vertical characterization tests. Low amplitude input is swept in frequency to pass through the body vertical bending resonance. For this vehicle, it was very easy to determine whether a bending resonance had been achieved because the car structural flexibility made it possible to visually observe the resonances.

The torsion tests were performed in a manner similar to the roll characterization tests. Similarly, the lateral bending tests were an extension of the lateral suspension characterization tests.

5.3.4 Test Results

5.3.4.1 Parameter Identification

One of the objectives of the vehicle characterization tests was to validate a new computer program designed to assist in identifying vehicle parameters. This program, known as Rail Vehicle IDentification (RVID), has been under development for the AAR by Massachusetts Institute of Technology (MIT).[8]

5.3.4.2 Secondary Vertical Suspension

The secondary vertical suspension consists of the main truck springs, with variable rate friction damping provided by friction castings. The damping rate is dependent on the deflection of the control springs with full compression providing the highest friction level. Figure 9 shows the general arrangement of the friction castings in the bolster. As the bolster moves downward the control springs as well as the main springs are compressed, increasing the vertical load on the friction castings. Due to the wedge shape, this increases the lateral load against the side frame and hence increases the vertical friction damping.

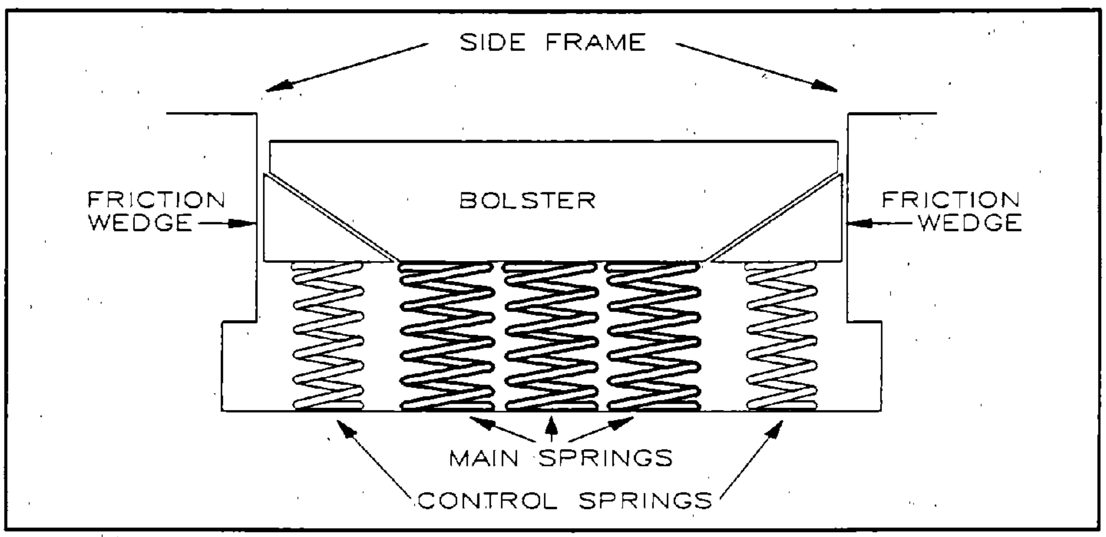

Figure 9. Diagram of Secondary Vertical Suspension (Bolster to Side Frame Connection)

To determine the characteristics of this suspension, data from quasi-static test runs were analyzed. The measured rail vertical forces at each wheel on one side were summed together. These two left and right vertical forces were then plotted against their respective vertical suspension displacements.

RVID was used to assist in identifying the suspension characteristics. Vertical force and displacement data were input to the program, with the controlled variable being the displacement. RVID output estimates of the vertical suspension forces were calculated based on a simple hysteresis loop friction model.

Figures 10 through 15 show the RVID results for the left, right and "average" suspensions. Data shown are plots of the force versus time and force versus displacement for each case. Plotted are the actual test data, the RVID estimates of vehicle response, and the error (difference) between the two. The average suspension is based on the average of the left and right suspension displacements and forces.

The left suspension (Figure 11) can clearly be seen to have greater friction than the right suspension (Figure 13), with the variable friction level also being noticeable. The sudden increase in force at the lower left corner of the force versus displacement plots is an indication that the springs have bottomed at the end of their strokes. The error is greatest at this point, with RVID having difficulty matching the test results when the friction shoes are locked up and not moving.

These characteristics show, as expected, that the friction levels are greatest for the fully compressed position (lower left corner of the plots). The variability between the friction at the fully extended and fully compressed positions is less than expected. This is especially true for the right hand side suspension shown in Figure 13.

The estimated piece-wise linear (PWL) characteristics calculated by RVID for these three suspensions are given in Table 1. The average suspension characteristic was used as input to the NUCARS model for all four vertical suspensions in the preliminary pre-test modeling. The actual left and right characteristics were substituted for the lead truck during the post-test modeling using the actual track geometries (ENSCO measured track).

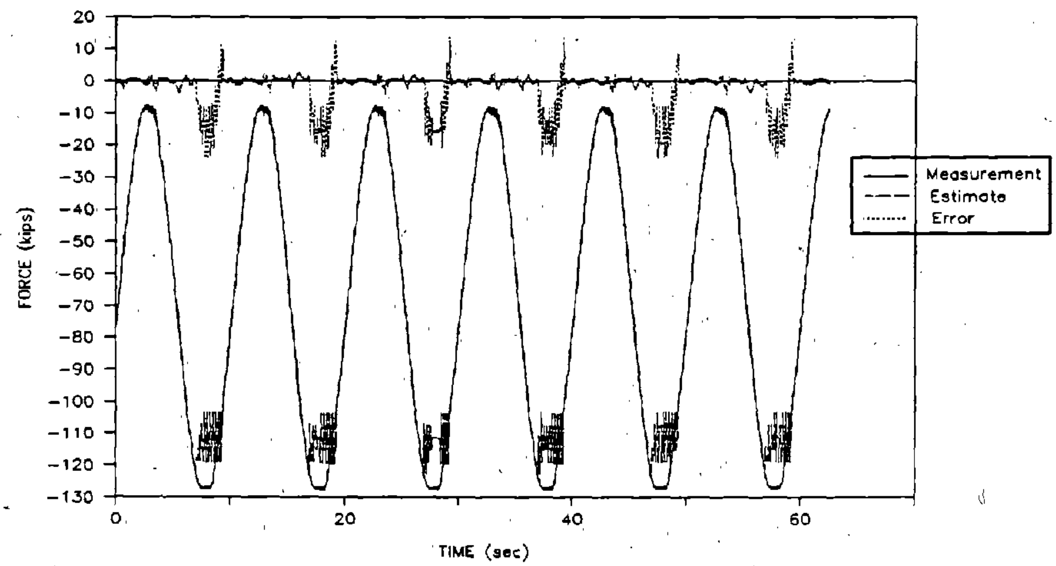

Figure 10. Vertical Force Time History of the Left Side Secondary Vertical Suspension

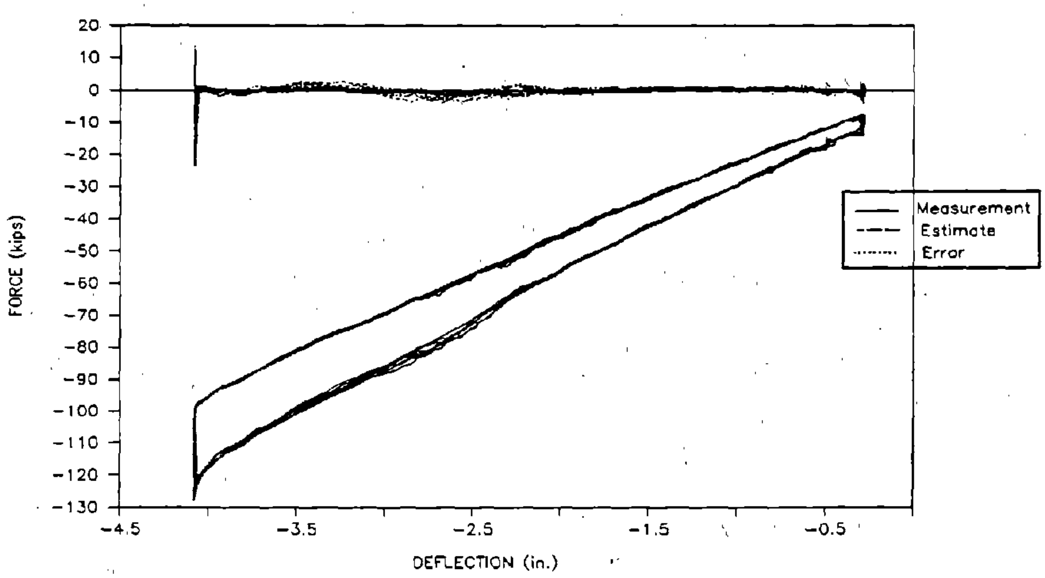

Figure 11. Vertical Force Versus Vertical Displacement Characteristic of the Left Side Secondary Vertical Suspension

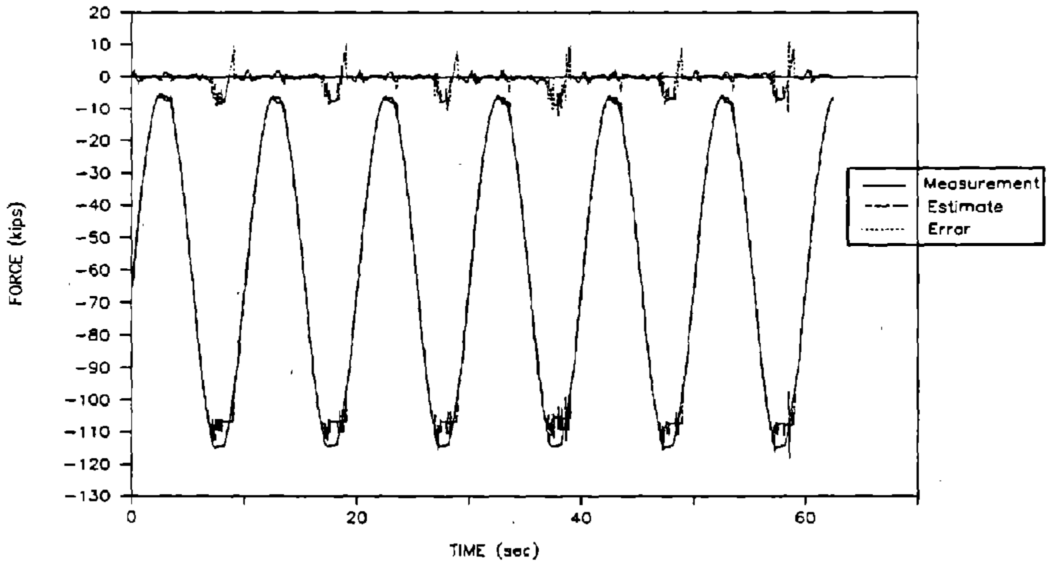

Figure 12. Vertical Force Time History of the Right Side Secondary Vertical Suspension

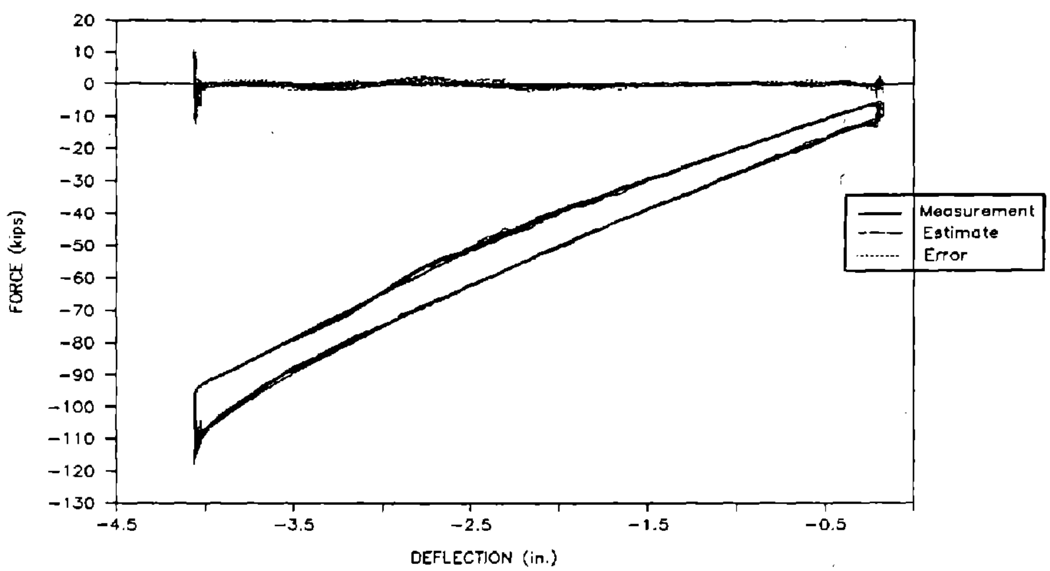

Figure 13. Vertical Force Versus Vertical Displacement Characteristic of the Right Side Secondary Vertical Suspension

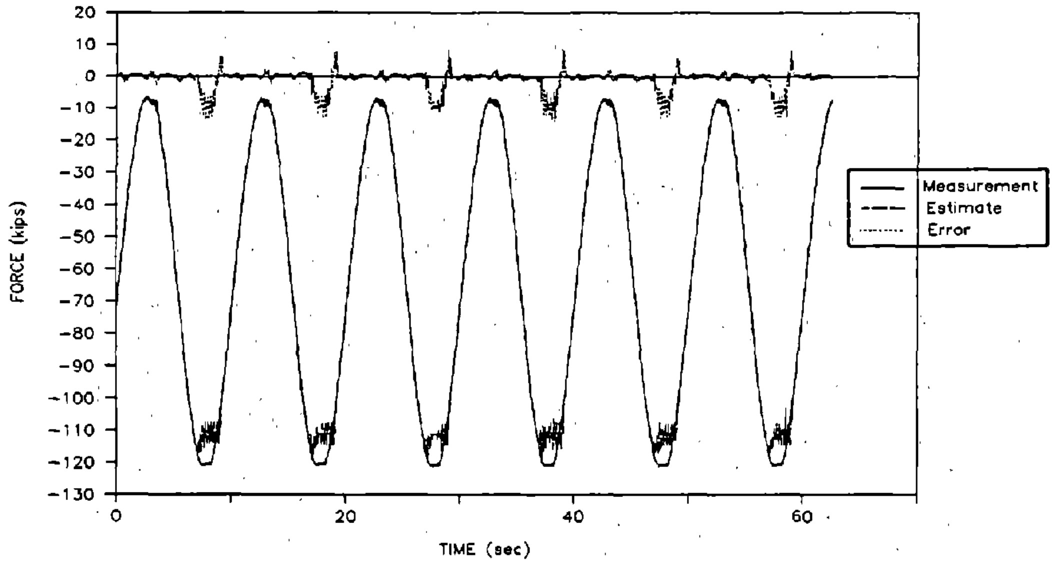

Figure 14. Vertical Force Time History of the "Average" Secondary Vertical Suspension

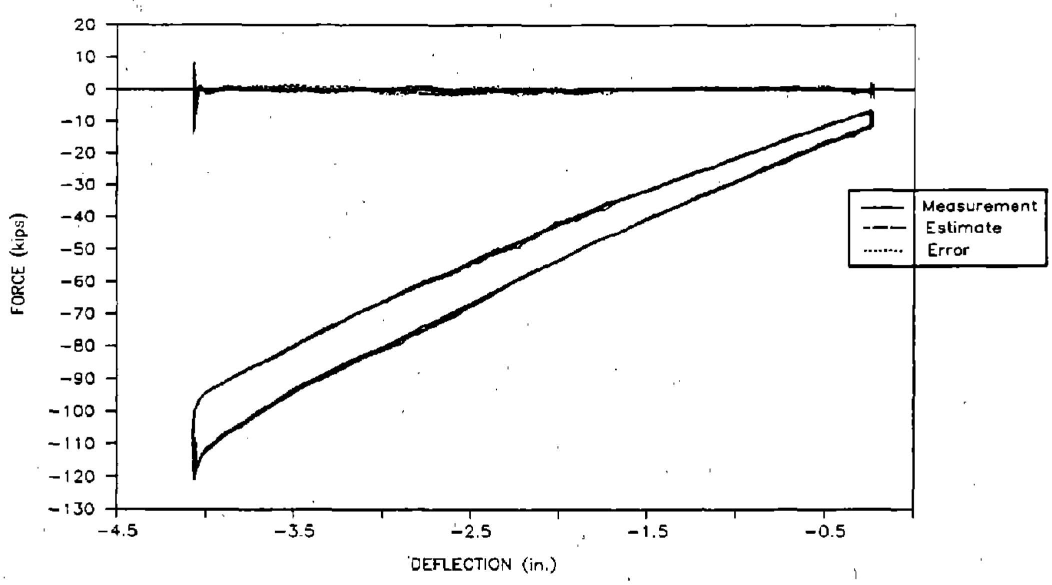

Figure 15. Vertical Force Versus Vertical Displacement Characteristic of the "Average" Secondary Vertical Suspension

Table 1. Estimated Piece-Wise Linear Characteristics for the Secondary Vertical Suspension

	DOWN STROKE			UP STROKE		
	Force (lb)	Displacements (in.)	Number of Break Points	Force (lb)	Displacements (in.)	Number of Break Points
Right	-1.25x10^5	-4.05		-1.3x10^5	-4.05	
	-9.36x10^4	-3.98		-1.07x10^5	-4.01	
	-7.4x10^4	-3.35	5	-8.88x0^4	-3.55	6
	-4.18x10^4	-2.12		-7.98x10^4	-3.23	
	-6.8x10^3	-0.19		-5.19x10^4	-2.05	
				-1.12x10^4	-0.19	
Left	-1.25x10^5	-4.07		-1.3x10^5	-4.07	
	-9.81x10^4	-4.00		-1.13x10^5	-3.94	
	-9.30x10^4	-3.93	6	-9.13x10^4	-3.15	5
	-7.67x10^4	-3.33		-5.28x10^4	-1.91	
	-4.36x10^4	-1.89		-1.26x10^4	-0.3	
	-7.97x10^3	-0.3				
Average	-1.25x10^5	-4.05		-1.3x10^5	-4.05	
	-1.12x10^5	-3.97		-1.11x10^5	-3.89	
	-9.06x10^4	-3.94	5	-1.0x10^5	-3.59	5
	-3.76x10^4	-1.77		-4.02x10^4	-1.46	
	-7.52x10^3	-0.22		-1.15x10^4	-0.22	

Comparisons of these vertical suspension tests were made with data from the roll characterization tests. Although it was not possible to bottom the suspension during roll, Figure 16 shows that the behavior in roll is very similar to vertical bounce.

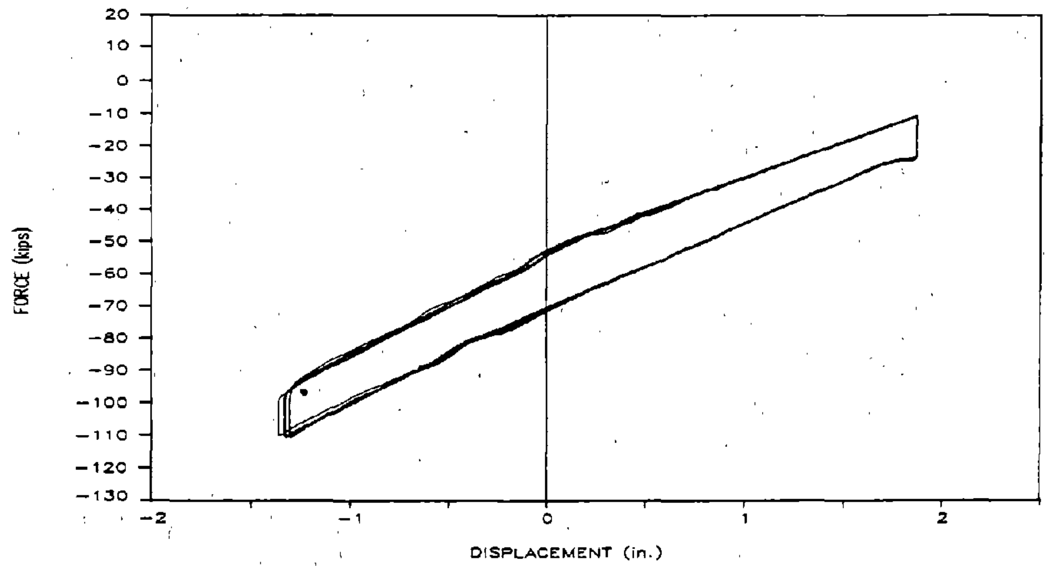

Figure 16. Vertical Force Versus Vertical Deflection Characteristic for the Right Side Secondary Vertical Suspension During Roll Input

5.3.4.3 Primary Lateral Suspension

The primary suspension is provided by rubber shear pads mounted between the axle bearing adaptors and the truck side frames. These are circular and therefore expected to have uniform stiffness in the lateral and longitudinal directions. The manufacturer's specifications for the vertical stiffness of these pads is very high; 7,500,000 lb/in. This stiffness is so high as to have no influence on the dynamic performance of the vehicle. Therefore no attempts were made to characterize the primary suspension in the vertical direction.

The lateral suspension was characterized from the lateral suspension tests. RVID was used to assist in identifying the shear pad characteristics. Due to the nature of the test arrangement, it was not possible to isolate the forces being transmitted through each individual shear pad. The axles act as solid links connecting the left and right sides while the side frames transmit forces from lead to trail axle. Therefore, although the individual displacements across the shear pads were measured with LVDTs, the results had to be averaged to develop the average characteristic, as shown in Figures 17 and 18.

The sharp upturn and downturn at the two ends of the force versus deflection plot (Figure 18) indicate the limits of travel as the bearing adaptor strikes the stops in the pedestal jaws. The intermediate portion of the plot shows an average stiffness of 38,095 lb/in, 19 percent stiffer than the manufacturer's specification of 32,000 lb/in. As can be seen in the time history plot, this data was taken from the first part of a frequency sweep; in this case the data runs from 0.1 Hz to 0.5 Hz. The match between estimated and actual test results is good with relatively small errors. The RVID program is very useful in developing these average results, as the software automatically assigns equal weight to the four individual inputs while performing its optimization calculations.

The resultant PWL data for the primary lateral suspension were also used to describe the longitudinal primary suspension, with adjustments as needed to reflect the longitudinal clearances between the bearing adaptor and side frame at the pedestal jaws.

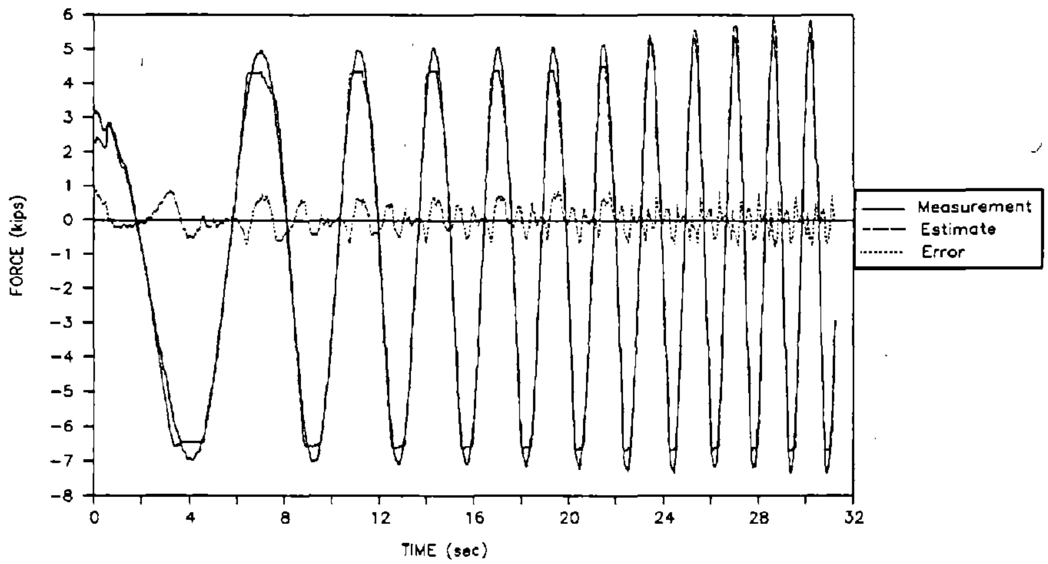

Figure 17. Lateral Force Time History of the Average Primary Lateral Shear Pad Suspension

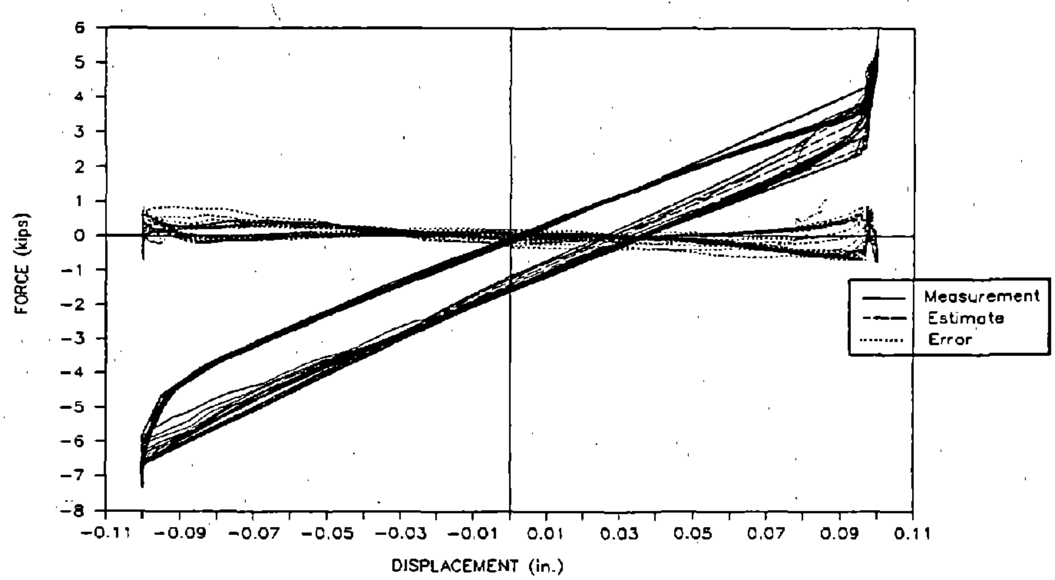

Figure 18. Lateral Force Versus Lateral Deflection Characteristic of the Average Primary Lateral Shear Pad Suspension

5.3.4.4 Secondary Lateral Suspension

The secondary lateral suspension is between the bolster and side frames. This is normally dominated by the friction snubbers and is expected to vary somewhat with the vertical deflection of the secondary suspension (main truck springs).

Again it was not possible to isolate the lateral forces being transmitted through the left and right secondary lateral suspensions. Therefore, only an average lateral suspension can be examined. Figure 19 shows a typical force versus deflection plot for the lateral suspension.

This shows a very wide friction band of about 20,000 pounds. The stiffness is approximately 18,000 lb/in. The upturn and downturn at the two ends again indicate that the full extent of lateral travel has been achieved with the bolster gibs striking the side frames. Figure 18 is asymmetric with greater negative travel than positive travel. This probably indicates that the rest position of the suspension is not centered between the gib stops.

At this time the RVID program has not successfully characterized this suspension. Therefore data for input to NUCARS was "eyeball estimated" from this plot. For the purposes of the pre-test modeling, the asymmetry was removed from the data.

It is believed that under dynamic conditions the friction level is much reduced from that shown in this data. Unfortunately, it was not possible to induce a lateral resonance in this suspension during the MSU tests so no higher frequency test data is available to test this hypothesis. Previous efforts at modeling vehicles with three-piece trucks had used a friction band width of only 10,000 pounds, and other test results supported this value. This is half of the measured friction level. It was therefore decided to halve the measured value for input to the NUCARS model.

The measured data is also only valid for a loaded vehicle. The characteristic is expected to change for the empty car. Therefore, the loaded car values used in the model were halved for pre-test modeling of the empty car. This lower value is based on the expectation that the variable friction snubbers would reduce the lateral suspension friction damping when the car was empty.

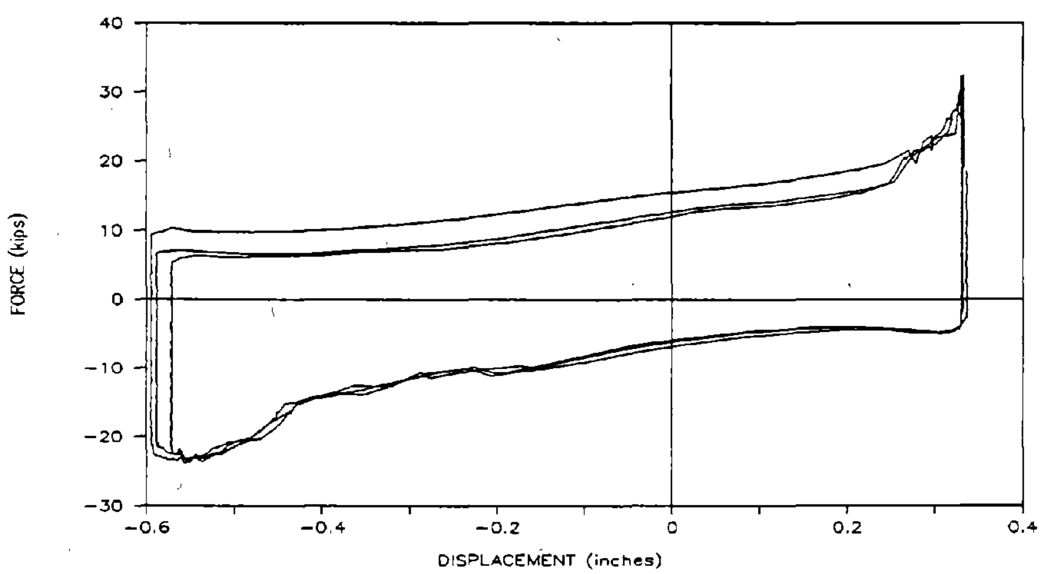

Figure 19. Secondary Lateral Suspension Lateral Force Versus Lateral Deflection Characteristic

5.3.4.5 Roll Suspension Characterization

The primary roll suspension is provided by the main vertical springs (secondary vertical suspension) working out of phase. Characterization of this suspension is unnecessary, having been achieved by characterizing the secondary vertical suspension. The secondary roll suspension acts between the body and truck bolster. It is provided by the action of the body center plate rocking in the bolster center bowl combined with the vertical deflections of the constant contact side bearings.

During the MSU tests, very little roll motion occurred between the body and truck bolster. Therefore, the pre-test NUCARS modeling made use of a standard data set frequently used by the AAR to represent a "typical" constant contact side bearing arrangement. This is based on data collected during previous tests of constant contact side bearings in the laboratory.

Subsequent analysis of the MSU test data provided a secondary roll characteristic, as shown in Figure 20. The test data are overlaid with the theoretical characteristic used in the pre-test NUCARS modeling. It is clear that they are completely different.

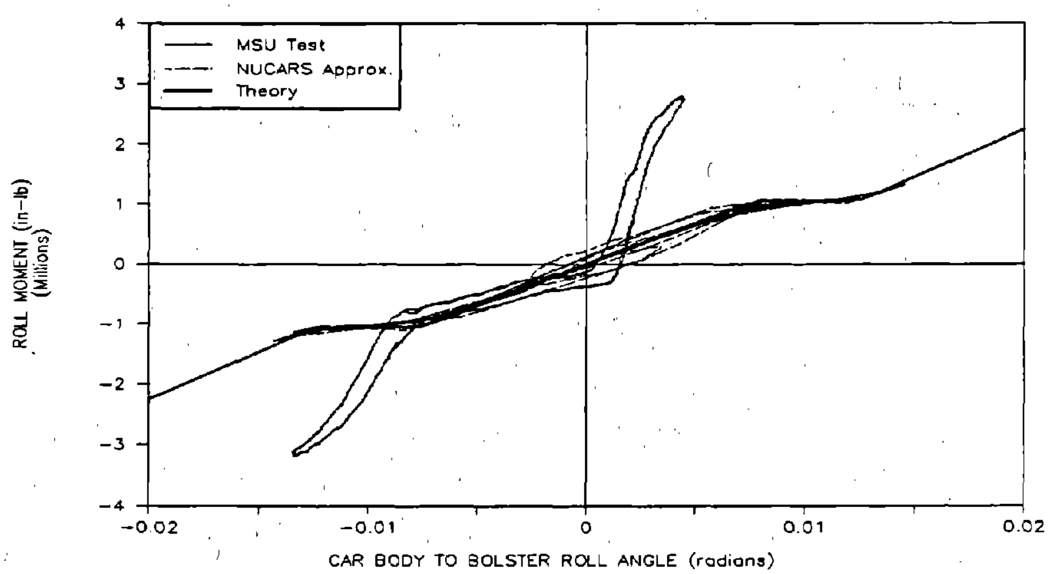

Figure 20. Roll Moment Versus Roll Angle Characteristic of the Secondary Roll Suspension

The slope of a force versus deflection plot indicates the stiffness of the suspension. The center sections of the two data sets show different stiffnesses. On the theoretical characteristic the center section has a low stiffness at each end which represents the car tipping on the corner of the center plate. The center section itself does not represent a true stiffness, but is instead a "leading edge slope" which is introduced as a mathematical convenience to represent what should theoretically be an infinitely steep slope. At even greater roll angles the stiffness increases as the side bearings are compressed and the vehicle begins to tip on the side bearings.

This is not matched by the measured characteristic, which shows instead a true stiffness in the center section which gets even stiffer at the ends. These stiffer end sections then taper to lower stiffness. Unfortunately roll displacements remained small so a complete comparison with the theoretical characteristic is not possible.

It is clear that a better understanding of the secondary roll suspension is needed. Current explanations of the behavior indicate that the center plate to center bowl interface is not flat with sharp corners to tip on, as theoretically modeled. Instead it is believed that the center must be rounded relative to the center bowl so that some rocking action occurs before actual tipping on the corners occurs. This combined with the fact that the corners are probably rounded off would give a completely different characteristic.

The characteristics of the constant contact side bearings may also be misunderstood. Compression of the side bearings would thus be misrepresented in the NUCARS model. An estimate of the roll characteristic from the MSU data was made for use in the post-test real track NUCARS modeling.

5.4 QUASI-STATIC (AIR BEARING) TRUCK ROTATION TESTS

5.4.1 Introduction

It is not possible to measure all truck suspension parameters on the MSU. These are mostly parameters that involve rotational and longitudinal motions of various suspension components. For this vehicle the following parameters needed to be measured:

1. Truck (center bowl) rotational breakaway torque
2. Inter-axle bending (primary shear pad longitudinal stiffness)
3. Inter-axle shear (truck warp stiffness)
4. Axle alignment

The general method for measuring these parameters involves floating the end of the car to be tested on an air table. This eliminates the friction between the truck and the ground. The opposite end of the car is jacked up so that the body remains level. The car body is restrained with chains to prevent it from moving. Hand operated hydraulic actuators are connected at appropriate locations to rotate the truck, move the axles, etc., as required for the particular parameter being measured.

Load cells are mounted in series with the actuators to measure the applied loads. Displacements of the various suspension components are measured with LVDTs or string potentiometers. All data were collected and digitally recorded using an HP 9826 desk top computer.

Tests on the PSMX 111 test car were performed in the Urban Rail Building (URB) at the TTC. Existing fixtures were used to react the various hydraulic loads applied. Tests were performed only on the A-end of the car, with the results being assumed to be similar for the B-end. The tests were performed with the car loaded. For some of the tests, an empty car was simulated by jacking the car up 2.25 inches until the truck springs were extended to the height expected for an empty car. At this point the weight of the load was transferred to the jacks, and the trucks carried only the weight of an empty car.

Figure 21 shows the PSMX 111 test car in position for the inter-axle shear test.

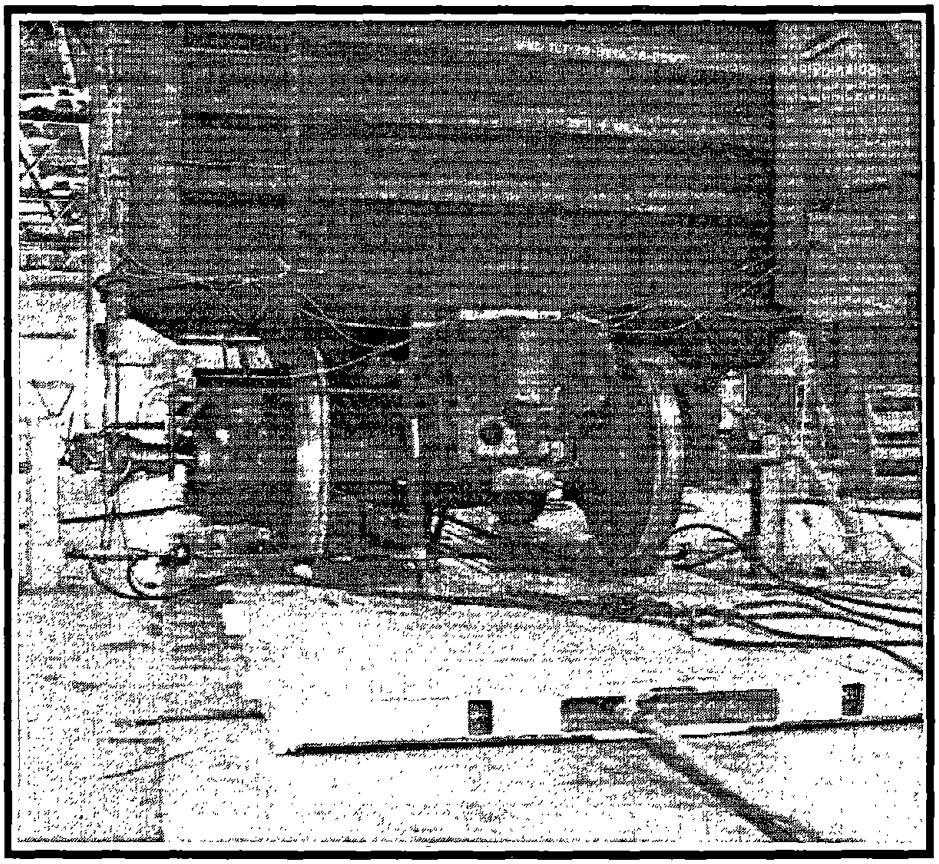

Figure 21. PSMX 111 Test Vehicle in Position on Air Tables for Inter-axle Shear Tests

5.4.2 Truck Rotation

Truck rotation tests are performed to measure the breakaway torque between truck bolster and car body. The breakaway torque is defined as the moment required to rotate the truck freely relative to the car body, overcoming the friction in the center bowl and side bearings.

To perform the test, one truck of the car was floated on a single air table. Actuators were attached at diagonally opposite corners of the air table and connected to reaction frames attached to the floor. When the actuators were operated they applied a moment to the air table and the truck, causing them to rotate relative to the car. Two string potentiometers were mounted between the car body and the truck bolster to measure the rotation.

When performing these tests, the truck typically moves in a series of rotational jerks, as illustrated in Figure 22. This figure shows the applied moment plotted against the rotational angle of the truck. The moment can be seen to build up to a certain breakaway level and then suddenly jerk into motion. This repeats several times as the motion temporarily relieves the moment, and it builds up again to the breakaway level.

Tests were performed with the constant contact side bearings installed and removed to measure their contribution to the overall breakaway torque. Both loaded and empty conditions were tested. Three runs were performed in each condition, and the results were averaged for final values of breakaway torque (Table 2).

Table 2. Lightweight Car 2 Truck Rotational Breakaway Torque

Test Condition	Side Bearing Condition	Breakaway Torque (lb-ft)
Loaded	Installed	16.5×10^3
Loaded	Removed	15.0×10^3
Empty	Installed	3.0×10^3
Empty	Removed	1.0×10^3

Results are as expected, with the side bearings appearing to contribute 2000 lb-ft to 3000 lb-ft of torque to the overall truck rotational moment. As expected, the loaded car required a much larger moment to rotate the truck than the empty car.

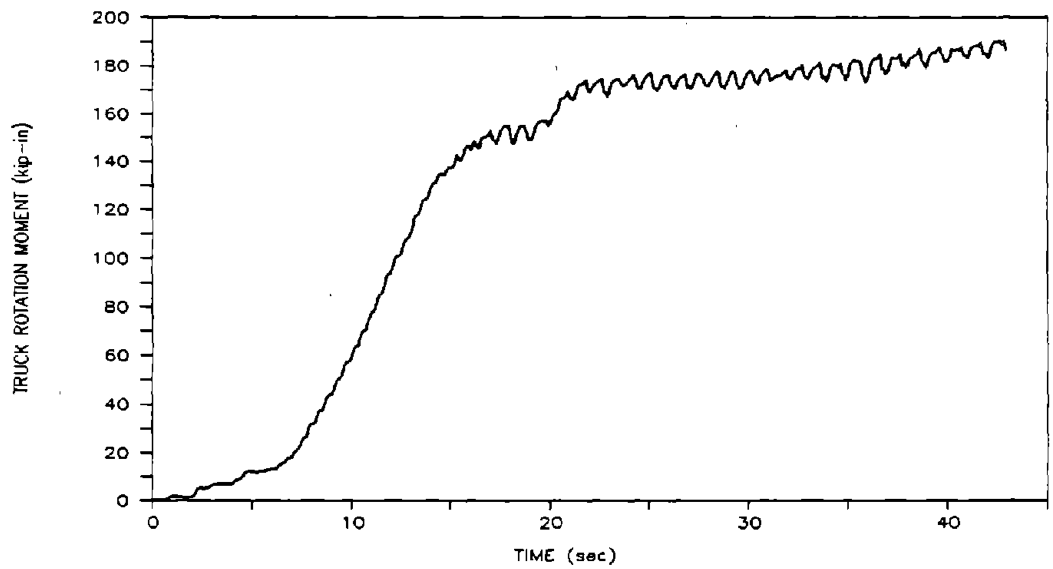

Figure 22. Example Time History of Truck Rotation Moment during Truck Rotation Test

5.4.3 Inter-axle Bending and Axle Spreading

These tests were performed to measure the longitudinal stiffness of the primary rubber shear pads located between the axle bearing adaptors and the truck side frames. Due to the symmetric design of the shear pads, it is expected that the results of this test will be similar to the dynamic lateral stiffness measurements made with the MSU.

The car was floated using two air tables, one under each axle of the A-end truck. Special end caps with extension rods were mounted to the axle ends. Hydraulic actuators were attached between the axle ends using these end caps, with one actuator on each side of the truck. The actuators acted to pull the axles together. Three different combinations of pulling were used: pulling on both sides simultaneously, and pulling on each side individually. Attempts were made to push the axles apart but this caused the actuator assembly to buckle.

LVDTs were mounted between bearing adaptors and the side frames to measure pad deflections. Three runs were made in each of the three combinations. The stiffness of each pad was calculated and averaged for all the runs. Table 3 lists the results. The average value of 27.7×10^3 lb/in is 13 percent less than the manufacturer's theoretical value of 32.0×10^3 lb/in, and is within the manufacturer's tolerance of 15 percent. This result is in conflict with the MSU test results (Section 5.3.4.3) of 38.1×10^3 lb/in. The shear pads are circular and were expected to have uniform characteristics for both lateral and longitudinal deflections. The MSU tests measured the lateral deflections while these tests measured the longitudinal deflections combined with a small amount of torsion. It is possible that the different loading conditions and variations in the mounting may account for the differences between the two sets of test results and the manufacturer's figures.

Past experience had shown large margins of error for some of the air table test measurements. Therefore more faith was put in the MSU test results so for the purposes of the pre-test modeling, the value of 38.1×10^3 lb/in was chosen.

Table 3. Lightweight Car 2 Primary Shear Pad Longitudinal Stiffness

Pad Location	Stiffness (lb/in)
Lead Left	28.1×10^3
Lead Right	26.8×10^3
Trail Left	25.8×10^3
Trail Right	30.3×10^3
AVERAGE	27.7×10^3

5.4.4 Inter-axle Shear

The inter-axle shear tests are performed to measure the warp (lozenging, tramming) stiffness of the truck. This is the combined rotational stiffnesses (around the vertical axis) between the bolster and side frame, and between the bearing adaptor and side frame (primary rubber shear pad). When modeling a truck, for convenience, these combined stiffnesses are usually lumped together as warp stiffness.

To measure warp stiffness, the car was mounted on two air tables, one for each axle of the A-end truck. Reaction frames were mounted to the floor, one on each side of the car, in line with the truck bolster. One hydraulic actuator was attached from each reaction frame to one of the air tables in such a way as to be in line with the truck bolster. This is illustrated in Figures 21 and 23. The actuators were operated to pull in opposite directions, thus pulling one axle to the left and one to the right, shearing the axles relative to each other. This action warps the truck, causing the side frames to rotate relative to the bolster.

A string potentiometer was attached between the two air tables to measure the axle shear displacement. LVDTs were mounted between bearing adaptors and side frames to measure the primary rubber shear pad lateral deflections. From these measurements, the warp rotations of the side frame relative to the bolster could be calculated.

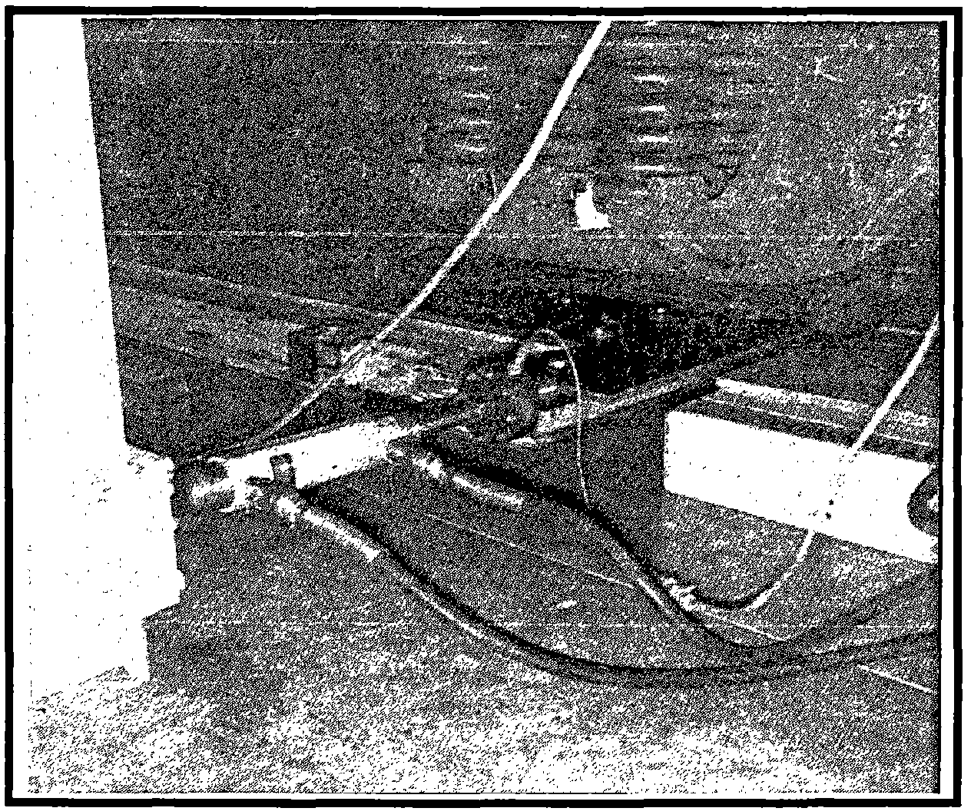

Figure 23. Lateral Actuator Position for Inter-axle Shear Tests

An example plot of applied moment plotted versus warp rotation angle is shown in Figure 24. This apparently has a two stage characteristic which has increasing stiffness for large warp angles. Three runs were made, and the results averaged together to give first and second stage warp stiffnesses of 10.98×10^6 and 15.14×10^6 lb-in/radian, per side of each truck. A friction band of 60.0×10^3 lb-in was estimated from the plots by assuming that the initial rise represents one side of the friction band.

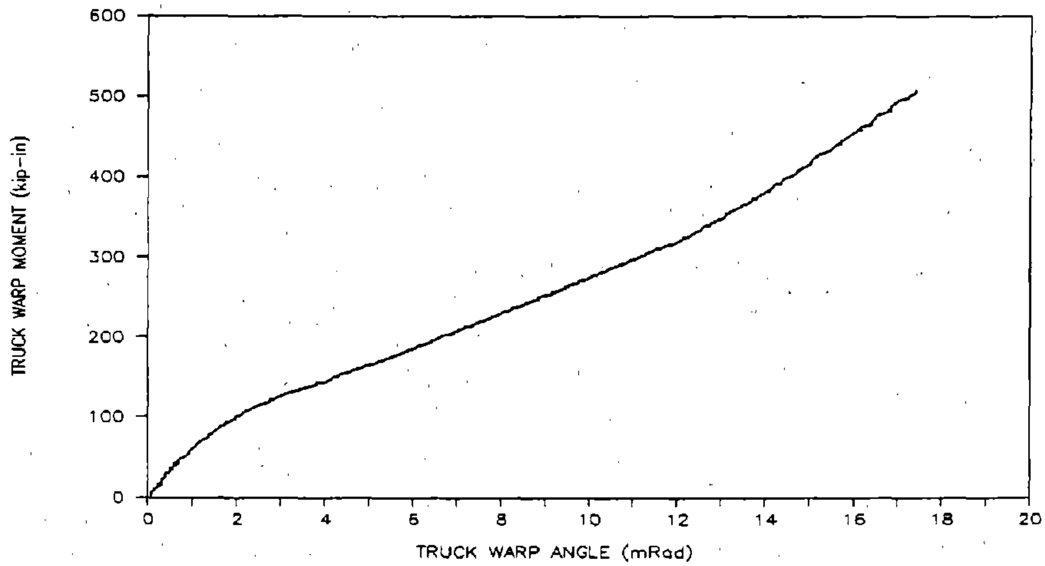

Figure 24. Truck Warp Moment Versus Warp Angle Characteristic

5.4.5 Axle Alignment

Axle alignment measurements are made to determine the natural resting position of the axles when all external alignment forces are removed, except vertical load. Axle misalignments can have a large effect on the dynamic performance of railroad vehicles. Relatively small misalignments on normal three-piece trucks have been shown to cause increased wheel/rail wear and rolling resistance. In order to correctly model this vehicle, the static alignments must be measured.

The vehicle was lifted on the air tables, with one table under each axle of the A-end truck. With all wheel/rail friction forces removed, the axles are now free to take up their natural alignment. The air tables were then gently deflated and the car allowed to settle on the ground with the axles holding their alignment.

Four machinist's scales were mounted perpendicular to the wheel rims on one side of the truck with the scales extending laterally from the sides of the wheels. Figure 25 is a diagram of the measurement scheme. An optical transit was positioned, as shown in the figure, so that a line of sight established dimensions TA2 and LA1 to be the same (+/- 0.01 inch). This line of sight established the datum position. By sighting along the datum line dimensions, TA1 and LA2 were determined.

Axle alignments were calculated using these dimensions from the formulas given in Figure 25. From the results tabulated in Table 4, it can be seen that the axles are very close to being parallel, but the truck is slightly warped.

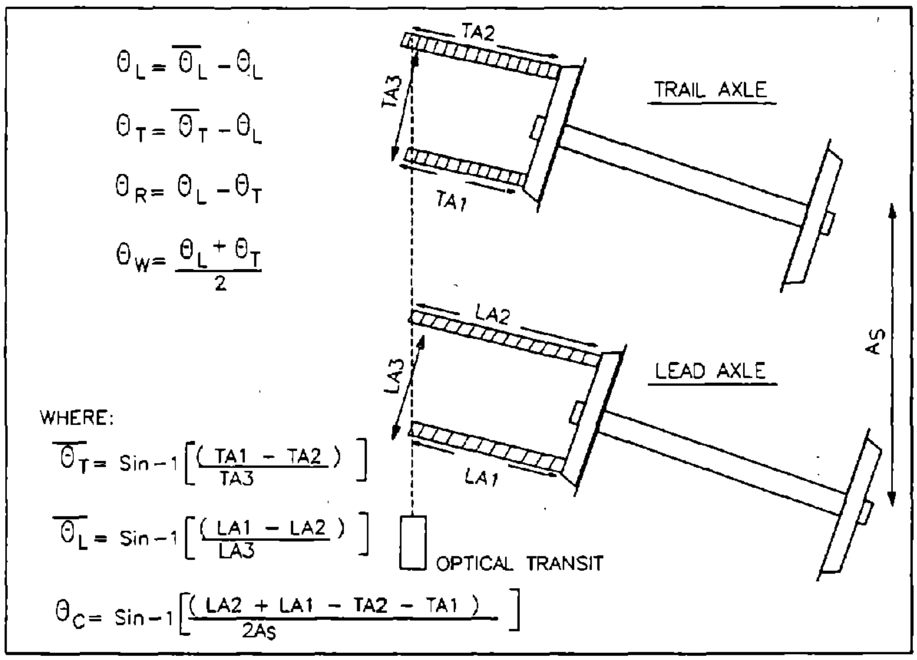

Figure 25. Diagram Showing Schematic for Determining Axle Alignment Measurement

Table 4. Lightweight Car 2 Axle Misalignments

Misalignments		(milliradians)	(degrees)
$\theta_{(L)}$	Lead Axle Misalignment	-1.8	-0.103
$\theta_{(T)}$	Trail Axle Misalignment	-2.0	-0.115
$\theta_{(R)}$	Radial Misalignment	0.2	0.011
$\theta_{(W)}$	Warp (Shear) Misalignment	-1.9	-0.109

6.0 PRELIMINARY NUCARS MODEL PREDICTIONS

6.1 NUCARS INPUT DATA

The input data for the NUCARS modeling consists of:

- Vehicle and suspension mass and inertial parameters
- Vehicle dimensional data
- Suspension characteristics (stiffness, damping and location)
- Wheel/rail profile geometry
- Input track geometry

Data for the first three of these was obtained by direct measurement from the vehicle characterization tests or from manufacturers' specifications. The wheel/rail profile geometry and input track geometry were theoretical formulations.

The vehicle and suspension masses were determined by weighing the car and the various components. From these masses the various rotational inertias were calculated based on the physical geometry of the parts and the masses previously measured. Most of the suspension characteristics used as input were determined from either the MSU tests or the Air Table tests. The characteristics of the roll connection between car body and truck bolster were determined from manufacturers' data as the test data was not analyzed before the NUCARS modeling efforts began. For the same reason the pre-test modeling did not include the car body flexible mode parameters.

NUCARS system files for the empty and loaded car, resulting from the characterization tests are shown in Tables 1 and 2 of Appendix 2. The wheel/rail profile geometry used was a theoretical CN-Heumann profile wheel on a theoretical new AREA 136-pound rail. This profile and the required theoretical Chapter XI track geometries used are defined in the NUCARS data files. The exact formulation of each Chapter XI test zone is given in the following sections.

6.2 NUCARS ANALYSES

In general, the analyses performed are all those required by Chapter XI. In some conditions extra analyses were performed to gain a greater understanding of the vehicle's behavior. The results of all analyses were compared with the Chapter XI safety criteria as then written, which are briefly summarized in Table 5. These criteria have been updated since the pre-test predictions and track tests were completed. The revised criteria are summarized in Table 7, Section 7.

Table 5. AAR Chapter XI Criteria for Assessing the Requirements for Field Service (as Used for the Pre-test Analyses)

REGIME	SECTION	CRITERION	LIMITING VALUE
Hunting (empty)	11.5.2	minimum critical speed (mph)	70
		maximum lateral acceleration (g)	1.0
		maximum sum L/V axle	1.3*
Constant curving (empty & loaded)	11.5.3	maximum wheel L/V	0.8
		or maximum sum L/V axle	1.3
Spiral (empty & loaded)	11.5.4	minimum vertical load (percent)	10**
		maximum wheel L/V	0.8
Twist, Roll (empty & loaded)	11.6.2	maximum roll (deg) ***	6
		maximum sum L/V axle	1.3
		minimum vertical load (percent)	10**
Pitch, Bounce (loaded)	11.6.3	minimum vertical load (percent)	10**
Yaw, Sway (loaded)	11.6.4	maximum L/V truck side	0.6*
		maximum sum L/V axle	1.3*
Dynamic curving (loaded)	11.6.5	maximum wheel L/V	0.8*
		or maximum sum L/V axle	1.3*
		maximum roll (deg)**	6
		minimum vertical load (percent)	10**
Vertical curve	11.7.2	To Be Determined	
Horizontal curve	11.7.3	To Be Determined	

*Not to exceed indicated value for a period greater than 50 milliseconds per exceedence

**Not to fall below indicated value for a period greater than 50 milliseconds per exceedence

***Peak-to-peak

6.3 LATERAL STABILITY ON UNPERTURBED TRACK (HUNTING)

NUCARS predictions were made to analyze the tendency of the vehicle to develop sustained lateral oscillations of the axle between the two rails, known as hunting. Analyses were performed as required by Chapter XI for tangent track and 50-minute-curved track with 6 inches of superelevation. This curvature was chosen as it represents the actual curved track hunting test section at the TTC.

Normally it is expected that only empty vehicles will exhibit a tendency to hunt, so Chapter XI only requires analyses in the empty condition. Previous tests with a vehicle using the same design of truck indicated that there might be a possibility of lateral instability with the loaded car so analyses were made for both loaded and empty car.

The general method for simulating hunting involves setting the vehicle to run on the appropriate curved or tangent track. A single lateral perturbation is introduced into the track to induce lateral oscillations of the wheel sets. If these lateral oscillations are sustained or grow in magnitude as the vehicle progresses down the track, hunting is occurring. If the oscillations die away, the vehicle is stable.

6.3.1 Empty Tangent Track Hunting

The predictions for the empty vehicle show a definite tendency to hunt, with sustained oscillations being evident at 55 mph. Figure 26 shows the Chapter XI limiting criteria for car body lateral accelerations of 1.0 g peak-to-peak being achieved at 57.5 mph. These oscillations are sustained for more than 20 seconds. At 65 mph the limit of any single peak-to-peak oscillation exceeding 1.5 g is also reached. By 70 mph, it is predicted that the vehicle derails due to excessive lateral and yaw motion of the axles.

Figure 27 shows that the axle sum L/V ratios do not exceed 1.2 for speeds up to 70 mph, just within the Chapter XI limiting criteria. Although no Chapter XI limit is set for individual wheel L/V for hunting, the usual limit of 0.8 is exceeded for both left and right wheels on all axles. This is illustrated for axle 1 in Figure 28 and indicates that the axle sum L/V criterion taken by itself may not be an adequate indicator of safe vehicle performance. In most cases the axle sum L/V provides a less conservative prediction of approaching derailment, especially when the wheel set angle of attack relative to the rails is low. In this instance, however, the angles of attack exceed 7 milliradians, and derailment was predicted to occur at 70 mph.

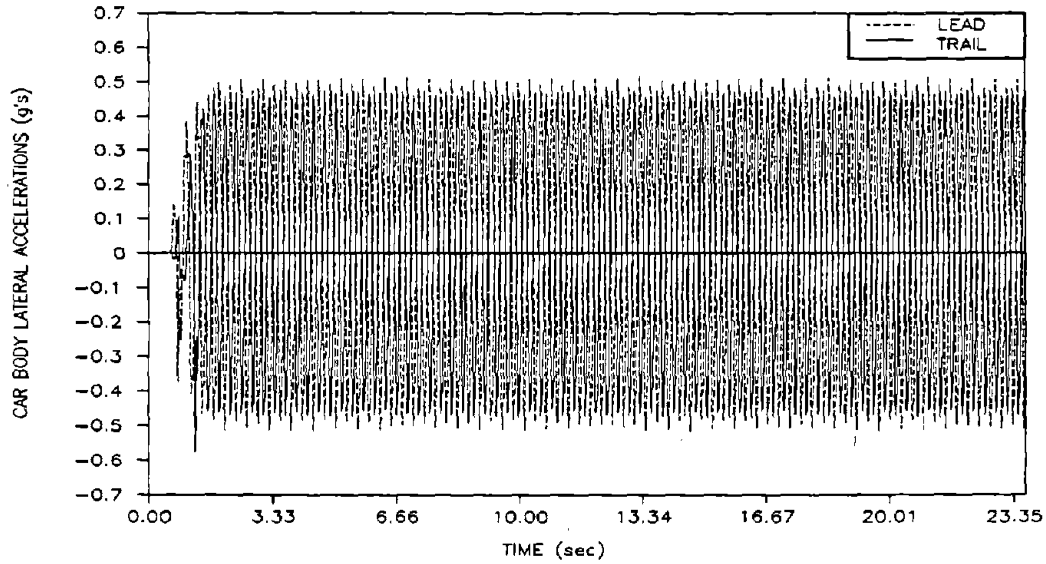

Figure 26. Pre-test Predictions of Lateral Car Body Acceleration for the Empty Car While Hunting at a Speed of 57.5 mph Tangent Track

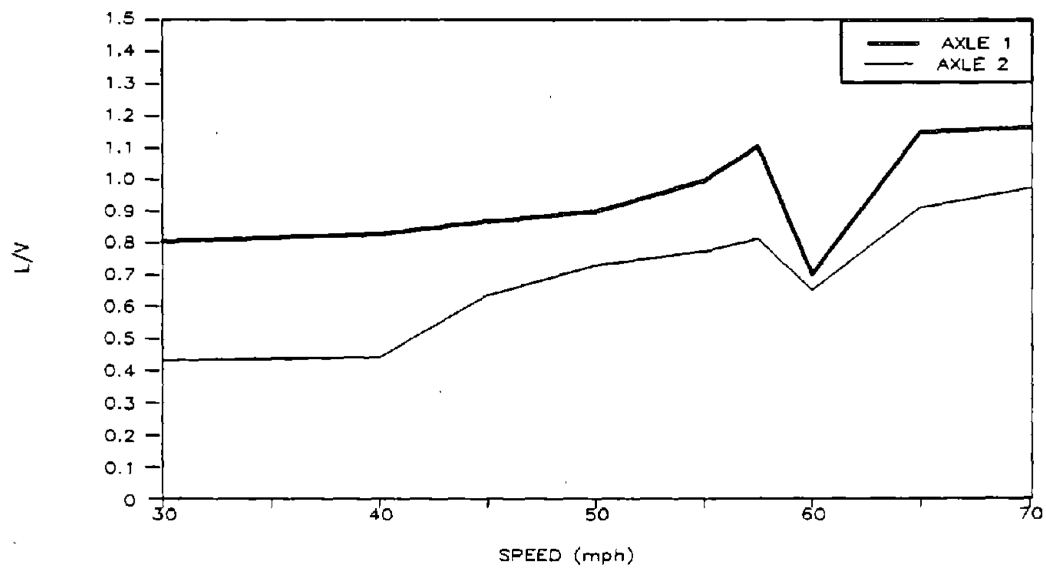

Figure 27. Pre-test Predictions of Maximum Axle Sum L/V Ratio Versus Speed of Empty Car on Tangent Track

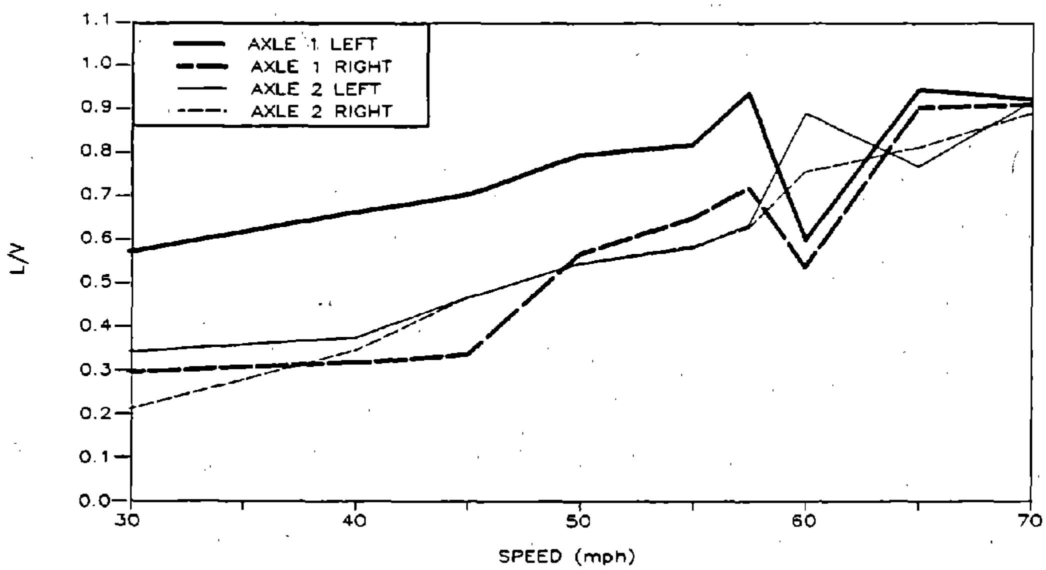

Figure 28. Pre-test Predictions of Maximum Wheel L/V Ratio Versus Speed for the Empty Car on Tangent Track

6.3.2 Empty 50-Minute-Curved Track Hunting

In the 50-minute-curve, sustained oscillations are not predicted for any speed for the empty vehicle. Figures 29 and 30 show that at speeds up to 70 mph the peak-to-peak car body lateral accelerations do not exceed 0.5 g, and the axle sum L/V ratios do not exceed 0.6. These values are well within the Chapter XI limiting criteria. Hunting is not therefore predicted for the 50-minute-curve.

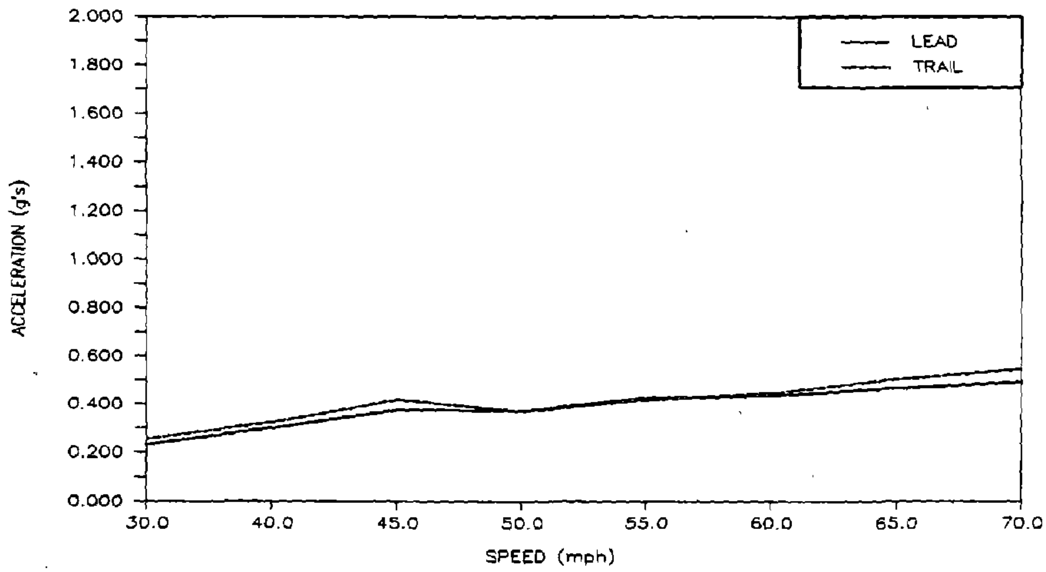

Figure 29. Pre-test Predictions of Lateral Acceleration of the Empty Car on 50-Minute-Curved Track

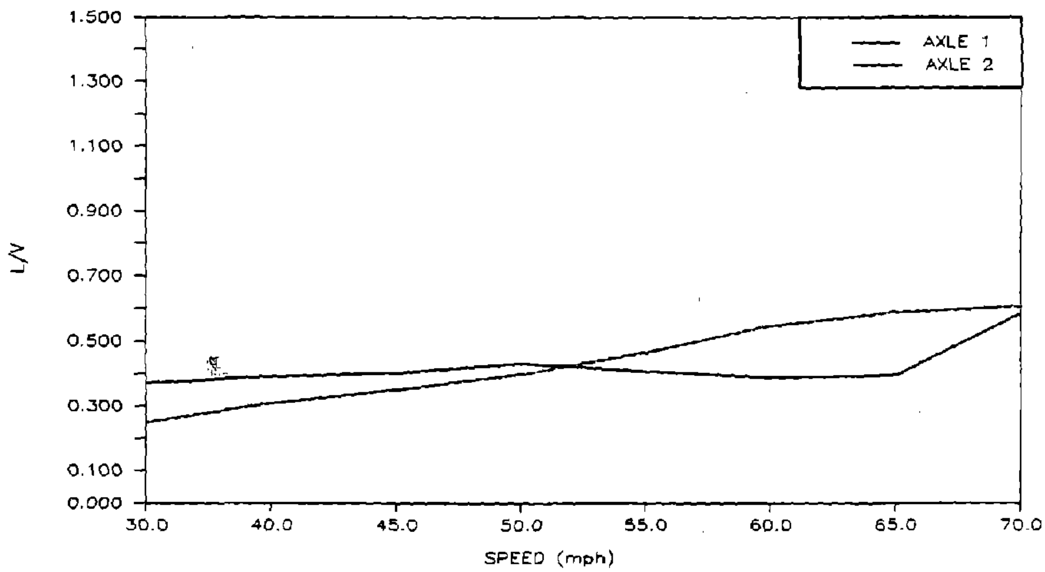

Figure 30. Pre-test Predictions of Maximum Axle Sum L/V Ratios Versus Speed for the Empty Car on 50-Minute-Curved Track

6.3.3 Loaded Tangent Track Hunting

The loaded vehicle is predicted to exhibit oscillatory behavior, beginning at 65 mph and being sustained at 70 mph, as illustrated in Figure 31. Although sustained oscillations are taking place, no Chapter XI limiting criteria are exceeded, with car body lateral accelerations remaining below 0.6 g peak-to-peak and axle sum L/V ratios remaining below 0.75.

Because it is unusual for normal loaded freight vehicles to exhibit hunting activity, this behavior is considered significant even though no Chapter XI safety criteria are exceeded.

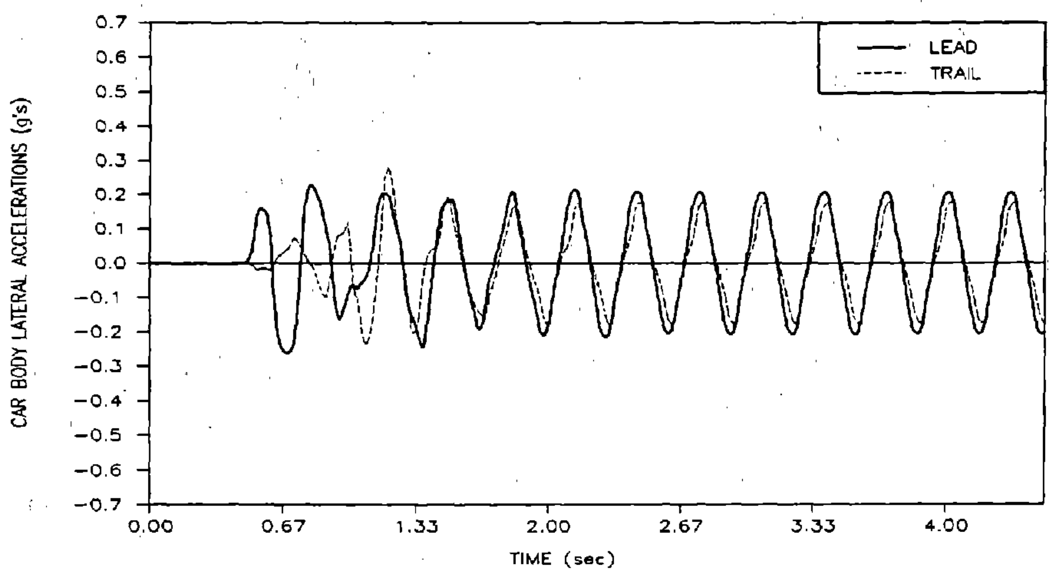

Figure 31. Pre-test Predictions of Lateral Car Body Acceleration for the Loaded Car at 70 mph on Tangent Track

6.3.4 Loaded 50-Minute-Curved Track Hunting

In the 50-minute-curve no sustained lateral oscillations are predicted for the loaded vehicle. Maximum axle sum L/V ratios are below 0.35 and the maximum lateral car body accelerations are less than 0.35 g at speeds up to 70 mph.

6.4 TWIST AND ROLL

Twist and roll analyses evaluate the roll dynamic performance of a vehicle negotiating track with varying cross level such as may occur on staggered jointed rail. This is simulated by defining ten 39-foot segments of rail with the joints lowered by 0.75 inches from the centers. The left and right rails have the joints offset by 19.5 feet providing a varying cross level.

Predictions were made for both the loaded and empty vehicle as required by Chapter XI.

6.4.1 Empty Twist and Roll

The predictions for the empty vehicle in the twist and roll section show a significant roll resonance at 35 mph, shown in Figure 32, with a maximum peak-to-peak roll angle of 5-degrees. Figure 32 also plots the roll angle of the body relative to the truck bolsters. This data indicates that the body roll is almost entirely due to the body rolling relative to the truck bolster. Peak-to-peak spring deflections are at most 0.3 inches at this resonant speed, and therefore contribute little to the car body roll.

Figure 33 shows that at the resonant speed the maximum axle sum L/V ratios are less than 0.6 but rise to 1.4 on the lead axle at 60 mph. At 65 and 70 mph the vehicle derails. Similarly the minimum vertical wheel loads are greater than 30 percent of the static value at resonance but begin to fall at the higher speeds.

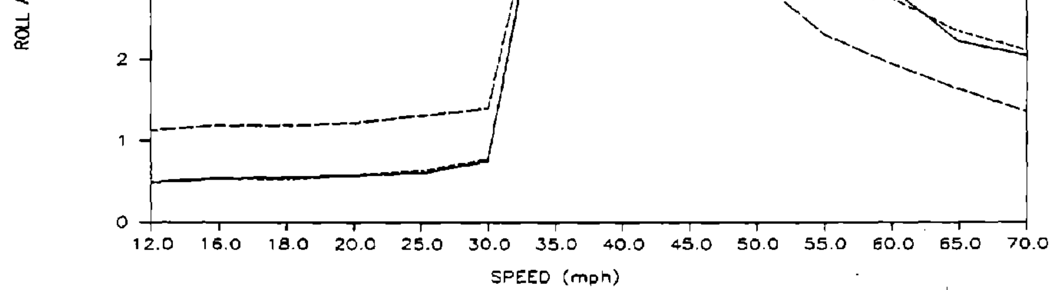

Figure 32. Pre-test Predictions of Maximum Car Body and Bolster Roll Angles for the Empty Car in the Twist and Roll Test Zone

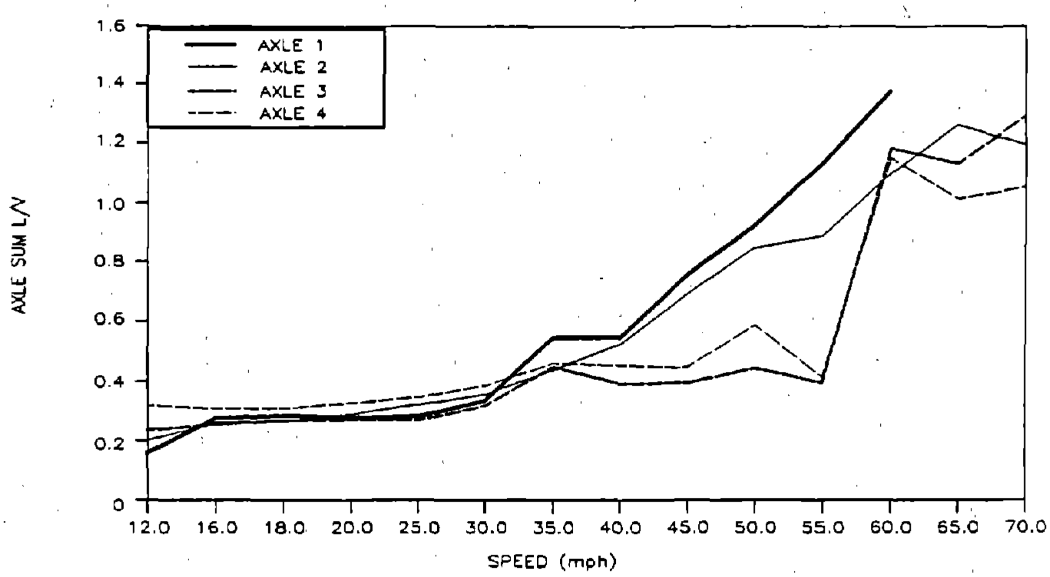

Figure 33. Pre-test Predictions of Maximum Axle Sum L/V Ratio for the Empty Car in the Twist and Roll Test Zone

Increased L/V ratios, lower vertical forces, and consequent derailment at speeds above 60 mph do not appear to be caused by roll phenomena, but by lateral oscillations of the wheel sets. Figure 34 shows distance histories of the lateral positions of the four axles traversing the test zone at 70 mph. At the 100-foot distance, which marks the beginning of the test section, the wheel sets begin lateral oscillations at a frequency of about 3 Hz. This corresponds to the hunting frequency predicted in Section 6.3.1. This could easily be excited by the passing frequency of the 39-foot perturbations, which at 70 mph is 2.6 Hz.

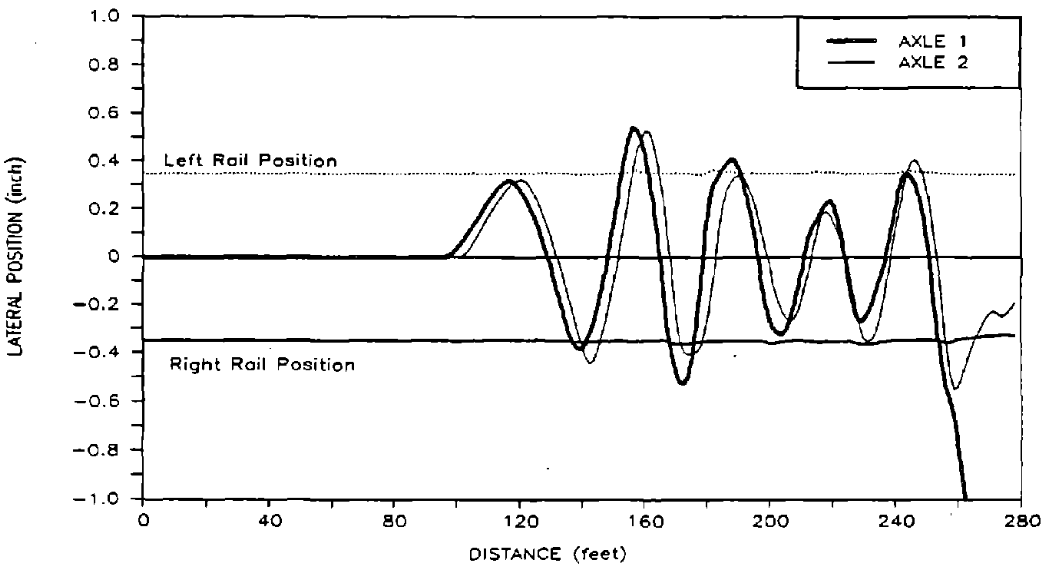

Figure 34. Pre-test Predictions of Time History of Axle Lateral Motion of the Empty Car at 70 mph in the Twist and Roll Test Zone

The predictions therefore indicate that the vehicle exceeds the Chapter XI limiting criteria because it is hunting. The twist and roll perturbations are only acting to excite the hunting behavior.

6.4.2 Loaded Twist and Roll

The predictions for the loaded vehicle in the twist and roll test section indicate performance well within Chapter XI limiting criteria. The peak-to-peak car body roll angles illustrated in Figure 35 reach a maximum of 3-degrees at 25 mph. This roll resonant speed is confirmed with minimum wheel loads and maximum axle sum L/V ratios all occurring at this speed. All criteria remained well within Chapter XI limits, with the minimum wheel loads remaining greater than 45 percent of the static values. The axle sum L/V ratios show peaks at the 25 mph resonant speed, although the maximum value of 0.4 is reached at 70 mph for axles 1 and 2 and at 55 mph for axles 3 and 4, as shown in Figure 36.

It appears in Figure 36 that a secondary resonant condition is occurring at 55 mph to 70 mph. This is probably related to lateral oscillations of the axles due to the mild hunting discussed in Section 6.3.3, and does not appear to be a roll phenomenon. If roll was occurring it would be evident in the car body roll angle data in Figure 35.

The favorable roll behavior in the loaded condition is probably due to the low center of gravity of the vehicle.

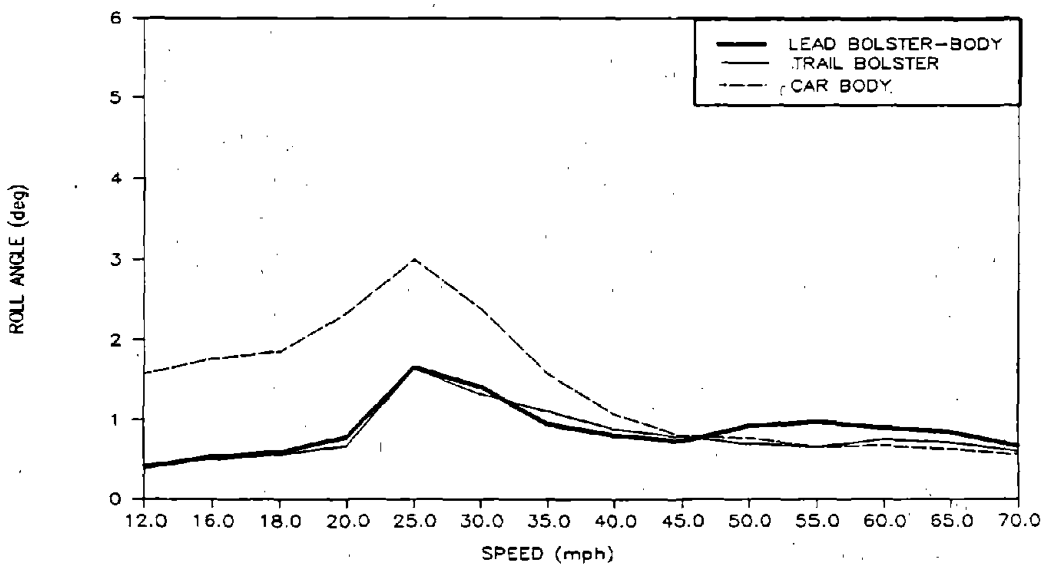

Figure 35. Pre-test Predictions of Maximum Car Body and Bolster Roll Angles for the Loaded Car in the Twist and Roll Test Zone

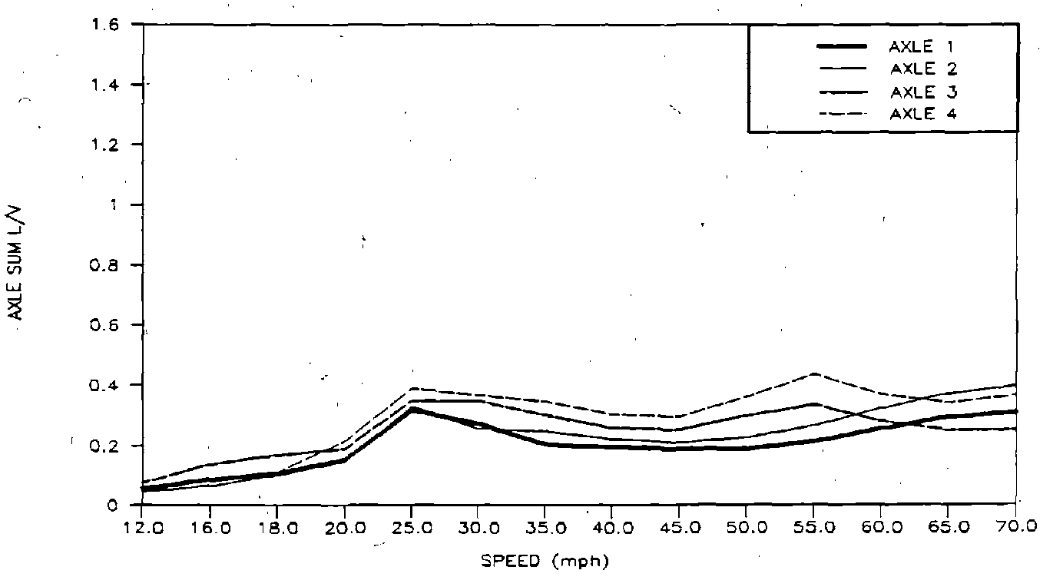

Figure 36. Pre-test Predictions of Maximum Axle Sum L/V Ratios for the Loaded Car in the Twist and Roll Test Zone

6.5 PITCH AND BOUNCE

Pitch and bounce analyses evaluate vertical dynamic performance of a vehicle negotiating track with a varying vertical profile such as might be caused by track with parallel low rail joints. This is simulated similarly to the twist and roll with ten 39-foot sections of rail, low by 0.75 inches at the joints. In this case, however, the joints are set in phase with each other giving no cross level variation but a varying vertical profile.

Chapter XI requires tests and analyses only for the loaded car. The results of the Lightweight Car 1 project indicated worse performance for the empty car than the loaded car in pitch and bounce. Therefore, both loaded and empty car analyses were performed.

6.5.1 Empty Pitch and Bounce

The predictions for the empty vehicle in the pitch and bounce test section show performance well within Chapter XI limits. No resonant condition is predicted. Minimum wheel loads of 62 percent of the static value occur at 70 mph, as shown in Figure 37. It appears that any resonance is likely to be above 70 mph.

6.5.2 Loaded Pitch and Bounce

The loaded car also is predicted to perform well within Chapter XI limits. Minimum wheel loads, shown in Figure 38, are all above 75 percent of the static values. It appears that a mild resonance is occurring near 70 mph. This is confirmed by the peak-to-peak spring deflections shown in Figure 39, which show a maximum of 0.92 inches at 65 mph, and fall slightly at 70 mph.

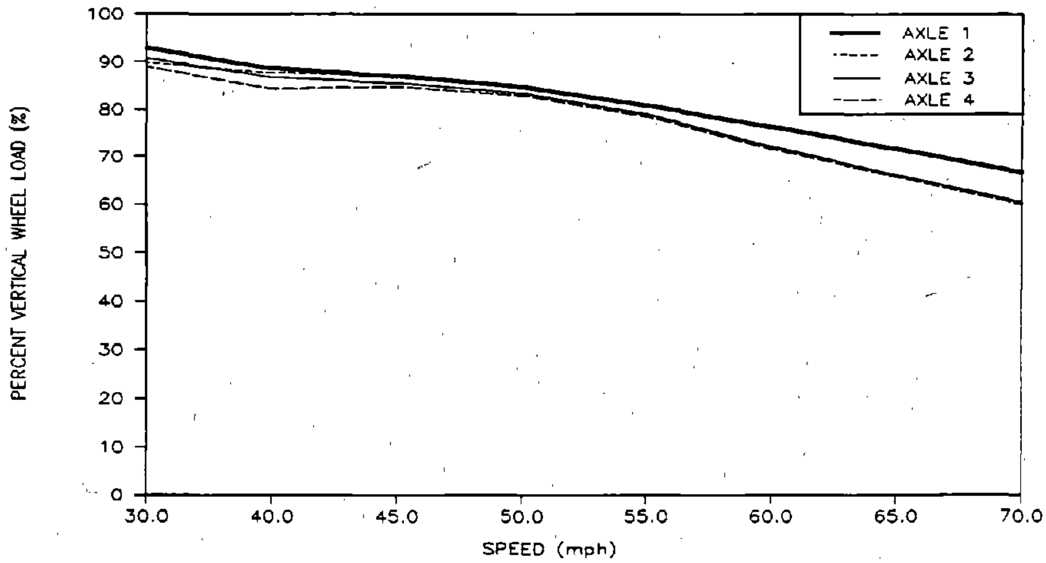

Figure 37. Pre-test Predictions of Minimum Percent of Vertical Wheel Load for the Empty Car with Respect to Static Load in the Pitch and Bounce Test Zone

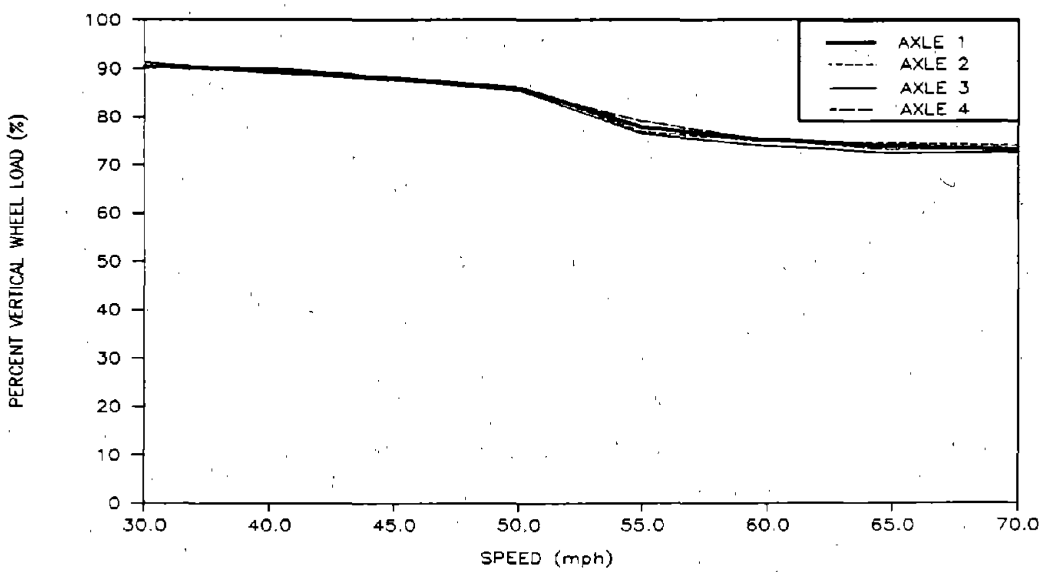

Figure 38. Pre-test Predictions of Minimum Percent of Vertical Wheel Load for the Loaded Car with Respect to Static Load in the Pitch and Bounce Test Zone

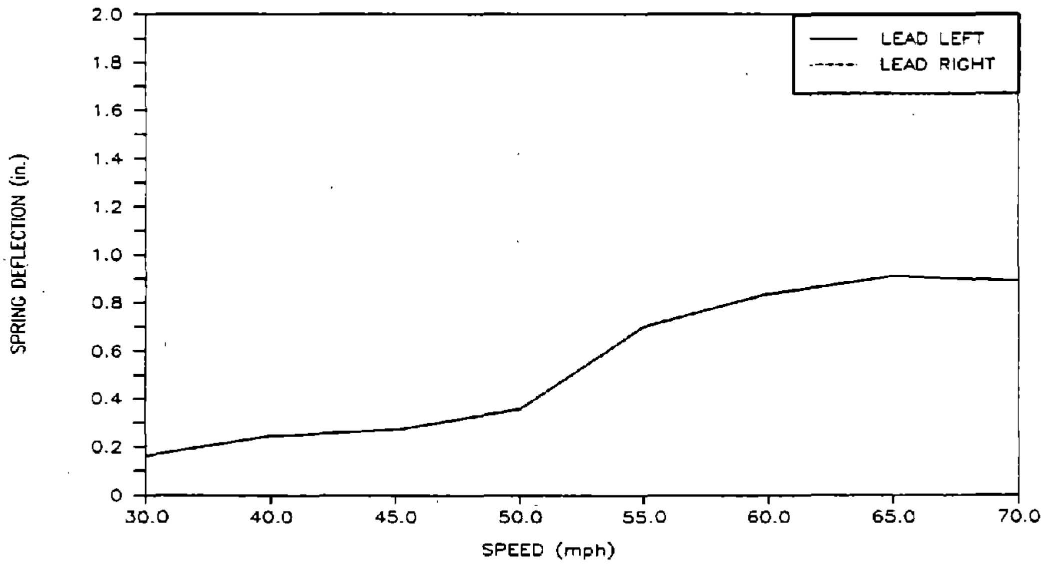

Figure 39. Pre-test Predictions of Maximum Vertical Spring Deflections for the Loaded Car in the Pitch and Bounce Test Zone

6.6 SINGLE BOUNCE

The single bounce also analyzes the vertical dynamic performance of a vehicle, but only over a single perturbation of large amplitude. This is intended to simulate the effect of a sudden increase in vertical track stiffness such as occurs at grade crossings. The track section was defined as a segment of track 24 feet long raised by 2 inches. Ramps 6 feet long lead up to and down from the raised section, making the perturbed section 36 feet long.

Both empty and loaded predictions were made although Chapter XI only requires loaded car tests and analyses.

6.6.1 Empty Single Bounce

The predicted performance of the empty vehicle on the single bounce is considerably different than on the multiple bounce section. Predicted vertical wheel loads are reduced to 10 percent of the static values at speeds of 40 mph, for axles 1 and 3. This is clearly shown in Figure 40. At 45 mph, axles 1 and 3 show minimum loads close to zero and the other two axles are approaching 10 percent. Derailment is predicted for 50 mph and 55 mph, although above these speeds minimum vertical wheel loads begin to increase along with spring deflections. The spring deflection data (Figure 41) indicates a resonance at 60 mph to 65 mph.

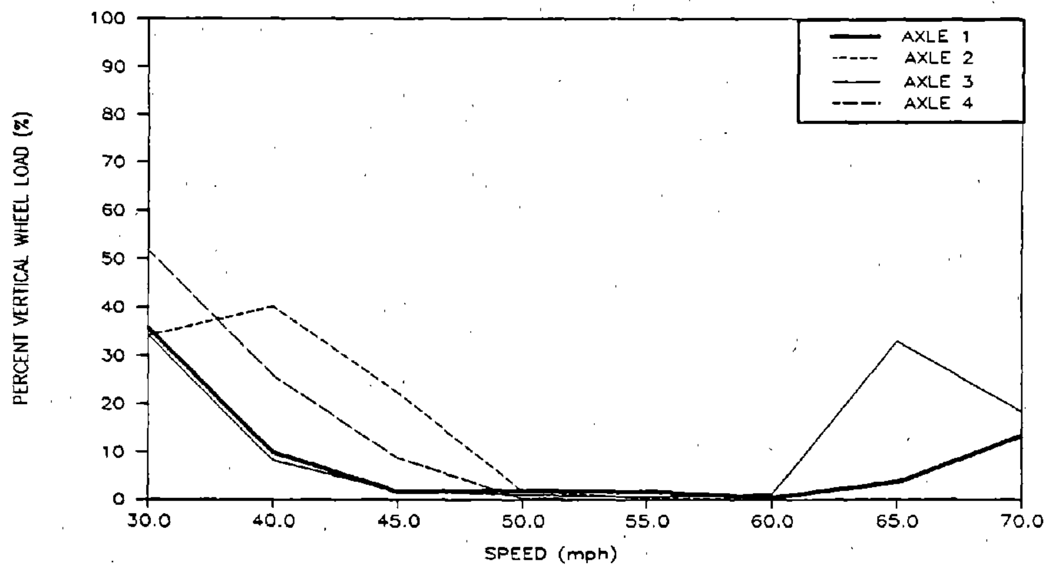

Figure 40. Pre-test Predictions of Minimum Percent Vertical Wheel Load for the Empty Car Passing Over the Single Bump

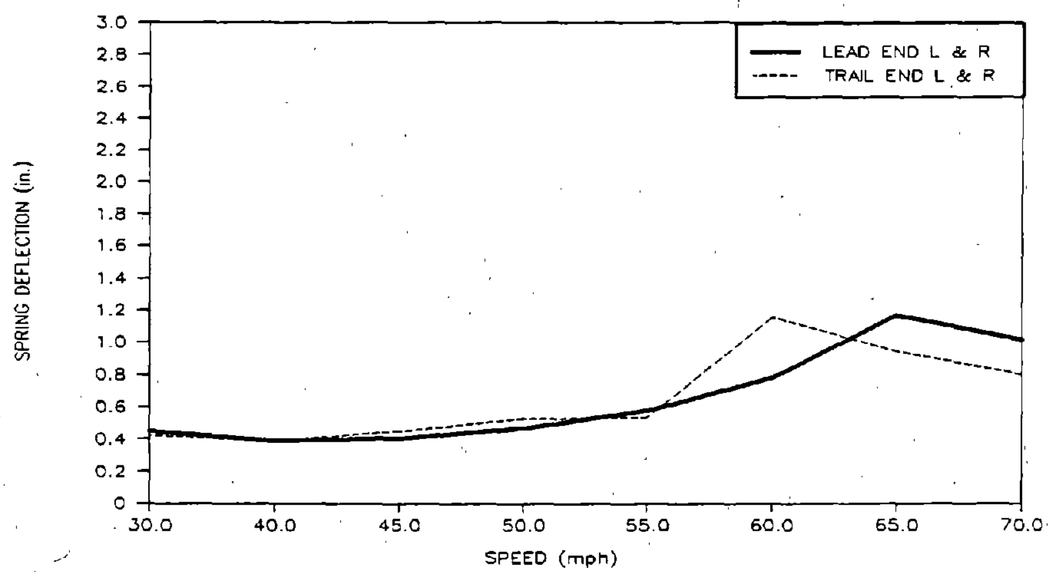

Figure 41. Pre-test Predictions of Maximum Vertical Spring Deflection for the Empty Car Passing Over the Single Bump

It appears that at the lower speeds the vertical suspension remains "locked up" so that rather than deflecting the springs the vehicle "jumps," unloading the wheels. When at the higher speeds the suspension breaks free, the wheels can then remain in contact with the rails while the body moves up and down on the suspension.

Although tests and analyses in the empty condition are not required by Chapter XI, these results indicate a possible need to revise the requirements.

6.6.2 Loaded Single Bounce

The loaded vehicle's predicted performance is much less severe. Figure 42 shows that the minimum wheel loads get no lower than 25 percent of the static values with resonance occurring at 65 mph for axles 3 and 4. This resonance is confirmed by the peak-to-peak spring deflection data displayed in Figure 42. The lead axles appear to be approaching a resonance at 70 mph or above.

The spring deflection data in Figure 43 clearly indicates that the suspension is free to move allowing the wheel sets to remain in contact with the rails, while the body moves up and down in response to the perturbation.

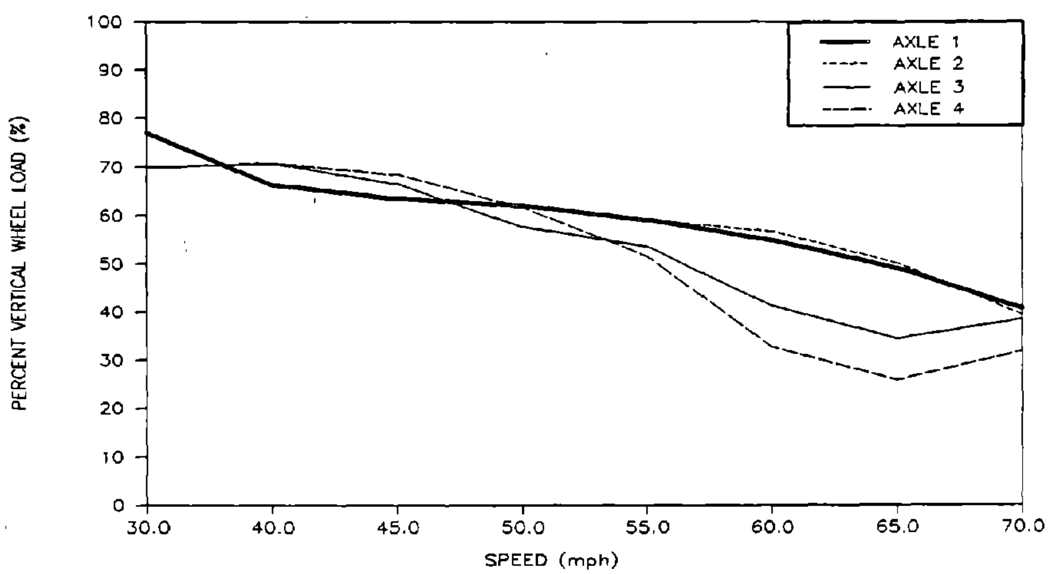

Figure 42. Pre-test Predictions of Minimum Percent Vertical Wheel Load for the Loaded Car Passing Over the Single Bump

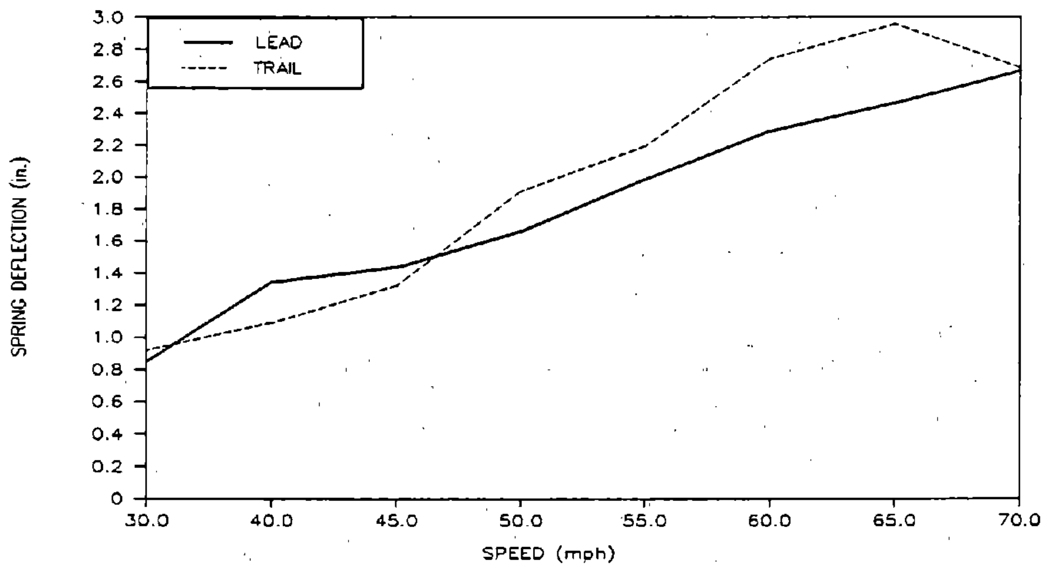

Figure 43. Pre-test Predictions of Maximum Vertical Spring Deflection for the Loaded Car Passing Over the Single Bump

6.7 YAW AND SWAY

The yaw and sway analyses evaluate the lateral dynamics of a vehicle negotiating track with a sinusoidal lateral alignment perturbation and wide track gage. This is defined by five segments of track with sinusoidal lateral alignment variation, 1.25 inches amplitude, 39-foot wavelength. Throughout this zone, track gage is held to 57.5 inches, 1 inch wider than standard.

Chapter XI only requires yaw and sway analyses and tests for the loaded vehicle. The results of the previous Lightweight Car 1 project and the predictions for this vehicle over other test sections indicated poor performance under many conditions when empty. It was therefore decided to also perform yaw and sway analyses and tests for this car in the empty condition.

6.7.1 Empty Yaw and Sway

Results of the empty vehicle yaw and sway analyses are contrary to the behavior expected based on previous test experiences. At all speeds above 30 mph derailments are predicted. The predictions all show the axles steering sharply into the perturbations, with the axles being drawn deeper into each successive perturbation until they eventually derail. This occurs earlier in the test section for increased speeds.

Figure 44 illustrates this phenomenon. Axle 1 is shown running down the track within the flangeway clearance of each wheel. Note that the axle is centered between the rails in the tangent portion of the test section. As the wheel set enters the test zone, it appears to be drawn to run into flange contact with first the left and then the right rail, alternating with increasing amplitude until the wheel overshoots and the flange climbs the rail.

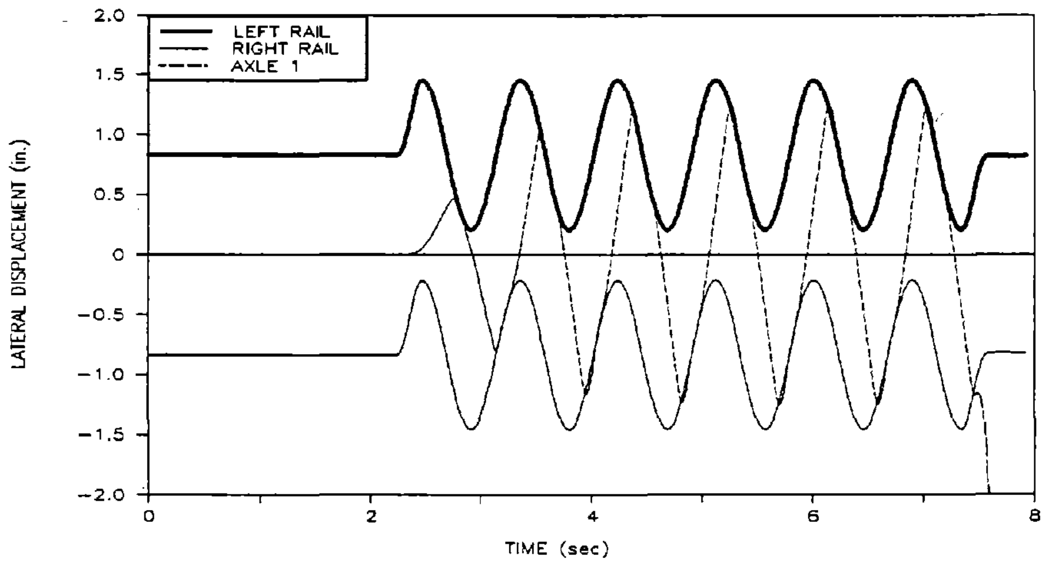

Figure 44. Pre-test Predictions of Lateral Position of Axle 1 for the Empty Car in Relation to the Lateral Rail Position in the Yaw Sway Test Zone at 30 mph

These results are far more severe than what might be considered likely for an ordinary freight car. It is possible that the descriptions of the lateral and yaw suspensions of the vehicle's trucks are inaccurate leading to anomalous predictions. Similar excessively severe predictions for the yaw sway test zone occur with NUCARS for most vehicles, including the Frontrunner car of the Lightweight Car 1 test program. Analyses of the actual track test data have been made to develop dynamic characteristics for these suspensions. The post-test predictions have been made with these new characteristics to determine their effects.

6.7.2 Loaded Yaw and Sway

The loaded vehicle predictions for yaw and sway are similar to the empty vehicle predictions. Similar behavior is seen with the axles still running in alternate left and right wheel flange contact.

6.8 STEADY STATE CURVING

The steady state curving analyses are intended to evaluate the ability of a vehicle to negotiate track curves. Chapter XI requires analysis of a vehicle negotiating a single curve between 7- and 15-degrees of curvature, with a balance speed between 20 and 30 mph. Analyses are to be performed at speeds representing cant deficiencies of -3, 0, and +3 inches.

For the purposes of this project, predictions were made at curvatures of 4-, 7.5-, 10- and 12-degrees, with superelevation of 3, 3, 4, and 5 inches respectively. Predictions were made at cant deficiencies of -3, -1.5, 0, +1.5, and +3 inches. This was to permit a greater understanding of the mechanisms controlling this vehicle's curving behavior. These curves were chosen because they match the curves available for track tests at the TTC. For the 4- and 7.5-degree curves the -3 inch underbalance conditions could not be modeled. Because the track has 3 inches superelevation in these curves, the -3 inch underbalance speed would be 0 mph.

As per Chapter XI requirements, analyses were performed for both the loaded and empty car.

6.8.1 Empty Steady State Curving

Predictions for the empty vehicle negotiating steady curves show performance well within Chapter XI limiting criteria. A maximum single wheel L/V ratio of 0.55 is achieved on the lead axle outside wheel in the 12-degree curve with +3 inches of imbalance. This same condition also generates a maximum axle sum L/V of 0.95, again on the lead axle.

Behavior for this truck when the vehicle is empty is similar to that for normal three-piece trucks. The leading axle generates the largest L/V ratios, as shown in Figure 45. The trailing axle forces are much lower. The large L/V ratios are due to the large angle of attack (AOA) the leading axle takes up relative to the rails. Figure 46 demonstrates that the trailing axle generates an AOA only one tenth the AOA of the leading axle of the leading truck. The AOA increases with curvature in the same manner as a normal three-piece truck.

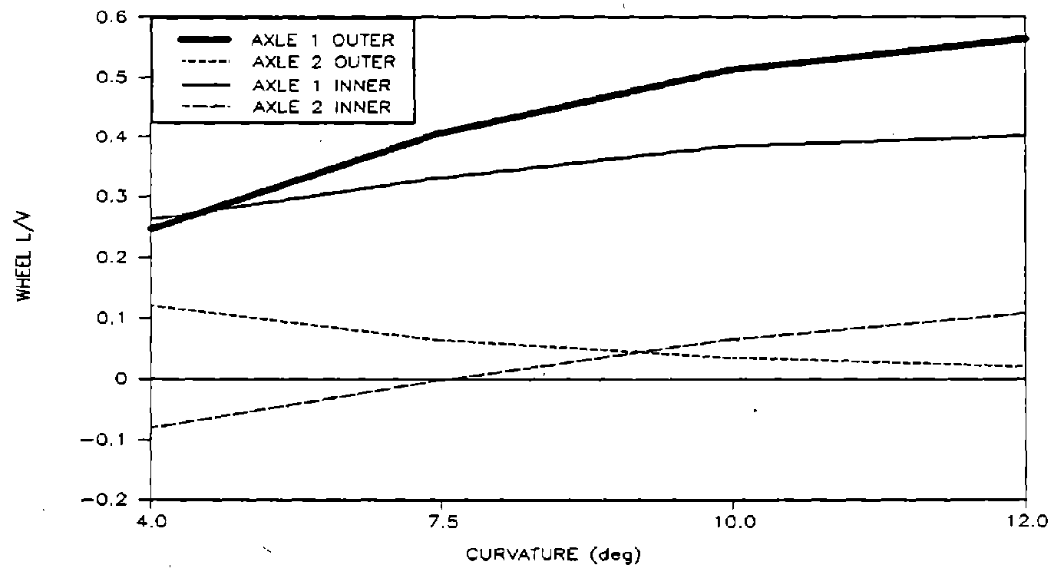

Figure 45. Pre-test Predictions of Maximum Individual Wheel L/V Ratio at +3 Inches Cant Deficiency

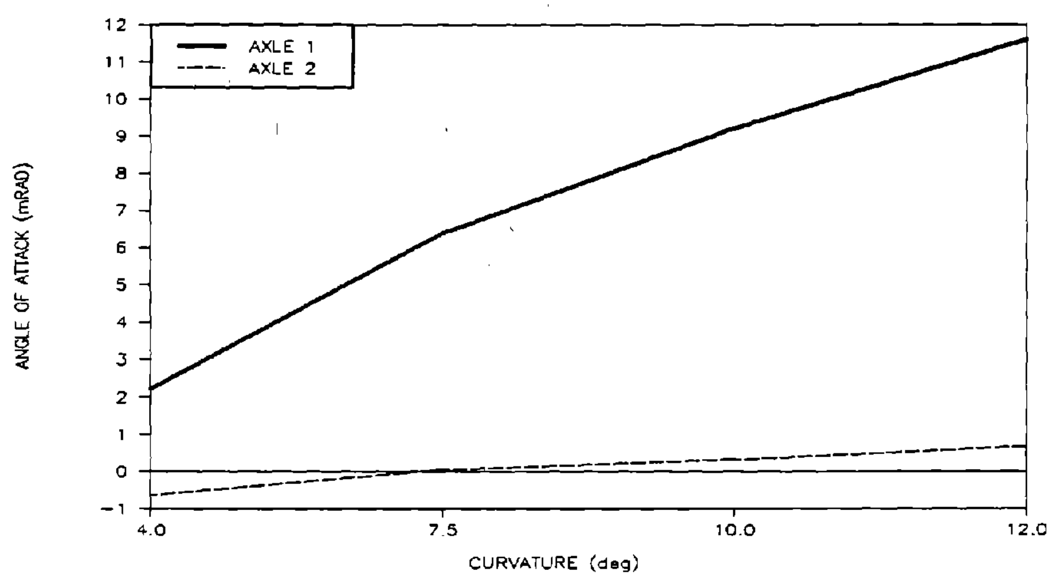

Figure 46. Pre-test Predictions of Angle of Attack of Axles 1 and 2 at +3 Inches Cant Deficiency

6.8.2 Loaded Steady State Curving

The loaded vehicle is predicted to perform better than the empty vehicle. The maximum wheel L/V generated is 0.42 on the lead outside wheel in the 12-degree curve at -3 inches of (cant deficiency) imbalance. The same condition also generates the maximum axle sum L/V ratio of 0.82 on the lead axle. Trailing axle L/V ratios all remain low. These results are clearly evident in Figure 47.

These lower ratios are due to the loaded vehicle "steering" better than the empty one. This is demonstrated in Figure 48 which plots AOA against speed. Angles of attack are one-third less than those predicted for the empty car. Note that in these figures no data is presented for the 4- and 7.5-degree curves. This is because these curves have only 3 inches of superelevation. To achieve -3 inches of imbalance on this amount of superelevation, the vehicle would have to be standing still.

This better performance is almost certainly due to the presence of the rubber shear pads between the bearing adaptor and side frame. The higher axle loads, when loaded, allow sufficient longitudinal forces to develop between the wheels and rails to deflect the pads, allowing the axles to steer. At the lower loads of the empty car longitudinal forces are insufficient to deflect the pads. With a conventional three-piece truck, the friction between bearing adaptor and side frame is so great as to prevent virtually all motion, inhibiting the truck from steering well.

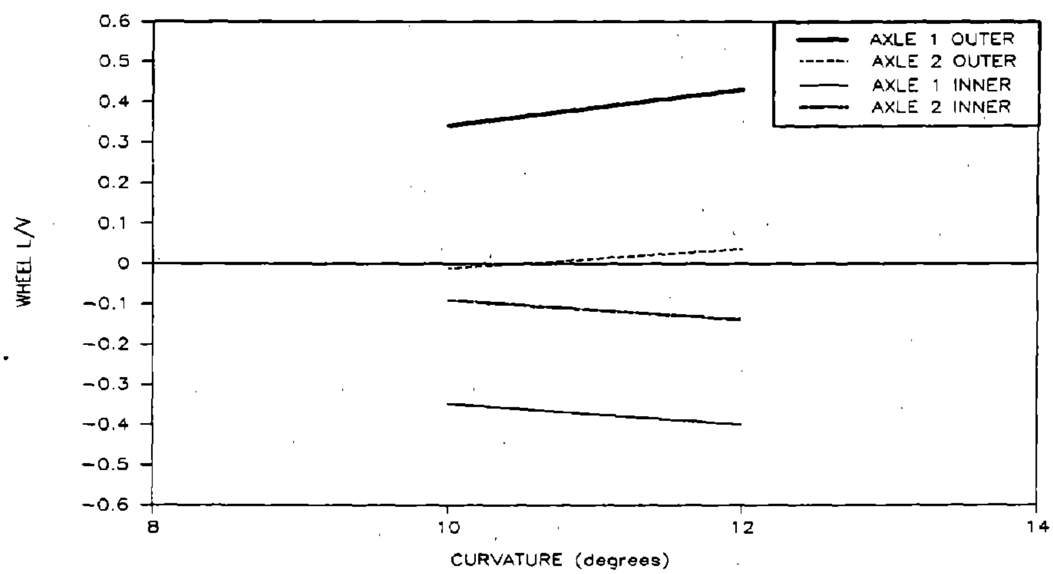

Figure 47. Pre-test Predictions of Maximum Wheel L/V at -3 Inches Cant Deficiency

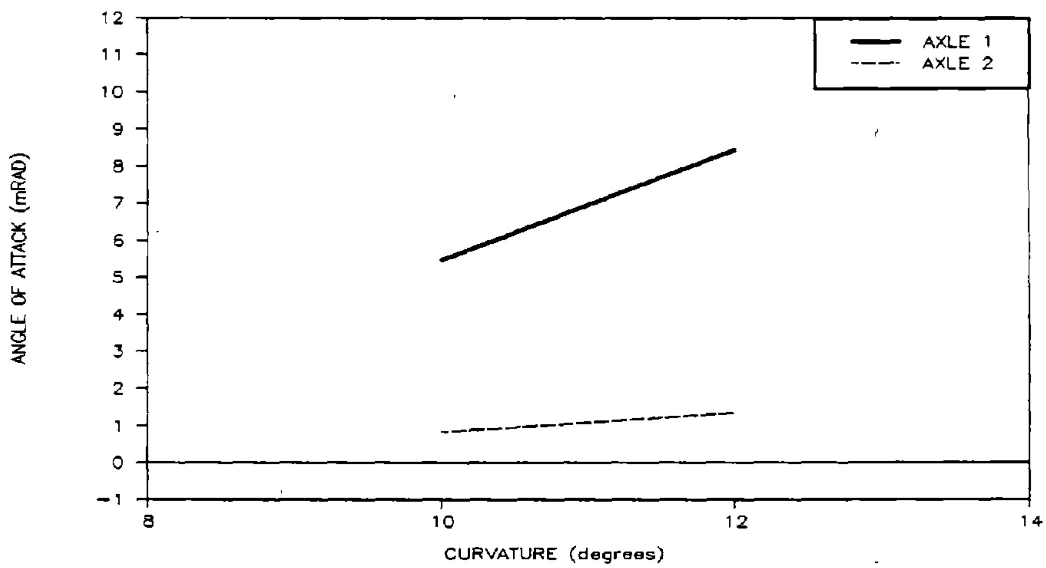

Figure 48. Pre-test Predictions of Angle of Attack of Axles 1 and 2 for the Loaded Car at -3 Inches Cant Deficiency

6.9 CURVE ENTRY/EXIT

The curve entry/exit analyses are intended to evaluate the dynamic performance of a vehicle as it negotiates the entry and exit spiral to a curve. In order to perform the analysis under extreme conditions, the spiral is defined to be a "bunched spiral" in which the change in superelevation occurs in the center portion of the spiral rather than being evenly distributed over the entire length of the spiral. In order to match the bunched spiral in place at the TTC, the NUCARS predictions are for a 200-foot spiral leading into a 12-degree curve with 5 inches of superelevation. The change in superelevation occurs in the central 100 feet of the spiral.

As required by Chapter XI, the analyses were performed for the empty and loaded car, entering and exiting this spiral.

6.9.1 Empty Curve Entry

Predictions for the empty vehicle entering the bunched spiral (Figures 49 and 50) show the highest wheel L/V ratios of 0.65 occurring on the leading inside wheel, during the last 50 feet of the spiral, while running with +3 inches of imbalance (31.1 mph). At the same time, the same wheel shows the maximum unloading to be 65 percent of the static vertical load. This performance is well within Chapter XI limits.

This position in the spiral comes after all the superelevation change has occurred. It is therefore to be expected that significant unloading might occur in this region.

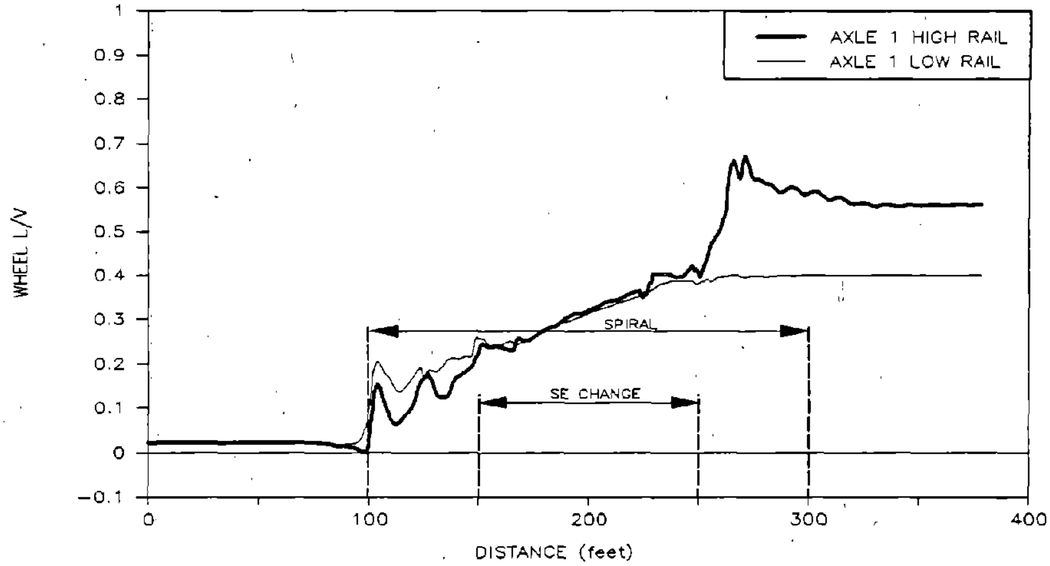

Figure 49. Pre-test Predictions of Maximum Wheel L/V for the Empty Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

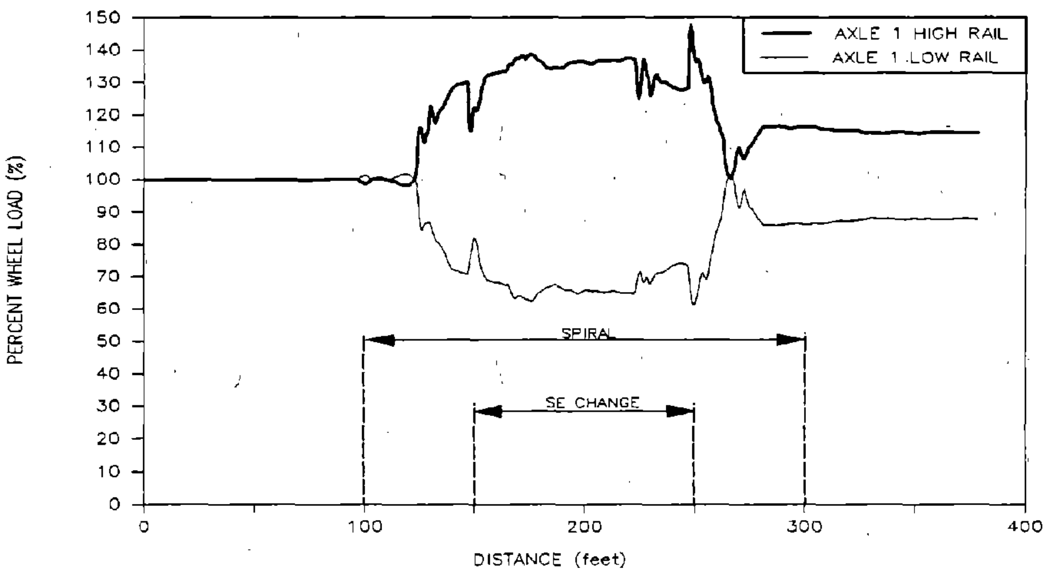

Figure 50. Pre-test Predictions of Minimum Percent Vertical Load of Individual Wheels for the Empty Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

6.9.2 Empty Curve Exit

The empty vehicle exiting the bunched spiral produces different results, shown in Figures 51 and 52. While running at +3 inches of imbalance (31.1 mph), the lead outside wheel generates the largest L/V ratio of 0.58. This occurs in the center 100 feet of the spiral where all the change in superelevation takes place. The same wheel also drops to 68 percent of the static wheel load in the same place, as well as further down the track in the last 50 feet where there is no superelevation.

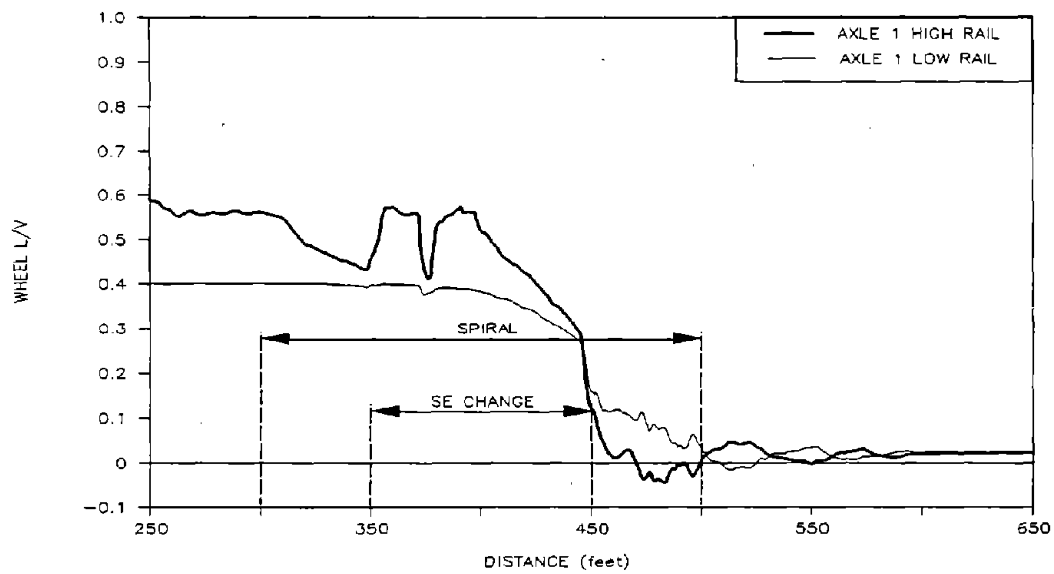

Figure 51. Pre-test Predictions of Maximum of Wheel L/V for the Empty Car Exiting the Bunched Spiral to Tangent Track at 31.1 mph

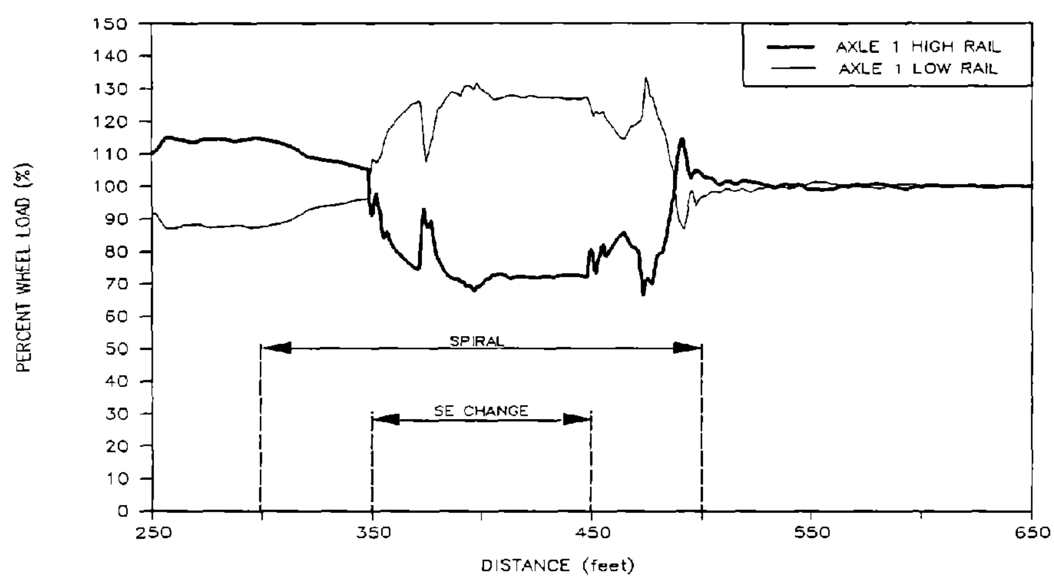

Figure 52. Pre-test Predictions of Minimum Percent Wheel Vertical Load for the Empty Car Exiting the Bunched Spiral to Tangent Track at 31.1 mph

6.9.3 Loaded Curve Entry

The loaded vehicle entering the bunched spiral is predicted to reach a maximum wheel L/V of 0.45 on the lead outside wheel. As shown in Figure 53, this occurs just as the superelevation reaches the maximum, 50 feet before the end of the spiral, while running at +3 inches of imbalance (31.1 mph).

The minimum vertical wheel load, shown in Figure 54, is reached by the lead inside wheel at the same speed. This minimum of 65 percent of the static load occurs just as the superelevation is beginning, 75 feet from the start of the spiral.

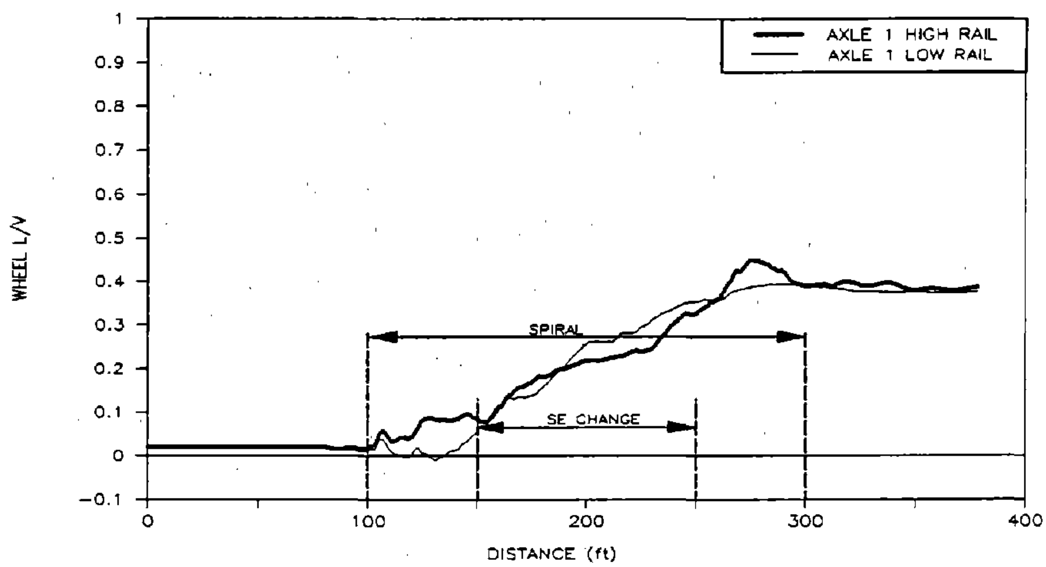

Figure 53. Pre-test Predictions of Maximum Wheel L/V for the Loaded Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

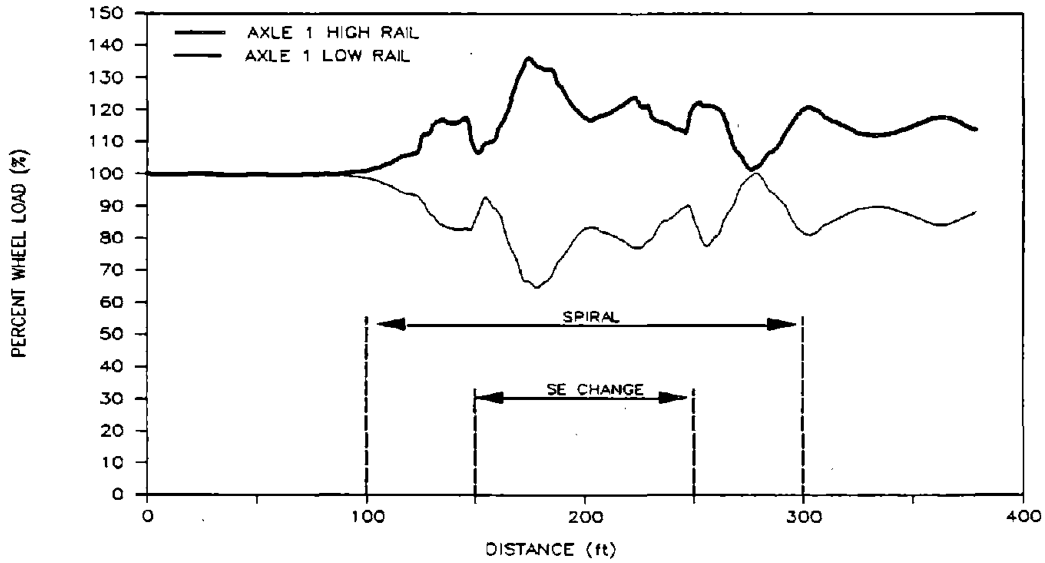

Figure 54. Pre-test Predictions of Minimum Percent Wheel Vertical Load for the Loaded Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

6.9.4 Loaded Curve Exit

In the exit of the bunched spiral, maximum wheel L/V ratio of 0.45 is predicted for the lead outside wheel while running at -3 inches of imbalance (15.5 mph), as shown in Figure 55. The same speed causes the maximum wheel unloading of 65 percent on the same wheel in the middle of the spiral (Figure 56).

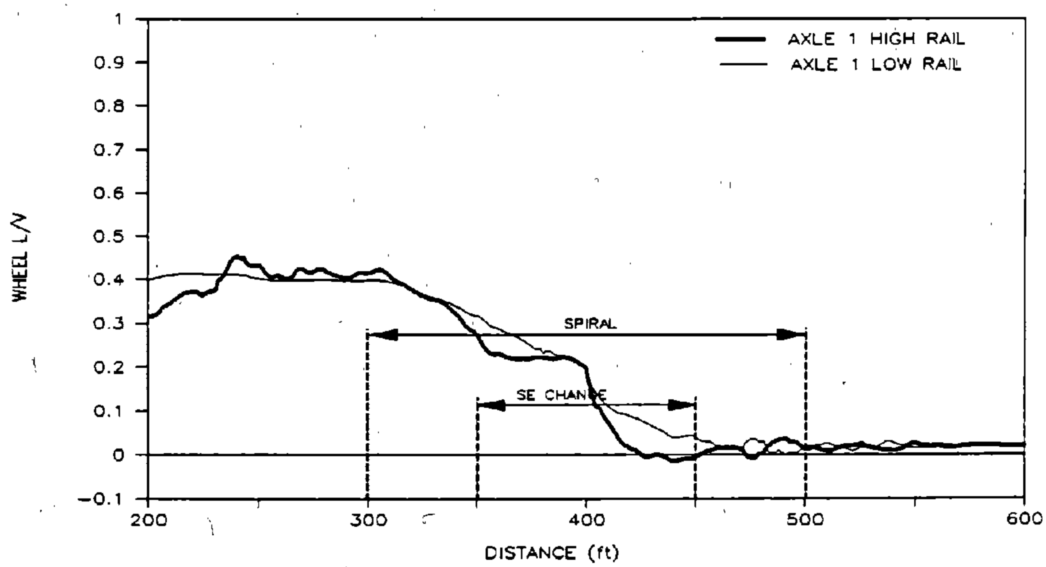

Figure 55. Pre-test Predictions of Maximum Wheel L/V for the Loaded Car Exiting the Bunched Spiral to Tangent Track at 15.5 mph

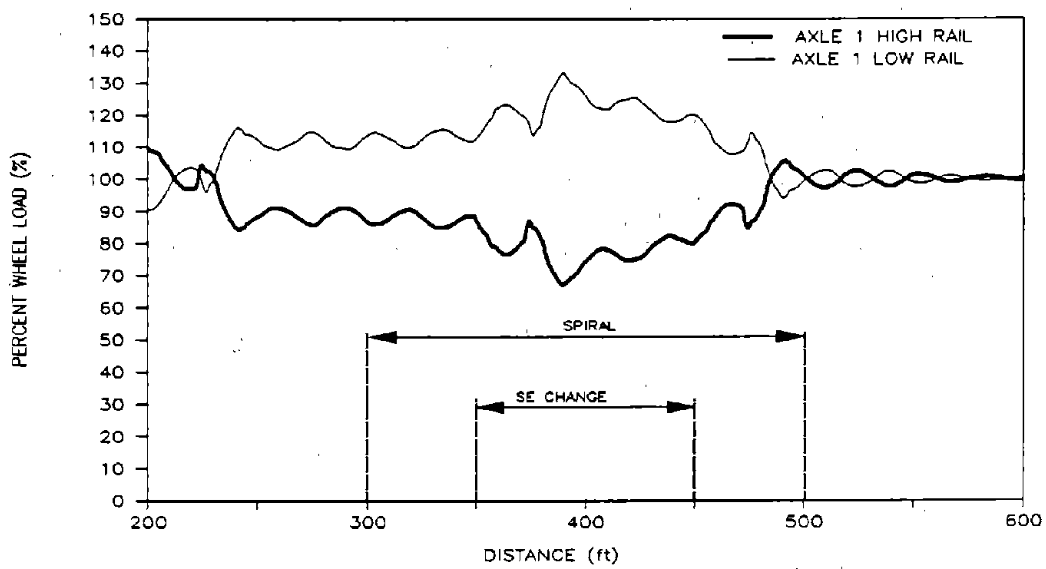

Figure 56. Pre-test Predictions of Minimum Percent Wheel Vertical Load for the Loaded Car Exiting the Bunched Spiral to Tangent Track at 15.5 mph

6.10 DYNAMIC CURVING

The dynamic curving analyses are to evaluate vehicle dynamic performance while negotiating a steady curve with vertical and lateral perturbations. This Chapter XI section is based on a 10-degree curve with 4 inches of superelevation. The curve contains a 200-foot-long twist and roll test section, similar to the tangent track twist and roll section The perturbations consist of 0.5 inch amplitude low rail joints at a wavelength of 39 feet. The outside rail is also given outward cusps such that the track gage is widened to 57.5 inches at every outer rail low joint. The inside rail has no lateral perturbations.

As per Chapter XI requirements, both empty and loaded vehicles were modeled.

6.10.1 Empty Dynamic Curving

Predictions for the empty vehicle negotiating the dynamic curve suffered from the same problems as the yaw-sway predictions. At balance speed (24.1 mph) and above the vehicle was predicted to derail in the second lateral cusp. Figure 57 plots the lateral position of the lead wheel set relative to the two rails, at balance speed. The wheel runs in flange contact until the beginning of the first cusp, and then moves slightly away from contact until just after the peak. The wheel then appears to begin to climb the flange at the valley between the cusps, drops back into ordinary flange contact and finally derails after the second peak.

Individual wheel L/V ratios, axle sum L/V ratios, minimum wheel loads, and body roll angles are all within Chapter XI limits at -3 and -1.5 inches of imbalance (12 and 19 mph). No indication is given from these parameters that a derailment is likely at higher speeds. Therefore, the predictions of derailment are doubtful. Further predictions need to be made to determine the validity of these results and the source of any possible errors.

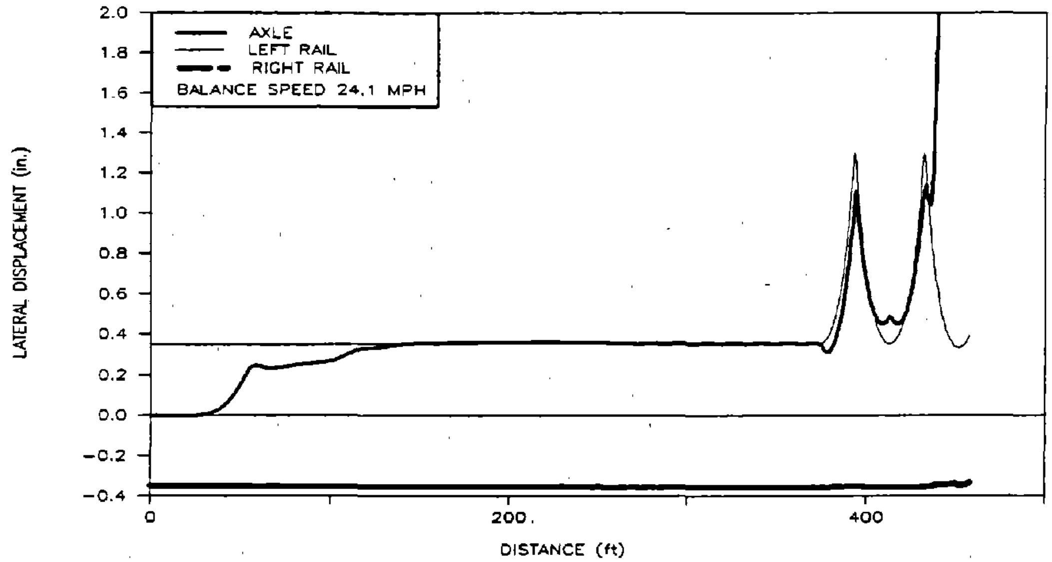

Figure 57. Pre-test Predictions of Lateral Position of Axle 1, Left and Right Rails as the Empty Car Negotiates the Dynamic Curve at Balance Speed

6.10.2 Loaded Dynamic Curving

The predictions for the loaded vehicle are all well within Chapter XI limiting criteria for all speeds modeled. The maximum L/V ratio is 0.5 on the lead axle outside wheel while running with -3 inches of imbalance (12 mph), while the maximum axle sum L/V of 0.95 occurs on the lead axle at +1.5 inches of imbalance (28.2 mph), as illustrated in Figure 58.

Car body peak-to-peak roll angles are small, reaching a maximum of 1.3-degrees at balance speed (24.1 mph). This corresponds well to the predicted roll resonance speed of 25 mph in the twist and roll test zone.

A minimum wheel load of 65 percent of the static value is reached at 3 inches of imbalance (31.9 mph) on the trailing axle inside wheel. Figure 59 shows the minimum wheel loads.

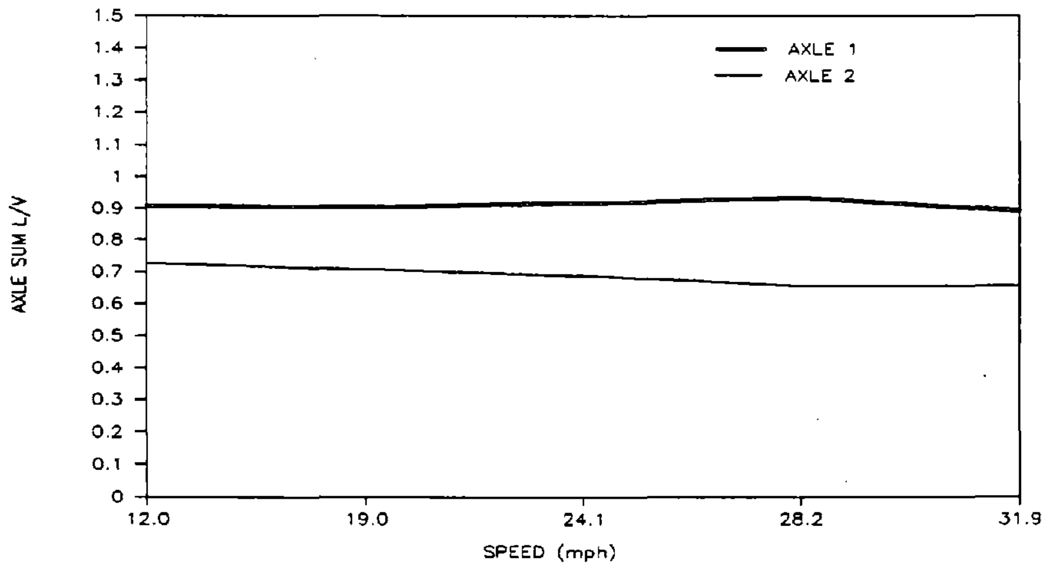

Figure 58. Pre-test Predictions of Maximum Axle Sum L/V Ratios for the Loaded Car Negotiating the Dynamic Curve

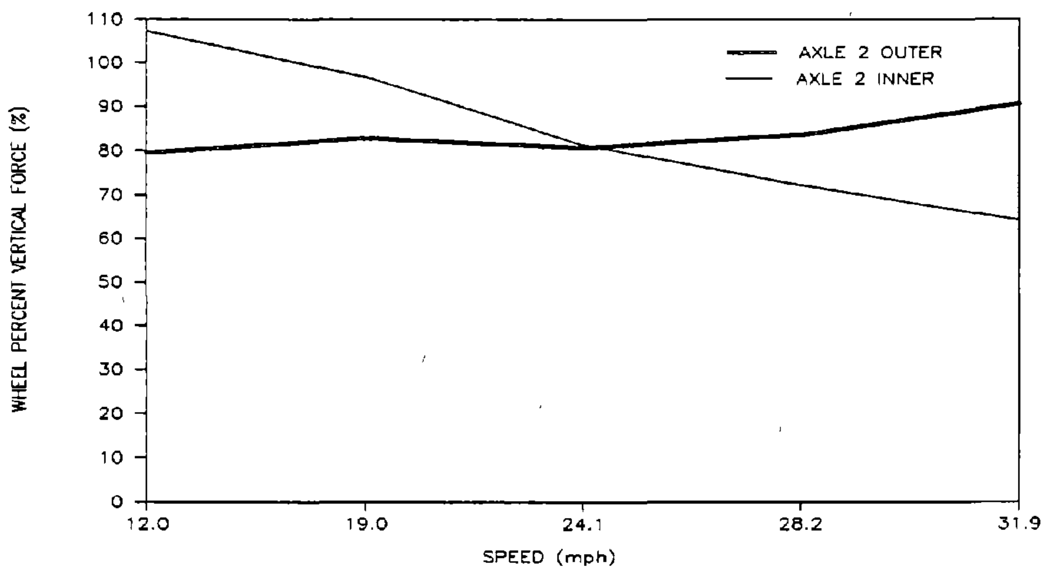

Figure 59. Pre-test Predictions of Minimum Percent Wheel Vertical Load for the Loaded Car in the Dynamic Curve

6.11 SUMMARY OF PRE-TEST NUCARS PREDICTIONS

Successful predictions were made for all conditions except the yaw and sway test zone and the empty dynamic curve. Results from these simulations appear erroneous as they do not match anticipated behavior. It is suspected that errors in the definition of the yaw and lateral suspension characteristics may have affected these results. Table 6 summarizes the predictions in terms of Chapter XI limiting criteria.

Chapter XI limiting criteria were predicted to be exceeded for the following test regimes. This indicated that careful monitoring of the track tests would be required.

1. Empty Car Tangent Hunting
2. Empty Car Pitch and Bounce (test not required by Chapter XI)
3. Empty Car Single Bounce (test not required by Chapter XI)
4. Empty and Loaded Car Yaw and Sway (possibly erroneous results)
5. Empty Car Dynamic Curving (possibly erroneous results)

Table 6. Summary Results of Pre-test NUCARS Predictions

	CHAPTER XI CRITERIA					
TEST CASE	Maximum Lateral Acceleration	Maximum Wheel L/V	Maximum Axle Sum L/V	Minimum % Vertical Wheel Load	Minimum Car Roll Angle	COMMENTS
Tangent Hunting Empty	1g P-P @ 70 mph	0.8 @ 55 mph (not required by Chapter XI)	1.15 @ 70 mph	N/A	N/A	Exceeds Chapter XI @ 57.5 mph. Derails at 70 mph.
Tangent Hunting Loaded	0.6g P-P @ 70 mph	N/A	0.75 @ 70 mph	N/A	N/A	Sustained oscillations at 70 mph do not exceed Chapter XI limits. (Test not required by Chapter XI)
Curved Hunting Empty	0.25 g @ 70 mph	N/A	0.6 @ 70 mph	N/A	N/A	No hunting predicted.
Curved Hunting Loaded	0.35 g @ 70 mph	N/A	0.35 @ 70 mph	N/A	N/A	No hunting Predicted. (Test not required by Chapter XI)
Twist & Roll Empty	N/A	N/A	1.4 @ 60 mph 0.6 @ 35 mph resonant speed	30% @ 35 mph	5.3 deg @ 35 mph	Derailment above 60 mph due to hunting. Roll resonance at 35 mph.
Twist & Roll Loaded	N/A	N/A	0.4 @ 55 mph 0.35 @ 25 mph resonant speed	45% @ 25 mph	3 deg @ 25 mph	Roll resonance at 25 mph. Secondary resonance at 55 mph.
Pitch & Bounce Empty	N/A	N/A	N/A	75% @ 70 mph	N/A	No resonance predicted. (Test not required by Chapter XI)
Pitch & Bounce Loaded	N/A	N/A	N/A	75% @ 70 mph	N/A	Mild resonance 65-70 mph

Table 6. Summary Results of Pre-test NUCARS Predictions (Continued)

TEST CASE	CHAPTER XI CRITERIA: Maximum Lateral Acceleration	Maximum Wheel L/V	Maximum Axle Sum L/V	Minimum % Vertical Wheel Load	Minimum Car Roll Angle	COMMENTS
Single Bounce Empty	N/A	N/A	N/A	10% @ 40 mph	N/A	Derailment at 50 and 55 mph. Bounce resonance at 60-65 mph.
Single Bounce Loaded	N/A	N/A	N/A	25% @ 65 mph	N/A	Bounce resonance 65-70 mph.
Yaw & Sway Empty						Results appear inaccurate. Derailment predicted above 30 mph.
Yaw & Sway Loaded						Results appear inaccurate. Derailment predicted above 30 mph.
Steady State Curving Empty	N/A	0.55 @ 12 deg curve -3.0 in. unbalance	0.82 @ 12 deg curve -3.0 in. unbalance	N/A	N/A	Curving performance similar to 3-piece truck.
Steady State Curving Loaded	N/A	0.42 @ 12 deg curve -3.0 in. unbalance	0.82 @ 12 deg curve -3.0 in. unbalance	N/A	N/A	Curving performance improved by rubber primary shear pads.
Curve Entry Empty	N/A	0.65 @ +3.0 in. unbalance	N/A	65% @ +3.0 in. unbalance	N/A	Occurs after all superelevation change has occurred.
Curve Exit Empty	N/A	0.58 @ +3.0 in. unbalance	N/A	68% @ +3.0 in. unbalance	N/A	Maximum L/V in center of spiral and superelevation. Minimum wheel load where superelevation is zero.
Curve Entry Loaded	N/A	0.45 @ +3.0 in. unbalance	N/A	65% @ +3.0 in. unbalance	N/A	Maximum L/V near end of spiral, with maximum curvature. Minimum wheel load where superelevation is zero.
Curve Exit Loaded	N/A	0.45 @ -3.0 in. unbalance	N/A	65% @ -3.0 in. unbalance	N/A	Maximum L/V at beginning of spiral, with maximum curvature. Minimum wheel load at center of spiral.
Dynamic Curving Empty						Derailment at balance speed and above. Inaccurate results.
Dynamic Curving Loaded	N/A	0.5 @ -3.0 in. unbalance	0.95 @ +1.5 in. unbalance	65% @ +3.0 in. unbalance	1.3 deg @ balance speed	Roll resonance speed 24.1 mph matches with Twist & Roll resonance (25 mph).

7.0 PROCEDURES FOR TRACK TESTS AND POST-TEST MODEL PREDICTIONS

7.1 INTRODUCTION

Track tests followed the general Chapter XI guidelines. Tests were performed on all the required track test zones with the car both loaded and empty. For certain test regimes, Chapter XI only requires testing in the loaded condition. Previous experience with the Car 1 tests and results of the pre-test NUCARS modeling indicated that these tests should be performed for the empty condition as well. In addition, curving tests were performed over a range of curves from 4- to 12-degrees of curvature, in order to fully evaluate the test vehicle's performance. Chapter XI requires testing on only one curve.

Prior to the track tests, all test zones were measured with an inertial track geometry measurement system developed by ENSCO Corporation. These geometry measurements were used as input for the post-test NUCARS model predictions.

After completion of the track tests and the preliminary test data analyses, post-test NUCARS predictions of vehicle performance were made using the ENSCO track geometry. In addition some vehicle suspension characteristics in the NUCARS model were altered based on dynamic suspension characteristics developed from the preliminary test results. Wheel and rail profiles used as input were also based on measurements of the actual instrumented wheel profiles and the rail profiles in the individual test zones.

All test results and NUCARS predictions presented here are compared to the applicable Chapter XI performance guidelines. The Chapter XI limiting criteria have changed slightly since the pre-test predictions were made. The criterion for wheel L/V ratio has been increased from 0.8 to 1.0, and the criterion for axle sum L/V ratio has been increased from 1.3 to 1.4. These new criteria have been applied in the following analysis. A synopsis of the criteria as used in these analyses is given in Table 7.

Table 7. AAR Chapter XI Criteria for Assessing the Requirements for Field Service (as Used for Final Analyses)

REGIME	SECTION	CRITERION	LIMITING VALUE
Hunting (empty)	11.5.2	minimum critical speed (mph)	70
		maximum lateral acceleration (g)	1.0
		maximum sum L/V axle	1.4*
Constant curving (empty & loaded)	11.5.3	maximum wheel L/V	1.0
		or maximum sum L/V axle	1.4
Spiral (empty & loaded)	11.5.4	minimum vertical load (percent)	10**
		maximum wheel L/V	1.0*
		or	
		maximum sum L/V axle	
			1.4*
Twist, Roll (empty & loaded)	11.6.2	maximum roll (deg) ***	6
		maximum sum L/V axle	1.4
		minimum vertical load (percent)	10**
Pitch, Bounce (loaded)	11.6.3	minimum vertical load (percent)	10**
Yaw, Sway (loaded)	11.6.4	maximum L/V truck side	0.6*
		maximum sum L/V axle	1.4*
Dynamic curving (loaded)	11.6.5	maximum wheel L/V	1.0*
		or maximum sum L/V axle	1.4*
		maximum roll (deg) ***	6
		minimum vertical load (percent)	10**
Vertical curve	11.7.2	To Be Determined	
Horizontal curve	11.7.3	To Be Determined	

*Not to exceed indicated value for a period greater than 50 milliseconds per exceedence
**Not to fall below indicated value for a period greater than 50 milliseconds per exceedence
***Peak-to-peak

7.2 TEST CONSIST

The test train consisted of a locomotive, the DOT 211 instrumentation car, the PSMX 111 test vehicle, and a hopper car used as a buffer car. The buffer car was included only for those tests for which Chapter XI requires one. The PSMX 111 test car is shown in Figure 1. The lading for the loaded car tests consisted of expanded shale ballast. Due to the very dense nature of this material, it only filled the car to approximately three fourths of its cubic capacity.

The denser lading would cause the center of gravity to be lower than might be expected for a load of coal. In addition the roll and pitch moments of inertia would be somewhat less due to the volume occupied by the lading being less than for a normal coal load. These factors would probably reduce the tendency of the car to roll relative to the same car filled to its cubic capacity. These might also lead to a slightly increased roll resonant frequency.

7.3 INSTRUMENTATION

Instrumentation installed on PSMX 111 was based around two instrumented wheel sets designed to measure vertical, lateral and longitudinal loads at the wheel/rail interface. These were installed in the lead truck for all tests, as shown in Figure 60. Chapter XI normally requires the instrumented wheel sets to be installed in the lead axle of each truck, except for the yaw and sway, and dynamic curving tests, which require both axles of the lead truck to be instrumented. This deviation was done to allow measurement of all the forces in a truck under all conditions. This permitted better comparisons with the model predictions. This arrangement also permitted the calculation of dynamic suspension characteristics from the test data.

Additional instrumentation included displacement transducers measuring deflections of various suspension elements, accelerometers to measure lateral car body and axle motions, and roll gyro's to measure car body roll angles. Figures 2 and 60 also show some of these transducers in place to measure the vertical spring deflections and the lateral and longitudinal deflections of the primary shear pads at each bearing adaptor. Appendix 3 provides a complete list of the instrumentation used for the tests. This is considerably more extensive than the minimum required by Chapter XI. The extra instrumentation was installed to gain an understanding of the vehicle's dynamic behavior and to permit dynamic measurement of various suspension characteristics.

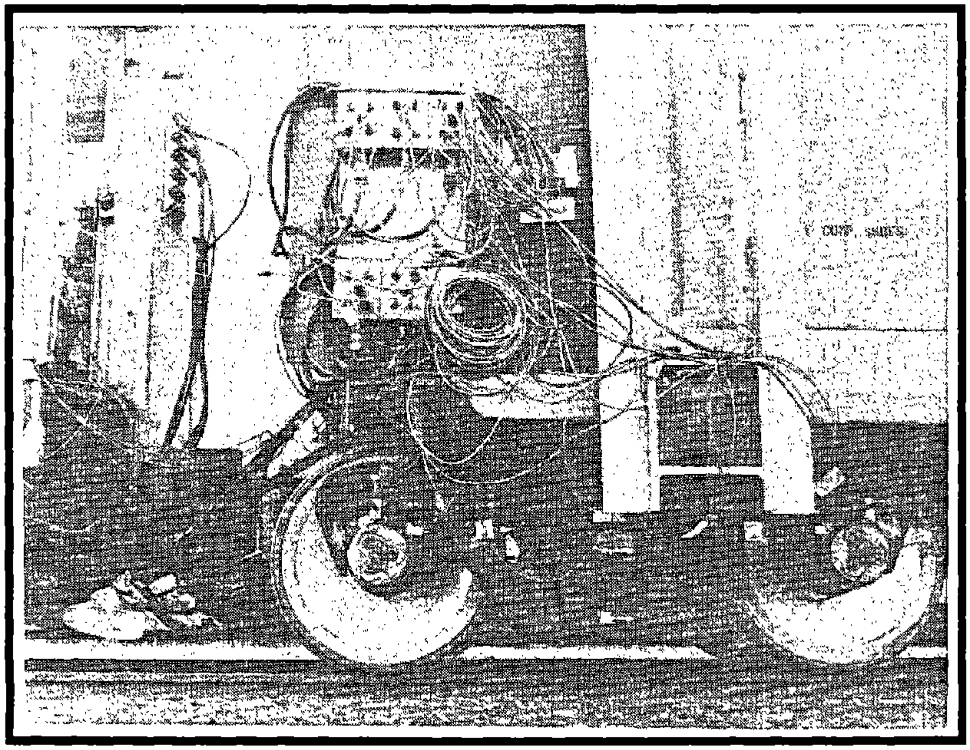

Figure 60. Instrumented Wheel Sets Installed in the PSMX 111 Aluminum Coal Gondola

All data was sampled digitally at 500 Hz and stored for later analysis on digital disks. Data collection and storage was accomplished using the HP 3000 based computer system shown in Figure 61. Selected data channels were also displayed on strip chart recorders for monitoring test safety.

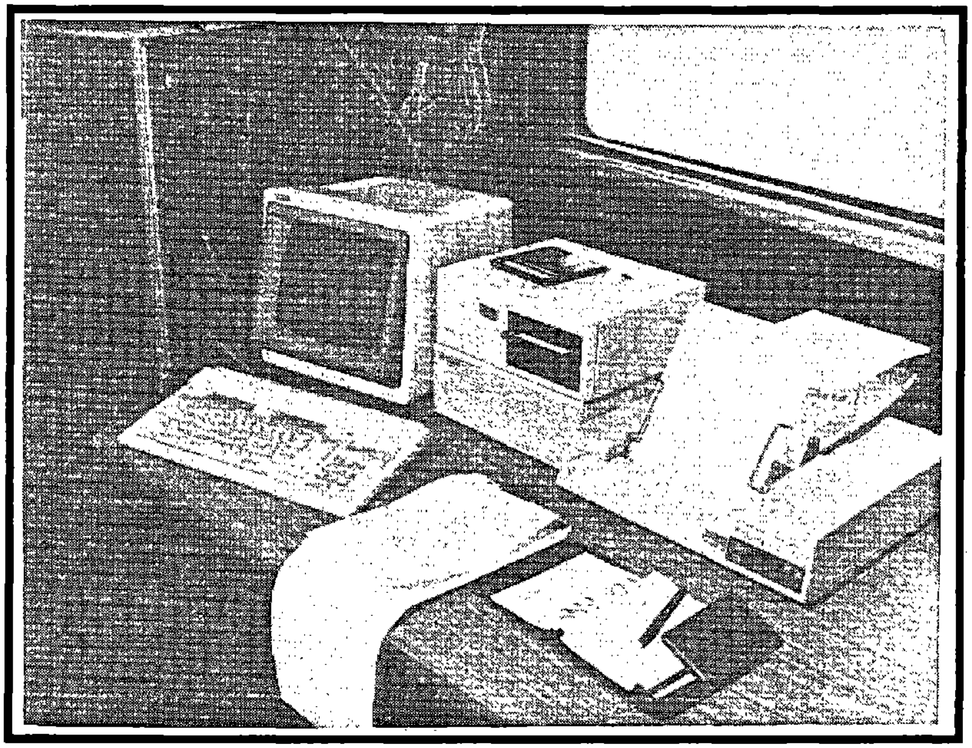

Figure 61. HP 3000 Desk Top Computer Based Data Collection System

Unlike previous test projects, the data from the instrumented wheel sets was not preprocessed to calculate vertical and lateral wheel forces. Instead the raw strain signals were recorded, and all forces were calculated post-test. This was done because doubt had arisen about the accuracy of the software used in the microprocessors which preprocessed the strain signals into lateral and vertical wheel forces. This made actual testing more cumbersome because the wheel forces had to be calculated after each test run to determine whether unsafe limits had been achieved.

Video cameras were installed on arms in front of the lead wheels of the lead truck. Video tapes were made of each run to aid in determining the lateral position of the wheels relative to the rails.

For the curving tests, devices for measuring wheel set AOA relative to the track were installed in the 4- and 10-degree curves. These allow measurement, at a single location in the track, the angle of attack of each wheel set in a train. The AOA frames, shown in Figure 62, have small light beams which shine across the rail heads on each rail. As each wheel passes through the light beams, the time of passage and difference in time between left and right wheel allow calculation of wheel speed and AOA. Data for each train pass is calculated and stored for later analysis.

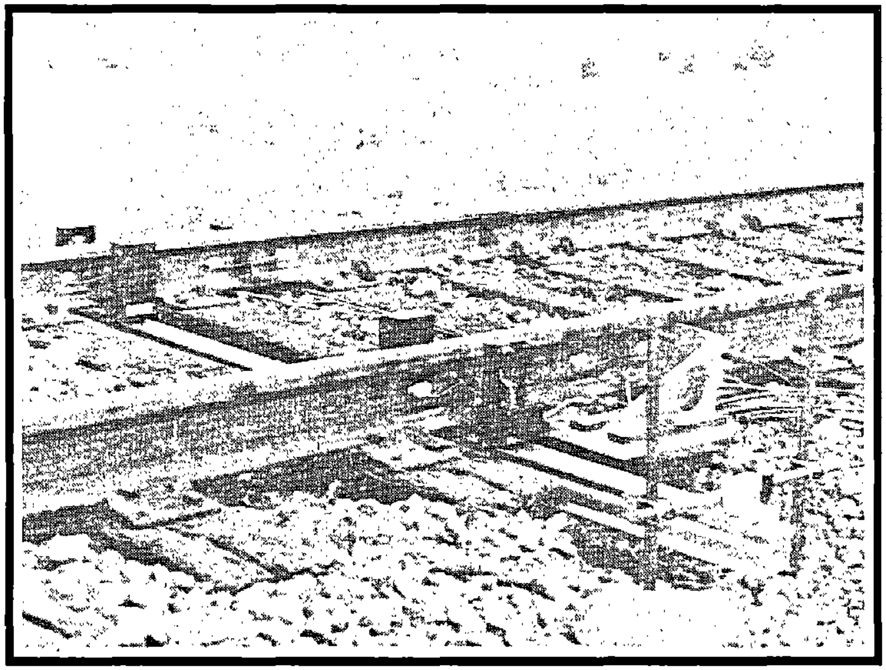

Figure 62. Wheel Set AOA Measurement System Installed in the Track

7.4 TEST DATA ANALYSIS

The first step in test data analysis was to process the raw wheel set strain signals to calculate the vertical and lateral wheel/rail interaction forces at each wheel. Initial attempts used software that was functionally identical to the software used with the microprocessors that are frequently used with the instrumented wheel sets to provide real time processing of the strain signals. The standard microprocessors were not used during these tests. A careful analysis of the results of this processing indicated that under many circumstances the calculated forces were incorrect. This occurred most often when the lateral to vertical force ratios (L/Vs) were high. This was determined to be due to errors in the software and some of the underlying principles applied in the analysis.

It was therefore decided to attempt to develop new software to correct this problem. This process, which was conducted with AAR funds, took considerably longer than originally anticipated. It also resulted in the development of new designs for building instrumented wheel sets which should be more accurate than the ones used for this project. The new software was developed to interface with the TTC's existing data analysis software, which runs on HP 3000 series desk top computers.

The TTC's analysis package performs digital filtering, calculation of statistics, and arithmetical combinations of data from several measurements to create "synthetic" data channels and many other functions. Initially all test data was to be digitally filtered at 15 Hz to comply with the Chapter XI requirements. It was subsequently found that the instrumented wheel set data from the empty car tests were extremely noisy. These data were therefore filtered at 5 Hz in an attempt to obtain usable data. This was only partially successful. Therefore much of the wheel set force data for the empty car is subject to a large error band. This problem has multiple sources:

- The extreme lightness of the car is close to the minimum force measurable by the wheel sets
- The filtering may have suppressed some legitimate peaks in the data
- Loose wires were found in some of the wheel set strain gage circuits post-test, which may have caused intermittent dropouts
- A flat spot was present on one wheel during some of the tests, which may have caused occasional spikes in the output

7.5 ENSCO TRACK GEOMETRY MEASUREMENTS

Measurements of the actual track geometry were required for the post-test data analysis. The actual track was expected to deviate from the ideal geometry used in the pre-test NUCARS predictions. The measured geometry was used in a specially modified version of NUCARS.

The ENSCO track geometry measurement system (TGMS) was mounted under a loaded hopper car to provide a measurement of the track profile in its deflected condition. The basic components of this system consisted of:

- Servo Controlled Laser Gage Measurement System
- Cross Level and Curvature System (inertial gyroscope based with integrator)
- Alignment System (accelerometer and rate gyro based)
- Profile System (accelerometer based)

Data were collected on digital disk and processed post-test into the vertical and lateral rail geometry histories required by NUCARS. Data were collected for all the test zones except the single vertical bump, which was not installed at the time. During the post processing, it was discovered that the data for the yaw and sway test zone were incorrectly recorded. Therefore post-test modeling was done with mathematically defined input for these two zones.

7.6 POST-TEST NUCARS MODELING

Post-test NUCARS predictions were made for all test zones except the single vertical bump and the yaw and sway test zones using the ENSCO measured track geometry as input. NUCARS predictions for the yaw and sway, and single vertical bump were made with mathematically defined track geometry. Measurements of the instrumented wheel set cross sectional profiles were made. These were combined with measurements of the rail profiles in each test zone to give different wheel on rail profile geometries for each test zone.

The NUCARS input files describing the test vehicle were based upon those used for the pre-test predictions. A few changes were made with regard to the yaw suspension characteristics to more accurately model the PSMX 111 car. As described in Section 5.4, the yaw rotational characteristics for the bolster to side frame connection and bolster center bowl to body connection were measured under quasi-static conditions on the air bearing tables. These tests could not accurately measure the force versus velocity characteristics of these connections. To obtain more accurate characteristics for the post-test modeling, these suspension characteristics were recalculated from measurements made during the hunting, and yaw and sway track tests. This process is described in the following subsection. The recalculated characteristics were included in the NUCARS input files for the post-test modeling.

In addition, the "average" vertical suspension characteristics for the main spring groups were replaced with their individual characteristics. These were also slightly modified by changing the friction characteristics to correspond with friction levels extrapolated from the track test data. This primarily consisted of increasing the slope of the linear viscous band by a factor of 100. This slope defines the suspension velocity at which the full friction force is attained. By increasing the slope, the full friction is achieved at lower velocities, providing a more accurate representation of the true friction characteristic. This increased slope made the NUCARS simulations more time consuming however, requiring much smaller integration time steps.

The final modifications included all measured axle misalignments. The resultant loaded and empty car NUCARS input files are given in Tables 3 and 4 of Appendix 2.

Pre-test NUCARS predictions had been made using a design case new CN Heumann wheel profile running on new AREA 136-pound rails. For the post-test modeling, measurements were made of the actual rail profiles in each test zone. The profiles of the instrumented wheel sets used during the tests were also measured and averaged together to represent the actual test conditions. Individual wheel/rail profile geometries were then computed for each test zone and used for the NUCARS predictions.

7.6.1 Calculation of Dynamic Yaw and Lateral Suspension Characteristics

One shortcoming of the suspension characterization tests performed on the MSU and the air tables is the lack of dynamic input for various yaw suspension connections. This prevents the accurate measurement of the damping forces in yaw. The design of most three-piece freight trucks, such as the ones used in this test, have several yaw connections. Many previous efforts to describe these connections refer to the resistance to turning of these connections as a yaw stiffness with little regard to the damping due to friction.

Recent investigations by the AAR indicate that these connections are more accurately represented as being dominated by friction, with little if any stiffness in parallel. In the case of the bolster center bowl to body connection, there is usually no stiffness, just rotational friction. The side frame to bolster yaw connection is dominated by the friction wedges, although there is some stiffness due to the torsion of the vertical suspension springs and the deflection of the wedge control springs. This connection dominates what has traditionally been referred to as warp or lozenging stiffness.

The traditional method of measuring the breakout friction and stiffness of these yaw connections has been with air tables as described in Section 5.4. It is believed that the friction level measured on the air tables may be different than the dynamic friction level found when a vehicle is hunting or negotiating the yaw and sway test zone. It was believed that some of the difficulties experienced in the pre-test modeling of the yaw and sway, and dynamic curving test zones may have arisen from inaccurate modeling of these characteristics.

To obtain more accurate characteristics for the post-test modeling, these suspension characteristics were recalculated from measurements made during the hunting, and yaw and sway track tests. These recalculated values have been used in all the post-test NUCARS modeling.

The bases of these calculations are the forces measured by the instrumented wheel sets. The total turning moment contributed by the wheels was calculated from the longitudinal and lateral forces measured by the wheel sets. The inertial forces were calculated from the lateral accelerations measured at the bearing adaptors, and multiplied by the known masses of the various suspension components (axles, side frames, and bolster). These inertial forces were transformed into moments and then subtracted from the turning moments previously calculated to give the yaw moments being transmitted through the yaw connections.

The bolster to side frame yaw angles and the bolster to car body yaw angles were measured during the track tests. The calculated yaw moments were plotted against these yaw angles and against the calculated yaw velocities to develop dynamic stiffness and damping characteristics for these connections.

Figure 63 shows an example yaw suspension characteristic calculated by these means. This example plots the average yaw moment for the side frame to bolster connection against the yaw velocity. The resultant characteristic is the warp damping characteristic for the lead truck. Test data is from a 60 mph run with the loaded car when the truck appeared to be hunting despite the loaded condition.

The result is obviously very noisy. This noise comes from two main sources. The first source is the instrumented wheel sets which have an estimated accuracy of + /- 2000 pounds which can cause a subsequent error in the calculated moments. This problem is accentuated in the case of the empty car, because the forces acting on it are relatively small. The second source is the accelerometers used for calculating the inertial forces which were mounted on the axle bearing adaptors. These are separated from the truck side frames by the rubber primary shear pads. Thus the accelerations of the side frame inertial forces will be somewhat in error due to being based on the similar, but not identical motions, of the axles. Despite these sources of error, the results show a recognizable friction characteristic, with a rapid crossing from positive to negative moment at low rotational velocities. For higher angular velocities, the moment is relatively constant, near 150 kip-in.

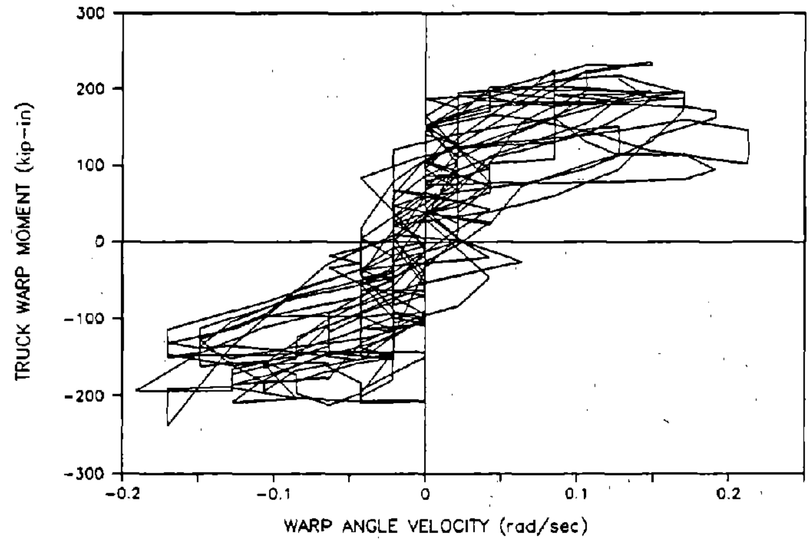

Figure 63. Measured Average Truck Warp Friction Damping Characteristic for the Loaded Car at 60 mph

Similar plots were generated for both bolster to side frame yaw connections, the center bowl yaw connection, and the lateral side frame to bolster connections all on the lead truck. No calculations were made for the trailing truck since it did not have instrumented wheel sets. Plots were generated for only a few test runs, mostly hunting, and yaw and sway runs where a significant amount of lateral and yaw motion was occurring. Average values of suspension friction damping were extrapolated from these plots and averaged together to obtain idealized characteristics, as shown in Figure 64. Also shown in Figure 64 are the characteristics obtained from the MSU and air table tests, as used in the pre-test modeling. Table 8 summarizes the new dynamic characteristics. These have been included into all post-test NUCARS predictions.

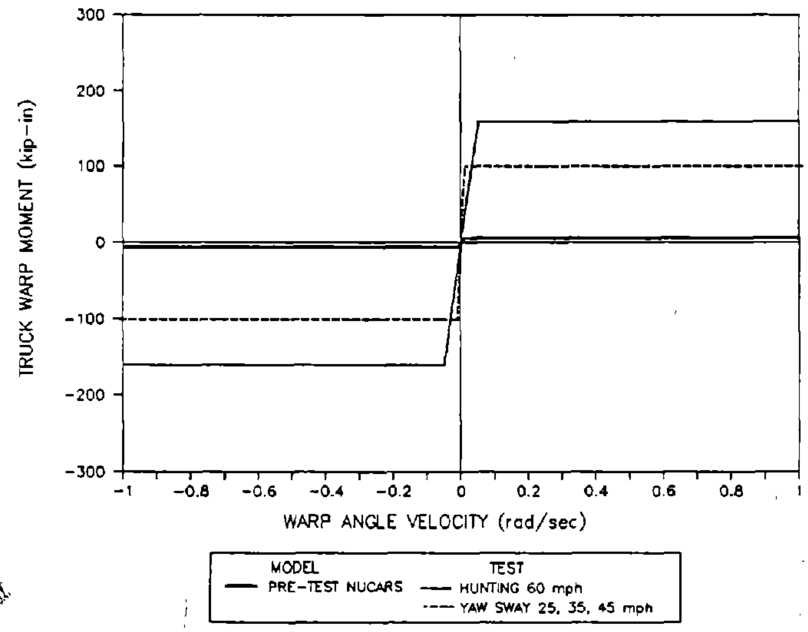

Figure 64. Extrapolated Average Truck Warp Friction Damping Characteristics for the Loaded Car in Several Different Test Conditions

It can be seen that in several instances the dynamic test data produces different stiffnesses and friction levels than measured in the original characterization tests included in the pre-test NUCARS model. It is interesting to note that in most cases the friction levels measured in the dynamic tests exceeds the levels measured during the quasi-static air table tests. The original expectation had been that the dynamic measurements would show lower levels than the air table tests. It had been expected that the shaking of the car, while in motion, would allow the friction to "break out" at lower levels. No explanation is yet available for this contradictory result.

In the case of the center bowl rotation, these values are three to five times greater than the values measured during the air table tests. This implies a very large coefficient of friction at the center bowl and/or the side bearings. It is believed that these values are much too high. This was confirmed by preliminary modeling which showed the bolster remaining locked relative to the car body when simulated on most curves. Therefore, the final modeling makes use of the original characteristics measured on the air tables.

Note that a small stiffness was measured for the center bowl yaw connection. This is probably due to the flexing of the rubber side bearing blocks.

In a recent AAR research program,[9] very accurate measurements of dynamic lateral, vertical and yaw characteristics have been made. The preliminary results are showing similar results to the characteristics calculated shown in Table 8. This confirms the validity of the test measurements.

Table 8. Comparison of Quasi Static (Original Model) and Measured Dynamic Suspension Characteristics

Suspension Characteristics	STIFFNESS		DAMPING	
	Original Model	Dynamic Test	Original Model	Dynamic Test
Empty Truck Warp	10.98×10^6 and 15.14×10^6 lb-in/rad (2 stages)	10.98×10^6 and 15.14×10^6 lb-in/rad (2 stages)	3.0×10^4 lb-in friction	5.0×10^4 lb-in friction
Loaded Truck Warp	10.98×10^6 and 15.14×10^6 lb-in/rad (2 stages)	10.98×10^6 and 15.14×10^6 lb-in/rad (2 stages)	3.0×10^4 lb-in friction	1.0×10^5 lb-in friction
Empty Center Bowl Yaw	0	1.0×10^4 lb-in/rad	3.616×10^4 lb-in friction	2.0×10^5 lb-in friction
Loaded Center Bowl Yaw	0	2.0×10^4 lb-in/rad	1.985×10^5 lb-in friction	1.0×10^6 lb-in friction
Empty Bolster to Side Frame Lateral	9000.0 lb/in	6666.0 lb/in	3000.0 lb friction	3000.0 lb friction
Loaded Bolster to Side Frame Lateral	1.8×10^4 lb/in	6666.0 lb/in	6000.0 lb friction	13000.0 lb friction

8.0 TEST AND MODEL RESULTS

Results for the track tests are presented in comparison with the post-test NUCARS predictions. In most instances, the data are compared to the Chapter XI limiting criteria for the test zones. A few additional data channels are also shown to enhance the comparisons. In most instances these compare the deflections of various suspension components during the test with the deflections predicted by NUCARS.

8.1 LATERAL STABILITY ON UNPERTURBED TRACK (Hunting)

8.1.1 Empty Tangent Track Hunting

The NUCARS predictions and test results for the root-mean-square (RMS) axle and car body lateral accelerations are shown in Figures 65 to 67. Model predictions match the test results well up to 45 mph, with the model showing slightly greater levels of acceleration than the test. The tests were halted at 55 mph because severe lateral axle motion was occurring. The sustained car body lateral acceleration levels exceeded 1.0 g peak to peak, as shown in Figure 68. These levels were not sustained for more than 20 seconds, so this result does not exceed the Chapter XI limit. The model does not show these extreme levels at 45 mph, but above that speed derailment was predicted.

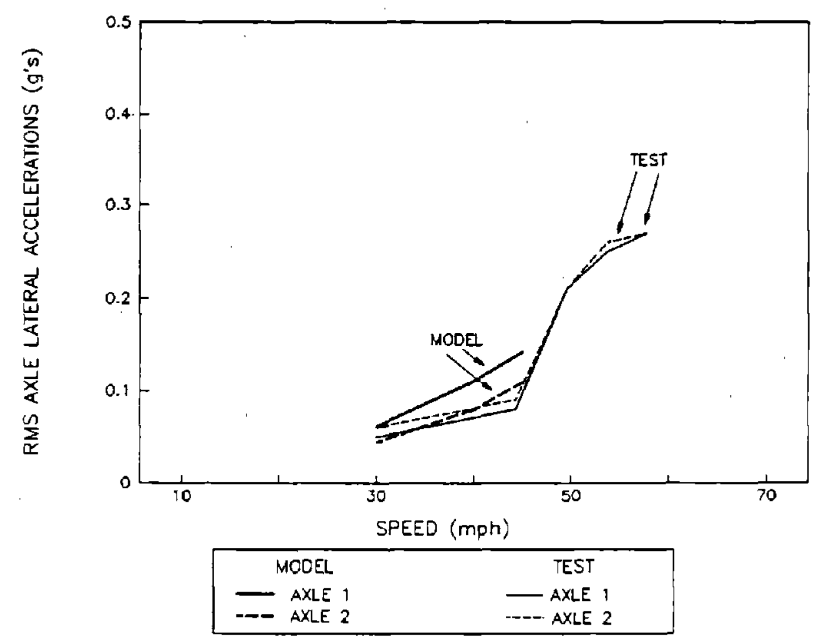

Figure 65. RMS Lateral Axle Accelerations for the Lead Truck of the Empty Car while Hunting on Tangent Track

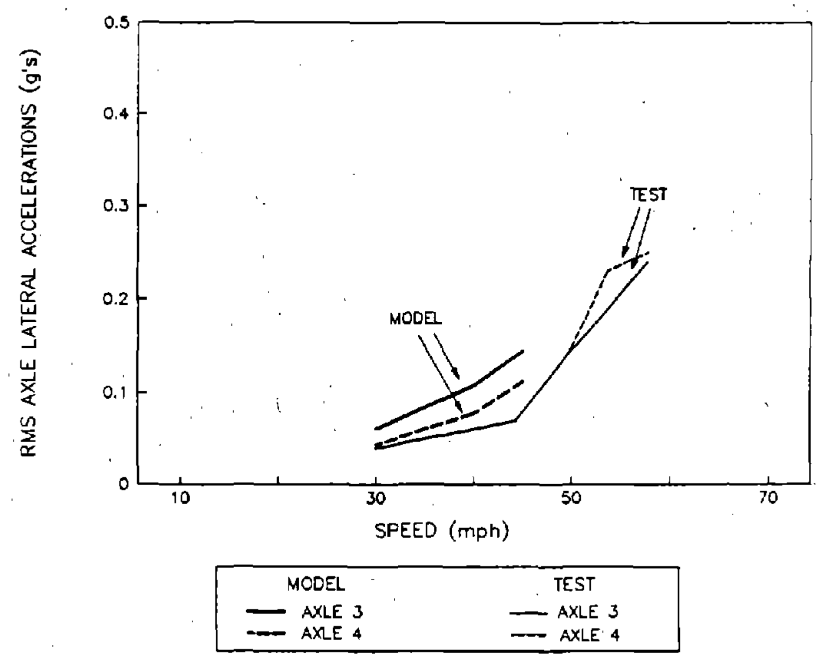

Figure 66. RMS Lateral Axle Accelerations for the Trail Truck of the Empty Car while Hunting on Tangent Track

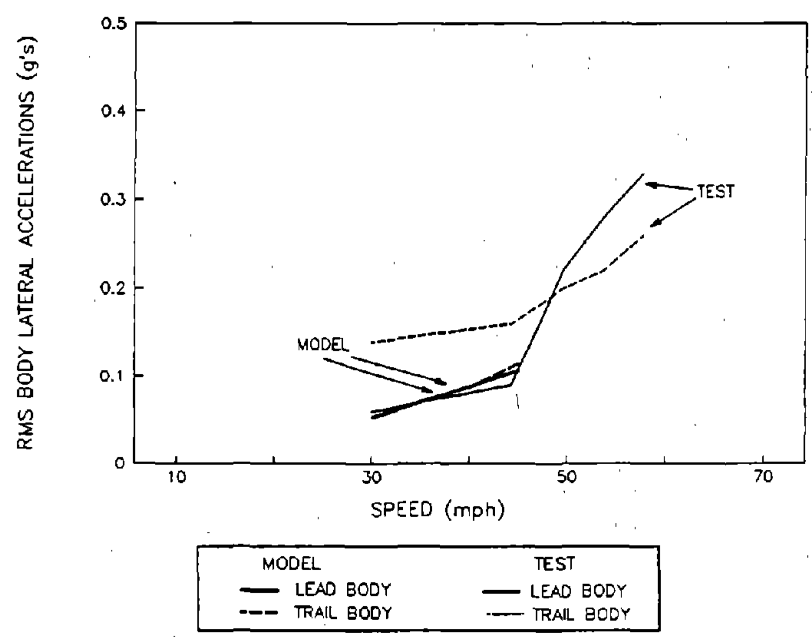

Figure 67. RMS Lateral Car Body Accelerations for the Empty Car while Hunting on Tangent Track

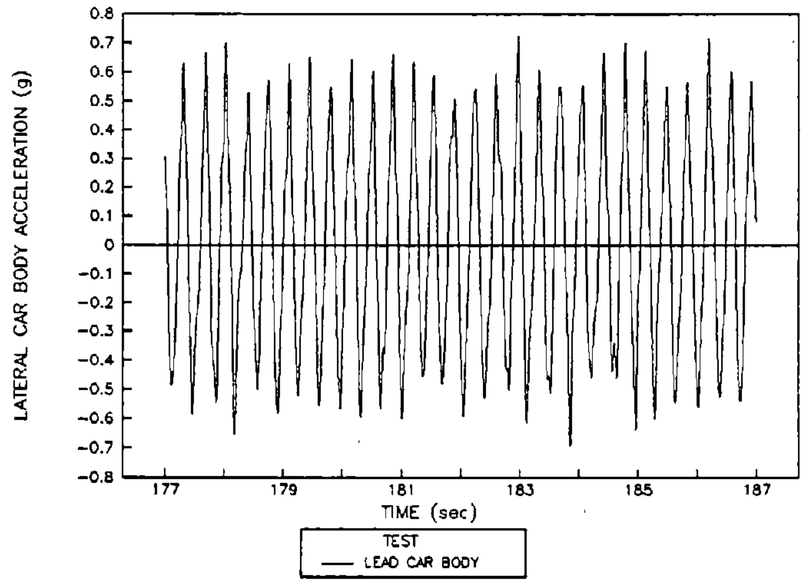

Figure 68. Measured Lateral Car Body Accelerations for the Empty Car while Hunting at 55 mph on Tangent Track

The sharp upward trend in the test results at 55 mph indicates that a threshold has been reached. The model matches these results up to this threshold and then predicts derailment.

The measured and predicted axle sum L/V ratios are shown in Figure 69. The test results indicate very high L/V ratios, exceeding the Chapter XI limit of 1.4. These extreme values, especially in the case of axle 2, are suspected to be in error. The values are so high that they should indicate derailment, which is contrary to the actual test experience. It is probable that the very light weight of this vehicle is causing errors in the force measurements. The fact that the model predicted derailment does indicate, however, that an unsafe operating condition had been achieved.

Although severe hunting was occurring, the car body accelerations did not exceed Chapter XI limits. This is almost certainly due to the primary rubber shear pads isolating the axle motions from the car body.

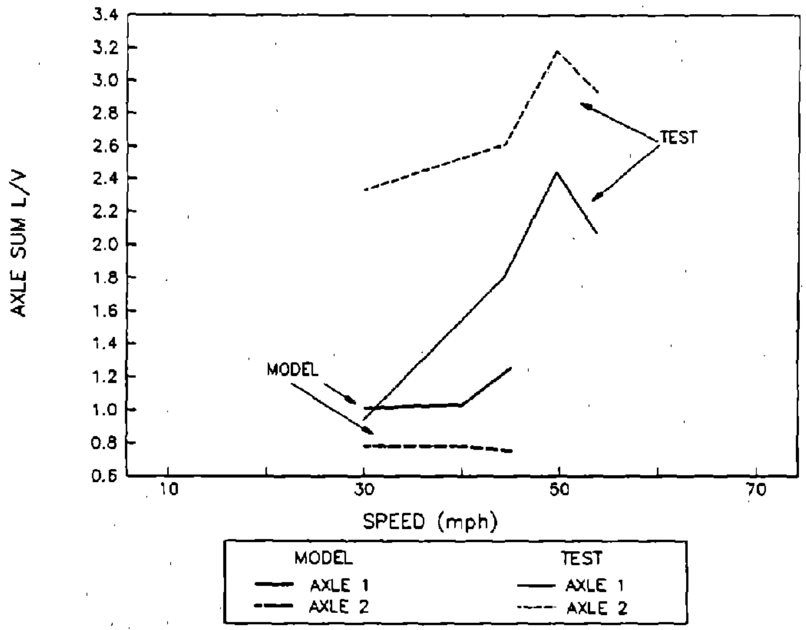

Figure 69. Axle Sum L/V Ratios for the Empty Car while Hunting on Tangent Track

8.1.2 Loaded Tangent Track Hunting

Chapter XI does not require loaded hunting tests, because previous test experience has shown that very few loaded cars hunt. Preliminary tests of the PSMX 111 and the pre-test NUCARS predictions indicated that PSMX 111 might have a lateral instability problem when loaded.

Unfortunately the test data for the loaded car was not collected correctly and was therefore not available for analysis. Subjective observations of the test crew were noted in the test log book and indicate that hunting was occurring at speeds 50 mph and above. Data recorded on strip charts during the tests indicate that car body lateral accelerations reached a maximum of 1.25 g at 65 mph, being sustained for more than 20 seconds. This is in excess of the Chapter XI limit. This hunting phenomena also occurred while trying to make test runs through the other tangent track test zones. The unsafe behavior caused testing to be limited to 60 mph in all test zones.

NUCARS simulations of loaded car hunting are shown in Figure 70. The RMS acceleration levels at speeds above 60 mph are similar to those seen during the empty car tests at 45 mph. This shows that hunting is predicted for the loaded car as well, in agreement with the test observations. In contrast with these results the pre-test predictions showed only minor oscillatory behavior and no true loaded car hunting.

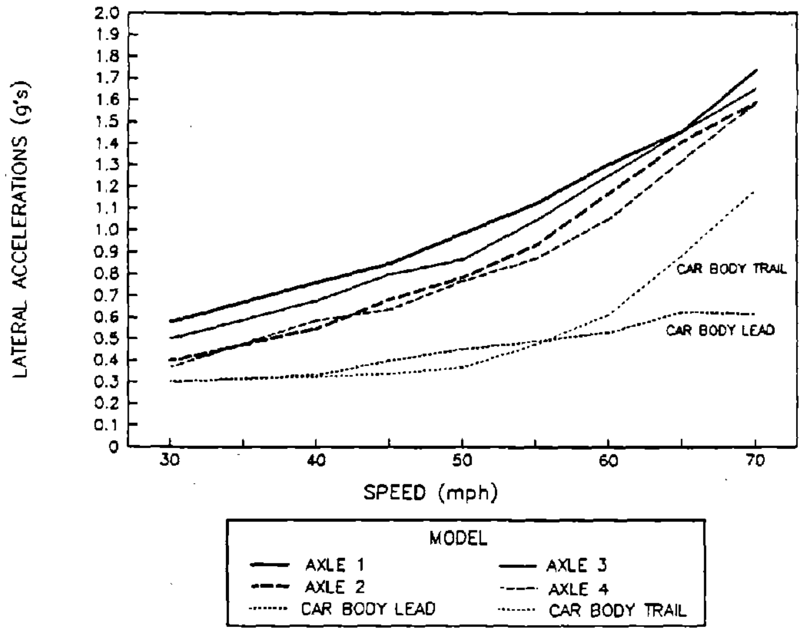

Figure 70. Predicted RMS Lateral Accelerations for the Loaded Car while Hunting on Tangent Track

8.1.3 Empty 50-Minute-Curved Track Hunting

Comparisons of RMS lateral accelerations for the test and NUCARS are shown in Figures 71 through 73. As with the tangent track results, higher acceleration levels were predicted by NUCARS than were measured. Similar to the tangent hunting tests, the data shows a sharp increase at 45 mph, which is not matched by the NUCARS predictions. However NUCARS does predict derailment at 45 mph and above.

This result is in contrast with the pre-test predictions which showed no hunting behavior. The difference is probably due to the different values used for the truck warp and lateral bolster to side frame suspension characteristics, and also due to the steeper leading edge slopes used for some of the friction damping characteristics. The method of exciting hunting was also different. For the pre-test model the car was subjected to a single lateral perturbation and then allowed to run on perfect track. Hunting was defined to occur if lateral oscillations of the car did not die out. For the post-test model the actual track roughness was allowed to excite the car throughout the length of the simulation. This more accurately simulates the actual test conditions. If lateral oscillations were excited and sustained then hunting was occurring.

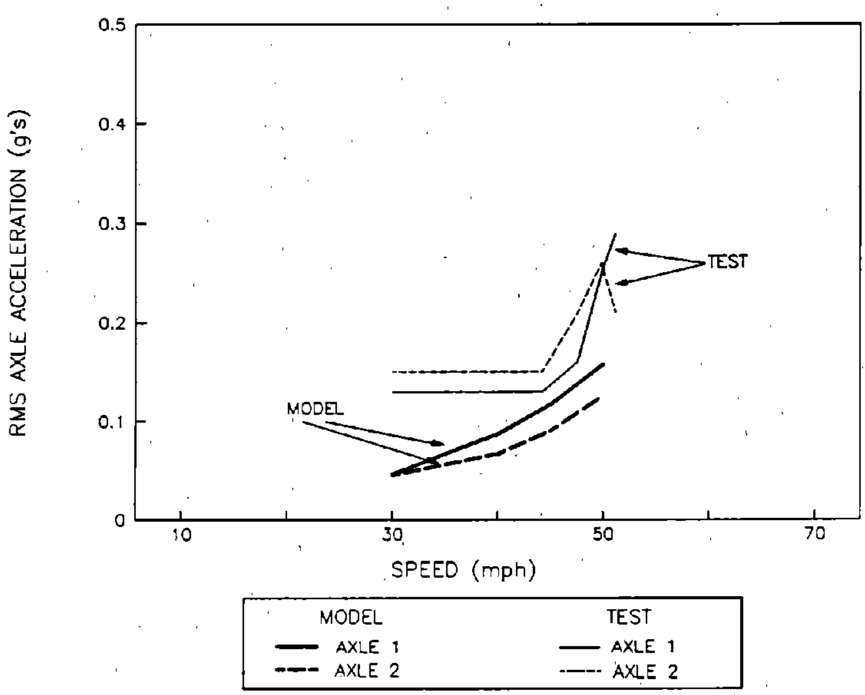

Figure 71. RMS Lateral Axle Accelerations for the Lead Truck of the Empty Car while Hunting on a 50-Minute Curve

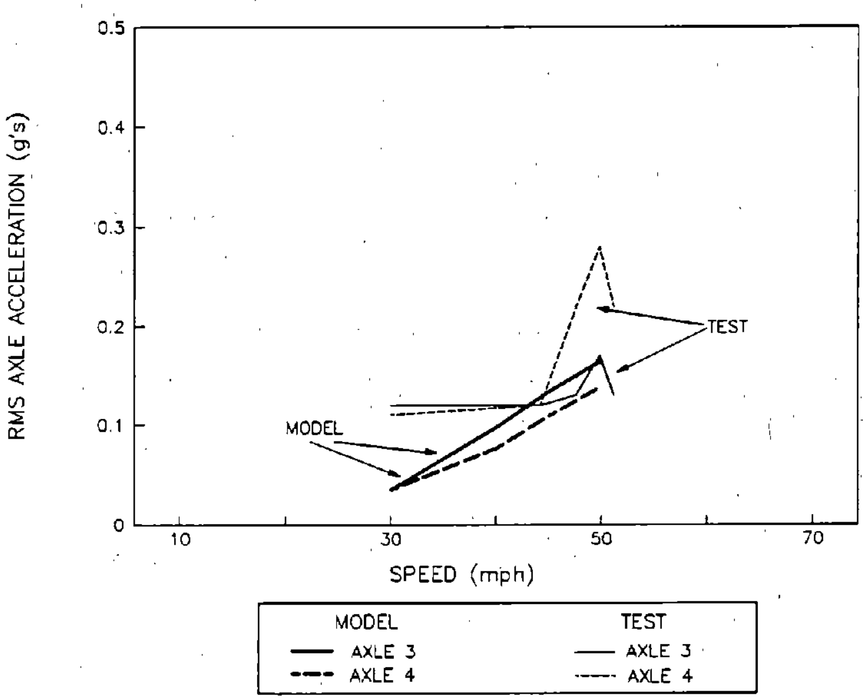

Figure 72. RMS Lateral Axle Accelerations for the Trail Truck of the Empty Car while Hunting on a 50-Minute Curve

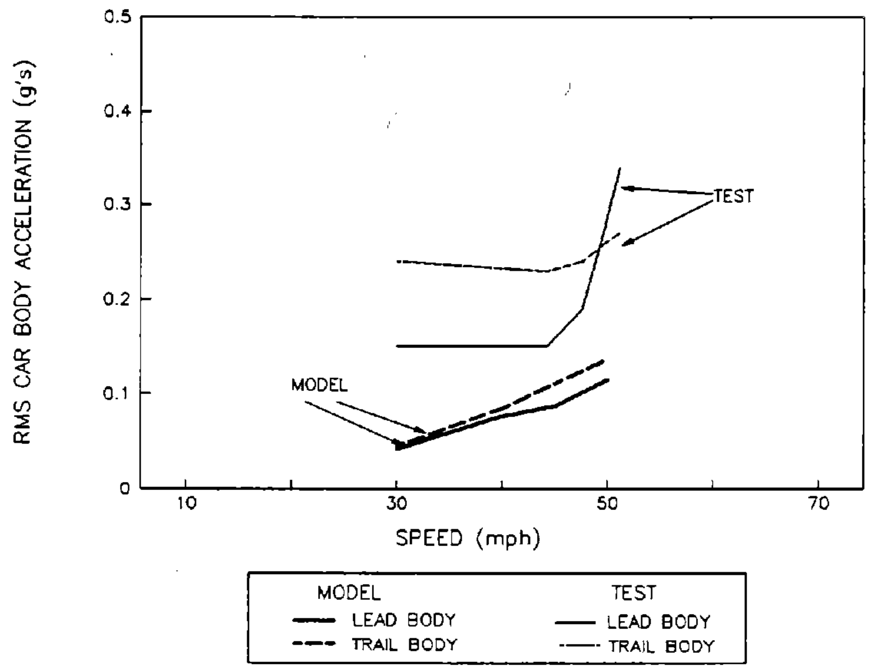

Figure 73. RMS Lateral Car Body Accelerations for the Empty Car while Hunting on a 50-Minute Curve

8.1.4 Loaded 50-Minute-Curved Track Hunting

This test was not required by Chapter XI but was conducted in conjunction with the loaded tangent hunting tests. The test data for these tests was also incorrectly recorded so no test data is available. The subjective observations of the test crew do indicate that in the curve the car did hunt at speeds from 50 mph and up.

NUCARS predictions of RMS lateral accelerations are shown in Figure 74. Results are similar to the loaded tangent hunting tests and also indicate hunting at speeds above 60 mph. These results are contrary to the pre-test predictions which did not show any loaded car hunting in the curve.

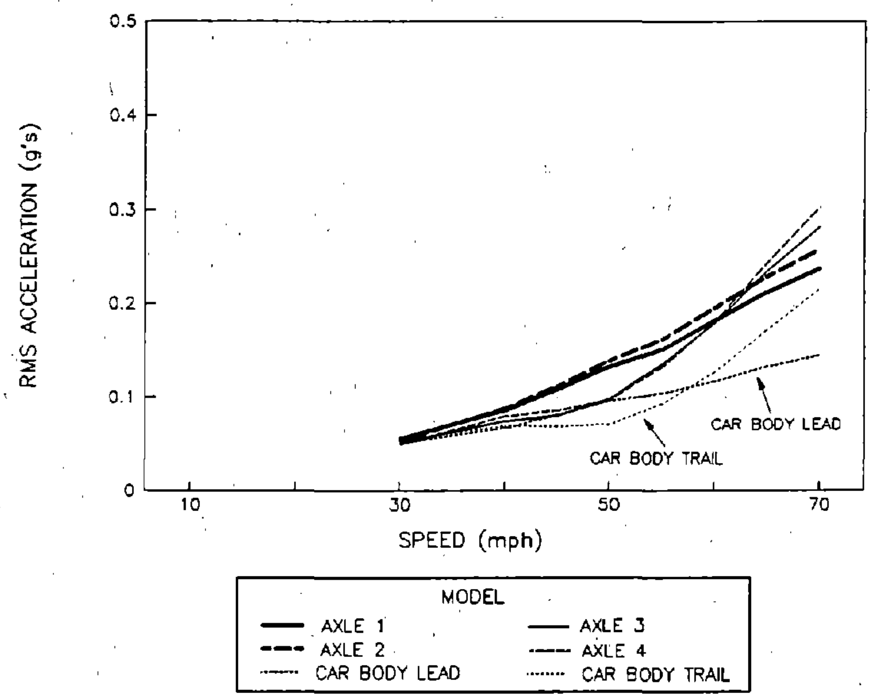

Figure 74. Predicted RMS Lateral Accelerations for the Loaded Car while Hunting on a 50-Minute Curve

8.2 PITCH AND BOUNCE

8.2.1 Empty Pitch and Bounce

The Chapter XI criterion for the pitch and bounce tests are that the minimum vertical wheel load shall not be less than 10 percent of the static load. This test is required only for loaded cars. Figures 75 and 76 show the minimum percent wheel loads for the lead and trail axles (axles 1 and 2) of the lead truck on the empty car. The NUCARS predictions indicate that up to 70 mph the minimum wheel load does not become less than 15 percent of static load, with a resonance at 60 mph. It should be noted however that with this lightweight vehicle 15 percent of the empty car static wheel load is only 750 pounds!

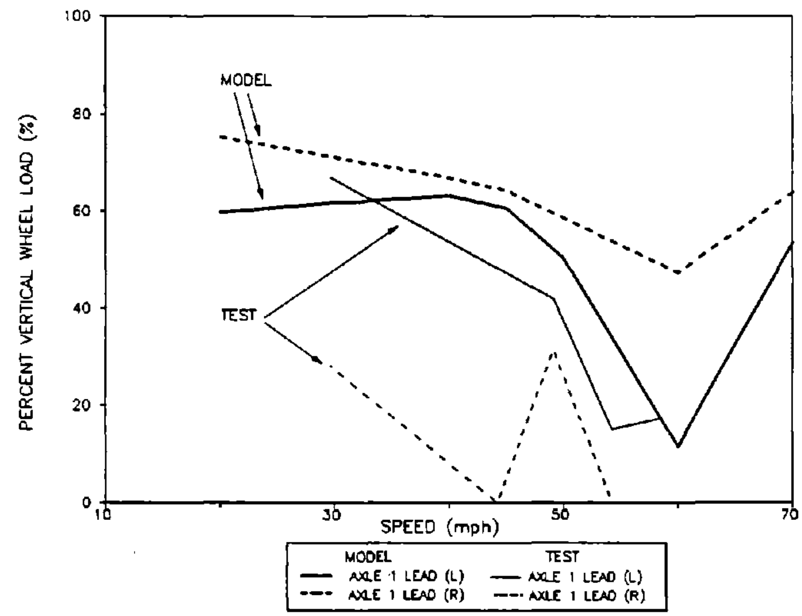

Figure 75. Minimum Percent Vertical Wheel Load for Axle 1 of the Empty Car in the Pitch and Bounce Test Zone

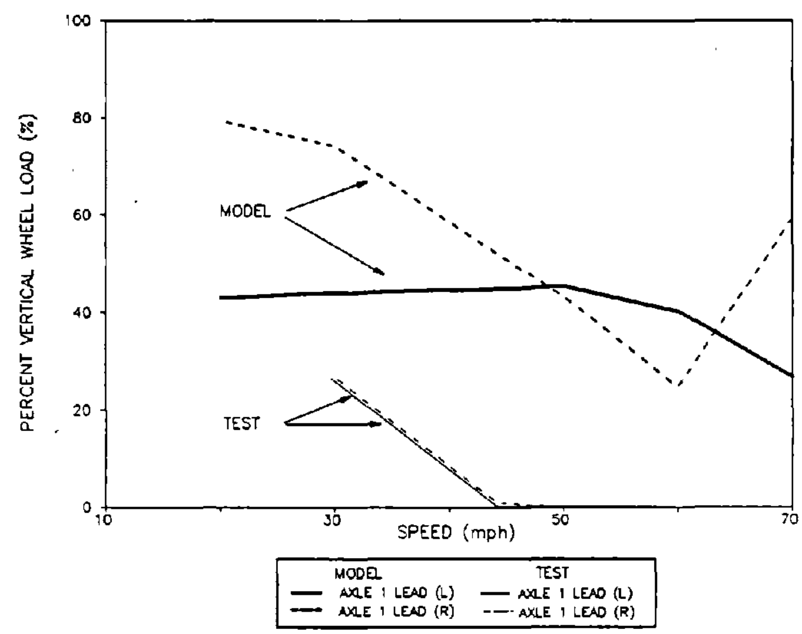

Figure 76. Minimum Percent Vertical Wheel Load for Axle 2 of the Empty Car in the Pitch and Bounce Test Zone

The test data indicates wheel lift (0% wheel load) starting at 45 mph. These are likely to be erroneous results. During these runs, three of the four measuring wheels were indicating very low wheel loads prior to the actual test zone in perfectly smooth track sections. The lead left wheel, which was functioning well, does show good correlation with the NUCARS predictions. A comparison of the predicted and measured vertical spring deflections on the lead truck is shown in Figure 77. This shows a good correlation between test and model results and reinforces the belief that the vertical force test data is erroneous. The possible sources of error in these force measurements are outlined in Section 7.4.

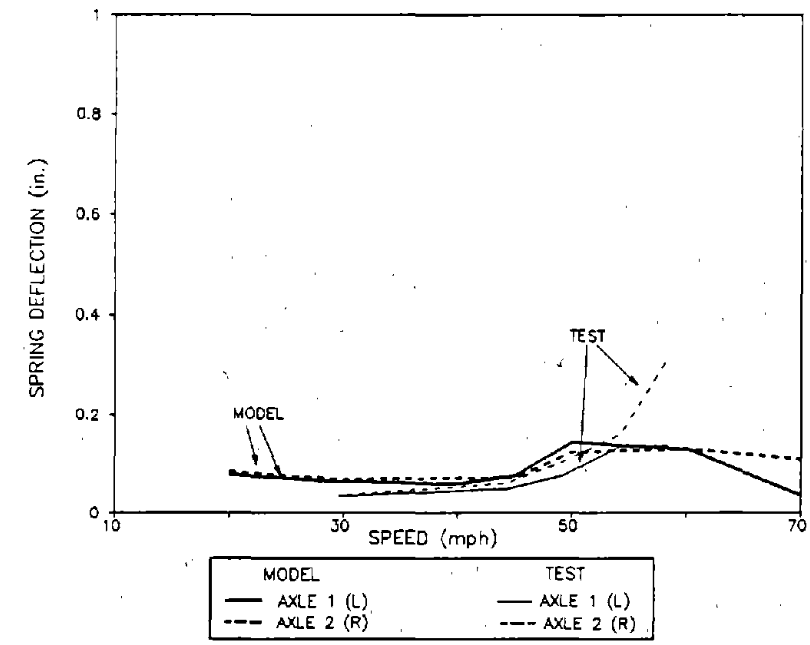

Figure 77. Maximum Peak to Peak Vertical Spring Deflections for the Lead Truck of the Empty Car in the Pitch and Bounce Test Zone

The pre-test predictions (Section 6.5.1) did not show the same trend as the post-test predictions with no true resonance apparent at speeds up to 70 mph. The post-test predictions do indicate lower minimum wheel loads for the worst case at 60 mph, but still within Chapter XI limits. The close match between post-test predictions and test measurements of spring deflections indicates that the post-test predictions are a better match.

8.2.2 Loaded Pitch and Bounce

Figures 78 and 79 show the minimum percent wheel loads for the lead and trail axles (axles 1 and 2) of the lead truck on the loaded car. The NUCARS predictions show a resonant speed of 50 mph with the minimum wheel load greater than 40 percent of static load. The test data matches the minimum loads reasonably well although the test resonant speed appears to be at 55 mph.

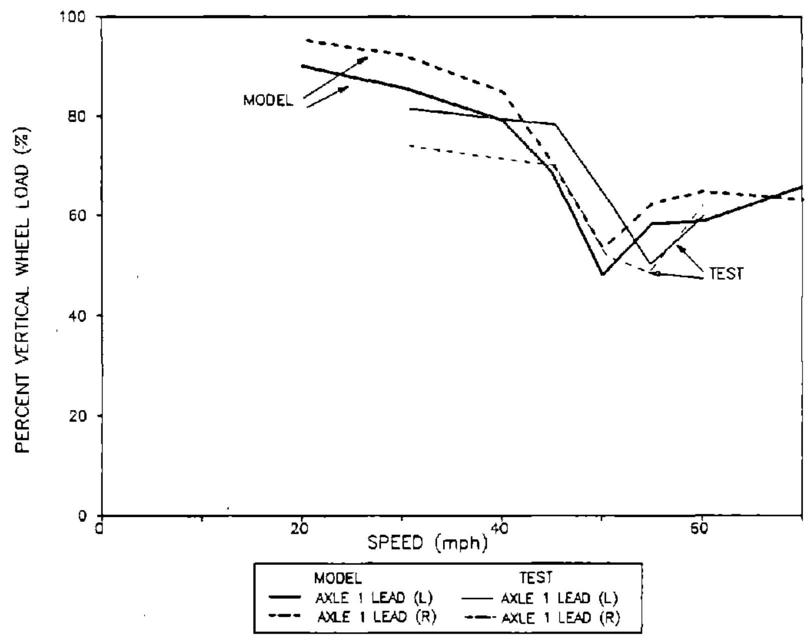

Figure 78. Minimum Percent Vertical Wheel Loads for Axle 1 of the Loaded Car in the Pitch and Bounce Test Zone

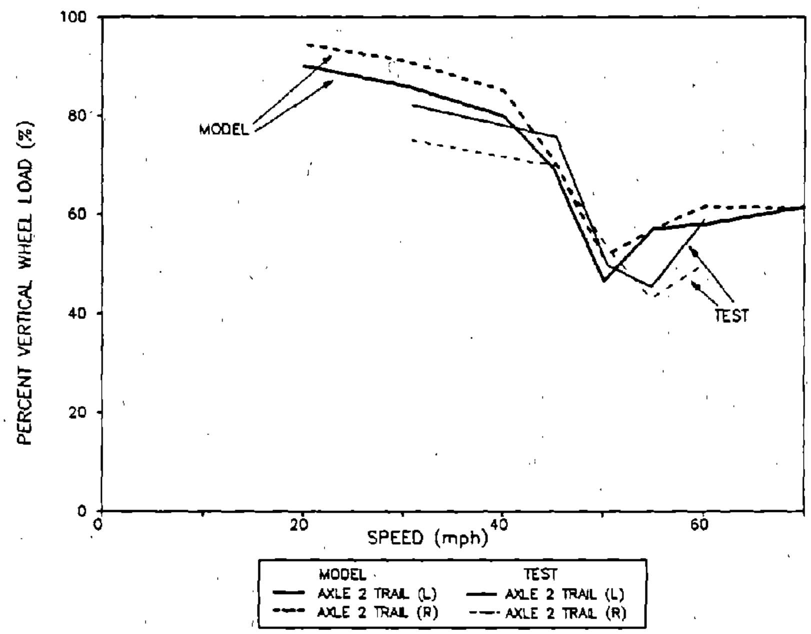

Figure 79. Minimum Percent Vertical Wheel Loads for Axle 2 of the Loaded Car in the Pitch and Bounce Test Zone

The predictions for vertical spring deflections, Figures 80 and 81, show a similar match with the test data. Predicted resonant speed is 50 mph while the test resonant speed is again 55 mph. The characteristic shape of the data curves is very similar with a narrow resonant peak. This indicates that the simulation of the friction damping with the increased linear viscous slope in the vertical suspension is correct.

The pre-test NUCARS predictions (Section 6.5.2) indicated a resonant speed near 65 mph with very little wheel unloading. Peak to peak spring deflections were also less than 1 inch. These do not match test results at all. The post-test NUCARS predictions match the test results much more closely.

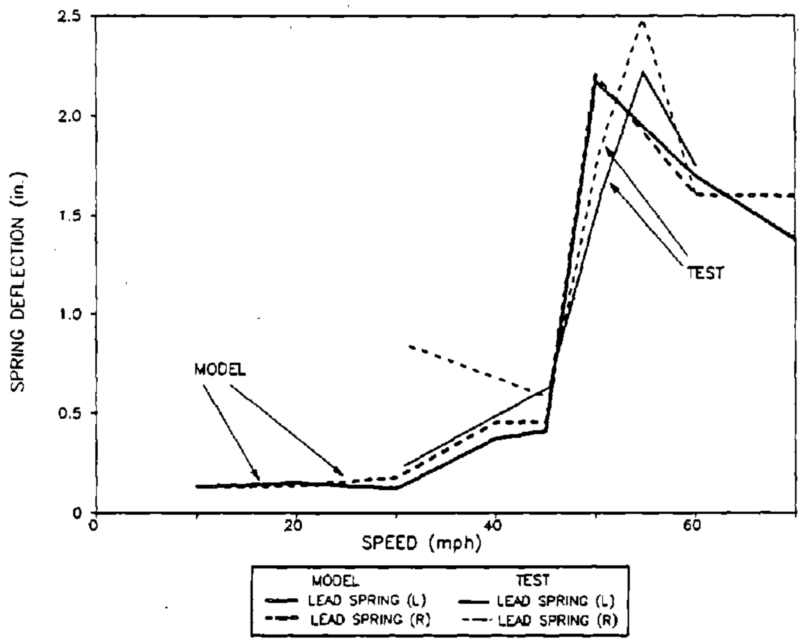

Figure 80. Maximum Peak to Peak Vertical Spring Deflections for the Lead Truck of the Loaded Car in the Pitch and Bounce Test Zone

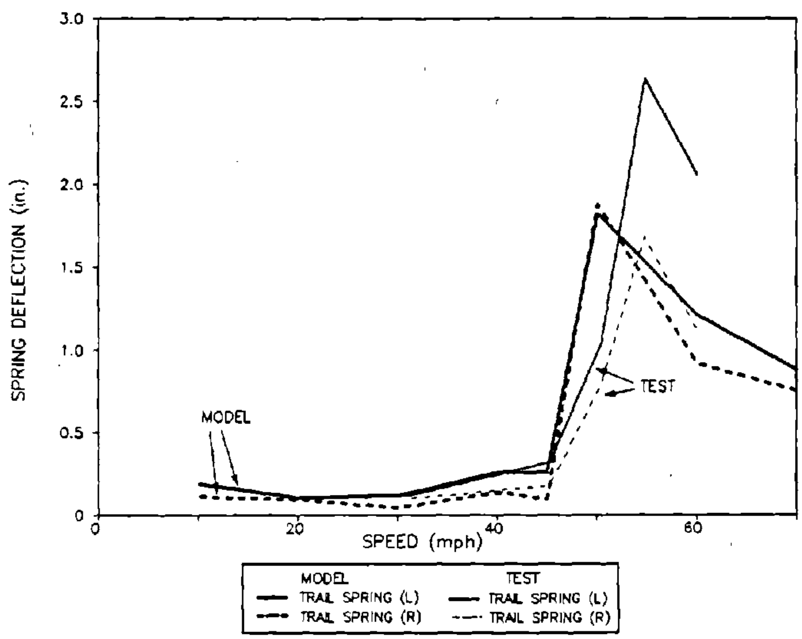

Figure 81. Maximum Peak to Peak Vertical Spring Deflections for the Trail Truck of the Loaded Car in the Pitch and Bounce Test Zone

8.3 SINGLE BOUNCE

The single bounce test zone was not measured by the ENSCO TGMS. Therefore the post-test modeling results presented in this section were made with mathematically defined track geometry in the same manner as the pre-test NUCARS predictions. Wheel/rail cross sectional geometries measured at the test zone were used however.

8.3.1 Empty Single Bounce

Figures 82 and 83 show the minimum percent wheel loads for the lead and trail axles (axles 1 and 2) of the lead truck on the empty car in the single bounce test. The NUCARS predictions indicate that at speeds of 40 mph and up there is wheel lift (zero vertical wheel load). The test data is less clear. On two wheels, the test data indicates wheel lift for all speeds, while for the other two wheels, wheel lift does not occur until above 35 mph. As discussed in Section 7.4, it is believed that the wheel force data is probably in error due to the extremely low static wheel loads.

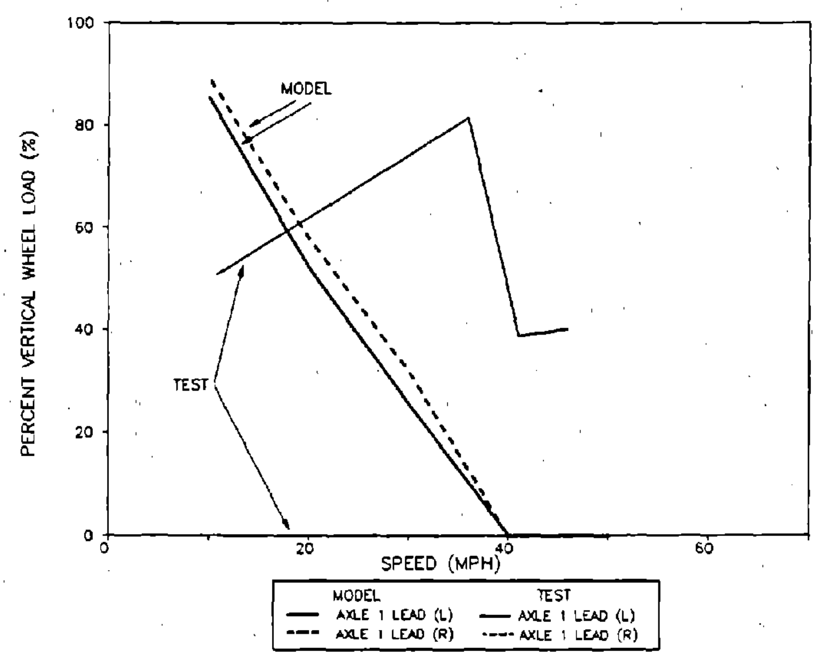

Figure 82. Minimum Percent Vertical Wheel Loads for Axle 1 of the Empty Car Passing over the Single Bump

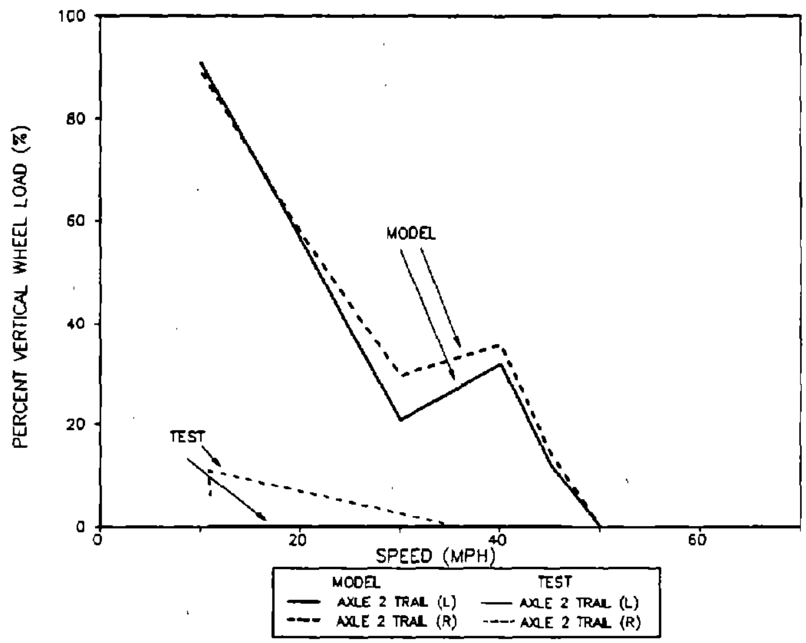

Figure 83. Minimum Percent Vertical Wheel Loads for Axle 2 of the Empty Car Passing over the Single Bump

The predicted vertical suspension deflection data shown in Figures 84 and 85 do match well with the test data. The very small amplitudes of motion indicate that the friction wedges stay "locked up," even at the higher vehicle speeds. Eventually this causes the wheel sets to lift from the rails as predicted, near 40 mph.

Good correlation is shown between predicted and measured suspension deflections, while the measured wheel set forces appear erroneous. Tests of an empty car over the single bounce test zone are not required by Chapter XI, however, the criterion of 10 percent minimum wheel load appears to have been exceeded at 40 mph. Uncertainty about the test data makes it unclear whether this also occurred at lower speeds.

The pre-test predictions (Section 6.6.1) correlate fairly well with the post-test predictions. Wheel lift was predicted at a slightly higher speed of 45 mph. The spring deflections do not match the test as well, with greater deflections being predicted pre-test. This is probably due to the revised post-test friction wedge description with its steeper linear viscous leading edge slope. This keeps the suspension locked up until resonance is reached. The post-test modeling matches the test results better in this regard.

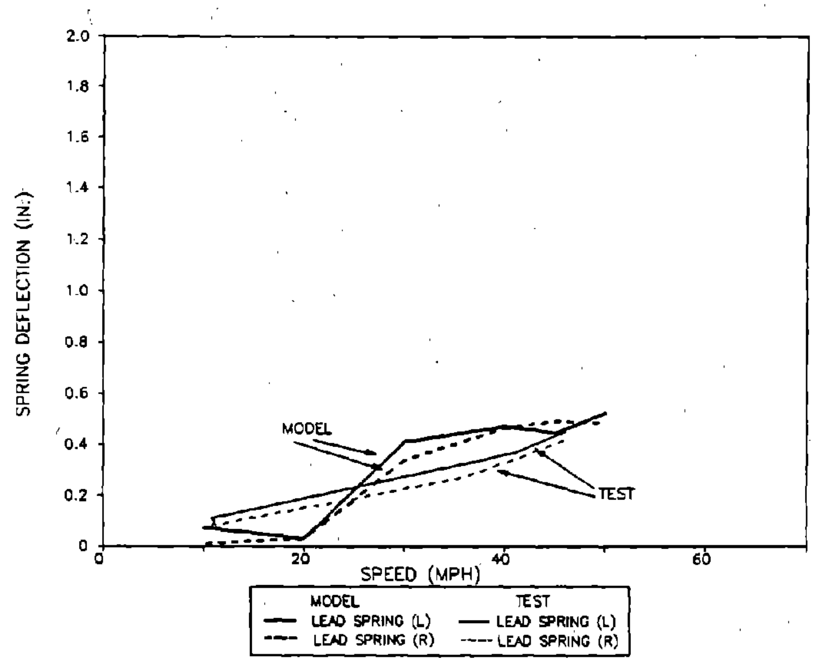

Figure 84. Maximum Peak to Peak Vertical Spring Deflections for the Lead Truck of the Empty Car Passing over the Single Bump

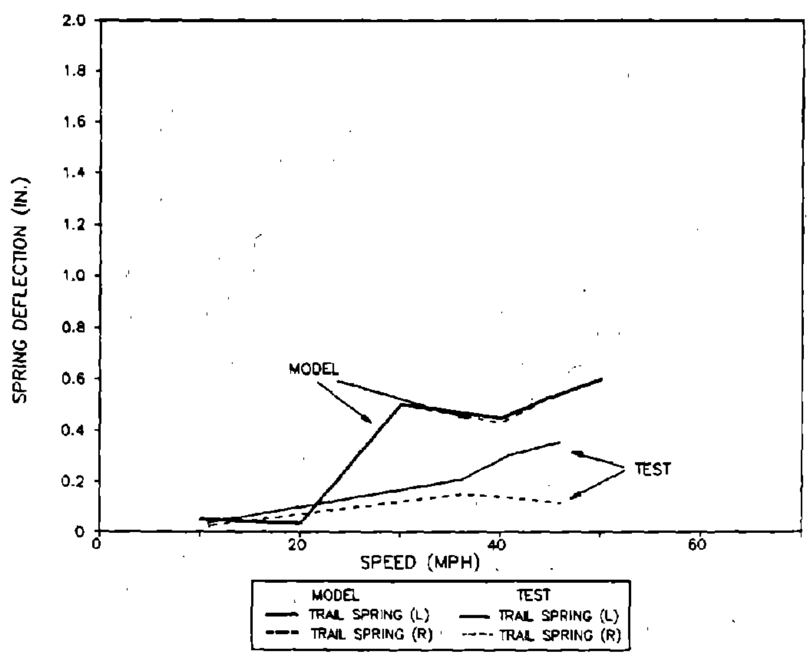

Figure 85. Maximum Peak to Peak Vertical Spring Deflections for the Trail Truck of the Empty Car Passing over the Single Bump

8.3.2 Loaded Single Bounce

Figures 86 and 87 show the minimum percent wheel loads for the lead and trail axles (axles 1 and 2) of the lead truck of the loaded car in the single bounce test zone. The NUCARS model predicts that at 70 mph the minimum wheel load reaches 34 percent of static load, with the trend indicating that higher speeds will further decrease the wheel loads. The test data for two wheels match the predictions, while the others show much greater wheel unloading. The lead right wheel shows wheel lift between 35 and 40 mph. This exceeds the Chapter XI limiting criteria of 10 percent minimum wheel load.

The predicted vertical suspension deflection data shown in Figures 88 and 89 match the test data better, with the lead test data being greater than the predictions. The trail springs showed the reverse, with less deflection than was predicted. This indicates a good correlation between test and the NUCARS model. One possible reason for this disparity in correlation could be the values of track stiffness and damping assumed in the model. The test zone is designed with a very rapid change in vertical profile which acts very much like an impact to the suspension. Inaccuracies in modeling the vertical stiffness in the track could alter the effects of this rapid change.

Pre-test predictions were very similar to the post-test predictions.

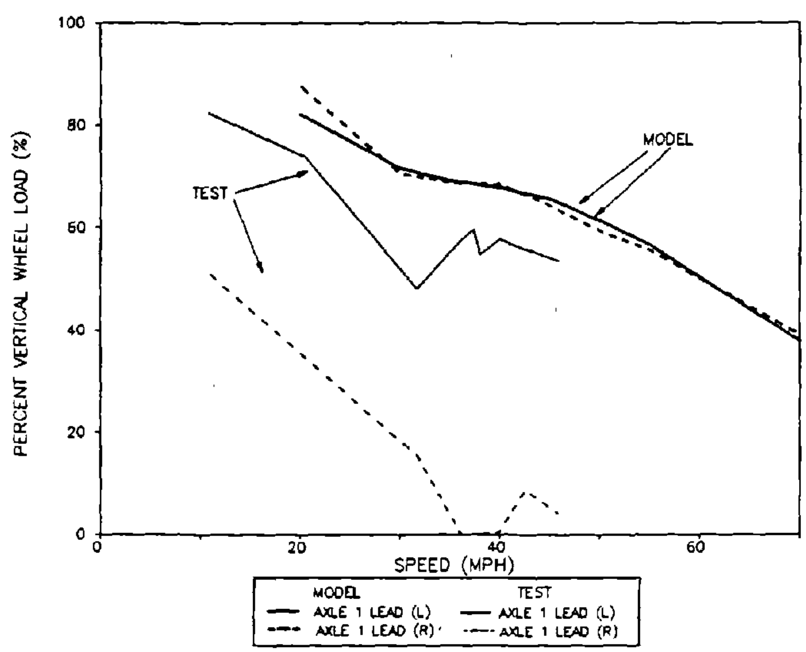

Figure 86. Minimum Percent Vertical Wheel Loads for Axle 1 of the Loaded Car Passing over the Single Bump

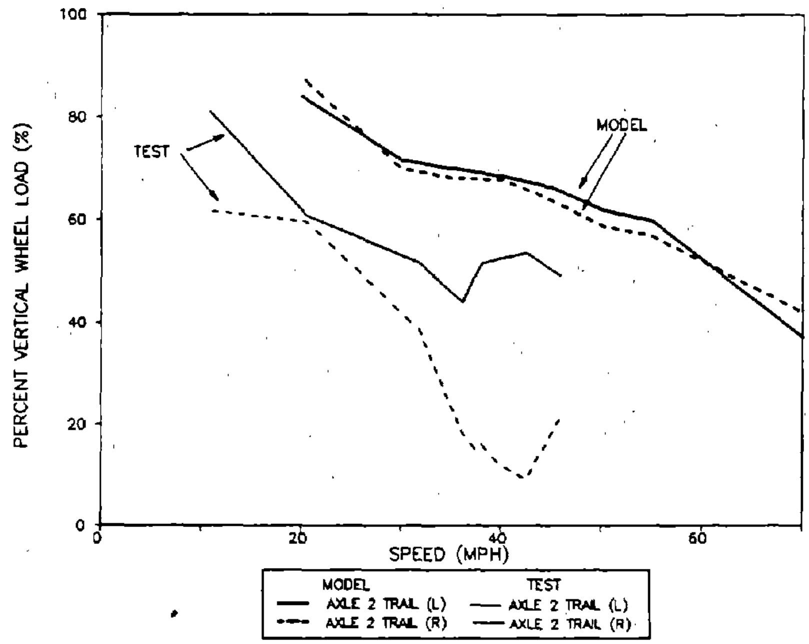

Figure 87. Minimum Percent Vertical Wheel Loads for Axle 2 of the Loaded Car Passing over the Single Bump

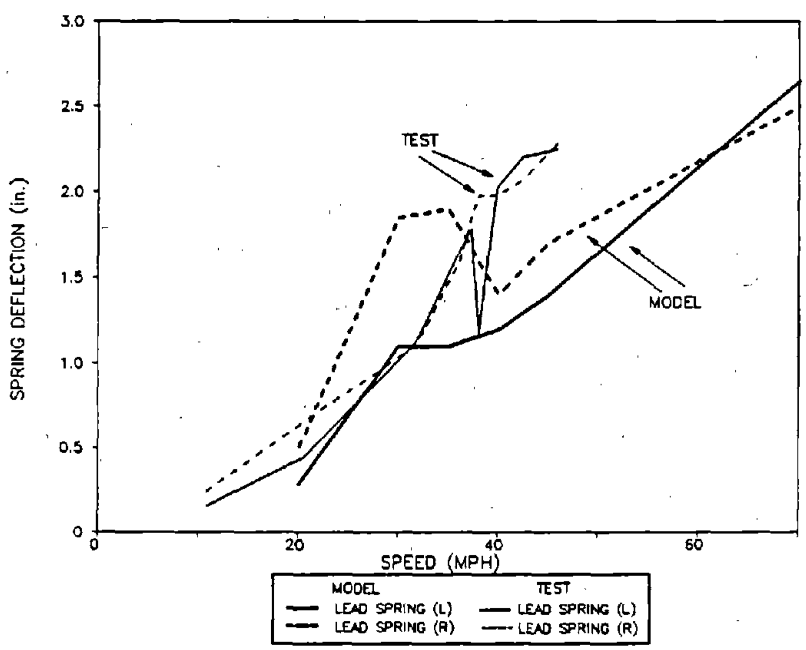

Figure 88. Maximum Peak to Peak Vertical Spring Deflections for the Lead Truck of the Loaded Car Passing over the Single Bump

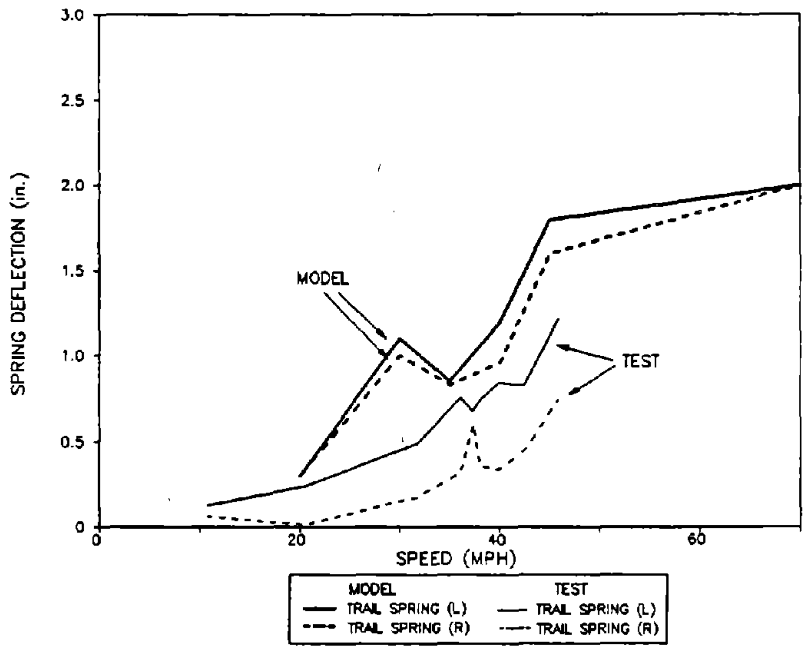

Figure 89. Maximum Peak to Peak Vertical Spring Deflections for the Trail Truck of the Loaded Car Passing over the Single Bump

8.4 TWIST AND ROLL

8.4.1 Empty Twist and Roll

Figures 90 and 91 compare the predicted lead truck (axles 1 and 2) vertical wheel loads with the loads measured during the empty twist and roll track tests. Predicted resonance is between 30 and 35 mph, with a minimum wheel load of 22 percent of the static load. The test results match this resonant speed showing wheel lift at speeds from 30 to 50 mph. There is some doubt about the accuracy of the measured wheel force data for the empty car, for the reasons stated in previous Sections.

Comparisons of predicted vertical suspension deflections in Figures 92 and 93 also show good agreement on resonant speed, although the model predictions show a lower amplitude of motion. This is probably due to an inaccuracy in the representation of the action of the friction wedge.

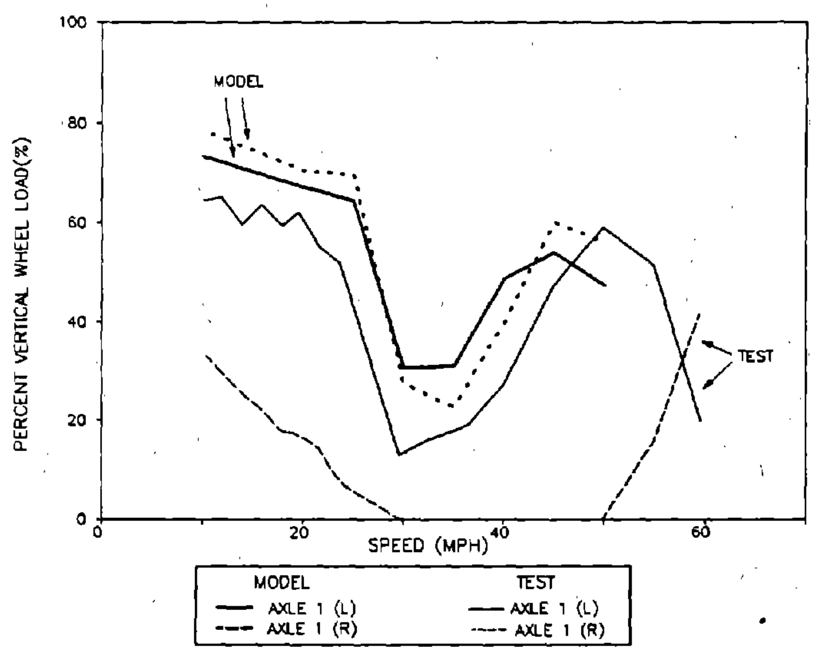

Figure 90. Minimum Percent Vertical Wheel Loads for Axle 1 of the Empty Car in the Twist and Roll Test Zone

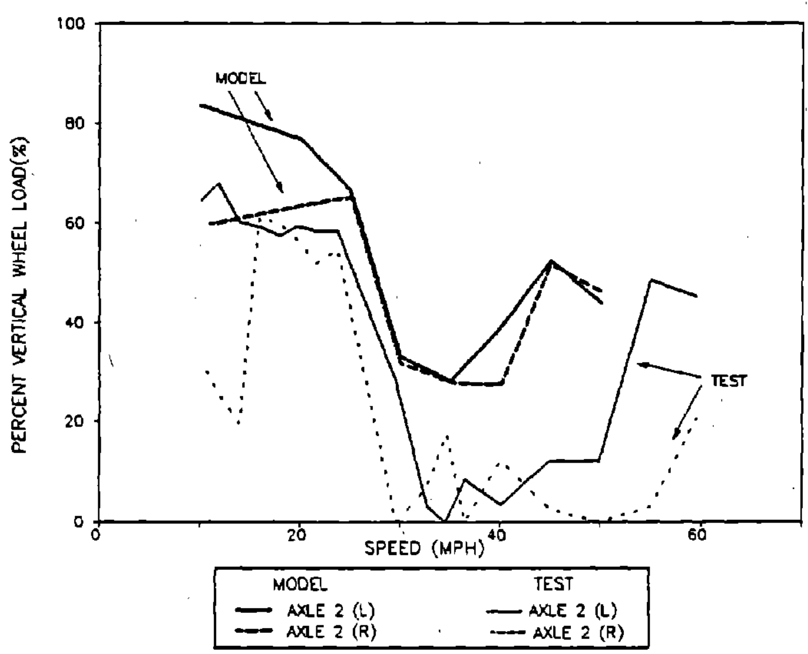

Figure 91. Minimum Percent Vertical Wheel Loads for Axle 2 of the Empty Car in the Twist and Roll Test Zone

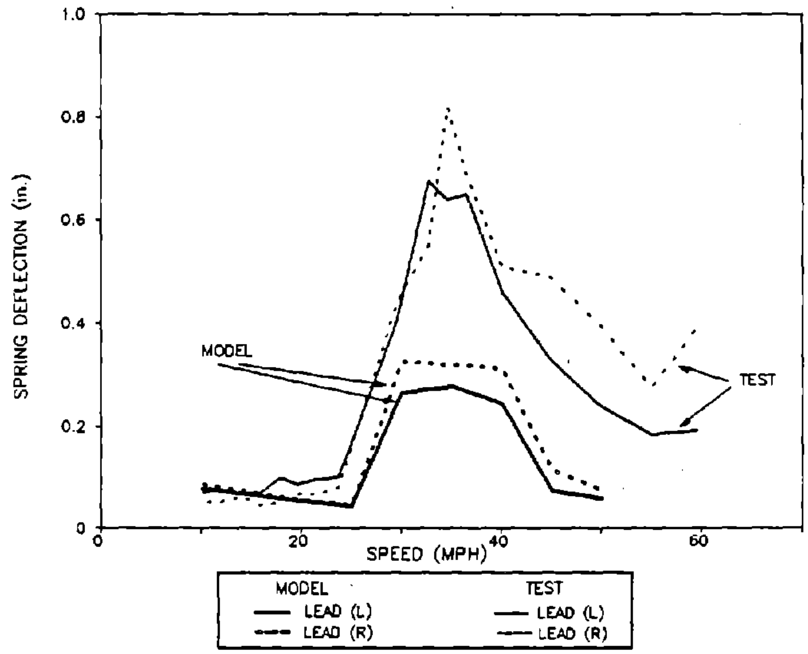

Figure 92. Maximum Peak to Peak Vertical Spring Deflections for the Lead Truck of the Empty Car in the Twist and Roll Test Zone

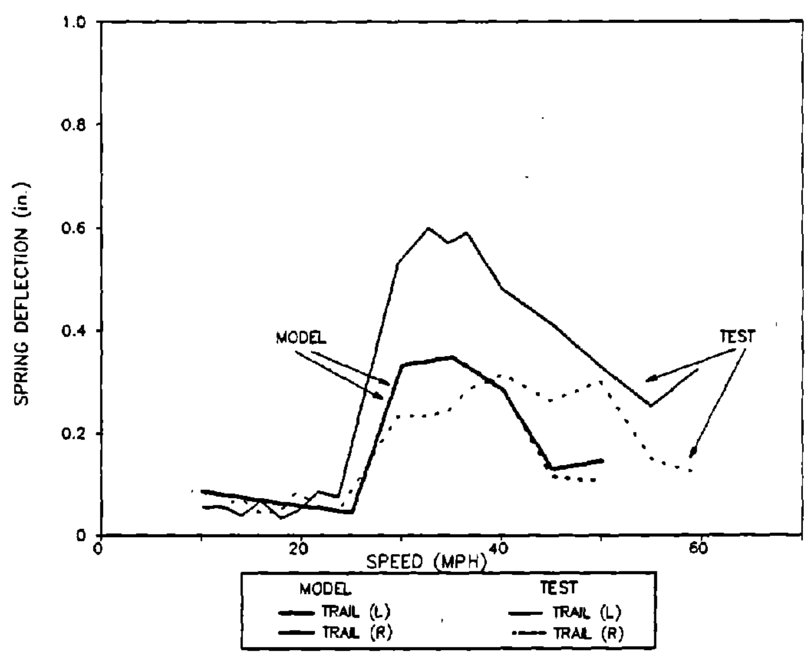

Figure 93. Maximum Peak to Peak Vertical Spring Deflections for the Trail Truck of the Empty Car in the Twist and Roll Test Zone

The friction wedge acts in both the vertical and lateral directions, but the total friction available remains a constant. Thus when there is some lateral motion coupled with the vertical motion as in these twist and roll tests, some of the friction force is "used up" in the lateral motion, leaving a lower breakout force for the vertical motion. The current NUCARS model does not vary the level of vertical friction to account for lateral motion. Thus the breakout friction for these predictions was probably too high, reducing total suspension travel. The AAR is developing a new method for modeling this cross coupling, and it will be included in the next version of NUCARS.

Comparisons of predicted and measured car body and car body relative to bolster roll angles are shown in Figure 94. The predicted resonant speed is 30 mph while the measured resonance is at 33 mph. Both model and test results show the car body roll angle exceeding the Chapter XI criteria of 6-degrees peak to peak. The body to bolster roll angles match the total body motion showing that the body is moving separately from the bolster with little roll action occurring at the bolster itself, as indicated by the relatively small vertical spring deflections.

The pre-test predictions (Section 6.4.1) showed a roll resonance at 35 mph, while the post-test predicted resonance is at 30 mph. These results bracket the measured resonance of 33 mph. Unlike the post-test predictions, the pre-test predictions show no Chapter XI roll criteria being exceeded until hunting response causes derailment at speeds above 60 mph. Thus the post-test modeling more accurately predicts vehicle behavior.

Based on both the vertical wheel load data and the car body roll angle predictions, the car exceeds the Chapter XI limiting criteria for empty car twist and roll.

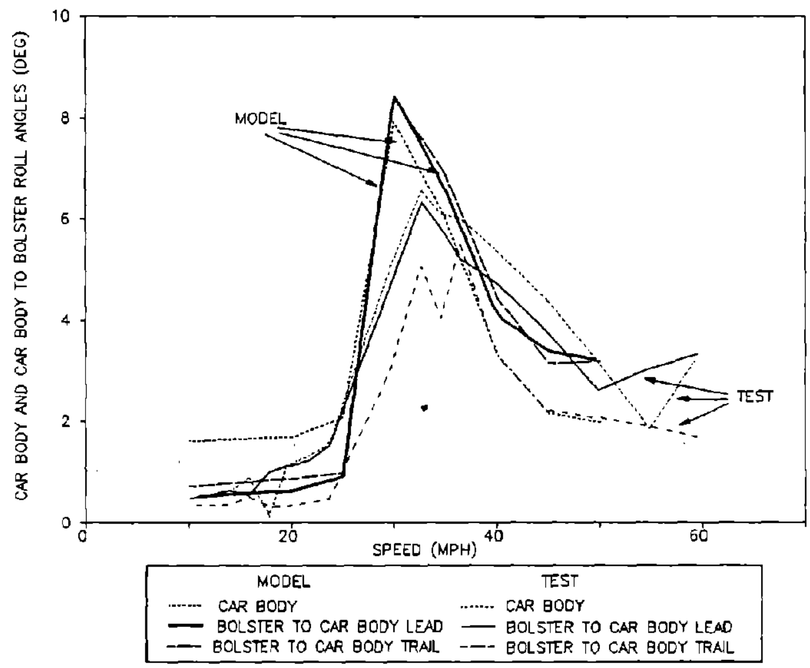

Figure 94. Maximum Peak to Peak Car Body and Car Body to Bolster Roll Angles of the Empty Car in the Twist and Roll Test Zone

8.4.2 Loaded Twist and Roll

Figures 95 and 96 compare the predicted lead truck vertical wheel loads with the loads measured during the loaded twist and roll track tests. Predicted resonance is between 20 and 25 mph, with a minimum wheel load of 40 percent of the static load. The test results show a much sharper resonance, showing wheel lift at 19 mph. The reason for NUCARS failing to predict wheel lift is probably due to not modeling the coupling between vertical and lateral motions of the friction wedge, as discussed for the empty car twist and roll results.

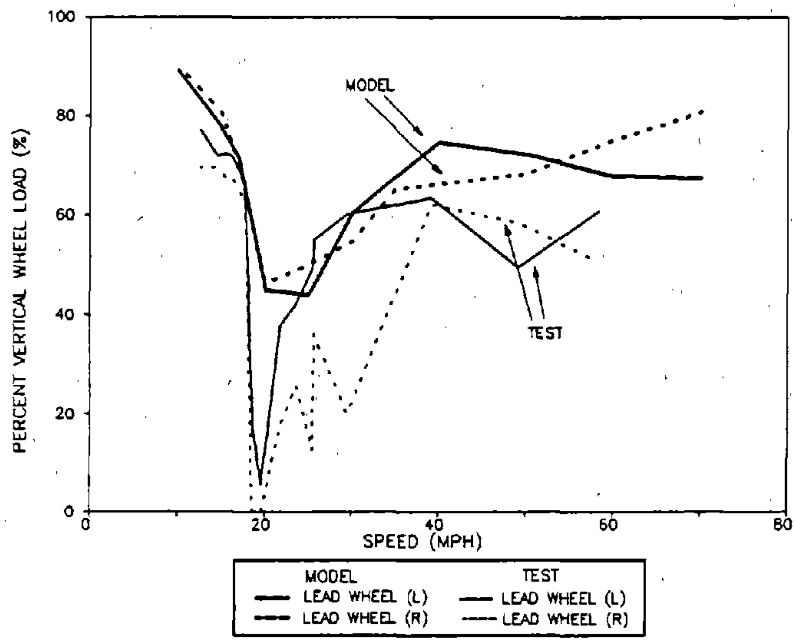

Figure 95. Minimum Percent Vertical Wheel Loads for Axle 1 of the Loaded Car in the Twist and Roll Test Zone

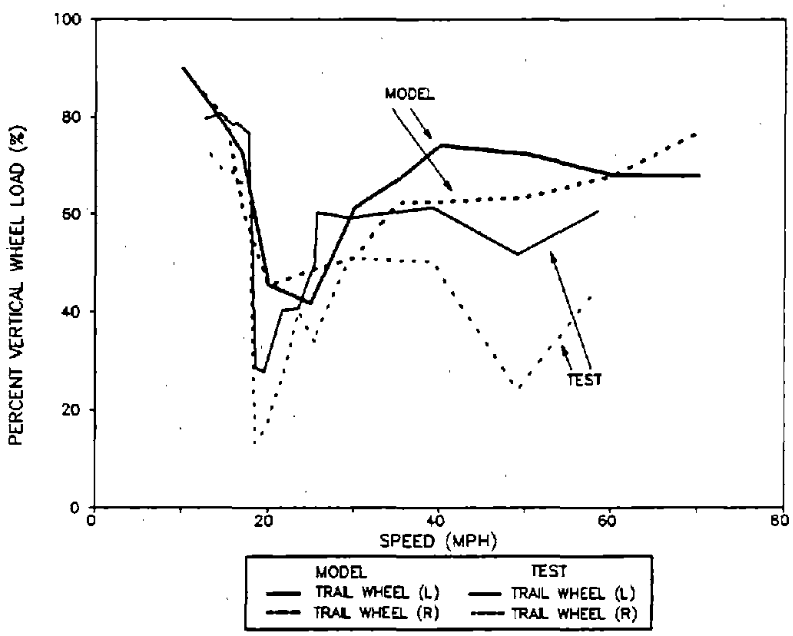

Figure 96. Minimum Percent Vertical Wheel Loads for Axle 2 of the Loaded Car in the Twist and Roll Test Zone

Comparisons of predicted vertical suspension deflections in Figures 97 and 98 confirm these conclusions about the model inaccuracies. The predicted suspension deflections show a much lower amplitude resonant peak than the test data. It is obvious that the model is over damped.

Comparisons of predicted and measured car body and car body relative to bolster roll angles are shown in Figure 99. The predicted resonances match well with the test data in the 19 mph to 23 mph speed range. Both model and test results show the car body roll angle well within the Chapter XI criteria of 6-degrees peak to peak.

The post and pre-test model results (Section 6.4.2) are very similar. This indicates that the errors in the simulations have little to do with how the track input is defined, further confirming that the friction wedge simulations are inadequate.

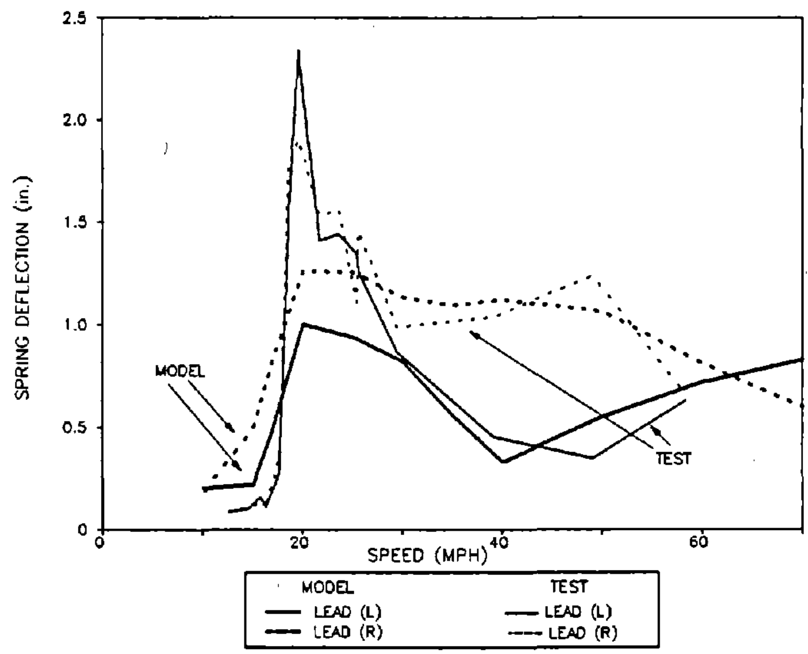

Figure 97. Maximum Peak to Peak Vertical Spring Deflections for the Lead Truck of the Loaded Car in the Twist and Roll Test Zone

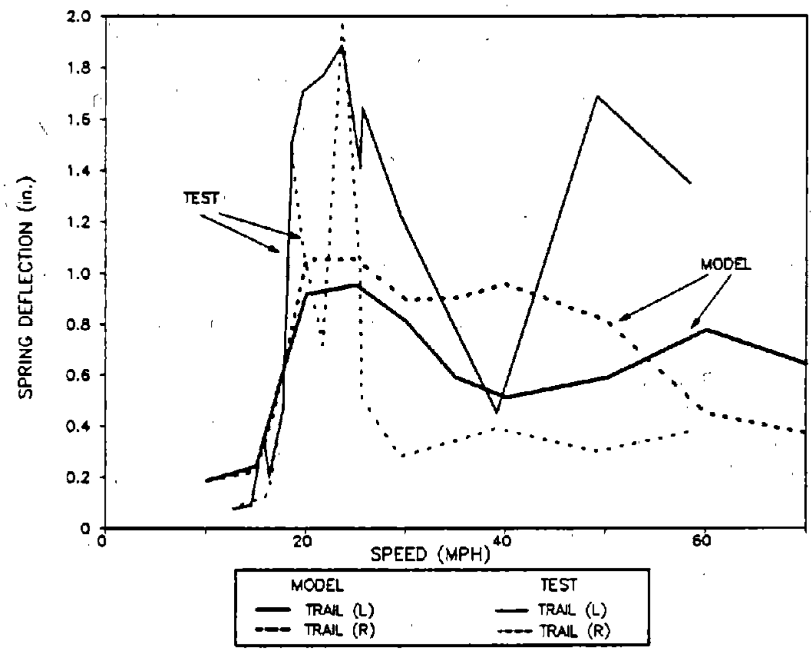

Figure 98. Maximum Peak to Peak Vertical Spring Deflections for the Trail Truck of the Loaded Car in the Twist and Roll Test Zone

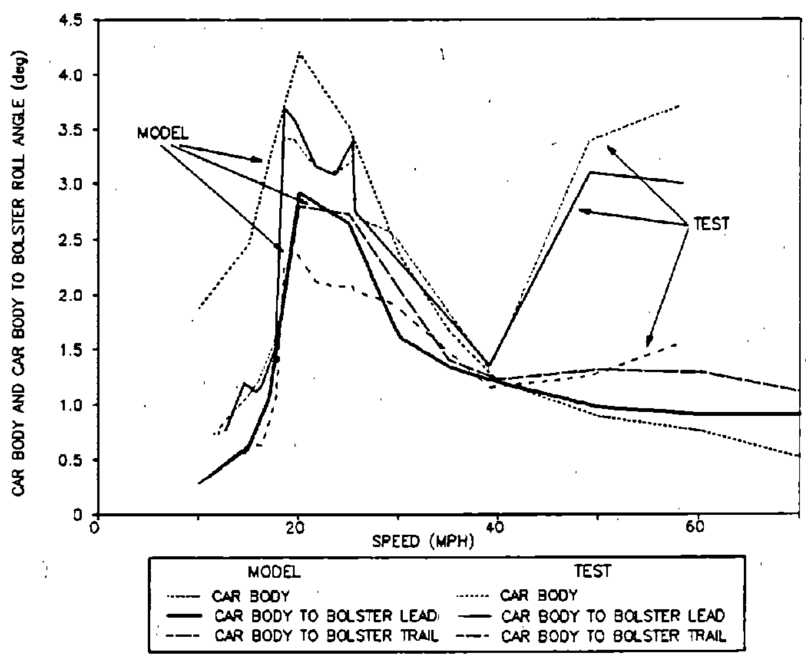

Figure 99. Maximum Peak to Peak Car Body and Car Body to Bolster Roll Angles of the Loaded Car in the Twist and Roll Test Zone

8.5 YAW AND SWAY

The ENSCO, TGMS measurements of the yaw and sway test zone were not recorded correctly. Therefore the post-test modeling results presented in this section were made with mathematically defined track geometry in the same manner as the pre-test NUCARS predictions. Wheel/rail cross sectional geometries measured at the test zone were used however. The NUCARS vehicle description system files for both loaded and empty car include the revised increased leading edge slope for the friction wedge characteristics described in Section 7.6.

8.5.1 Empty Yaw and Sway

Figures 100 and 101 show the maximum axle sum L/V ratios and the maximum truck side L/V ratios for the lead truck of the empty car. Measured L/V ratios are well below the Chapter XI limits of 1.4 for the axle sum and 0.6 for the truck side L/V. The NUCARS predictions show worse performance for all speeds up to 70 mph with the L/V ratios over the Chapter XI limits at all speeds. The very large predicted axle sum L/V ratios are the result of predicted wheel lifts on the non-flanging wheels. No derailments were predicted however.

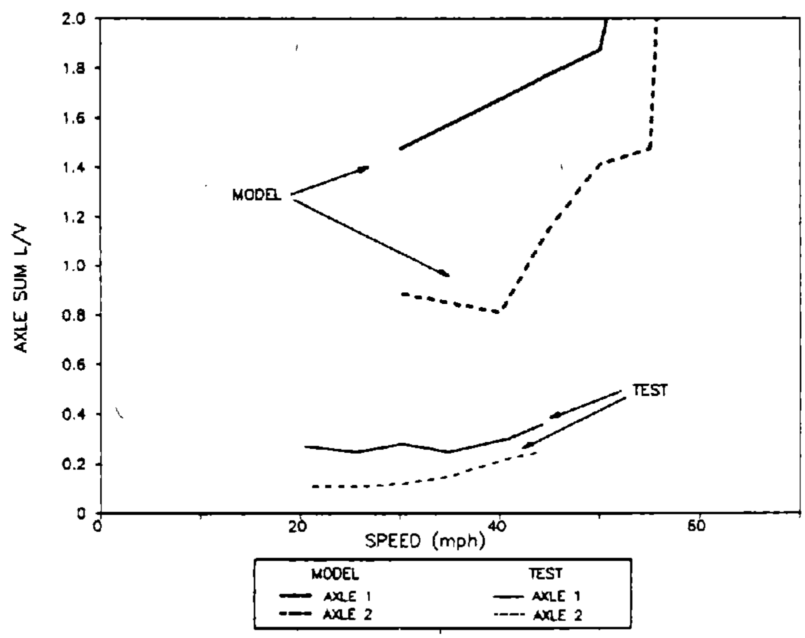

Figure 100. Maximum Axle Sum L/V Ratios for the Empty Car in the Yaw and Sway Test Zone

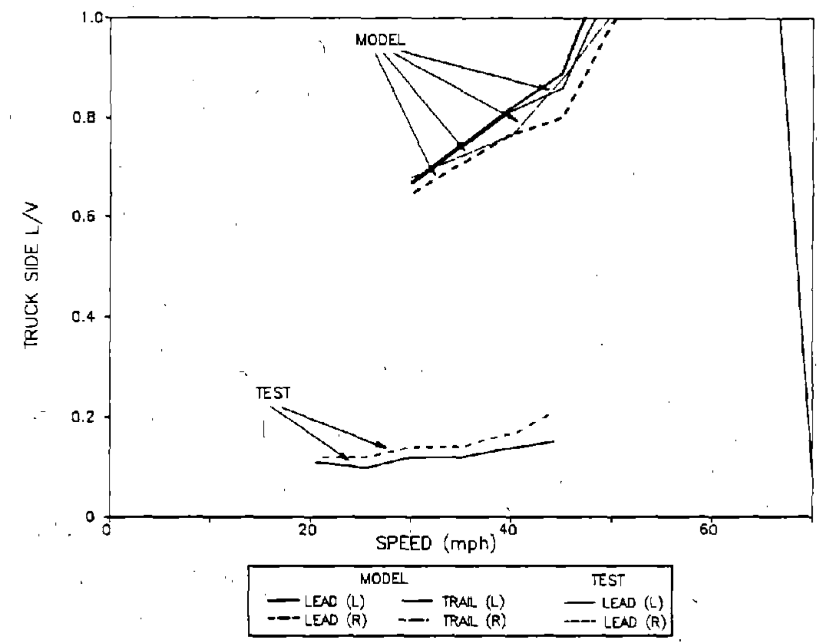

Figure 101. Maximum Truck Side L/V Ratios for the Empty Car in the Yaw and Sway Test Zone

Track tests were conducted up to only 45 mph, but the trends indicate that performance would remain within Chapter XI limits up to 70 mph. It is possible that these discrepancies between the test and model could be due to the measurement inaccuracies of the instrumented wheel set when lightly loaded.

The track tests were stopped at 45 mph because at the time the raw strain signal data appeared to indicate that very high L/V ratios were occurring. Analysis of the strain data with the software then available appeared to confirm this unsafe behavior. Subsequent analysis with the revised processing software produced the results shown here. It is apparent that the lack of reliable instrumented wheel set processing software could have caused the tests to be ended prematurely.

Maximum predicted lateral forces are greater than those measured during the tests, but the general trends are similar.

Pre-test NUCARS predictions had predicted derailment at nearly all speeds in the yaw and sway test zones. The post-test predictions shown here do not show derailment but exceed Chapter XI safety criteria while the track tests show even better performance. The improvements in predicted performance come from the revised yaw suspension characteristics and the inclusion of the measured axle misalignments as described in Sections 7.6 and 7.6.1.

The revised suspension characteristics allow the truck to warp more easily thus allowing the truck to warp as it steers through the perturbations. The axle misalignments cause the creep forces between the wheel and rail to be higher on tangent track than they would be for a perfectly aligned truck. Thus when the truck enters laterally perturbed track, it takes less creepage between the wheel and the rail to saturate the lateral forces. When not in flange contact, the creep force between wheel and rail is limited (saturated) by the coefficient of friction between wheel and rail. With the maximum lateral force reduced the maximum L/V ratios are also reduced.

It is suspected however that the primary shear pad stiffness used in the model was still too large. This would prevent the axles from steering into the perturbations as much as was expected. This combined with the previously discussed inaccuracies in the vertical suspension characteristics would result in the predicted wheel lifts evident in this data.

8.5.2 Loaded Yaw and Sway

Figures 102 and 103 show the maximum axle sum L/V ratios and the maximum truck side L/V ratios for the lead truck of the loaded car. The predicted axle sum L/V ratio exceeds the Chapter XI limit of 1.4 on axle 3 at 35 mph. Maximum axle sum L/V ratios on the other axles remain below 1.4 up to 45 mph. The truck side L/V ratio is also predicted to exceed the Chapter XI limit of 0.6 over the same speed range. Above 50 mph sharp increases in L/V are predicted indicating imminent derailment. These results are matched well by the test data. Tests were terminated at 45 mph when the truck side L/V ratio exceeded 0.8, but the trend indicates that at higher speeds results would match the predictions.

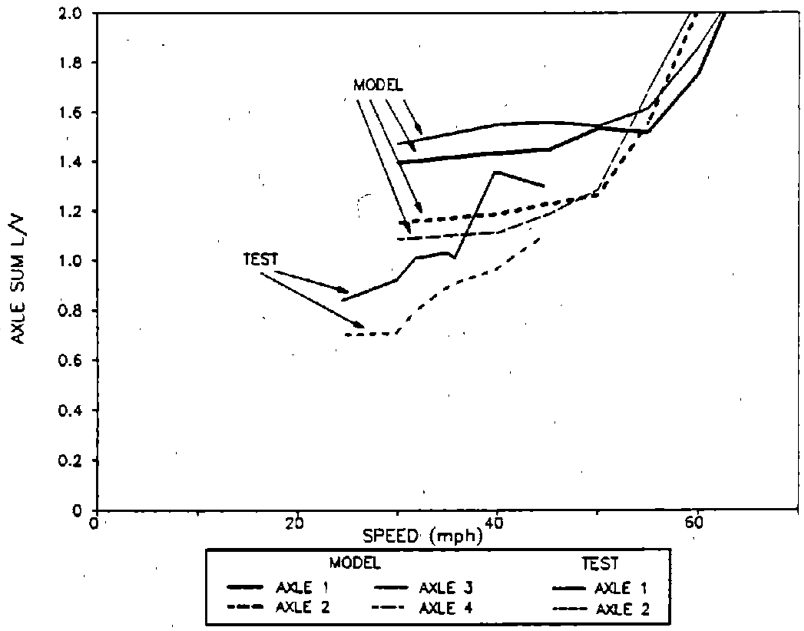

Figure 102. Maximum Axle Sum L/V Ratios for the Loaded Car in the Yaw and Sway Test Zone

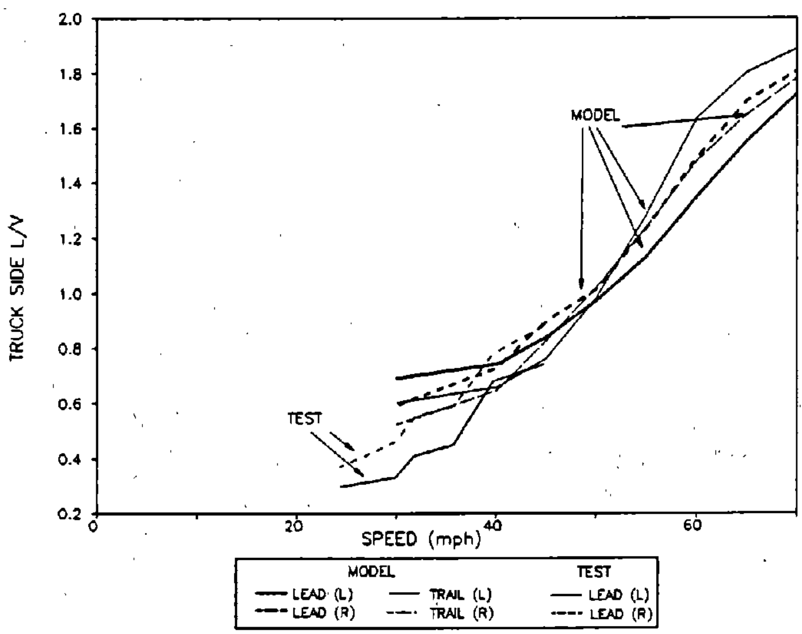

Figure 103. Maximum Truck Side L/V Ratios for the Loaded Car in the Yaw and Sway Test Zone

These post-test predictions contrast sharply with the pre-test predictions, in which derailment was predicted for speeds 30 mph or more. As with the empty car predictions (Section 8.5.1), the only differences are the revised system description files and the use of measured wheel/rail profile geometries. The differences between test results and NUCARS predictions could be due to using mathematically defined track instead of actual measured track geometry, as well as the inaccurate modeling of the friction wedge when it is moving in two directions. It is obvious however that the lateral behavior is very sensitive to changes to lateral and yaw suspension characteristics.

8.6 STEADY STATE CURVING

Chapter XI only requires testing on a single curve with greater than 7-degrees curvature. For this project, the curving tests were conducted on a loop with 4-, 7.5-, 10- and 12-degree curves. This allowed fulfilling the Chapter XI requirements as well as exploring the details of the vehicle's curving performance over a wide range of curves.

The spiral leading into the 12-degree curve is also designed to meet the requirements for the Chapter XI bunched spiral tests, while perturbations have been installed in a portion of the 10-degree curve to fulfill the requirements of the dynamic curving tests. These perturbations limited the maximum test speed for the loaded car to balance speed due to excessive dynamic action.

8.6.1 Empty Steady State Curving

The test data for the empty car were again plagued by dubious wheel force measurements. Therefore comparisons between test and model are dependent on the wheel set AOA measurements made in the 4- and 10-degree curves.

A comparison of predicted and measured AOA in the 4-degree curve for the lead truck is shown in Figure 104. The trailing truck showed similar results. The model predictions show the lead axles (axles 1 and 3) of each truck running at an AOA of about 4 milliradians, while the trail axles (axles 2 and 4) are running nearly radial at close to 0 milliradians. The test data shows that in most instances the lead axles are closer to being radial indicating better actual curving performance than was predicted. The scatter in the test data is to be expected because the measurement is made at a single location in the curve while the model prediction is an average value over 200 to 300 feet of curve.

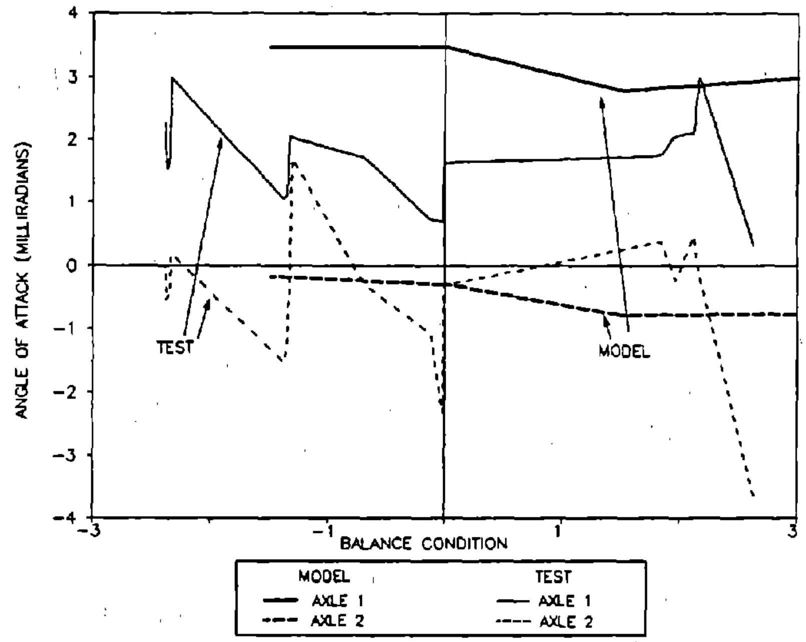

Figure 104. Axle Angles of Attack for the Empty Car in the 4-Degree Curve

Small AOA's normally indicate small L/V ratios and hence good curving behavior. The predicted axle sum L/V ratios shown in Figure 105 were all well within Chapter XI limits. Therefore it is expected that the test which had lower AOA's, than were predicted, would show even lower L/V ratios.

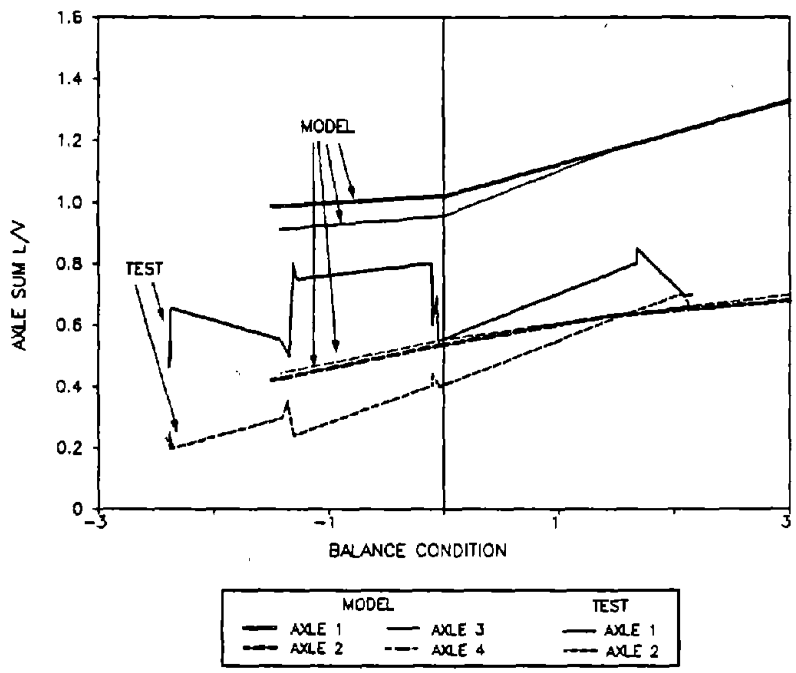

Figure 105. Maximum Axle Sum L/V Ratios for the Empty Car in the 4-Degree Curve

A comparison of the predicted and measured AOA for the lead truck in the 10-degree curve is given in Figure 106. In this instance the data match is within the accuracy and repeatability of the AOA measurement. This indicates that in the sharper curves the ability of the primary rubber shear pads to allow the axles to steer is exceeded.

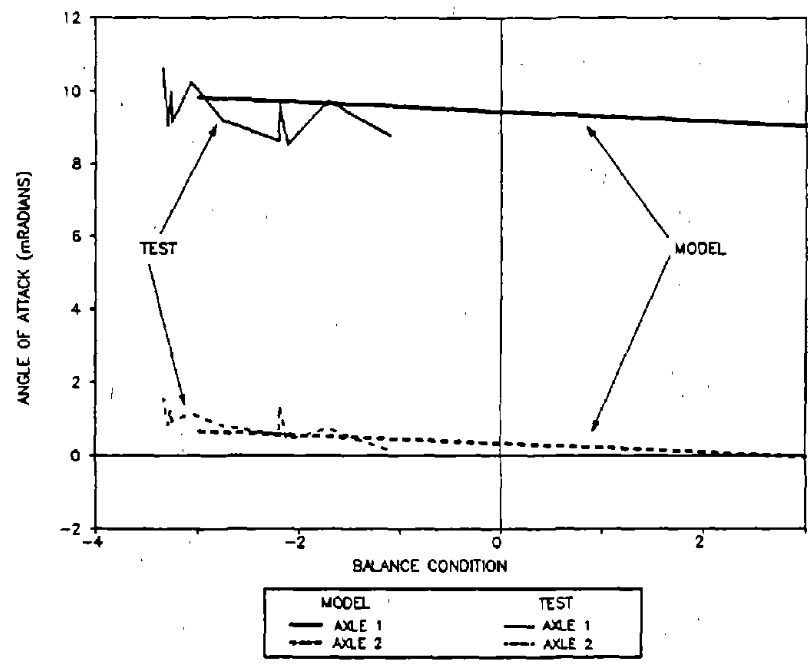

Figure 106. Axle Angles of Attack for the Empty Car in the 10-Degree Curve

The predicted L/V ratios in the 10-degree curve were again all within Chapter XI limits, and it is assumed that the test results would have been similar.

As expected, the maximum predicted L/V ratios occurred in the 12-degree curve. These were all within Chapter XI limits. An example of these results is shown in Figure 107, which compares predicted and measured maximum axle sum L/V ratios for the different balance conditions in the 12-degree curve. In some instances, the test data appears to exceed the Chapter XI limit of 1.4. This is due to the inaccuracies of the instrumented wheel sets when very lightly loaded. Otherwise the trends shown in the test data match the NUCARS predictions.

The pre-test predictions were similar to the post-test predictions, and do not show the better curving behavior exhibited by the test vehicle.

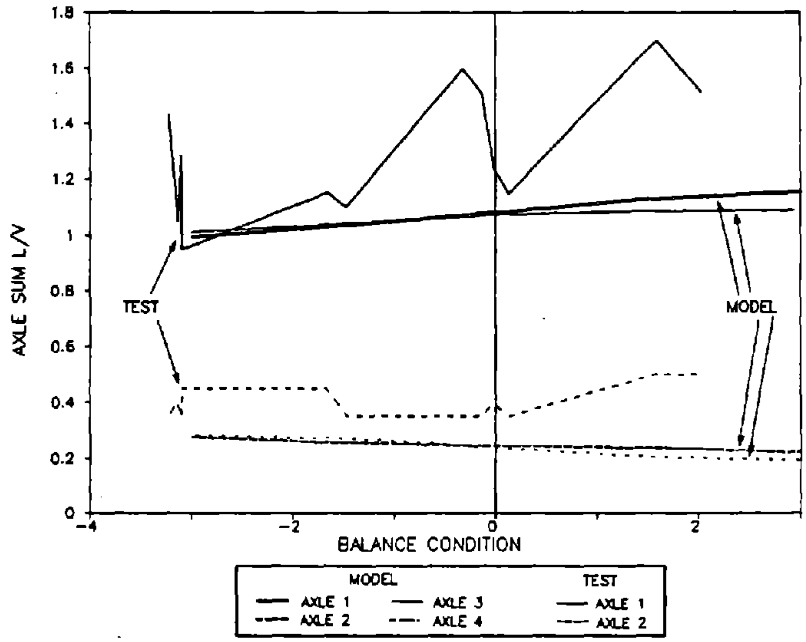

Figure 107. Maximum Axle Sum L/V Ratios for the Empty Car in the 12-Degree Curve

8.6.2 Loaded Steady State Curving

The comparison between test and model predictions for the loaded car is similar to the empty car results. For the 4- and 7.5-degree curves the test vehicle negotiates the curve better than predicted by NUCARS. This is illustrated by comparing the predicted and measured axle sum L/V ratios shown in Figures 108 and 109. The model consistently predicts larger L/V ratios for the lead axle than the trail axle of each truck. The test data shows the lead and trail axle L/V ratios to be similar and very low, indicating the axles are aligning themselves well to the track.

Both test and model results show performance well within Chapter XI limits in both 4- and 7.5-degree curves.

For the 10- and 12-degree curves the match is much better. The maximum axle sum L/V ratios shown in Figures 110 and 111 match very well. Unfortunately only below balance speed data is available for the 10-degree curve. This is because the excessive dynamic action in the dynamic curve portion of the 10-degree curve prevented collecting steady state data at the higher speeds. Neither model nor test results exceeded any Chapter XI limits for the 10- and 12-degree curves.

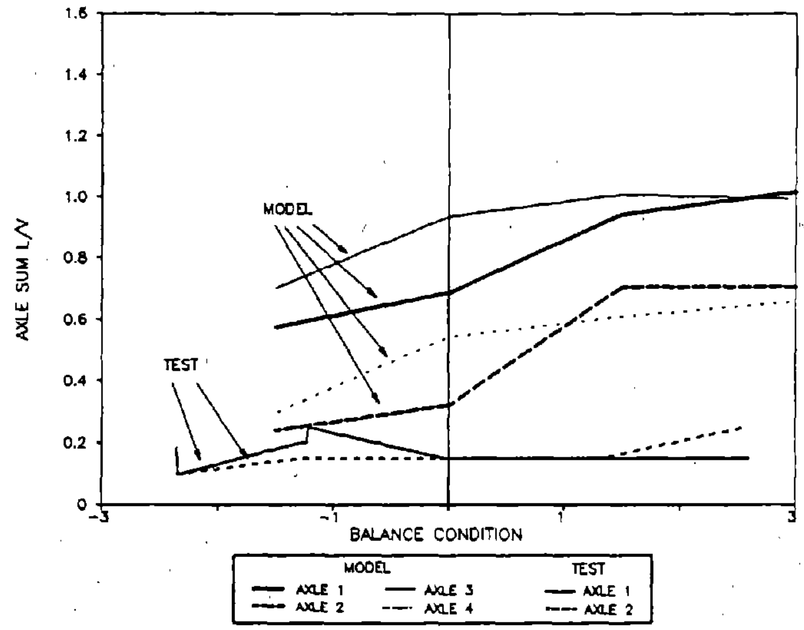

Figure 108. Maximum Axle Sum L/V Ratios for the Loaded Car in the 4-Degree Curve

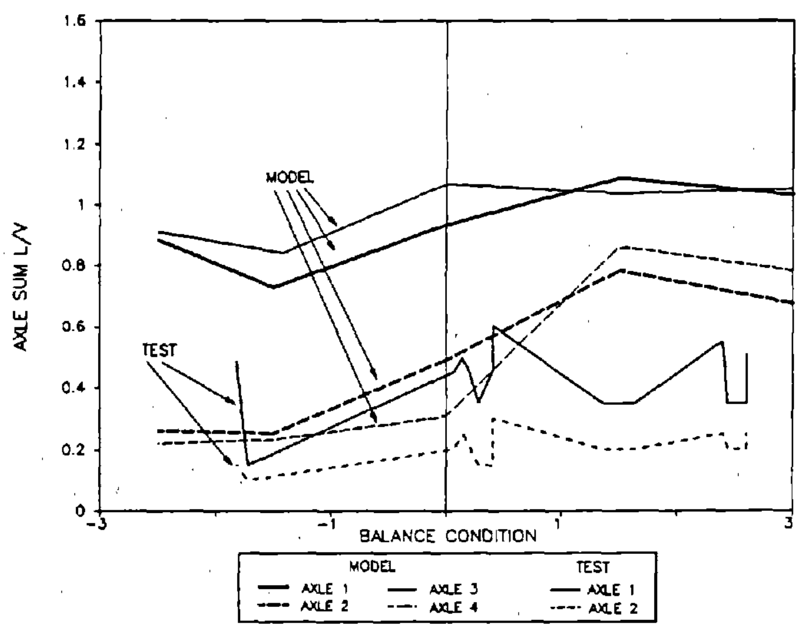

Figure 109. Maximum Axle Sum L/V Ratios for the Loaded Car in the 7.5-Degree Curve

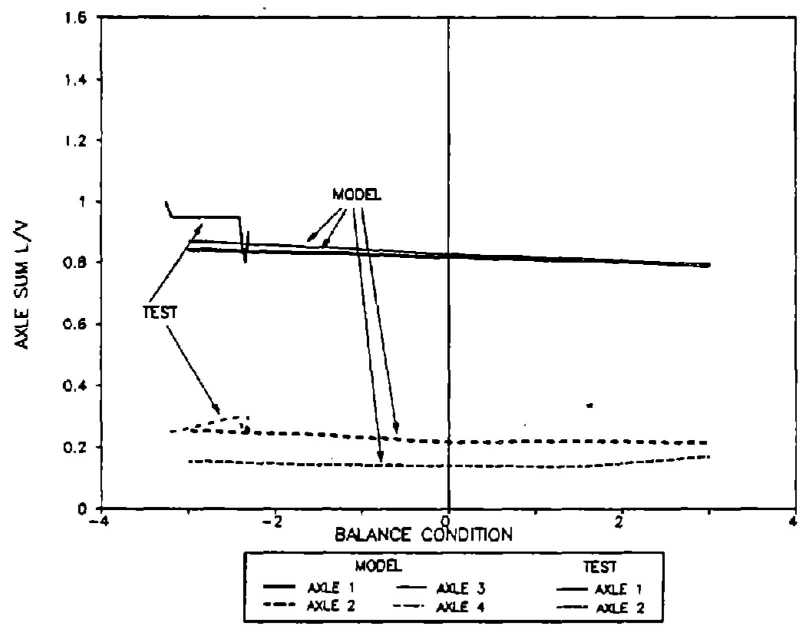

Figure 110. Maximum Axle Sum L/V Ratios for the Loaded Car in the 10-Degree Curve

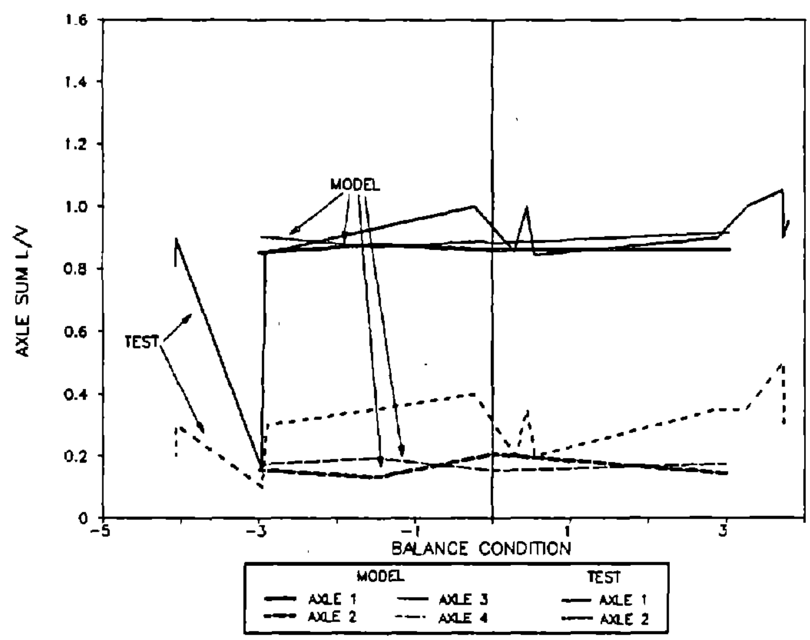

Figure 111. Maximum Axle Sum L/V Ratios for the Loaded Car in the 12-Degree Curve

The reason for the mismatch between test and model predictions for the lower degrees of curvature is unclear. It is obvious that the axles are steering into the curve better than predicted. The ability to steer is affected by several parameters including:

1. Truck warp stiffness
2. Bolster to body yaw friction
3. Primary shear pad stiffness
4. Wheel set conicity
5. Presence or absence of 2 point contact between wheels and rails

The truck warp stiffness and damping values used in this modeling were derived from the test data. These are probably correct for reasons explained in Section 7.6.1 above.

The bolster to body center bowl yaw friction could be a little too high. Figure 112 compares predicted and measured average center bowl yaw displacement for the loaded car in the 10-degree curve. The test data shows somewhat greater yaw angles indicating that there was less friction to prevent the bolster from turning relative to the body.

The primary shear pads may also have been modeled as being too stiff. Unfortunately no estimates of its stiffness were made from the test data. This stiffness was measured by two methods during the characterization tests giving two values; one 25 percent less than the other. The design value is approximately midway between the two measured values. As discussed in Section 5.4.3 the higher test value is a result from the inter-axle bending test while the lower value is a result from the lateral dynamic test on the MSU. The modeling used the higher value for both longitudinal and lateral characteristics because it was assumed that the characteristics should be symmetric for all horizontal deflections. It is suspected that this is the most likely inaccuracy.

The pre-test predictions match the test data better than the post-test predictions. The pre-test predictions do show some tendency for the axles to steer into the curves, although not to the degree seen during the tests.

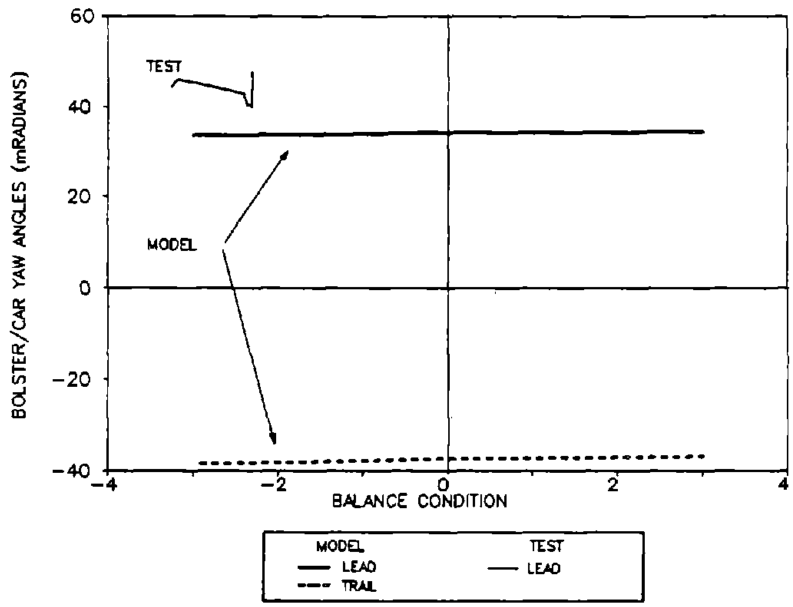

Figure 112. Average Bolster to Car Body Yaw Angles for the Loaded Car in the 10-Degree Curve

8.7 CURVE ENTRY/EXIT

8.7.1 Empty Curve Entry

Maximum wheel L/V ratios and minimum percent wheel loads for the empty car negotiating the bunched entry spiral are shown in Figures 113 through 116. The NUCARS predictions show performance well within Chapter XI limits. The test data however shows much worse performance. Wheel lift is indicated at one speed with L/V ratios much greater than the Chapter XI limit of 1.0. Due to the problems encountered with the empty car wheel force measurements, these results are dubious.

Pre-test predictions showed similar results to the post-test predictions.

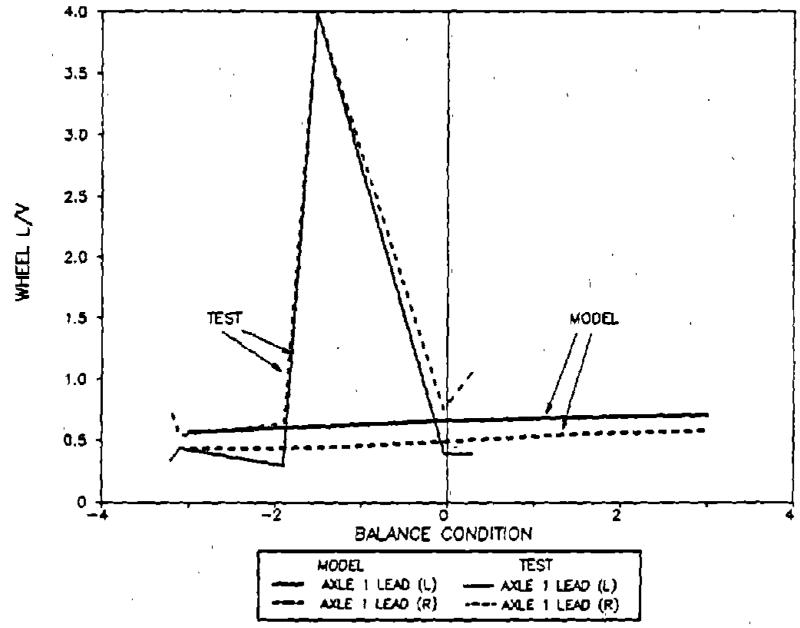

Figure 113. Maximum Wheel L/V Ratios for Axle 1 of the Empty Car in the Bunched Entry Spiral to the 12-Degree Curve

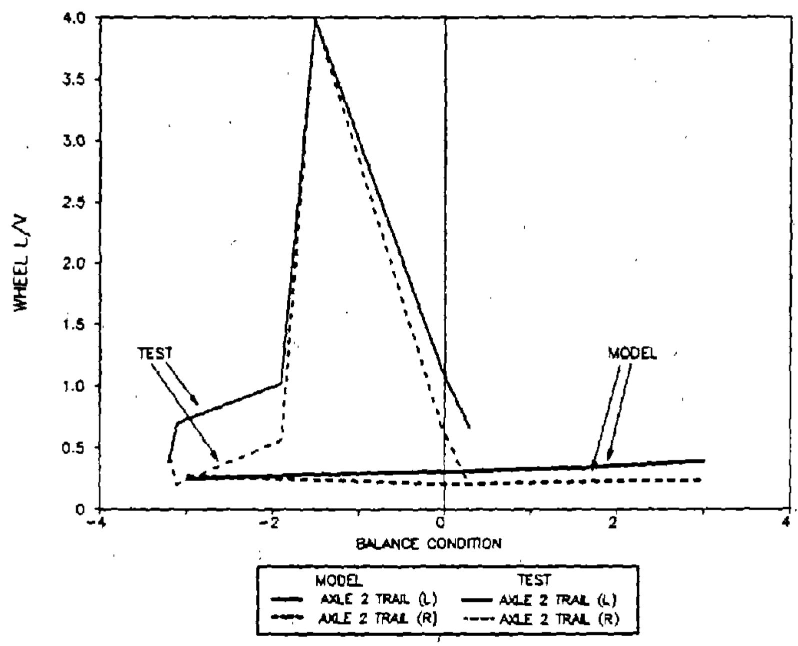

Figure 114. Maximum Wheel L/V Ratios for Axle 2 of the Empty Car in the Bunched Entry Spiral to the 12-Degree Curve

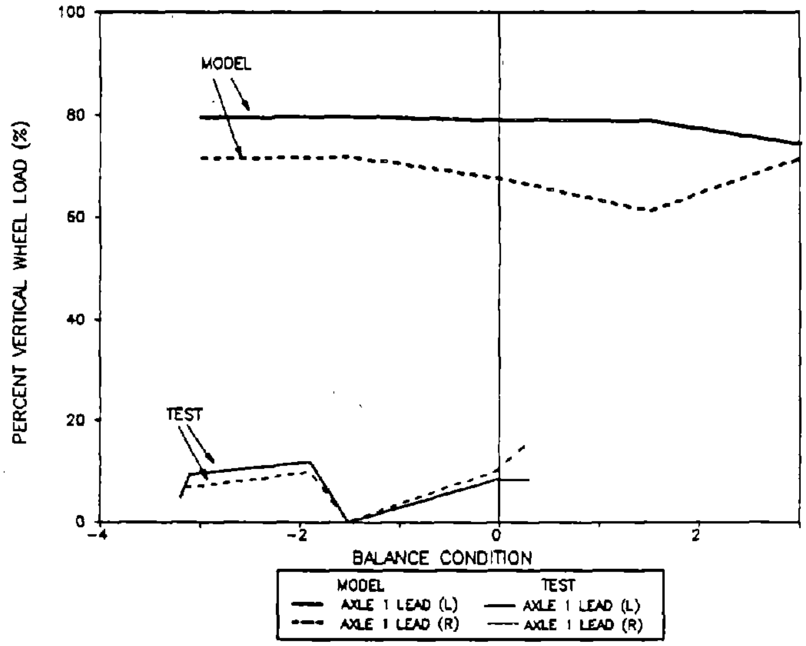

Figure 115. Minimum Percent Vertical Wheel Load for Axle 1 of the Empty Car in the Bunched Entry Spiral to the 12-Degree Curve

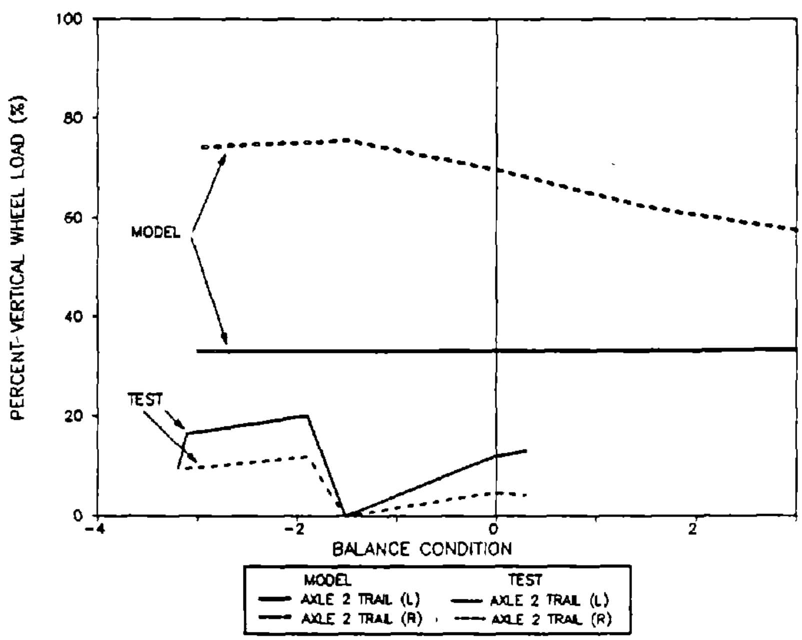

Figure 116. Minimum Percent Vertical Wheel Load for Axle 2 of the Empty Car in the Bunched Entry Spiral to the 12-Degree Curve

8.7.2 <u>Empty Curve Exit</u>

The curve exit predictions and test results shown in Figures 117 through 120 for the empty car are similar to the curve entry. NUCARS predictions are all well within Chapter XI limits. The test results appear to be plagued by the poor quality wheel force data found in most of the empty car tests. Again pre-test predictions were matched by the post-test predictions.

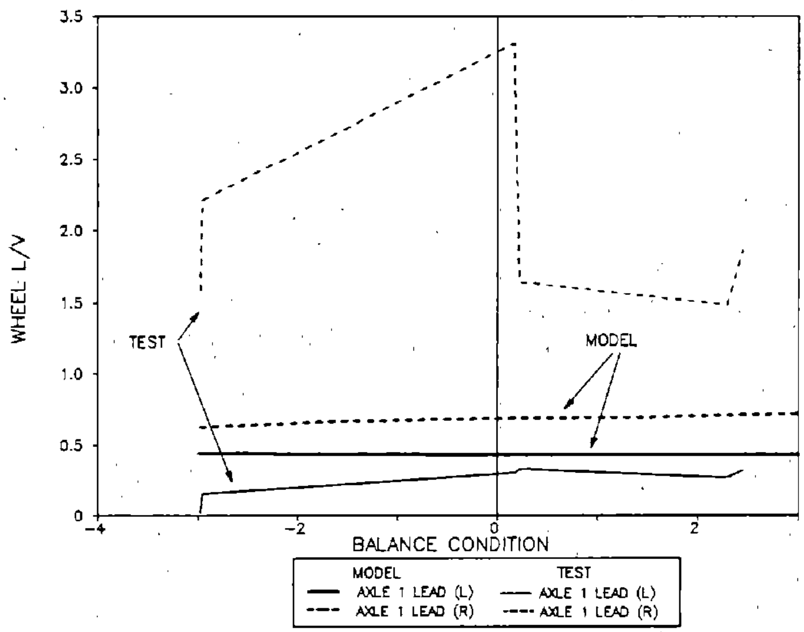

Figure 117. Maximum Wheel L/V Ratios for Axle 1 of the Empty Car in the Bunched Exit Spiral of the 12-Degree Curve

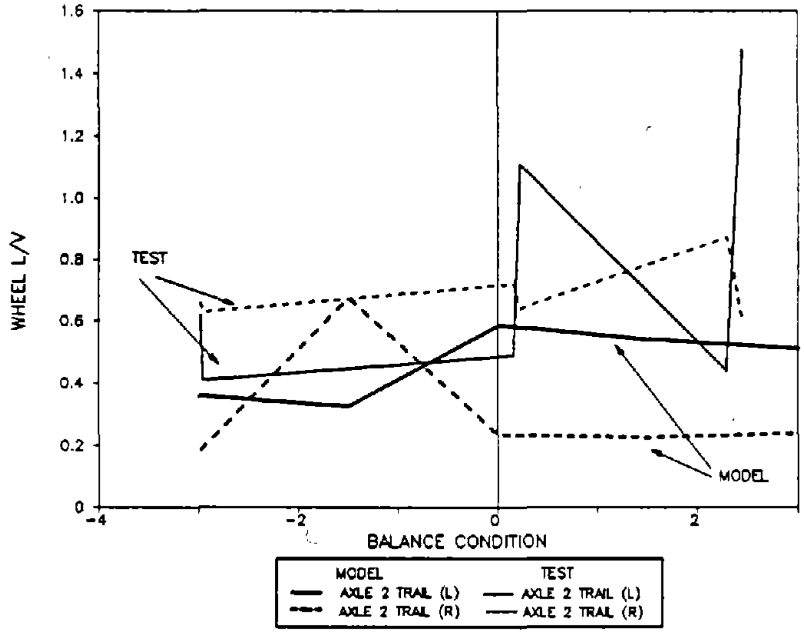

Figure 118. Maximum Wheel L/V Ratios for Axle 2 of the Empty Car in the Bunched Exit Spiral of the 12-Degree Curve

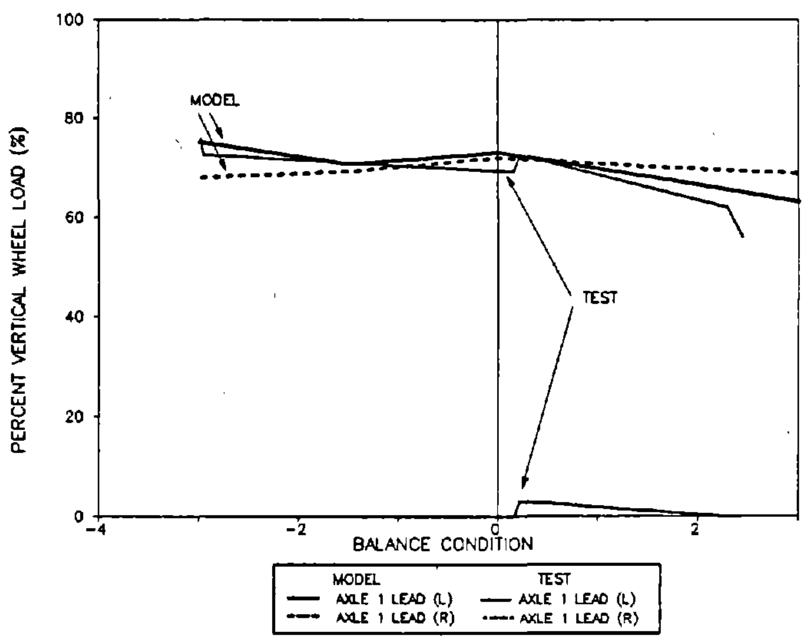

Figure 119. Minimum Percent Vertical Wheel Load for Axle 1 of the Empty Car in the Bunched Exit Spiral of the 12-Degree Curve

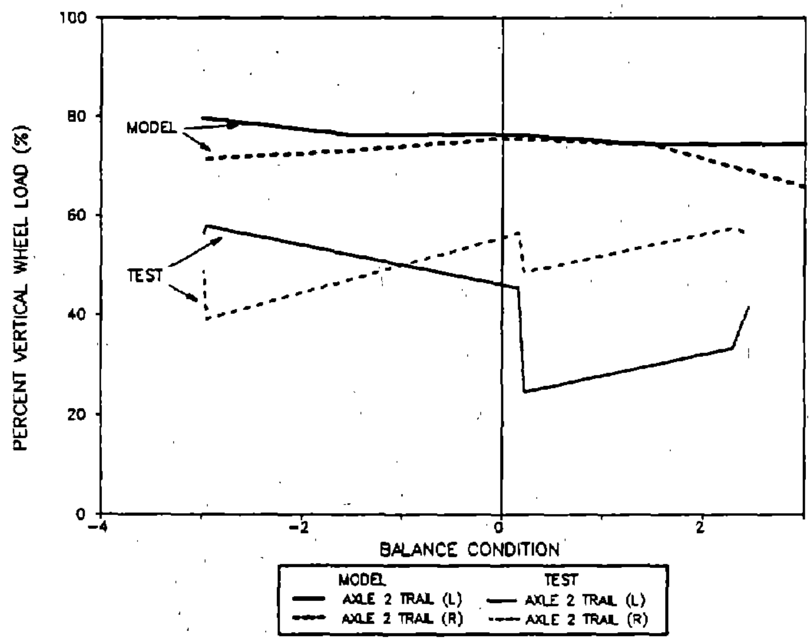

Figure 120. Minimum Percent Vertical Wheel Load for Axle 2 of the Empty Car in the Bunched Exit Spiral of the 12-Degree Curve

8.7.3 Loaded Curve Entry

Maximum wheel L/V ratios and minimum percent vertical wheel loads for the loaded car are shown in Figures 121 through 124. NUCARS predictions are similar to the test results, up to the balance speed for the 12-degree test curve. Speeds higher than the balance speed were not tested because the test train could not attain that speed without traversing the dynamic curving test zone at an unsafe speed.

The very low L/V ratios measured during the test on the lead axle (axle 1) at -3 inches of imbalance appears to be anomalous. It is possible that for that one run the axles were steering better into the curve than would be expected. Both test and model results are well within the Chapter XI limits, with results being similar to the pre-test predictions.

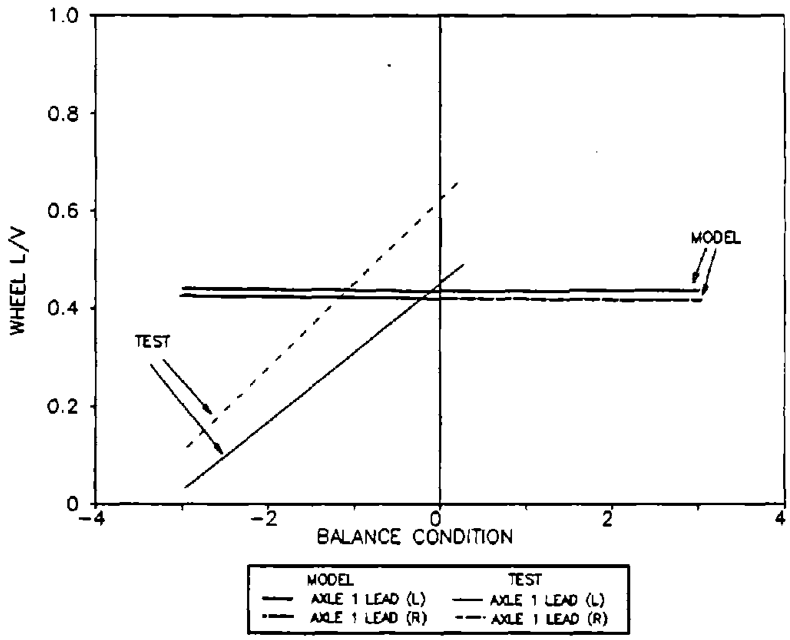

Figure 121. Maximum Wheel L/V Ratios for Axle 1 of the Loaded Car in the Bunched Entry Spiral to the 12-Degree Curve

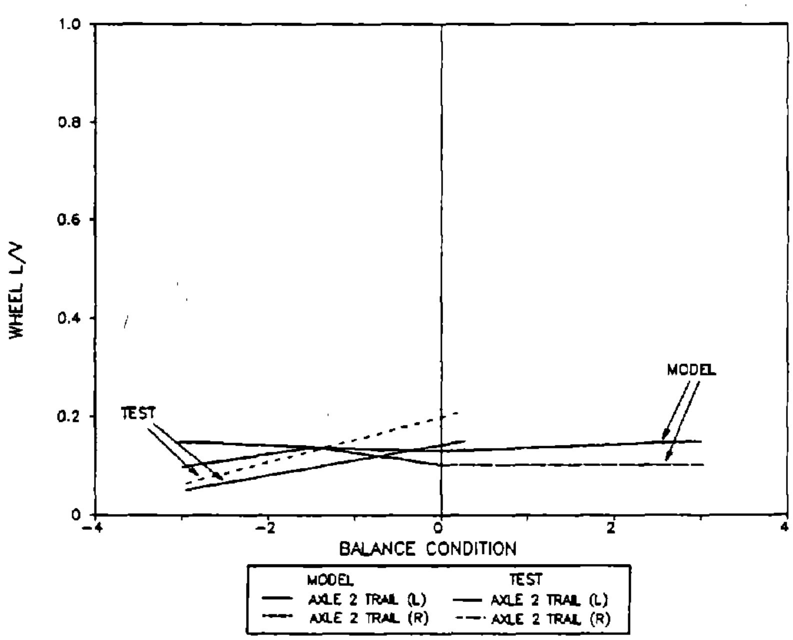

Figure 122. Maximum Wheel L/V Ratios for Axle 2 of the Loaded Car in the Bunched Entry Spiral to the 12-Degree Curve

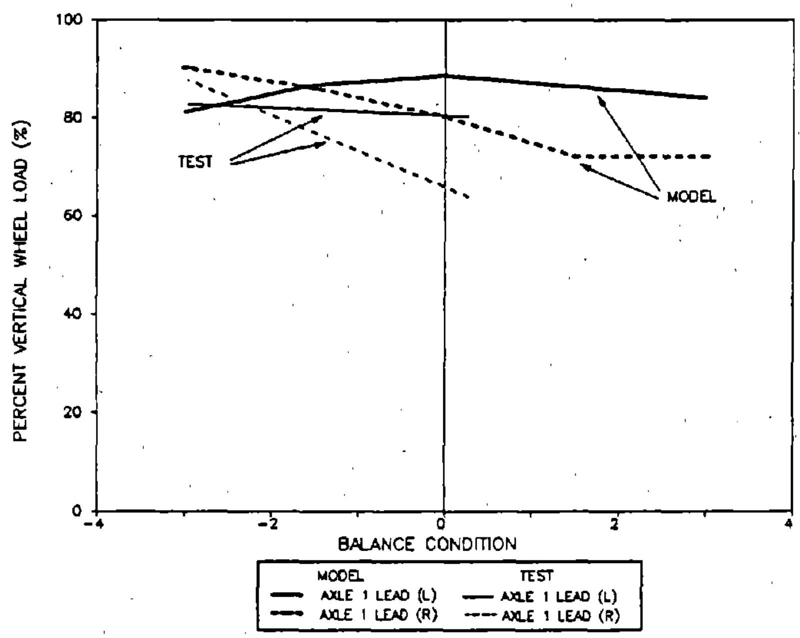

Figure 123. Minimum Percent Vertical Wheel Load for Axle 1 of the Loaded Car in the Bunched Entry Spiral to the 12-Degree Curve

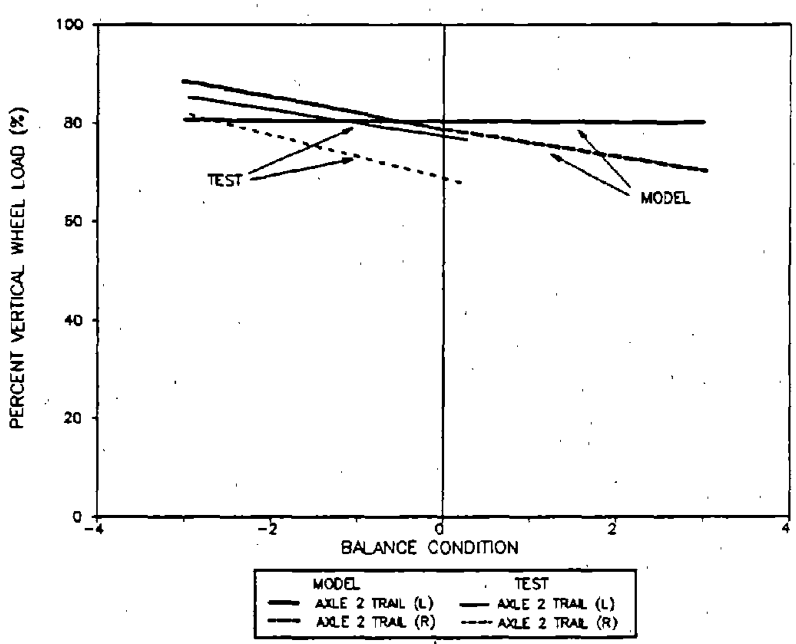

Figure 124. Minimum Percent Vertical Wheel Load for Axle 2 of the Loaded Car in the Bunched Entry Spiral to the 12-Degree Curve

8.7.4 Loaded Curve Exit

The curve exit predictions and test results are compared in Figures 125 through 128. The test results show lower wheel L/V ratios than the NUCARS predictions. This confirms the results of the steady curving analysis which showed the test vehicle has better curving performance than the NUCARS model. Both NUCARS and test results are well within the Chapter XI limits, with the post-test predictions being similar to the pre-test predictions.

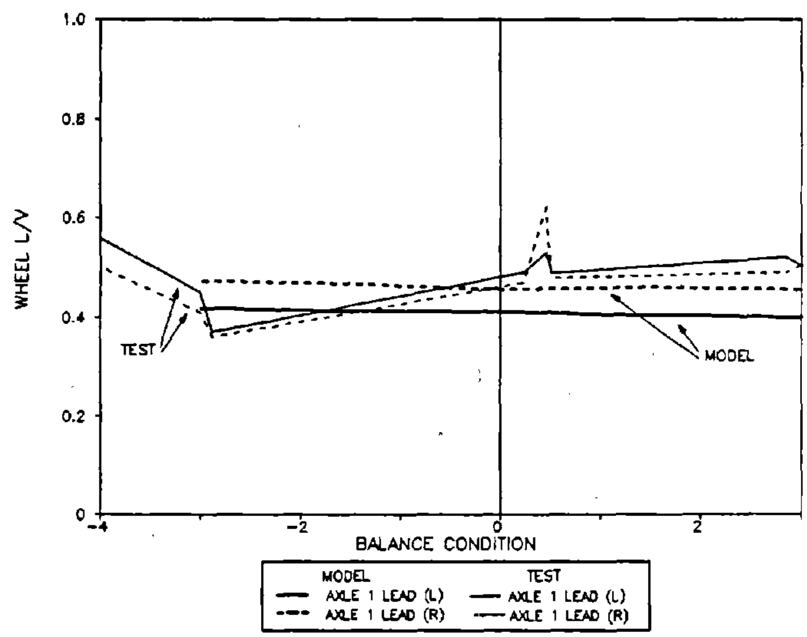

Figure 125. Maximum Wheel L/V Ratios for Axle 1 of the Loaded Car in the Bunched Exit Spiral of the 12-Degree Curve

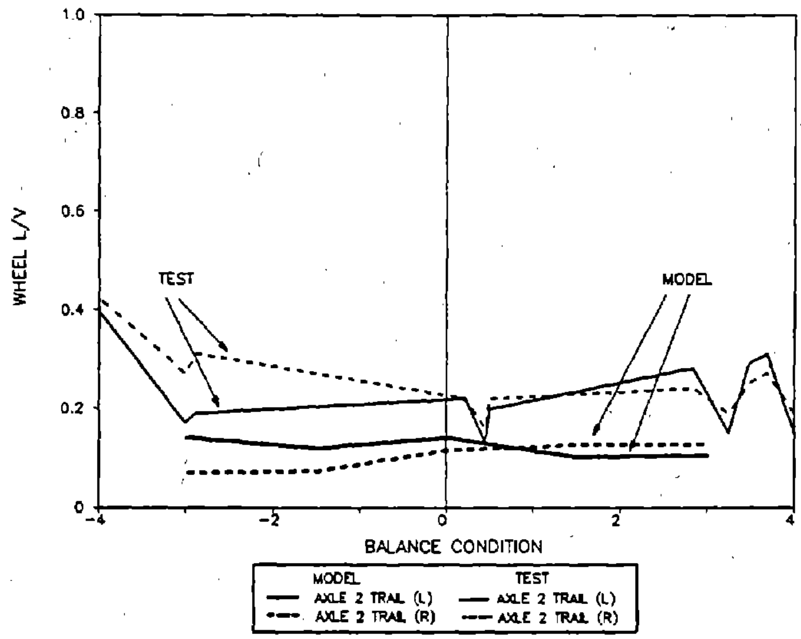

Figure 126. Maximum Wheel L/V Ratios for Axle 2 of the Loaded Car in the Bunched Exit Spiral of the 12-Degree Curve

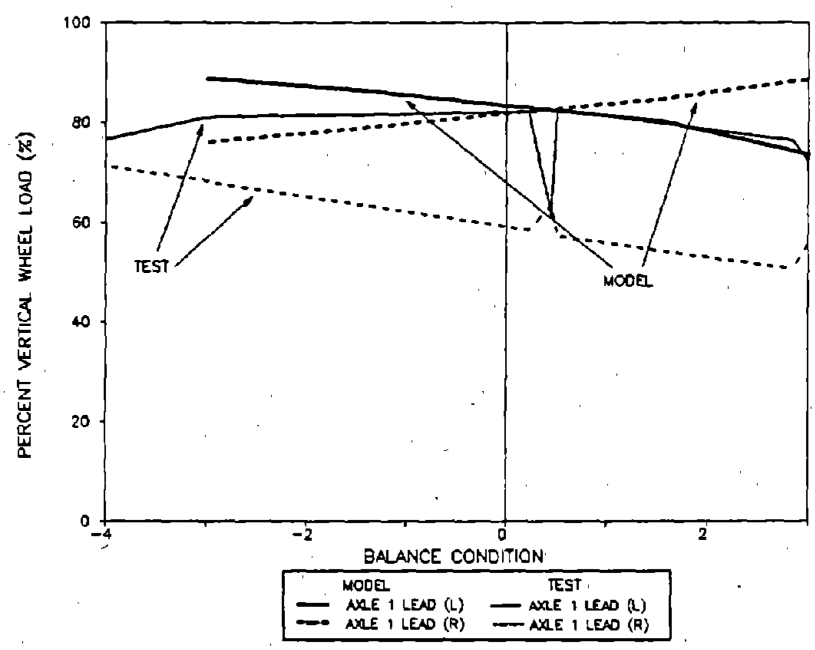

Figure 127. Minimum Percent Vertical Wheel Load for Axle 1 of the Loaded Car in the Bunched Exit Spiral of the 12-Degree Curve

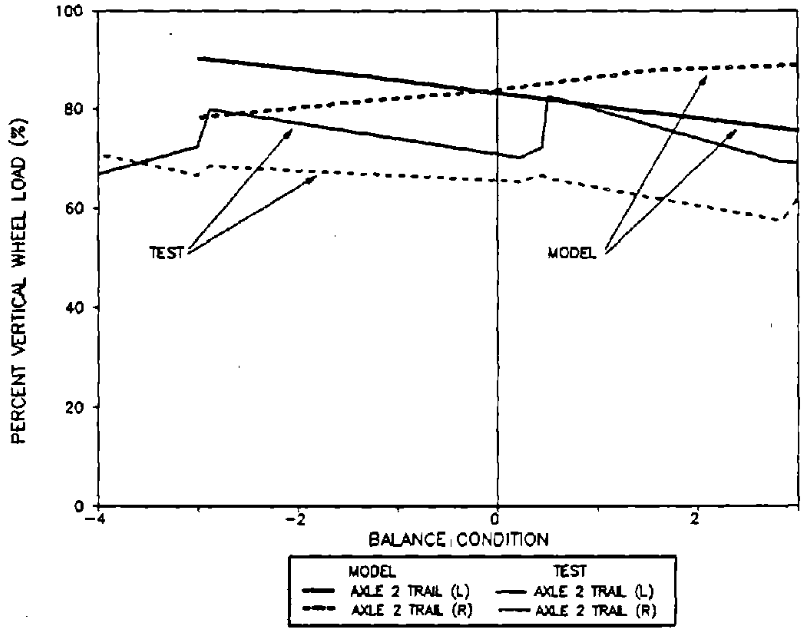

Figure 128. Minimum Percent Vertical Wheel Load for Axle 2 of the Loaded Car in the Bunched Exit Spiral of the 12-Degree Curve

8.8 DYNAMIC CURVING

The dynamic curving test zone is located in a 10-degree curve with 4 inches of superelevation. This gives a balance speed of 24 mph, with -3 and +3 inches of imbalance being 12 mph and 32 mph respectively.

8.8.1 Empty Dynamic Curving

Dynamic curving test results for the empty car are all affected by poor quality wheel force data. Negative vertical wheel forces were recorded on all four wheels, showing that the wheel force results are dubious. NUCARS predictions for the lead axle maximum wheel L/V ratios, shown in Figure 129 indicate that Chapter XI limits are exceeded at the 3-inch overbalance condition.

A comparison of the predicted and measured car body roll and car body to bolster roll angles is shown in Figure 130. Although the Chapter XI limit of 6-degrees peak to peak is not exceeded, large body roll angles are predicted at the 3-inch overbalance condition. This is close to the resonant speed found by both test and model in the twist and roll test section.

The pre-test predictions showed derailment at balance speed and above. Although the tests were not conducted above 1.5 inches of imbalance the good match with post-test predictions of roll angle indicate that the vehicle would safely negotiate the dynamic curve up to the 3-inch overbalance speed, although the maximum allowable roll angle is being approached at that speed.

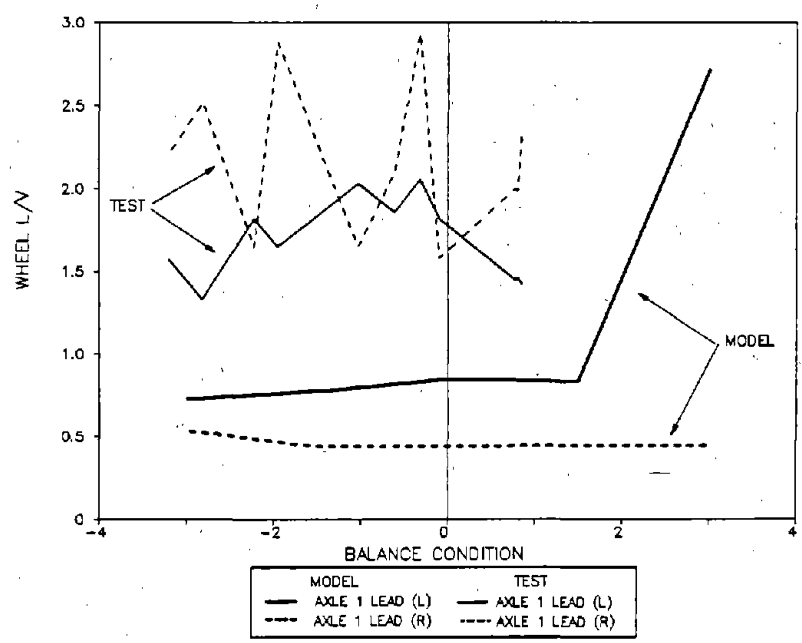

Figure 129. Maximum Wheel L/V Ratios for Axle 1 of the Empty Car in the 10-Degree Dynamic Curve

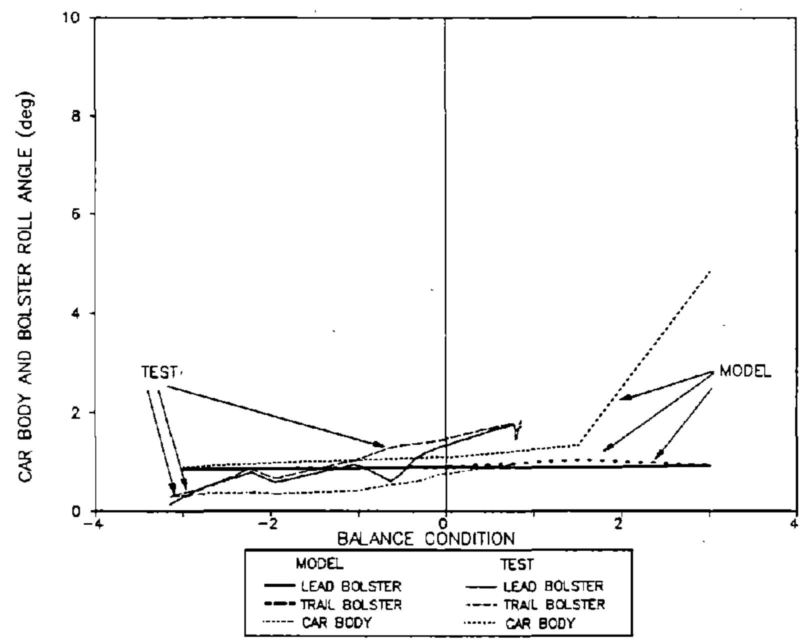

Figure 130. Maximum Peak to Peak Car Body and Bolster Roll Angles of the Empty Car in the 10-Degree Dynamic Curve

8.8.2 Loaded Dynamic Curving

A comparison of the predicted and measured minimum percent wheel loads in the dynamic curve for the loaded car is shown in Figures 131 and 132. The NUCARS predictions indicate good performance at all speeds while the test results show wheel lift at balance speed. No tests were performed above balance speed due to the possibility of derailment. This indicates that considerably more dynamic motion was occurring during the test than was predicted by NUCARS. This could be due to the inaccurate modeling of the friction wedge described in Section 2.4. The combined lateral and vertical action in the dynamic curve could very easily reduce the available friction in the vertical direction. The reduced vertical friction would allow the vehicle greater dynamic action and hence wheel lift.

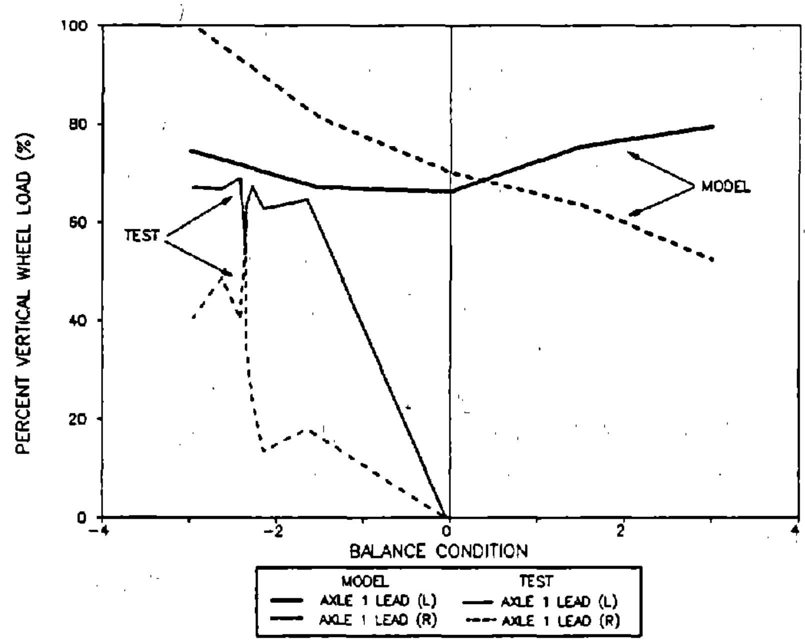

Figure 131. Minimum Percent Vertical Wheel Load for Axle 1 of the Loaded Car in the 10-Degree Dynamic Curve

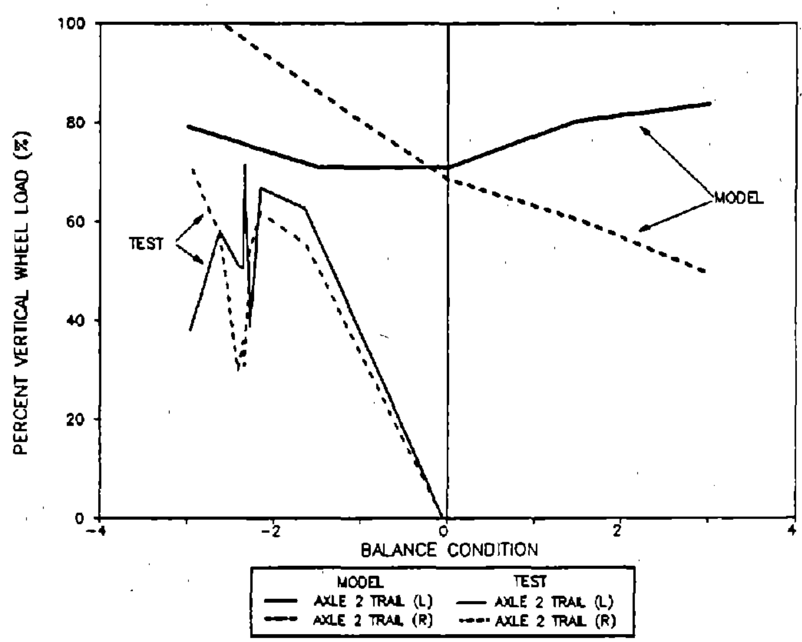

Figure 132. Minimum Percent Vertical Wheel Load for Axle 2 of the Loaded Car in the 10-Degree Dynamic Curve

Chapter XI limits for wheel L/V and axle sum L/V were also exceeded during the test, as shown in Figures 133 and 134. The loaded car showed a roll resonance at the -1.5 inch unbalance condition. This corresponds to the 19 mph roll resonance found in the twist and roll test section.

This resonant peak can be seen in the comparison of predicted and measured roll angles shown in Figure 135. The test data shows a roll angle peak at -1.5 inches of imbalance. The model data peaks at balance speed, which is approximately the same as the predicted resonant speed in the twist and roll test zone. Predicted angles are greater than those measured, but none come close to the 6-degree Chapter XI limit. It is obvious therefore that the NUCARS model does not accurately replicate the dynamic curving track tests. This is confusing because NUCARS did produce good correlation for the twist and roll test sections.

The pre-test predictions were similar to the post-test predictions, and therefore did not predict the severe test results.

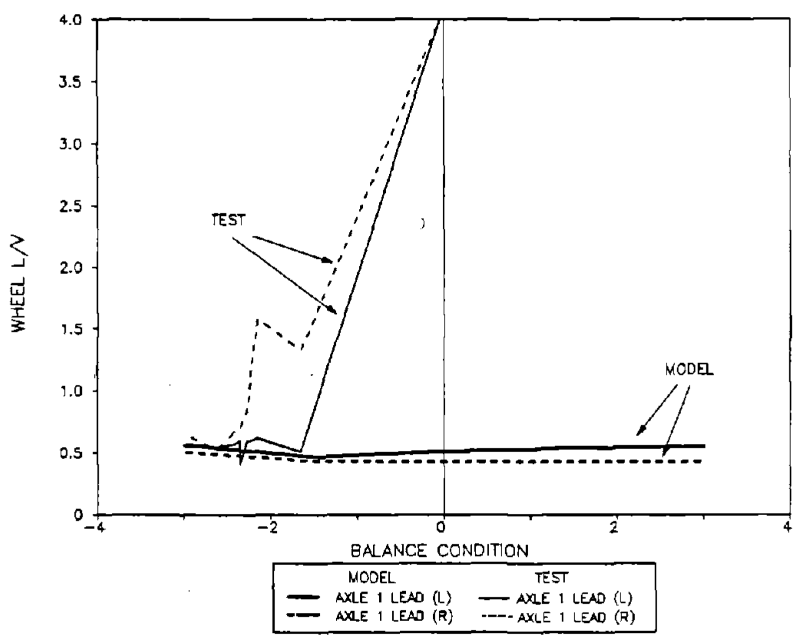

Figure 133. Maximum Wheel L/V Ratios for Axle 1 of the Loaded Car in the 10-Degree Dynamic Curve

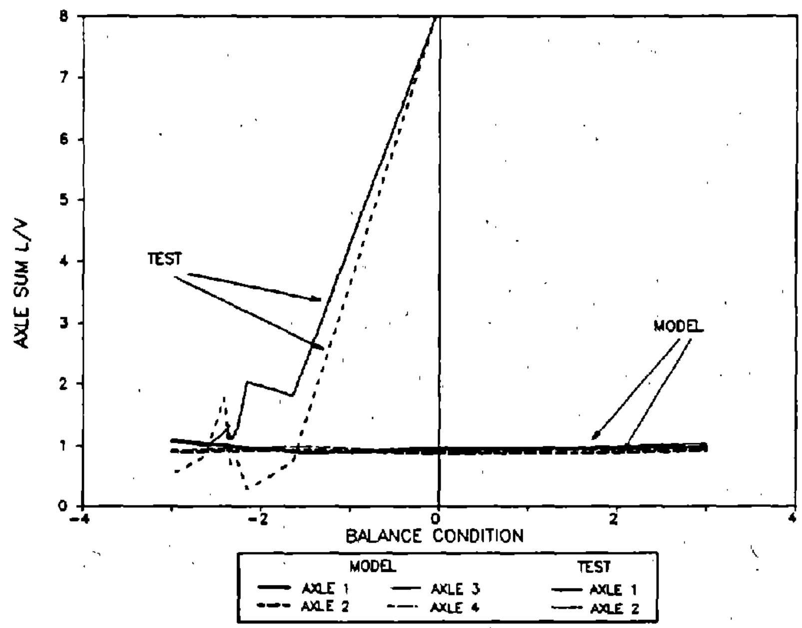

Figure 134. Maximum Axle Sum L/V Ratios for the Loaded Car in the 10-Degree Dynamic Curve

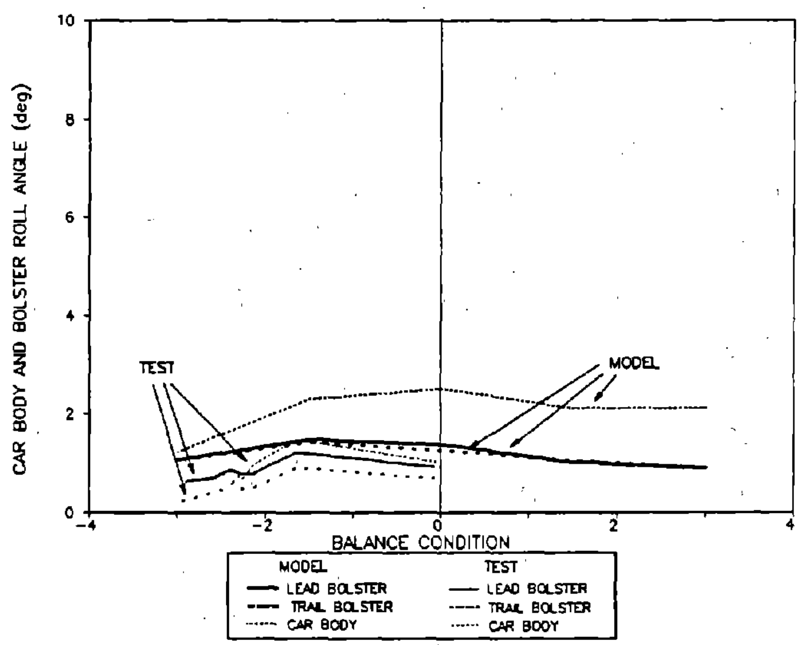

Figure 135. Maximum Peak to Peak Roll Angles for the Car Body and Bolsters of the Loaded Car in the 10-Degree Dynamic Curve

8.9 SUMMARY OF TRACK TEST AND MODELING RESULTS

NUCARS predictions were made for all track test conditions. Predictions were made with measured track geometry for all but the single bounce, and yaw and sway test zones. Mathematically defined track geometry was used for these two test zones. A summary of the tests and NUCARS modeling is given in Table 9. Both pre-test and post-test predictions are tabulated to compare results.

Chapter XI limiting criteria were predicted and/or measured to be exceeded for the following test regimes. As noted, due to the possible errors in the empty car wheel force data, some results could be erroneous.

1. Empty Car Tangent Hunting (model and test)
2. Empty Car Curved Hunting (model and test)
3. Loaded Car Tangent Hunting (model)
4. Loaded Car Curved Hunting (model)
5. Empty Car Pitch and Bounce (questionable test)
6. Empty Car Single Bounce (model and test)
7. Loaded Car Single Bounce (test)
8. Empty Car Twist and Roll (model and test)
9. Empty Car Yaw and Sway (model)
10. Loaded Car Yaw and Sway (model and test)
11. Empty Car Bunched Spiral Entry (questionable test)
12. Empty Car Bunched Spiral Exit (questionable test)
13. Empty Car Dynamic Curve (questionable test)
14. Loaded Car Dynamic Curve (test)

Table 9. Summary Comparison of NUCARS Predictions and Track Test Results

CHAPTER XI RESULTS				
TEST CASE	PRE-TEST NUCARS PREDICTION	POST-TEST NUCARS PREDICTION	TRACK TEST RESULT	COMMENTS
Tangent Hunting Empty	1g P-P @ 57.5 mph	derailment @ 50 mph	1g p-p @ 55 mph	Model and test exceed Ch. XI
Curved Hunting Empty	0.25g P-P @ 70 mph	derailment @ 50 mph	1g p-p @ 55 mph	Model and test exceed Ch. XI
Tangent Hunting Loaded	0.6g P-P @ 70 mph sustained oscillations do not exceed Chapter XI	1g p-p @ 60 mph	sustained oscillations @ 50mph	Model predictions exceed Ch. XI, confirm test observations (not required by Ch. XI)
Curved Hunting Loaded	0.35 g P-P @ 70 mph	1g p-p @ 60 mph	sustained oscillations @ 50mph	Model predictions exceed Ch. XI, confirm test observations (not required by Ch. XI)
Twist & Roll Empty	5.3 deg P-P 30 % min wheel load 35 mph resonance	22 % min wheel load 7.9 deg p-p roll angle 30 mph resonance	0% min wheel load (dubious) 6.8 deg p-p roll angle 33 mph resonance	Model and test exceed Ch. XI
Twist & Roll Loaded	3 deg P-P 45 % min wheel load 25 mph resonance	40% min wheel load 4.2 deg p-p roll angle 20 mph resonance	0% min wheel load 3.5 deg p-p roll angle 19 mph resonance	Test exceeds Ch. XI, test suspension deflections much greater than predicted.
Pitch & Bounce Empty	75 % min wheel load @ 70 mph (no resonant speed)	15 % min wheel load 60 mph resonance	0 % min wheel load (dubious data) 60 mph resonance	Test forces dubious, model suspension deflections match test (not required by Ch. XI)
Pitch & Bounce Loaded	75 % min wheel load 65-70 mph resonance	45 % min wheel load 50 mph resonance	42 % min wheel load 55 mph resonance	Model matches test
Single Bounce Empty	10 % min wheel load 40 mph resonance	0 % min wheel load @ 40 mph	0 % min wheel load (dubious data)	Model matches test, test forces dubious (not required by Ch. XI)
Single Bounce Loaded	25 % min wheel load 65 mph resonance	34 % min wheel load @ 70 mph	0 % min wheel load @ 35 mph	Test exceeds Ch. XI, test suspension deflections much greater than predicted.
Yaw & Sway Empty	Erroneous results Derailment above 30 mph	1.8 axle sum L/V @ 50 mph 0.65 truck side L/V @ 30 mph	0.5 axle sum L/V @ 45 mph 0.2 truck side L/V @ 45 mph	Model exceeds Ch. XI, test forces dubious, test does not match model (not required by Ch. XI)
Yaw & Sway Loaded	Erroneous results Derailment above 30 mph	1.45 axle sum L/V @ 25 mph 0.7 truck side L/V @ 25 mph	1.37 axle sum L/V @ 40 mph 0.79 truck side L/V @ 40 mph	Model and test exceed Ch. XI

Table 9. Summary Comparison of NUCARS Predictions and Track Test Results--(Continued)

CHAPTER XI RESULTS				
TEST CASE	PRE-TEST NUCARS PREDICTION	POST-TEST NUCARS PREDICTION	TRACK TEST RESULT	COMMENTS
Steady State Curving Empty	0.55 wheel L/V 0.82 axle sum L/V 12-deg curve -3 in. imbalance	0.79 wheel L/V 1.15 axle sum L/V 12-deg curve	1.1 wheel L/V 1.7 axle sum L/V 12-deg curve	Test forces dubious, model matches test for sharp curves
Steady State Curving Loaded	0.42 wheel L/V 0.82 axle sum L/V 12-deg curve -3 in. imbalance	0.47 wheel L/V 0.85 axle sum L/V	0.55 wheel L/V 1.05 axle sum L/V	Model and test match for sharp curves
Curve Entry Empty	.65 wheel L/V 65% min. wheel load +3.0 in. imbalance	0.74 wheel L/V 25 % min wheel load +3.0 in. imbalance	4.0 wheel L/V (dubious) 0 % min wheel load (dubious)	Test wheel forces dubious
Curve Exit Empty	.58 wheel L/V 68% min wheel load +3.0 in. imbalance	0.68 wheel L/V 59 % min wheel load +3.0 in. imbalance	3.3 wheel L/V (dubious) 0 % min wheel load (dubious)	Test wheel forces dubious
Curve Entry Loaded	0.45 wheel L/V 65 % min wheel load -3.0 in. imbalance	0.43 wheel L/V -3.0 in. imbalance 71 % min wheel load +3.0 in. imbalance	0.65 wheel L/V 63 % min wheel load	Test shows slightly poorer response than model but well within Ch. XI limits
Curve Exit Loaded	0.45 wheel L/V 65 % min wheel load -3.0 in imbalance	0.48 wheel L/V -3.0 in. imbalance 72 % min wheel load +3.0 in. imbalance	0.62 wheel L/V 50 % min wheel load	Test shows slightly poorer response than model but well within Ch. XI limits
Dynamic Curving Empty	Dubious results Derailment at balance speed and above	2.75 wheel L/V 3.0 axle sum L/V 30 % minimum wheel load 4.8 deg p-p roll angle +3.0 in. imbalance	2.9 wheel L/V 3.25 axle sum L/V 0 % minimum wheel load 0.9 deg p-p roll angle +1.0 in. imbalance	Test wheel forces dubious, model and test suspension deflections similar (not required by Ch. XI)
Dynamic Curving Loaded	0.5 wheel L/V 0.95 axle sum L/V 65% min wheel load 1.3 deg p-p roll angle roll resonance @ 24 mph balance speed	0.6 wheel L/V 1.1 axle sum L/V 42 % minimum wheel load 2.45 deg p-p roll angle +3.0 in. imbalance	4.0 wheel L/V 8.0 axle sum L/V 0 % minimum wheel load 1.45 deg p-p roll angle balance speed	Test exceeds Ch. XI, shows greater suspension deflections than model

9.0 CONCLUSIONS AND RECOMMENDATIONS

9.1 VEHICLE CHARACTERIZATION

The MSU proved to be a successful facility for performing vehicle characterizations. Tests were accomplished in less time than with previous facilities such as the VTU. Results are also believed to be more accurate. The data collection (and control) system allowed immediate post-test analysis of some data, allowing real time verification of successful test runs. Post-test data analysis was also simplified due to the easy transfer of data from collection to analysis computers.

Subsequent to this test the MSU has been used to characterize the suspensions of many other vehicles, including all the prototype Rail Garrison cars and the test locomotive for the FRA Locomotive Heavy Axle Load Test. The MSU represents a major advance in the TTC's capability for determining vehicle suspension parameters.

The RVID parameter identification software was used with some success to identify suspension characteristics. Its use for identifying more complex characteristics, such as the car body to truck bolster roll suspension, has, however, proved difficult. Satisfactory results for these suspensions have yet to be achieved. In addition, some doubt remains about the characteristics developed for the secondary lateral suspension.

Successful attempts were made to develop dynamic suspension characteristics from the track test data for the lateral and yaw suspensions. These were to be compared to the MSU and air table test results. Revisions to the NUCARS suspension characteristics for the post-test modeling were based on these comparisons. Some of the characteristics developed by these means matched well with results being found in other AAR research programs and were felt to be trustworthy for use in this project. However, the data developed for the center bowl friction appeared totally erroneous, being 3 to 5 times larger than was reasonable. This data was discarded.

The general method of determining suspension characteristics from test data appeared successful however. More accurate instrumented wheel sets will be required to provide reliable results by these means, as well as more extensive instrumentation to measure the accelerations of all the trucks' component parts. Doubt was thrown on the primary shear pad suspension characteristics measured on the MSU. It is likely that estimates of the shear pad stiffness made from the track test data could have clarified this problem.

9.2 NUCARS PREDICTIONS

NUCARS predictions were made after completion of the track tests using actual measured track geometry as input. Pre-test predictions were made using mathematically defined track input. In general the post-test predictions match the test results more closely than the pre-test predictions, for two reasons:

1. The post-test predictions were made with track input that was an exact measurement of most of the test zones.
2. The post predictions were made with revised lateral and yaw suspension characteristics derived from dynamic measurements made during the track tests. Measured axle misalignments were also included.

Accurate modeling of the friction characteristics was shown to have a significant influence on the NUCARS predictions. A small change (altering the slope of the linear viscous band) had an enormous effect on the pitch and bounce simulations, ultimately producing very good correlation with the test data especially in the pitch and bounce test section. This method of improving the modeling of friction characteristics is a compromise which requires increasing the run time for the computer simulations, due to the required decrease in integration time step. A new method for simulating friction has been developed for inclusion in the next version of NUCARS, which should alleviate this problem.

Correlation between test and model was not as good where combined vertical and lateral motion in the main spring group was occurring. This is also attributable to inaccurate modeling of friction suspension elements; in this case the friction wedge. Current versions of NUCARS model the vertical and lateral friction forces generated by this wedge as independent from each other. This is inaccurate when combined vertical and lateral motion is occurring. New methods are currently being developed to model this behavior more accurately in the next version of NUCARS.

The modeling of the primary shear pad stiffness also appears to have been incorrect. This was probably too stiff, preventing a good correlation with the test data in the curves and possibly the yaw and sway test zone. It appears to have had more effect on the empty car predictions than the loaded car.

There was some evidence that the center bowl friction level used in the modeling was too high. Lowered center bowl friction levels in combination with reduced primary shear stiffness would probably result in improved correlation with the test results for the curving, and yaw and sway test regimes. Attempts to identify a dynamic center bowl friction characteristic from the test data were made. Unfortunately these resulted in unreasonably high values. Preliminary modeling attempts made with this data produced very poor results and were discarded. It is expected that improved instrumentation could produce better characteristics in order to more accurately measure this crucial parameter under dynamic conditions.

9.3 INSTRUMENTED WHEEL SETS

Results of the empty car tests show the instrumented wheel sets used for these tests **are not** accurate enough for testing an empty car. The key measurement tools for the Chapter XI tests are the instrumented wheel sets. It is apparent from the tests of the empty car that the accuracy of the wheel sets is in doubt. There is a large amount of scatter in the force data for the empty car. The data also had a lot of noise, which required filtering at 5 Hz.

The static load of the empty car is approximately 5000 pounds per wheel. In order to measure a minimum wheel load of 10 percent, the resolution of the wheel set must be better than 250 pounds. This is less than these wheel sets are capable of measuring.

No published figures are available for the measurement accuracy and resolution of these instrumented wheel sets. Estimates by AAR engineers indicate a minimum resolution of 2000 pounds. Examination of the test data from the empty car tests showed that the average vertical wheel load measured while running on smooth tangent track could vary from as low as 1000 pounds up to 7000 pounds. During dynamic tests, negative vertical wheel loads (a physical impossibility) were occasionally recorded.

Appendix B of Chapter XI (attached as Appendix A of this report) specifies minimum standards for measurement accuracy for instrumented wheel sets. This requires the wheel set to be capable of resolving vertical forces to within 5 percent and lateral forces to within 10 percent of the actual loads with a minimum resolution of 0.5 percent of the static load. For PSMX 111, this implies a resolution of +/- 25 pounds. It is probably not practical to build a wheel set with this sort of accuracy. An alternative specification could be written in terms of absolute minimum resolution, such as 1000 pounds with an error band of +/- 500 pounds.

A review of currently available instrumented wheel sets indicates that none is likely to be able to resolve the very low loads imparted by a light car such as PSMX 111. It is doubtful that any are capable of better than + /- 1000 pound accuracy. The measurement accuracy of the L/V ratios is even worse because these are taken as a ratio of two measured forces.

For example, assume a wheel set was capable of measuring vertical and lateral forces to within + /- 1000 pounds. Assume the actual loads were 4000 pounds vertical and 4000 pounds lateral, which gives an L/V of 1.0. If the wheel set was in error by + 1000 pounds vertically and -1000 pounds laterally it would have read 5000 pound vertical load and 3000 pound lateral for an L/V of 0.6. Thus the measurement error would have produced a 40 percent error in the L/V. L/V of 1.0 would indicate unsafe performance, while a value of 0.6 would not cause concern.

The forgoing analysis shows that the empty car test results should be viewed with caution. Most of the force and L/V data is subject to a very wide margin of error. This also sheds new light on the results obtained from Lightweight Car 1 tests. All of the NUCARS modeling for that project was performed for the empty car. Thus all comparisons with test data were made for a vehicle with a static wheel load of 6700 pounds. This light load plus the fact that wheel set force data was only calculated four times for every wheel revolution could explain some of the scatter seen in the empty car test data and some of the mismatches with model predictions for the two-axle Frontrunner car.

9.4 PSMX 111 TEST VEHICLE PERFORMANCE

The PSMX 111 exceeded the Chapter XI limiting criteria in several test zones. Much of the poor behavior appears related to the shear pads which allow the axles to oscillate laterally and in yaw promoting hunting behavior. The hunting occurred in all tangent track test zones for both the loaded and empty car. Hunting is normally only expected to occur on empty vehicles.

This same freedom of motion in the shear pads does however give the car good curving performance. Steady curving, curve entry and curve exit gave no performance problems. The few wheel lifts seen during the empty car curve entry and exit tests are probably false readings by the instrumented wheel sets.

The wheel lifts and large roll angles seen in the empty car single bounce and empty car twist and roll test zones are probably related to the very small suspension motions that occurred. The PSMX 111 is equipped with variable friction damping in the vertical suspension. Despite this the friction levels measured during the characterization tests were very large at the fully extended position of the suspension. This high friction kept the truck suspension mostly locked up during the empty car tests causing the car to roll and lift off the rails in response to the perturbations.

It is normally expected that variable friction damping would provide much less damping at the fully extended position of the suspension than was measured. The trucks used for this test were very new. It is possible that the dampers were not fully broken in and had higher friction than expected.

It must be noted that the PSMX 111 was equipped with special trucks for the purpose of this test. These were different than the trucks originally installed under this car. The original trucks had a two stage spring arrangement. This would provide a much softer spring rate for the empty car while providing a similar stiffness for the loaded car. Car response would obviously change. It is not known how much effect this would have had on overall vehicle performance.

9.5 CHAPTER XI TESTS AND CRITERIA

9.5.1 Minimum Wheel Load Criterion.

Consideration should be given to modifying the Chapter XI criterion for minimum vertical wheel load to include an absolute lower limit. For example, the criterion could state that the minimum vertical wheel load shall be no less than 1000 pounds or 10 percent of the static wheel load, whichever is greater.

The criterion for minimum vertical wheel load is currently set at 10 percent of the static wheel load. For a lightweight car such as the PSMX 111, this is only 500 pounds when the car is empty. Aluminum coal gondolas similar to the PSMX 111 are becoming more common. This may be too low a limit. It is conceivable that further developments of skeleton flatcars could achieve even lower empty car static wheel loads with proportionately smaller 10 percent minimum loads.

9.5.2 Empty Car Pitch and Bounce, Single Bounce, Yaw and Sway, and Dynamic Curving Tests

Test results and NUCARS predictions indicate that tests and analyses of the empty car should be included for all test sections. This conclusion reiterates a conclusion of the Lightweight Car 1 test program.

The intent of Chapter XI is that all new vehicles be submitted to an extensive pre-test analysis. The results of this analysis is to be used to guide the choice of which (if any) tests are to be performed. Thus only cars that appear likely to show poor performance in all test regimes, both loaded and empty, would be subjected to all the tests. If this intent is followed only a few new designs would see a significant increase in the cost of Chapter XI because of the addition of the extra empty regimes. The benefit would be to ensure that all vehicles are examined for safety in regimes where some vehicles have shown poor performance.

The pitch and bounce, single bounce, yaw and sway, and dynamic curving tests are not currently required by Chapter XI for the empty car. The PSMX 111 car did fail the single bounce, with this result being confirmed by the NUCARS predictions. In most other test regimes, wheel lift or high L/V ratios were indicated by the test data, but the unreliable instrumented wheel sets cast some doubt on this data, which is not corroborated by the post-test predictions.

The predictions of the empty car in the dynamic curve show the car coming close to exceeding some limiting criteria. This is confirmed by the test data. The predictions for the empty car in the yaw and sway zone showed L/V ratios in excess of Chapter XI limits. It appears that these test zones should also be included for the empty car tests.

9.5.3 Loaded Car Lateral Stability (Hunting) Tests

Loaded car lateral stability test should be considered for inclusion in Chapter XI. These tests in curves and tangent track are not currently required by Chapter XI for the loaded car. The loaded PSMX 111 car appeared to fail the lateral stability tests on both curved and tangent track. Post-test predictions also indicate that it would have failed these tests. It is unusual for loaded freight cars with three-piece trucks to hunt. It is probable that the hunting is related to the soft rubber primary shear pads installed on this car.

The Chapter XI tests are supposed to evaluate all new designs of freight cars for safety performance. It is very possible that other new designs of car might also have unforeseen high speed stability problems when loaded due to an unusual or different suspension design. Inclusion of a loaded car hunting test (and the empty car tests mentioned in section 9.5.2) would give the Car Engineering Committee the option of requesting the test(s) if the vehicle design and pre-test analyses suggest it might be necessary.

9.5.4 Lateral Acceleration Criteria for Lateral Stability Tests

The lateral car body acceleration measurement required by Chapter XI does not necessarily measure vehicle safety performance but is more a measure of vehicle ride quality. Consideration should be given to changing or deleting this requirement.

The results of the empty car hunting tests showed sustained lateral axle motions indicating that severe hunting was occurring. The lateral car body accelerations did not however exceed the Chapter XI limits of 1.5 g for a single occurrence or 1.0 g for the 20 second sustained period. The tests were terminated because the accelerometers on the axles showed extreme motion. The axle sum L/V ratio data is dubious but appears to support the assumption that dangerous hunting was occurring. These results were supported by NUCARS, which predicted derailment.

For vehicles with ordinary three-piece trucks the car body lateral acceleration measurement may give some indication of vehicle lateral behavior. However, in this instance the soft rubber primary shear pad suspension acted to improve the ride quality of the car by isolating the axle motions. The Chapter XI acceleration criteria therefore failed to detect the dangerous vehicle behavior. In light of the poor performance of the instrumented wheel sets, when lightly loaded, a better criterion is needed. This could be based on axle rather than car body lateral accelerations.

9.5.5 Roll Angle Criteria

A measurement of truck bolster to car body roll angle should considered for inclusion as a Chapter XI limiting criteria. This can be done inexpensively with two string potentiometers.

The Chapter XI limiting criterion for car body roll angle is 6-degrees peak to peak. This is intended to limit car body to bolster roll angle in order to prevent the car body center plate from lifting out of the bolster center bowl.[10] During the empty car twist and roll tests the bolster to car body roll angles did exceed 6-degrees, showing that the center plate was lifting out of the center bowl. This is a dangerous situation that should be detected. Car body roll of itself is not dangerous except as it affects clearances in tunnels and with wayside structures and other trains. Although it did not occur during this project, it is conceivable that the car body roll angle could be less than 6-degrees while at the same time the center plate lifts out of the center bowl.

Two string potentiometers per truck were all the instrumentation required to measure this bolster to car body angle. This measurement could be added to those required by Chapter XI. It may be possible to then delete the requirement for measuring car body roll angle alone. Car body roll angle is usually measured with a roll gyro which is less reliable and more expensive to use than a string potentiometer.

A second observation also applies to the car body roll angle measurement. Many new design cars such as the two axle Frontrunner car (Lightweight Car 1) do not have trucks, center bowls or center plates. It seems meaningless to apply this criterion to these cars. If a car body roll angle criterion is desirable it should be based on a study of the relationship of roll angle to derailment and wheel climb potential. In the case of vehicles which carry separately suspended loads (such as trailers), it may also be necessary to set a limit on safe load roll angle.

9.5.6 Summary

As car builders design new freight cars for higher productivity, lower weight, and improved dynamic performance it is likely that many new designs will not have three-piece trucks, or even two-axle trucks. Chapter XI was written to provide a means of evaluating the performance of these new designs. Many of the suggestions and observations listed in the subsections above are due to the fact that the wording of Chapter XI appears to imply that the test vehicle has two-axle trucks and a single car body. Interpreting the requirements for unusual cars can be difficult, and some of the measurements may not be applicable. It is suggested that the wording be altered for more general application. Chapter XI bases evaluation of car safety performance on wheel/rail interaction forces. Supplemental measurements such as roll angles and accelerations were intended to determine safety performance of any design of car. Some of these appear to be based on current design practices and could be changed for clarity and a better safety evaluation.

In some cases, it is essential to use the computer model to support the track test results to determine compliance with the established safety criteria. In the case of the empty car tests the instrumented wheel sets were so unreliable that the NUCARS predictions were required to verify the wheel set force measurements. It may be difficult to ever build an instrumented wheel set capable of measuring empty car forces as accurately as required by Chapter XI. Therefore computer simulations should be carried out to assist in interpreting test results. In order to make valid comparisons between test and model, a small amount of additional instrumentation to measure suspension deflections would be required. In the case of a traditional vehicle with three-piece trucks, this could be as few as four string potentiometers measuring spring deflections, with two more measuring bolster rotation angles.

Two issues involving repeatabilty and accuracy of Chapter XI type tests and analyses arise from the model results.

The first issue arises from the fact that small changes in friction level in a suspension can cause large changes in vehicle behavior. This means that simulation of actual vehicle performance will be difficult without careful measurement of the suspension characteristics. The friction levels are also known to change considerably over the lifetime of a vehicle. Thus it is likely that a result predicted or measured when a vehicle is new is likely to change over time. This implies that to properly analyze vehicle safety performance, a parameter variation should be made of the critical suspension elements to examine the effects of wear and changing friction levels. Similar parameter studies would also have to be conducted to account for substitution of similar but not identical components, such as changing side bearing designs.

The second repeatability issue involves the difference between simulation results obtained with measured track and mathematical representations of the perturbations. This is really two problems, the first being that the mathematical representation does not include any surface roughness information. This surface roughness probably acts to break out the friction in the various suspensions, giving an apparent reduction in damping. This problem could be remedied in the NUCARS model by imposing a random surface roughness onto the mathematical perturbation shapes. The second problem involves the accuracy to which the test perturbations are constructed and maintained. If the perturbations are not maintained within certain limits the tests will not be repeatable. Research must be conducted to identify these limits, and these limits should be specified in Chapter XI. The tolerances must not be set too tightly however, or else the cost of maintaining the tracks will become prohibitive.

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APPENDIX A

CHAPTER XI
SERVICE-WORTHINESS TESTS AND ANALYSES FOR NEW FREIGHT CARS

CHAPTER XI
SERVICE-WORTHINESS TESTS AND ANALYSES FOR NEW FREIGHT CARS
Adopted 1987

11.1. PURPOSE AND SCOPE

This chapter presents guidelines for testing and analysis to ascertain the interchange-service worthiness of freight cars. The regimes of vehicle performance to be examined are divided into two sections. Section 1 covers structural static and impact requirements. Section 2 covers vehicle dynamic performance, with the following regimes to be examined: hunting, car body twist and roll, pitch and bounce, yaw and sway and longitudinal train action.

Braking performance, structural fatigue life, car handling, and other design considerations must be considered in accordance with requirements outlined by other chapters of this specification.

The methods presented provide acceptable approaches to the analysis and measurement of car parameters and performance. Other rational methods may be proposed at the time of submission for design approval. Their use and applicability must be agreed to by the Car Construction Committee.

11.2. STATIC AND IMPACT TEST REQUIREMENTS

Application for approval of new and untried types of cars, along with supporting data specified in paragraph 1.2.3, shall be submitted to the Director—Technical Committees Freight Car Construction prior to initiation of official AAR testing. A proposed testing schedule and testing procedures will be submitted sufficiently in advance of tests to permit review and approval of the proposal and assignment of personnel to witness tests as AAR observers. Tests will be in conformity with the following and all costs are to be borne by the applicant, including observers.

11.2.1. TEST CONDITIONS

11.2.1.1.

A car of the configuration proposed for interchange service must be utilized for all tests. Deviation from such configuration is only permitted with the explicit permission of the Car Construction Committee.

During impact tests, the test car will be the striking car and shall be loaded to AAR maximum gross rail load for the number and size of axles used under car (see 2.1.5.17). Exceptions to this procedure will be considered by the Car Construction Committee when justified by the applicant.

Cars designed for bulk loading shall have a minimum of 85% of the total volume filled.

Cars designed for general service, other than bulk loading, shall be loaded so that the combined center of gravity of car and loading is as close as practicable to the center of gravity computed in accordance with the requirements of 2.1.3, except that general service flat cars may be loaded by any practicable method. The loads shall be rigidly braced where necessary, and various types of loads should be used to test each component to its maximum load.

The test car may be equipped with any AAR-approved draft gear or any AAR-approved cushioning device for which the car was designed.

11.2.1.2.

The cars, other than the test car, shall be of seventy ton nominal capacity, loaded to the allowable gross weight on rails prescribed in 2.1.5.17. A high density granular material should be used to load cars to provide a low center of gravity, and the load should be well braced to prevent shifting. Such cars shall be equipped with draft gears meeting the requirements of AAR Specification M-901, except at the struck end where M-901E rubber friction gear shall be used.

Free slack between cars is to be removed, draft gears are not to be compressed. No restraint other than handbrake on the last car is to be used.

11.2.2. INSTRUMENTATION

The coupler force shall be measured by means of a transducer complying with AAR Specification M-901F, or other approved means. Instrumentation used for recording of other data shall be generally acceptable type properly calibrated and certified as to accuracy.

Speed at impact shall be recorded.

11.2.3. STATIC TESTS

11.2.3.1. COMPRESSIVE END LOAD

A horizontal compressive static load of 1,000,000 lbs, shall be applied at the centerline of draft to the draft system of car/unit structure interface areas, and sustained for a minimum 60 seconds. The car/unit structure tested shall simulate an axially loaded beam having rotation free-translation fixed end restraints. (See Figure 11.2.3.1).

No other restraints, except those provided by the suspension system in its normal running condition, are permissible. Multi-unit car must have each structurally different unit subjected to such test, also two empty units joined together by their connector shall undergo this test to verify the connectors compressive adequacy and its anti-jackknifing properties.

The test is to be performed with the car subjected to the most adverse stress or stability conditions (empty and/or loaded).

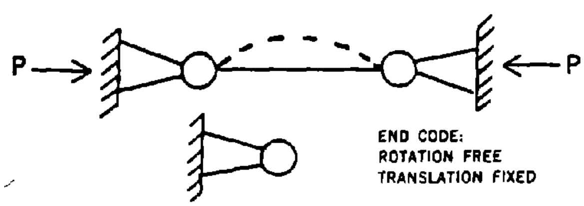

Figure 11.2.3.1

11.2.3.2. COUPLER VERTICAL LOADS

A vertical upward load shall be applied to the coupler shank immediately adjacent to the striker face or to the face of the cushion unit body at one end of the car, sufficient in magnitude to lift the fully loaded car free of the truck nearest the applied load, and held for sixty seconds. Cushion underframe cars having sliding sill are excluded from the requirements of this paragraph.

For cushion underframe cars having sliding sills, a vertical upward load shall be applied to the sliding sill in a plane as near the ends of the fixed center sills as practicable, sufficient in magnitude to lift the fully loaded car free of the truck nearest the applied load, and held for sixty seconds.

For all cars, a load of 50,000 pounds shall be applied in both directions to the coupler head as near to the pulling face as practicable and held for sixty seconds.

11.2.3.3. CURVE STABILITY

The test consist is to undergo a squeeze and draft load of 200,000 lbs. without car body-suspension separation or wheel lift. Load application shall simulate a static load condition and shall be of minimum 20 seconds sustained duration.

Cars consisting of more than two units shall be tested with a minimum of three units in the test consist. The number of units used shall generate maximum load in the critical L/V location of the car.

For the purpose of this test, wheel lift is defined as a separation of wheel and rail exceeding 1/8″ when measured 2⅝″ from the rim face at the inside of curve for buff and outside for draft.

Empty car shall be subjected to squeeze and draft load on a curve of not less than 10 degrees. The curve is to have ½″ maximum superelevation. The test car is to be coupled to a "base car" as defined in paragraph 2.1.6.1. or a like car which ever is most severe and a "long car" having 90′ over strikers, 66′ truck centers, 60″ couplers and conventional draft gear.

The test consist shall have means for measuring and recording coupler forces.

11.2.3.4. RETARDER AND "HOT BOX" DETECTION

Cars with other than conventional 3 piece trucks must be operated while fully-loaded over a hump and through a retarder. Retarder shall be operated to determine capability to brake the test cars. Such cars must also demonstrate their compatibility with hot box detection systems or be equipped with on-board hot box detection systems.

11.2.3.5. JACKING

Vertical load capable of lifting a fully loaded car/unit shall be applied at designated jacking locations sufficient to lift the unit and permit removal of truck or suspension arrangement nearest to the load application points.

11.2.3.6. TWIST LOAD

Loaded car/unit shall be supported on the side bearings or equivalent load points only. Diagonally opposite bearing or load point support shall be lowered through a distance resulting from a calculated 3″ downward movement of one wheel of the truck or suspension system supporting it. No permanent deformation of car/unit structure shall be produced by this test.

11.2.4. IMPACT TESTS

These requirements apply to all cars except those exempted by other specification requirements.

11.2.4.1. SINGLE CAR IMPACT

The loaded car shall be impacted into a string of standing cars consisting of three nominal 70-ton capacity cars, loaded to maximum gross weight on rails as described in paragraph 2.1.5.17. with sand or other granular material, equipped with M-901E rubber-friction draft gear at the struck end and with the hand brake on the last car on the non-struck end of the string tightly set. Free slack between cars is to be removed; however, draft gears are not to be compressed. No restraint other than handbrake on the last car is to be used.

A series of impacts shall be made on tangent track by the striking car at increments of two miles per hour starting at six miles per hour until a coupler force of 1,250,000 pounds or a speed of fourteen miles per hour has been reached, whichever occurs first.

A car consisting of two or more units must also undergo impact testing as outlined above with the leading unit of the test car being empty for a two-unit car, or with the first two units being empty for a three (or more) unit car. No carbody-suspension disengagement or wheel lift is permitted during the partially loaded impact tests.

11.2.4.2. DYNAMIC SQUEEZE

(Optional–May be performed in lieu of or in addition to static end compression test if requested by the Car Construction Committee.)

The striking and standing car groups shall each consist of six cars, in which the test car may be the lead car in either group. All cars except the test car shall be as prescribed in 11.2.1.2. The brakes shall be set on all standing cars after all slack between cars has been eliminated. There shall be no precompression of the draft gears. The standing cars shall be on level tangent track. The striking cars, coupled together, shall be adjusted, if necessary, to restore the original conditions.

A series of impacts shall be made at increments of two miles per hour starting at six miles per hour until a coupler force of 1,250,000 pounds or a speed of fourteen miles per hour has been reached, whichever occurs first.

11.2.5. INSPECTION

A visual inspection of the test car shall be made after each static test and after each impact. Following the impact tests, the car shall be unloaded and inspected.

Any permanent damage to any major structural part of the car, found before or after all tests are completed, will be sufficient cause for disapproval of the design. Damage will be considered permanent when the car requires shopping for repairs.

11.3. TRACK-WORTHINESS ASSESSMENT

11.3.1. METHODOLOGY

Regimes are identified, representative of the performance of the car in service. Tests are defined for each regime. The results of the tests are an indication of the car's track-worthiness. In most regimes, analytic methods are also available to permit prediction to be made of the performance of the car, to the degree of accuracy required.

The characteristic properties of the car body and its suspension, required for the analysis, shall be supported by evidence of their validity. Characterization tests, such as those defined in Appendix A, are required to verify the values used in the analyses.

11.3.2. TRACK-WORTHINESS CRITERIA

The criteria applied to the analyses and tests are chosen from a consideration of the processes by which cars deviate from normal and required guidance. They are also subject to the requirement of observability in tests. Typical of these are lateral and vertical forces, the lateral over vertical force (L/V) ratios, dynamic displacements, and accelerations of the masses. These criteria are based on considerations of the processes of wheel climb, rail and track shift, wheel lift, coupler and component separation and structural integrity.

The values chosen for the criteria selected have been used in tests on cars presently in service. Those included in the body of this chapter are shown in Table 11.1. Values worse than these are regarded as having a high risk of unsafe behavior. Values better than these are regarded as indicating the likelihood of safe car performance.

Table 11.1 Criteria for Assessing the Requirements for Field Service

Regime	Section	Criterion	Limiting Value
Hunting (empty)	11.5.2	minimum critical speed (mph)	70
		maximum lateral acceleration (g)	1.0
		maximum sum L/V axle	1.4*
Constant curving (empty and loaded)	11.5.3	95th percentile maximum wheel L/V	1.0
		or	
		95th percentile maximum sum L/V axle	1.4
Spiral (empty and loaded)	11.5.4	minimum vertical load (%)	10 **
		maximum wheel L/V	1.0*
		or	
		maximum sum L/V axle	1.4*
Twist, Roll (empty and loaded)	11.6.2	maximum roll (deg)***	6
		maximum sum L/V axle	1.4
		minimum vertical load (%)	10 **
Pitch, Bounce (loaded)	11.6.3	minimum vertical load (%)	10 **
Yaw, Sway (loaded)	11.6.4	maximum L/V truck side	0.6*
		maximum sum L/V axle	1.4*
Dynamic curving (loaded)	11.6.5	maximum wheel L/V	1.0*
		or	
		maximum sum L/V axle	1.4*
		maximum roll (deg)***	6
		minimum vertical load (%)	10 **
Vertical curve	11.7.2	to be added****	
Horizontal curve	11.7.3	to be added****	

* Not to exceed indicated value for a period greater than 50 milliseconds per exceedence
** Not to fall below indicated value for a period greater than 50 milliseconds per exceedence
*** Peak-to-peak
**** See the introduction to section 11.7.1

11.4. GLOSSARY OF TERMS

Radial misalignment of axles in a truck or car is the difference in yaw angle in their loaded but otherwise unforced condition. It causes a preference to curving in a given direction.

Lateral misalignment is the difference in lateral position between axles. It causes both axles to be yawed in the same direction on straight track.

Inter-axle shear stiffness, equivalent to the lozenge or tramming stiffness in 3-piece trucks, is the stiffness between axles in a truck or car found by shearing the axles in opposite directions along their axes, and measuring the lateral deflection between them.

Inter-axle bending stiffness is the stiffness in yaw between axles in a truck or car.

Bounce is the simple vertical oscillation of the body on its suspensions in which the car body remains horizontal.

Pitch of the body is the rotation about its transverse axis through the mass center.

Body yaw is the rotation of the body about a vertical axis through the mass center.

Body roll is the rotation about a longitudinal axis through the mass center.

Upper and lower center roll are the coupled lateral motion and roll of the body center of mass. They combine to give an instantaneous center of rotation above or below the center of mass. When below the center of mass, the motion is called lower center roll. When above, the motion is called upper center roll.

Sway is the coupled body mode in roll and yaw and it occurs where the loading is not symmetrical.

Unbalance is used in this chapter to mean the additional height in inches, which if added to the outer rail in a curve, at the designated car speed, would provide a single resultant force, due to the combined effects of weight and centrifugal force on the car, having a direction perpendicular to the plane of the track. Thus, the unbalance (U) is defined as:

$$\text{Unbalance } U = \frac{V^2 D}{1480} - H$$

where,
- D is the degree of the curve.
- V is the vehicle speed in mph.
- H is the height, in inches, of the outer rail over the inner rail in the curve.

Effective conicity, E, of a wheel on a rail is its apparent cone angle used in the calculation of the path of the wheel on the rail. It is defined as:

$$E = A\left(\frac{R_w}{R_w - R_R}\right)$$

where,
- A is the angle of the contact plane, between the wheel and rail, to the plane of the track.
- R_w is the transverse profile radius of the wheel.
- R_R is the transverse profile radius of the rail.

The effective conicity of the modified Heumann wheel of Figure 8.1 on AREA 132 1b rail, under conditions of tight gage, is between 0.1 and 0.3.

Three ratios of lateral (L) to vertical (V) forces are used as criteria in the assessment of car performance. These are:

(1) **The individual wheel L/V**, (or wheel L/V). This is defined as the ratio of the lateral force to the vertical force between the wheel and rail on any individual wheel. It is used to assess the proximity of the wheel to climbing the rail.

(2) **The instantaneous sum of the absolute wheel L/V's on an axle**, (or sum L/V axle). This is defined as the sum of the absolute values of the individual wheel L/V's on the same axle, as given in the following algebraic equation. They must be measured at the same time.

$$\text{Sum L/V axle} = \left| \text{L/V (left whl)} \right| + \left| \text{L/V (right whl)} \right|$$

It is used to assess the proximity of the wheel to climbing the rail and is more appropriate where the angle of attack of the flanging wheel to the rail does not result in full slippage at the area of contact.

(3) **The truck side L/V**, (or L/V truck side). This is defined as the total sum of the lateral forces between the wheels and rails on one side of a truck divided by the total sum of the vertical forces on the same wheels of the truck, as given in the following algebraic expression.

$$\text{Truck side L/V} = \frac{\Sigma\text{L (truck side)}}{\Sigma\text{V (truck side)}}$$

It is used to indicate the proximity to moving the rail laterally.

11.5. SINGLE CAR ON UNPERTURBED TRACK

11.5.1. GENERAL

The regimes described in this section are chosen to test the track-worthiness of the car running on premium track. They are required to establish the safety of the car from derailment under conditions basic to its performance in service and are carried out under operating conditions similar to those found in normal service, but without the effects of dynamic variations due to adjacent cars or large perturbations associated with poor track.

The parameters used in the analysis shall be confirmed in characterization tests described in Appendix A. The results of the following analyses and tests shall be included for the consideration of approval by the Car Construction Committee.

11.5.2. LATERAL STABILITY ON TANGENT TRACK (HUNTING)

This requirement is designed to ensure the absence of hunting, which can result from the transfer of energy from forward motion into a sustained lateral oscillation of the axle between the wheel flanges, in certain car and suspension designs. The analyses and tests are required to show that the resulting forces between the wheel and rail remain within the bounds necessary to provide an adequate margin of safety from any tendency to derail.

11.5.2.1. PREDICTIONS AND ANALYSES

An analysis shall be made of the critical speed at which continuous full flange contact is predicted to commence, using a validated mathematical model and the parameters measured for the empty test car. This analysis shall include predictions on tangent and on 1/2 and 1 degree curves.

The analytic requirement is that no hunting be predicted for the empty car below 70 miles per hour assuming a coefficient of friction of 0.5 and an effective conicity of 0.15, for the modified Heumann wheel profile given in Figure 8.1 of Chapter VIII, on new AREA 136 lb. rail, for axle lateral displacements up to +/− 0.2 in. on track with standard gauge.

11.5.2.2. TEST PROCEDURE AND CONDITIONS

The empty test car shall be placed at the end of the test consist, behind a stable buffer car, and operated at speeds up to 70 miles per hour on tangent class 5 or better track, with dry rail.

All axles of the lead unit or car shall be equipped with modified Heumann profile wheels as shown in Figure 8.1 of Chapter VIII, with the machining grooves worn smooth on the tread.

The rail profile shall be new AREA 136 lb. or an equivalent which, with the Heumann wheel specified, gives an effective conicity of at least 0.15 for lateral axle displacements of +/− 0.2 inch from the track center. The track gage may be adjusted in order to achieve this minimum effective conicity. If hunting is predicted for curved track in section 11.5.2.1, a special hunting test in shallow curves may be requested.

11.5.2.3. INSTRUMENTATION AND CRITERIA

The leading axle of both trucks on an end unit or car, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets, and each truck location on the end unit or car shall be equipped with a lateral accelerometer on the deck above the center of the truck.

Sustained truck hunting shall be defined as a sustained lateral acceleration greater than 1 g peak-to-peak for at least 20 consecutive seconds. No occurrences of greater than 1.5 g peak-to-peak are permitted within the same time period. The instantaneous sum of the absolute values of the L/V ratios shall not exceed 1.4 on any instrumented axle. Components of the measured accelerations and forces having frequencies above 15 hertz are to be filtered out.

The car shall not experience sustained truck hunting during the test. A record of maximum lateral acceleration and the wheel L/V's on the same axle, against speed, at the worst location, shall be submitted as required test data.

11.5.3. OPERATION IN CONSTANT CURVES

This requirement is designed to ensure the satisfactory negotiation of track curves. The analyses and tests are required to show that the resulting forces between the wheel and rail are safe from any tendency to derail and to confirm other predictions of the car behavior relating to the guidance of the car and absence of interferences.

11.5.3.1. PREDICTIONS AND ANALYSES

An analysis shall be made of the wheel forces and axle lateral displacements and yaw angles on a single car, empty and fully loaded, using a validated mathematical model. The model shall include a fundamental representation of the rolling contact forces using the geometry of the profiles of the wheel and rail, and car parameters from the measurements described in Appendix A.

Either the individual wheel L/V shall be less than 1.0 on all wheels measured, or the instantaneous sum of the absolute wheel L/Vs on any axle shall be less than 1.4, for any curve up to 15 degrees. The range of unbalance assumed shall be −3 inches to +3 inches, with a coefficient of friction of 0.5 and modified Heumann profiled wheels on new AREA 132 lb. or 136 lb. rail.

11.5.3.2. TEST PROCEDURE AND CONDITIONS

The test car shall be operated at constant speeds equivalent to unbalances of −3, 0, and +3 inches. The tests shall be run with the test car in both empty and fully loaded conditions, between two heavy buffer cars, one of which may be replaced by an instrumentation car. A complete set of tests shall be carried out in both directions and with the test consist turned in each direction, on dry rail.

The wheels of the test car shall have less than 5000 miles wear on the new profiles specified for production, except that those on instrumented wheelsets shall have modified Heumann profiles. The rail profiles shall have a width at the top of the head not less than 95 percent of the original value when new. The test curve shall be of not less than 7 degrees with a balance speed of 20 to 30 mph, and with class 5 or better track.

11.5.3.3. INSTRUMENTATION AND CRITERIA

The leading axle of both trucks on an end unit or car, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets. The lateral and vertical forces and their ratio, L/V, shall be measured for the length of the body of the curve, which must be at least 500 ft., and their maxima and means computed. Measured force components having frequencies above 15 hertz are to be filtered out.

Either the individual wheel L/V shall be less than 1.0 on all wheels measured, or the instantaneous sum of the absolute wheel L/Vs on any axle shall be less than 1.4. A record of L/V on both wheels of the instrumented axles, for each test run, shall be submitted as required test data.

11.5.4. SPIRAL NEGOTIATION AND WHEEL UNLOADING

This requirement is designed to ensure the satisfactory negotiation of spirals leading into and away from curves. The analyses and tests are required to show that the resulting forces between the wheel and rail show an adequate margin of safety from any tendency to derail, especially under reduced wheel loading, and to confirm other predictions of the car behavior.

11.5.4.1. PREDICTIONS AND ANALYSES

An analysis shall be carried out of the lateral and vertical wheel forces on a single car, with the car loaded asymmetrically, consistent with AAR loading rules, to give maximum wheel unloading.

The analysis shall be made for a speed equivalent to a mean unbalance at the car center of −3 inches to +3 inches with a coefficient of friction of 0.5 and modified Heumann wheel and new AREA 132 1b. or 136 1b. rail profiles.

The predicted lateral-to-vertical force ratio shall not exceed 1.0 or the sum of the absolute values of L/V on any axle shall not exceed 1.4, and no vertical wheel load shall be less than 10 percent of its static value, in a bunched spiral, with a change in superelevation of 1 inch in every 20 ft, leading into a curve of at least 7 degrees and a minimum of 3 inches superelevation.

11.5.4.2. TEST PROCEDURE AND CONDITIONS

This test may be carried out concurrently with the previous test, paragraph 11.5.3.2. The test car shall be operated, empty and fully loaded, between two heavy buffer cars, one of which may be an instrumentation car, at constant speeds equivalent to an unbalance of −3, 0, and +3 inches at the maximum curvature.

The wheels of the test car shall have less than 5000 miles wear on the new profiles specified for production, except that those on instrumented wheelsets shall have modified Heumann profiles. The rail profiles shall have a width at the top of the head not less than 95 percent of the original value when new.

The maximum curvature shall be not less than 7 degrees, with a minimum of 3 inches superelevation. A bunched spiral, with a change in superelevation of not less than 1 inch in every 20 ft., is required. The track shall be class 5 or better and dry. Tests shall be run in both directions and with the consist turned.

11.5.4.3. INSTRUMENTATION AND CRITERIA

The leading axle on both trucks on an end unit or car, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets.

The lateral and vertical forces and their ratio, L/V, shall be measured continuously through the bunched spiral, in both directions, and their maxima and minima computed. Measured force components having frequencies above 15 hertz are to be filtered out.

The maximum L/V ratio on any wheel shall not exceed 0.8, and the vertical wheel load shall not be less than 10 percent of the measured static value. A record of L/V's and vertical forces on both wheels of the two worst axles in a car, and car body roll angle, for each test, shall be submitted as required test data.

11.6. SINGLE CAR ON PERTURBED TRACK

11.6.1. GENERAL

The analyses and tests described in this section are designed to establish the trackworthiness of the car under conditions associated with variations in the track geometry. They include the dynamic response due to perturbations in the track but exclude the dynamic effects due to coupling with adjacent cars.

The investigations are designed to demonstrate that the car design provides an adequate margin of safety from structural damage and from any tendency to derail.

The tests shall be completed and their results found satisfactory by the AAR observers. The results identified shall be added as required data for the consideration of the Car Construction Committee.

11.6.2. RESPONSE TO VARYING CROSS-LEVEL (TWIST AND ROLL)

This requirement is designed to ensure the satisfactory negotiation of oscillatory cross-level excitation of cars, such as occurs on staggered jointed rail, which may lead to large car roll and twist amplitudes. The analyses and tests are required to show that the resulting forces between the wheel and rail show an adequate margin of safety from any tendency to derail.

11.6.2.1. PREDICTIONS AND ANALYSES

A review shall be made of any tests and analyses for the natural frequency and damping of the car body, in the roll and twist modes, in the empty and fully loaded conditions, and an estimate made of the speed of the car at each resonance.

The maximum amplitude of the carbody in roll and twist, the maximum instantaneous sum of the absolute values of the wheel L/V ratios on any axle, the minimum vertical wheel load, and the number of cycles to reach them, shall be predicted at resonant speed of 70 mph or below, on tangent track, with staggered jointed rails of 39 ft. length, and a maximum cross-level at the joints of 0.75 in. as shown in Fig. 11.1.

The instantaneous sum of the absolute values of the wheel L/V ratios on any axle shall be less than 1.3, the predicted roll angle of the carbody shall not exceed 6 degrees peak-to-peak, and the vertical wheel load shall not be less than 10 percent of its static value, within 10 rail lengths of the start, at any speed at or below 70 mph.

11.6.2.2. TEST PROCEDURE AND CONDITIONS

The test car shall be between two cars chosen for their stable performance. Tests shall be carried out with the test car empty and fully loaded.

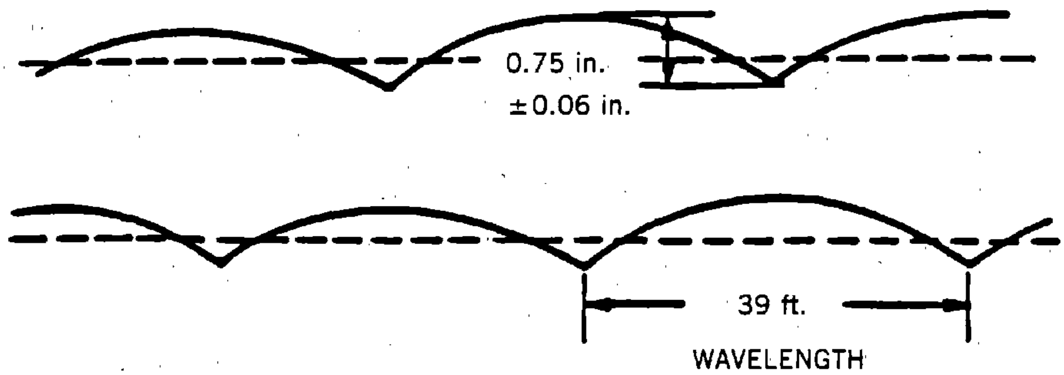

Figure 11.1.

TRACK CROSS LEVEL FOR THE TWIST AND ROLL TEST

The test shall be on tangent track with staggered 39 ft. rails on good ties and ballast, shimmed to a cross level of 0.75 in., low at each joint as shown in Fig. 11.1, over a test zone length of 400 ft., but otherwise held to class 5 or better.

The test shall be carried out at constant speed, increasing in 2 mph steps from well below any predicted resonance until it is passed, or approaching it from a speed above that expected to give a resonant condition. The test shall be stopped if an unsafe condition is encountered or if the maximum of 70 mph is reached. It shall be regarded as unsafe if a wheel lifts or if the car body roll angle exceeds 6 degrees, peak-to-peak.

11.6.2.3. INSTRUMENTATION AND CRITERIA

The leading axle of both trucks on an end unit or car, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets. The car body roll angle shall also be measured at a minimum of each end of an end unit.

The wheel forces, the mean roll angle and difference in roll between ends for each unit, shall be measured continuously through the test zone. Measured force components having frequencies above 15 hertz are to be filtered out.

The sum of the absolute values of wheel L/V on any instrumented axle shall not exceed 1.4, the roll angle of the carbody of any unit shall not exceed 6 degrees peak-to-peak and the vertical wheel load shall not be less than 10 percent of its static value at any speed tested.

A record of the vertical loads measured at the axle with the lowest measured vertical load, and the roll angles measured at each end of the most active unit of the car, taken at the resonant speeds for each car load, shall be submitted as required test data.

11.6.3. RESPONSE TO SURFACE VARIATION (PITCH AND BOUNCE)

This requirement is designed to ensure the satisfactory negotiation of the car over track which provides a continuous or transient excitation in pitch and bounce, and in particular the negotiation of grade crossings and bridges, where changes in vertical track stiffness may lead to sudden changes in the loaded track profile beyond those measured during inspection. The analyses and tests are required to show that the resulting forces between the wheel and rail show an adequate margin of safety from any tendency for the car to derail, to uncouple, or to show interference either between subsystems of the car or between the car components and track.

11.6.3.1. PREDICTIONS AND ANALYSES

A review shall be made of any tests and analyses for the natural frequency and damping of the car body, fully loaded, in the modes of pitch and bounce, and an estimate made of the resonant speed of the car when excited by a track wavelength of 39 feet.

The vertical wheel load shall be predicted at these speeds or at 70 mph, whichever is greater, for a continuous near sinusoidal excitation with a vertical amplitude to the track surface of 0.75 inches peak-to-peak and a single symmetric vertical bump in both rails, of the shape and amplitude shown in Fig. 11.2, predicted vertical wheel load shall not be less than 10 percent of its static value at any resonant speed at or below 70 mph, within 10 rail lengths of the start of the continuous sinusoid or following the single bump.

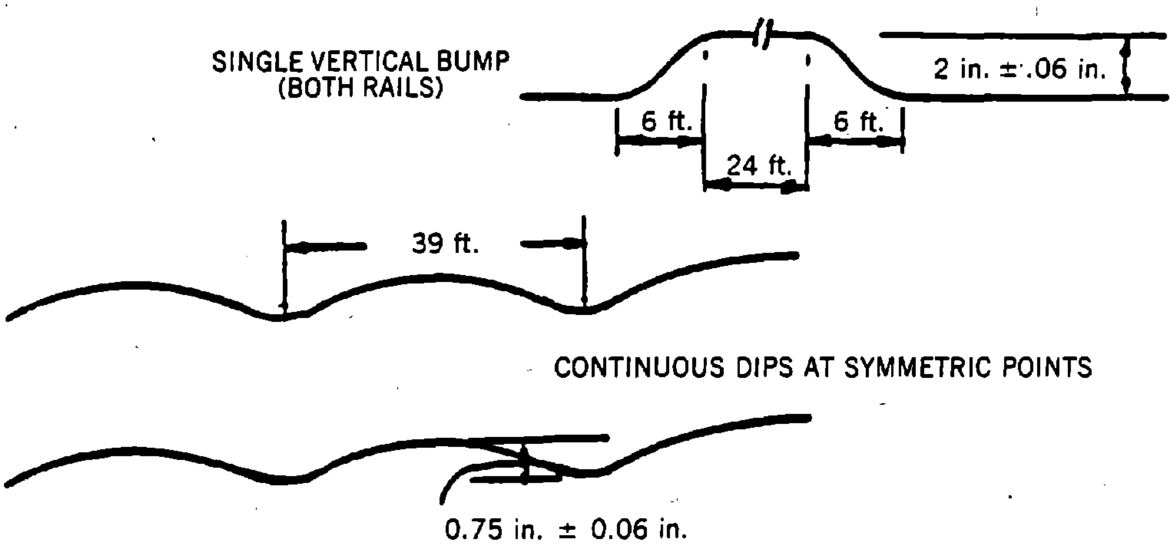

Figure 11.2.

TRACK SURFACE VARIATION FOR PITCH AND BOUNCE

11.6.3.2. TEST PROCEDURE AND CONDITIONS

The fully loaded test car shall be tested between two light cars that have at least 45 ft. truck center spacing.

Tests shall be carried out on tangent track with surface deviations providing a continuous, near sinusoidal, excitation with a vertical amplitude to the track surface of 0.75 inches peak-to-peak and a single symmetric vertical bump in both rails of the shape and amplitude shown in Fig. 11.2. These tests may be carried out separately, or together, with a separation of at least 100 feet. The track shall otherwise be held to class 5 or better.

Testing shall start at constant speed well below any predicted resonant speed, increasing in 5 mph steps until an unsafe condition is encountered, the resonance is passed, or the maximum of 70 mph is reached. The speed at which resonance is expected may be approached from a higher speed, using steps to decrease the speed. It shall be regarded as unsafe if any wheel lifts.

11.6.3.3. INSTRUMENTATION AND CRITERIA

The leading axle on both trucks on an end unit or car, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets. The vertical wheel forces shall be measured continuously through the test zone. Measured force components having frequencies above 15 hertz are to be filtered out.

The vertical wheel load shall not be less than 10 percent of its static value on any wheel at any speed tested. A record of the vertical loads measured on the axle with the lowest vertical load shall be submitted as required test data.

11.6.4. RESPONSE TO ALIGNMENT VARIATION ON TANGENT TRACK (YAW AND SWAY)

This requirement is designed to ensure the satisfactory negotiation of the car over track with misalignments which provide excitation in yaw and sway. The analyses and tests are required to show that the resulting forces between the wheel and rail show an adequate margin of safety from any tendency for the car forces to move the track or rail or to give interference either between subsystems of the car or between the car components and track.

11.6.4.1. PREDICTIONS AND ANALYSES

A review shall be made of the previous tests and analyses for the natural frequency and damping of the car body, fully loaded, in the yaw and roll modes. These may combine in a natural motion referred to as sway, which, if present, must be included in this analysis. Using the values for frequency and damping identified, an estimate shall be made of the resonant speed of the car, in each mode.

The car shall be assumed to be excited by a symmetric, sinusoidal track alignment deviation of wavelength 39 feet, on tangent track. The ratio of the sum of the lateral to that of the vertical forces on all wheels on one side of any truck shall be predicted at resonance or at 70 mph, whichever is greater, for a sinusoidal double amplitude of 1.25 inches peak-to-peak on both rails and a constant wide gage of 57.5 inches, as shown in Fig. 11.3.

The predicted truck side L/V shall not exceed 0.6, and the sum of the absolute values of L/V on any axle shall not exceed 1.4, at any speed at or below 70 mph, within 5 rail wavelengths of the start.

11.6.4.2. TEST PROCEDURE AND CONDITIONS

The fully loaded test car shall be placed at the end of the test consist, behind a buffer car of at least 45 feet truck center spacing, chosen for its stable performance.

Tests shall be carried out on dry tangent track, with symmetric, sinusoidal alignment deviations of wave length 39 feet, alignment amplitude 1.25 inches peak-to-peak and a constant wide gage of 57.5 inches, over a test zone of 200 feet as shown in Fig. 11.3. The track shall otherwise be held to class 5 or better.

The wheels of the test car shall have less than 5000 miles wear on the new profiles specified for production, except that those on instrumented wheelsets shall have modified Heumann profiles. The rail profiles shall have a width at the top of the head not less than 95 percent of the original value when new.

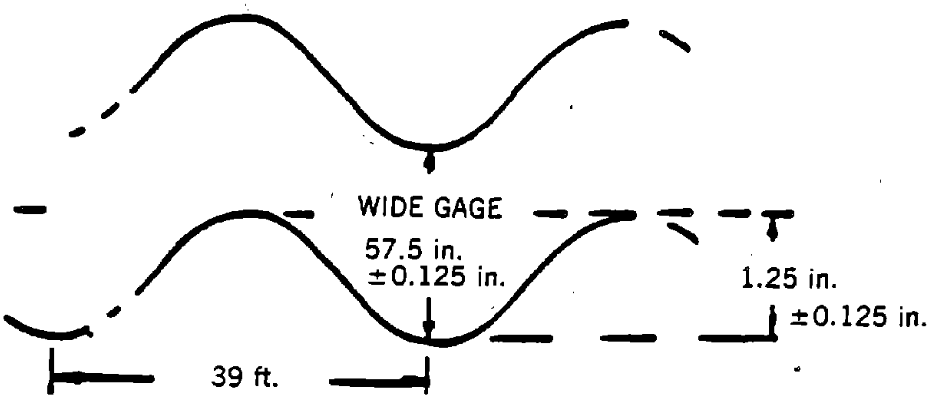

Figure 11.3.

TRACK ALIGNMENT VARIATIONS FOR YAW AND SWAY

Testing shall start at constant speed well below any predicted resonant speed, increasing in 5 mph steps until an unsafe condition is encountered, the resonance is passed, or the maximum of 70 mph is reached. It shall be regarded as unsafe if the ratio of total lateral to vertical forces, on any truck side measured, exceeds 0.6 for a duration equivalent to 6 feet of track.

11.6.4.3. INSTRUMENTATION AND CRITERIA

All axles on the truck estimated to provide the worst total truck side L/V, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets. The wheel forces shall be measured continuously through the test zone. Measured force components having frequencies above 15 hertz are to be filtered out.

The truck side L/V measured shall not exceed 0.6 for a duration equivalent to 6 feet of track, and the sum of the absolute values of L/V on any axle shall not exceed 1.4, at any speed at or below 70 mph. A record of the lateral and vertical loads, measured on the truck with the largest truck side L/V, shall be submitted as required test data.

11.6.5. ALIGNMENT, GAGE AND CROSS-LEVEL VARIATION IN CURVES (DYNAMIC CURVING)

This requirement is designed to ensure the satisfactory negotiation of the car over jointed track with a combination of misalignments at the outer rail joints and crosslevel due to low joints on staggered rails at low speed. The analyses and tests are required to show that the resulting forces between the wheel and rail show an adequate margin of safety from any tendency for the car forces to cause the wheel to climb the rail or to move the track or rail or to give unwanted interference, either between subsystems of the car, or between the car components and track.

11.6.5.1. PREDICTIONS AND ANALYSES

A review shall be made of the previous tests and analyses for the natural frequencies and response of the car body, fully loaded, in the yaw and roll modes.

No analysis is presently available, which can predict the results accurately for this test, for all possible designs. It is therefore necessary to provide additional safety features in the running of the test program to prevent unexpected derailments or unnecessary damage.*

*Analyses suitable for predictions of new car performance in this test are under development and will be added later.

11.6.5.2. TEST PROCEDURE AND CONDITIONS

The test car shall be operated between two cars that are loaded to provide them with a low center of gravity. If suitable, an instrumentation car may be used as one of these cars.

Tests shall be carried out on dry rail, in a curve of between 10 and 15 degrees with a balance speed of between 15 and 25 mph, with the test car empty and fully loaded.

The wheels of the test car shall have less than 5000 miles wear on the new profiles specified for production, except that those on instrumented wheelsets shall have modified Heumann profiles. The rail profiles shall have a width at the top of the head not less than 95 percent of the original value when new.

The track shall consist of staggered rails, 39 feet long, on good ties and ballast, shimmed to provide a cross level of 0.5 inch, low at each joint, over the test zone length of 200 feet, as shown in Figure 11.4.

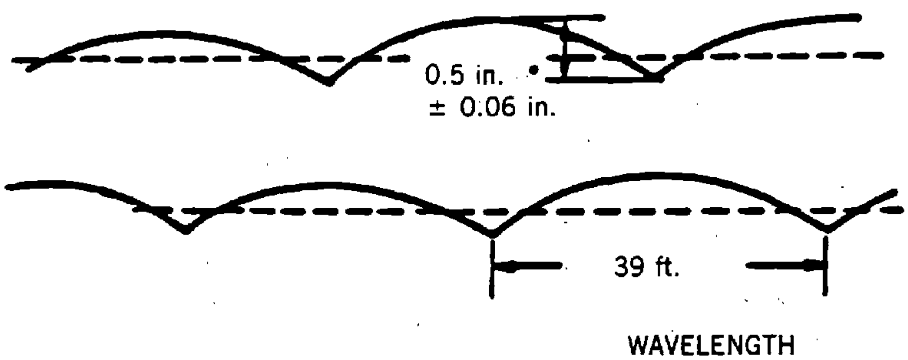

Figure 11.4.

CROSS LEVEL FOR DYNAMIC CURVING TESTS

Combined gage and alignment variation shall be provided in the test zone by shimming the outer rail in the form of an outward cusp, giving a maximum gage of 57.5 inches at each outer rail joint and a minimum gage of 56.5 inches at each inner rail joint, the inner rail being within class 5 standards for alignment in curves, as given in Figure 11.5.

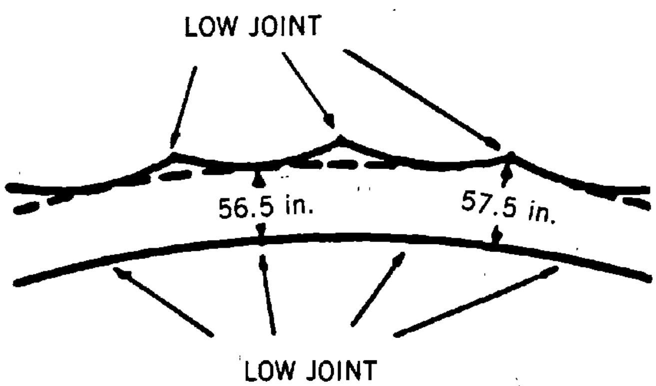

Figure 11.5.

GAGE AND ALIGNMENT VARIATION IN DYNAMIC CURVING

It is recommended that a guard rail be used to prevent unpredicted derailment; however, it must not be in contact with the wheel during normal test running. The test shall be carried out at constant speeds up to 3 inches of overbalance, increasing in 2 mph steps from well below any predicted lower center roll resonance until it is passed. The resonance may be approached from a speed above that predicted to give a lower center roll resonance.

The test shall be stopped if an unsafe condition is encountered or if the maximum unbalance is reached. It shall be regarded as unsafe if a wheel lifts, the instantaneous sum of the absolute L/V values of the individual wheels on any axle exceeds 1.4, or car body roll exceeds 6 degrees, peak-to-peak.

11.6.5.3. INSTRUMENTATION AND CRITERIA

The leading axle on both trucks on an end unit or car, or each axle on an end unit or car with single-axle trucks, shall be equipped with instrumented wheelsets. The car body roll angle shall also be measured at one end of the lead unit. The lateral and vertical wheel forces and the roll angle shall be measured continuously through the test zone. Measured force components having frequencies above 15 hertz are to be filtered out.

The maximum roll angle shall not exceed 6 degrees, peak-to-peak, the vertical wheel load shall not be less than 10 percent of its static value, the individual wheel L/V shall be less than 1.0 or the instantaneous sum of the absolute wheel L/Vs on any axle shall be less than 1.4, at any test speed.

A record of both wheel loads measured on the axle with the lowest measured vertical load and largest measured lateral load, and the roll angles measured, taken at the resonant speeds for each car load, shall be submitted as required test data.

11.7. COUPLED CARS AND UNITS

11.7.1. GENERAL

The tests described in this section will be designed to establish the track-worthiness of the car under conditions associated with the realistic operation of cars within a train. This may include severe transient forces due to coupling with adjacent cars. These forces may have a significant effect on the stability of cars and may lead to derailment. The investigations will be designed to demonstrate that the car design provides an adequate margin of safety from structural damage and from any tendency to derail.

11.7.2. VERTICALLY CURVED TRACK *

* This section to be added at a later date

11.7.3. HORIZONTALLY CURVED TRACK +

+ Investigations are currently underway which will allow the addition of this section in the near future.

APPENDIX A
VEHICLE CHARACTERIZATION
Adopted 1987

1.0. GENERAL

The characteristic properties of the car body and its suspension, required for analysis of its track-worthiness, must be supported by test results providing evidence of their validity. Forces and motions between suspension components and the body modal frequencies of the car, as assembled, can vary significantly from the values calculated or specified in the design, and may be important to the safe performance of the vehicle.

1.1. TEST CAR

It is important that characterizations be carried out on the particular car in the same condition that it is to be track tested so that accurate predictions of its performance can be made. For cars with more than one type of suspension, at least one of each type should be tested.

The tests apply to all new car suspensions, including trucks retrofitted with devices such as inter-axle connections, sideframe cross-bracing and additional suspension elements, which have not been tested previously.

Tests for horizontal characteristics of the suspension of trucks with at least two axles, may be carried out with the truck separated from the body. In this case static vertical loads must be applied to simulate those due to the body or bodies and the rotational and lateral characteristics between the truck and body must be measured separately.

Where connections exist between the truck and body that may affect the truck characteristics, such as with a truck steered through links to the body, and for all cars with single axle trucks, the suspension characteristics must be tested while connected to the body.

Where the truck is at the junction of two articulated bodies, both must be simulated or used in the suspension characterization tests specified.

1.2. TEST LOADS

Modal tests, and tests for the horizontal and vertical suspension characteristics are required with vertical loads equivalent to the car in the loaded condition required for the analyses in which the results will be used. This includes tests to measure the alignment of the axles to each other and to other elements in the system.

1.3. GENERAL PROCEDURE

In tests for the suspension characteristics, the recommended procedure is to load the suspension and to measure the load and displacement, or velocity, across the particular suspension element, in the required direction. These should be recorded up to the required maximum and down to the required minimum identified.

The loads may be applied, either through automatic cycling at an appropriate frequency or through manual increase and decrease of load through at least two complete cycles. If manual loading is used, delays and intermediate load reversals between measurements should be avoided. For the determination of stiffness and frictional energy dissipation, the frequency of cycling must be between 0.2 and 0.5 hertz.

Graphs of load versus displacement or velocity are desirable for the determination of the required stiffness or damping.

2.0. TESTS WITH THE WHEELS RESTRAINED

2.1. GENERAL

In the tests described in this section, the wheels are rigidly attached to the rails or supporting structure and the frame is moved relative to them.

The methods described are not suitable for trucks having steering links, which couple the lateral or roll motion of the body or truck frame to the yaw motion of the axles. In such a case, provision must be made for unrestrained longitudinal movement of the wheels, discussed in section 3. The steering links may be disconnected to measure the characteristics of suspension elements in the unsteered condition.

All tests require that the actuators and restraining links, other than those at the wheels, have the equivalent of ball joints at both ends to allow for motion perpendicular to their axis.

2.2. VERTICAL SUSPENSION STIFFNESS

For this test, equal measured vertical loads are applied across the spring groups in the range from zero to 1.5 times the static load, if possible, and at least to the static load of the fully loaded car. Vertical actuators are attached to each side of the body or the structure simulating it. The load may also be applied by adding dead load or a combination of both dead and actuator loads.

Vertical deflections are required across all significant spring elements under load. It is important to report any differences in the measurements taken between each axle and frame or sideframe.

2.3. TOTAL ROLL STIFFNESS

A roll test is required if the roll characteristic between the body and axle includes movement at or forces due to elements other than the vertical suspension, such as clearances at sidebearings, or anti-roll bars.

For the roll test, two vertical actuators are required as in the vertical test, but with the loads in the actuators in opposite directions. The range of roll moments, in inch-pounds, applied to the truck should be between plus and minus 30 times its static load, in pounds, or until the wheels lift. The roll angle across all suspension elements may be measured directly or deduced from displacements.

2.4. TOTAL LATERAL STIFFNESS

The lateral stiffness characteristic may be found by attaching an actuator to apply loads laterally to the body or bodies, which should be positioned as if on tangent track. If the lateral motion of the truck frame is coupled to its yaw through a steering mechanism, it should be disconnected to prevent the yaw resistance of the frame from affecting the measurement of lateral stiffnesses.

The minimum and maximum lateral loads applied per truck should be minus and plus one fifth of the static load carried. Measurements are required of the lateral displacements across all suspension elements.

2.5. INTER-AXLE TWIST AND EQUALIZATION

This test is carried out with only one axle fixed to the track. One wheel of the other axle in the car or truck is jacked up to a height of 3 inches, and the vertical load and displacement are measured. The stiffness between the axles in twist is the ratio of the load to the displacement multiplied by the square of the gage. It is a measure of the truck equalization.

3.0. TESTS WITH UNRESTRAINED WHEELS

3.1. GENERAL

These tests involve movements in the suspension system and axles relative to other elements of the system or to other axles, without restraint between the wheel and rail, but with the normal static vertical load.

The shear resistance between the rail and the wheel must be eliminated by the provision of a device having very low resistance, such as an air bearing, under each axle.

3.2. AXLE ALIGNMENT

Both radial and lateral misalignments may be deduced from measurements of the yaw angle of each axle from a common datum. The radial misalignment between axles is half the difference in their yaw angles, taken in the same sense, and the lateral misalignment is their mean yaw angle.

In the case of trucks which have significant clearance between the axle and frame, it may be necessary to establish the axle in the center of the clearance for the purpose of identifying the mean axle misalignments.

3.3. LONGITUDINAL STIFFNESS

A longitudinal load must be applied to the axle, equivalent to a single load at its center, and cycled between tension and compression up to half the static load on the axle.

The load may be applied directly between axles, or between the test axle and ground through an appropriate structure, with the body or truck frame restrained. The load may also be applied directly between the axle and frame, or in the case of a car with single axle trucks, between the axle and the body.

The longitudinal deflection across each spring element must be measured and the results plotted.

Where the load is applied directly between the axles of a truck or car, this measurement may be combined with the inter-axle shear test in section 3.4., or the inter-axle bending stiffness test in section 3.5.

3.4. AXLE LATERAL AND INTER-AXLE SHEAR STIFFNESS

The inter-axle shear stiffness may be found by shearing the axles, or moving them in opposite directions along their axes, and measuring the shear or lateral deflection between them. The shear force on each axle must be at least one tenth of the static vertical axle load.

This test may be combined with the inter-axle longitudinal test of section 3.3., where the required load can be achieved.

In the case of direct inter-axle loading, the locations of the applied force and restraint are such that they are equal and opposite, diagonally across the truck or car.

The actuator and restraint each provide two components of force on the axle to which they are attached. One component lies along the direction of the track and provides tension and compression, as in section 3.3., for the longitudinal stiffness. The other component lies along the axle and applies the required shear force bewteen axles. This component may be applied separately with a suitable arrangement of actuators and restraints.

Measurements are made of the lateral misalignment of the axles during the load cycle. The shear stiffness is the ratio of shear force to the lateral misalignment.

For single axle trucks, a test similar to that described above may be used to determine the lateral stiffness, with force applied laterally between ground and the axle with the body restrained, or with the truck frame restrained in the case of trucks having more than one axle. For trucks which also provide steering through coupling axle lateral motion to its yaw angle, this test may be preferred over the lateral test of section 2.4. for finding the lateral stiffness, since the axles are free to yaw.

3.5. AXLE YAW AND INTER-AXLE BENDING STIFFNESS

The inter-axle bending stiffness may be found by yawing the axles in the opposite directions and measuring the yaw angle between them. The yaw moment applied, in inch-pounds, must be at least equal to the axle load in pounds.

This test may be combined with the inter-axle longitudinal test of section 3.3. If this is done, the test is carried out by applying an effective force on the axle a known distance laterally from the truck centerline.

In the case of direct inter-axle loading the restraint must be applied to the axle, at the other end of the car or truck, on the same side as the applied force. The applied and restraining forces each provide a longitudinal force and a yaw moment on the axle to which they are attached. The force provides the tension and compression as in section 3.3. for the longitudinal stiffness and the moment is applied between the truck axles in yaw. This moment may be applied independently of the longitudinal force.

Measurements are made of the resulting radial mis-alignment of the axles during the load cycle. The bending stiffness is the ratio of applied bending moment to the radial misalignment.

A similar test of the axle yaw stiffness may be arranged with forces applied in yaw between a single axle and ground, with the body restrained, or with the truck frame restrained in the case of trucks having more than one axle.

3.6. YAW MOMENT BETWEEN THE SUSPENSION AND BODY

The required yaw stiffness and breakout torque between the car body and truck must be measured by applying a yaw moment, using actuators in equal and opposite directions at diagonally opposite corners of the truck to rotate the truck in yaw. The car body must be restrained.

The applied yaw moment must be increased until gross rotation is observed, representing the breakout torque, or to the limit recommended for the yaw of the secondary suspension.

The angle in yaw between the car body and truck bolster or frame must be measured.

4.0. RIGID AND FLEXIBLE BODY MODAL CHARACTERISTICS

4.1. GENERAL

Tests are required to identify the rigid and flexible body modal frequencies and damping. The rigid body modal frequencies may be compared to predictions using estimated or measured body masses, and inertias and the suspension parameters measured according to the requirements of sections 2. and 3. Tests and estimates should be made with the car in the empty and fully loaded state.

4.2. TEST CAR BODY

For cars consisting of more than one coupled unit, tests for body modes are required on one of each of the unit bodies having a different structural design. Dead loads may be added to give the required additional loading to any shared suspensions.

Where coupling exists between the modes of adjacent bodies, such as in roll or torsion, this may be examined in a dynamic analysis, validated for the case of tests without coupling.

The frequency and modal damping are only required for the flexible body modes which are predicted to have a natural frequency below 12 hertz.

4.3. GENERAL PROCEDURE

Transient or continuous excitation may be applied, using one or more actuators or dropping the car in a manner to suit the required mode of excitation.

The modal frequency and damping are required for an amplitude typical of the car running on class 2 track.

In the case of the rigid body modes, the actuators must be located at the rail level or the level of the truck frame with the body free to oscillate on its suspension. In the case of the flexible body modes, the excitation may be applied directly to the body.

The frequency in hertz may be determined from the wavelength in the transient test, or from the peak response, or from the 90 degree phase shift between the response and excitation where continuous excitation is used.

The percentage modal damping may be determined using the logarithmic decrement in transient tests or the bandwidth of the response from a range of frequencies.

4.4. RIGID BODY MODES

The rigid body modes for the car are:

Body bounce
Body pitch
Body yaw and sway
Lower center roll
Upper center roll

In the case where the normal load on the body is not centered between the suspensions, the body bounce mode may be coupled to the body pitch. The required measurement of bounce and pitch may be achieved by two vertical measurements at the ends of the car. Their weighted sum provides bounce and their weighted difference pitch. The weighting is dependent on their position relative to the center of mass.

Yaw and sway are deduced from lateral measurements made at each end of the body, a known distance from its mass center, similarly to the determination of pitch.

Measurement of the upper and lower center roll modes are determined from lateral displacements taken at two heights, or by a single lateral displacement and a roll angle measurement.

4.5. FLEXIBLE BODY MODES

The flexible body modes for the car are:

Torsion
Vertical bending
Lateral bending

Determination of the frequency and damping in the torsion mode requires excitation and measurement of roll at one end of the car.

The excitation is similar to that for roll but resonance occurs at a higher frequency. The response between the ends of the car is out of phase for modes number 1,3, and in phase for modes number 2,4, although it is unlikely that modes above 2 will be significant.

Vertical or lateral bending modes are measured as a response to the vertical or lateral excitation at one end or both ends of the car. The first bending mode has a maximum amplitude at or near the car center. The second bending mode has a node or point of minimum response at the center.

5.0. PARAMETER ESTIMATION*

* Tests are presently being conducted to examine this method.

APPENDIX B

SPECIFICATION FOR INSTRUMENTED WHEELSETS FOR CHAPTER XI (M-1001) TESTING

Adopted 1989

1.0. INTRODUCTION

Instrumented wheelsets to be used in acceptance testing of new and untried cars under Chapter XI of AAR Standard M-1001 must meet the requirements of this specification. Load measuring wheelsets are a critical transducer for a wide range of the Chapter XI vehicle dynamics tests. Calibrated wheelsets will be required to accurately measure lateral and vertical wheel/rail forces, as well as wheel lateral to vertical force (L/V) ratios. A verification of wheelset accuracy is performed through a three-step process consisting of calibration, analysis, and field procedures.

2.0. INSTRUMENTED WHEELSET SPECIFICATIONS

To be accepted for Chapter XI testing, a load measuring wheelset design must meet the following specifications:

2.1.

Vertical wheel load measurements must be within +/− 5 percent of the actual vertical load. This accuracy is to be maintained for loads ranging from 0 to 200 percent of the static wheel load. The minimum signal resolution is to be no less than 0.5 percent of the static wheel load.

2.2.

Lateral wheel load measurements must be within +/− 10 percent of the actual lateral load. This accuracy is to be maintained for loads ranging from 0 to 100 percent of the static (vertical) wheel load. The minimum signal resolution is to be no less than 0.5 percent of the static (vertical) wheel load.

2.3.

Maintain the above stated accuracy requirements, at all times, for:

2.3.1.

All potential load cases (longitudinal loads of up to 60 percent of the static (vertical) wheel load, lateral loads of up to 100 percent of the static (vertical) wheel load, and vertical loads of up to 200 percent of the static wheel load).

2.3.2.

All potential wheel/rail contact conditions including full flange contact, outside tread contact, two-point contact, and flange contact at high wheelset angles of attack.

2.3.3.

An operating speed (for dynamic wheelset output) of from 5 to 80 mph.

2.3.4.

Signals from 0 to 30 Hertz.

2.3.5.

Over a recommended operating ambient temperature range of 0 to 110 degrees Fahrenheit. Any restrictions in the operating temperature range are to be noted.

2.4.

Wheelset reprofiling or recalibration requirements due to profile wear are to be documented. Temperature compensation arrangements and operating limitations due to ambient temperature swings are to be detailed as well. The wheelsets are to be equipped with the modified Heumann profile shown in Figure 8.1 of Chapter VIII of AAR Standard M-1001.

3.0. VERIFICATION

Wheelset accuracy is to be substantiated through calibration, analysis, and testing. A minimum number of required wheelset static tests to calibrate and verify wheelset output are described. Since dynamic calibration of load measuring wheelsets has proven difficult, further verification of wheelset accuracy relies on required static and dynamic analyses. A limited set of simple experimental procedures are then prescribed to confirm proper wheelset function under field conditions.

3.1. STATIC CALIBRATION

Static tests to determine the wheelset calibration factors are required of all instrumented wheelsets. Documentation in support of the calibration tests is to include a complete description of the calibration stand and the calibration procedure. Calibration for vertical and lateral loads is to include testing for a minimum of six wheel rotational positions (0, 60, 120, 180, 240, and 300 degrees). Calibration for vertical loads is to include testing for a minimum of three contact point lateral positions (on tape line and one inch), respectively, to the flange and wheel face of the tape line. Each calibration sequence is to be repeated at least once to verify measurement repeatability.

The static calibration tests are as follows:

3.1.1.

Using an appropriate loading scheme, vertical loads ranging from 0 to 200 percent of the static wheel load are to be applied with a minimum of 5 equally spaced inputs (0, 50, 100, 150, and 200 percent of the static wheel load). Strain gauge output for both vertical and lateral force circuits is to be recorded.

3.1.2.

Using an appropriate loading scheme, lateral wheel loads are to be applied at the wheel tread ranging from −100 to 100 percent of the static wheel load with a minimum of 10 equally spaced inputs (+/− 20, 40, 60, 80, and 100 percent). A vertical force equivalent to the static wheel load is to be applied simultaneously. Both vertical and lateral force strain gauge outputs are to be recorded.

The static calibration report is to include raw measurement values and the derived calibration factors. The calibration report must also include a table comparing the applied forces and, given the calibration factors obtained during the testing, the measured forces. It is assumed here that the calibration factors will represent average values independent, for example, of wheelset rotational position.

3.2. ANALYSIS

The following theoretical analyses are required to verify theoretical wheelset accuracy for load combinations that cannot satisfactorily be applied using a conventional static loading frame. It is assumed that finite element or similar calculations will have been performed beforehand to obtain the theoretical wheelset calibration factors. Any variations in wheelset output or accuracy due to rotational position are to be described.

Static finite element or similar calculations to verify theoretical wheelset accuracy for the following scenarios:

3.2.1.

Single point contact at one inch toward the wheel face from the wheel tape line for a vertical load of 50 and 200 percent of the static wheel load in combination with a lateral load of −25 and 25 percent of the static wheel load (giving a total of four load combinations).

3.2.2.

Single point contact on the flange (defined as being at a point giving a rolling radius one-half inch greater than that obtained at the tape line) for a vertical load of 100 and 150 percent of the static wheel load in combination with a lateral load of 25, 50, and 75 percent of the static wheel load (giving a total of six load combinations).

3.2.3.

Single point contact at the wheel tape line for a vertical load equal to the static wheel load in combination with a longitudinal load of −50, −25, 25, and 50 percent of the static wheel load and a lateral load of 10 percent of the static wheel load (for a total of four load combinations). Note that a negative longitudinal load is defined here as a load directed in the sense of the wheel rotation.

3.2.4.

Single point contact at the flange for a vertical load of 75 percent of the static wheel load in combination with a longitudinal load of −50, −25, 25, and 50 percent of the static wheel load and a lateral load of 50 percent of the static wheel load (for a total of four load combinations).

3.2.5.

Two-point contact with the first point of contact at one-half inch toward the wheel face from the wheel tape line and the second point of contact at the flange and displaced −0.5, 0, and 0.5 inches longitudinally from the mid-plane axis of the wheelset. The loading at the tread contact is to be a vertical load of 50 percent of the static wheel load in combination with a longitudinal load of −25 percent and a lateral load of −10 percent of the static wheel load. The loading at the flange contact is to be a vertical load of 75 percent of the static wheel load in combination with a longitudinal load of 50 percent and a lateral load of 50 percent of the static wheel load (for a total of three calculation cases).

3.2.6.

Single point contact at the tape line for a wheel with a radius one-quarter inch less than nominal and a vertical load equal to the static wheel load in combination with a lateral load of 10 percent of the static wheel load.

3.2.7.

Single point contact at the flange for a wheel with a radius one-quarter inch less than nominal and a vertical load equal to 75 percent of the static wheel load in combination with a lateral load of 50 percent of the static wheel load.

Results for the twenty-three static calculation cases described above are to be given as the percent deviation of the predicted lateral and vertical force values from the applied values.

A single dynamic finite element or similar calculation to verify theoretical wheelset accuracy under dynamic conditions:

3.2.8.

This calculation is to verify that no wheelset vibration modes are present with natural frequencies below 30 Hertz. If such modes exist, a dynamic calculation is to be performed for the following wheelset input: single point contact at the wheel tape line for a vertical load equal to the static wheel load in combination with a time varying longitudinal load with an amplitude of 25 percent and a lateral load with an amplitude of 10 percent of the static wheel load. The mean longitudinal and lateral force are both to be zero. The calculation is to consider an input frequency ranging from 0 to 30 Hertz where the lateral and longitudinal force signals are 90 degrees out of phase. The boundary condition to be used for both this calculation and the wheelset natural frequency calculation is to fix the wheelset in the longitudinal, lateral, vertical, and rotational sense at the bearing centerline (axle top dead center).

The results of the dynamic calculation are to be given as the mean value and amplitude of the predicted lateral and vertical forces as functions of the wheelset rotational position.

3.3. TEST PROCEDURES

The following experimental analyses are required:

3.3.1.

A zero speed jacking test to set the wheelset zero followed by a slow speed roll (at ten, twenty, and thirty miles per hour) along tangent track to verify that wheel vertical load signals are within +/− 5 percent of the calibrated scale axle load for constant speed operation on level tangent track. Wheelset signals will be evaluated on the basis of mean values for a randomly chosen output segment having a minimum duration of ten seconds.

3.3.2.

A steady-state curving test to confirm that net truck or car lateral loads are within +/− 10 percent of the theoretical value for constant speed operation on constant radius track at speeds corresponding to +3, 0, and −3 inches cant deficiency. Both curvature and superelevation of the track need to be constant and accurate. Wheelset accuracy is to be verified on a sharp curve (7 degrees curvature and above) for curving with hard flange contact. Wheelset signals will be evaluated on the basis of mean values for a randomly chosen output segment having a minimum duration of ten seconds.

3.3.3.

As an alternative to this test a zero speed jacking test is suggested using equal and opposing lateral loads applied (via a hydraulic jack) to the wheel backs. Measured lateral loads are to be within +/− 5 percent of the applied value for loads ranging from 0 to 50 percent of the static (vertical) wheel load.

APPENDIX B

NUCARS SYSTEM INPUT FILES
for Pre-Test and Post Test Analyses

Table 1. Pre-Test NUCARS System File for Empty PSMX 111 Car

```
-SYSTEM FILE (.SYS) for the program NUCARS Version 1.0
 N.B. Parameters are in lb., in. & sec. unless otherwise stated.
-Enter a title up to 80 characters long between the lines,
-----------------------------------------------------------------------------
 Empty Lightweight Car # 2   10/18/89
-----------------------------------------------------------------------------
-FOR THE BODIES
-Provide the number of heavy bodies including axles (IMM), and the number
  of input or light bodies (IBIN, used for input degrees of freedom )
     IMM    IBIN
     11      8
-List the number, name, in single quotes up to 15 characters long, and
  position of each body, (and axle body), relative to a datum on the system
  center, in inches, followed by the number of degrees of freedom required,
  followed by a list of the degrees of freedom for each, in turn,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, 7=epsx, 8=epsy, 9=epsz.
  The 4 degrees of freedom required for each axle are  2 3 4 6
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & list of DoF's
   1      'Carbody        '  -281.0   0.0   57.2    8   2 3 4 5 6 7 8 9
   2      'Lead Bolster   '   -35.0   0.0   18.0    4   2 3 4 6
   3      'Trail Bolster  '  -521.0   0.0   18.0    4   2 3 4 6
   4      'Ld Lt Sideframe'   -35.0  39.5   18.0    5   1 2 3 5 6
   5      'Ld Rt Sideframe'   -35.0 -39.5   18.0    5   1 2 3 5 6
   6      'Tl Lt Sideframe'  -521.0  39.5   18.0    5   1 2 3 5 6
   7      'Tl Rt Sideframe'  -521.0 -39.5   18.0    5   1 2 3 5 6
   8      'Axle 1         '     0.0   0.0   18.0    4   2 3 4 6
   9      'Axle 2         '   -70.0   0.0   18.0    4   2 3 4 6
  10      'Axle 3         '  -486.0   0.0   18.0    4   2 3 4 6
  11      'Axle 4         '  -556.0   0.0   18.0    4   2 3 4 6
  continue the body list with the number and position of each input body,
  relative to the same datum, in inches, followed by the number of input
  degrees of freedom required, followed by a list of the degrees of freedom,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, the number of the input history
  for each degree of freedom, in turn, followed by a choice of input phase
  lag for the input to this body, 0 = no, 1 = yes.
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & DoF list  Input list  Lag
  12      'Axle 1 Lt Wheel'     0.0  29.75   0.0   2    2 3      1  3       1
  13      'Axle 1 Rt Wheel'     0.0 -29.75   0.0   2    2 3      2  4       1
  14      'Axle 2 Lt Wheel'   -70.0  29.75   0.0   2    2 3      1  3       1
  15      'Axle 2 Rt Wheel'   -70.0 -29.75   0.0   2    2 3      2  4       1
  16      'Axle 3 Lt Wheel'  -486.0  29.75   0.0   2    2 3      1  3       1
  17      'Axle 3 Rt Wheel'  -486.0 -29.75   0.0   2    2 3      2  4       1
  18      'Axle 4 Lt Wheel'  -556.0  29.75   0.0   2    2 3      1  3       1
  19      'Axle 4 Rt Wheel'  -556.0 -29.75   0.0   2    2 3      2  4       1
-For all heavy bodies with flexible modes, give the position of each body
  geometric center, in the X direction from the datum, backward is -ve, its
  length in inches, the natural frequencies, in Hz., and the damping ratios
  in twist, vertical & lateral bending, as required.
  Body #     X-Posn    X-Length    Nat Frequencies(Hz.)       Damping Ratios
    1        -278.0     606.0        9.4  26.2   9.9         0.1   0.1    0.2
-List the mass, roll, pitch and yaw inertias, in order,
  for each heavy body, including axles,
  54.25     2.12e5    1.754e6   1.791e6
   4.77     3.48e3    0.0       3.48e3
   4.77     3.48e3    0.0       3.48e3
   3.0      0.0       1.37e3    1.37e3
   3.0      0.0       1.37e3    1.37e3
   3.0      0.0       1.37e3    1.37e3
   3.0      0.0       1.37e3    1.37e3
   7.09     5.41e3    1.38e3    5.41e3
   7.09     5.41e3    1.38e3    5.41e3
   7.09     5.41e3    1.38e3    5.41e3
   7.09     5.41e3    1.38e3    5.41e3
```

-FOR THE CONNECTIONS (including suspensions) Identify the following parameters,

-Number of connections:

IALLC
68

-Complete the following tables for each connection, identifying:
a name, in single quotes up to 20 characters long;
its position relative to the chosen datum in x, y, z inches;
the number of the body at each end, 0 for an earth in local track coords.;
a number indicating the degree(s) of freedom, translational 1,2,3 or
rotational 4,5,6; in x,y,z resp., including 2 for lateral wheel motion;
the type 1 - parallel pair of spring and damper characteristics
2 - series pair of spring and damper characteristics
3 - device with hysteresis between 2 PWL characteristics,
e.g. carriage spring or load sensitive suspension
4 - lateral/longitudinal suspension of the wheel on rail
in the track plane
5 - connection force as a history of the distance moved
and the identification number for each of type 1, 2 and 3;
axle number for type 4; input function number for type 5.

Note - single characteristics are treated as parallel pairs with the
missing characteristic set to zero in the subsequent table.

-Complete for all connections in turn,

Conn #	' 20 CHARACTER NAME '	Posn in X,	Y	& Z	Body1	Body2	DoF.	Type	Number
1	'Ld Bols-Bod Lt CB Vt'	-35.0	8.0	27.0	1	2	3	1	1
2	'Ld Bols-Bod Rt CB Vt'	-35.0	-8.0	27.0	1	2	3	1	1
3	'Tr Bols-Bod Lt CB Vt'	-521.0	8.0	27.0	1	3	3	1	1
4	'Tr Bols-Bod Rt CB Vt'	-521.0	-8.0	27.0	1	3	3	1	1
5	'Ld Bols-Bod Lt SB Vt'	-35.0	25.0	27.0	1	2	3	1	2
6	'Ld Bols-Bod Rt SB Vt'	-35.0	-25.0	27.0	1	2	3	1	2
7	'Tr Bols-Bod Lt SB Vt'	-521.0	25.0	27.0	1	3	3	1	2
8	'Tr Bols-Bod Rt SB Vt'	-521.0	-25.0	27.0	1	3	3	1	2
9	'Lead Bols-Bod CB Lat'	-35.0	0.0	27.0	1	2	2	1	3
10	'Trail Bols-Bod CB Lt'	-521.0	0.0	27.0	1	3	2	1	3
11	'Lead Bols-Bod CB Yaw'	-35.0	0.0	27.0	1	2	6	1	4
12	'Trl Bols-Bod CB Yaw '	-521.0	0.0	27.0	1	3	6	1	4
13	'Ld Bols-Sdfm Lt Vert'	-35.0	39.5	18.0	2	4	3	3	1
14	'Ld Bols-Sdfm Rt Vert'	-35.0	-39.5	18.0	2	5	3	3	1
15	'Tr Bols-Sdfm Lt Vert'	-521.0	39.5	18.0	3	6	3	3	1
16	'Tr Bols-Sdfm Rt Vert'	-521.0	-39.5	18.0	3	7	3	3	1
17	'Ld Bols-Sdfm Lt Lat '	-35.0	39.5	18.0	2	4	2	1	5
18	'Ld Bols-Sdfm Rt Lat '	-35.0	-39.5	18.0	2	5	2	1	5
19	'Tr Bols-Sdfm Lt Lat '	-521.0	39.5	18.0	3	6	2	1	5
20	'Tr Bols-Sdfm Rt Lat '	-521.0	-39.5	18.0	3	7	2	1	5
21	'Ld Bols-Sdfm Lt Yaw '	-35.0	39.5	18.0	2	4	6	1	6
22	'Ld Bols-Sdfm Rt Yaw '	-35.0	-39.5	18.0	2	5	6	1	6
23	'Tr Bols-Sdfm Lt Yaw '	-521.0	39.5	18.0	3	6	6	1	6
24	'Tr Bols-Sdfm Rt Yaw '	-521.0	-39.5	18.0	3	7	6	1	6
25	'Ld Bols-Sdfm Lt Long'	-35.0	39.5	18.0	2	4	1	1	7
26	'Ld Bols-Sdfm Rt Long'	-35.0	-39.5	18.0	2	5	1	1	7
27	'Tr Bols-Sdfm Lt Long'	-521.0	39.5	18.0	3	6	1	1	7
28	'Tr Bols-Sdfm Rt Long'	-521.0	-39.5	18.0	3	7	1	1	7
29	'Ax 1 Lt BA-Sdfm Long'	0.0	39.5	21.0	4	8	1	1	9
30	'Ax 1 Rt BA-Sdfm Long'	0.0	-39.5	21.0	5	8	1	1	9
31	'Ax 2 Lt BA-Sdfm Long'	-70.0	39.5	21.0	4	9	1	1	9
32	'Ax 2 Rt BA-Sdfm Long'	-70.0	-39.5	21.0	5	9	1	1	9
33	'Ax 3 Lt BA-Sdfm Long'	-486.0	39.5	21.0	6	10	1	1	9
34	'Ax 3 Rt BA-Sdfm Long'	-486.0	-39.5	21.0	7	10	1	1	9
35	'Ax 4 Lt BA-Sdfm Long'	-556.0	39.5	21.0	6	11	1	1	9
36	'Ax 4 Rt BA-Sdfm Long'	-556.0	-39.5	21.0	7	11	1	1	9
37	'Ax 1 Lt BA-Sdfm Lat '	0.0	39.5	21.0	4	8	2	1	9
38	'Ax 1 Rt BA-Sdfm Lat '	0.0	-39.5	21.0	5	8	2	1	9
39	'Ax 2 Lt BA-Sdfm Lat '	-70.0	39.5	21.0	4	9	2	1	9
40	'Ax 2 Rt BA-Sdfm Lat '	-70.0	-39.5	21.0	5	9	2	1	9
41	'Ax 3 Lt BA-Sdfm Lat '	-486.0	39.5	21.0	6	10	2	1	9
42	'Ax 3 Rt BA-Sdfm Lat '	-486.0	-39.5	21.0	7	10	2	1	9
43	'Ax 4 Lt BA-Sdfm Lat '	-556.0	39.5	21.0	6	11	2	1	9
44	'Ax 4 Rt BA-Sdfm Lat '	-556.0	-39.5	21.0	7	11	2	1	9
45	'Ax 1 Lt BA-Sdfm Vert'	0.0	39.5	21.0	4	8	3	1	1

46	'Ax 1 Rt BA-Sdfm Vert'	0.0	-39.5	21.0	5	8	3	1	1
47	'Ax 2 Lt BA-Sdfm Vert'	-70.0	39.5	21.0	4	9	3	1	1
48	'Ax 2 Rt BA-Sdfm Vert'	-70.0	-39.5	21.0	5	9	3	1	1
49	'Ax 3 Lt BA-Sdfm Vert'	-486.0	39.5	21.0	6	10	3	1	1
50	'Ax 3 Rt BA-Sdfm Vert'	-486.0	-39.5	21.0	7	10	3	1	1
51	'Ax 4 Lt BA-Sdfm Vert'	-556.0	39.5	21.0	6	11	3	1	1
52	'Ax 4 Rt BA-Sdfm Vert'	-556.0	-39.5	21.0	7	11	3	1	1
53	'Ax 1 Lt Whl/Rail Vrt'	0.0	29.75	0.0	8	12	3	1	8
54	'Ax 1 Rt Whl/Rail Vrt'	0.0	-29.75	0.0	8	13	3	1	8
55	'Ax 2 Lt Whl/Rail Vrt'	-70.0	29.75	0.0	9	14	3	1	8
56	'Ax 2 Rt Whl/Rail Vrt'	-70.0	-29.75	0.0	9	15	3	1	8
57	'Ax 3 Lt Whl/Rail Vrt'	-486.0	29.75	0.0	10	16	3	1	8
58	'Ax 3 Rt Whl/Rail Vrt'	-486.0	-29.75	0.0	10	17	3	1	8
59	'Ax 4 Lt Whl/Rail Vrt'	-556.0	29.75	0.0	11	18	3	1	8
60	'Ax 4 Rt Whl/Rail Vrt'	-556.0	-29.75	0.0	11	19	3	1	8
61	'Ax 1 Lt Whl/Rail lat'	0.0	29.75	0.0	8	12	2	4	1
62	'Ax 1 Rt Whl/Rail Lat'	0.0	-29.75	0.0	8	13	2	4	1
63	'Ax 2 Lt Whl/Rail Lat'	-70.0	29.75	0.0	9	14	2	4	2
64	'Ax 2 Rt Whl/Rail Lat'	-70.0	-29.75	0.0	9	15	2	4	2
65	'Ax 3 Lt Whl/Rail Lat'	-486.0	29.75	0.0	10	16	2	4	3
66	'Ax 3 Rt Whl/Rail Lat'	-486.0	-29.75	0.0	10	17	2	4	3
67	'Ax 4 Lt Whl/Rail Lat'	-556.0	29.75	0.0	11	18	2	4	4
68	'Ax 4 Rt Whl/Rail Lat'	-556.0	-29.75	0.0	11	19	2	4	4

-List for each pair of type 1 - parallel connections, its number, followed by the identification numbers of the piecewise linear characteristics for the stiffness and damping respectively, zero if absent, and the combined force or moment limit in extn & compn, lb or lb-in., 0.0 in extension at the vertical rail/wheel conn. allows valid wheel lift. (If no limit exists, set the F-values outside the expected range.)

Pair #	Stiff PWL	Damp PWL	F-extn.	F-compn.
1	1	2	0.0e8	-1.0e8
2	3	4	0.0e8	-1.0e8
3	5	6	1.0e8	-1.0e8
4	0	7	1.0e8	-1.0e8
5	8	9	1.0e8	-1.0e8
6	10	11	1.0e8	-1.0e8
7	12	13	1.0e8	-1.0e8
8	14	15	0.0e8	-1.0e8
9	18	19	1.0e8	-1.0e8

-List for each pair of type 2 - series connections, its number, followed by the identification numbers of the piecewise linear characteristics for the stiffness and damping respectively, and the stroke limit in extension & compression for the pair, in or rad, and the stiffness of the stop at the limit in lb/in or lb-in/rad. (If no limit exists, set the S-values outside the expected range.)

Pair #	Stiff PWL	Damp PWL	S-extn.	S-compn.	Stop K

-List the type 3 - hysteresis loop characteristics, giving to each a number, identification numbers for the extension and compression PWLs, a linear viscous damping in lb-sec/in or lb-in-sec/rad, and extn/compn force limits.

Loop #	Extn PWL	Comp PWL	LVB damping	F-extn	F-compn
1	16	17	4.49e4	0.0e8	-1.0e8

-List the type 4 - axle to track characteristics, the general lateral rail stiffness and damping coefficient, and, for each axle, IAX, an identification number, IBDAX, its general body number, WRAD, the nominal wheel radius and INDWH, a wheel rotation index, 1 for solid, 2 for independent wheels, ITRQ, traction torque input nos. for left and right wheels, 0 for none, and, for independent wheels, KWHL, DWHL, the axle torsional stiffness and damping.

Lateral Rail Stiffness lb/in	Lateral Rail Damping lb-sec/in
4.0e5	4.0e3

IAX	IBDAX	WRAD	INDWH	ITRQ-L	ITRQ-R	KWHL	DWHL
1	8	18.0	1	0	0	0.0	0.0
2	9	18.0	1	0	0	0.0	0.0
3	10	18.0	1	0	0	0.0	0.0
4	11	18.0	1	0	0	0.0	0.0

```
-How many different piecewise linear, (PWL), characteristics are required
    19
-List the data required for the connection characteristics,
  PWL, the piece-wise linear function no., IBP, the no. of Break Points in each
  PWL, Ordinate, lb or lb-in, over abscissa, in or rad, at each Break Point
  N.B. (1) Extension is assumed to be positive for both ordinate and abscissa
       (2) 0.0 for the first break point indicates symmetry about the origin
 PWL     IBP     Ordinates over Abscissae
 1       2       0.0      1.0e6
                 0.0      1.0
 2       2       0.0      1.0e3
                 0.0      1.0
 3       5      -1.0e6   -5.174e3   -2.3e3    0.0      0.0
                -1.3125  -0.3125     0.0      0.21     1.0
 4       3       0.0      7.0e2      7.0e2
                 0.0      0.007      1.0
 5       2       0.0      1.0e6
                 0.0      1.0
 6       2       0.0      1.0e3
                 0.0      1.0
 7       3       0.0      3.616e4    3.616e4
                 0.0      0.002      1.0
 8       3       0.0      4.05e3     5.0e5
                 0.0      0.45       0.55
 9       3       0.0      3.0e3      3.0e3
                 0.0      0.01       1.0
 10      4       0.0      1.4e5      8.1e5     1.698e6
                 0.0      0.01275    0.057     0.058
 11      3       0.0      3.0e4      3.0e4
                 0.0      0.003      1.0
 12      2       0.0      1.0e6
                 0.0      1.0
 13      2       0.0      1.0e3
                 0.0      1.0
 14      2       0.0      1.0e5
                 0.0      1.0
 15      2       0.0      1.0e3
                 0.0      1.0
 16      5      -1.25e5  -1.12e5  -9.06e4  -3.76e4  -7.52e3
                -4.0542  -3.9684  -3.9387  -1.7753  -0.2212
 17      5      -1.30e5  -1.11e5  -1.00e5  -4.02e4  -1.15e4
                -4.0542  -3.8853  -3.5914  -1.4563  -0.2212
 18      3       0.0      4080.0   8241.0
                 0.0      0.1071   0.1204
 19      3       0.0      650.0    642.5
                 0.0      0.1      1.0
```

Table 2. Post-Test NUCARS System File for Loaded PSMX 111 Car

```
-SYSTEM FILE (.SYS) for the program NUCARS Version 1.0
 N.B. Parameters are in lb., in. & sec. unless otherwise stated.
-Enter a title up to 80 characters long between the lines,
---------------------------------------------------------------------------
 Loaded Lightweight Car # 2   10/18/89
---------------------------------------------------------------------------
-FOR THE BODIES
-Provide the number of heavy bodies including axles (IMM), and the number
  of input or light bodies (IBIN, used for input degrees of freedom )
     IMM     IBIN
     11       8
-List the number, name, in single quotes up to 15 characters long, and
  position of each body, (and axle body), relative to a datum on the system
  center, in inches, followed by the number of degrees of freedom required,
  followed by a list of the degrees of freedom for each, in turn,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, 7=epsx, 8=epsy, 9=epsz.
  The 4 degrees of freedom required for each axle are  2 3 4 6
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & list of DoF's
   1      'Carbody        '  -275.0   0.0   68.07   8   2 3 4 5 6 7 8 9
   2      'Lead Bolster   '   -35.0   0.0   18.0    4   2 3 4 6
   3      'Trail Bolster  '  -521.0   0.0   18.0    4   2 3 4 6
   4      'Ld Lt Sideframe'   -35.0  39.5   18.0    5   1 2 3 5 6
   5      'Ld Rt Sideframe'   -35.0 -39.5   18.0    5   1 2 3 5 6
   6      'Tl Lt Sideframe'  -521.0  39.5   18.0    5   1 2 3 5 6
   7      'Tl Rt Sideframe'  -521.0 -39.5   18.0    5   1 2 3 5 6
   8      'Axle 1         '     0.0   0.0   18.0    4   2 3 4 6
   9      'Axle 2         '   -70.0   0.0   18.0    4   2 3 4 6
  10      'Axle 3         '  -486.0   0.0   18.0    4   2 3 4 6
  11      'Axle 4         '  -556.0   0.0   18.0    4   2 3 4 6
  continue the body list with the number and position of each input body,
  relative to the same datum, in inches, followed by the number of input
  degrees of freedom required, followed by a list of the degrees of freedom,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, the number of the input history
  for each degree of freedom, in turn, followed by a choice of input phase
  lag for the input to this body, 0 = no, 1 = yes.
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & DoF list  Input list  Lag
  12      'Axle 1 Lt Wheel'     0.0  29.75   0.0   2   2 3       1  3       1
  13      'Axle 1 Rt Wheel'     0.0 -29.75   0.0   2   2 3       2  4       1
  14      'Axle 2 Lt Wheel'   -70.0  29.75   0.0   2   2 3       1  3       1
  15      'Axle 2 Rt Wheel'   -70.0 -29.75   0.0   2   2 3       2  4       1
  16      'Axle 3 Lt Wheel'  -486.0  29.75   0.0   2   2 3       1  3       1
  17      'Axle 3 Rt Wheel'  -486.0 -29.75   0.0   2   2 3       2  4       1
  18      'Axle 4 Lt Wheel'  -556.0  29.75   0.0   2   2 3       1  3       1
  19      'Axle 4 Rt Wheel'  -556.0 -29.75   0.0   2   2 3       2  4       1
-For all heavy bodies with flexible modes, give the position of each body
  geometric center, in the X direction from the datum, backward is -ve, its
  length in inches, the natural frequencies, in Hz., and the damping ratios
  in twist, vertical & lateral bending, as required.
  Body #    X-Posn    X-Length    Nat Frequencies(Hz.)      Damping Ratios
   1        -278.0     606.0       3.6    7.7    4.7        0.2    0.2    0.4
-List the mass, roll, pitch and yaw inertias, in order,
  for each heavy body, including axles,
 634.47     1.011e6    1.608e7    1.636e7
   4.77     3.48e3     0.0        3.48e3
   4.77     3.48e3     0.0        3.48e3
   3.0      0.0        1.37e3     1.37e3
   3.0      0.0        1.37e3     1.37e3
   3.0      0.0        1.37e3     1.37e3
   3.0      0.0        1.37e3     1.37e3
   7.09     5.41e3     1.38e3     5.41e3
   7.09     5.41e3     1.38e3     5.41e3
   7.09     5.41e3     1.38e3     5.41e3
   7.09     5.41e3     1.38e3     5.41e3
```

-FOR THE CONNECTIONS (including suspensions) Identify the following parameters,

-Number of connections:

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-Complete the following tables for each connection, identifying:

a name, in single quotes up to 20 characters long;
its position relative to the chosen datum in x, y, z inches;
the number of the body at each end, 0 for an earth in local track coords.;
a number indicating the degree(s) of freedom, translational 1,2,3 or rotational 4,5,6; in x,y,z resp., including 2 for lateral wheel motion;
the type 1 - parallel pair of spring and damper characteristics
2 - series pair of spring and damper characteristics
3 - device with hysteresis between 2 PWL characteristics, e.g. carriage spring or load sensitive suspension
4 - lateral/longitudinal suspension of the wheel on rail in the track plane
5 - connection force as a history of the distance moved
and the identification number for each of type 1, 2 and 3; axle number for type 4; input function number for type 5.

Note - single characteristics are treated as parallel pairs with the missing characteristic set to zero in the subsequent table.

-Complete for all connections in turn,

Conn #	' 20 CHARACTER NAME '	Posn in X,	Y	& Z	Body1	Body2	DoF.	Type	Number
1	'Ld Bols-Bod Lt CB Vt'	-35.0	8.0	25.0	1	2	3	1	1
2	'Ld Bols-Bod Rt CB Vt'	-35.0	-8.0	25.0	1	2	3	1	1
3	'Tr Bols-Bod Lt CB Vt'	-521.0	8.0	25.0	1	3	3	1	1
4	'Tr Bols-Bod Rt CB Vt'	-521.0	-8.0	25.0	1	3	3	1	1
5	'Ld Bols-Bod Lt SB Vt'	-35.0	25.0	25.0	1	2	3	1	2
6	'Ld Bols-Bod Rt SB Vt'	-35.0	-25.0	25.0	1	2	3	1	2
7	'Tr Bols-Bod Lt SB Vt'	-521.0	25.0	25.0	1	3	3	1	2
8	'Tr Bols-Bod Rt SB Vt'	-521.0	-25.0	25.0	1	3	3	1	2
9	'Lead Bols-Bod CB Lat'	-35.0	0.0	25.0	1	2	2	1	3
10	'Trail Bols-Bod CB Lt'	-521.0	0.0	25.0	1	3	2	1	3
11	'Lead Bols-Bod CB Yaw'	-35.0	0.0	25.0	1	2	6	1	4
12	'Trl Bols-Bod CB Yaw '	-521.0	0.0	25.0	1	3	6	1	4
13	'Ld Bols-Sdfm Lt Vert'	-35.0	39.5	18.0	2	4	3	3	1
14	'Ld Bols-Sdfm Rt Vert'	-35.0	-39.5	18.0	2	5	3	3	1
15	'Tr Bols-Sdfm Lt Vert'	-521.0	39.5	18.0	3	6	3	3	1
16	'Tr Bols-Sdfm Rt Vert'	-521.0	-39.5	18.0	3	7	3	3	1
17	'Ld Bols-Sdfm Lt Lat '	-35.0	39.5	18.0	2	4	2	1	5
18	'Ld Bols-Sdfm Rt Lat '	-35.0	-39.5	18.0	2	5	2	1	5
19	'Tr Bols-Sdfm Lt Lat '	-521.0	39.5	18.0	3	6	2	1	5
20	'Tr Bols-Sdfm Rt Lat '	-521.0	-39.5	18.0	3	7	2	1	5
21	'Ld Bols-Sdfm Lt Yaw '	-35.0	39.5	18.0	2	4	6	1	6
22	'Ld Bols-Sdfm Rt Yaw '	-35.0	-39.5	18.0	2	5	6	1	6
23	'Tr Bols-Sdfm Lt Yaw '	-521.0	39.5	18.0	3	6	6	1	6
24	'Tr Bols-Sdfm Rt Yaw '	-521.0	-39.5	18.0	3	7	6	1	6
25	'Ld Bols-Sdfm Lt Long'	-35.0	39.5	18.0	2	4	1	1	7
26	'Ld Bols-Sdfm Rt Long'	-35.0	-39.5	18.0	2	5	1	1	7
27	'Tr Bols-Sdfm Lt Long'	-521.0	39.5	18.0	3	6	1	1	7
28	'Tr Bols-Sdfm Rt Long'	-521.0	-39.5	18.0	3	7	1	1	7
29	'Ax 1 Lt BA-Sdfm Long'	0.0	39.5	21.0	4	8	1	1	9
30	'Ax 1 Rt BA-Sdfm Long'	0.0	-39.5	21.0	5	8	1	1	9
31	'Ax 2 Lt BA-Sdfm Long'	-70.0	39.5	21.0	4	9	1	1	9
32	'Ax 2 Rt BA-Sdfm Long'	-70.0	-39.5	21.0	5	9	1	1	9
33	'Ax 3 Lt BA-Sdfm Long'	-486.0	39.5	21.0	6	10	1	1	9
34	'Ax 3 Rt BA-Sdfm Long'	-486.0	-39.5	21.0	7	10	1	1	9
35	'Ax 4 Lt BA-Sdfm Long'	-556.0	39.5	21.0	6	11	1	1	9
36	'Ax 4 Rt BA-Sdfm Long'	-556.0	-39.5	21.0	7	11	1	1	9
37	'Ax 1 Lt BA-Sdfm Lat '	0.0	39.5	21.0	4	8	2	1	9
38	'Ax 1 Rt BA-Sdfm Lat '	0.0	-39.5	21.0	5	8	2	1	9
39	'Ax 2 Lt BA-Sdfm Lat '	-70.0	39.5	21.0	4	9	2	1	9
40	'Ax 2 Rt BA-Sdfm Lat '	-70.0	-39.5	21.0	5	9	2	1	9
41	'Ax 3 Lt BA-Sdfm Lat '	-486.0	39.5	21.0	6	10	2	1	9
42	'Ax 3 Rt BA-Sdfm Lat '	-486.0	-39.5	21.0	7	10	2	1	9
43	'Ax 4 Lt BA-Sdfm Lat '	-556.0	39.5	21.0	6	11	2	1	9
44	'Ax 4 Rt BA-Sdfm Lat '	-556.0	-39.5	21.0	7	11	2	1	9
45	'Ax 1 Lt BA-Sdfm Vert'	0.0	39.5	21.0	4	8	3	1	1

```
46        'Ax 1 Rt BA-Sdfm Vert'    0.0 -39.5  21.0    5     8    3     1    1
47        'Ax 2 Lt BA-Sdfm Vert'  -70.0  39.5  21.0    4     9    3     1    1
48        'Ax 2 Rt BA-Sdfm Vert'  -70.0 -39.5  21.0    5     9    3     1    1
49        'Ax 3 Lt BA-Sdfm Vert' -486.0  39.5  21.0    6    10    3     1    1
50        'Ax 3 Rt BA-Sdfm Vert' -486.0 -39.5  21.0    7    10    3     1    1
51        'Ax 4 Lt BA-Sdfm Vert' -556.0  39.5  21.0    6    11    3     1    1
52        'Ax 4 Rt BA-Sdfm Vert' -556.0 -39.5  21.0    7    11    3     1    1
53        'Ax 1 Lt Whl/Rail Vrt'    0.0  29.75  0.0    8    12    3     1    8
54        'Ax 1 Rt Whl/Rail Vrt'    0.0 -29.75  0.0    8    13    3     1    8
55        'Ax 2 Lt Whl/Rail Vrt'  -70.0  29.75  0.0    9    14    3     1    8
56        'Ax 2 Rt Whl/Rail Vrt'  -70.0 -29.75  0.0    9    15    3     1    8
57        'Ax 3 Lt Whl/Rail Vrt' -486.0  29.75  0.0   10    16    3     1    8
58        'Ax 3 Rt Whl/Rail Vrt' -486.0 -29.75  0.0   10    17    3     1    8
59        'Ax 4 Lt Whl/Rail Vrt' -556.0  29.75  0.0   11    18    3     1    8
60        'Ax 4 Rt Whl/Rail Vrt' -556.0 -29.75  0.0   11    19    3     1    8
61        'Ax 1 Lt Whl/Rail lat'    0.0  29.75  0.0    8    12    2     4    1
62        'Ax 1 Rt Whl/Rail Lat'    0.0 -29.75  0.0    8    13    2     4    1
63        'Ax 2 Lt Whl/Rail Lat'  -70.0  29.75  0.0    9    14    2     4    2
64        'Ax 2 Rt Whl/Rail Lat'  -70.0 -29.75  0.0    9    15    2     4    2
65        'Ax 3 Lt Whl/Rail Lat' -486.0  29.75  0.0   10    16    2     4    3
66        'Ax 3 Rt Whl/Rail Lat' -486.0 -29.75  0.0   10    17    2     4    3
67        'Ax 4 Lt Whl/Rail Lat' -556.0  29.75  0.0   11    18    2     4    4
68        'Ax 4 Rt Whl/Rail Lat' -556.0 -29.75  0.0   11    19    2     4    4
-List for each pair of type 1 - parallel connections, its number, followed by
  the identification numbers of the piecewise linear characteristics
  for the stiffness and damping respectively, zero if absent, and
  the combined force or moment limit in extn & compn, lb or lb-in.,
  0.0 in extension at the vertical rail/wheel conn. allows valid wheel lift.
  (If no limit exists, set the F-values outside the expected range.)
     Pair #     Stiff PWL     Damp PWL       F-extn.      F-compn.
       1           1             2           0.0e8        -1.0e8
       2           3             4           0.0e8        -1.0e8
       3           5             6           1.0e8        -1.0e8
       4           0             7           1.0e8        -1.0e8
       5           8             9           1.0e8        -1.0e8
       6          10            11           1.0e8        -1.0e8
       7          12            13           1.0e8        -1.0e8
       8          14            15           0.0e8        -1.0e8
       9          18            19           1.0e8        -1.0e8
-List for each pair of type 2 - series connections, its number, followed by
  the identification numbers of the piecewise linear characteristics
  for the stiffness and damping respectively, and the stroke limit
  in extension & compression for the pair, in or rad, and the stiffness
  of the stop at the limit in lb/in or lb-in/rad.
  (If no limit exists, set the S-values outside the expected range.)
     Pair #     Stiff PWL    Damp PWL     S-extn.      S-compn.    Stop K
-List the type 3 - hysteresis loop characteristics, giving to each a number,
  identification numbers for the extension and compression PWLs, a linear
  viscous damping in lb-sec/in or lb-in-sec/rad, and extn/compn force limits.
     Loop #      Extn PWL    Comp PWL      LVB damping      F-extn    F-compn
       1            16          17           4.49e4         0.0e8    -1.0e8
-List the type 4 - axle to track characteristics, the general lateral rail
  stiffness and damping coefficient, and, for each axle, IAX, an identification
  number, IBDAX, its general body number, WRAD, the nominal wheel radius and
  INDWH, a wheel rotation index, 1 for solid, 2 for independent wheels, ITRQ,
  traction torque input nos. for left and right wheels, 0 for none, and, for
  independent wheels, KWHL, DWHL, the axle torsional stiffness and damping.
       Lateral Rail Stiffness lb/in       Lateral Rail Damping lb-sec/in
                 4.0e5                              4.0e3
    IAX       IBDAX      WRAD     INDWH       ITRQ-L      ITRQ-R      KWHL      DWHL
     1          8        18.0       1            0           0         0.0       0.0
     2          9        18.0       1            0           0         0.0       0.0
     3         10        18.0       1            0           0         0.0       0.0
     4         11        18.0       1            0           0         0.0       0.0
-How many different piecewise linear, (PWL), characteristics are required
     19
```

```
-List the data required for the connection characteristics,
  PWL, the piece-wise linear function no., IBP, the no. of Break Points in each
  PWL, Ordinate, lb or lb-in, over abscissa, in or rad, at each Break Point
  N.B. (1) Extension is assumed to be positive for both ordinate and abscissa
       (2) 0.0 for the first break point indicates symmetry about the origin
PWL     IBP     Ordinates over Abscissae
1       2       0.0     1.0e6
                0.0     1.0
2       2       0.0     1.0e3
                0.0     1.0
3       5       -1.0e6  -5.174e3  -2.3e3    0.0      0.0
                -1.3125 -0.3125    0.0      0.21     1.0
4       3       0.0     7.0e2     7.0e2
                0.0     0.007     1.0
5       2       0.0     1.0e6
                0.0     1.0
6       2       0.0     1.0e3
                0.0     1.0
7       3       0.0     1.985e5   1.985e5
                0.0     0.002     1.0
8       3       0.0     8.1e3     1.0e6
                0.0     0.45      0.55
9       3       0.0     6.0e3     6.0e3
                0.0     0.01      1.0
10      4       0.0     1.4e5     8.1e5     1.698e6
                0.0     0.01275   0.057     0.058
11      3       0.0     3.0e4     3.0e4
                0.0     0.003     1.0
12      2       0.0     1.0e6
                0.0     1.0
13      2       0.0     1.0e3
                0.0     1.0
14      2       0.0     1.0e5
                0.0     1.0
15      2       0.0     1.0e3
                0.0     1.0
16      5       -1.25e5 -1.12e5 -9.06e4 -3.76e4 -7.52e3
                -4.0542 -3.9684 -3.9387 -1.7753 -0.2212
17      5       -1.30e5 -1.11e5 -1.00e5 -4.02e4 -1.15e4
                -4.0542 -3.8853 -3.5914 -1.4563 -0.2212
18      3       0.0     4080.0  8241.0
                0.0     0.1071  0.1204
19      3       0.0     650.0   642.5
                0.0     0.1     1.0
```

Table 3. Post-Test NUCARS System File for Empty PSMX 111 Car

```
-SYSTEM FILE (.SYS) for the program NUCARS Version 1.0
 N.B. Parameters are in lb., in. & sec. unless otherwise stated.
-Enter a title up to 80 characters long between the lines,
--------------------------------------------------------------------------------
LTWT CAR #2 EMPTY REAL TRACK SUSP. CHARS. Vert LVB = 4.49e6 revised 9/16/91
--------------------------------------------------------------------------------
-FOR THE BODIES
-Provide the number of heavy bodies including axles (IMM), and the number
  of input or light bodies (IBIN, used for input degrees of freedom )
     IMM    IBIN
     11      8
-List the number, name, in single quotes up to 15 characters long, and
  position of each body, (and axle body), relative to a datum on the system
  center, in inches, followed by the number of degrees of freedom required,
  followed by a list of the degrees of freedom for each, in turn,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, 7=epsx, 8=epsy, 9=epsz.
  The 4 degrees of freedom required for each axle are  2 3 4 6
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & list of DoF's
   1      'Carbody        '  -281.0   0.0   57.2    8   2 3 4 5 6 7 8 9
   2      'Lead Bolster   '   -35.0   0.0   18.0    4   2 3 4 6
   3      'Trail Bolster  '  -521.0   0.0   18.0    4   2 3 4 6
   4      'Ld Lt Sideframe'   -35.0  39.5   18.0    5   1 2 3 5 6
   5      'Ld Rt Sideframe'   -35.0 -39.5   18.0    5   1 2 3 5 6
   6      'Tl Lt Sideframe'  -521.0  39.5   18.0    5   1 2 3 5 6
   7      'Tl Rt Sideframe'  -521.0 -39.5   18.0    5   1 2 3 5 6
   8      'Axle 1         '     0.0   0.0   18.0    4   2 3 4 6
   9      'Axle 2         '   -70.0   0.0   18.0    4   2 3 4 6
  10      'Axle 3         '  -486.0   0.0   18.0    4   2 3 4 6
  11      'Axle 4         '  -556.0   0.0   18.0    4   2 3 4 6
  continue the body list with the number and position of each input body,
  relative to the same datum, in inches, followed by the number of input
  degrees of freedom required, followed by a list of the degrees of freedom,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, the number of the input history
  for each degree of freedom, in turn, followed by a choice of input phase
  lag for the input to this body, 0 = no, 1 = yes.
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & DoF list  Input list  Lag
  12      'Axle 1 Lt Wheel'     0.0  29.75   0.0   2    2 3       1  3        1
  13      'Axle 1 Rt Wheel'     0.0 -29.75   0.0   2    2 3       2  4        1
  14      'Axle 2 Lt Wheel'   -70.0  29.75   0.0   2    2 3       1  3        1
  15      'Axle 2 Rt Wheel'   -70.0 -29.75   0.0   2    2 3       2  4        1
  16      'Axle 3 Lt Wheel'  -486.0  29.75   0.0   2    2 3       1  3        1
  17      'Axle 3 Rt Wheel'  -486.0 -29.75   0.0   2    2 3       2  4        1
  18      'Axle 4 Lt Wheel'  -556.0  29.75   0.0   2    2 3       1  3        1
  19      'Axle 4 Rt Wheel'  -556.0 -29.75   0.0   2    2 3       2  4        1
-For all heavy bodies with flexible modes, give the position of each body
  geometric center, in the X direction from the datum, backward is -ve, its
  length in inches, the natural frequencies, in Hz., and the damping ratios
  in twist, vertical & lateral bending, as required.
  Body #     X-Posn     X-Length     Nat Frequencies(Hz.)       Damping Ratios
   1         -278.0      606.0         9.4  26.2   15.9         0.05  0.05    0.1
-List the mass, roll, pitch and yaw inertias, in order,
  for each heavy body, including axles,
  57.84      2.26e5     1.872e6    1.911e6
   5.09      3.71e3     0.0        3.71e3
   5.09      3.71e3     0.0        3.71e3
   3.20      0.0        1.46e3     1.46e3
   3.20      0.0        1.46e3     1.46e3
   3.20      0.0        1.46e3     1.46e3
   3.20      0.0        1.46e3     1.46e3
   7.56      5.77e3     1.47e3     5.77e3
   7.56      5.77e3     1.47e3     5.77e3
   7.56      5.77e3     1.47e3     5.77e3
   7.56      5.77e3     1.47e3     5.77e3
```

-FOR THE CONNECTIONS (including suspensions) Identify the following parameters,

-Number of connections:

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-Complete the following tables for each connection, identifying:

a name, in single quotes up to 20 characters long;
its position relative to the chosen datum in x, y, z inches;
the number of the body at each end, 0 for an earth in local track coords.;
a number indicating the degree(s) of freedom, translational 1,2,3 or
rotational 4,5,6; in x,y,z resp., including 2 for lateral wheel motion;
the type 1 - parallel pair of spring and damper characteristics
2 - series pair of spring and damper characteristics
3 - device with hysteresis between 2 PWL characteristics,
e.g. carriage spring or load sensitive suspension
4 - lateral/longitudinal suspension of the wheel on rail
in the track plane
5 - connection force as a history of the distance moved
and the identification number for each of type 1, 2 and 3;
axle number for type 4; input function number for type 5.

Note - single characteristics are treated as parallel pairs with the
missing characteristic set to zero in the subsequent table.

-Complete for all connections in turn,

Conn #	'20 CHARACTER NAME '	Posn in X,	Y &	Z	Body1	Body2	DoF.	Type	Number
1	'Ld Bols-Bod CB Vt'	-35.0	0.0	27.0	1	2	3	1	1
2	'Ld Bols-Bod CB Roll'	-35.0	0.0	27.0	1	2	4	1	2
3	'Tr Bols-Bod CB Vt'	-521.0	0.0	27.0	1	3	3	1	1
4	'Tr Bols-Bod CB Roll'	-521.0	0.0	27.0	1	3	4	1	2
5	'Lead Bols-Bod CB Lat'	-35.0	0.0	27.0	1	2	2	1	3
6	'Trail Bols-Bod CB Lt'	-521.0	0.0	27.0	1	3	2	1	3
7	'Lead Bols-Bod CB Yaw'	-35.0	0.0	27.0	1	2	6	1	4
8	'Trl Bols-Bod CB Yaw '	-521.0	0.0	27.0	1	3	6	1	4
9	'Ld Bols-Sdfm Lt Vert'	-35.0	39.5	18.0	2	4	3	3	2
10	'Ld Bols-Sdfm Rt Vert'	-35.0	-39.5	18.0	2	5	3	3	3
11	'Tr Bols-Sdfm Lt Vert'	-521.0	39.5	18.0	3	6	3	3	1
12	'Tr Bols-Sdfm Rt Vert'	-521.0	-39.5	18.0	3	7	3	3	1
13	'Ld Bols-Sdfm Lt Lat '	-35.0	39.5	18.0	2	4	2	1	5
14	'Ld Bols-Sdfm Rt Lat '	-35.0	-39.5	18.0	2	5	2	1	5
15	'Tr Bols-Sdfm Lt Lat '	-521.0	39.5	18.0	3	6	2	1	5
16	'Tr Bols-Sdfm Rt Lat '	-521.0	-39.5	18.0	3	7	2	1	5
17	'Ld Bols-Sdfm Lt Yaw '	-35.0	39.5	18.0	2	4	6	1	6
18	'Ld Bols-Sdfm Rt Yaw '	-35.0	-39.5	18.0	2	5	6	1	6
19	'Tr Bols-Sdfm Lt Yaw '	-521.0	39.5	18.0	3	6	6	1	6
20	'Tr Bols-Sdfm Rt Yaw '	-521.0	-39.5	18.0	3	7	6	1	6
21	'Ld Bols-Sdfm Lt Long'	-35.0	39.5	18.0	2	4	1	1	7
22	'Ld Bols-Sdfm Rt Long'	-35.0	-39.5	18.0	2	5	1	1	7
23	'Tr Bols-Sdfm Lt Long'	-521.0	39.5	18.0	3	6	1	1	7
24	'Tr Bols-Sdfm Rt Long'	-521.0	-39.5	18.0	3	7	1	1	7
25	'Ax 1 Lt BA-Sdfm Long'	0.0	39.5	21.0	4	8	1	1	10
26	'Ax 1 Rt BA-Sdfm Long'	0.0	-39.5	21.0	5	8	1	1	11
27	'Ax 2 Lt BA-Sdfm Long'	-70.0	39.5	21.0	4	9	1	1	12
28	'Ax 2 Rt BA-Sdfm Long'	-70.0	-39.5	21.0	5	9	1	1	13
29	'Ax 3 Lt BA-Sdfm Long'	-486.0	39.5	21.0	6	10	1	1	9
30	'Ax 3 Rt BA-Sdfm Long'	-486.0	-39.5	21.0	7	10	1	1	9
31	'Ax 4 Lt BA-Sdfm Long'	-556.0	39.5	21.0	6	11	1	1	9
32	'Ax 4 Rt BA-Sdfm Long'	-556.0	-39.5	21.0	7	11	1	1	9
33	'Ax 1 Lt BA-Sdfm Lat '	0.0	39.5	21.0	4	8	2	1	9
34	'Ax 1 Rt BA-Sdfm Lat '	0.0	-39.5	21.0	5	8	2	1	9
35	'Ax 2 Lt BA-Sdfm Lat '	-70.0	39.5	21.0	4	9	2	1	9
36	'Ax 2 Rt BA-Sdfm Lat '	-70.0	-39.5	21.0	5	9	2	1	9
37	'Ax 3 Lt BA-Sdfm Lat '	-486.0	39.5	21.0	6	10	2	1	9
38	'Ax 3 Rt BA-Sdfm Lat '	-486.0	-39.5	21.0	7	10	2	1	9
39	'Ax 4 Lt BA-Sdfm Lat '	-556.0	39.5	21.0	6	11	2	1	9
40	'Ax 4 Rt BA-Sdfm Lat '	-556.0	-39.5	21.0	7	11	2	1	9
41	'Ax 1 Lt BA-Sdfm Vert'	0.0	39.5	21.0	4	8	3	1	1
42	'Ax 1 Rt BA-Sdfm Vert'	0.0	-39.5	21.0	5	8	3	1	1
43	'Ax 2 Lt BA-Sdfm Vert'	-70.0	39.5	21.0	4	9	3	1	1
44	'Ax 2 Rt BA-Sdfm Vert'	-70.0	-39.5	21.0	5	9	3	1	1
45	'Ax 3 Lt BA-Sdfm Vert'	-486.0	39.5	21.0	6	10	3	1	1

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46       'Ax 3 Rt BA-Sdfm Vert' -486.0 -39.5  21.0    7     10    3     1     1
47       'Ax 4 Lt BA-Sdfm Vert' -556.0  39.5  21.0    6     11    3     1     1
48       'Ax 4 Rt BA-Sdfm Vert' -556.0 -39.5  21.0    7     11    3     1     1
49       'Ax 1 Lt Whl/Rail Vrt'    0.0  29.75  0.0    8     12    3     1     8
50       'Ax 1 Rt Whl/Rail Vrt'    0.0 -29.75  0.0    8     13    3     1     8
51       'Ax 2 Lt Whl/Rail Vrt'  -70.0  29.75  0.0    9     14    3     1     8
52       'Ax 2 Rt Whl/Rail Vrt'  -70.0 -29.75  0.0    9     15    3     1     8
53       'Ax 3 Lt Whl/Rail Vrt' -486.0  29.75  0.0   10     16    3     1     8
54       'Ax 3 Rt Whl/Rail Vrt' -486.0 -29.75  0.0   10     17    3     1     8
55       'Ax 4 Lt Whl/Rail Vrt' -556.0  29.75  0.0   11     18    3     1     8
56       'Ax 4 Rt Whl/Rail Vrt' -556.0 -29.75  0.0   11     19    3     1     8
57       'Ax 1 Lt Whl/Rail lat'    0.0  29.75  0.0    8     12    2     4     1
58       'Ax 1 Rt Whl/Rail Lat'    0.0 -29.75  0.0    8     13    2     4     1
59       'Ax 2 Lt Whl/Rail Lat'  -70.0  29.75  0.0    9     14    2     4     2
60       'Ax 2 Rt Whl/Rail Lat'  -70.0 -29.75  0.0    9     15    2     4     2
61       'Ax 3 Lt Whl/Rail Lat' -486.0  29.75  0.0   10     16    2     4     3
62       'Ax 3 Rt Whl/Rail Lat' -486.0 -29.75  0.0   10     17    2     4     3
63       'Ax 4 Lt Whl/Rail Lat' -556.0  29.75  0.0   11     18    2     4     4
64       'Ax 4 Rt Whl/Rail Lat' -556.0 -29.75  0.0   11     19    2     4     4
-List for each pair of type 1 - parallel connections, its number, followed by
  the identification numbers of the piecewise linear characteristics
  for the stiffness and damping respectively, zero if absent, and
  the combined force or moment limit in extn & compn, lb or lb-in.,
  0.0 in extension at the vertical rail/wheel conn. allows valid wheel lift.
  (If no limit exists, set the F-values outside the expected range.)
     Pair #     Stiff PWL     Damp PWL       F-extn.       F-compn.
       1            1            2            0.0e8         -1.0e8
       2            3            4            1.0e8         -1.0e8
       3            5            6            1.0e8         -1.0e8
       4            7            8            1.0e8         -1.0e8
       5            9           10            1.0e8         -1.0e8
       6           11           12            1.0e8         -1.0e8
       7           13           14            1.0e8         -1.0e8
       8           15           16            0.0e8         -1.0e8
       9           19           20            1.0e8         -1.0e8
      10           21           20            1.0e8         -1.0e8
      11           22           20            1.0e8         -1.0e8
      12           23           20            1.0e8         -1.0e8
      13           24           20            1.0e8         -1.0e8
-List for each pair of type 2 - series connections, its number, followed by
  the identification numbers of the piecewise linear characteristics
  for the stiffness and damping respectively, and the stroke limit
  in extension & compression for the pair, in or rad, and the stiffness
  of the stop at the limit in lb/in or lb-in/rad.
  (If no limit exists, set the S-values outside the expected range.)
     Pair #     Stiff PWL    Damp PWL     S-extn.      S-compn.     Stop K
-List the type 3 - hysteresis loop characteristics, giving to each a number,
  identification numbers for the extension and compression PWLs, a linear
  viscous damping in lb-sec/in or lb-in-sec/rad, and extn/compn force limits.
     Loop #      Extn PWL     Comp PWL      LVB damping      F-extn     F-compn
       1            17           18           4.49e6          0.0e8      -1.0e8
       2            25           26           4.49e6          0.0e8      -1.0e8
       3            27           28           4.49e6          0.0e8      -1.0e8
-List the type 4 - axle to track characteristics, the general lateral rail
  stiffness and damping coefficient, and, for each axle, IAX, an identification
  number, IBDAX, its general body number, WRAD, the nominal wheel radius and
  INDWH, a wheel rotation index, 1 for solid, 2 for independent wheels, ITRQ,
  traction torque input nos. for left and right wheels, 0 for none, and, for
  independent wheels, KWHL, DWHL, the axle torsional stiffness and damping.
       Lateral Rail Stiffness lb/in      Lateral Rail Damping lb-sec/in
                 4.0e5                               4.0e3
    IAX        IBDAX      WRAD      INDWH      ITRQ-L    ITRQ-R      KWHL      DWHL
     1           8        18.0        1           0         0         0.0       0.0
     2           9        18.0        1           0         0         0.0       0.0
     3          10        18.0        1           0         0         0.0       0.0
     4          11        18.0        1           0         0         0.0       0.0
-How many different piecewise linear, (PWL), characteristics are required
     28
```

-List the data required for the connection characteristics,
PWL, the piece-wise linear function no., IBP, the no. of Break Points in each
PWL, Ordinate, lb or lb-in, over abscissa, in or rad, at each Break Point
N.B. (1) Extension is assumed to be positive for both ordinate and abscissa
(2) 0.0 for the first break point indicates symmetry about the origin

PWL	IBP	Ordinates over Abscissae					
1	2	0.0	1.0e6				
		0.0	1.0				
2	2	0.0	1.0e3				
		0.0	1.0				
3	5	0.0	8.94e4	8.94e4	2.79e5	2.79e5	
		0.0	0.0012	0.0263	0.0323	1.0	
4	3	0.0	2.0e4	2.0e4			
		0.0	0.1e-3	1.0e-3			
5	2	0.0	1.0e6				
		0.0	1.0				
6	2	0.0	1.0e3				
		0.0	1.0				
7	2	0.0	1.0e4				
		0.0	1.0				
8	3	0.0	36.16e3	36.16e3			
		0.0	0.002	1.0			
9	3	0.0	3.0e3	3.0e4			
		0.0	0.45	0.55			
10	3	0.0	3.0e3	3.0e3			
		0.0	0.01	1.0			
11	4	0.0	1.4e5	8.1e5	1.698e6		
		0.0	0.01275	0.057	0.058		
12	3	0.0	5.0e4	5.0e4			
		0.0	0.005	1.0			
13	2	0.0	1.0e6				
		0.0	1.0				
14	2	0.0	1.0e3				
		0.0	1.0				
15	2	0.0	1.0e5				
		0.0	1.0				
16	2	0.0	1.0e3				
		0.0	1.0				
17	5	-1.25e5	-1.12e5	-9.06e4	-3.76e4	-7.52e3	
		-4.0542	-3.9684	-3.9387	-1.7753	-0.2212	
18	5	-1.30e5	-1.11e5	-1.00e5	-4.02e4	-1.15e4	
		-4.0542	-3.8853	-3.5914	-1.4563	-0.2212	
19	3	0.0	4080.0	8241.0			
		0.0	0.1071	0.1204			
20	3	0.0	650.0	642.5			
		0.0	0.1	1.0			
21	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.0496	-0.0363	0.0708	0.1779	0.1912	
22	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.1912	-0.1779	-0.0708	0.0363	0.0496	
23	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.0407	-0.0274	0.0797	0.1868	0.2001	
24	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.2001	-0.1868	-0.0797	0.0274	0.0407	
25	6	-1.25e5	-9.81e4	-9.30e4	-7.67e4	-4.36e4	-7.97e3
		-4.07	-4.00	-3.93	-3.33	-1.89	-0.30
26	5	-1.30e5	-1.13e5	-9.13e4	-5.28e4	-1.26e4	
		-4.07	-3.94	-3.15	-1.91	-0.30	
27	5	-1.25e5	-9.36e4	-7.40e4	-4.18e4	-6.8e3	
		-4.05	-3.98	-3.35	-2.12	-0.19	
28	6	-1.30e5	-1.07e5	-8.88e4	-7.48e4	-5.19e4	-1.12e4
		-4.05	-4.01	-3.55	-3.23	-2.05	-0.19

Table 4. Post-Test NUCARS System File for Loaded PSMX 111 Car

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-SYSTEM FILE (.SYS) for the program NUCARS Version 1.0
 N.B. Parameters are in lb., in. & sec. unless otherwise stated.
-Enter a title up to 80 characters long between the lines,
--------------------------------------------------------------------------------
LTWT CAR #2 LOADED REAL TRACK SUSP. CHAR.  VERT. LVB = 4.49e6  revised 9/16/91
--------------------------------------------------------------------------------
-FOR THE BODIES
-Provide the number of heavy bodies including axles (IMM), and the number
  of input or light bodies (IBIN, used for input degrees of freedom )
     IMM    IBIN
     11      8
-List the number, name, in single quotes up to 15 characters long, and
  position of each body, (and axle body), relative to a datum on the system
  center, in inches, followed by the number of degrees of freedom required,
  followed by a list of the degrees of freedom for each, in turn,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, 7=epsx, 8=epsy, 9=epsz.
  The 4 degrees of freedom required for each axle are  2 3 4 6
  Body #  ' 15 CHAR NAME  ' Posn in X, Y & Z  No. & list of DoF's
   1      'Carbody        '  -275.0   0.0   68.07   8   2 3 4 5 6 7 8 9
   2      'Lead Bolster   '   -35.0   0.0   18.0    4   2 3 4 6
   3      'Trail Bolster  '  -521.0   0.0   18.0    4   2 3 4 6
   4      'Ld Lt Sideframe'   -35.0  39.5   18.0    5   1 2 3 5 6
   5      'Ld Rt Sideframe'   -35.0 -39.5   18.0    5   1 2 3 5 6
   6      'Tl Lt Sideframe'  -521.0  39.5   18.0    5   1 2 3 5 6
   7      'Tl Rt Sideframe'  -521.0 -39.5   18.0    5   1 2 3 5 6
   8      'Axle 1         '     0.0   0.0   18.0    4   2 3 4 6
   9      'Axle 2         '   -70.0   0.0   18.0    4   2 3 4 6
  10      'Axle 3         '  -486.0   0.0   18.0    4   2 3 4 6
  11      'Axle 4         '  -556.0   0.0   18.0    4   2 3 4 6
  continue the body list with the number and position of each input body,
  relative to the same datum, in inches, followed by the number of input
  degrees of freedom required, followed by a list of the degrees of freedom,
  from  1=x, 2=y, 3=z, 4=phi, 5=theta, 6=psi, the number of the input history
  for each degree of freedom, in turn, followed by a choice of input phase
  lag for the input to this body, 0 = no, 1 = yes.
  Body #  ' 15 CHAR NAME  '  Posn in X, Y & Z  No. & DoF list  Input list  Lag
  12      'Axle 1 Lt Wheel'     0.0  29.75   0.0   2   2 3       1  3       1
  13      'Axle 1 Rt Wheel'     0.0 -29.75   0.0   2   2 3       2  4       1
  14      'Axle 2 Lt Wheel'   -70.0  29.75   0.0   2   2 3       1  3       1
  15      'Axle 2 Rt Wheel'   -70.0 -29.75   0.0   2   2 3       2  4       1
  16      'Axle 3 Lt Wheel'  -486.0  29.75   0.0   2   2 3       1  3       1
  17      'Axle 3 Rt Wheel'  -486.0 -29.75   0.0   2   2 3       2  4       1
  18      'Axle 4 Lt Wheel'  -556.0  29.75   0.0   2   2 3       1  3       1
  19      'Axle 4 Rt Wheel'  -556.0 -29.75   0.0   2   2 3       2  4       1
-For all heavy bodies with flexible modes, give the position of each body
  geometric center, in the X direction from the datum, backward is -ve, its
  length in inches, the natural frequencies, in Hz., and the damping ratios
  in twist, vertical & lateral bending, as required.
  Body #     X-Posn    X-Length    Nat Frequencies(Hz.)      Damping Ratios
   1         -278.0     606.0        2.0   5.3   4.8        0.1   0.04    0.12
-List the mass, roll, pitch and yaw inertias, in order,
  for each heavy body, including axles,
 632.21      1.007e6   1.600e7   1.630e7
   5.09      3.71e3    0.0       3.71e3
   5.09      3.71e3    0.0       3.71e3
   3.20      0.0       1.46e3    1.46e3
   3.20      0.0       1.46e3    1.46e3
   3.20      0.0       1.46e3    1.46e3
   3.20      0.0       1.46e3    1.46e3
   7.56      5.77e3    1.47e3    5.77e3
   7.56      5.77e3    1.47e3    5.77e3
   7.56      5.77e3    1.47e3    5.77e3
   7.56      5.77e3    1.47e3    5.77e3
```

-FOR THE CONNECTIONS (including suspensions) Identify the following parameters,
-Number of connections:

IALLC
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-Complete the following tables for each connection, identifying:
a name, in single quotes up to 20 characters long;
its position relative to the chosen datum in x, y, z inches;
the number of the body at each end, 0 for an earth in local track coords.;
a number indicating the degree(s) of freedom, translational 1,2,3 or rotational 4,5,6; in x,y,z resp., including 2 for lateral wheel motion;
the type 1 - parallel pair of spring and damper characteristics
2 - series pair of spring and damper characteristics
3 - device with hysteresis between 2 PWL characteristics, e.g. carriage spring or load sensitive suspension
4 - lateral/longitudinal suspension of the wheel on rail in the track plane
5 - connection force as a history of the distance moved
and the identification number for each of type 1, 2 and 3;
axle number for type 4; input function number for type 5.

Note - single characteristics are treated as parallel pairs with the missing characteristic set to zero in the subsequent table.

-Complete for all connections in turn,

Conn #	' 20 CHARACTER NAME '	Posn in X,	Y	& Z	Body1	Body2	DoF.	Type	Number
1	'Ld Bols-Bod CB Vt'	-35.0	0.0	25.0	1	2	3	1	1
2	'Ld Bols-Bod CB Roll'	-35.0	0.0	25.0	1	2	4	1	2
3	'Tr Bols-Bod CB Vt'	-521.0	0.0	25.0	1	3	3	1	1
4	'Tr Bols-Bod CB Roll'	-521.0	0.0	25.0	1	3	4	1	2
5	'Lead Bols-Bod CB Lat'	-35.0	0.0	25.0	1	2	2	1	3
6	'Trail Bols-Bod CB Lt'	-521.0	0.0	25.0	1	3	2	1	3
7	'Lead Bols-Bod CB Yaw'	-35.0	0.0	25.0	1	2	6	1	4
8	'Trl Bols-Bod CB Yaw '	-521.0	0.0	25.0	1	3	6	1	4
9	'Ld Bols-Sdfm Lt Vert'	-35.0	39.5	18.0	2	4	3	3	2
10	'Ld Bols-Sdfm Rt Vert'	-35.0	-39.5	18.0	2	5	3	3	3
11	'Tr Bols-Sdfm Lt Vert'	-521.0	39.5	18.0	3	6	3	3	1
12	'Tr Bols-Sdfm Rt Vert'	-521.0	-39.5	18.0	3	7	3	3	1
13	'Ld Bols-Sdfm Lt Lat '	-35.0	39.5	18.0	2	4	2	1	5
14	'Ld Bols-Sdfm Rt Lat '	-35.0	-39.5	18.0	2	5	2	1	5
15	'Tr Bols-Sdfm Lt Lat '	-521.0	39.5	18.0	3	6	2	1	5
16	'Tr Bols-Sdfm Rt Lat '	-521.0	-39.5	18.0	3	7	2	1	5
17	'Ld Bols-Sdfm Lt Yaw '	-35.0	39.5	18.0	2	4	6	1	6
18	'Ld Bols-Sdfm Rt Yaw '	-35.0	-39.5	18.0	2	5	6	1	6
19	'Tr Bols-Sdfm Lt Yaw '	-521.0	39.5	18.0	3	6	6	1	6
20	'Tr Bols-Sdfm Rt Yaw '	-521.0	-39.5	18.0	3	7	6	1	6
21	'Ld Bols-Sdfm Lt Long'	-35.0	39.5	18.0	2	4	1	1	7
22	'Ld Bols-Sdfm Rt Long'	-35.0	-39.5	18.0	2	5	1	1	7
23	'Tr Bols-Sdfm Lt Long'	-521.0	39.5	18.0	3	6	1	1	7
24	'Tr Bols-Sdfm Rt Long'	-521.0	-39.5	18.0	3	7	1	1	7
25	'Ax 1 Lt BA-Sdfm Long'	0.0	39.5	21.0	4	8	1	1	10
26	'Ax 1 Rt BA-Sdfm Long'	0.0	-39.5	21.0	5	8	1	1	11
27	'Ax 2 Lt BA-Sdfm Long'	-70.0	39.5	21.0	4	9	1	1	12
28	'Ax 2 Rt BA-Sdfm Long'	-70.0	-39.5	21.0	5	9	1	1	13
29	'Ax 3 Lt BA-Sdfm Long'	-486.0	39.5	21.0	6	10	1	1	9
30	'Ax 3 Rt BA-Sdfm Long'	-486.0	-39.5	21.0	7	10	1	1	9
31	'Ax 4 Lt BA-Sdfm Long'	-556.0	39.5	21.0	6	11	1	1	9
32	'Ax 4 Rt BA-Sdfm Long'	-556.0	-39.5	21.0	7	11	1	1	9
33	'Ax 1 Lt BA-Sdfm Lat '	0.0	39.5	21.0	4	8	2	1	9
34	'Ax 1 Rt BA-Sdfm Lat '	0.0	-39.5	21.0	5	8	2	1	9
35	'Ax 2 Lt BA-Sdfm Lat '	-70.0	39.5	21.0	4	9	2	1	9
36	'Ax 2 Rt BA-Sdfm Lat '	-70.0	-39.5	21.0	5	9	2	1	9
37	'Ax 3 Lt BA-Sdfm Lat '	-486.0	39.5	21.0	6	10	2	1	9
38	'Ax 3 Rt BA-Sdfm Lat '	-486.0	-39.5	21.0	7	10	2	1	9
39	'Ax 4 Lt BA-Sdfm Lat '	-556.0	39.5	21.0	6	11	2	1	9
40	'Ax 4 Rt BA-Sdfm Lat '	-556.0	-39.5	21.0	7	11	2	1	9
41	'Ax 1 Lt BA-Sdfm Vert'	0.0	39.5	21.0	4	8	3	1	1
42	'Ax 1 Rt BA-Sdfm Vert'	0.0	-39.5	21.0	5	8	3	1	1
43	'Ax 2 Lt BA-Sdfm Vert'	-70.0	39.5	21.0	4	9	3	1	1
44	'Ax 2 Rt BA-Sdfm Vert'	-70.0	-39.5	21.0	5	9	3	1	1
45	'Ax 3 Lt BA-Sdfm Vert'	-486.0	39.5	21.0	6	10	3	1	1

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46      'Ax 3 Rt BA-Sdfm Vert' -486.0 -39.5  21.0    7     10     3     1     1
47      'Ax 4 Lt BA-Sdfm Vert' -556.0  39.5  21.0    6     11     3     1     1
48      'Ax 4 Rt BA-Sdfm Vert' -556.0 -39.5  21.0    7     11     3     1     1
49      'Ax 1 Lt Whl/Rail Vrt'    0.0 29.75   0.0    8     12     3     1     8
50      'Ax 1 Rt Whl/Rail Vrt'    0.0 -29.75  0.0    8     13     3     1     8
51      'Ax 2 Lt Whl/Rail Vrt'  -70.0 29.75   0.0    9     14     3     1     8
52      'Ax 2 Rt Whl/Rail Vrt'  -70.0 -29.75  0.0    9     15     3     1     8
53      'Ax 3 Lt Whl/Rail Vrt' -486.0 29.75   0.0   10     16     3     1     8
54      'Ax 3 Rt Whl/Rail Vrt' -486.0 -29.75  0.0   10     17     3     1     8
55      'Ax 4 Lt Whl/Rail Vrt' -556.0 29.75   0.0   11     18     3     1     8
56      'Ax 4 Rt Whl/Rail Vrt' -556.0 -29.75  0.0   11     19     3     1     8
57      'Ax 1 Lt Whl/Rail lat'    0.0 29.75   0.0    8     12     2     4     1
58      'Ax 1 Rt Whl/Rail Lat'    0.0 -29.75  0.0    8     13     2     4     1
59      'Ax 2 Lt Whl/Rail Lat'  -70.0 29.75   0.0    9     14     2     4     2
60      'Ax 2 Rt Whl/Rail Lat'  -70.0 -29.75  0.0    9     15     2     4     2
61      'Ax 3 Lt Whl/Rail Lat' -486.0 29.75   0.0   10     16     2     4     3
62      'Ax 3 Rt Whl/Rail Lat' -486.0 -29.75  0.0   10     17     2     4     3
63      'Ax 4 Lt Whl/Rail Lat' -556.0 29.75   0.0   11     18     2     4     4
64      'Ax 4 Rt Whl/Rail Lat' -556.0 -29.75  0.0   11     19     2     4     4
-List for each pair of type 1 - parallel connections, its number, followed by
  the identification numbers of the piecewise linear characteristics
  for the stiffness and damping respectively, zero if absent, and
  the combined force or moment limit in extn & compn, lb or lb-in.,
  0.0 in extension at the vertical rail/wheel conn. allows valid wheel lift.
  (If no limit exists, set the F-values outside the expected range.)
     Pair #     Stiff PWL     Damp PWL        F-extn.       F-compn.
       1           1             2            0.0e8         -1.0e8
       2           3             4            1.0e8         -1.0e8
       3           5             6            1.0e8         -1.0e8
       4           7             8            1.0e8         -1.0e8
       5           9            10            1.0e8         -1.0e8
       6          11            12            1.0e8         -1.0e8
       7          13            14            1.0e8         -1.0e8
       8          15            16            0.0e8         -1.0e8
       9          19            20            1.0e8         -1.0e8
      10          21            20            1.0e8         -1.0e8
      11          22            20            1.0e8         -1.0e8
      12          23            20            1.0e8         -1.0e8
      13          24            20            1.0e8         -1.0e8
-List for each pair of type 2 - series connections, its number, followed by
  the identification numbers of the piecewise linear characteristics
  for the stiffness and damping respectively, and the stroke limit
  in extension & compression for the pair, in or rad, and the stiffness
  of the stop at the limit in lb/in or lb-in/rad.
  (If no limit exists, set the S-values outside the expected range.)
     Pair #     Stiff PWL    Damp PWL     S-extn.     S-compn.    Stop K
-List the type 3 - hysteresis loop characteristics, giving to each a number,
  identification numbers for the extension and compression PWLs, a linear
  viscous damping in lb-sec/in or lb-in-sec/rad, and extn/compn force limits.
     Loop #      Extn PWL     Comp PWL      LVB damping     F-extn     F-compn
       1           17            18           4.49e6        0.0e8      -1.0e8
       2           25            26           4.49e6        0.0e8      -1.0e8
       3           27            28           4.49e6        0.0e8      -1.0e8
-List the type 4 - axle to track characteristics, the general lateral rail
  stiffness and damping coefficient, and, for each axle, IAX, an identification
  number, IBDAX, its general body number, WRAD, the nominal wheel radius and
  INDWH, a wheel rotation index, 1 for solid, 2 for independent wheels, ITRQ,
  traction torque input nos. for left and right wheels, 0 for none, and, for
  independent wheels, KWHL, DWHL, the axle torsional stiffness and damping.
       Lateral Rail Stiffness lb/in       Lateral Rail Damping lb-sec/in
                 4.0e5                              4.0e3
    IAX        IBDAX      WRAD       INDWH      ITRQ-L     ITRQ-R     KWHL     DWHL
     1           8        18.0         1          0          0        0.0      0.0
     2           9        18.0         1          0          0        0.0      0.0
     3          10        18.0         1          0          0        0.0      0.0
     4          11        18.0         1          0          0        0.0      0.0
-How many different piecewise linear, (PWL), characteristics are required
     28
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-List the data required for the connection characteristics,
PWL, the piece-wise linear function no., IBP, the no. of Break Points in each
PWL, Ordinate, lb or lb-in, over abscissa, in or rad, at each Break Point
N.B. (1) Extension is assumed to be positive for both ordinate and abscissa
(2) 0.0 for the first break point indicates symmetry about the origin

PWL	IBP	Ordinates over Abscissae					
1	2	0.0	1.0e6				
		0.0	1.0				
2	2	0.0	1.0e3				
		0.0	1.0				
3	5	0.0	9.77e5	9.77e5	3.054e6	3.054e6	
		0.0	0.0135	0.0263	0.0323	1.0	
4	3	0.0	2.0e5	2.0e5			
		0.0	0.1e-3	1.0			
5	2	0.0	1.0e6				
		0.0	1.0				
6	2	0.0	1.0e3				
		0.0	1.0				
7	2	0.0	2.0e4				
		0.0	1.0				
8	3	0.0	1.985e5	1.9850e5			
		0.0	0.002	1.0			
9	3	0.0	3.0e3	3.0e4			
		0.0	0.45	0.55			
10	3	0.0	4.0e3	4.0e3			
		0.0	0.1	1.0			
11	4	0.0	1.4e5	8.1e5	1.698e6		
		0.0	0.01275	0.057	0.058		
12	3	0.0	100.0e3	100.0e3			
		0.0	0.05	1.0			
13	2	0.0	1.0e6				
		0.0	1.0				
14	2	0.0	1.0e3				
		0.0	1.0				
15	2	0.0	1.0e5				
		0.0	1.0				
16	2	0.0	1.0e3				
		0.0	1.0				
17	5	-1.25e5	-1.12e5	-9.06e4	-3.76e4	-7.52e3	
		-4.0542	-3.9684	-3.9387	-1.7753	-0.2212	
18	5	-1.30e5	-1.11e5	-1.00e5	-4.02e4	-1.15e4	
		-4.0542	-3.8853	-3.5914	-1.4563	-0.2212	
19	3	0.0	4080.0	8241.0			
		0.0	0.1071	0.1204			
20	3	0.0	650.0	642.5			
		0.0	0.1	1.0			
21	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.0496	-0.0363	0.0708	0.1779	0.1912	
22	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.1912	-0.1779	-0.0708	0.0363	0.0496	
23	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.0407	-0.0274	0.0797	0.1868	0.2001	
24	5	-8241.0	-4080.0	0.0	4080.0	8241.0	
		-0.2001	-0.1868	-0.0797	0.0274	0.0407	
25	6	-1.25e5	-9.81e4	-9.30e4	-7.67e4	-4.36e4	-7.97e3
		-4.07	-4.00	-3.93	-3.33	-1.89	-0.30
26	5	-1.30e5	-1.13e5	-9.13e4	-5.28e4	-1.26e4	
		-4.07	-3.94	-3.15	-1.91	-0.30	
27	5	-1.25e5	-9.36e4	-7.40e4	-4.18e4	-6.8e3	
		-4.05	-3.98	-3.35	-2.12	-0.19	
28	6	-1.30e5	-1.07e5	-8.88e4	-7.48e4	-5.19e4	-1.12e4
		-4.05	-4.01	-3.55	-3.23	-2.05	-0.19

APPENDIX C

INSTRUMENTATION LISTS FOR TRACK TESTS

Table 1. Instrumentation Summary

Channel #	Description	Name	Transducer Type	Range	Resolution
55	Automatic Location Device	ALD1		+1-5V	1V
54	Consist Speed	SPD1	Tach	0-100 mph	.1 mph
	A-Truck Leading Instrumented Wheelset				
0	Left Wheel Vertical Force	4V1A			x
1	Left Wheel Vertical Force	4V2A	I.W.	0-100 kips	100 lb
2	Left Wheel Vertical Force	4V3A			
3	Left Wheel Lateral Force	4L1A	I.W.	+/- 50 kips	
4	Left Wheel Lateral Force	4L2A			
5	Left Wheel Position	4P0A			
6	Right Wheel Vertical Force	4V1B	I.W.	0-100 kips	100 lb
7	Right Wheel Vertical Force	4V2B			
8	Right Wheel Vertical Force	4V3B			
9	Right Wheel Lateral Force	4V1B	I.W.	+/- 50 kips	50 lb
10	Right Wheel Lateral Force	4L2B			
11	Right Wheel Position	4P0A			
12	Longitudinal Force	4L0N	I.W.	+/- 20 kips	50 lb
	A-Truck Trailing Instrumented Wheelset				
13	Left Wheel Vertical Force	5V1A	I.W.		
14	Left Wheel Vertical Force	5V2A			
15	Left Wheel Vertical Force	5V3A			
16	Left Wheel Lateral Force	5L1A	I.W.	+/- 50 kips	
17	Left Wheel Lateral Force	5L2A			
18	Left Wheel Position	5P0A			
19	Right Wheel Vertical Force	5V1B	I.W.	0-100 kips	100 lb
20	Right Wheel Vertical Force	5V2B			
21	Right Wheel Vertical Force	5V3B			
22	Right Wheel Lateral Force	5L1A	I.W.	+/- 50 kips	50 lb
23	Right Wheel Lateral Force	5L1B			
24	Right Wheel Position	5P0B			
25	Longitudinal Force	5L0N	I.W.	+/- 20 kips	50 lb
	Displacements and Accelerations				
26	A-Truck Left Spring Displacement	LDZ1	Stringpot	+/- 3 in	0.01
27	A-Truck Right Spring Displacement	RDZ2	"	"	"
50	B-Truck Left Spring Displacement	LDZ3	"	"	"
49	B-Truck Right Spring Displacement	RDZ4	"	"	"
28	A-Truck Left Carbody to Truck Bolster	LDZ5	Stringpot	+/- 3 in	0.01
29	A-Truck Left Carbody to Truck Bolster	RDZ6	"	"	"
32	Lead Left Side Frame to Bearing Adapter Displacement	LDY1	LVDT	+/- .25 in	0.001

33	Lead Right Side Frame to Bearing Adapter Displacement	RDY1	"	"	"
34	Lead Left Side Frame to Bearing Adapter Displacement	LDY2	"	"	"
35	Lead Right Side Frame to Bearing Adapter Displacement	RDY2	"	"	"
36	A-Truck (+) Bolster to Carbody Displacement	1YD5	Stringpot	+/- 5 in	0.01
37	A-Truck (-) Bolster to Carbody Displacement	2YD6	"	"	"
38	A-Truck (+) Left Side Frame to Bolster	3YD7	Stringpot	+/- 3 in	0.01
39	A-Truck (-) Left Side Frame to Bolster	4YD8	"	"	"
40	A-Truck (+) Right Side Frame to Bolster	5YD9	"	"	"
41	A-Truck (-) Right Side Frame to Bolster	YD10	"	"	"
42	Lead Left Side Frame to Bearing Adapter Displacement	LDX1	LVDT	+/- .25 in	0.001
43	Lead Right Side Frame to Bearing Adapter Displacement	RDX2	"	"	"
44	Trail Left Side Frame to Bearing Adapter Displacement	LDX3	"	"	"
45	Trail Right Side Frame to Bearing Adapter Displacement	RDX4	"	"	"
46	A-End Lateral Carbody Accelerometer Lower Left Corner Side Sill	AY1	Accel	+/- 5 g	0.01
48	B-End Lateral Carbody Accelerometer Lower Right Corner Side Sill	AY3	Accel	+/- 5 g	0.01
51	A-End Lead Axle Left Side Lateral Bearing Adapter Accelerometer	AY2	"	"	"
47	A-End Trail Axle Left Side Lateral Bearing Adapter Accelerometer	AY3	"	"	"
52	B-End Lead Axle Left Side Lateral Bearing Adapter Accelerometer	AY5	"	"	"
53	B-End Trail Axle Left Side Lateral Bearing Adapter Accelerometer	AY6	"	"	"
30	A-End Roll Rate Gyro	RG1	Gyro	+/- 8 deg	0.01
31	B-End Roll Rate Gyro	"	"	"	

TOTALS: 55 Recorded Data Channels

NOTES: I.W. - Instrumented Wheelset

Data Channel # refers to order as recorded by HP data collection system.