Safety Aspects of New Trucks and Lightweight Cars, Car 2

Interim Report

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16. Abstract

The Federal Railroad Administration (FRA) has sponsored a program to continue the validation of new techniques for testing and analysis that could be applied to the evaluation of the safety and track worthiness aspects of new freight car and suspension designs. The total program involves laboratory and on-track testing, and simulation of tests using a computer model. In partnership with the FRA, the Association of American Railroads (AAR) funded the development and first tests of a new suspension characterization facility for performing the laboratory tests. This report documents the results of the laboratory tests and the preliminary computer modeling performed prior to the on-track tests.

The laboratory tests consisted of performing the suspension characterizations using the newly developed facility. The test results were used as input to a computer model of the test vehicle.

The computer model was used to predict, prior to the on-track tests, the dynamic behavior of the test vehicle. Simulations were run for all the track test sections required by AAR Specification M-1001, Chapter XI. The vehicle was simulated in both empty and loaded condition.

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EXECUTIVE SUMMARY

Recent research efforts sponsored by the Federal Railroad Administration (FRA) have investigated the development and validation of new techniques for testing and analysis of the safety and track worthiness performance of new designs of freight cars and suspensions. At the same time the Association of American Railroads (AAR) has recently introduced new service worthiness test requirements as a part of the AAR's *Manual of Standards and Recommended Practices,* M-1001, Chapter XI.

Prior to the acceptance of Chapter XI, the FRA and AAR jointly sponsored a research program to develop new tests and safety criteria for the evaluation of the dynamic performance of new vehicles. The tests and criteria used in this program were the same as those now required by Chapter XI. This program tested a newly designed 2-axle, skeletal platform car for carrying a single highway trailer. This project became known as Car 1. To continue the development process, the FRA and AAR have sponsored a second test program to apply similar testing and analysis techniques to a different, newly designed freight car. In this project the AAR's contribution has been to develop a new facility for performing some of the required laboratory tests. The AAR also sponsored the first tests conducted using this facility. The objectives of the entire program were as follows:

- Determine whether relatively inexpensive procedures could be devised which could be used for the analysis and testing of the safety aspects of new designs of lightweight cars and trucks
- Evaluate the safety aspects of a new design of lightweight car and truck using these procedures

The car chosen for this second test (Car 2) was an aluminum bodied, coal gondola, PSMX 111, which was equipped with modified three-piece trucks. The modifications included primary shear pads at the bearing adaptors, and redesigned friction snubbers to increase the truck warp stiffness.

To meet the objectives entailed three major sub-tasks:

- Laboratory Tests
- On-track Tests
- Mathematical Modeling

This interim report details the results of the laboratory tests and the portion of mathematical modeling performed prior to the on-track tests.

Laboratory Tests

These tests were conducted to provide input data for the mathematical model. This input data includes suspension stiffnesses and damping, and rigid and flexible body modal characteristics, such as resonant frequencies and structural damping. For the Car 1 project, most of these tests were conducted on a machine known as the Vibration Test Unit (VTU), at the Transportation Test Center (TTC), Pueblo, Colorado. The VTU tests proved to be complicated and very expensive to run and provided lower quality than was desired. Therefore, the AAR sponsored the development of a simpler facility for conducting characterization tests.

This new facility, the Mini-Shaker Unit (MSU), utilizes two vertical and one lateral actuator which connect the car body to the ground. One end of the car is supported on four strain-gaged bars for measuring suspension loads. The MSU is much more effective in exciting the desired motions for the characterization tests.

A few tests that required characterization of the yaw and longitudinal suspensions, required floating the car on air bearings and using manual hydraulic actuators to move the suspensions. These tests were conducted in an identical manner to the Car 1 tests.

The following laboratory tests were successfully completed:

- Vertical and Lateral Suspension Characterization (MSU)
- Rigid and Flexible Body Modal Characterizations (MSU)
- Yaw and Longitudinal Suspension Characterization (Air Tables)

Mathematical Modeling

Using the parameters measured during the laboratory tests, simulations of the test car were made using the AAR's New and Untried Car Analytic Regime Simulation (NUCARS) computer model. NUCARS was used to simulate the car in loaded and empty condition over all the required Chapter XI test zones to predict the following dynamic behavior:

- Lateral stability on tangent track
- Constant curving and spiral negotiation
- Response to varying cross level (twist/roll)
- Response to surface variation (pitch/bounce)
- Response to alignment variation (yaw/sway)
- Response to alignment, gage and cross level variation in curves (dynamic curving)

These test zones are simulated using mathematical definitions of the track curves, and perturbations.

Successful predictions were made for all conditions except the yaw/sway test zone and the dynamic curve. Results from the simulations of these two zones appeared erroneous and did not predict behavior that was consistent with previous experience. It is suspected that errors in the definition of the yaw and lateral suspension characteristics may have affected these results.

Exceedance of Chapter XI limiting criteria was predicted for the following test regimes:

1. Empty Car Tangent Hunting
2. Empty Car Pitch and Bounce
3. Empty Car Single Bounce
4. Empty and Loaded Car Yaw and Sway (possibly erroneous results)
5. Empty Car Dynamic Curving (possibly erroneous results)

Future Modeling Efforts

Comparison of these predictions with track test results awaits analysis of the track test data. Additional NUCARS modeling will also be performed using actual measured track geometry for input. Refinements to some of the suspension characteristics will also be made based on suspension dynamic response during the track tests.

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1.0 INTRODUCTION

Recently there has been a significant increase in the rate of introduction of new designs of freight cars and trucks to the railroad industry. This has occurred for a number of reasons including the railroad industry's desire to carry greater loads at higher speeds 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 light-weight 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 1985 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 1988 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 the Appendix.

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 Trailer Train TTUX 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 NUCARS 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.

Due to unforeseen difficulties in analyzing the on-track test data, completion of the project has been delayed. This document is an interim report, describing the results of the vehicle characterization tests and the results of the pretest computer modeling.

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 pretest 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 tests were performed on the Vibration Test Unit (VTU) at the Transportation Test Center (TTC), Pueblo, Colorado, to measure most of these characteristics. 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 PRETEST ANALYSIS

As with Car 1, the pretest 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. Due to difficulties in analyzing this data, no track test results will be presented in this interim report.

3.5 POST TEST ANALYSIS

Because completion of these predictions depends on track test data, post test predictions will be presented in this interim report. The final report will contain the complete track test results compared with these post test "real track" NUCARS predictions. 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 is to be used for Car 2.

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 will be 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 will be used in the post test model predictions.

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).

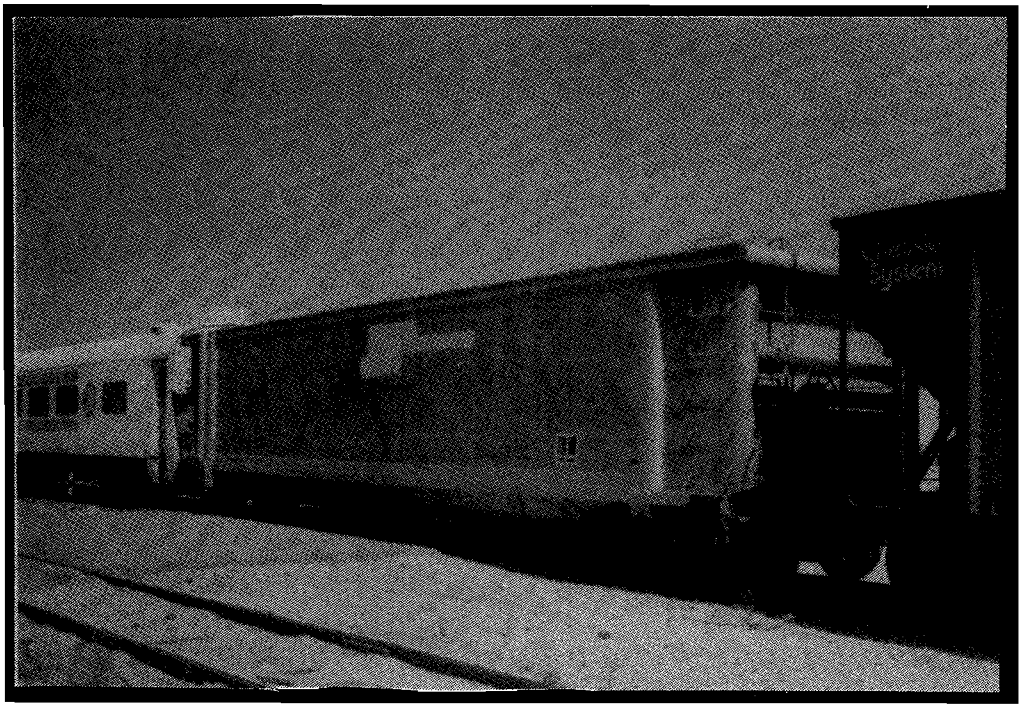

Figure 1. Test Vehicle: PSMX III Aluminum Coal Gondola

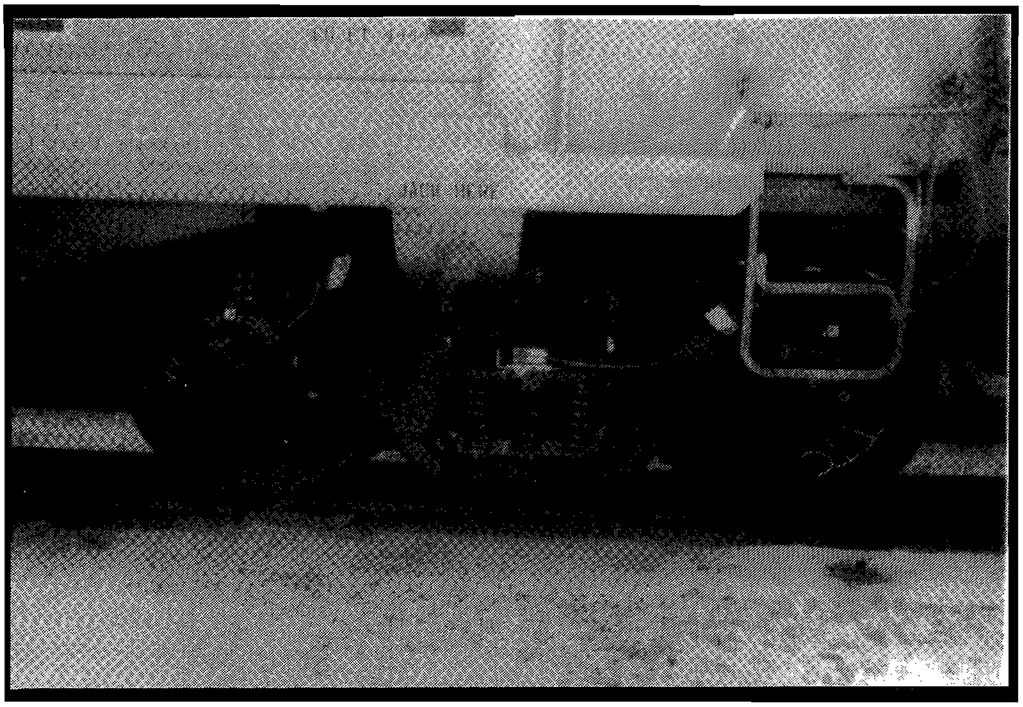

Figure 2. Test Truck: ASF Roadmaster Truck with Test Transducers Installed

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 very 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 come to be 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 desktop 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 crosstalk between vertical and lateral signals. These problems were confirmed by simple tests in the laboratory.

The second designs tested were strain gaged rail sections that had been developed by ENSCO under contract to the FRA. These were originally designed to be used as part of a sticking brake detector. These also proved to have considerably more crosstalk 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 crosstalk between vertical and lateral stains 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 crosstalk than any previous design.

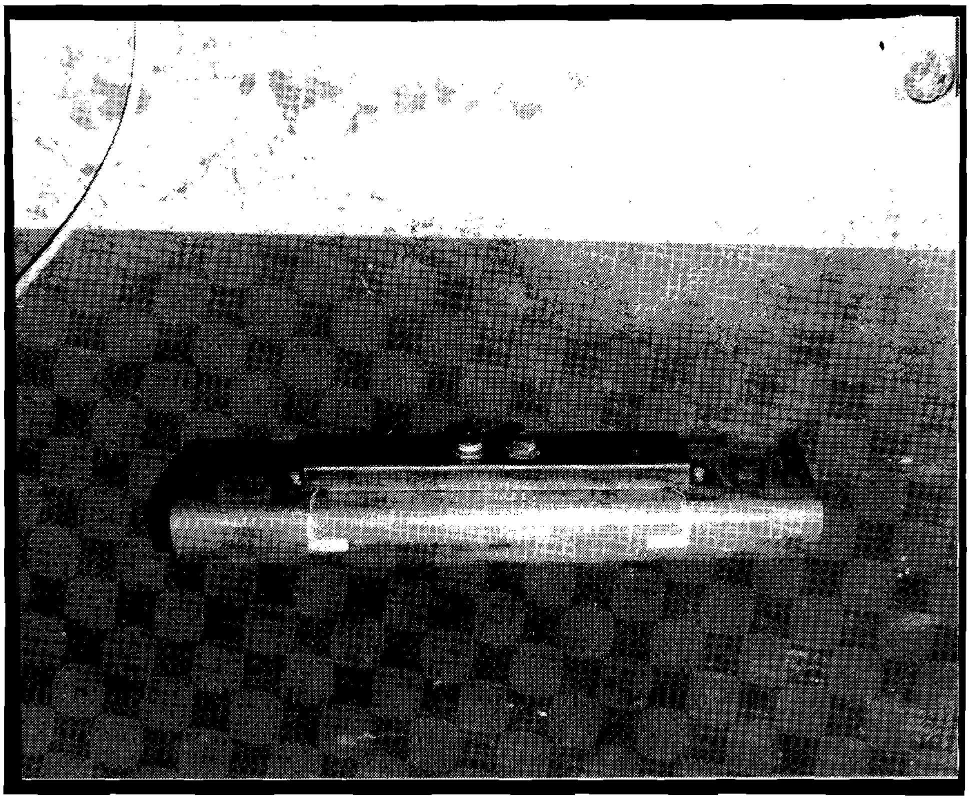

Figure 3. MSU Instrumented Rail, Showing Position of Strain Gages in Pockets on Top of Rail

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. Note that the VTU requires lifting the vehicle with cranes.

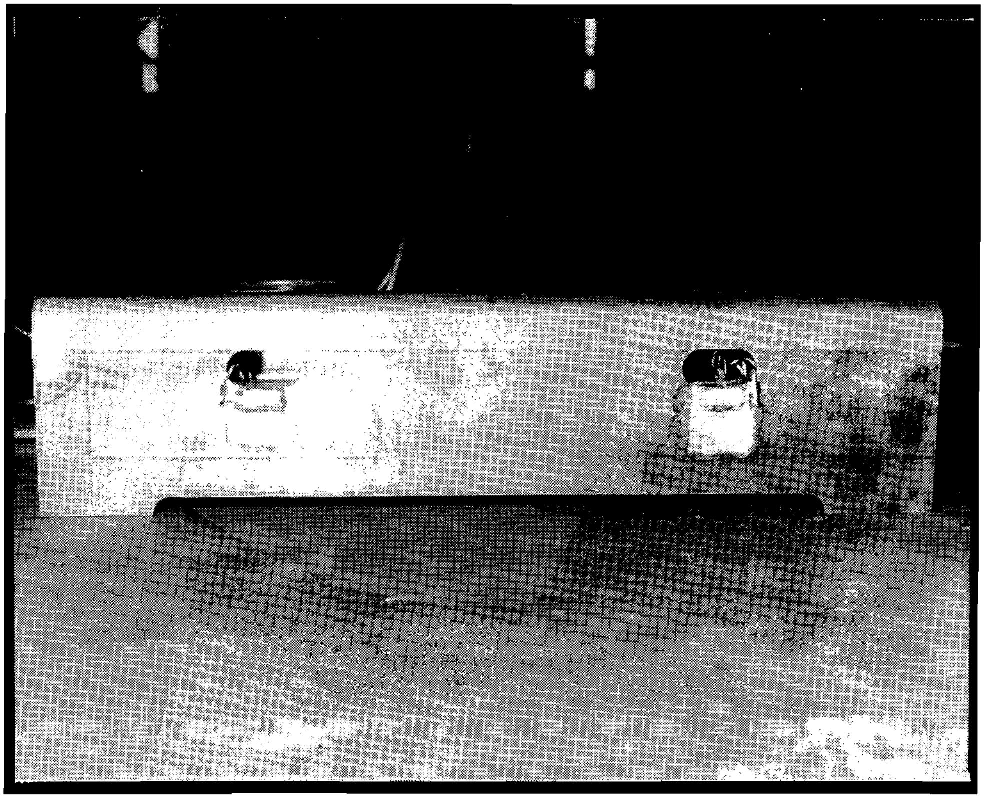

Figure 4. MSU Instrumented Rail Showing Holes for Strain Gages Machined in the Side

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, these 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.

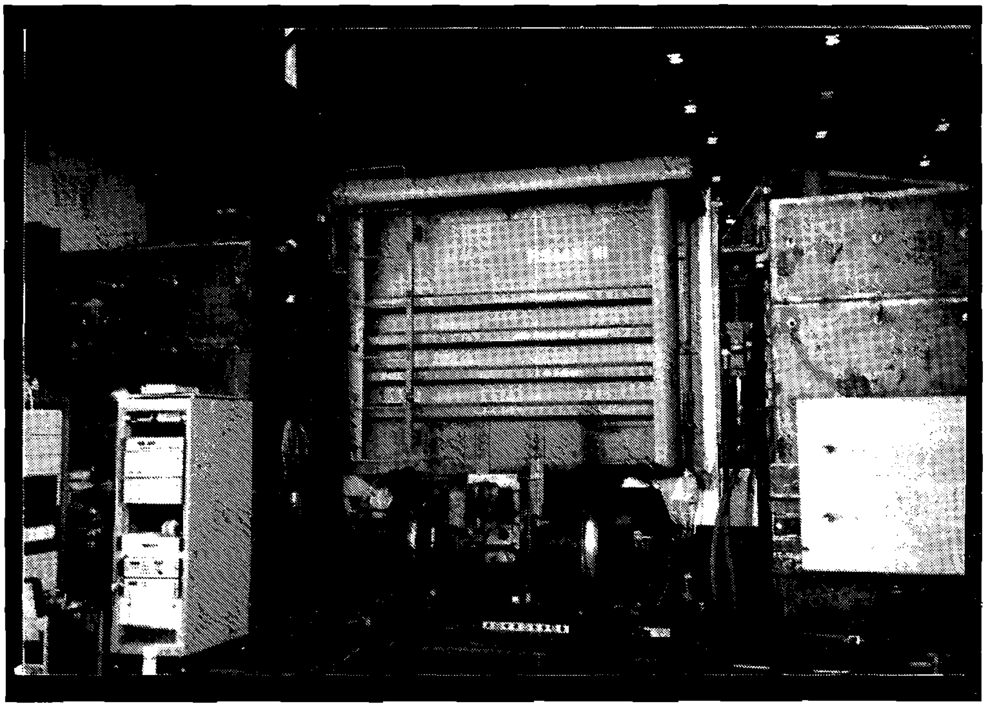

Figure 5. Test Vehicle Installed in MSU Showing Attachment of Hydraulic Actuators

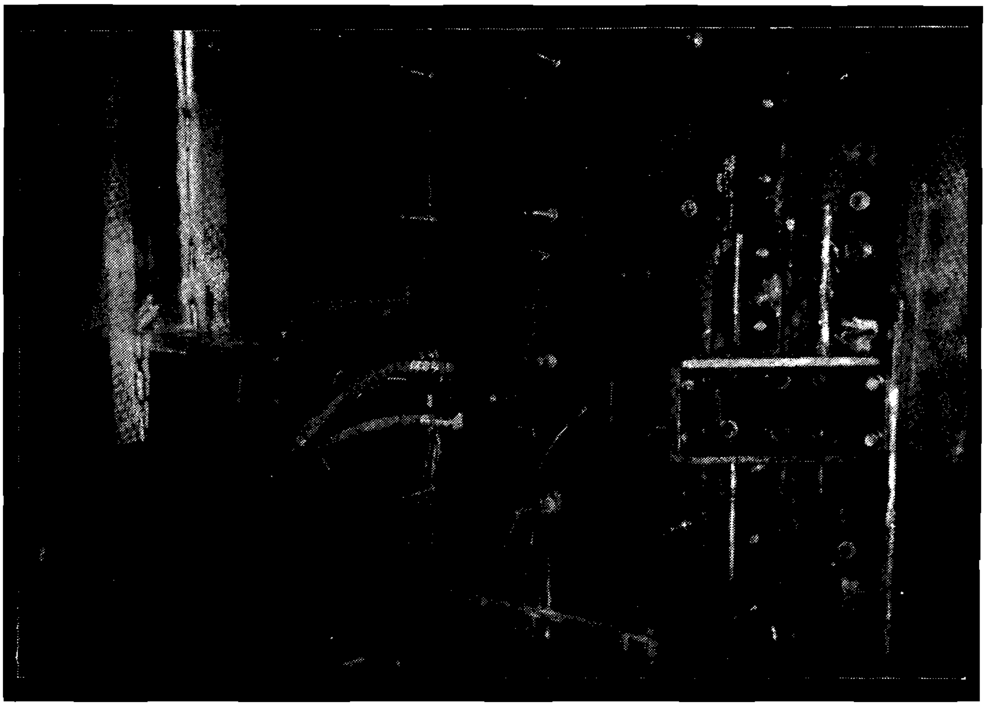

Figure 6. Attachment of MSU Lateral Actuator to Concrete Block and Test Vehicle

5.2.5 Control System And Data Acquisition

Control of the actuators was accomplished using a Hewlett-Packard (HP) 330 desktop 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.

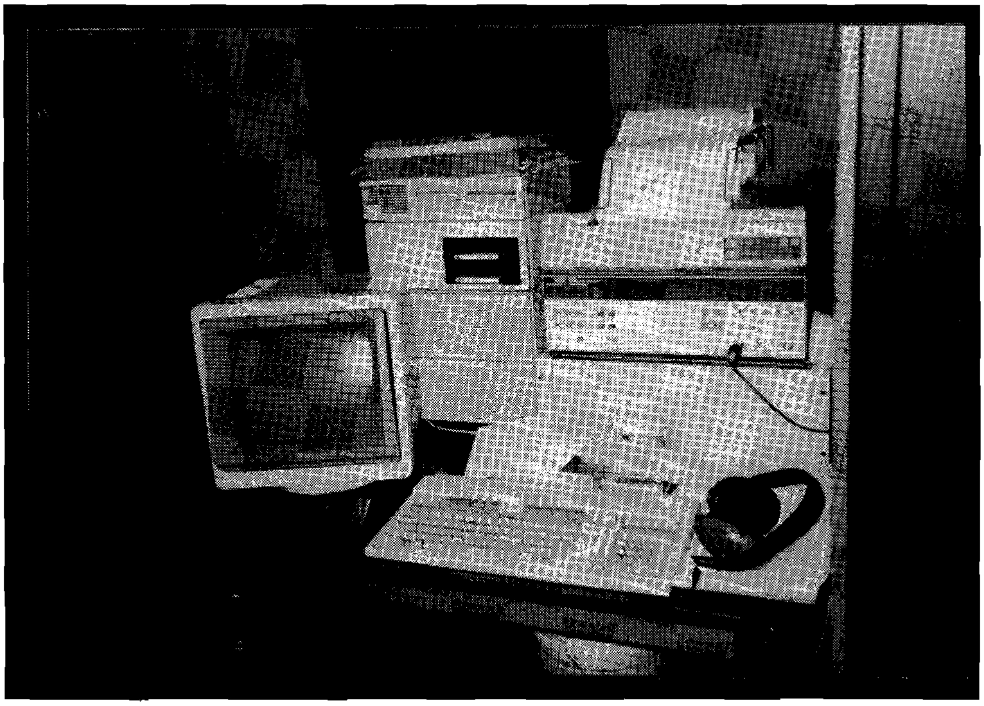

Figure 7. Hewlett-Packard Desktop Computer System Used for MSU Control and Data Acquisition

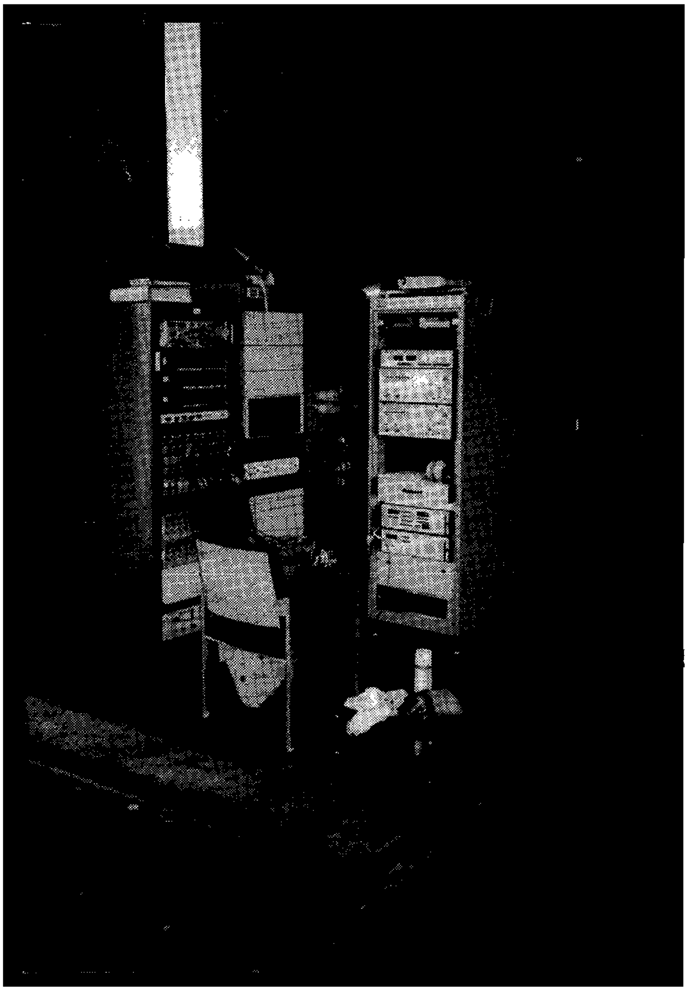

Figure 8. Signal Conditioning and MTS Hydraulic Control System for MSU

5.3 VEHICLE CHARACTERIZATION TESTS

5.3.1 Test Objectives

The objectives of performing vehicle characterization tests are:

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 five 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.

To determine the characteristics of this suspension, data from quasi-static test runs was 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 was 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.

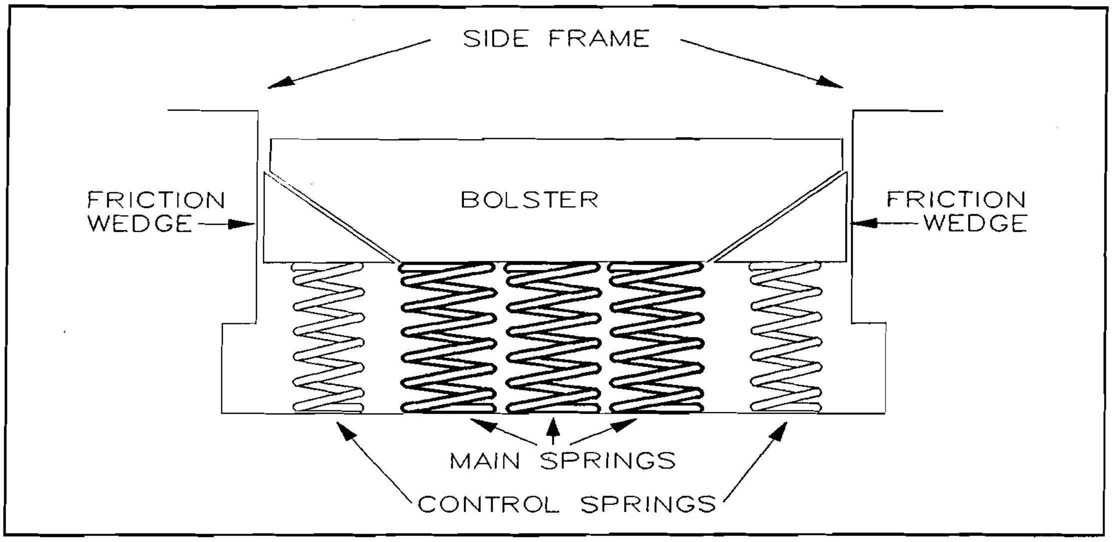

Figure 9. Diagram of Secondary Vertical Suspension (Bolster to Side Frame Connection)

Figures 10 to 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.

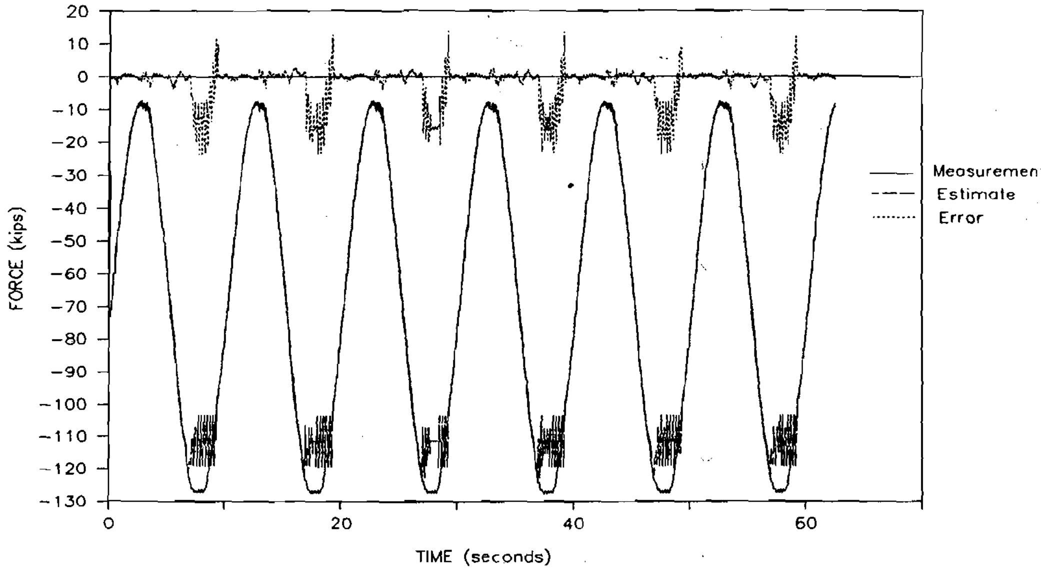

Figure 10. Vertical Force Time History of the Left Side Secondary Vertical Suspension

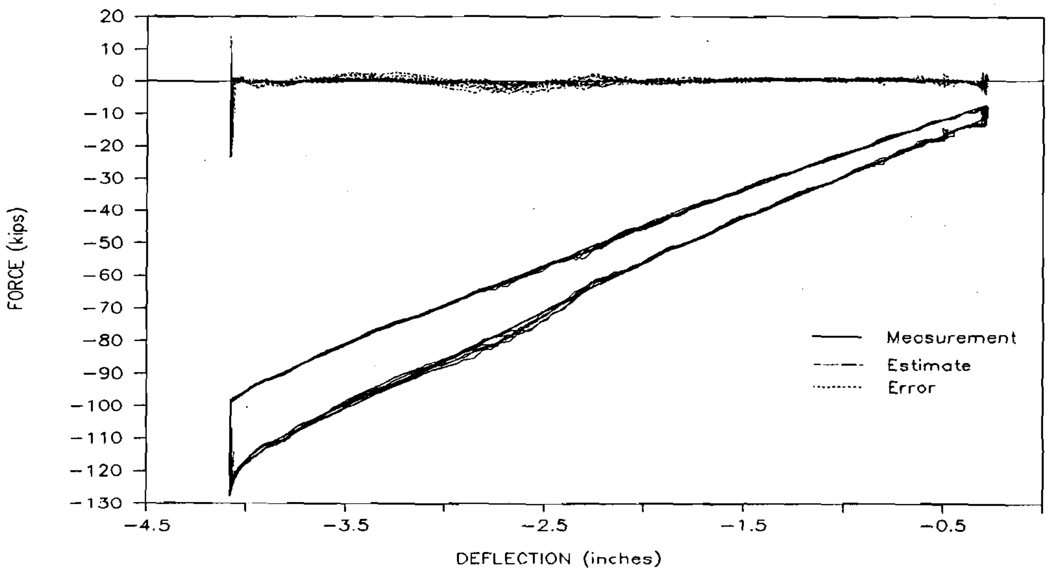

Figure 11. Vertical Force Versus Vertical Displacement Characteristic of the Left Side Secondary Vertical Suspension

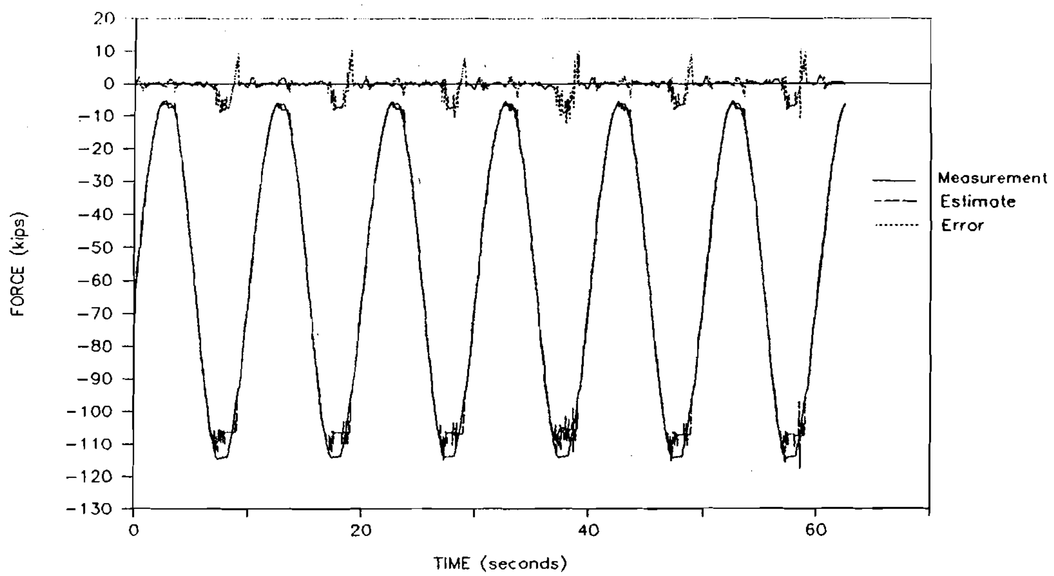

Figure 12. Vertical Force Time History of the Right Side Secondary Vertical Suspension

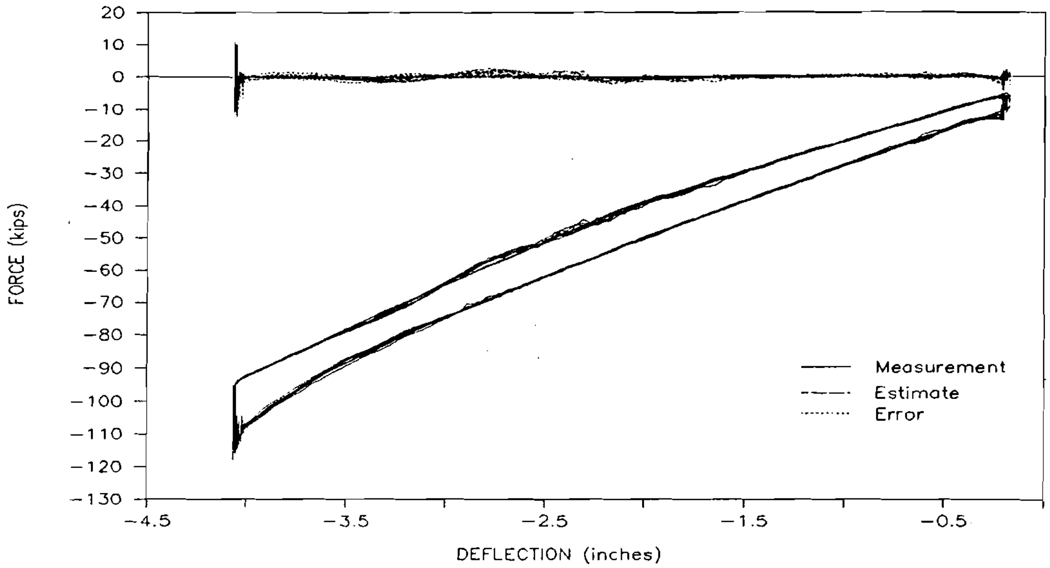

Figure 13. Vertical Force Versus Vertical Displacement Characteristic of the Right Side Secondary Vertical Suspension

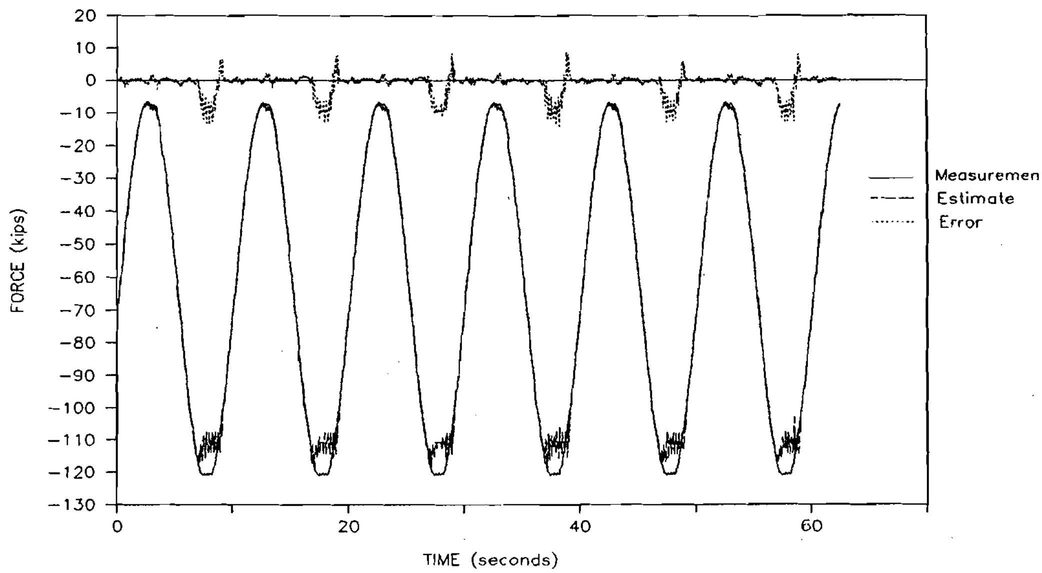

Figure 14. Vertical Force Time History of the "Average" Secondary Vertical Suspension

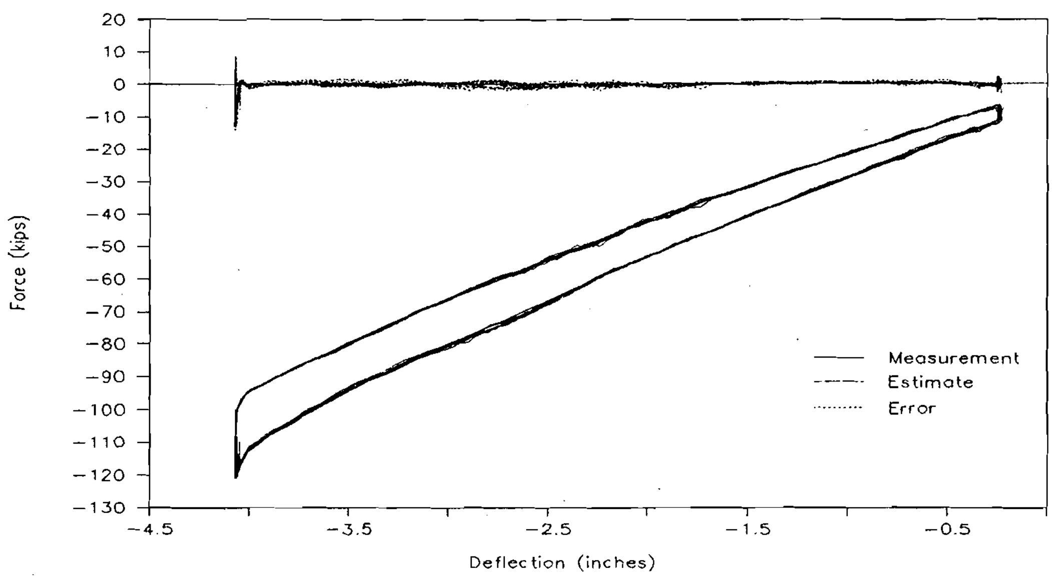

Figure 15. Vertical Force Versus Vertical Displacement Characteristic of the "Average" Secondary Vertical Suspension

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.

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 pretest modeling. The actual left and right characteristics will be substituted for the lead truck during the post test modeling using the actual track geometries (ENSCO measured track).

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	-1.25×10^{5}	-4.05		-1.3×10^{5}	-4.05	
	-9.36×10^{4}	-3.98		-1.07×10^{5}	-4.01	
	-7.4×10^{4}	-3.35	5	-8.88×0^{4}	-3.55	6
	-4.18×10^{4}	-2.12		-7.98×10^{4}	-3.23	
	-6.8×10^{3}	-0.19		-5.19×10^{4}	-2.05	
				-1.12×10^{4}	-0.19	
Left	-1.25×10^{5}	-4.07		-1.3×10^{5}	-4.07	
	-9.81×10^{4}	-4.00		-1.13×10^{5}	-3.94	
	-9.30×10^{4}	-3.93	6	-9.13×10^{4}	-3.15	5
	-7.67×10^{4}	-3.33		-5.28×10^{4}	-1.91	
	-4.36×10^{4}	-1.89		-1.26×10^{4}	-0.3	
	-7.97×10^{3}	-0.3				
Average	-1.25×10^{5}	-4.05		-1.3×10^{5}	-4.05	
	-1.12×10^{5}	-3.97		-1.11×10^{5}	-3.89	
	-9.06×10^{4}	-3.94	5	-1.0×10^{5}	-3.59	5
	-3.76×10^{4}	-1.77		-4.02×10^{4}	-1.46	
	-7.52×10^{3}	-0.22		-1.15×10^{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.

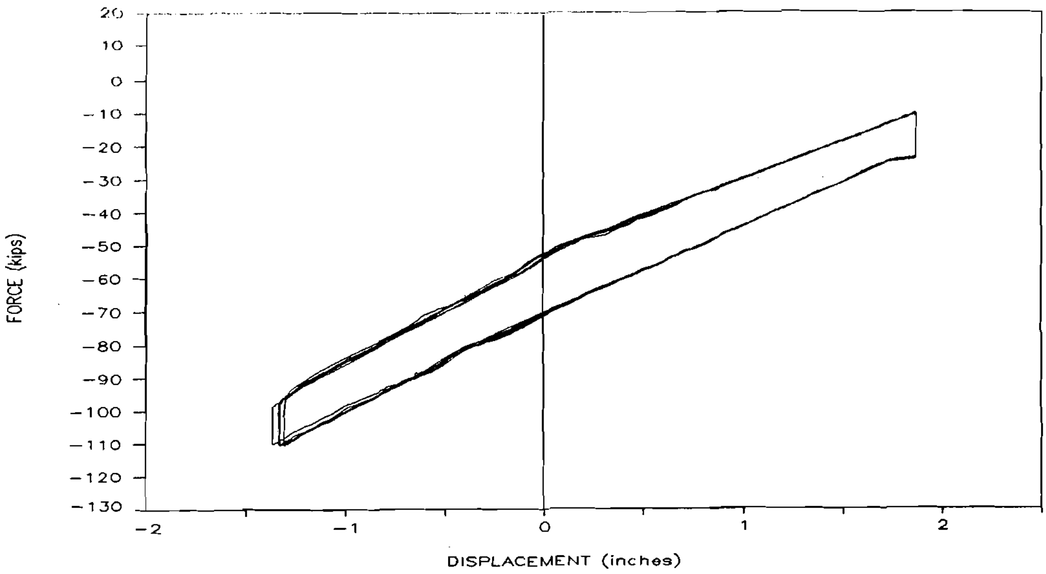

Figure 16. Vertical Force Versus Vertical Deflection Characteristic for the Right Side Secondary Vertical Suspension During Roll Input

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 effect 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.

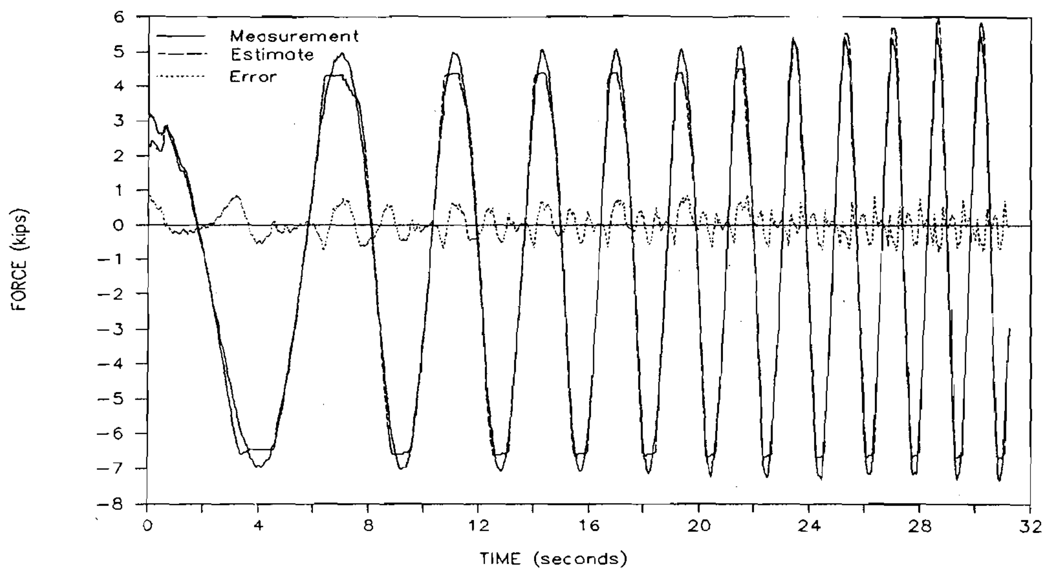

Figure 17. Lateral Force Time History of the Average Primary Lateral Shear Pad Suspension

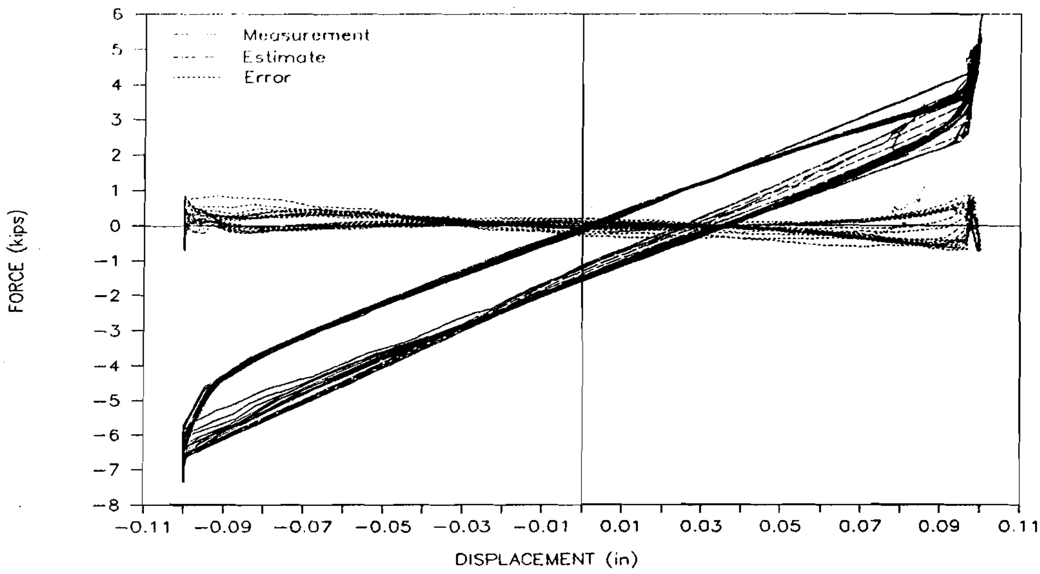

Figure 18. Lateral Force Versus Lateral Deflection Characteristic of the Average Primary Lateral Shear Pad Suspension

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 pretest 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. Therefore, for the purpose of pretest modeling, only half the friction level shown was used. Also this data is only valid for a loaded vehicle. The characteristic is expected to change for the empty car. Therefore, the loaded car values were halved for pretest modeling of the empty car. This lower value is based on the expectation that the variable friction dampers would reduce the lateral suspension friction damping when the car was empty.

During the track tests, transducers were mounted to measure lateral deflections of the secondary suspension. It is hoped that when plotted against the measured lateral wheel forces this data will allow the development of dynamic lateral suspension characteristics. This data will then be used for comparison with the MSU test data and to refine the NUCARS model.

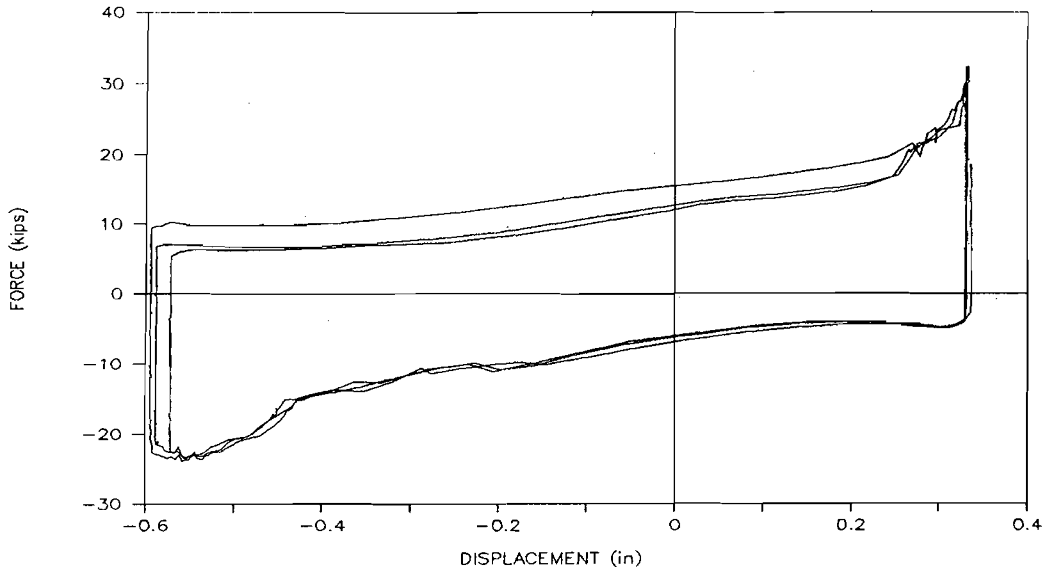

Figure 19. Secondary Lateral Suspension Lateral Force Versus Lateral Deflection Characteristic

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 pretest 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 is overlaid with the theoretical characteristic used in the pretest NUCARS modeling. It is clear that they are completely different.

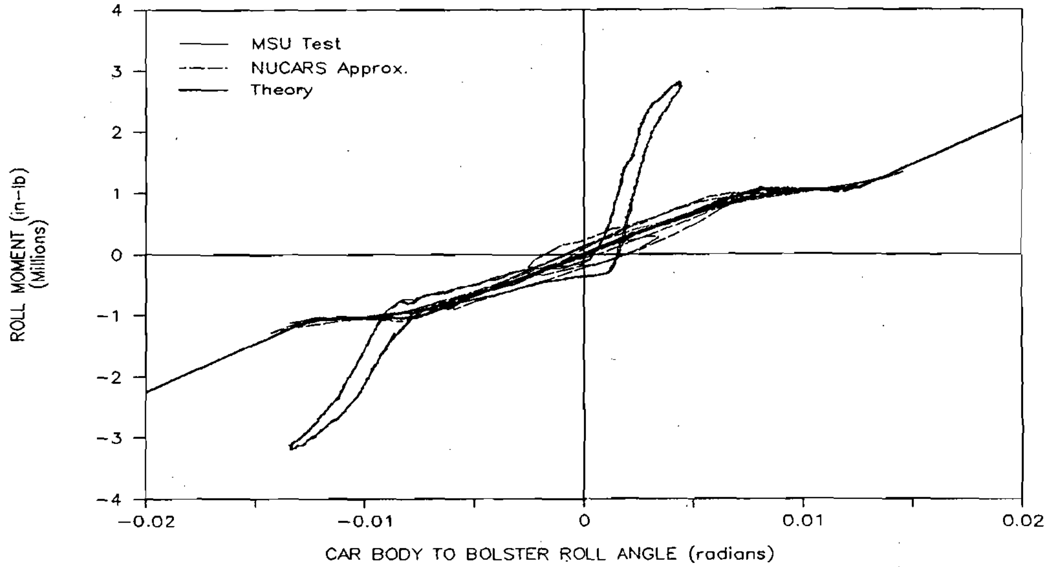

Figure 20. Roll Moment Versus Roll Angle Characteristic of the Secondary Roll Suspension

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 attempt will be made to estimate the roll characteristic from the MSU data 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 desktop 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.

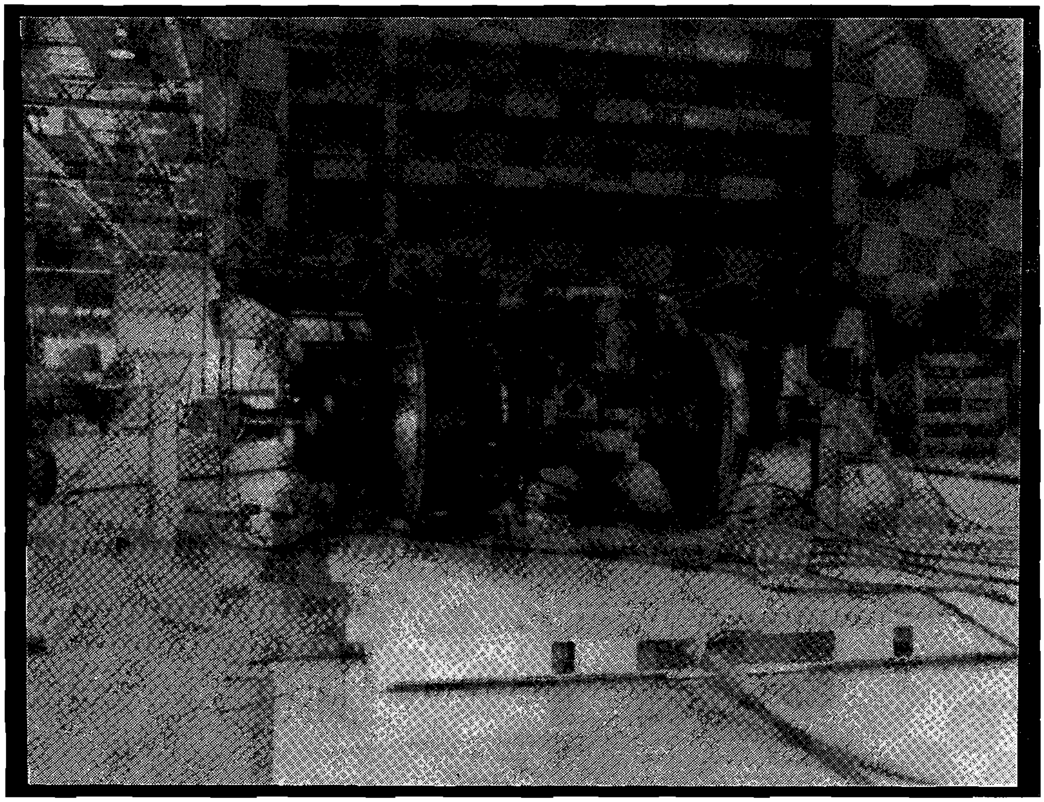

Figure 21. PSMX III Test Vehicle in Position on Air Tables for Inter-axle Shear Tests

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 applied which is required to allow the truck to rotate 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	17.9×10^3
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.

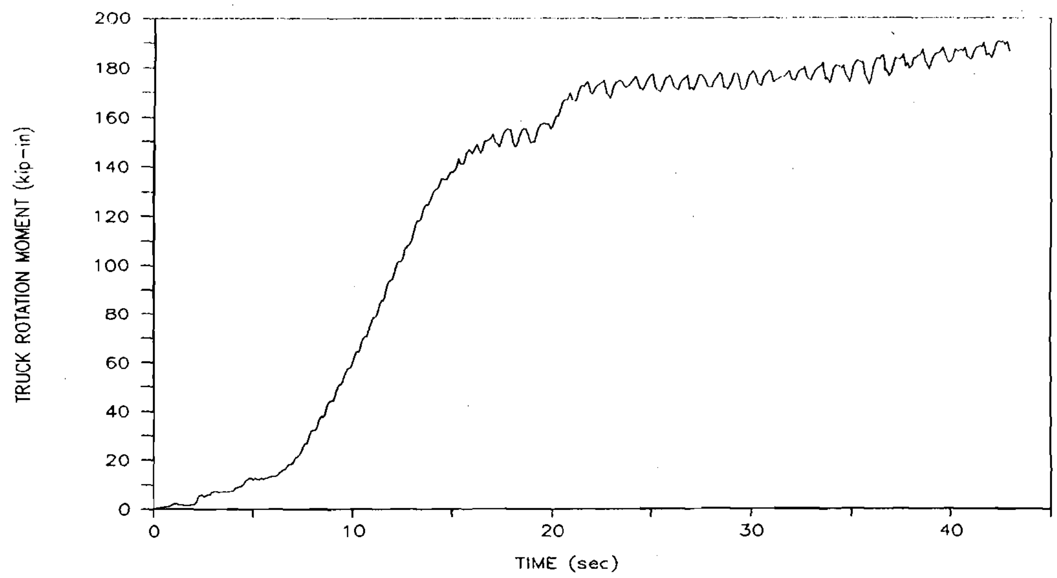

Figure 22. Example Time History of Truck Rotation Moment During Truck Rotation Test

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. No explanation is available to account for this difference. For the purposes of the pretest 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.

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.

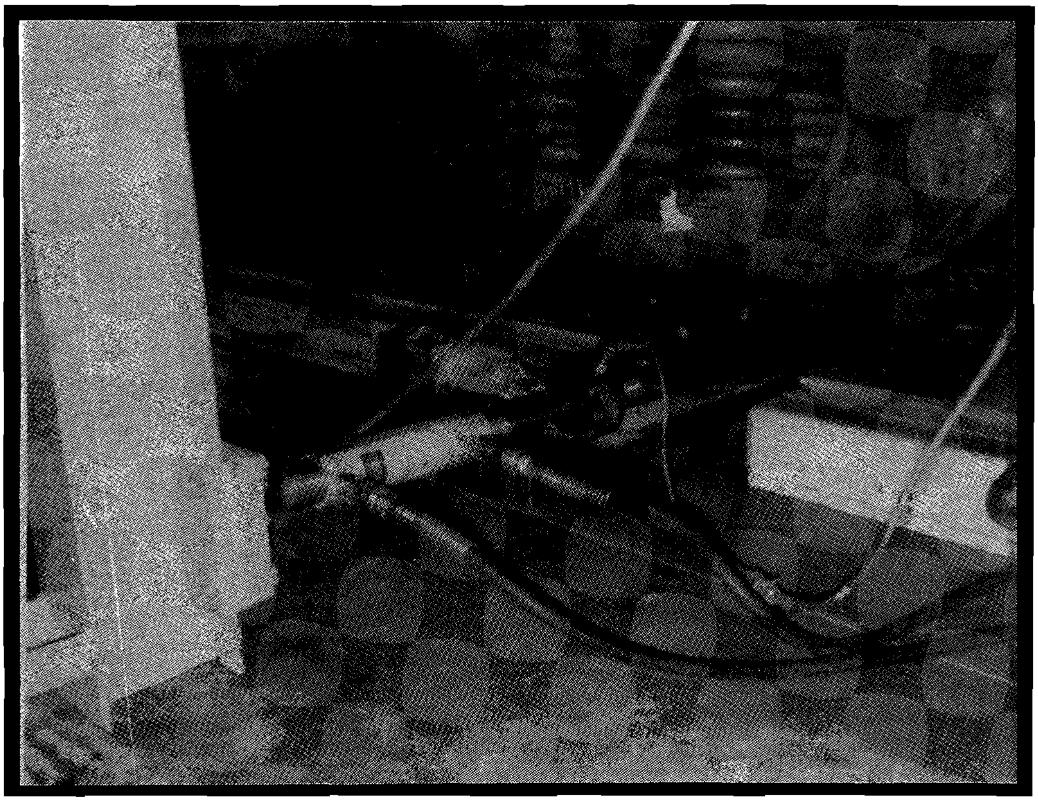

Figure 23. Lateral Actuator Position for Inter-axle Shear Tests

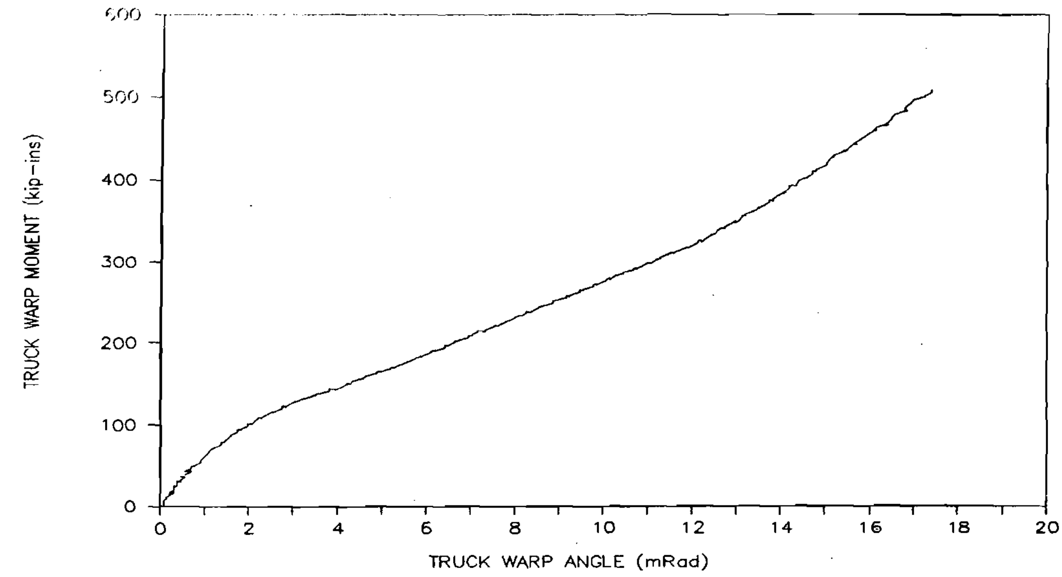

Figure 24. Truck Warp Moment Versus Warp Angle Characteristic

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. Large 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 machinists 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 ($\pm$ 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.

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

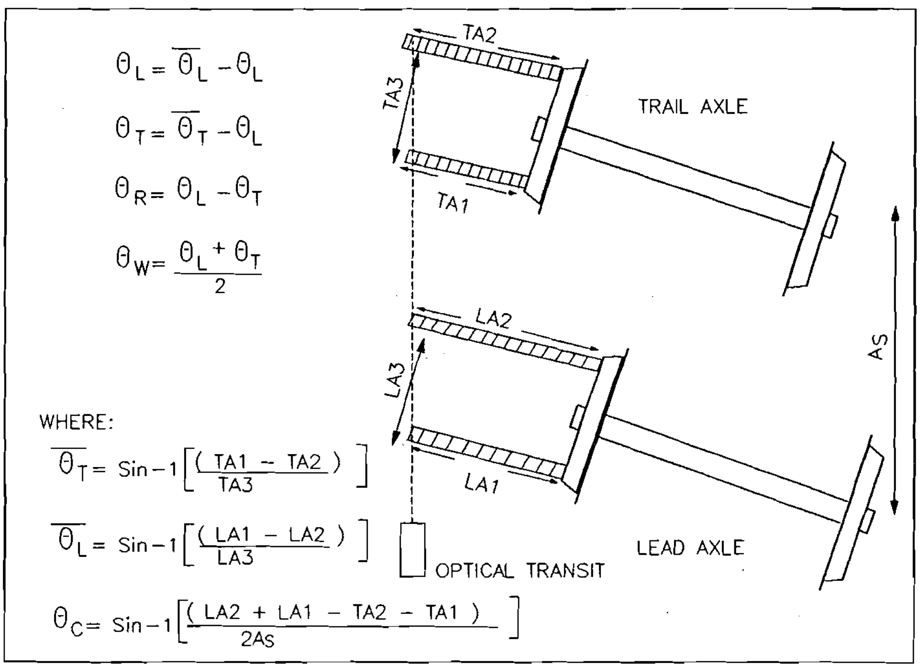

Figure 25. Diagram Showing Schematic for Determining Axle Alignment Measurement

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 pretest modeling did not include the car body flexible mode parameters.

Tables 5 and 6 are NUCARS system files resulting from the characterization tests. The wheel/rail profile geometry used was a theoretical CN-Heumann profile wheel on a theoretical new AREA 136 lb 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.

Table 5. NUCARS System File for Empty Test 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
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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	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

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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	-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 6. NUCARS System File for Loaded Test 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:

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 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		

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 are compared with established Chapter XI safety criteria, which are briefly summarized in Table 7.

Table 7. AAR Chapter XI 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	1.3
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 added*	
Horizontal curve	11.7.3	to be added*	

* Find asterisk in original Chapter XI table
** 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 does occur at 70 mph.

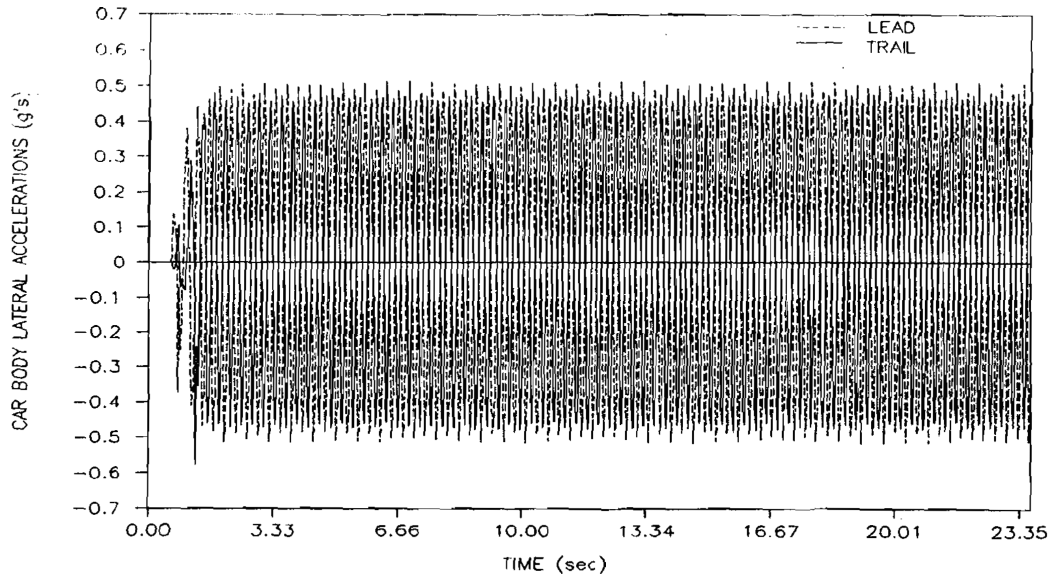

Figure 26. Lateral Car Body Acceleration for the Empty Car While Hunting at a Speed of 57.5 mph Tangent Track

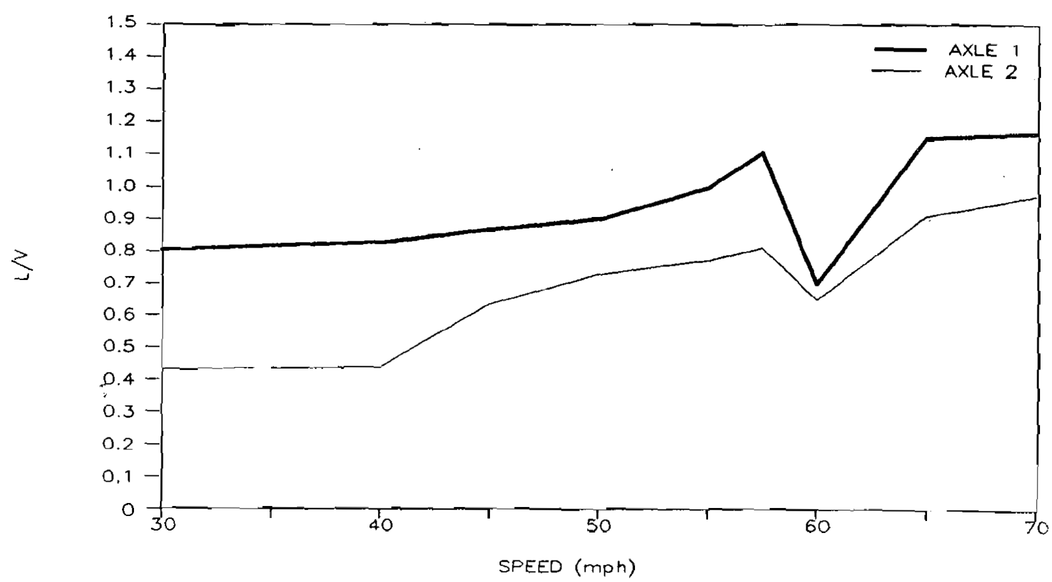

Figure 27. Maximum Axle Sum L/V Ratio Versus Speed of Empty Car on Tangent Track

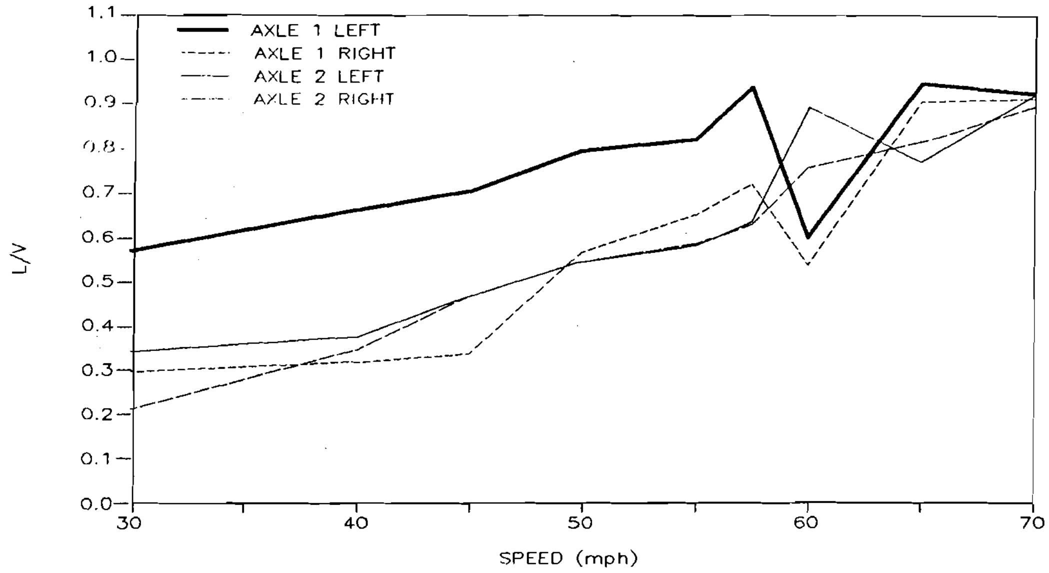

Figure 28. Maximum Wheel L/V Ratio Versus Speed for the Empty Car on Tangent Track

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 0.25 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.

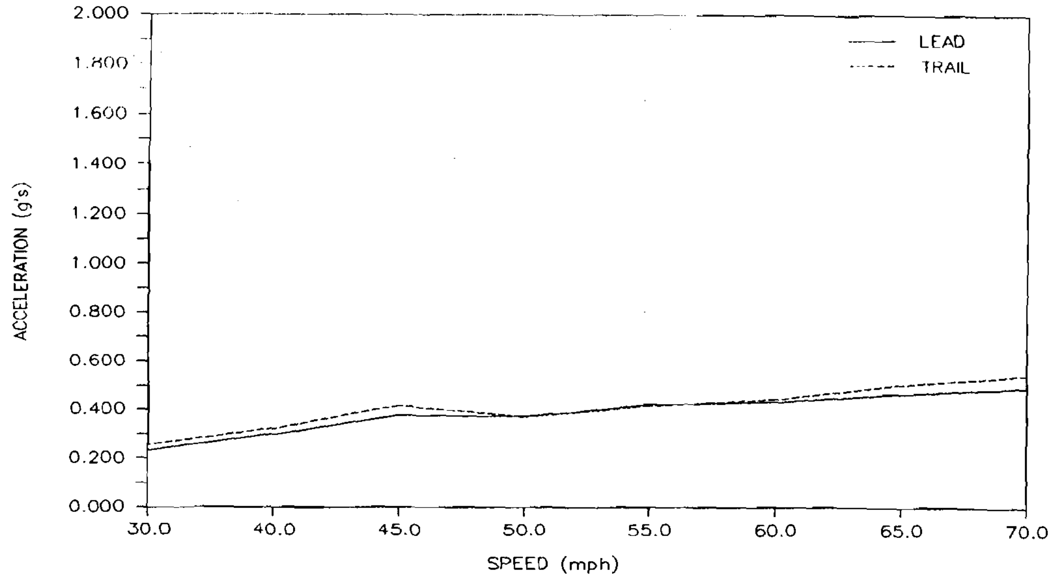

Figure 29. Lateral Acceleration of the Empty Car on 50-Minute-Curved Track

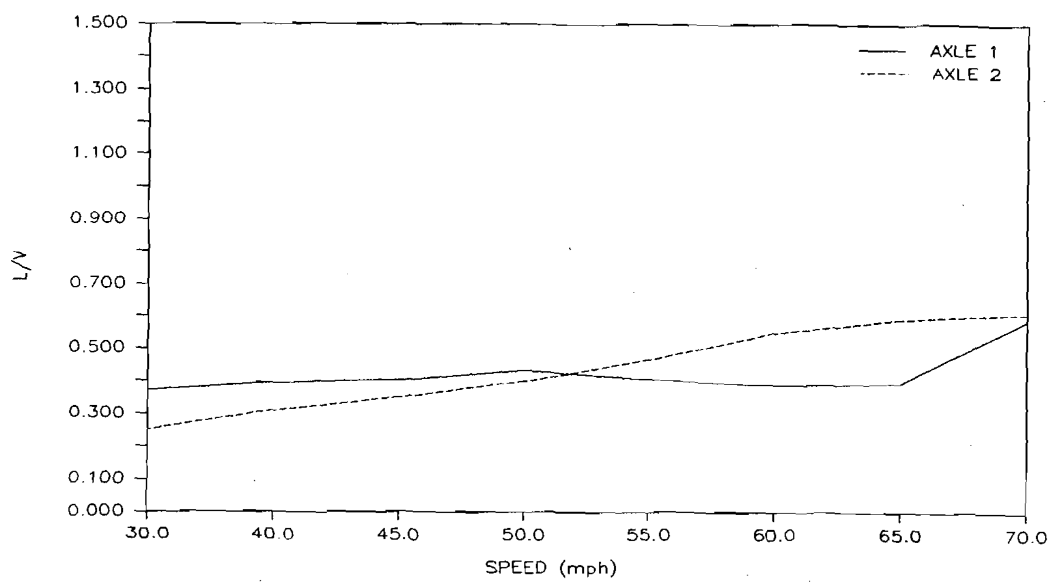

Figure 30. Maximum Axle Sum L/V Ratios Versus Speed for the Empty Car on 50-Minute-Curved Track

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.

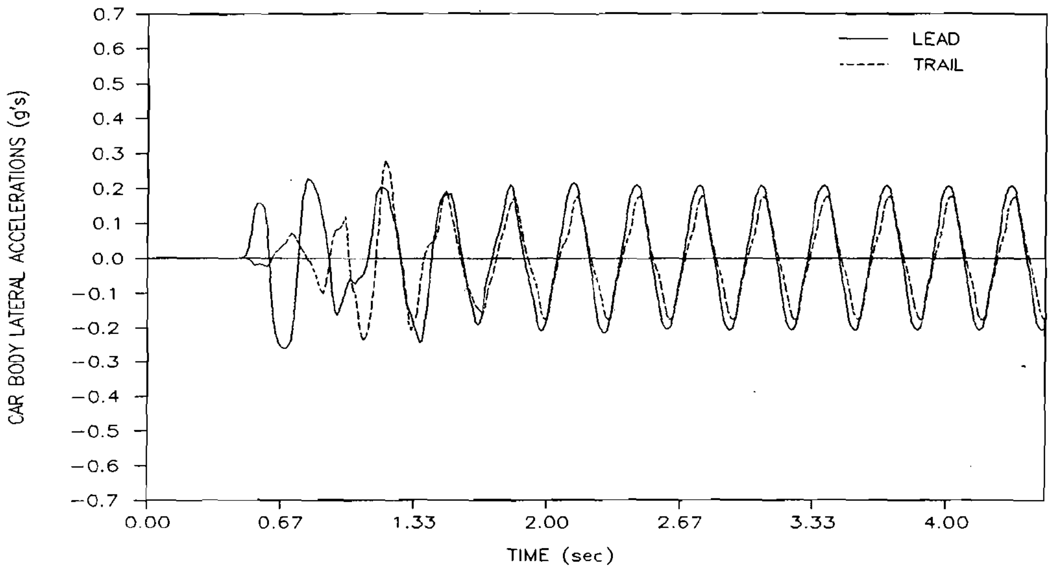

Figure 31. Lateral Car Body Acceleration for the Loaded Car at 70 mph on Tangent Track

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.

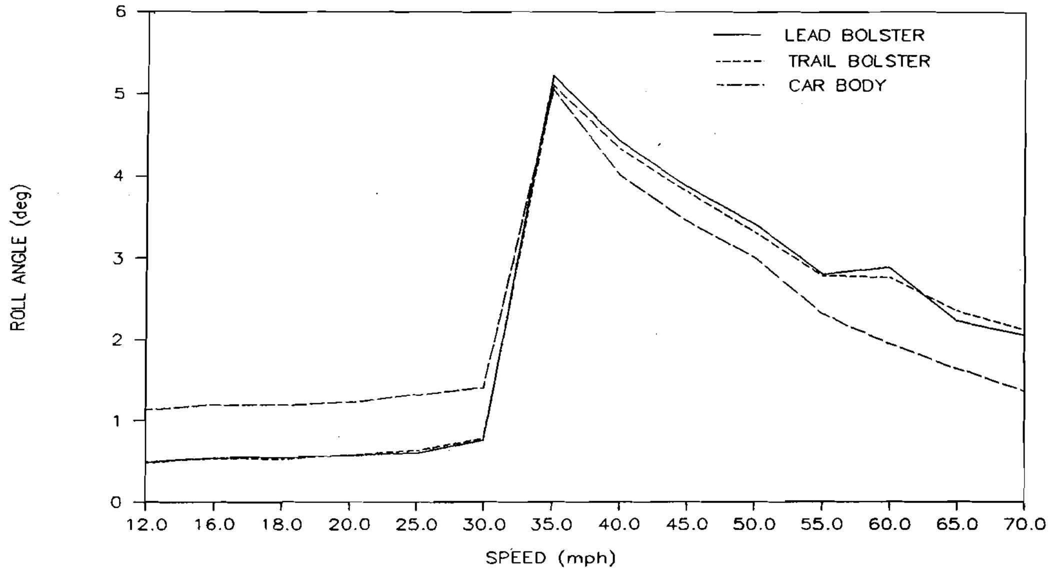

Figure 32. Maximum Car Body and Bolster Roll Angles for the Empty Car in the Twist and Roll Test Zone

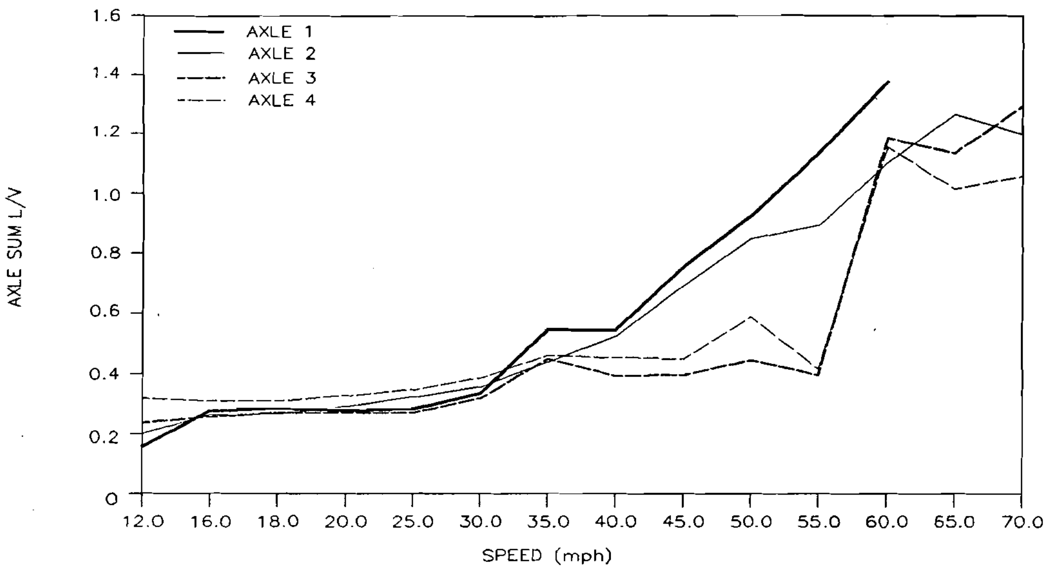

Figure 33. Maximum Axle Sum L/V Ratio for the Empty Car in the Twist and Roll Test Zone

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.

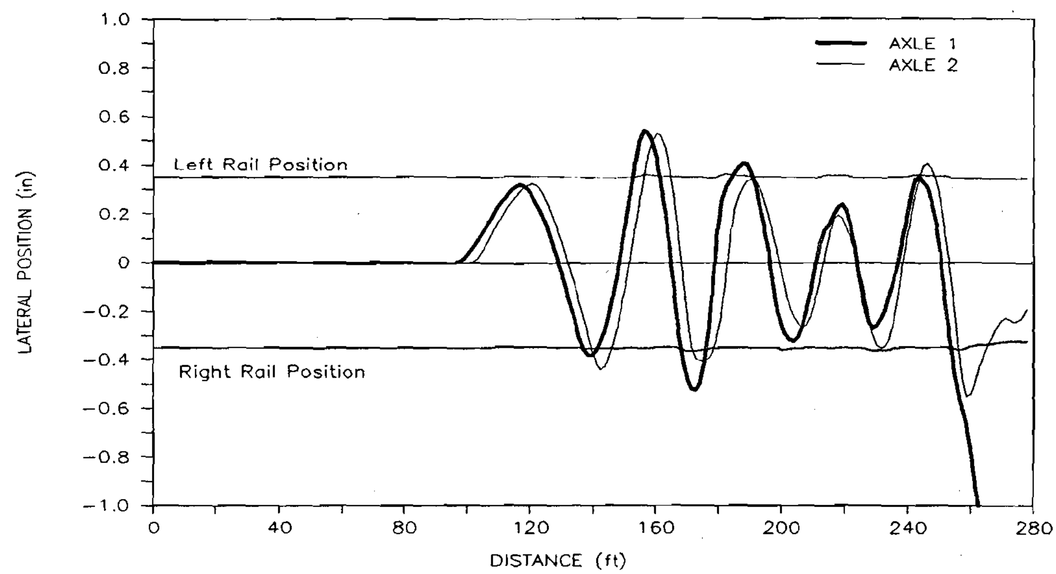

Figure 34. Time History of Axle Lateral Motion of the Empty Car at 70 mph in the Twist and Roll Test Zone

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.

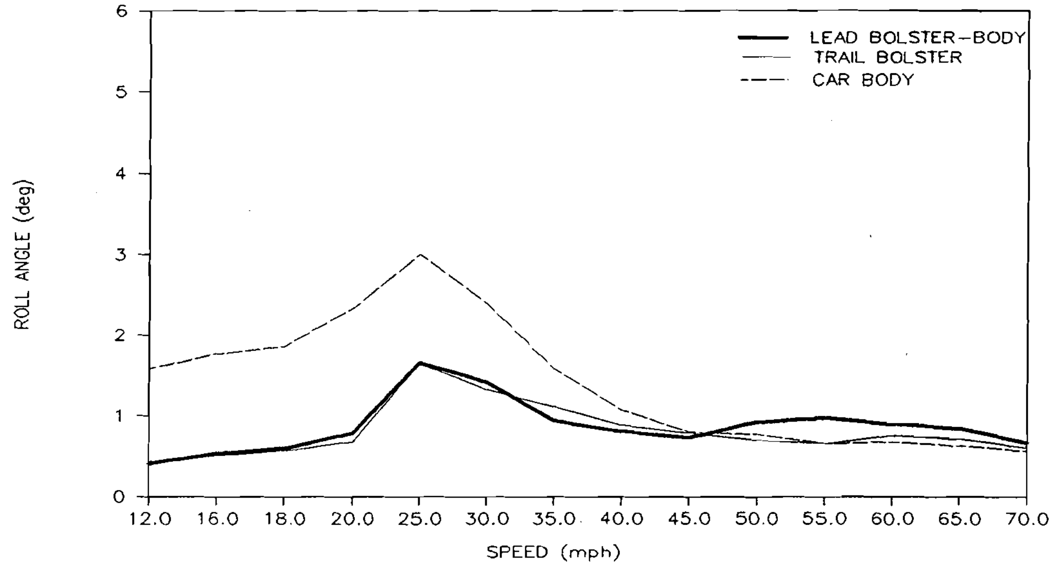

Figure 35. Maximum Car Body and Bolster Roll Angles for the Loaded Car in the Twist and Roll Test Zone

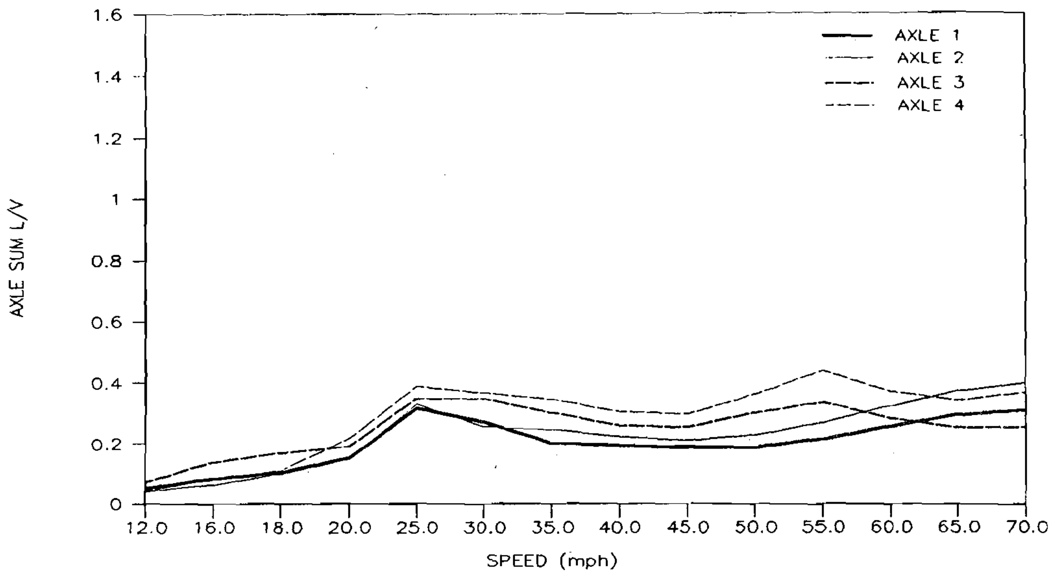

Figure 36. Maximum Axle Sum L/V Ratios for the Loaded Car in the Twist and Roll Test Zone

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 inphase 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.

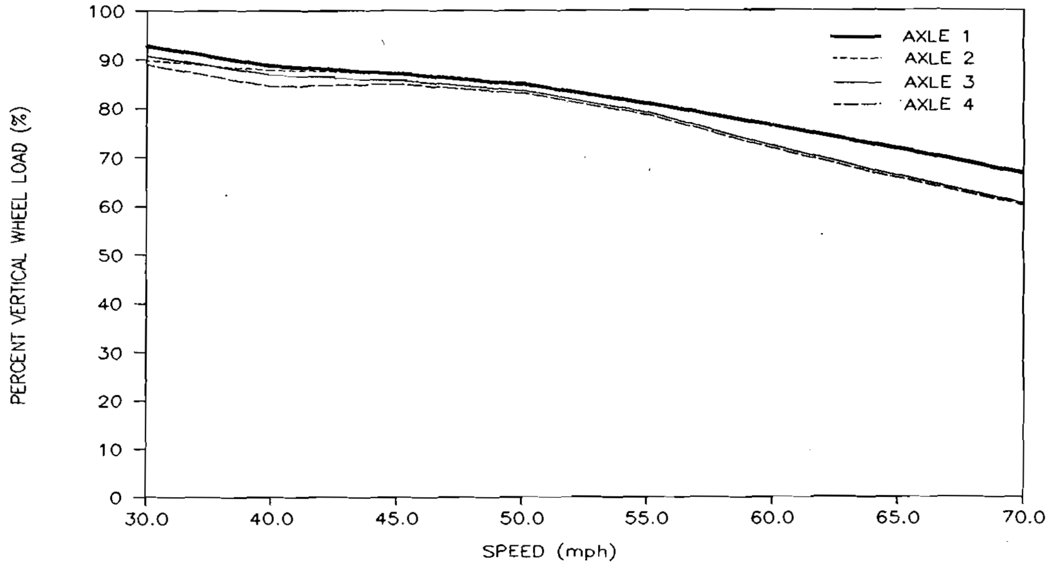

Figure 37. Minimum Percent of Vertical Wheel Load for the Empty Car with Respect to Static Load in the Pitch and Bounce Test Zone

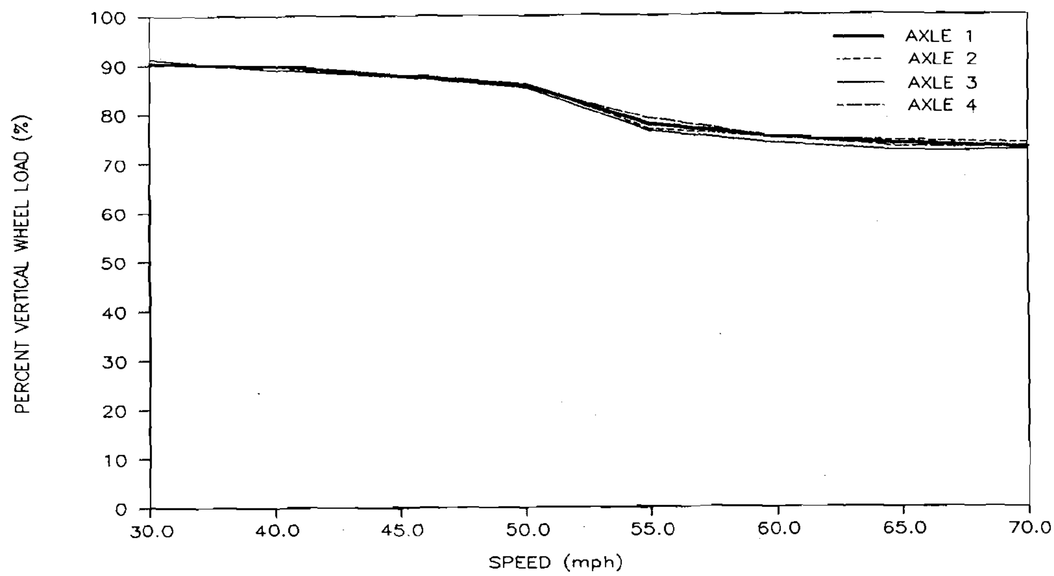

Figure 38. Minimum Percent of Vertical Wheel Load for the Loaded Car with Respect to Static Load in the Pitch and Bounce Test Zone

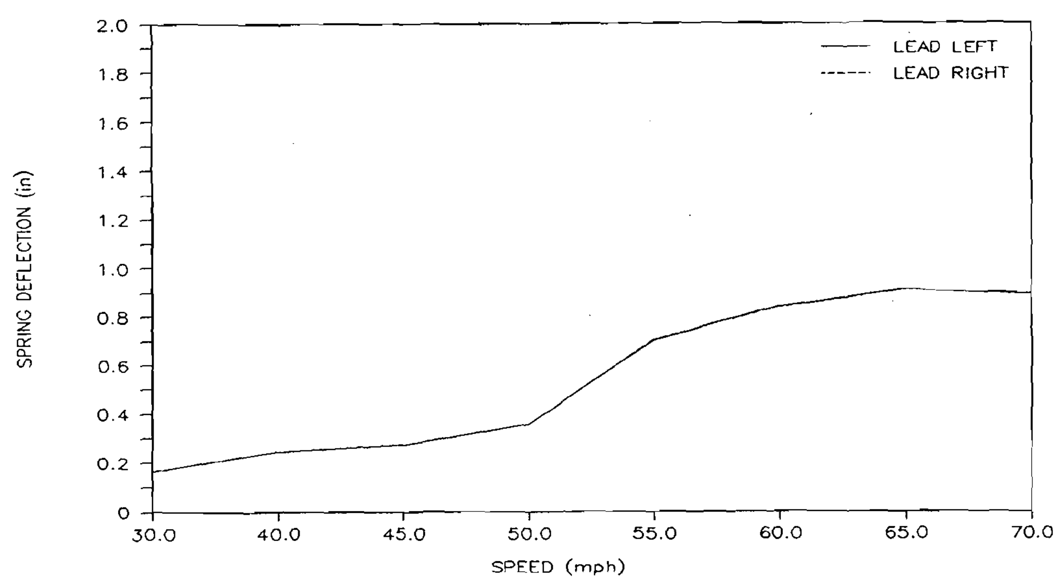

Figure 39. Maximum Vertical Spring Deflections for the Loaded Car in the Pitch and Bounce Test Zone

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.

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.

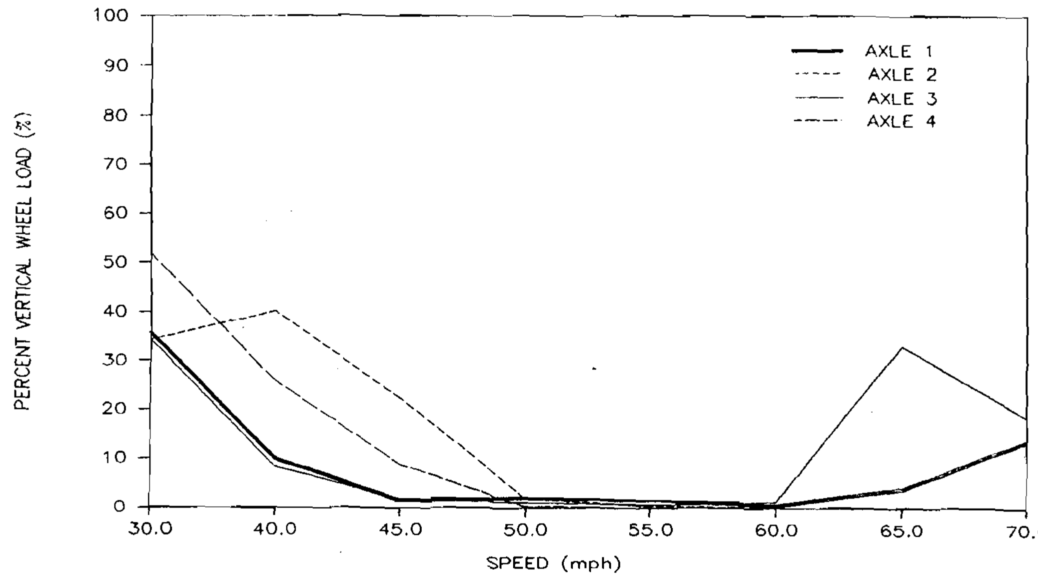

Figure 40. Minimum Percent Vertical Wheel Load for the Empty Car Passing Over the Single Bump

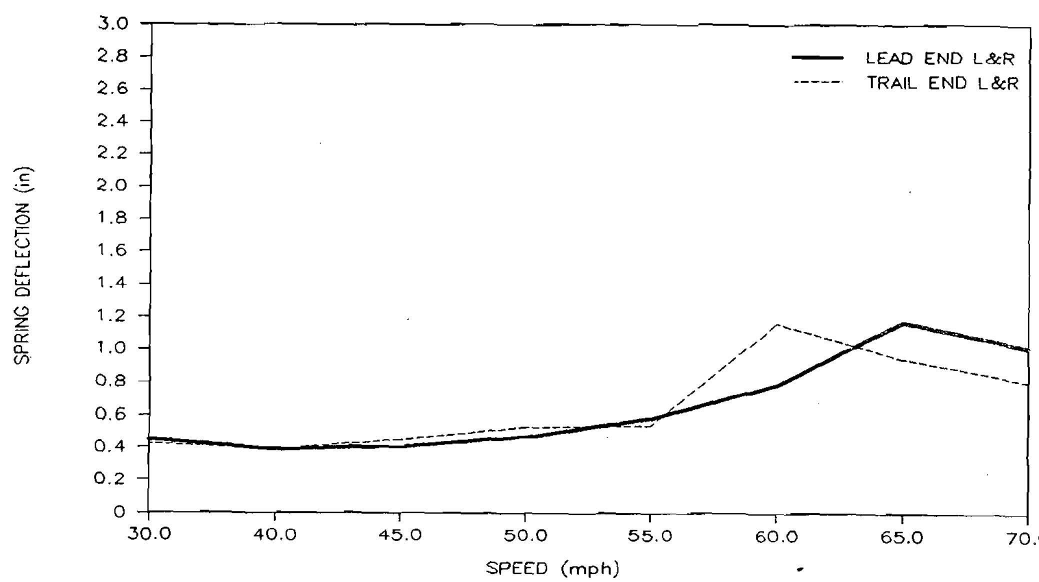

Figure 41. Maximum Vertical Spring Deflection for the Empty Car Passing Over the Single Bump

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.

Test experiences of the Chapter XI single bounce have indicated that the perturbation is too severe. Subsequent to these analyses and the track test program, a proposal was made to modify Chapter XI to lessen the severity of the single bounce. The redefined perturbation has ramps 18 feet long with a 6 foot long, 1.5 inch high center section. Predictions for operation over this new version will be made during the post test analyses.

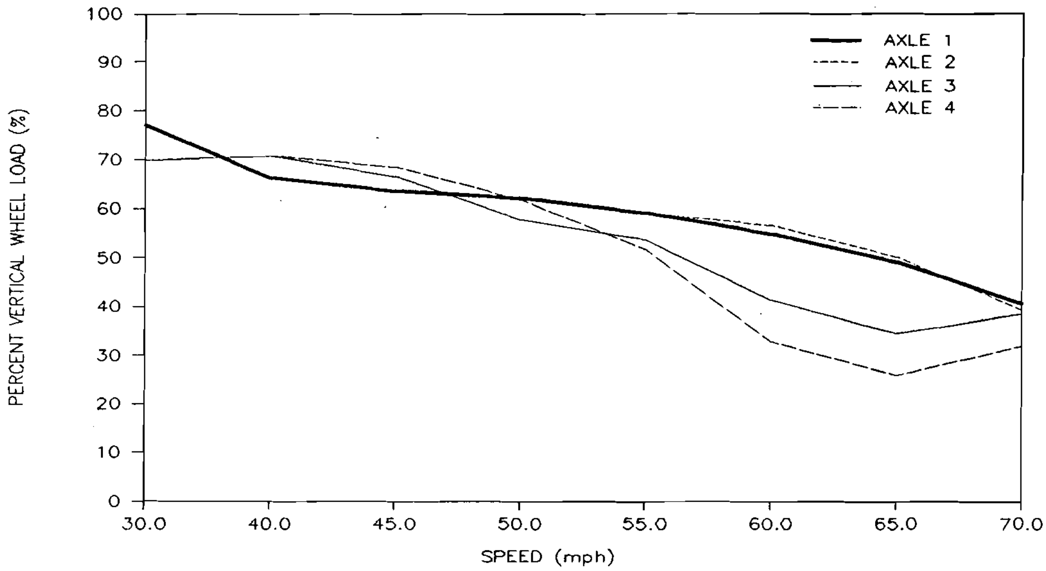

Figure 42. Minimum Percent Vertical Wheel Load for the Loaded Car Passing Over the Single Bump

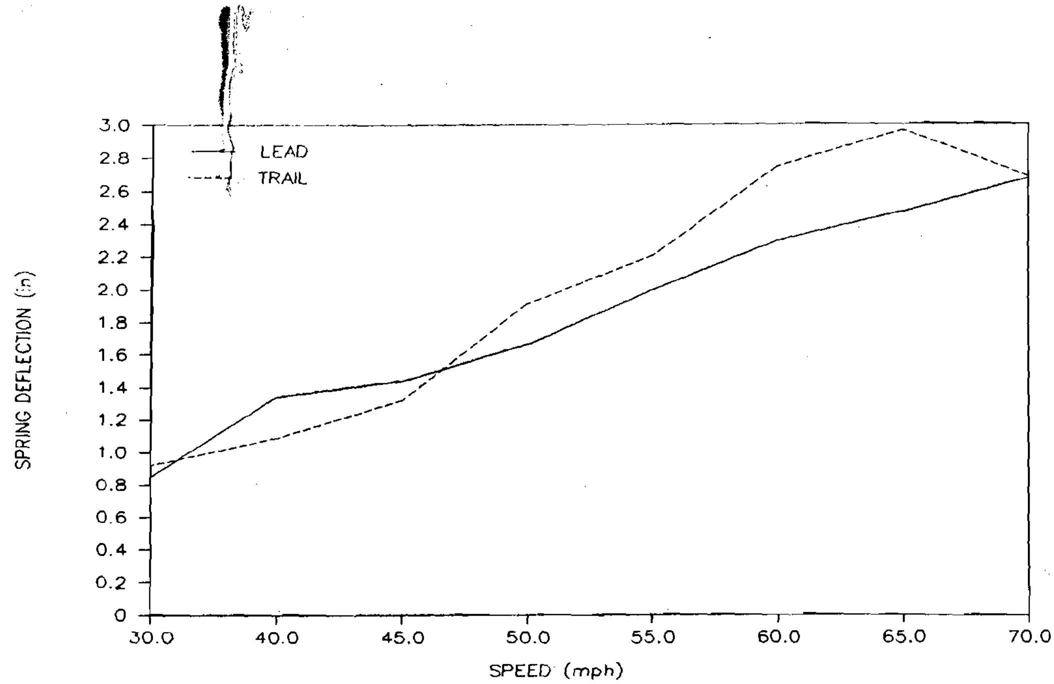

Figure 43. Maximum Vertical Spring Deflection for the Loaded Car Passing Over the Single Bump

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 appear to be inaccurate. 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.

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 front runner car of the Lightweight Car 1 test program. Analyses of the actual track test data are planned to develop dynamic characteristics for these suspensions. Further predictions will be 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.

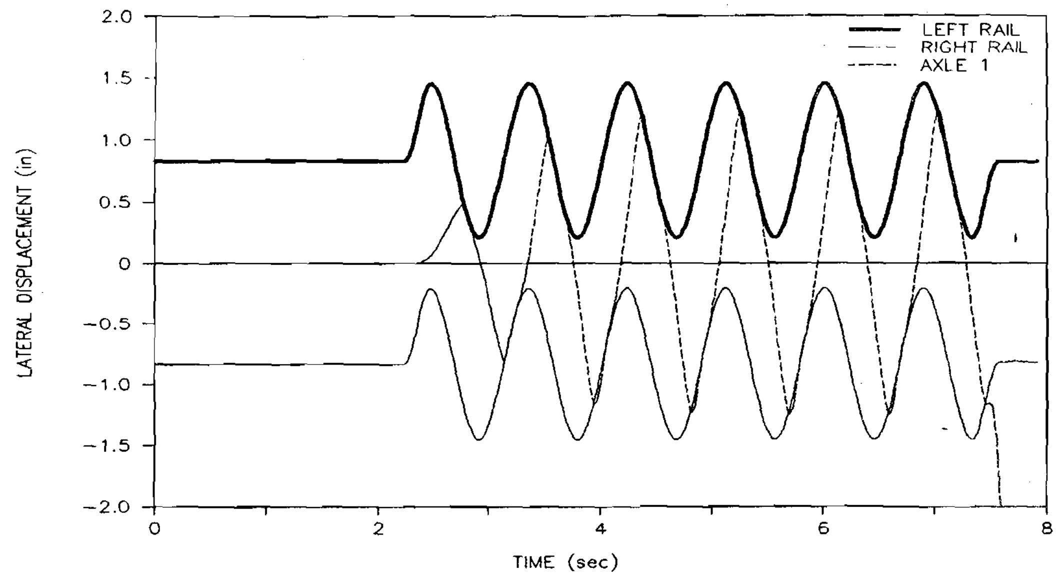

Figure 44. Lateral Position of Axle 1 for the Empty Car in Relation to the Lateral Rail Position in the Yaw Sway Test Zone at 30 mph

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 at 12.0 degrees with +3.0 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 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.

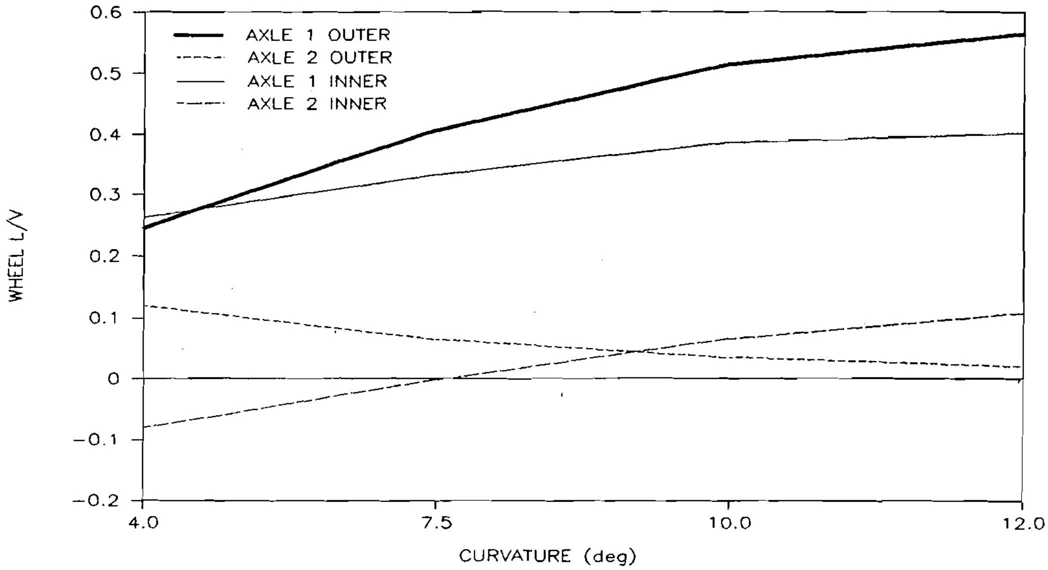

Figure 45. Maximum Individual Wheel L/V Ratio at Plus 3.0 Inches Cant Deficiency

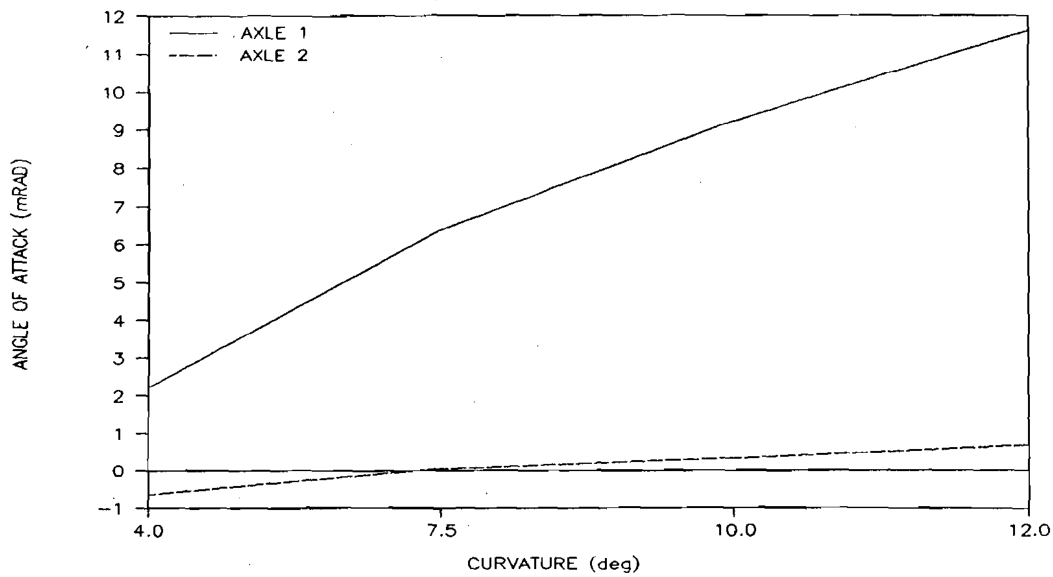

Figure 46. Angle of Attack of Axles 1 and 2 at Plus 3.0 Inches Cant Deficiency

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 -3.0 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 Figures 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 4.0 and 7.5 degree curves. This is because these curves have only 3 inches of superelevation. To achieve -3.0 inches of imbalance on this amount of superelevation the vehicle will have to be standing still.

This improved 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, preventing the truck from steering well.

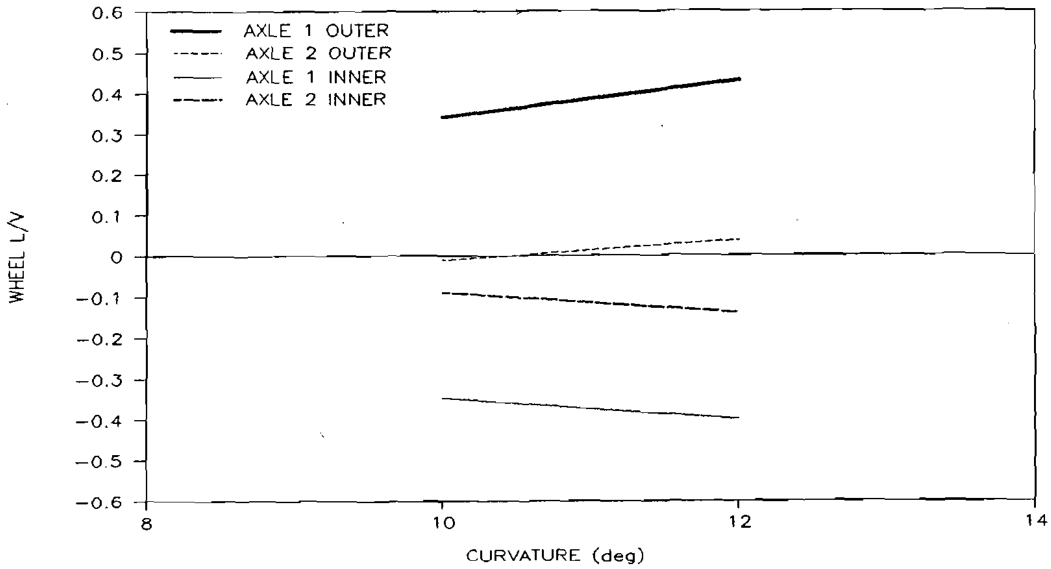

Figure 47. Maximum Wheel L/V at Minus 3.0 Inches Cant Deficiency

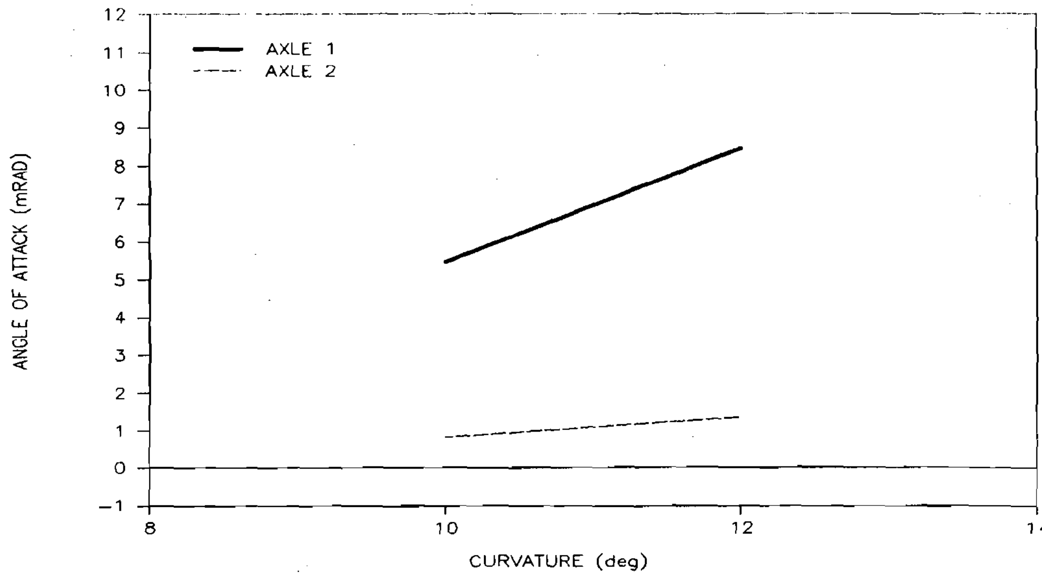

Figure 48. Angle of Attack of Axles 1 and 2 for the Loaded Car at Minus 3.0 Inches Cant Deficiency

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 occur 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 +3.0 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.

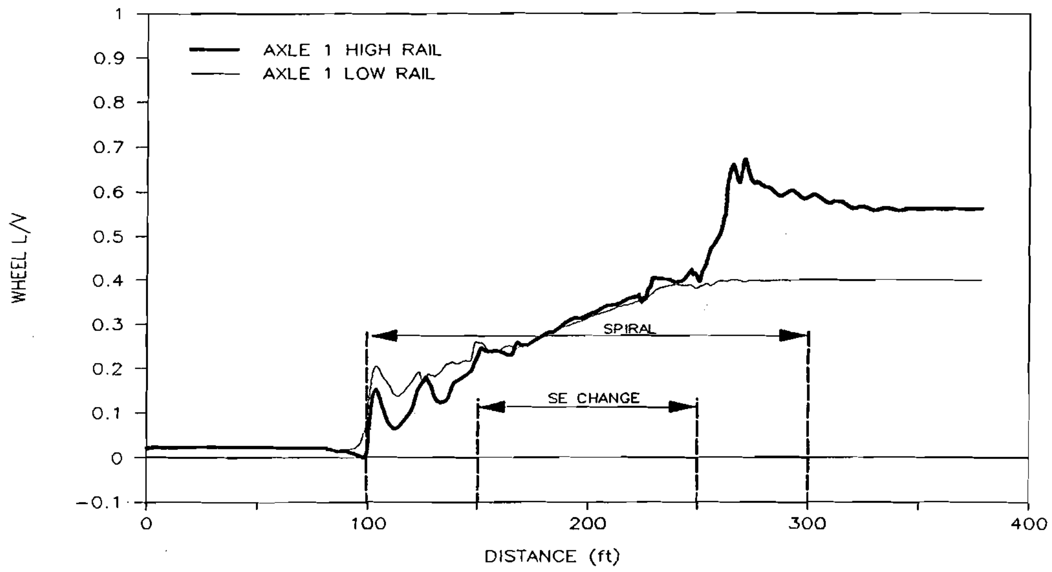

Figure 49. Maximum Wheel L/V for the Empty Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

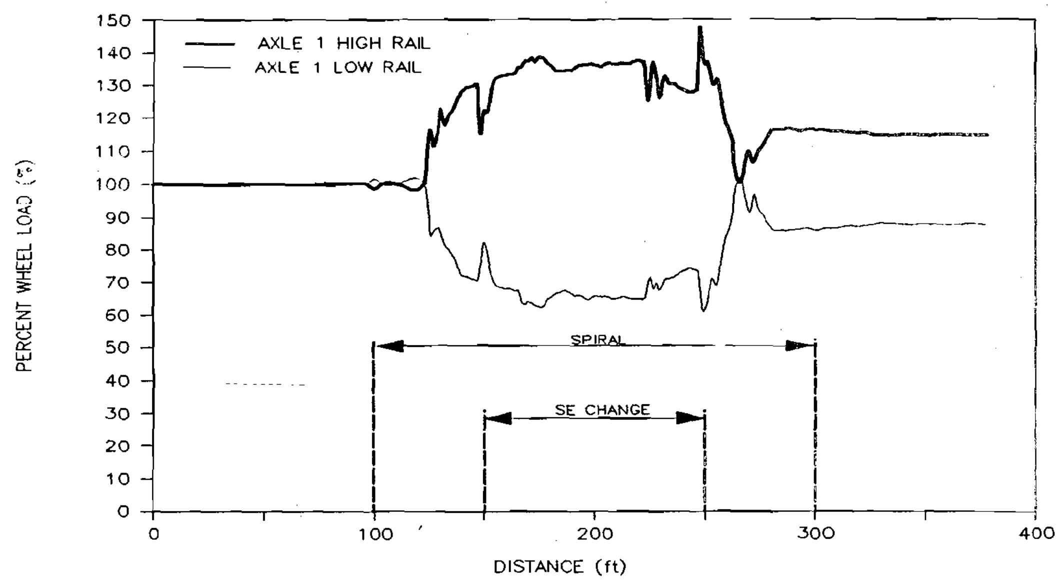

Figure 50. Minimum Percent Vertical Load of Individual Wheels for the Empty Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

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 +3.0 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.

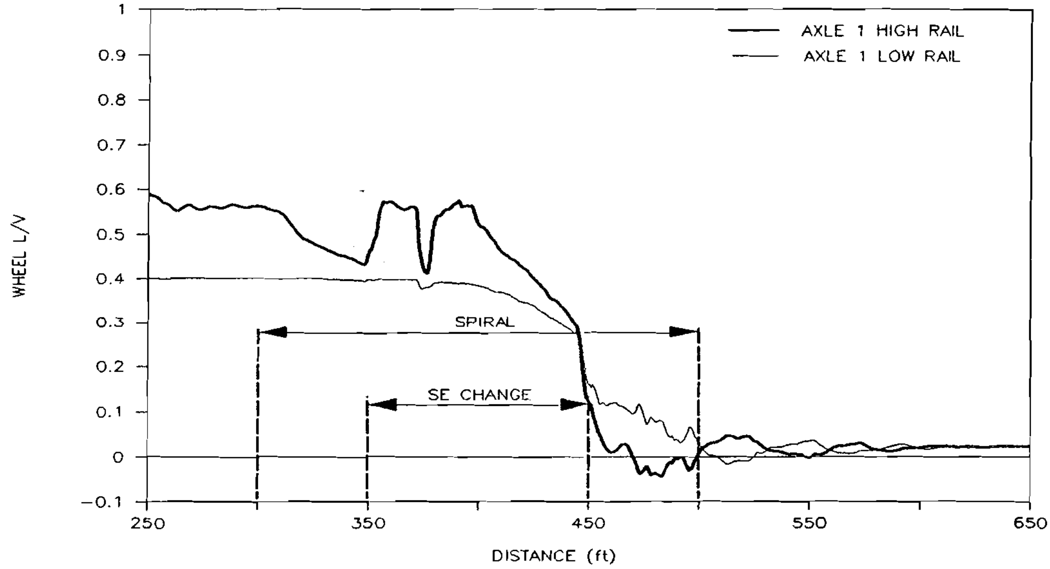

Figure 51. Maximum of Wheel L/V for the Empty Car Exiting the Bunched Spiral to Tangent Track at 31.1 mph

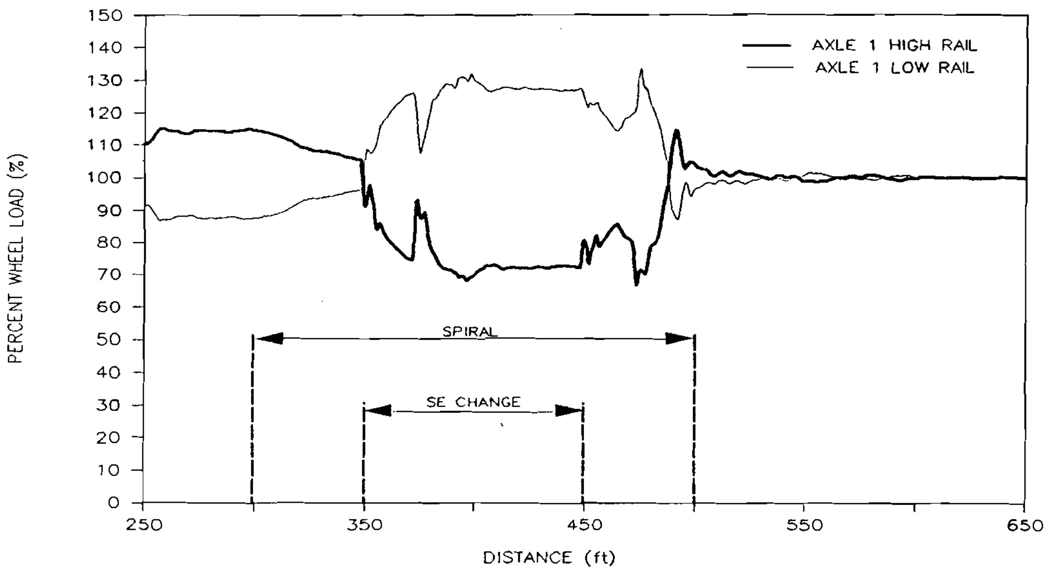

Figure 52. Minimum Percent Wheel Vertical Load for the Empty Car Exiting the Bunched Spiral to Tangent Track at 31.1 mph

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 +3.0 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.

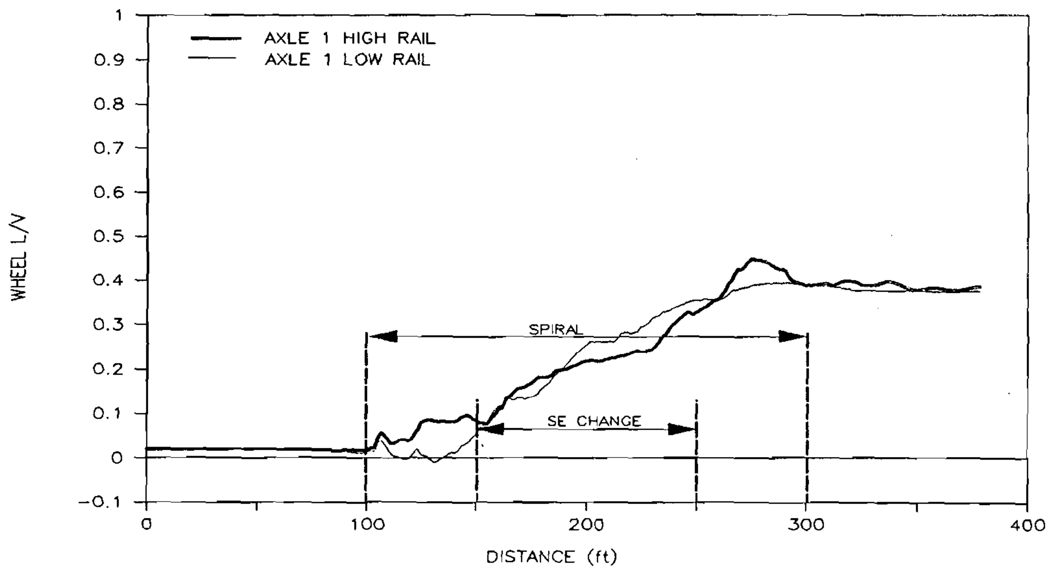

Figure 53. Maximum Wheel L/V for the Loaded Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

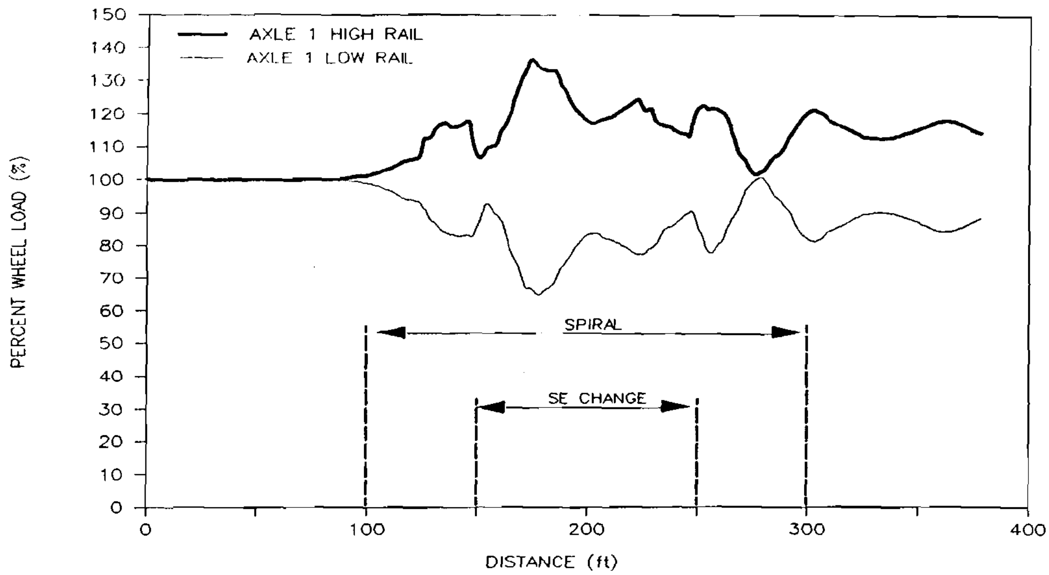

Figure 54. Minimum Percent Wheel Vertical Load for the Loaded Car Entering the Bunched Spiral from Tangent Track at 31.1 mph

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 -3.0 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).

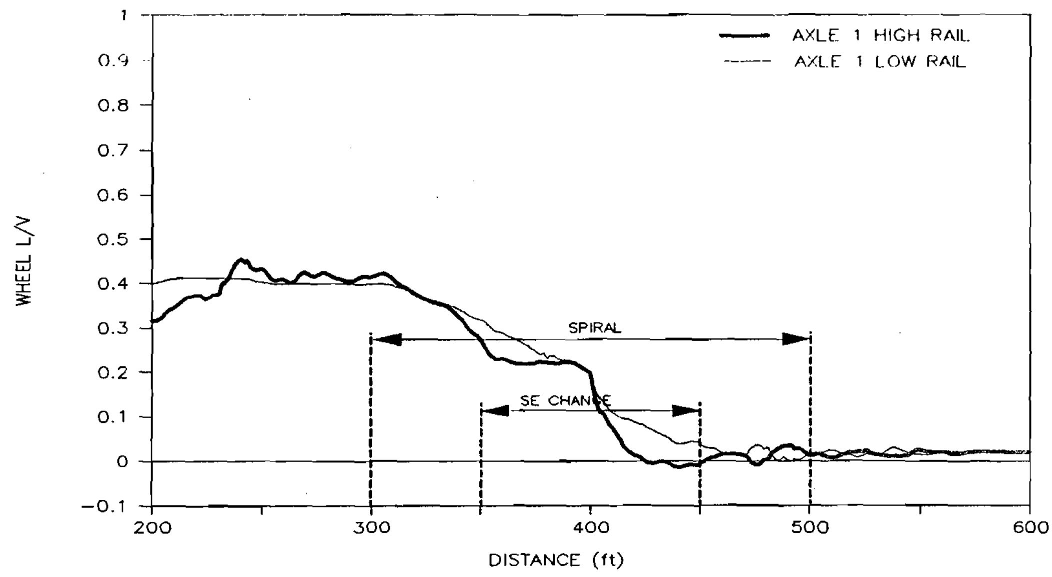

Figure 55. Maximum Wheel L/V for the Loaded Car Exiting the Bunched Spiral to Tangent Track at 15.5 mph

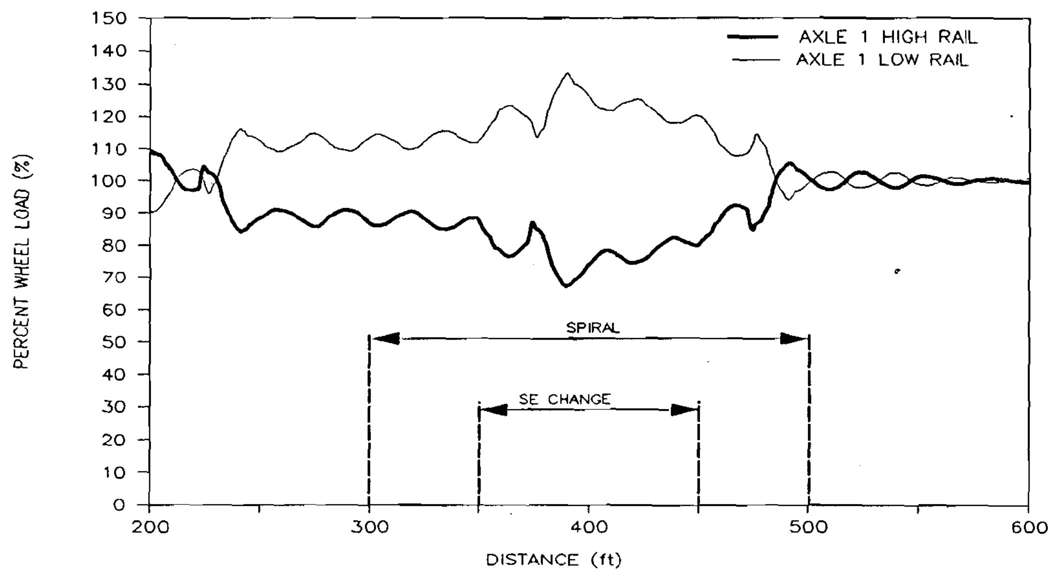

Figure 56. Minimum Percent Wheel Vertical Load for the Loaded Car Exiting the Bunched Spiral to Tangent Track at 15.5 mph

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 (12.0 and 19.0 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.

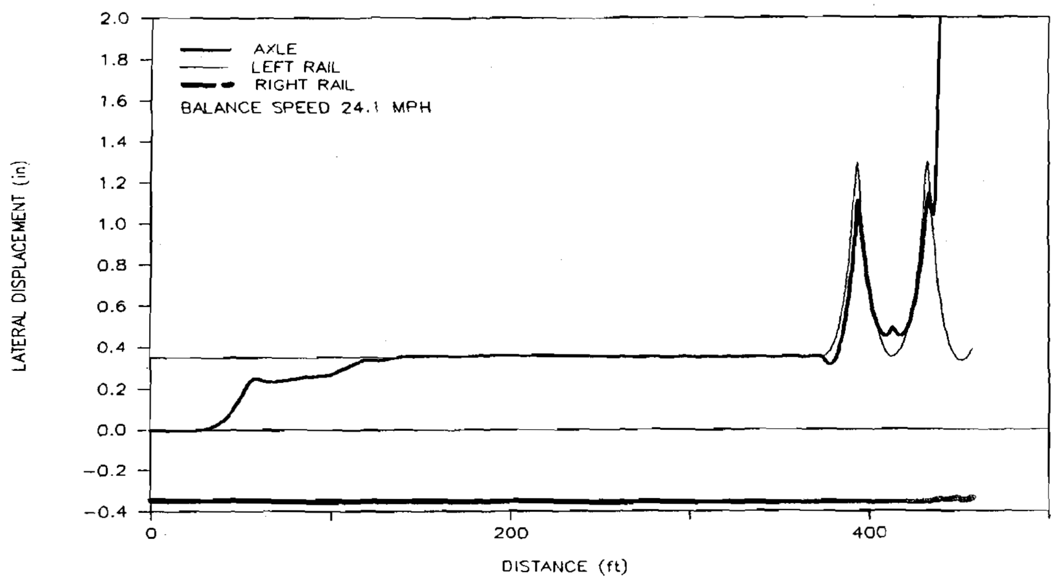

Figure 57. Lateral Position of Axle 1, Left and Right Rails as the Empty Car Negotiates the Dynamic Curve at Balance Speed

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 -3.0 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 3.0 inches of imbalance (31.9 mph) on the trail axle inside wheel. Figure 59 shows the minimum wheel loads.

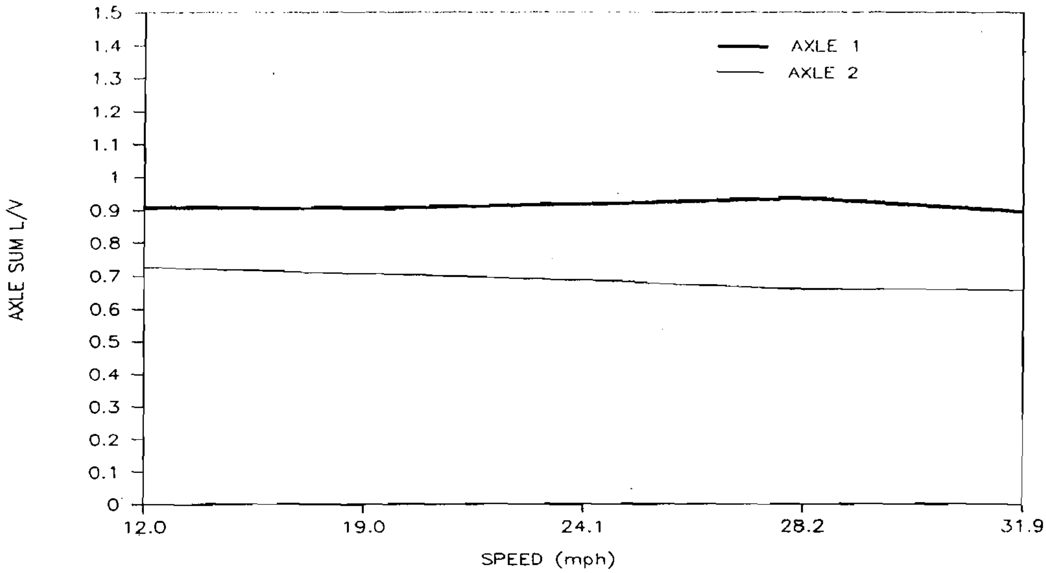

Figure 58. Maximum Axle Sum L/V Ratios for the Loaded Car Negotiating the Dynamic Curve

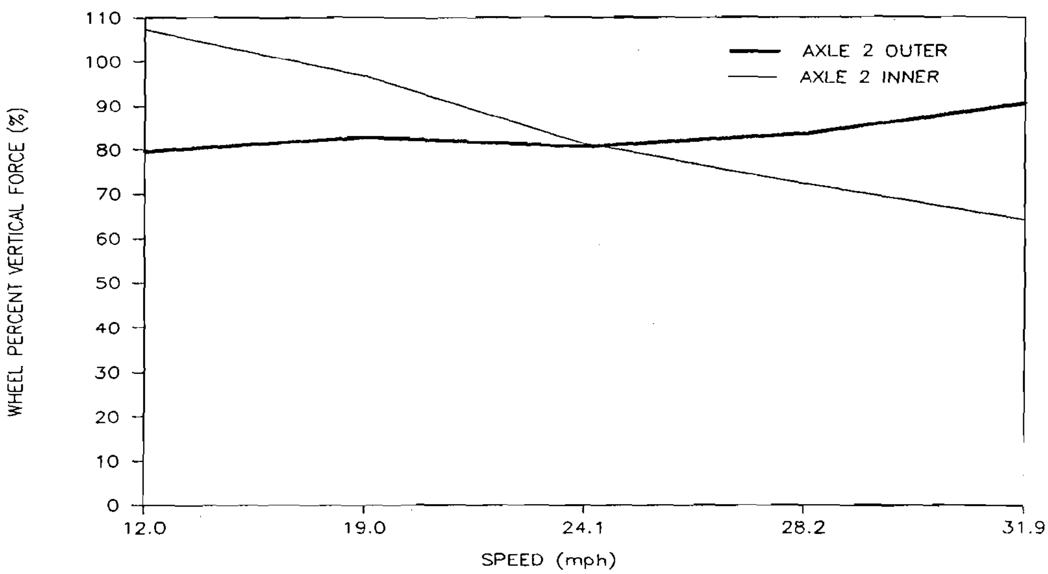

Figure 59. Minimum Percent Wheel Vertical Load for the Loaded Car in the Dynamic Curve

7.0 INTERIM CONCLUSIONS

7.1 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.

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

Attempts will be made to develop dynamic suspension characteristics from the track test data for the lateral and yaw suspensions. These will be compared to the MSU and air table test results. Revisions to the NUCARS suspension characteristics will be made as necessary based on these comparisons.

7.2 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 8 summarizes the predictions in terms of Chapter XI limiting criteria.

Exceedance of Chapter XI limiting criteria was predicted for the following test regimes. This indicates that careful monitoring of the track tests will 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 8. Summary Results of Pretest NUCARS Predictions

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
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
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.

Table 8. Summary Results of Pretest 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
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).

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APPENDIX

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

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Chapter X

SINGLE-AXLE SUSPENSION SYSTEM PARAMETERS AND DESIGN GUIDELINES

10.1 GENERAL

10.1.1.

The purpose of this chapter is to provide single-axle suspension parameters and design guidelines for the construction of two-axle freight cars.

10.1.2.

All two-axle cars are considered cars of an untried type, and their approval procedure requirements are outlined in section 1.2 of Chapter I. The suspension system and the car body will be treated as a single entity and shall be capable of being operated, singly or in combination with any other approved cars, on standard gauge rails, using AAR-approved wheelsets and roller bearings.

10.1.3.

Suspension must be designed as an integral part of the car. Performance and suitability must be proven through tests as a combined unit. Approval of the car or suspension will be based on tests of a specific combination of car and suspension only.

10.1.4.

Suspension must provide safe, dependable operation of the car over track of classes 1 through 6 under normal operating conditions, and the suspension system and attachment to car must not exhibit any structural or operational distress after completion of all required tests.

10.1.5.

The design of the suspension members and connections must be compatible with the design of the car body insofar as structural integrity is concerned. Overall design concept, static and dynamic parameters, and materials used in construction must be submitted and approved by the AAR Car Construction Committee before any official AAR tests are initiated.

10.1.6.

The design of the suspension system must satisfy applicable AAR and FRA requirements.

10.2. OPERATIONAL AND STRUCTURAL REQUIREMENTS

10.2.1.

Suspension system is to permit operation of the car at any speed up to and including 70 miles per hour within the outline of the track, worthiness criteria as specified in Chapter XI, Table 11.1, "Criteria for Assessing the Requirements for Field Service".

10.2.2. VACANT

10.2.3.

Suspension must permit self-steering of wheelset up to and including 4° curves. Longitudinal and lateral freedom of wheelset is to be restrained so as not to exceed ± 1.5″ lateral movement and a longitudinal movement of such magnitude as to permit safe operation of car on a curve of 150′ radius.

10.2.4.

Suspension must permit application of an on-tread brake system and must provide means for transmitting retardation forces from the wheelset to the car body. Reasonable access to the brake heads for replacement of brake shoes must be provided.

10.2.5.

Suspension structural members must permit unobstructed scanning of roller bearings and wheels by existing "hot box" detectors.

10.2.6.

Carbody suspension attachment is to be designed to sustain the forces developed by decelerating a fully-loaded car from 20 miles per hour to 0 miles per hour in a distance of 50 feet by application of retardation to wheels (e.g., use of track retarder).

10.2.7.

Suspension design, including all drawings, materials used in fabrication, data as to choice of loads and forces used in design of individual suspension components, stress and fatigue calculation or data must be submitted for review by the AAR Car Construction Committee before authorization for field test can be granted.

10.2.8.

Contemplated changes of arrangement, components, dimensions, material, fabrication processes, etc. in variance from the test vehicle must be submitted for approval by the AAR Car Construction Committee before implementation.

10.3. TESTING

10.3.1.

Since car and suspension is considered a unit, testing of the complete car as outlined in Chapter XI will be applicable.

10.3.2.

Semi-annual reports covering the suspension performance, wheel wear, and other operational and structural information shall be submitted to AAR until a minimum of 250,000 miles per car is attained.

CHAPTER X
APPENDIX A

PERFORMANCE SPECIFICATION FOR YAW-DAMPING DEVICES
ADOPTED 1987; Revised 1989

1.0. GENERAL

1.1.

The purpose of this specification is to provide an evaluation guide for wheelset or truck yaw-damping devices.

1.2.

The yaw damping characteristics are normally matched to a specific car-suspension system and therefore are not transferable to other suspension systems, or to other car structures equipped with similar suspension systems.

1.3.

The damper design shall provide for satisfactory functional and mechanical performance of at least 300,000 miles of maintenance-free service in the normal railroad environment.

1.4.

Request for approval of the device must include a set of detail and assembly drawings incorporating performance criteria, material specifications, and complete dimensional and tolerance data depicting the device.

1.5.

Hydraulic type yaw dampers must be equipped to permit visual inspection of fluid level required for proper operation.

2.0. PERFORMANCE CRITERIA

2.1.

The yaw damper of a design to be used in production must pass Section 11.5.2 and 11.5.3 performance tests on a car for which it is designed, as outlined in Chapter XI of the M-1001 Specifications, and must provide the required damping levels in both its "new" and "worn" condition.

2.2.

The yaw damper shall provide for "FREE" curving of the wheelset through all curves up to and inclusive of 4 degrees. "FREE" curving for purpose of this test is defined as the average measured reading during the test being no less than 90% of the axle-to-car body rotation angle when compared with readings obtained from the same car negotiating the same curve without the damper installed, (minimum three runs through curve in either direction should be performed at speeds of 10 and 30 mph on dry rail).

2.3.

At the completion of the test, the device will be removed by an AAR representative and sent to the AAR Technical Center for future comparison purposes. This device will be designated as test unit "witness", and will be retained by the AAR for reference purposes.

3.0. FIELD TEST REQUIREMENTS

3.1.

Upon successful completion of tests outlined in 2.1 and 2.2 which have been witnessed by authorized AAR representative, five (5) carsets of the design for field testing purpose can be granted. A lesser quantity can be authorized for prototype testing.

3.2. CONDITIONAL APPROVAL

To obtain conditional approval, five (5) sets of the device which have been placed in the field test must have completed at least 50,000 miles of service each, and fulfilled the following requirements:

3.2.1.

The devices shall be inspected at the end of test trial by an authorized AAR Representative for absence of mechanical distress and for proper functioning.

3.2.1.1.

A retest shall be conducted in accordance with the lateral stability requirements in Paragraph 2.1.

3.2.2.

Two devices chosen by an AAR Representative will be removed from cars. Tests may be performed at the Vendor's or other approved test facility and must be witnessed by AAR representatives. The AAR must be given at least ten (10) days notice prior to the start of testing.

3.2.3.

The following tests shall be performed on both the original "witness" device and a device removed from field service:

3.2.3.1.

The test device shall undergo a 5000 Ib. load suddenly applied tension or compression load at the middle and end positions of its stroke.

3.2.3.2.

Both the "witness" and the test device will undergo an extension and compression test. The resistance forces measured at velocities of .06 ft./sec.; 3 ft./sec.; and 1 ft./sec. shall not vary more than 10% from the comparable force obtained from the same test performed with the "witness" device. Devices using acceleration sensing as movement arresting mode, shall be tested at predetermined acceleration levels for comparison purposes and results shall not vary more than 10% from those of the witness device. Alternatively, for friction devices, the field test unit shall undergo a ±112-inch displacement induced by a 1 H_z sinusoidal velocity input. The device will be cycled about the nominal installed length. The double amplitude of the result force shall not vary more than 10% from that of the witness device.

3.2.3.3.

After completion of above tests, the devices shall be examined for performance and wear deterioration. Both items must be judged to be within the life expectancy of 300,800 miles maintenance free service.

3.2.4.

Successful completion of field test and lab test will qualify the device to receive conditional approval for service.

3.2.5.

Any external, internal, design, material or performance changes of an approved damper must be reported to AAR Car Construction Committee for review and concurrence before incorporation of such in production. Willful disregard of this provision will cause revocation of the approval.

3.2.6.

The "witness" device will be used for comparative evaluation and verification purposes of any future performance verifications of design improvements etc.

3.2.7.

Semiannual Service Reports are to be provided by the manufacturer until at least 10% of the authorized devices, no less than 400, have obtained a service life of 300,000 miles, and have been inspected by an authorized AAR Representative. Two such devices will be removed from a car sent to the lab for inspection and test as outlined in 3.2.3 using, however, the "worn" witness device data for comparison purposes.

3.2.8.

Upon completion of the above stated service experience, and satisfactory completion of test requirements outlined in Section 3.2.7, the device will be considered fully approved.

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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).

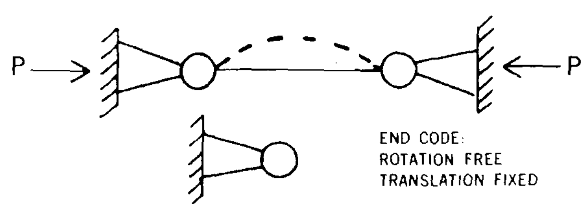

Figure 11.2.3.1

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 ⅛″ 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	1.3*
Constant curving (empty and loaded)	11.5.3	95th percentile maximum wheel L/V	0.8
		or	
		95th percentile maximum sum L/V axle	1.3
Spiral (empty and loaded)	11.5.4	minimum vertical load (%)	10 **
		maximum wheel L/V	0.8*
Twist, Roll (empty and loaded)	11.6.2	maximum roll (deg)***	6
		maximum sum L/V axle	1.3
		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	1.3*
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 (%)	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

4 1 90

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.

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

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 1.3 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 0.8 on all wheels measured, or the instantaneous sum of the absolute wheel L/Vs on any axle shall be less than 1.3, 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 0.8 on all wheels measured, or the instantaneous sum of the absolute wheel L/Vs on any axle shall be less than 1.3. 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 lb. or 136 lb. rail profiles.

The predicted lateral-to-vertical force ratio shall not exceed 0.8, 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.

11.5.1.3. 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 track-worthiness 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.

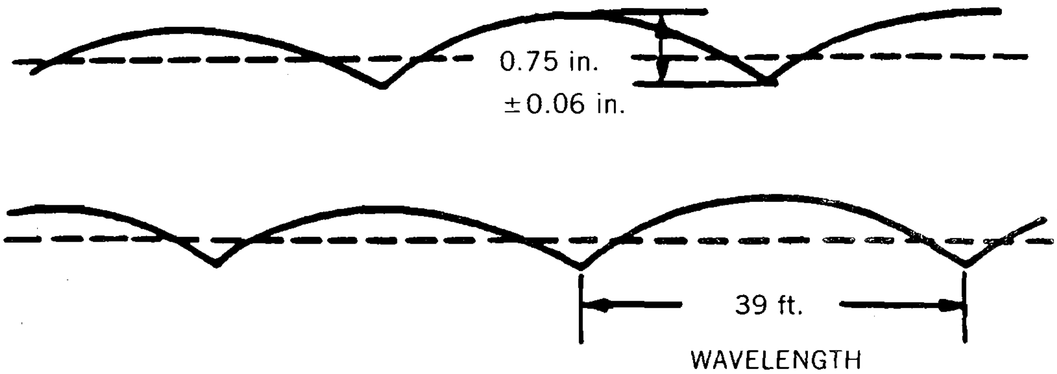

Figure 11.1.

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 1.3, 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.

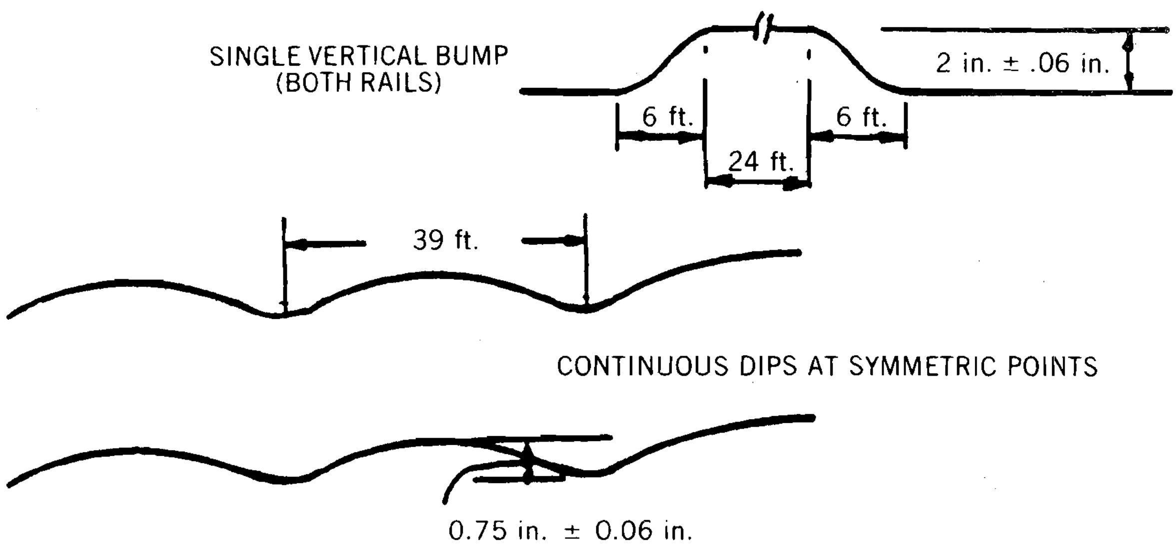

Figure 11.2.

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 1.3, 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.

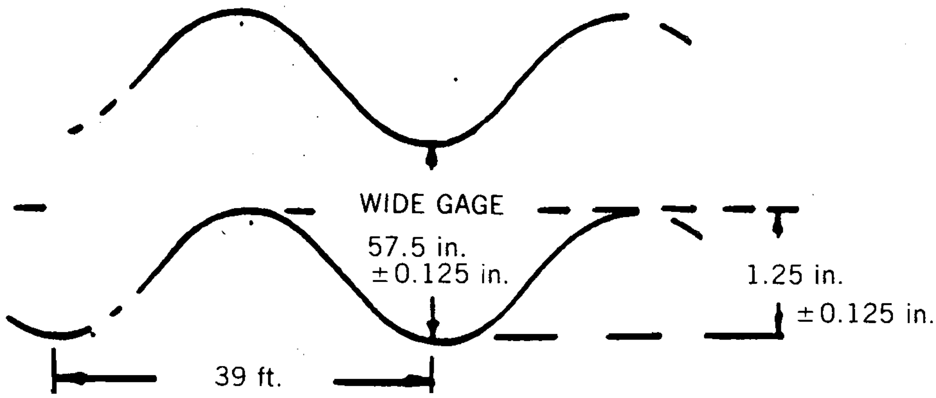

Figure 11.3.

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 1.3, 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.

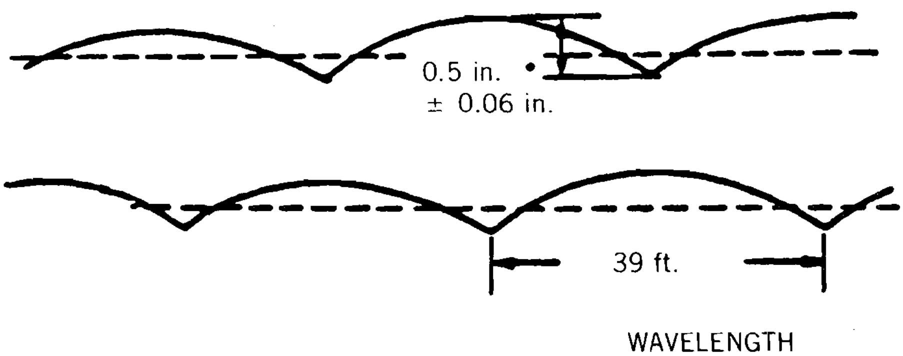

Figure 11.4.

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.

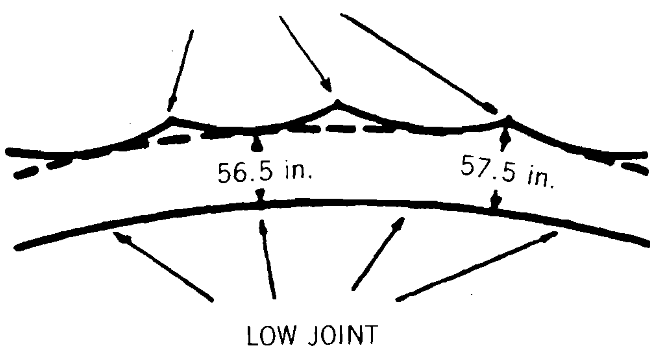

Figure 11.5.

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 1.3, 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 0.8, and the instantaneous sum of the absolute wheel L/Vs on any axle shall be less than 1.3, 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.

3.3.4.

A steady-state curving test to again confirm that total truck vertical loads are within +/- 5 percent of the theoretical value for constant speed operation on constant curvature track (for the test curve described above). Wheelset signals will be evaluated on the basis of mean values for a randomly chosen output segment having a minimum duration of ten seconds.

The test procedures prescribed above are also to be repeated and recorded at the start of each Chapter XI test series. A record of such results is to be kept for each Chapter XI certified wheelset. A minimum of the vertical load accuracy test is to be performed at the start of each daily test session.

4.0. RECORDS

4.1.

The theoretical analyses described are necessary only once for each wheelset design. The static calibration and field procedures must be performed for each wheelset produced to an accepted specification.

4.2.

An instrumented wheelset which has met these requirements will be so certified by the designated AAR representative.

4.3.

The designated AAR observer for Chapter XI testing will verify that the instrumented wheelsets to be used have been accepted for testing and the test procedures described in Section 3.3 above are completed satisfactorily.

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**THIS IS FINAL PAGE OF VOLUME I
SEE VOLUME II FOR
APPENDICES**