U.S. Department
of Transportation
**Federal Railroad
Administration**

PEACEKEEPER RAIL GARRISON TEST OF ENGINEERING MODEL MAINTENANCE CAR

Office of Research and
Development
Washington, DC 20590

Dominic DiBrito
Brent Whitsitt

Association of American Railroads
Transportation Test Center
Pueblo, CO 81001

October 1991

Document is available from
the Office of Research
and Development, Federal
Railroad Administration,
Washington, DC 20590

DISCLAIMER

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.

4. Title and Subtitle	5. Report Date
PEACEKEEPER RAIL GARRISON TEST OF ENGINEERING MODEL MAINTENANCE CAR	October 1991
	6. Performing Organization Association of American Railroads
7. Authors Dominic DiBrito Brent Whitsitt	8. Performing Organization Report No.
9. Performing Organization Name and Address Association of American Railroads Transportation Test Center P.O. Box 11130 Pueblo, CO 81001	10. Work Unit No. 11. Contract or Grant No. DTFR53-82-C-00282 Task Order 40
12. Sponsoring Agency Name and Address U.S. Department of Transportation Federal Railroad Administration Washington, D.C. 20590	13. Type of Report or Period Covered 14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

Tests were performed on the Peacekeeper Rail Garrison Maintenance Car (MC), according to specifications in Chapter XI of the AAR's, *Manual of Standards and Recommended Practices*. The unloaded MC was subjected to limited service worthiness testing. The tests, performed on the loaded MC, included vehicle characterization and track worthiness. These tests address vehicle safety performance for freight equipment. The primary measurement of safety as described in Chapter XI is the ratio of lateral to vertical wheel force (L/V ratio). No examination of ride comfort is addressed.

The MC performance in the Spiral Negotiation Test resulted in wheel unloading in the bunched spiral. The car body and trucks were unable to twist sufficiently to conform to the changing superelevation in the spiral. The MC also exceeded Chapter XI guidelines in the Dynamic Curving Test. The MC nearly exceeded Chapter XI guidelines in the Yaw and Sway Test with the perturbations at an amplitude of 1.0 inch rather than the specified 1.25 inches. It would most likely exceed Chapter XI guidelines if tested over 1.25-inch perturbations. The MC performed satisfactorily in the Pitch and Bounce Test, the Twist and Roll Test, and the Turnout and Crossover Test. The MC also negotiated 7.5- and 12-degree curves within Chapter XI guidelines. The unloaded MC showed no signs of lateral instability during the Hunting Test.

No suspension separation, wheel lift, or permanent deformation of the car body was seen during the Curve Stability Test which was performed on the unloaded MC.

17. Key Words	18. Distribution Statement
Chapter XI Vehicle Characterization Modal Response Track Worthiness Box Car	This document is available through National Technical Information Service Springfield, VA 22161

19. Security Classification (of the report)	20. Security Classification (of this page)	21. No. of Pages	22. Price
UNCLASSIFIED	UNCLASSIFIED		

Form DOT F 1700.7 (8-72)

EXECUTIVE SUMMARY

The Association of American Railroads (AAR), Transportation Test Center (TTC) conducted vehicle performance tests on a United States Air Force Peacekeeper Rail Garrison Maintenance Car following guidelines set forth in Chapter XI of the AAR's, M-1001, *Manual of Standards and Recommended Practices*. Measured performance met criteria described in Chapter XI for all tests except those titled Spiral Negotiation and Dynamic Curving.

The AAR, TTC, Pueblo, Colorado, has contracted with the Federal Railroad Administration (FRA) to perform vehicle performance tests on the Peacekeeper Rail Garrison (PKRG) rail cars according to specifications in Chapter XI, of the AAR's, M-1001, *Manual of Standards and Recommended Practices*.

The PKRG train will consist of two GP40 locomotives, a fuel car, a maintenance car, two security cars, two missile launch cars, and a launch control car. The overall objective of this test program was to examine the suitability of each PKRG car for railroad service through vehicle characterization, modeling, and static and dynamic on-track tests.

The fourth of the PKRG cars to be tested was the Maintenance Car (MC). The MC, which will carry the spare parts for the consist, is a four axle rail car with two 100-ton trucks, weighing 205,300 pounds. The car utilized concrete blocks bolted in a steel frame to simulate the actual "in service" weight. The car was manufactured by ITEL under contract to The Boeing Company.

Chapter XI states that values better than the criteria outlined in this report are regarded as indicating the likelihood of safe car performance. With the exception of dynamic curving and spiral negotiation, the loaded MC performed within the Chapter XI criteria. The main reason for acceptable MC performance was the truck spacing. Twist and roll, yaw and sway, and pitch and bounce contain perturbations of a 39-foot wavelength. It would be likely that a car with 39-foot truck spacing would be most sensitive to such perturbations or multiples of that wavelength. The truck spacing on this car was 64 feet. A wavelength of 39 feet was chosen to be most typical of excitation expected from the track due to the length of individual rail pieces in bolted track. Perturbations of other wavelength are possible but less likely. Multiples of 64 feet will provide more input to this car than the Chapter XI, 39-foot wavelengths.

By far, the most severe of all tests for the MC was the bunched spiral. At 25 mph the car experienced wheel lift. This was due to insufficient twisting of the car.

The MC exceeded Chapter XI criteria in dynamic curving primarily in the counter-clockwise direction. Exceedences in excess of 50 milliseconds were measured at speeds from 18 mph through 28 mph. The car curved better in the clockwise direction at speeds under 25 mph. This may have played a role in the dynamic curving directional dependency. The car also had some difficulty in curve entry. This may have played a role in the high L/V's in the counterclockwise direction as well. Soon after the car entered the 10-degree curve in that direction it encountered the dynamic curving perturbations as the test zone is at one end of the curve only. Even though the values were in excess of Chapter XI criteria, they were still low enough to allow the completion of the test at all speeds.

The MC barely stayed within Chapter XI guidelines in yaw and sway. However, the actual lateral perturbations over which the car was tested were 0.25 inches smaller in amplitude than those specified in Chapter XI. The car would probably have exceeded Chapter XI specifications if the perturbation amplitudes were 1.25 inches.

The car negotiated the 7.5- and 12-degree constant curve test zone within Chapter XI limits with the exception of the 12-degree clockwise run at 32 mph, which was not performed due to poor spiral negotiation performance at 25 mph.

The MC exhibited no lateral instability at speeds up to 70 mph. This was due, primarily, to the use of constant contact side bearings and due to the fact that the vehicle was tested loaded. Empty testing is specified by Chapter XI, however the United States Air Force chose to test loaded because the car should never be operated empty. The fact that the trucks were new and had tighter tolerances than worn trucks may have also been a factor.

The MC negotiated the turnout and crossover within the standard Chapter XI limits for wheel and axle L/V. This was a very important test, as switching is a frequent operation in a railroad environment.

The MC exhibited no suspension separation or wheel lift during the Service Worthiness Curve Stability Test.

The static brake tests showed that the MC braking system performance was within AAR specifications for the air brake system and the hand brake system.

The following recommendations are offered:

- Post test modeling should be performed to fine tune the computer simulation in light of measured car performance. This will validate the MC model for use in train modeling and future modeling for possible design changes or extension of results to regimes not tested.
- Post test modeling should be performed to examine car performance in yaw sway. The Yaw Sway Test should be modeled with the actual amplitudes in perturbations. If the model predictions match the test results, then predictions should be made with the Chapter XI specified perturbation amplitudes. This may indicate a possible need for design changes.
- Post test modeling should also include possible design changes for an improvement in spiral negotiation performance. If a solution is found, the OM car could be retrofitted and re-tested.
- If significant suspension changes are made, the car should be subjected to a limited Chapter XI re-test to include dynamic and constant curving, spiral negotiation, and hunting.

Table of Contents

Table of Figures

Tables

1.0 INTRODUCTION

The Association of American Railroads (AAR), Transportation Test Center (TTC), Pueblo, Colorado, has contracted with the Federal Railroad Administration (FRA) to perform vehicle performance tests on the Peacekeeper Rail Garrison (PKRG) rail cars according to specifications in Chapter XI, of the AAR's, M-1001, *Manual of Standards and Recommended Practices.*

These tests include static and quasi-static truck characterization, vehicle dynamic performance characterization, rail car service worthiness testing, and track worthiness testing.

The fourth of the PKRG cars to be tested was the Maintenance Car (MC). The MC, which will carry spare parts for the consist, is a four axle rail car with two 100-ton trucks, weighing 205,300 pounds. The car utilized concrete blocks bolted in a steel frame to simulate the actual "in service" weight. The car was manufactured by ITEL under contract to The Boeing Company.

Previously released reports concerning the MC testing, are Maintenance Car Vehicle Characterization Quick Look and MC Track Worthiness Quick Look. This report will encompass the entire MC test program.

2.0 OBJECTIVE

The overall objective of this test program was to examine the suitability of the PKRG MC for railroad service through vehicle characterization, modeling, and static and dynamic on-track tests.

To do this it was necessary to measure the static and quasi-static suspension characteristics of two 100-ton conventional three-piece trucks used under the MC. These parameters were required as input for mathematical models New and Untried Car Analytical Regime Simulation (NUCARS) and the Train Dynamics Model (TDM) used to predict individual rail car and full train performance respectively.

It was also necessary to measure the rigid body modal parameters of the MC to include bounce, pitch, roll, sway, and yaw. The first order flexible body modal parameters including vertical bending, lateral bending, and longitudinal body torsion (twist) were measured as well. These parameters were required for verification of the NUCARS model of the MC. They are also used to determine the vehicle natural frequency in important rail car dynamic modes. The frequency of the track perturbations and the car natural frequency enables calculation of the most critical test speed for that mode.

The service worthiness of the MC in curve stability was evaluated. The MC is not a new structural design so the impact, compressive end load, and jacking tests described in Chapter XI were not required.

Another further objective was to examine the vehicle dynamic performance in terms of track worthiness of the MC to include high speed stability (hunting), constant curving, curve entry and curve exit, pitch and bounce, twist and roll, dynamic curving, turnouts and crossovers, and yaw and sway. These characteristics were examined for the normal operating weight of 205,300 pounds which will be referred to as loaded.

3.0 TEST DESCRIPTION AND PROCEDURES

Chapter XI of the AAR's M-1001, *Manual of Standards and Recommended Practices* 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. Service Worthiness covers structural, static, and impact requirements. Track Worthiness covers vehicle dynamic performance. Chapter XI represents a realistic but severe environment for freight cars.

Vehicle characterization is used to predict the dynamic performance of freight cars. After the characteristics of the suspension and the car body system are determined, the results can be used to build a model to predict vehicle performance in the Chapter XI tests.

3.1 VEHICLE CHARACTERIZATION

Vehicle characterization, as set forth in Appendix A of Chapter XI, is divided into three groups: (1) static tests with wheels unrestrained, (2) quasi-static tests with restrained wheels, and (3) rigid and flexible body modal tests. These results allow comparison between measured and design values and are used as part of the NUCARS model input parameters.

3.1.1 Static Truck Characterization

Static tests with wheels unrestrained were conducted on air tables to determine rotational and longitudinal stiffnesses in the truck.

Static truck characterization was performed on the two 100-ton design trucks using air bearing tables. These tables utilize six air bearings to float an object off the ground on a cushion of air. This eliminates the friction between the wheels and the rail. Figure 3.1 shows an air bearing table.

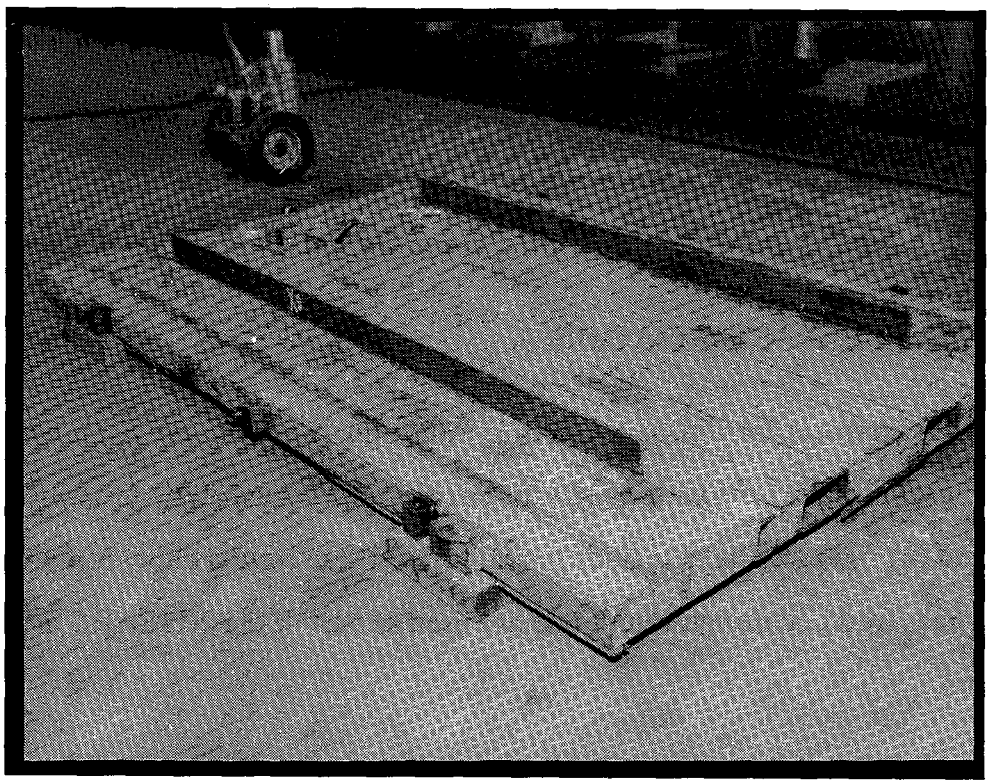


Figure 3.1 Air Bearing Table

The following tests were performed with air bearing tables:

- Truck Yaw Moment Test
- Axle Alignment Test
- Longitudinal Stiffness Test
- Inter-axle Yaw and Bending Test

3.1.1.1 Truck Yaw Moment Procedure

The Yaw Moment Test was done to determine the torque necessary to rotate the truck about the car body center plate. This break-away torque is related to the static friction between the center plate and center bowl and the friction between the car body and the constant contact side bearings. When the MC enters a curve, the wheel forces cause the truck to break away and rotate. The actuators and string pots were assembled on one table as shown in Figure 3.2.

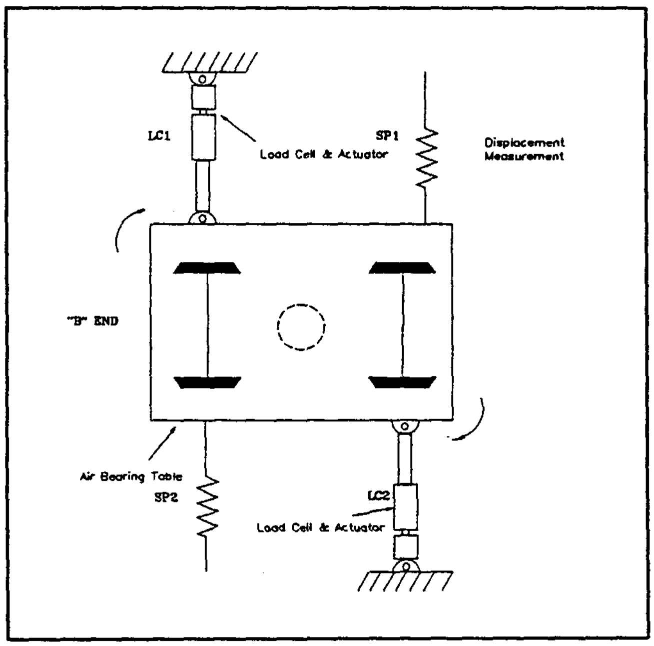


Figure 3.2 Truck Yaw Moment Test Setup

The trucks were displaced 1 inch, which equated to 27.8 milliradians (mrad). Each of the two trucks was tested in clockwise and counterclockwise directions by reversing the locations of the string pots and actuators.

3.1.1.2 Axle Alignment Procedure

The Axle Alignment Test was performed to determine the lateral and radial misalignment between the two axles in a truck. To allow each axle in the truck to align itself independently, two air tables were placed under one truck, one table under each axle. One optical transit and four precision scales were used in order to measure radial and lateral misalignments, in the arrangement shown in Figure 3.3.

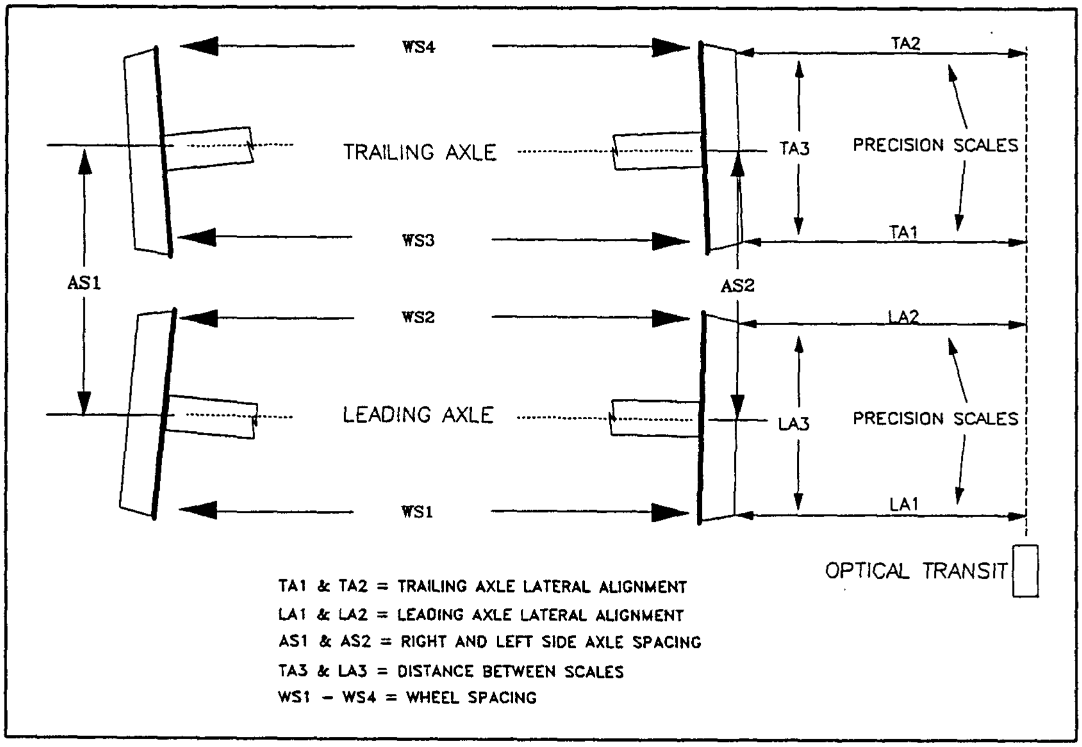


Figure 3.3 Axle Alignment Test Setup

Each time the tables were floated and set back down, the axle spacing on each side of the truck was measured. The scales were then put in place and the individual wheel mis-alignments calculated (Figure 3.4). The test was performed three times on each of the two trucks.

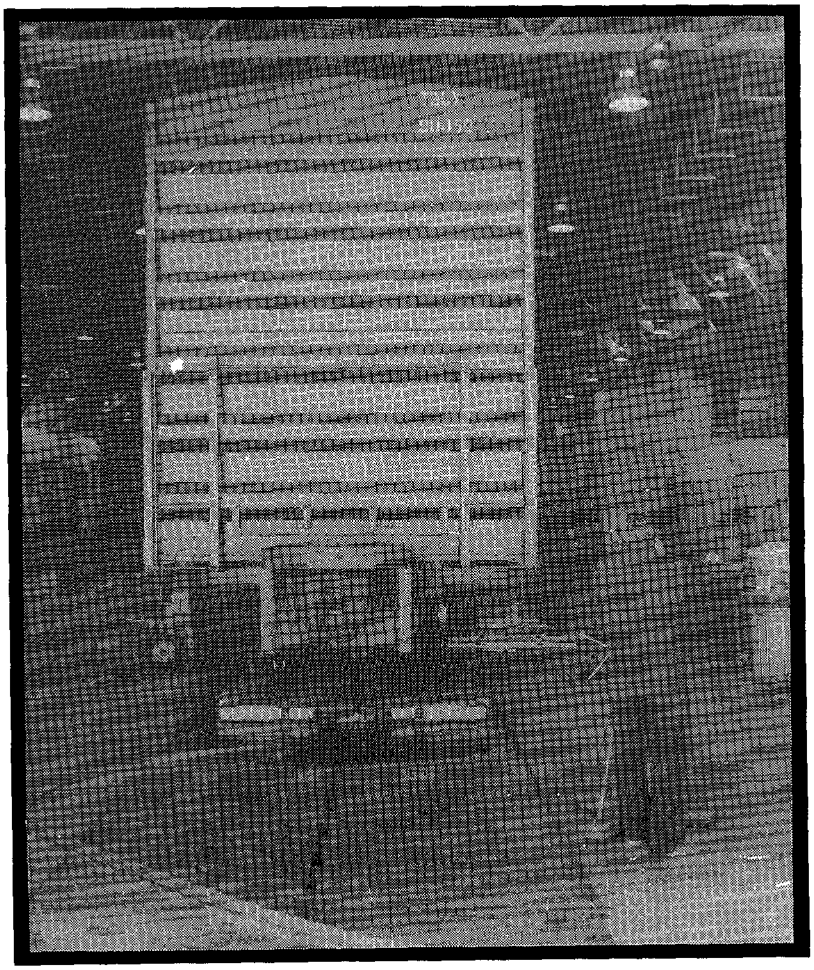


Figure 3.4 Axle Alignment Test

3.1.1.3 Longitudinal Stiffness Procedure

The air tables were left in the same configuration for longitudinal stiffness tests as they were in axle alignment tests. Actuators were connected between the ends of the axles on both sides of each truck via axle spuds bolted on the roller bearing end caps (Figure 3.5).

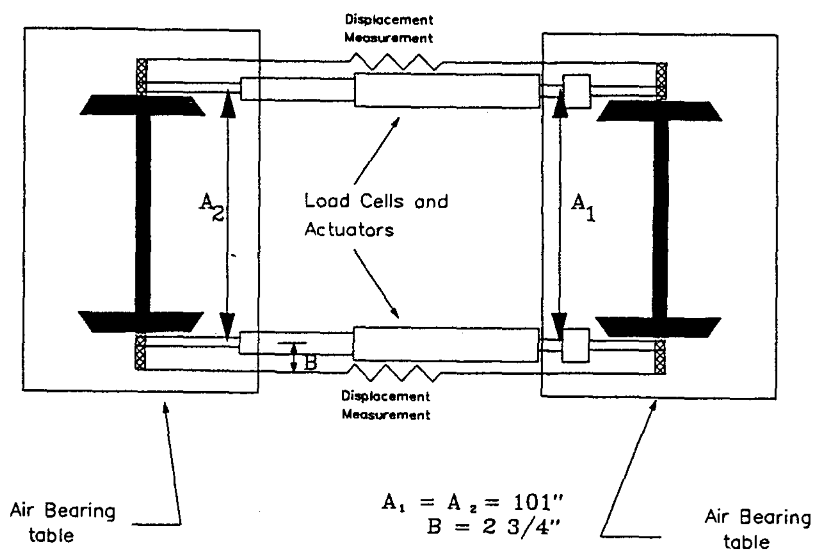


Figure 3.5 Truck Longitudinal Stiffness Test Setup

String pots were used to measure displacement between the two axles on each side of the truck. The axles were pushed apart and pulled together to determine the longitudinal stiffness. This test was repeated for the second truck.

3.1.1.4 Inter-axle Yaw and Bending Procedure

The Inter-Axle Yaw and Bending Test was performed in conjunction with the Longitudinal Stiffness Test. The axles were yawed by pushing them apart on one side of the truck while pulling them together on the opposite side of the truck. Figure 3.6 shows the actual test setup.

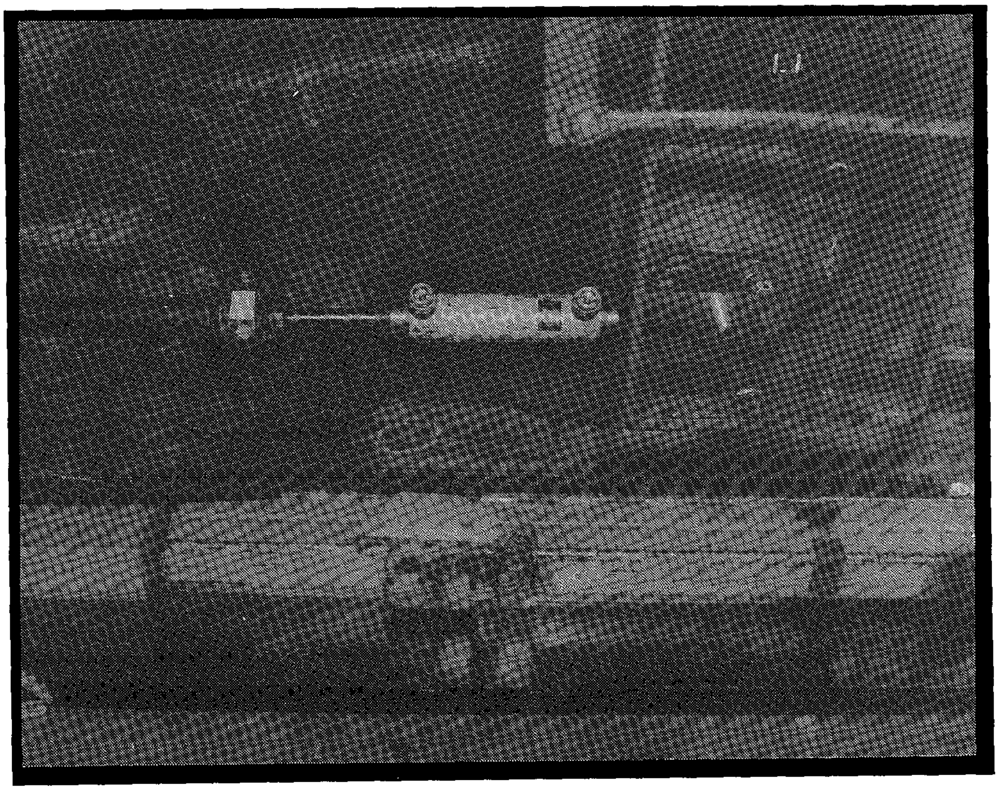


Figure 3.6 Longitudinal Stiffness and Inter-axle Yaw and Bending Stiffness Test Setup

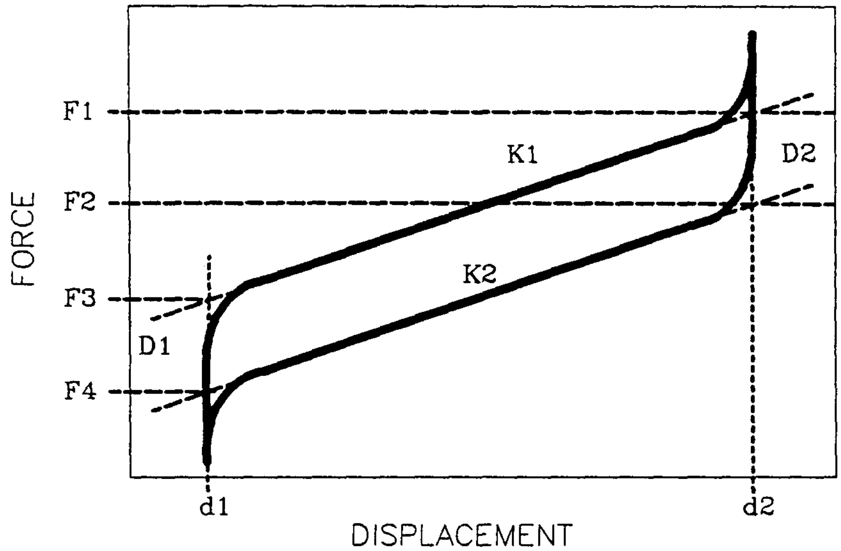
3.1.2 Quasi-Static Truck Characterization

Truck characterization tests were conducted to determine the dynamic suspension characteristics of the trucks. Quasi-static tests were conducted on the Mini-Shaker Unit (MSU) to measure the vertical and lateral displacement values at various truck component interfaces for given sinusoidal forces input at the car body sides.

Quasi-static truck characterization was performed on two 100-ton design "Ride Control" trucks. Each secondary suspension group was equipped with nine D-7 outer springs and five D-7 inner springs. The tests were performed on the MSU in the Rail Dynamics Laboratory (RDL). The following MSU tests were performed:

- Vertical Stiffness and Damping
- Lateral Stiffness and Damping
- Roll Stiffness

Stiffness and damping values for each component were obtained from force versus displacement plots as shown in the hysteresis curve in Figure 3.7. This stiffness is the slope of the characteristic lines. The damping is defined by the distance between the lines.



STIFFNESS	DAMPING
K1 = (F1 − F3) / (d2 − d1)	D1 = F3 − F4
K2 = (F2 − F4) / (d2 − d1)	D2 = F1 − F2

Figure 3.7 Stiffness and Damping From Typical Hysteresis Plot

The MSU utilized two 77 kip hydraulic actuators for vertical input excitation to the vehicle and, in separate testing, one 77 kip hydraulic actuator for lateral excitation. The actuators were attached to a reaction mass bolted to the floor of the RDL. The actuators were connected between the car body and the reaction mass with special brackets welded to the MC. Sinusoidal input signals were provided to the actuator control valves with a Hewlett-Packard (HP) 360 desktop computer teamed with a programmable function generator. The actuators were controlled with 0.1 and 0.25 Hz signals during the quasi-static tests. Each of the two 100-ton design trucks were individually tested under the A-end of the MC.

3.1.2.1 Vertical Stiffness and Damping Procedure

The Vertical Stiffness and Damping Test was conducted by cycling both vertical actuators in-phase at frequencies of 0.1 and 0.25 Hz. The actuators were extended and retracted to the full extent of the truck spring travel and to various levels below the maximum spring travel. It was determined, during the tests, that approximately ± 2 inches of actuator travel was sufficient to fully extend and compress the springs. Figure 3.8 shows the MSU in the vertical configuration.

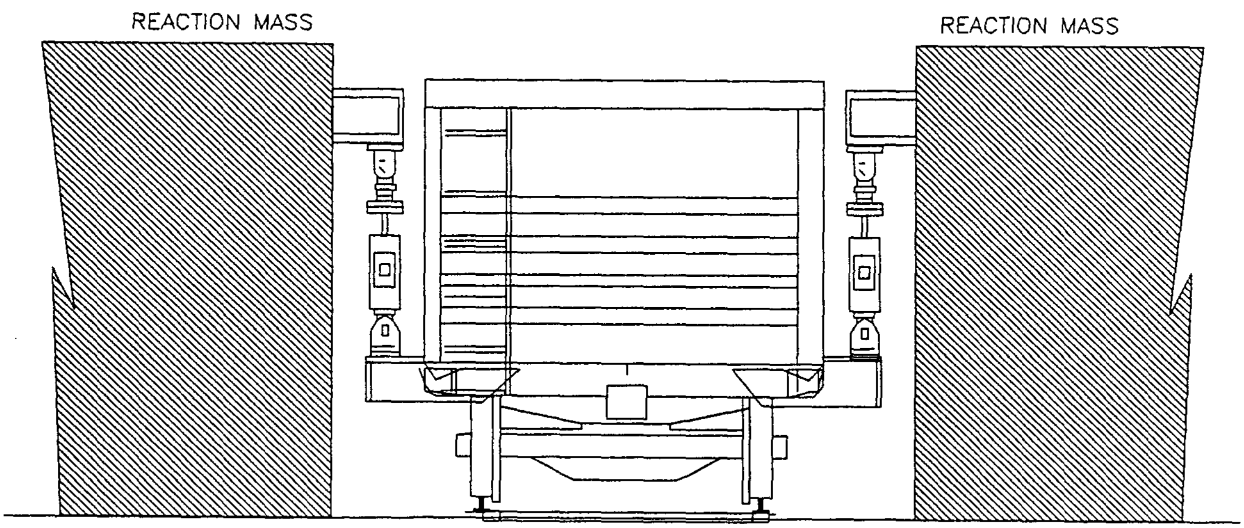


Figure 3.8 MSU in The Vertical Configuration

3.1.2.2 Roll Stiffness Procedure

The Roll Stiffness Test was very similar to the vertical test, with the exception that the vertical actuators were operated 180 degrees out-of-phase. Actuator displacements up to ±2 inches were tested.

3.1.2.3 Lateral Stiffness and Damping Procedure

The Lateral Stiffness and Damping Test required reconfiguration of the MSU to a single lateral actuator arrangement. The input was cycled at 0.1 and 0.25 Hz in the range from ±10 kips to ±1/5 the vertical static load of the car on that truck (±20 kips), which is the AAR Chapter XI criterion. Figure 3.9 shows the MSU in the lateral configuration.

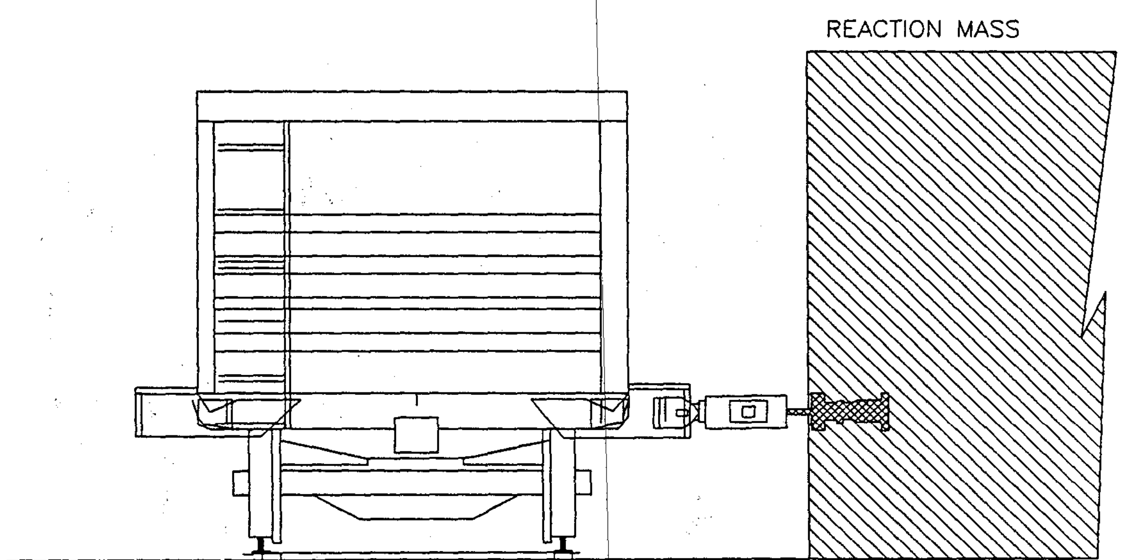


Figure 3.9 MSU in the Lateral Configuration

3.1.3 Modal Response

Modal characterization testing was conducted to determine the dynamic characteristics of the suspension and the car body as a system. The results of the rigid body tests were used as a comparison to the NUCARS prediction. The flexible body results were to be used as input parameters for the TDM. Vehicle characterization is necessary if the modeling results are to be extrapolated to include any conditions which were not tested, such as changes in car design, operation, or component degradation.

Modal response testing was performed on the MSU in the RDL during quasi-static truck characterization. The MC is on the MSU in Figure 3.10.

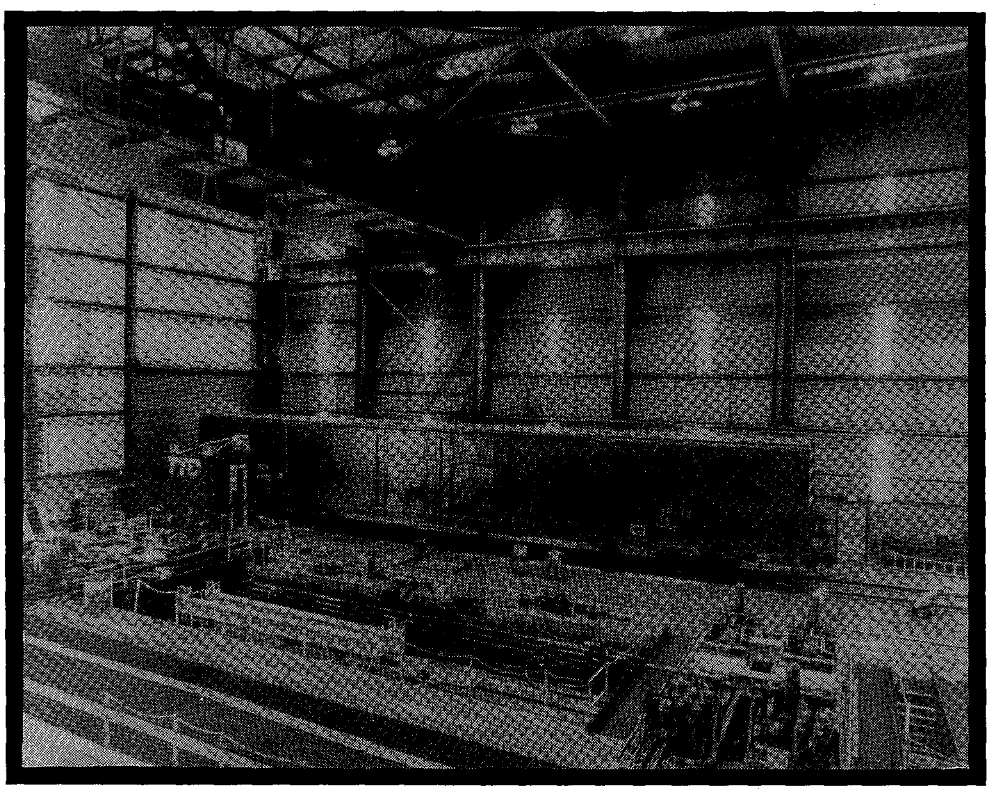


Figure 3.10 MC on the MSU

Modal response tesing was performed on the MC to determine the natural frequencies for the following modes:

Rigid Body Modes	Flexible Body Modes
•Pitch	•Vertical Bending
•Bounce	•Lateral Bending
•Roll	•Torsion (Twist)
•Yaw	
•Sway	

3.1.3.1 Rigid Body Vertical Procedure

The MSU was set up in the vertical test configuration for the rigid body vertical tests. The actuators were cycled in-phase with 5, 10, and 15 kip sinusoidal inputs. The frequency increased from 0.2 Hz to 10 Hz in 0.1 Hz steps with 10 cycles per step. Pitch and bounce were determined by the amplitude of the transfer function between the input force and accelerations measured along the car body and by the phase relationship of accelerations at the ends of the car body.

3.1.3.2 Rigid Body Roll Procedure

The MSU setup remained in the vertical configuration for the rigid body roll tests. The same procedure that was used during rigid body vertical testing was used with the actuators cycled 180 degrees out-of-phase. A roll frequency was determined by the amplitude of the transfer function between the input force and acceleration measured along the car body and by the phase relationship of accelerations at the right and left side of the car body.

3.1.3.3 Flexible Body Vertical Procedure

The MSU remained in the vertical test configuration for flexible body vertical testing. The actuators were cycled in-phase but they were in displacement control rather than force control. Displacement control was used for a constant acceleration (g) input. The actuators were swept from 3 Hz to 30 Hz at constant g of 0.1, 0.2, and 0.3. The displacement was decreased as the frequency increased to maintain constant acceleration. The frequency steps and numbers of cycles were calculated within the MSU control program and were unique for each test. Additional sweeps of 0.4g at 5Hz to 30 Hz and 0.5g at 10Hz to 30 Hz were also performed.

3.1.3.4 Flexible Body Twist Procedure

The Flexible Body Twist Test was performed in the vertical configuration. The inputs were identical to the Flexible Body Vertical Test; except, the actuators were cycled 180 degrees out-of-phase.

3.1.3.5 Rigid Body Lateral Procedure

The MSU was reconfigured to the lateral position for the Rigid Body Lateral Tests. Sinusoidal inputs of 5, 10, and 15 kips from 0.2 Hz to 10 Hz in 0.1 Hz steps at 10 cycles per step, were provided to the actuator control valve for input in to the MC. Yaw and sway frequencies were determined by the transfer function to input force and by the phase relationships between lateral displacements and accelerations at each end of the car body.

3.1.3.6 Flexible Body Lateral Procedure

The Flexible Body Lateral Test was performed with constant g inputs of 0.1, 0.2, and 0.3, from 3 Hz to 30 Hz and 0.4 g at 5 Hz to 30 Hz in a similar manner as the vertical tests.

3.2 SERVICE WORTHINESS

Service worthiness testing usually consists of four separate tests:

- Single Car Impact Test
- Compressive End Load Test
- Jacking Test
- Curve Stability Test

Since the MC was a conventional structural design, the impact, compressive end load, and jacking tests were not required.

3.2.1 Curve Stability Procedure

The Curve Stability Test was done to monitor car body suspension separation and wheel lift while the car was subjected to a static buff and draft (compression and tension) force. The test was conducted on a section of curved track with a limited amount of superelevation. Extremely short and long cars were connected adjacent to the MC to simulate the worst case situation.

The Curve Stability Test was conducted with the MC in the unloaded condition. The south wye of the Urban Rail Building (URB) at TTC was used for the test. The wye is a 10-degree curve with less than 0.5 inches of superelevation. The MC was subjected to static buff and draft loads of 200,000 pounds while being held in place for 20 seconds. The MC was monitored for wheel lift or any separation of the trucks and car body. Figures 3.11 and 3.12 show the Curve Stability Test.

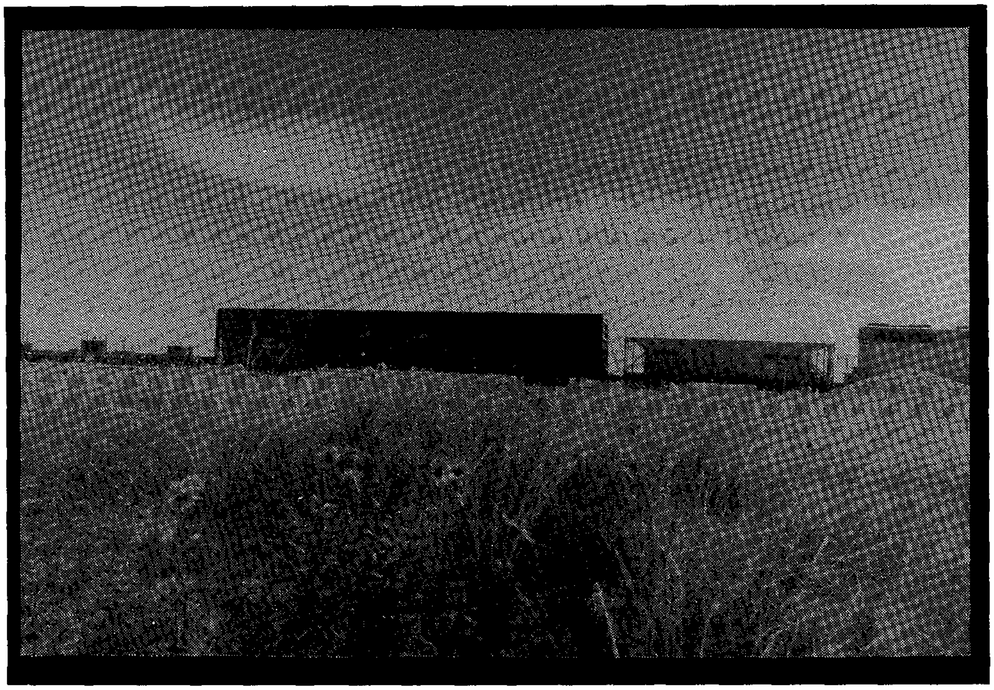


Figure 3.11 Curve Stability Test from Outside of Curve

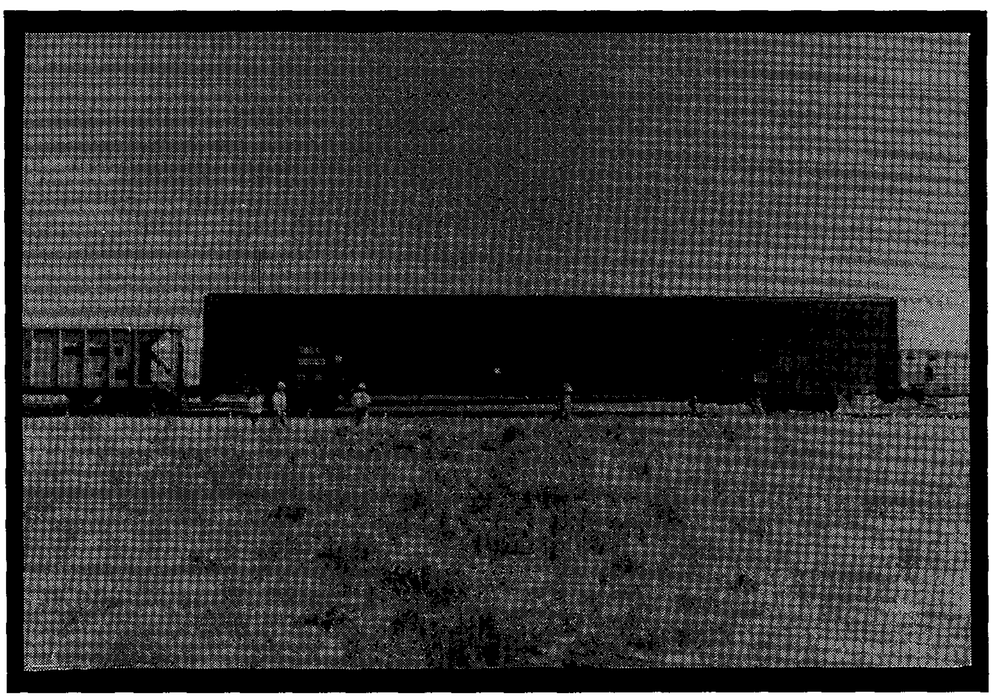


Figure 3.12 Curve Stability Test Showing MC from Inside of Curve

3.3 TRACK WORTHINESS

Track worthiness testing was conducted to assure an adequate margin of performance safety in normal operation of the rail car (60 mph). At the direction of the United States Air Force (USAF) track worthiness tests were conducted up to normal operating speeds of the car. The normal procedure is to test to 70 mph for all tangent track tests. All tests utilized instrumented wheel sets to measure lateral and vertical forces between the wheel and rail. These wheel sets have modified Heuman profiles which simulate worn wheels.

Two of the tests were performed on nominal track; hunting and constant curving. The remaining tests were performed on tracks intentionally misaligned to excite vehicle dynamic modes commonly associated with poor vehicle performance. A successful vehicle will be able to control vehicle response to these inputs.

Results were compared to the criteria stated in Chapter XI. The primary criteria are tendency to wheel climb derailment, as defined by the ratio of lateral to vertical wheel forces and tendency to cause rail rollover, as defined by the ratio of truck side lateral to vertical forces.

Predictions of MC performance were to be made with the NUCARS model using vehicle characterization data. Due to the tight PKRG test and TDM development schedule, it was not possible to make NUCARS predictions.

Track worthiness testing required specific buffer cars adjacent to the MC. The front buffer car was the T-7 Instrumentation Car and the rear buffer car changed, depending on the particular test. Figure 3.13 shows the core consist with a buffer car.

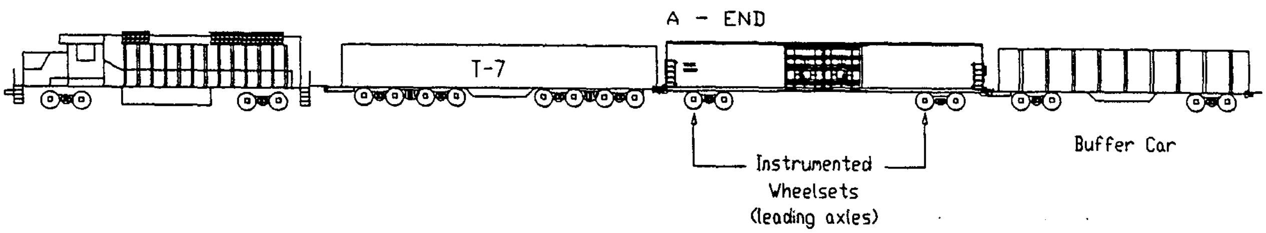


Figure 3.13 Track Worthiness Testing Core Consist

The MC track worthiness testing consisted of nine separate tests. All of the tests were conducted on TTC track with the car in the loaded configuration under which it will operate in actual service. Figure 3.14 is a track location diagram, the specific maps for each test are found in individual test description Sections 3.3.1 through 3.3.8. The normal upper limit speed for Chapter XI tangent track testing is 70 mph. The USAF limited testing for this car to 60 mph for all tests except one. The hunting test was performed at 70 mph.

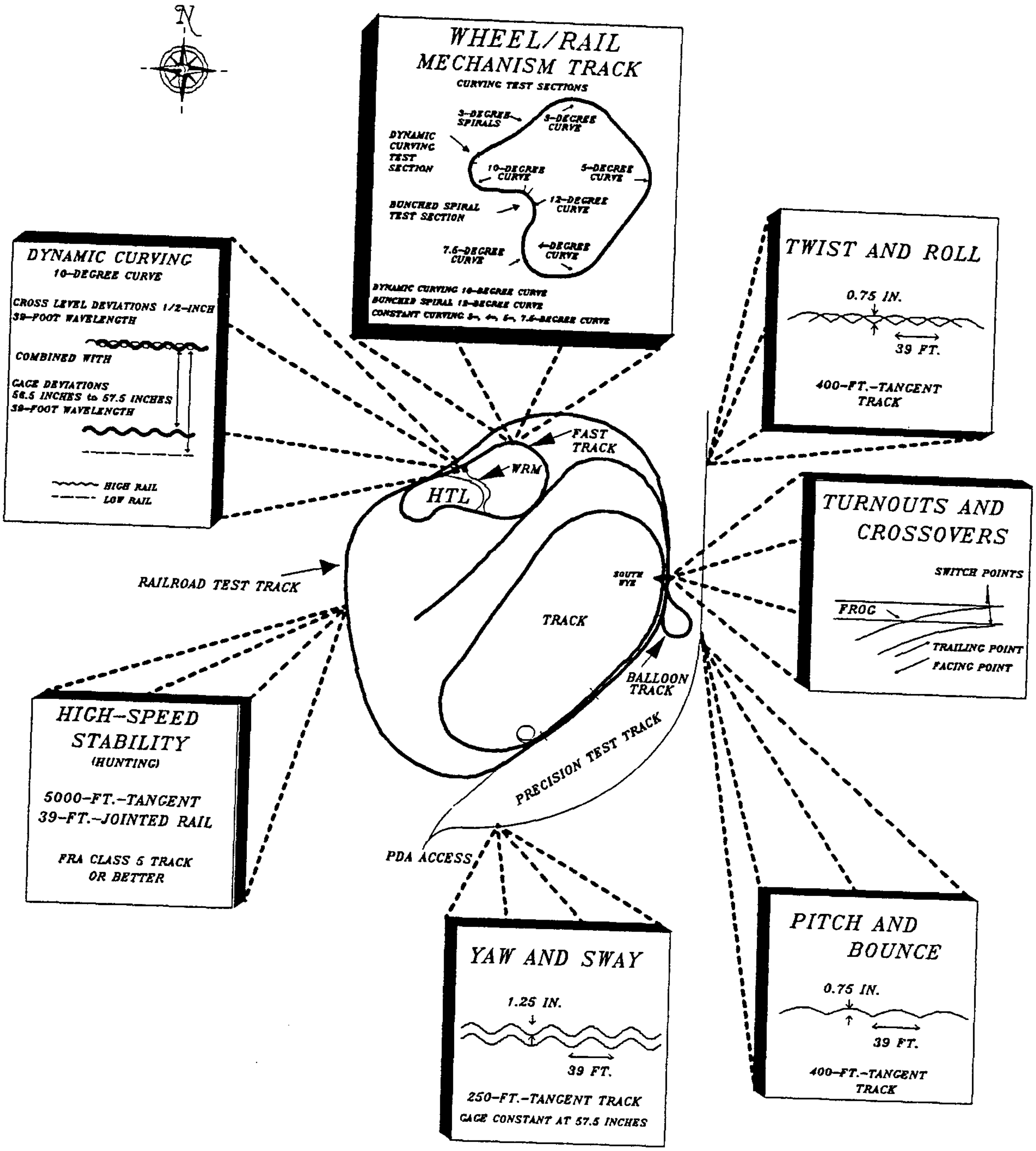


Figure 3.14 Track Location Diagram

Instrumented wheel sets were installed under the leading truck of the MC for yaw and sway (Figure 3.15).

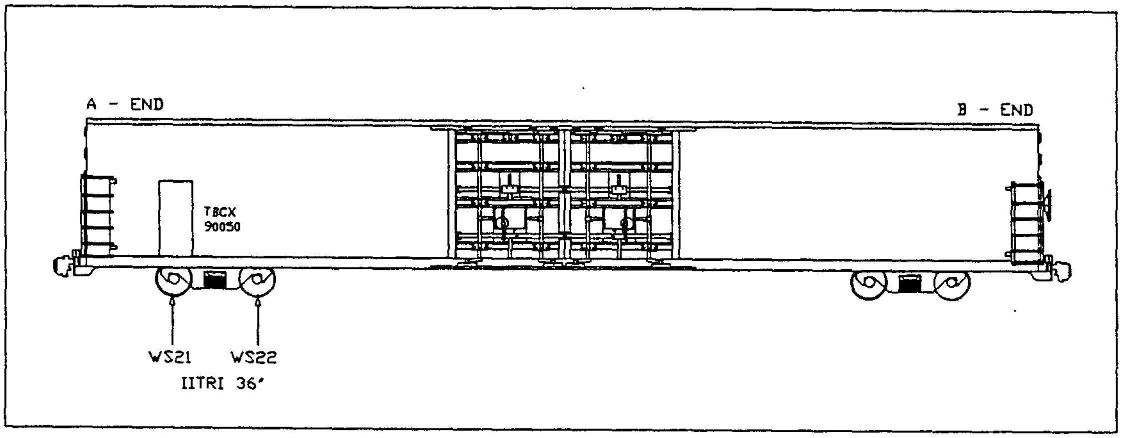


Figure 3.15 Yaw and Sway Instrumented Wheel Set Locations

The instrumented wheel sets were installed under the MC leading axle of each truck for all tests except yaw and sway (Figure 3.16).

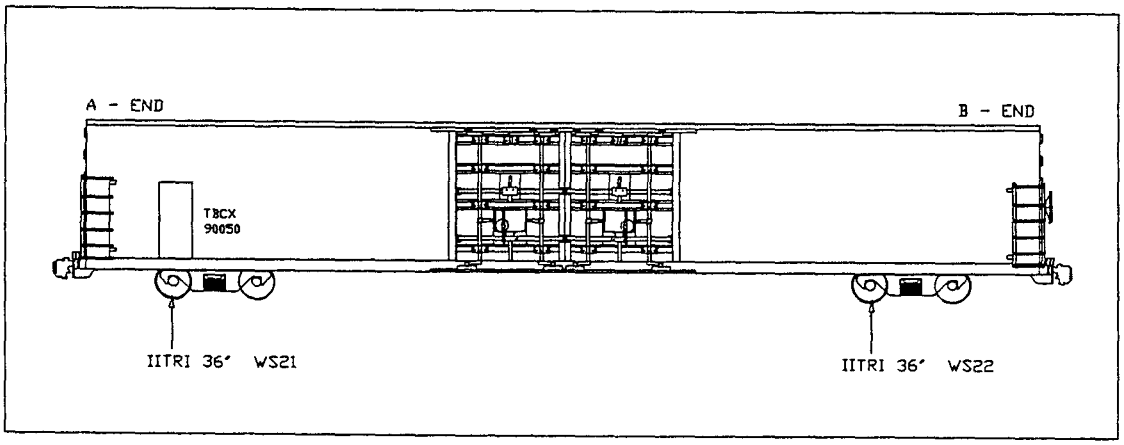


Figure 3.16 Instrumented Wheel Set Locations

3.3.1 Yaw and Sway Procedure

Yaw and sway testing was conducted to determine the ability of the car to negotiate laterally misaligned track. Track that generates input of this type may be found where the underlying ground is unstable and allows the track to shift in the lateral direction. The limiting truck side L/V criterion is 0.6 and the maximum axle sum L/V criterion is 1.3 sustained for 50 milliseconds (Chapter XI). The instrumented wheel sets were relocated to the leading truck to measure truck side L/V.

The Yaw and Sway Test was conducted in accordance with Section 11.6.4. Station 21+00 to 26+00 of the Precision Test Track (PTT) was the test site. This section had sinusoidal track alignment deviations of 39-foot wavelength and an amplitude of 1.0 inches peak-to-peak on both rails at a constant wide gage of 57.5 inches. These amplitudes were less than the 1.25 inches specified in Chapter XI. This was known before testing began, but it was impractical to adjust the perturbations due to cost and schedule. No trailing buffer car was used in the test consist. Figure 3.17 shows the test zone with 1.25-inch perturbations.

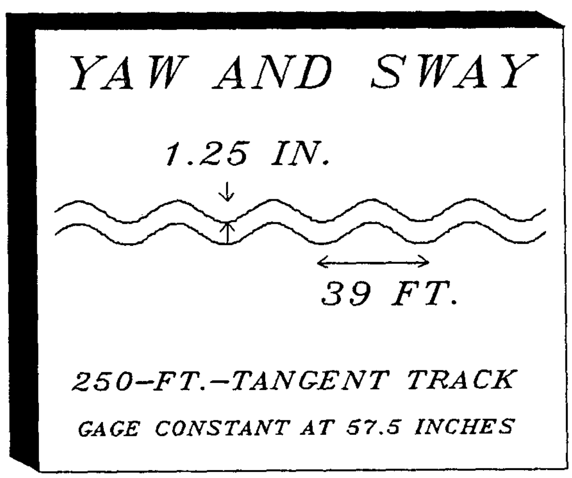


Figure 3.17 Yaw and Sway Test Facility

3.3.2 High Speed Stability Procedure

High speed stability testing was conducted to confirm that hunting (lateral oscillating instability in the trucks) did not occur within normal operating speeds of the car. Chapter XI limited the maximum lateral car body acceleration to 1.0 g peak-to-peak, sustained for 20 seconds or a single peak-to-peak occurrence of 1.5 g. The maximum axle sum L/V is limited to 1.3 sustained for 50 milliseconds. Hunting is inherent in typical railroad freight truck designs and is also seen in normally stable truck designs when components are allowed to wear beyond normal limits. If hunting occurs, the resonant speed is identified for operational considerations.

The MC was loaded and no trailing buffer car was used in the Hunting Test. The consist was operated at speeds up to 70 mph on 5,000 feet of tangent track with 39-foot jointed rail, Class 5 or better. Axle sum L/V's and car body lateral accelerations were monitored for any unsafe conditions. Figure 3.18 shows the hunting test track details.

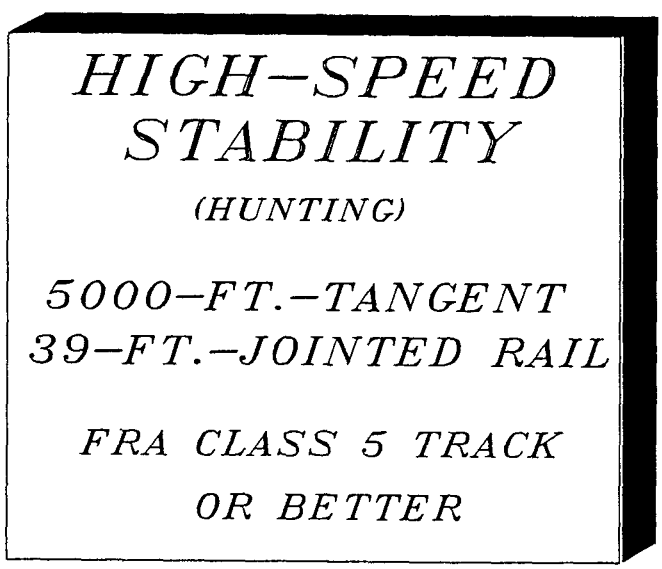


Figure 3.18 Hunting Test Track

3.3.3 Pitch and Bounce Procedure

The Pitch and Bounce Test was designed to determine the dynamic pitch and bounce response of the car as it is excited by vertical inputs from the track. Track which generates this type of input may be found at bridges, road crossings, and where there is a change in the underlying vertical support structure to the track. This phenomenon can also occur when rail joints on both rails are in-phase. The Chapter XI criterion is a minimum vertical wheel load of 10 percent of the static vertical wheel load sustained for 50 milliseconds. Figure 3.19 is a description of the test zone.

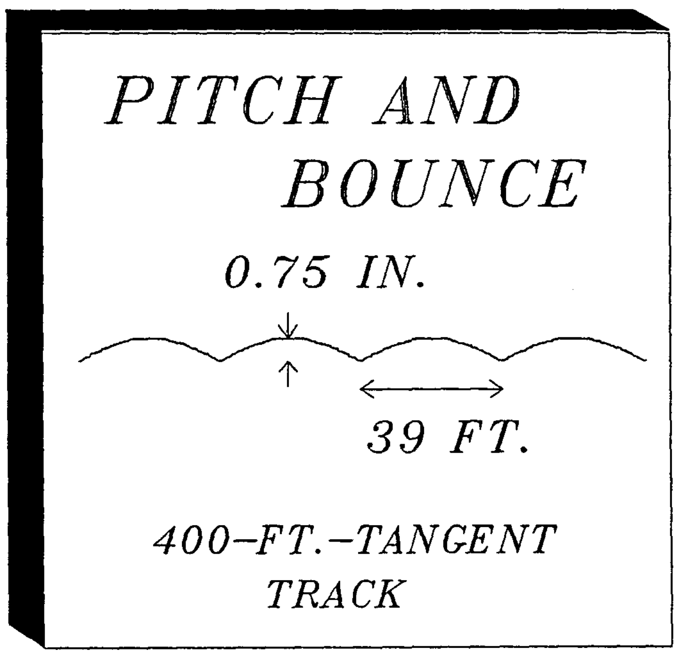


Figure 3.19 Pitch and Bounce Facility

The Pitch and Bounce Test was conducted on the PTT. The loaded MC was tested at speeds up to 60 mph on parallel jointed track with 0.75 inch vertical perturbations at 39-foot intervals in both rails. The Pitch and Bounce Test consist included a lightly loaded flatcar as a trailing buffer car per Chapter XI specifications. Figure 3.20 shows the test consist negotiating the pitch and bounce test zone

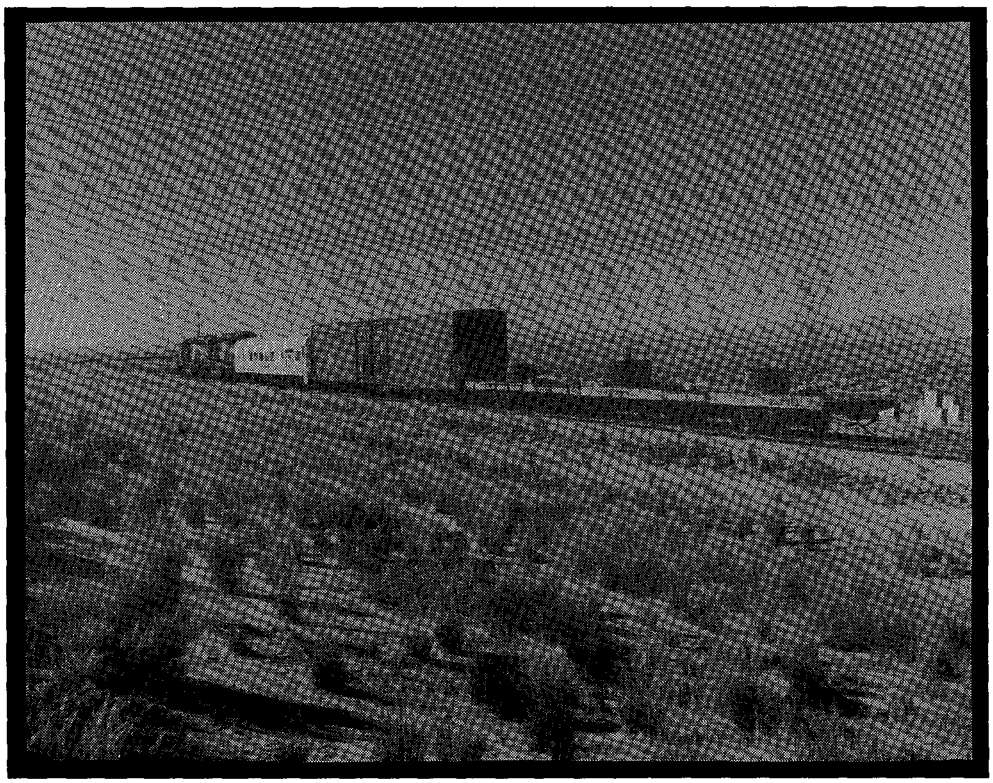


Figure 3.20 Pitch and Bounce Test Consist

3.3.4 Twist and Roll Procedure

The Twist and Roll Test was conducted to determine the car's ability to negotiate cross-level perturbations. These perturbations excite the natural twist and roll motions of the car. This type of track input may be found in locations where rail joints are staggered up to 180 degrees out-of-phase or at weak spots in the track structure. Three criteria were given for this test: (1) a maximum roll angle of 6 degrees peak-to-peak, (2) a maximum axle sum L/V of 1.3 sustained for 50 milliseconds, and (3) a minimum vertical wheel load of 10 percent of the static vertical wheel load sustained for 50 milliseconds (Chapter XI). Figure 3.21 describes the test zone.

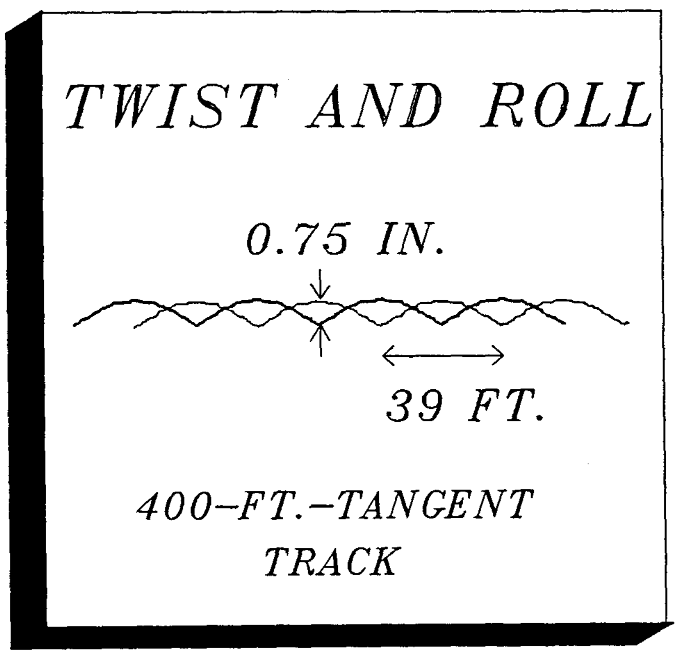


Figure 3.21 Twist and Roll Test Facility

The Twist and Roll Test was conducted on the PTT with the loaded car. The MC was tested up to 60 mph on staggered jointed rail with a cross-level of 0.75 inches at 39-foot intervals. Chapter XI specified a loaded buffer car with a truck spacing greater than 45 feet. Figure 3.22 shows the test consist approaching the twist and roll test zone with a lightly loaded flatcar as the trailing buffer car.

Figure 3.22 Twist and Roll Test Consist

3.3.5 Turnout and Crossover Procedure

The Turnout and Crossover Test is not listed in Chapter XI as an official test but was conducted to verify the operation of the vehicle through standard crossovers and turnouts with a margin of safety in wheel/rail forces. A turnout is an arrangement of a switch and a frog with closure rails, by which cars may be diverted from one track to another. The wheel/rail forces would indicate if there was a tendency for (1) wheel climb or (2) to induce large lateral forces into the track.

Testing was performed on the north turnout entering and exiting the URB (Switch No. 704). Chapter XI criteria for axle and wheel L/V's (1.3 and 0.8 respectively) were used as the limitation for proper operation. Figure 3.23 shows the turnout test facility.

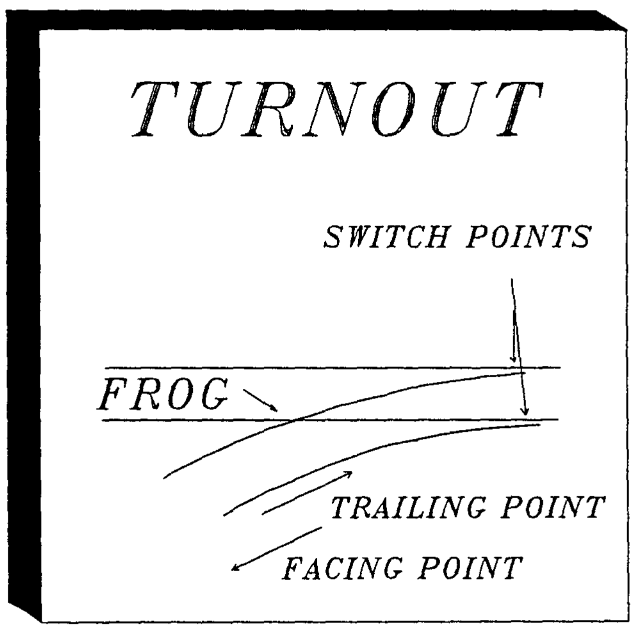


Figure 3.23 Turnout Test Facility

A crossover is an arrangement of two turnouts with track between the frogs arranged to allow passage between two parallel tracks. The Crossover Test was performed on switches 602-A and B between the Railroad Test Track (RTT) and the Transit Test Track (TTT) while operating in both north and south directions. The flatcar was used as the trailing buffer car for turnout and crossover testing. Figure 3.24 shows the crossover test facility.

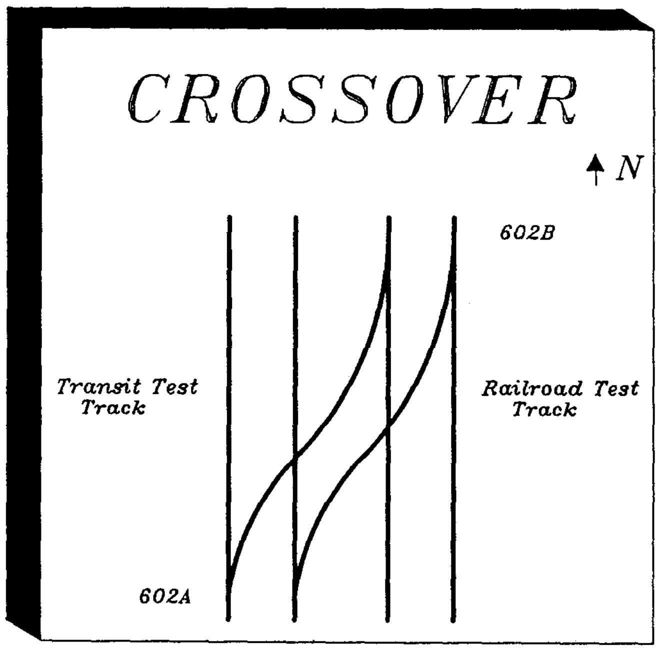


Figure 3.24 Crossover Test Facility

3.3.6 Dynamic Curving Procedure

The Dynamic Curving Test was designed to determine the ability of the car to negotiate track with simultaneous cross-level (vertical) and gage (lateral) misalignments. Four criteria were given in Chapter XI: (1) a maximum wheel L/V of 0.8, (2) a maximum axle sum L/V of 1.3, (3) a maximum roll angle of 6-degrees peak-to-peak, and (4) a minimum vertical wheel load of 10 percent of the static vertical wheel load. The 50 millisecond criteria applied to all but roll angle. Figure 3.25 outlines the Dynamic Curving Test Track.

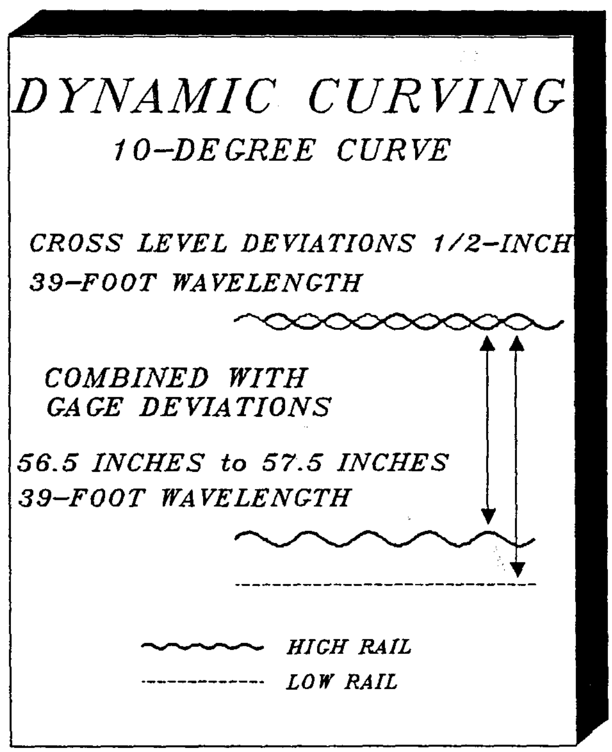


Figure 3.25 Dynamic Curving Test Facility

The Dynamic Curving Test was conducted on the 10-degree curve of the Wheel Rail Mechanisms (WRM) track with a loaded MC. The 10-degree curve was shimmed to provide cross level deviations of 0.5 inches combined with lateral perturbations which resulted in a maximum gage of 57.5 inches and a minimum gage of 56.5 inches. A 100-ton loaded gondola was used as the trailing buffer car (see Figure 3.26).

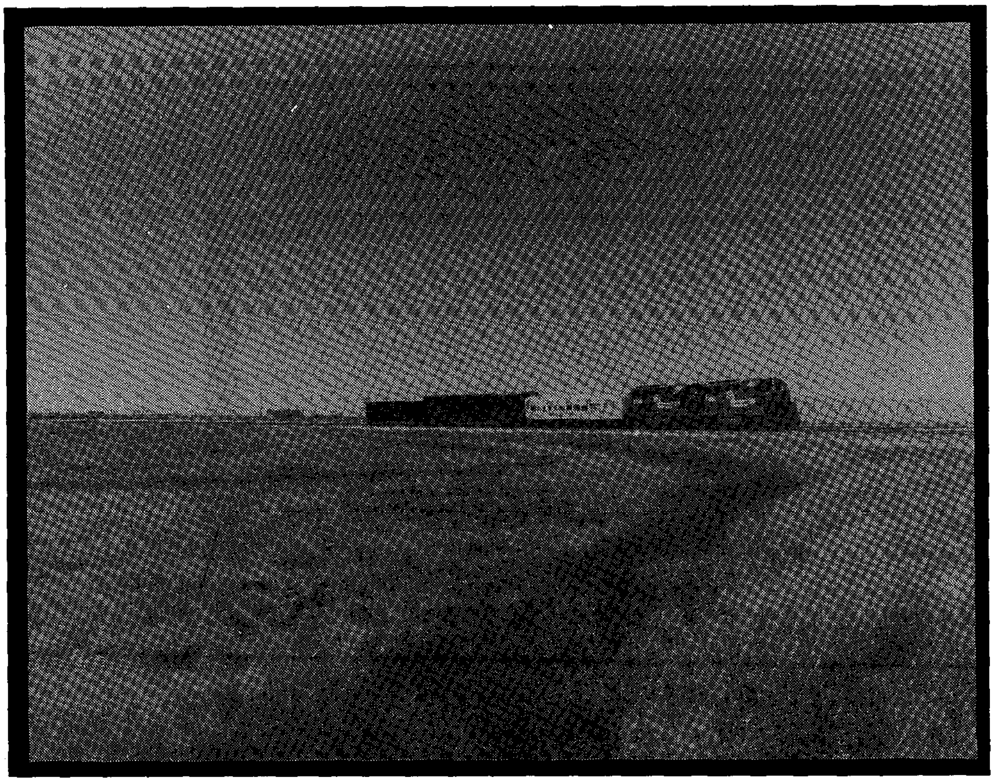


Figure 3.26 Dynamic Curving Test Consist

3.3.7 Constant Curving Procedure

The constant curving tests were designed to determine the car's ability to negotiate well maintained track curves. The test car was operated through nominal curves at typical operating speeds in the loaded condition. The 95th percentile maximum wheel L/V of 0.8 or axle sum L/V of 1.3 (Chapter XI, Table 11.1) was the limiting criteria. This test would verify that the car would not have wheel climb or impart large lateral forces to the rails during curving. The dynamic curving buffer car shown in Figure 3.26 was also used in constant curving and spiral negotiation. Figure 3.27 shows the WRM test track and associated curvatures.

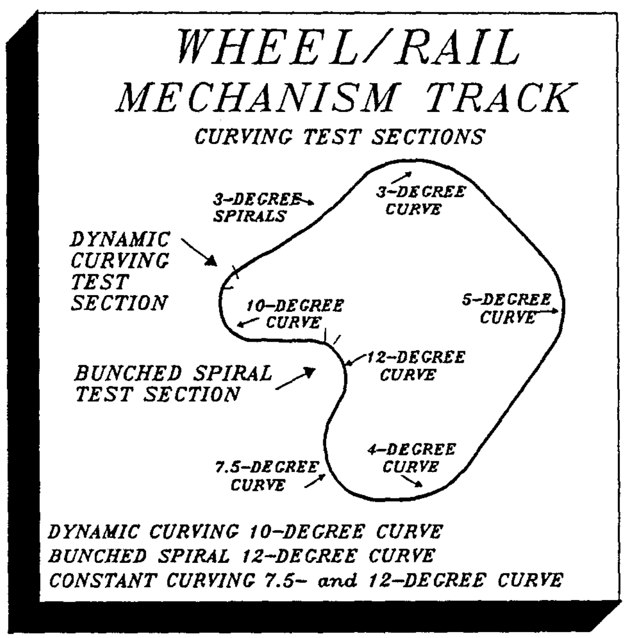


Figure 3.27 Constant Curving Test Facility

The MC was operated on various degrees of curvature and superelevation available on the WRM at speeds corresponding to 3 inches underbalance, balance, and 3 inches overbalance. The tests were run in both clockwise and counterclockwise directions. Wheel L/V's were monitored real time to ensure safe test operation. Table 3.1 is a tabulation of the balance speeds for the test curves on the WRM.

Table 3.1 WRM Curve Descriptions and Test Speeds

DEGREE OF CURVE	SUPER ELEVATION (inches)	BALANCE SPEED (mph)	+3 INCH SPEED (mph)	-3 INCH SPEED (mph)
7.5	3	24	32(34)	14(0)
10	4	24	32(32)	12(12)
12	5	25	32(31)	16(16)

Note: Speeds in ( ) are calculated, others are actual test speeds.

Test speeds were determined using the following equation:

$$V = \sqrt{1480 \frac{(U+H)}{D}}$$

Where: U = unbalance in inches

H = superelevation in inches

D = degree of curvature

V = speed in mph.

In some cases the track speed limit was lower than the calculated speed for +3 inches. A track speed limit of 32 mph for the 7.5-, 10-, and 12- degree curves was used in those cases. The speed calculated for -3 inches was zero or not possible in some cases (curves with less than 3 inches of superelevation). The following equation shows the method of test speed calculation for those cases.

$$V^*_{-3} = V_0 - (V_{+3} - V_0)$$

3.3.8 Spiral Negotiation Procedure

The spiral negotiation or curve entry and curve exit tests were performed in conjunction with the Constant Curving Test. A spiral is the transition from a curve to a tangent track. This transition includes constant rates of change in cross-level and curvature with distance. The purpose of the exaggerated bunched spiral is to twist the trucks and the car body. Chapter XI states that the minimum acceptable vertical load of a wheel is 10 percent of the static wheel load and that the maximum wheel L/V is 0.8, both sustained for 50 milliseconds. This data was monitored to verify that no wheel lift occurred and that no extreme wheel forces were measured.

The Spiral Negotiation Test was conducted on the WRM track with the MC in the loaded configuration. The Chapter XI specified bunched spiral section is found at the end of the 12-degree curve during clockwise operation. Tests were done at the same speeds as the Constant Curving Test and in both the clockwise and counterclockwise directions. Curve entry and exit performance was also examined for the 7.5- and 12-degree curves even though Chapter XI only specified the bunched spiral. Single wheel L/V's and axle L/V's were monitored for any unsafe condition.

3.4 STATIC BRAKE TEST

A static brake test was conducted to determine the static forces on the brake shoes at various brake cylinder pressures. This information was compared to accepted standards and was also used to ensure the compatibility between all car brake systems in the PKRG train.

The Static Brake Test was performed by the AAR with assistance from Blaine Consulting Services . The brake test was performed to ensure compliance with existing AAR and FRA Rules and Regulations. A single car test was performed on the MC, following specifications from the Westinghouse Air Brake Company instruction pamphlet entitled, *Single Car Testing Device Code of Tests for Freight Equipment*, No. 5039-4 Sup. 1, Standard S-486 April 1987.

Next, the Net Shoe Force Test was performed. Instrumented brake shoe load cells were installed at each wheel/brake interface on the A-end of the MC. Brake shoe forces were read from a digital readout for a series of different brake pipe reductions. A hand brake net shoe force test was performed while the instrumented brake shoes were in the A-end truck. The hand brake was applied in 1,000 pound (horizontal chain force) increments and brake shoe forces were measured and recorded. Both tests were repeated at the B-end. Figure 3.28 shows AAR's consultant operating the single car test device.

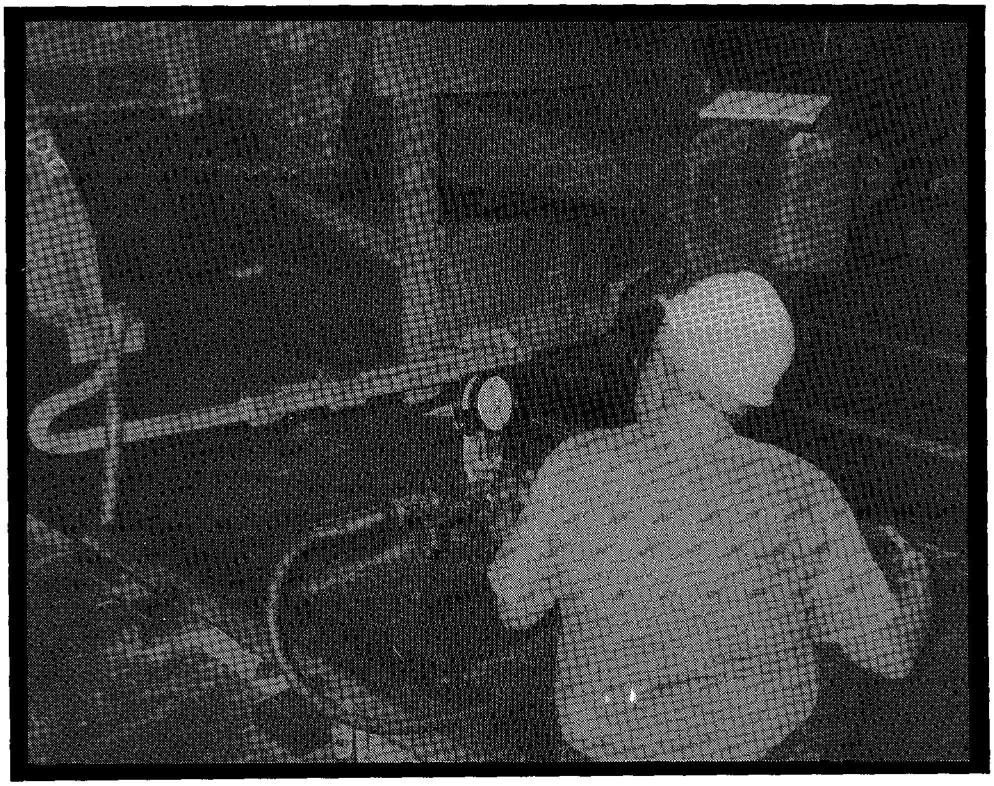


Figure 3.28 Static Brake Test with Single Car Test Device

4.0 TEST VEHICLES

4.1 MAINTENANCE CAR DESCRIPTION

The test vehicle was the PKRG Engineering Model (EM-1) of the MC numbered TBCX90050. Figures 4.1 and 4.2 show the MC from the side and B-end, respectively.

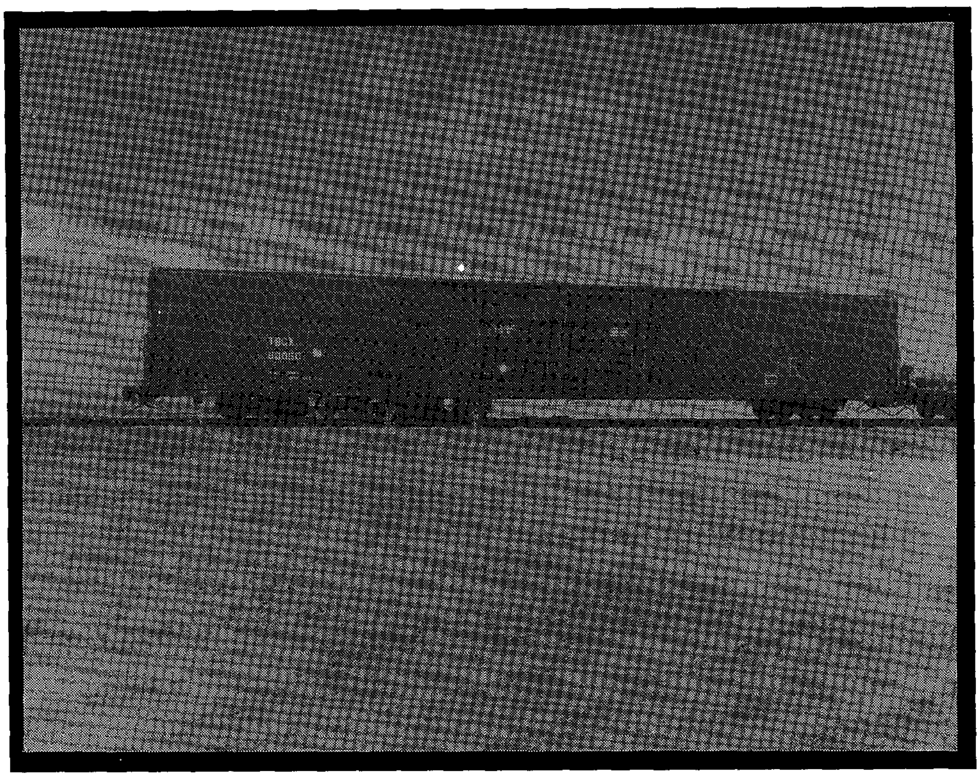


Figure 4.1 MC Side View

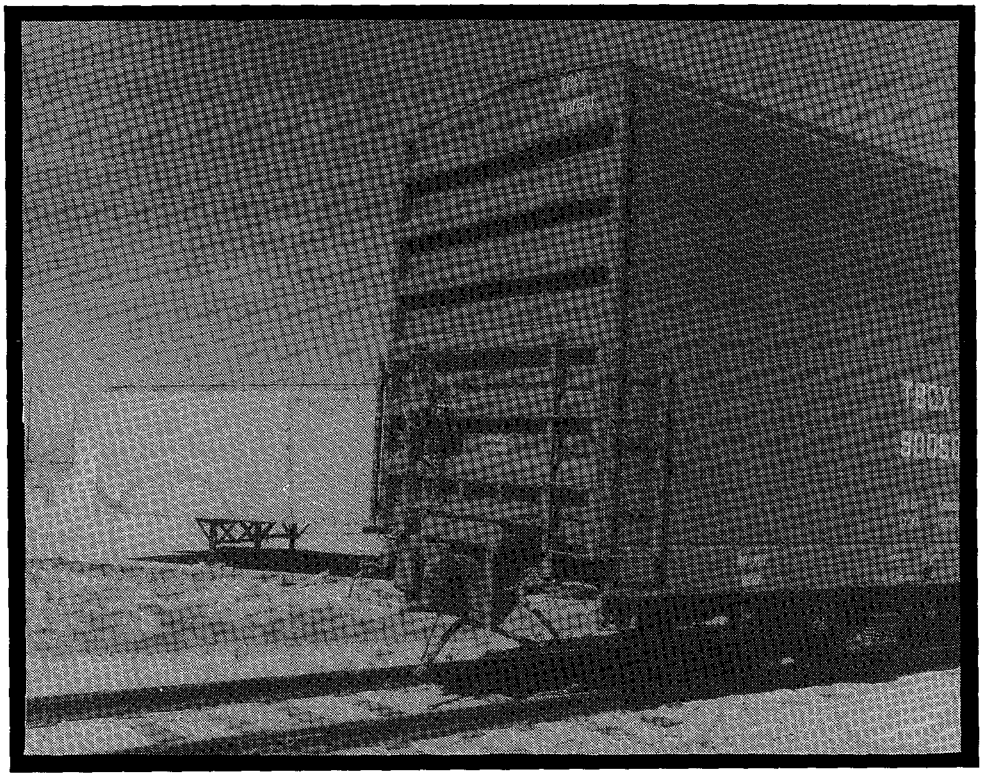


Figure 4.2 MC From B-end

The car was built by ITEL for Boeing, which is under contract to the USAF. The maintenance equipment and spare parts were simulated with concrete blocks bolted in a steel frame. The mass and center of gravity of the EM car were said to be the same as the Operational Model (OM). The loaded weight of the car was 205,300 pounds.

4.1.1 Running Gear

Two American Steel Foundries' (ASF) 100-ton design "Ride Control" trucks were utilized. Constant contact side bearings were used between the truck bolster and the car body bolster. There was no primary suspension. The secondary suspension system consisted of five inner and nine outer D-7 springs. The spring configuration is shown in Figure 4.3.

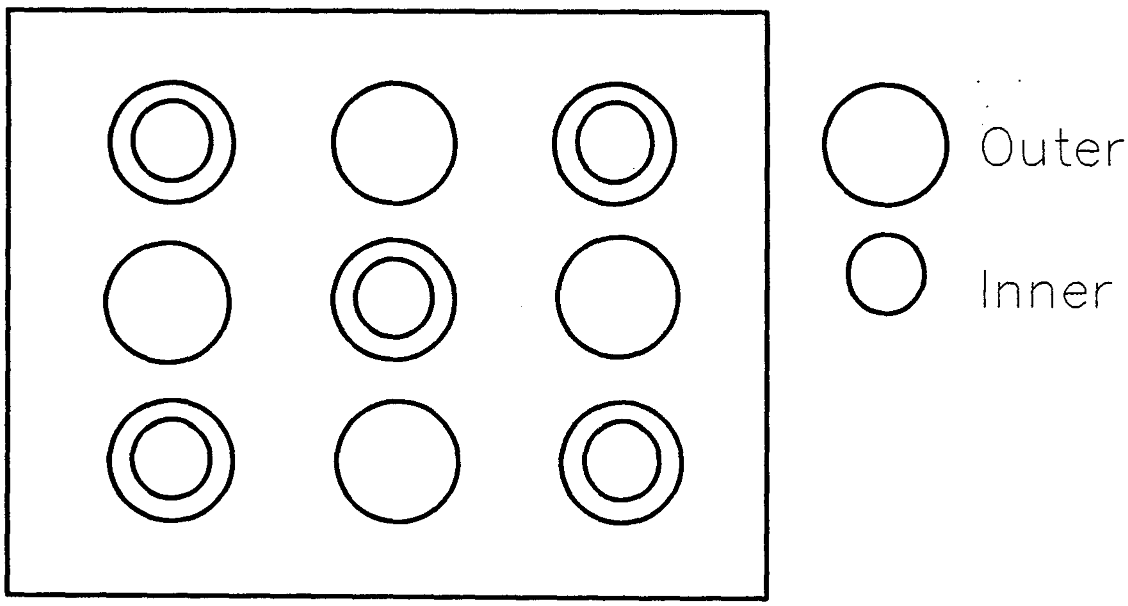


Figure 4.3 MC Spring Configuration

The axle spacing within each 100-ton design truck was 70 inches. The truck center spacing was 64 feet. The car body was 87 feet 1 inch long. The car length was 89 feet over strikers. Type H tight lock couplers and 901-E draft gear were used.

The 36-inch wheels arrived with AAR 1:20 profiles. All wheels were reprofiled to the AAR-1B profile for test. The wheel profile was verified with the profilometer shown in Figure 4.4.

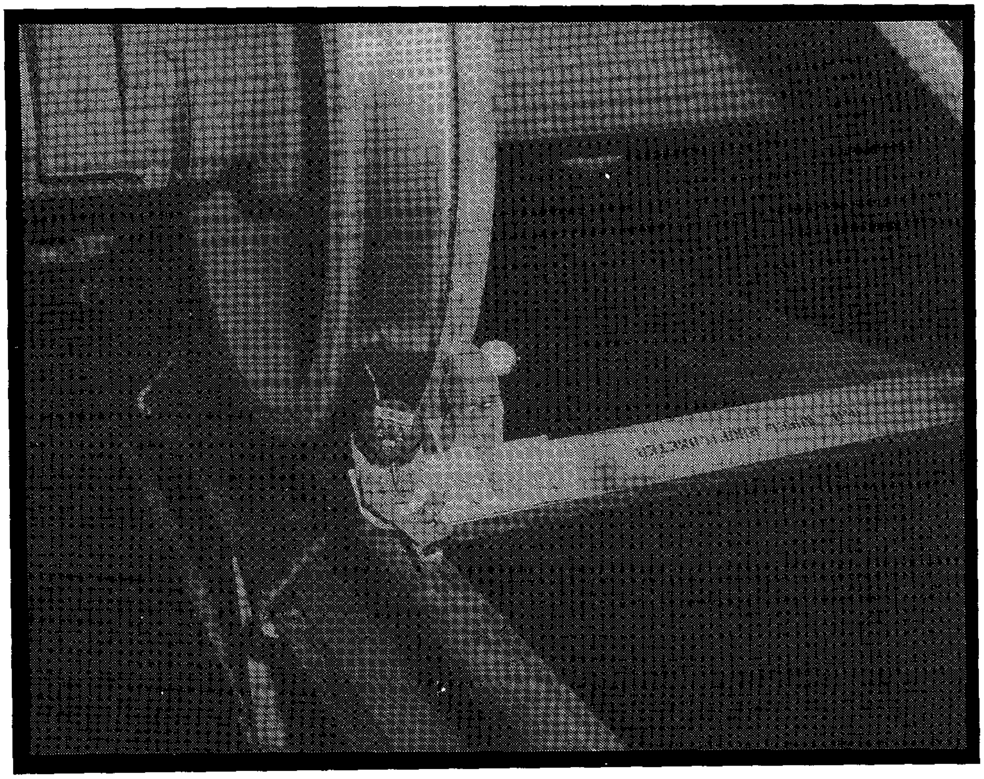


Figure 4.4 Wheel Profilometer

4.2 INSTRUMENTATION CAR DESCRIPTION

The instrumentation car used for the track worthiness testing of the MC was the DOTX207 (T-7) Instrumentation Car. The car was modified to allow installation of the instrumentation and computer equipment required for testing the MC.

4.3 LOCOMOTIVE DESCRIPTION

Dedicated locomotives were used for conducting all of the MC track worthiness testing. The locomotives were GP40-2, four axle models, similar to the locomotives being purchased for the PKRG trains. Other TTC locomotives were used for logistic moves, as required.

4.4 BUFFER CARS

A loaded 100-ton hopper car was used for the curving tests. A lightly loaded empty flatcar was used for pitch and bounce, twist and roll, and turnout and crossover tests. No buffer car was used for hunting or yaw and way testing.

4.5 TEST TRAIN CONFIGURATION

Figure 4.5 shows the standard test train configuration for the MC track worthiness testing. The MC always followed the T-7 Instrumentation Car and ran with the A-end leading.

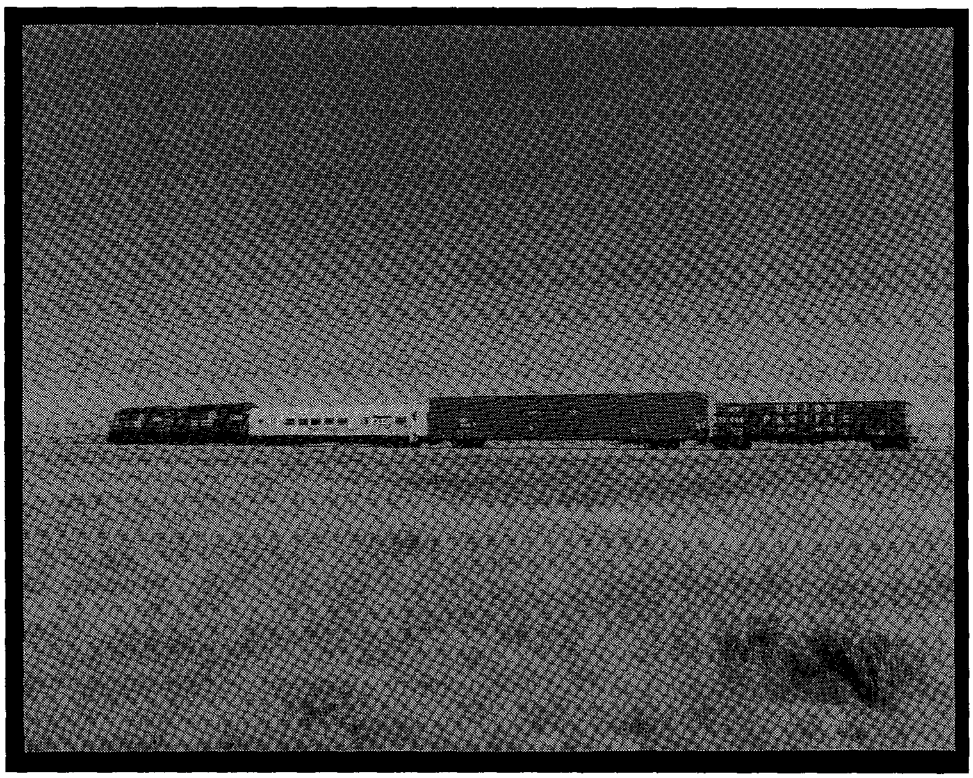


Figure 4.5 Standard Test Train Configuration

5.0 INSTRUMENTATION

5.1 STATIC TRUCK CHARACTERIZATION

Six pieces of instrumentation were used during the four individual air bearing table tests. Two load cells and four string pots were used according to the test requirements. Table 5.1 lists the transducers and where they were used during the test. All of the air bearing test measurements were recorded with a personal computer equipped with a Metrabyte analog to digital converter and Lotus Measure software. The data was stored on floppy disks in Lotus 1-2-3 format.

Table 5.1 Air Table Measurements

NAME	LOCATION & DESCRIPTION	TRANSDUCER TYPE	MEAS. RANGE
LC1	Left Side Actuator Force	Load cell	±10 kip
LC2	Right Side Actuator Force	Load cell	±10 kip
SP1	Left Side Displacement (Yaw Moment)	String pot	±10"
SP2	Right Side Displacement (Yaw Moment)	String pot	±10"
SP1	Left Side Displacement (Long Stiff)	String pot	±1"
SP2	Right Side Displacement (Long Stiff)	String pot	±1"

A load cell was connected to a hydraulic actuator to record the actuator force (Figure 5.1). The same actuator and load cell remained together throughout the air bearing tests.

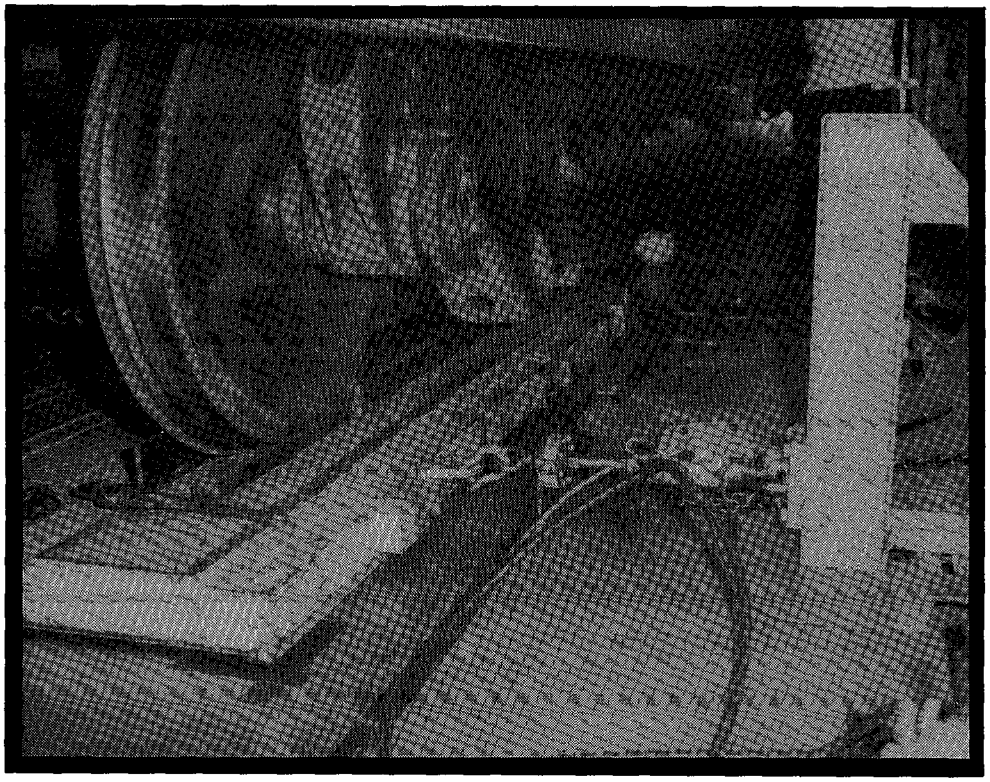


Figure 5.1 Air Table Force Transducer

The other type of transducer used during the air bearing tests was the string pot. These transducers were located appropriately to measure the displacements of the air tables. Figure 5.2 shows a typical setup during a truck rotation test.

Figure 5.2 Air Table Displacement Transducer

5.2 QUASI-STATIC TRUCK CHARACTERIZATION

Quasi-static and modal tests were performed on the MSU. Instrumentation used for quasi-static testing is listed, including transducers to measure actuator forces, rail forces, and suspension component displacements. Table 5.2 summarizes these channels. All of the measurements were collected on an HP 360 desktop computer and recorded onto an optical disk.

Table 5.2 Truck Characterization Measurements

NAME	LOCATION & DESCRIPTION	TRANSDUCER TYPE	MEAS. RANGE
VF01	Left Vertical Actuator Force	Load cell	±100 kip
AF02	Right Vertical Actuator Force	Load cell	±100 kip
VRF3	Lead Left Vertical Rail Force	Instr. rail	0-100 kip
VRF4	Lead Right Vertical Rail Force	Instr. rail	0-100 kip
VRF5	Trail Left Vertical Rail Force	Instr. rail	0-100 kip
VRF6	Trail Right Vertical Rail Force	Instr. rail	0-100 kip
LRF3	Lead Left Lateral Rail Force	Instr. rail	±60 kip
LRF4	Lead Right Lateral Rail Force	Instr. rail	±60 kip
LRF5	Trail Left Lateral Rail Force	Instr. rail	±60 kip
LRF4	Trail Right Lateral Rail Force	Instr. rail	±60 kip
DZ01	Left Vertical or Lateral Actuator Displacement	LVDT	±5"
DZ02	Right Vertical Actuator Displacement	LVDT	±5"
DZ03	Left Vertical Side Bearing Displacement	String pot	±5"
DZ04	Right Vertical Side Bearing Displacement	String pot	±5"
DZ05	Left Vertical Spring Displacement	String pot	±5"
DZ06	Right Vertical Spring Displacement	String pot	±5"
AY02	Right Lateral Spring Displacement	String pot	±1"
DY03	Left Lateral Spring Displacement	String pot	±1"
DY04	Lateral Body to Truck Bolster Displacement	String pot	± 5"

Figure 5.3 shows the transducer location for the right vertical spring displacement when characterizing the MC trucks. This configuration was used for both sides of each truck.

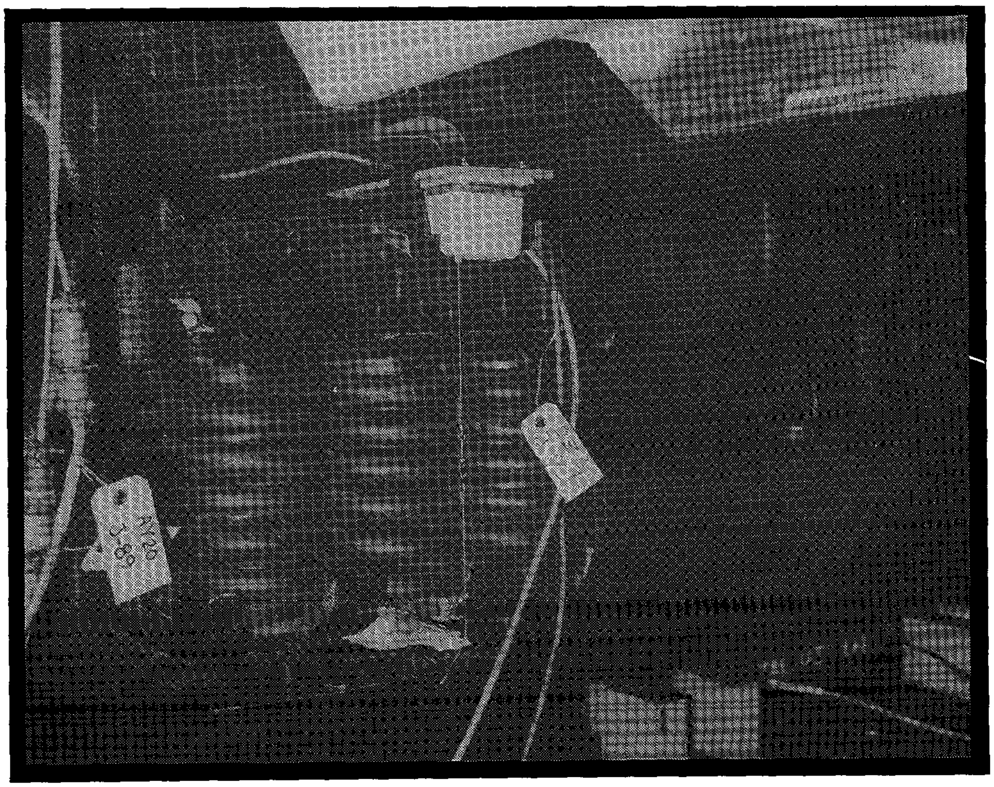


Figure 5.3 Spring Nest Vertical Displacement Transducer

Instrumented rails were used to record vertical and lateral wheel/rail forces. Figure 5.4 shows a wheel positioned on the instrumented rail. The rails were manufactured and calibrated by AAR for the PKRG program.

Figure 5.4 Instrumented Rail

5.3 MODAL RESPONSE

The Modal Response Test was performed on the MSU in the RDL. Figure 5.5 shows an MSU actuator and load cell configured for vertical excitation during quasi-static characterization.

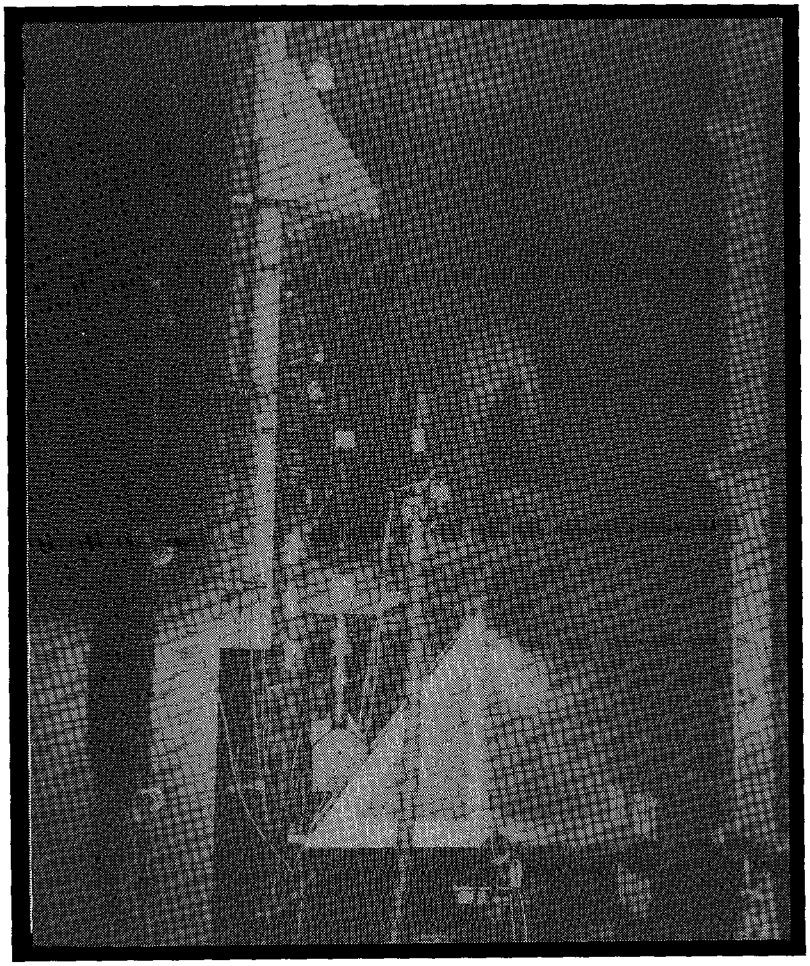


Figure 5.5 MSU Actuator and Load Cell in the Vertical Configuration

Car body-to-ground displacements were used to determine rigid body modes at frequencies less than 5 Hz. String pots were used to obtain these measurements. Figure 5.6 shows the installation of one of the car body-to-ground transducers.

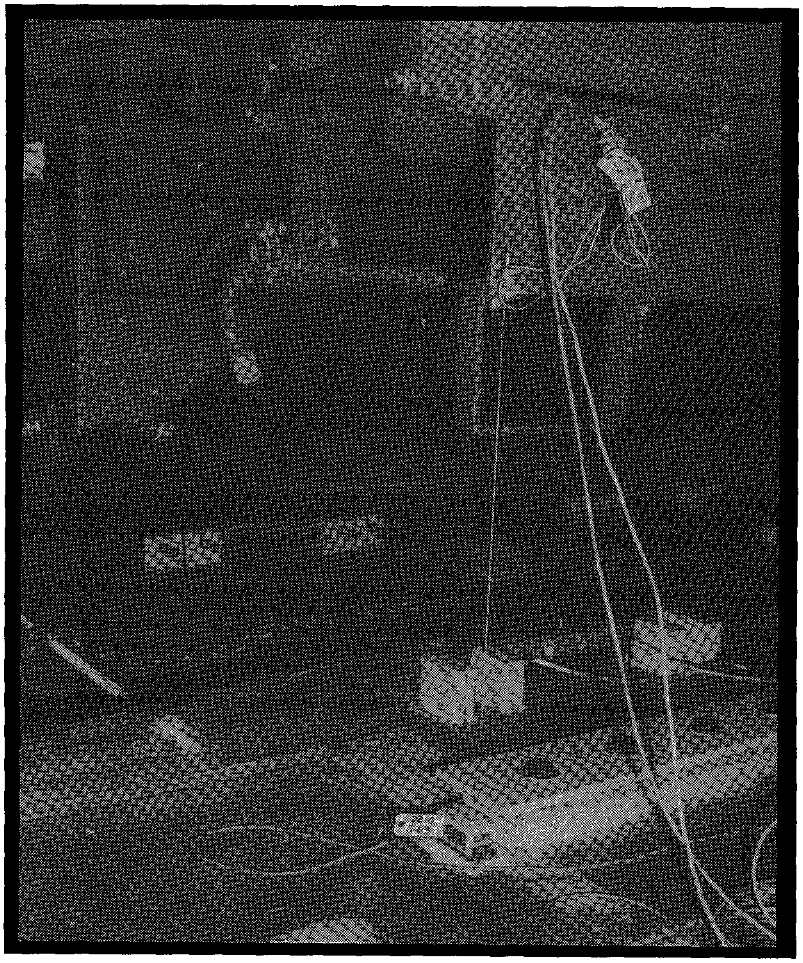


Figure 5.6 Car Body-to-Ground Displacement Transducer

Eight car body-to-ground displacement measurements were recorded. Figure 5.7 shows the location and data channel designation of all the displacement transducers.

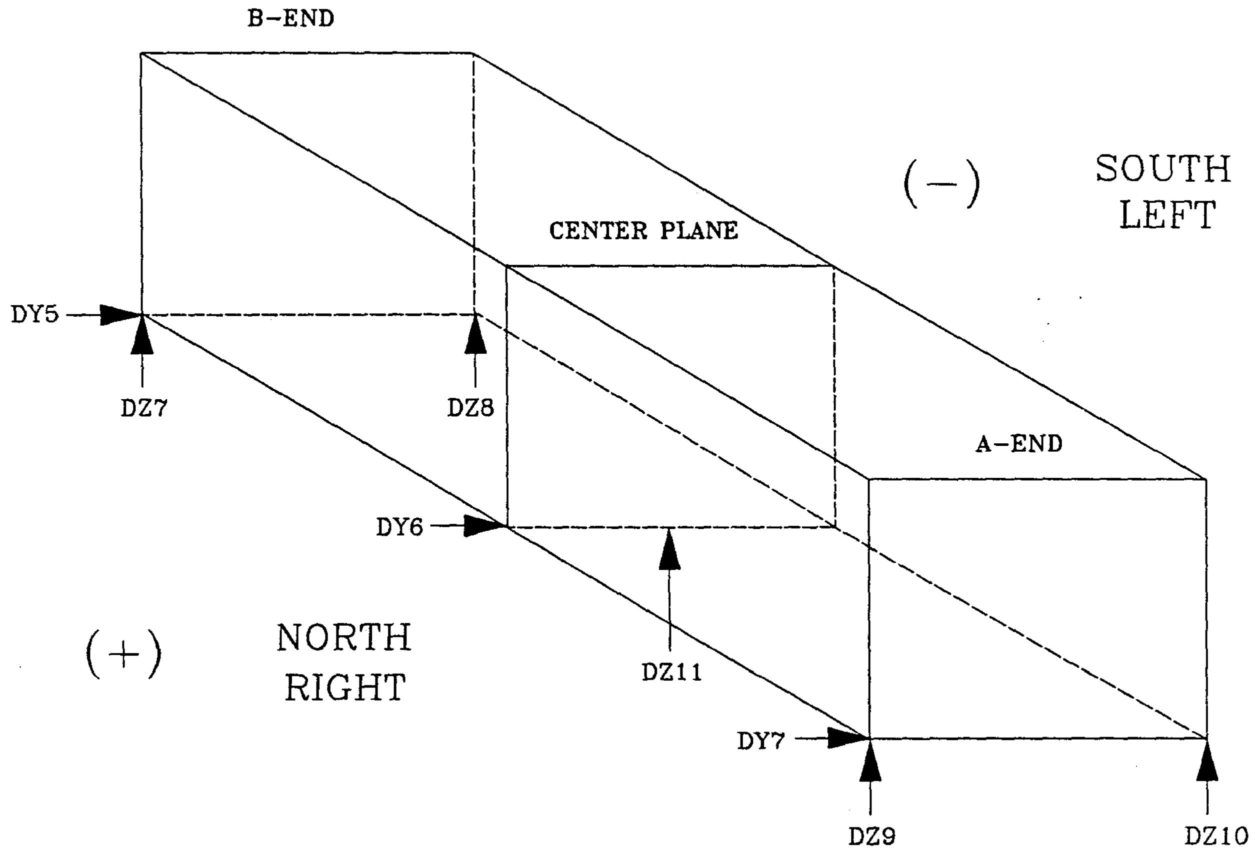


Figure 5.7 Car Body-to-Ground Displacement Locations

The primary source of data for all tests, was an array of car body accelerometers. Thirty-seven accelerometers were mounted on the MC at specific locations with an aluminum block and dental cement. Figure 5.8 shows a pair of accelerometers mounted on the side of the MC. One accelerometer monitored vertical movement and the other accelerometer monitored lateral movement.

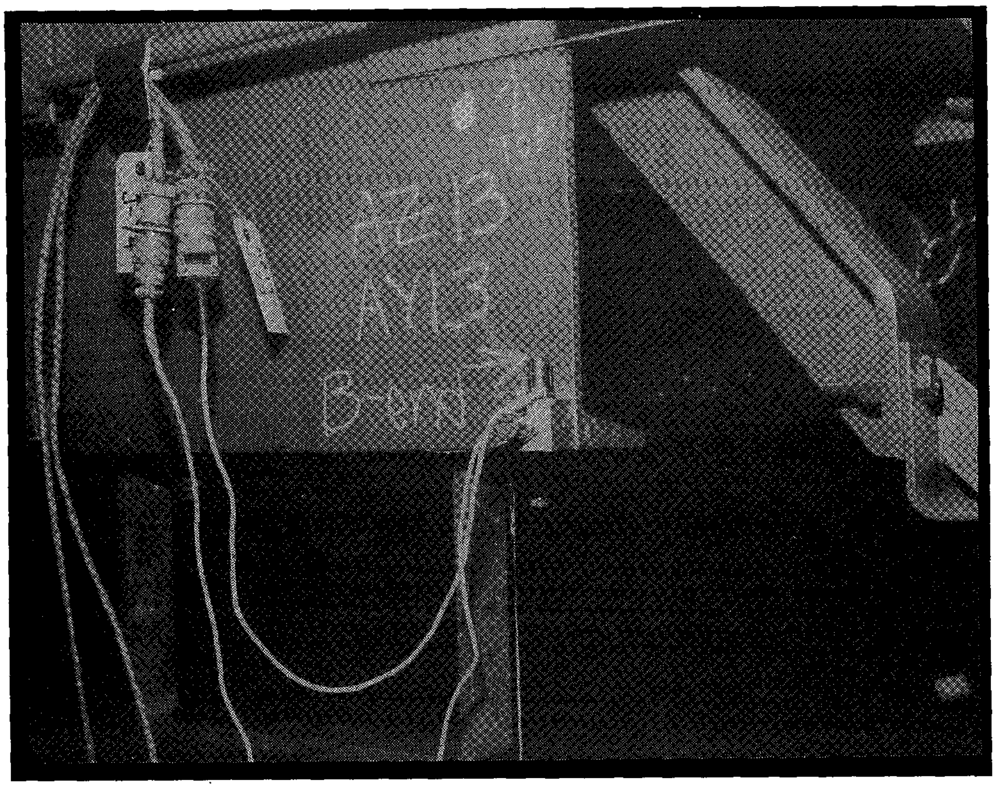


Figure 5.8 Two Car Body Accelerometers

The accelerometers were evenly spaced on the side of the MC and at other critical locations so that primary modes could be determined by transfer function amplitude and transducer phase relationships. Figure 5.9 shows the location and data channel designation of all car body accelerometers.

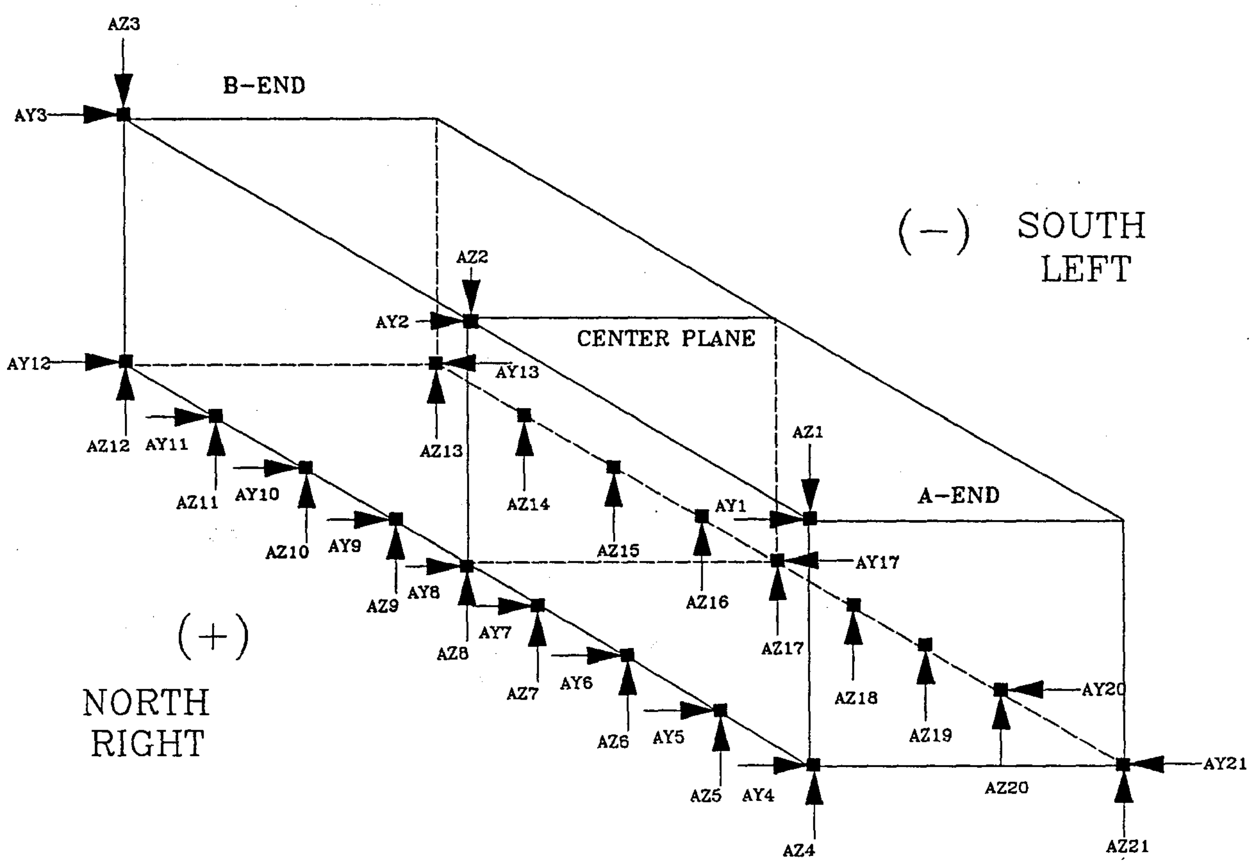


Figure 5.9 Car Body Accelerometer Locations

Vertical and lateral wheel forces were measured at each wheel on the A-end of the MC with the instrumented rails. The setup was very similar to that presented in Section 5.2.1. Figure 5.10 shows the instrumented rails installed on the MSU floor.

Figure 5.10 MSU Instrumented Rails

Table 5.3 is a list of the modal response measurements that were acquired with the quasi-static measurements listed in Table 5.2. Instrumented rail and actuator force were also used in modal analysis.

Table 5.3 Additional Measurements for Modal Response

NAME	LOCATION & DESCRIPTION	TRANSDUCER TYPE	MEAS. RANGE
DZ07	B-End Right Vertical Car Body Displacement	String Pot	±5"
DZ08	B-End Left Vertical Car Body Displacement	String Pot	±5"
DZ09	A-End Right Vertical Car Body Displacement	String Pot	±5"
DZ10	A-End Left Vertical Car Body Displacement	String Pot	±5"
DZ11	Center Vertical Car Body Displacement	String Pot	±5"
DY05	B-End Right Lateral Car Body Displacement	String Pot	±5"
DY06	Center Right Lateral Car Body Displacement	String Pot	±5"
DY07	A-End Right Lateral Car Body Displacement	String Pot	±5"
AZ1	Upper A-End Right Vertical Car Body Accelerometer	Accelerometer	±5 g
AY1	Upper A-End Right Lateral Car Body Accelerometer	Accelerometer	±5 g
AZ2	Upper Center Right Vertical Car Body Accelerometer	Accelerometer	±5 g
AY2	Upper Center Right Lateral Car Body Accelerometer	Accelerometer	±5 g
AZ3	Upper B-End Right Vertical Car Body Accelerometer	Accelerometer	±5 g
AY3	Upper B-End Right Lateral Car Body Accelerometer	Accelerometer	±5 g
AZ4	A-End Right Vertical Car Body Accelerometer	Accelerometer	±5 g
AY4	A-End Right Lateral Car Body Accelerometer	Accelerometer	±5 g
AZ5	Right Vertical Car Body Accelerometer - 1 Back	Accelerometer	±5 g
AY5	Right Lateral Car Body Accelerometer - 1 Back	Accelerometer	±5 g
AZ6	Right Vertical Car Body Accelerometer - 2 Back	Accelerometer	±5 g
AY6	Right Lateral Car Body Accelerometer - 2 Back	Accelerometer	±5 g
AZ7	Right Vertical Car Body Accelerometer - 3 Back	Accelerometer	±5 g
AY7	Right Lateral Car Body Accelerometer - 3 Back	Accelerometer	±5 g
AZ8	Right Vertical Car Body Accelerometer - Mid Car	Accelerometer	±5 g
AY8	Right Lateral Car Body Accelerometer - Mid Car	Accelerometer	±5 g
AZ9	Right Vertical Car Body Accelerometer - 5 Back	Accelerometer	±5 g
AY9	Right Lateral Car Body Accelerometer - 5 Back	Accelerometer	±5 g
AZ10	Right Vertical Car Body Accelerometer - 6 Back	Accelerometer	±5 g
AY10	Right Lateral Car Body Accelerometer - 6 Back	Accelerometer	±5 g
AZ11	Right Vertical Car Body Accelerometer - 7 Back	Accelerometer	±5 g
AY11	Right Lateral Car Body Accelerometer - 7 Back	Accelerometer	±5 g
AZ12	B-End Right Vertical Car Body Accelerometer	Accelerometer	±5 g
AY12	B-End Right Lateral Car Body Accelerometer	Accelerometer	±5 g
AZ13	B-End Left Vertical Car Body Accelerometer	Accelerometer	±5 g
AY13	B-End Left Lateral Car Body Accelerometer	Accelerometer	±5 g
AZ14	Left Vertical Car Body Accelerometer - 7 Back	Accelerometer	±5 g
AZ15	Left Vertical Car Body Accelerometer - 6 Back	Accelerometer	±5 g
AZ16	Left Vertical Car Body Accelerometer - 5 Back	Accelerometer	±5 g
AZ17	Left Vertical Car Body Accelerometer - Mid Car	Accelerometer	±5 g
AZ18	Left Vertical Car Body Accelerometer - 3 Back	Accelerometer	±5 g
AZ19	Left Vertical Car Body Accelerometer - 2 Back	Accelerometer	±5 g
AZ20	Left Vertical Car Body Accelerometer - 1 Back	Accelerometer	±5 g
AY20	Left Lateral Car Body Accelerometer - 1 Back	Accelerometer	±5 g
AZ21	A-end Left vertical Car body Accelerometer	Accelerometer	±5 g
AY21	A-end Left vertical Car body Accelerometer	Accelerometer	±5 g

5.4 CURVE STABILITY

The only instrumentation required in the Curve Stability Test was a load cell and a feeler gage. The load cell was assembled on a coupler which was installed in a locomotive (Figure 5.11). This coupler measured the compressive or tensile force in the consist.

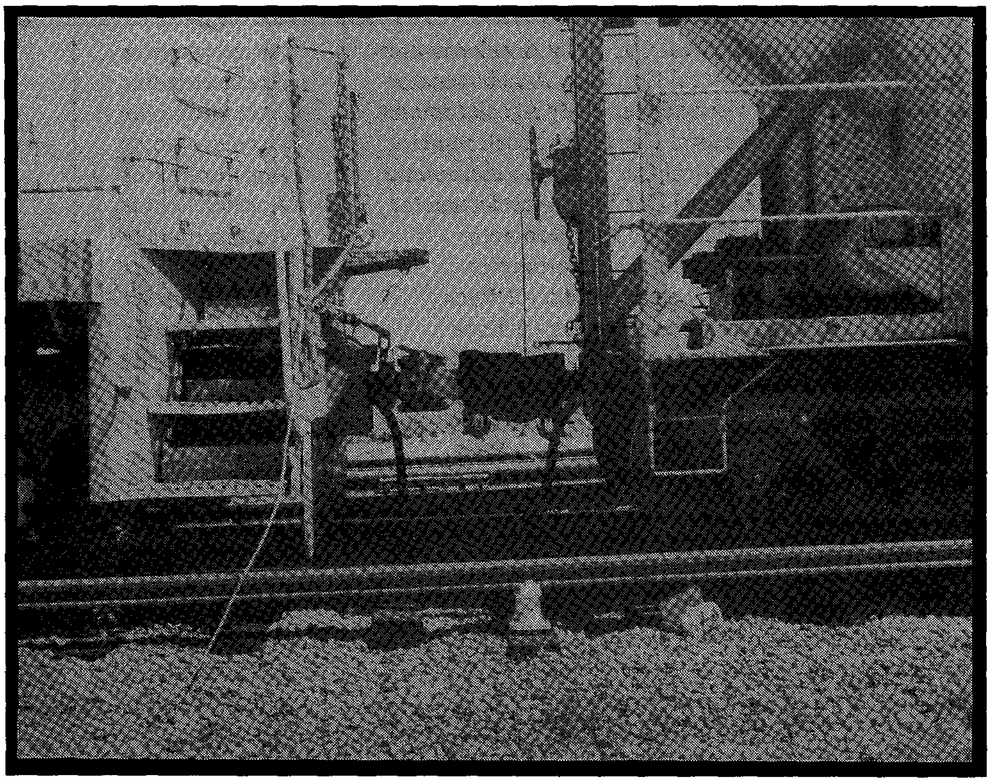


Figure 5.11 Curve Stability Instrumented Coupler

The feeler gage used, was simply a 1/8-inch steel bar that was placed under a wheel to measure if there was any wheel lift. Figure 5.12 shows an MC wheel being checked during a test.

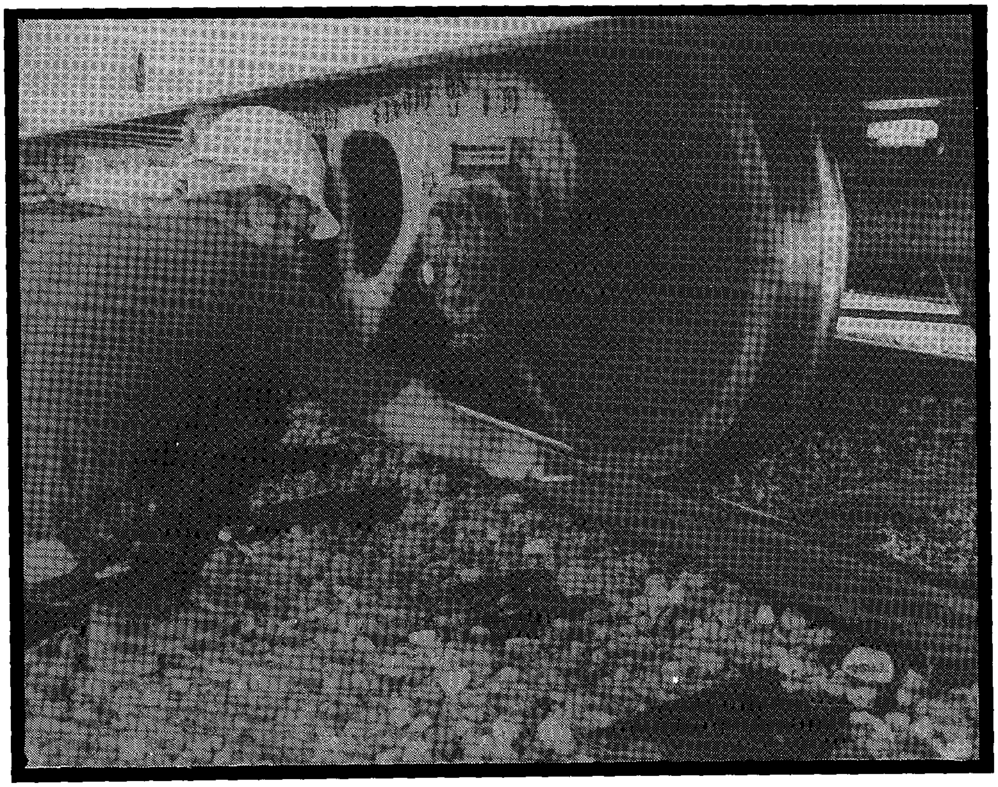


Figure 5.12 Curve Stability Wheel Lift Gage

5.5 TRACK WORTHINESS

The MC was fitted with instrumented wheel sets, accelerometers, roll gyros, and string pots. The following sections describe each part of the instrumentation package.

5.5.1 Instrumented Wheel Sets

Four 36-inch instrumented wheel sets were procured by TTC for the PKRG Program. They were manufactured by the Illinois Institute of Technology Research Institute (IITRI). The instrumented wheel sets used standard wheels and axles machined smooth and strain gaged. Vertical and lateral wheel force, and axle torque were calculated from the strain gage output.

Each wheel used six strain gage bridges. Three strain gage bridges were used to measure vertical force; two were used to measure lateral force, and one was used to indicate lateral wheel tread position on the rail. Axle torque was measured with a strain gage bridge on the axle. The raw analog strain gage signals were acquired with a 386 based computer system and an analog to digital (AD) converter. The signals were processed to produce digital output in the form of left and right side vertical wheel force, lateral wheel force, and axle torque. The digital signals were then converted to analog. Those analog signals were displayed on strip charts and acquired on the Hewlett-Packard (HP) Data Acquisition System (DAS) with the output from other transducers. Figure 5.13 shows an IITRI wheel set installed under the MC.

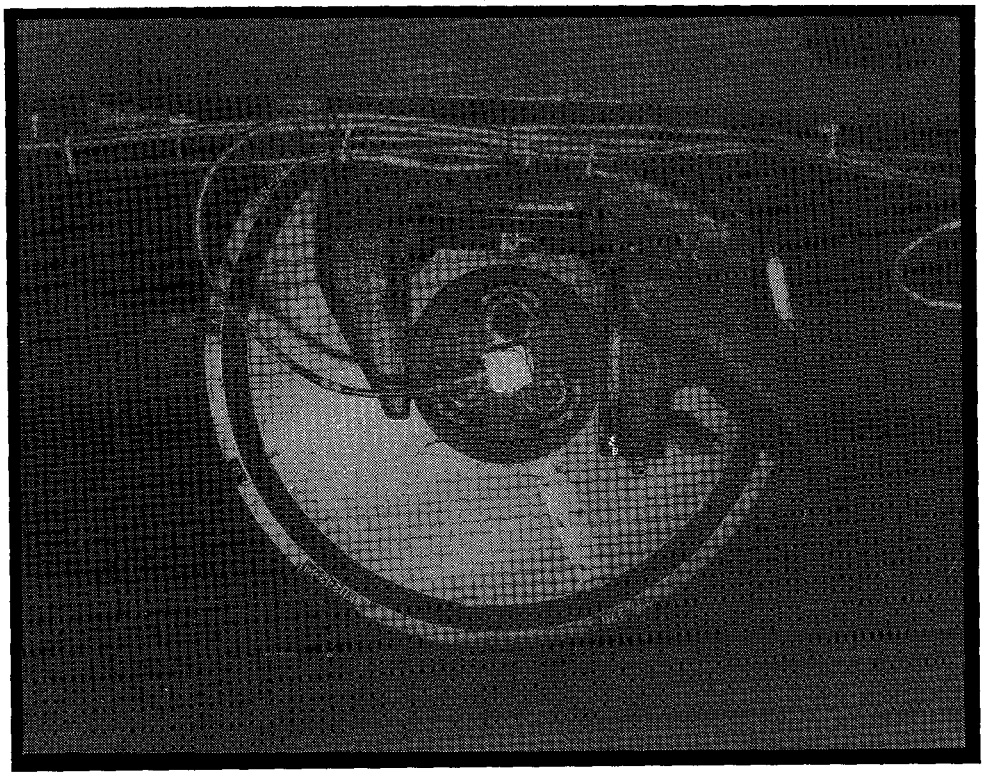


Figure 5.13 IITRI Instrumented Wheel Set

Real time indicators of potential derailment in the form of wheel force and lateral to vertical force ratios were displayed for test safety. Car body accelerations and roll angles were also displayed real time.

5.5.2 Roll Gyros

Chapter XI requires the measurement of car body roll angle. This was accomplished with two roll rate gyros. The gyros were installed on each end of the car at floor level, as shown in Figure 5.14.

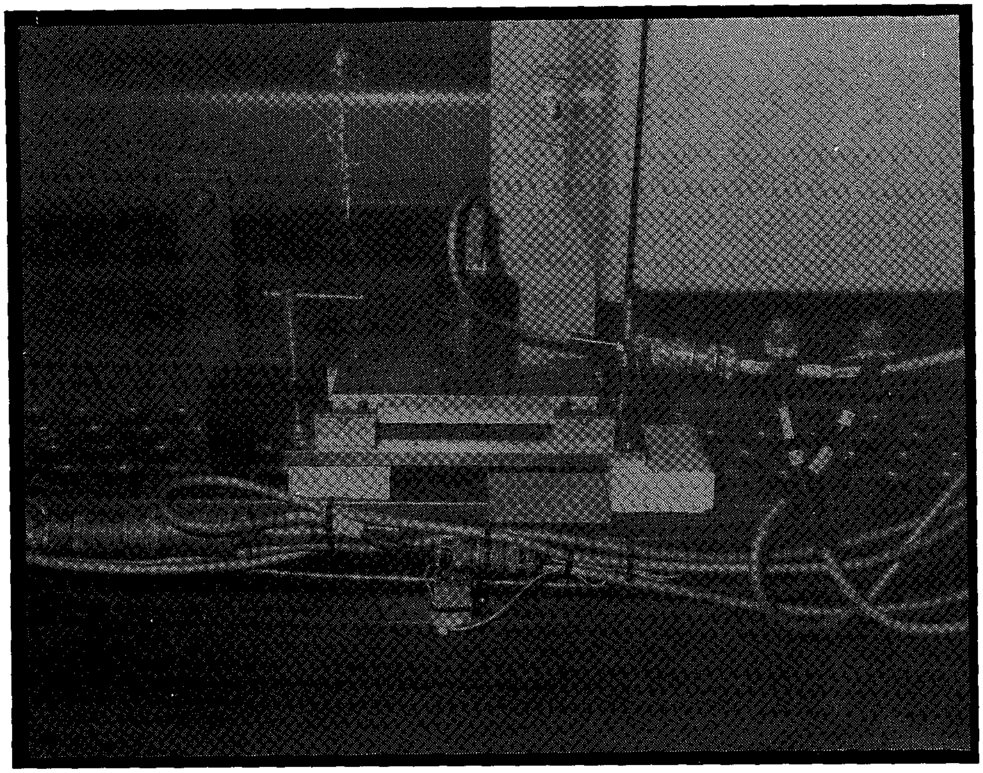


Figure 5.14 Roll Gyro at B-end of MC

The output signal was a roll rate. This was electronically integrated and output to the DAS as an analog roll angle.

5.5.3 Lateral Accelerometers

Columbia 5 g accelerometers were installed in a lateral orientation at the A- and B-ends on the roll gyro base plates. They were utilized for the Hunting Test criteria, 1.0 g peak-to-peak sustained for 20 seconds or a single occurrence of 1.5 g peak-to-peak.

5.5.4 Additional Measurements

A large number of accelerometers were installed in vertical, lateral, and longitudinal orientations on the car and running gear at the request of Boeing. Truck spring nest and side bearing displacements were also measured. All transducers are listed by channel name in Appendix A.

5.5.5 Data Acquisition System

Analog signals from 131 signal conditioners were multiplexed and digitized with a HP-6944 multiprogrammer. Digital signals were acquired with a HP-360 computer. Analog to digital counts were stored with their proper engineering unit conversions on one file. Data files were stored on a 650 megabyte optical disk.

5.5.6 Chart Recorders

Processed instrumented wheel set information was displayed real time on two chart recorders. Roll angle, lateral acceleration, and other pertinent measurements were displayed real time on two other recorders.

5.5.7 Video System

Four video cameras were mounted under the MC to record the leading wheel of each truck. The video signals were split to two monitors and then to VHS video recorders. On-screen annotation and audio were recorded for each test run. The video signals were stored on a VHS format video tape.

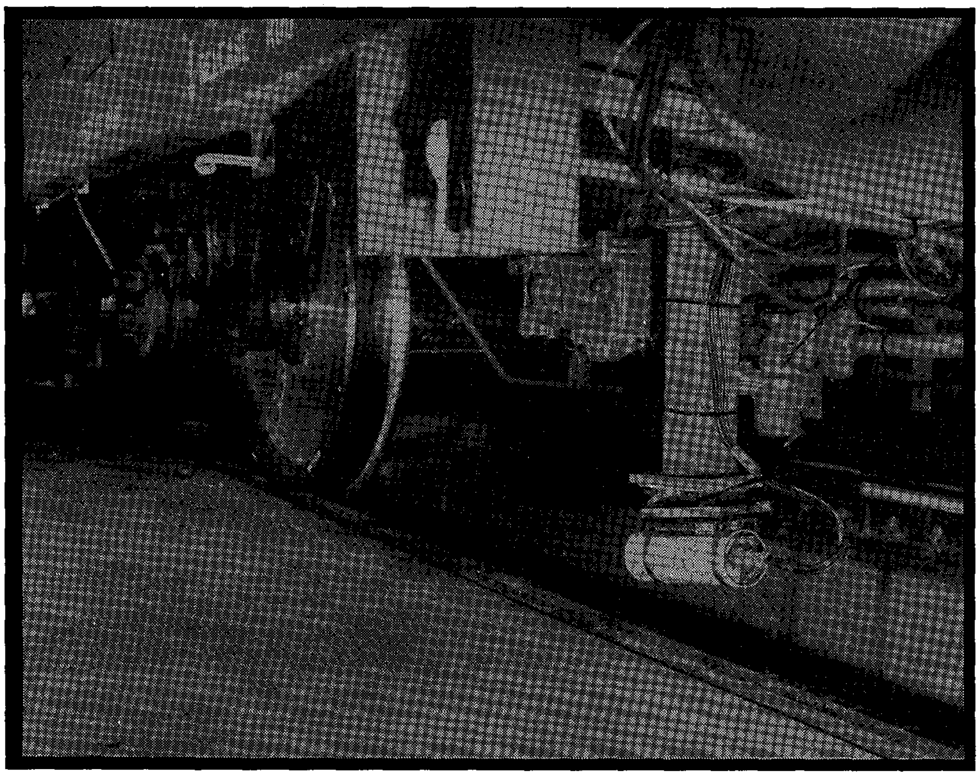


Figure 5.15 MC Track Worthiness Video

5.6 STATIC BRAKE TEST

The Static Brake Test was performed at TTC. A compressor was used to supply air to the MC brake system. A single car test device was connected between the compressor and the MC. The single car test device was used to control the brakes on the MC. An air gage was installed in the brake line of the MC to measure brake cylinder pressure. The brake shoes were replaced with four instrumented shoes to measure the brake shoe force. An instrumented shear pin was installed in the hand brake chain to measure the hand brake force that was applied during the test. All measurements were displayed with a digital readout (Figure 5.16).

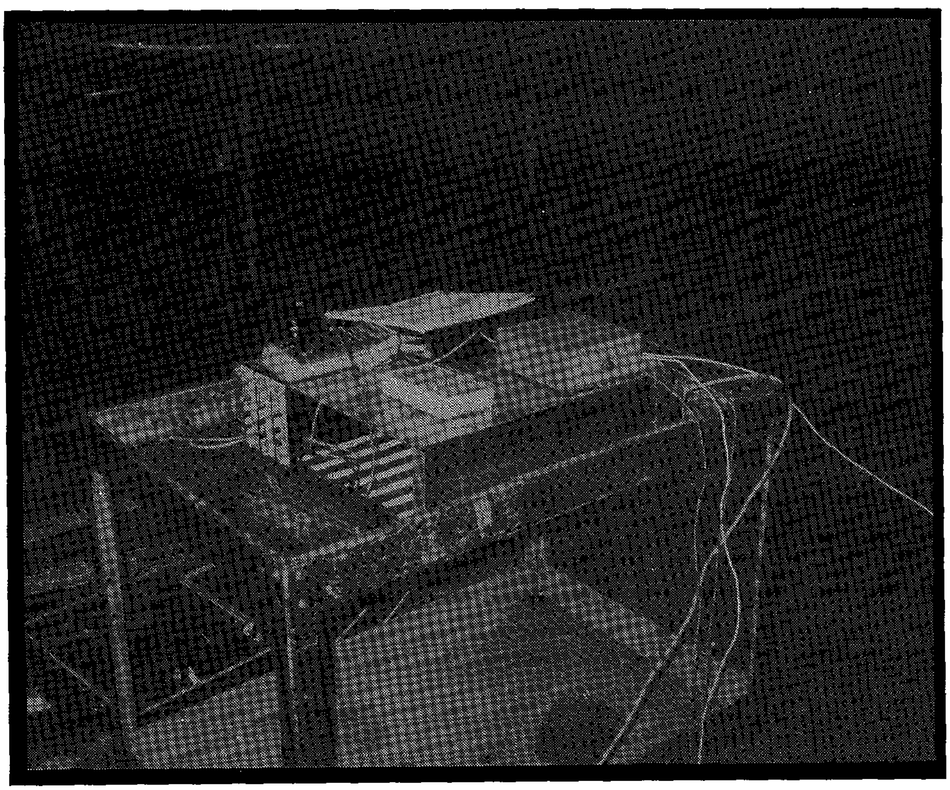


Figure 5.16 Brake Test Readout Device

6.0 RESULTS

The results of MC testing are presented in four sections:

- Vehicle Characterization
- Service Worthiness
- Track Worthiness
- Static Brake Test

6.1 VEHICLE CHARACTERIZATION

Vehicle characterization consisted of the following three tests and results are presented by those headings.

- Static Truck Characterization
- Quasi-Static Truck Characterization
- Modal Characterization

6.1.1 Static Characterization Test Results

Static characterization results include truck yaw moment, axle box longitudinal stiffness, inter-axle yaw and bending, and axle alignment.

6.1.1.1 100-Ton Truck Yaw Moment

Actuator force versus truck displacement was plotted for each actuator after each test. Figure 6.1 is a typical force versus displacement plot.

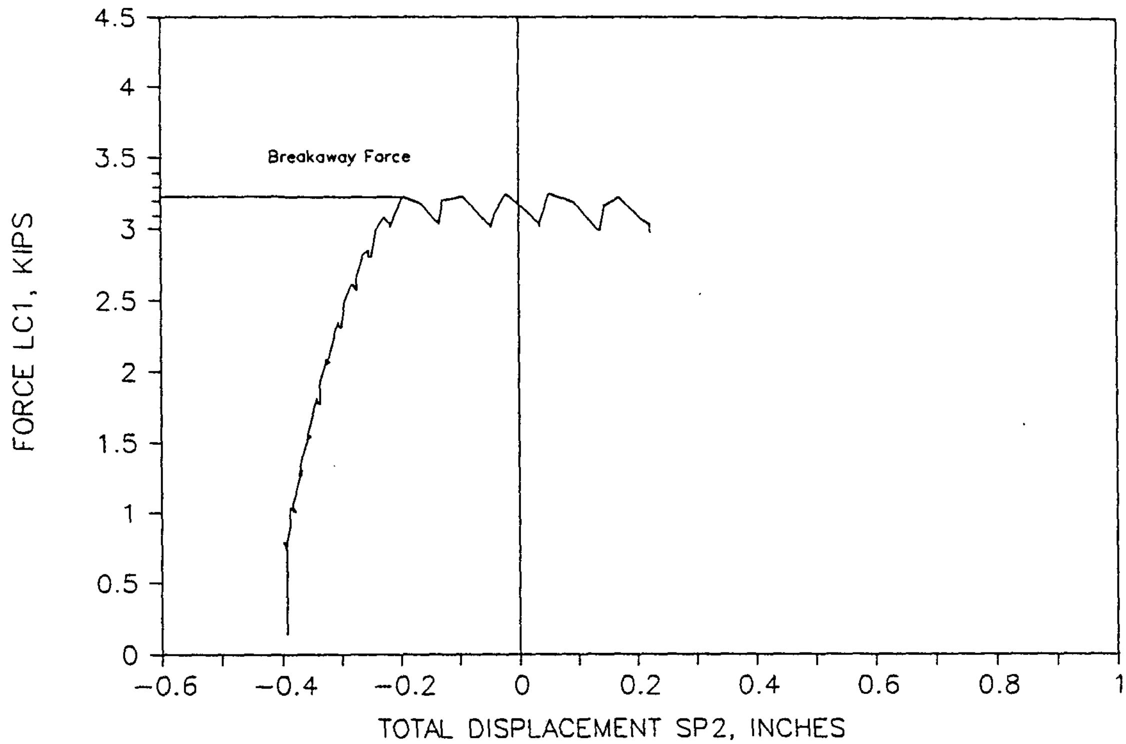


Figure 6.1 Force Versus Displacement, First Actuator from Yaw Moment Testing

The force increased with relatively small displacement until the static friction was overcome. At that point, the truck rotated with virtually no increase in force. This was called the breakaway point. Since two actuators were used, the actual breakaway torque or yaw moment was calculated by summing the two breakaway torques. A plot very similar to the one in Figure 6.1 is shown in Figure 6.2. It is force versus displacement for the second actuator.

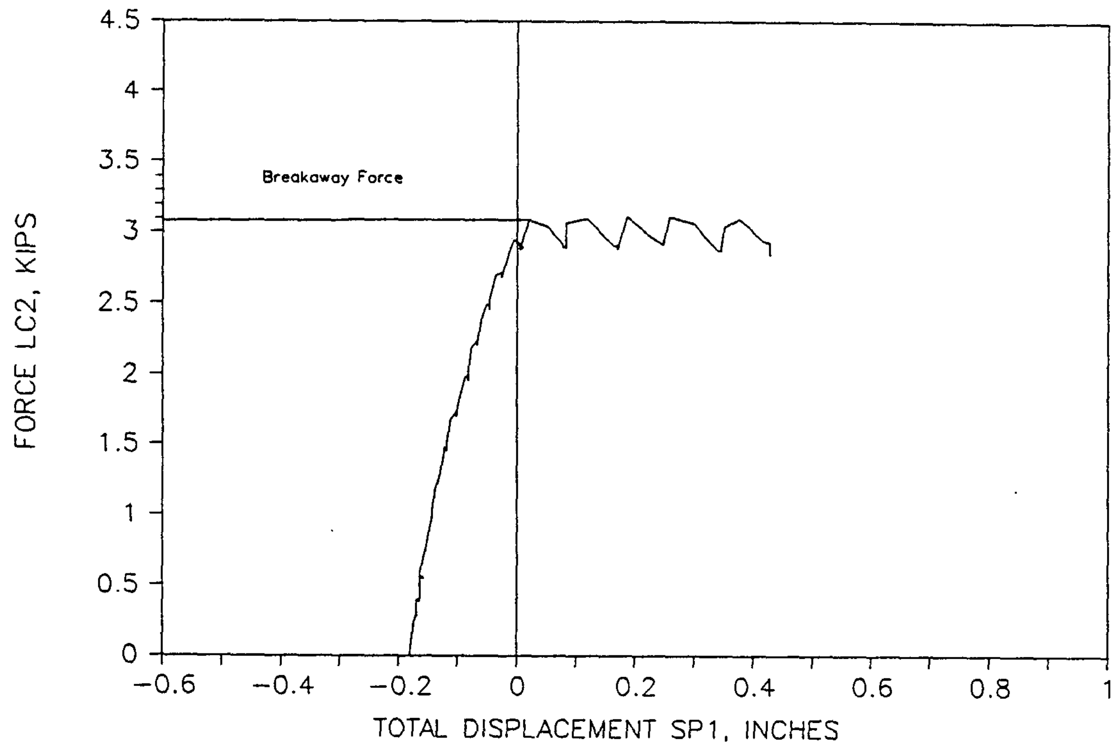


Figure 6.2 Force Versus Displacement, Second Actuator from Yaw Moment Testing

In this case, the breakaway force on each actuator was slightly more than 3,000 pounds. The plots are not smooth because a hand pump was utilized to drive the actuators. The perpendicular distance from each actuator to the truck center pin was approximately 36 inches. The yaw moment or breakaway torque was then calculated by multiplying the sum of the two forces by the distance of 36 inches. The yaw moment for run BMCA002, shown in Figures 6.1 and 6.2, was ≅ 227,500 in-lbs. The average yaw moment from all runs for both trucks was approximately 222,000 in-lbs. Truck yaw moment will depend on the friction level at the center plate and side bearings and on the side bearing preload. Tables 6.1 and 6.2 show a summary of the yaw moment test results.

Table 6.1 Yaw Moment Test Results For Truck 1 (B-end)

DIRECTION	RUN NO.	FORCE 1 (kips)	FORCE 2 (kips)	TOTAL (kips)	MOMENT (in-lbs)
CW	053	2.69	2.75	5.44	195,840
CW	054	2.85	2.82	5.67	204,120
CW	055	2.849	2.95	5.779	208,764
CW AVERAGE : ≅ 202,900 in-lbs					
STANDARD DEVIATION : ≅ 6,500					
CCW	049	2.47	2.56	5.03	181,080
CCW	050	2.68	2.77	5.45	196,200
CCW	051	2.8	2.9	5.7	205,200
CCW	052	2.8	2.91	5.71	205,560
CCW AVERAGE : ≅ 197,000 in-lbs					
STANDARD DEVIATION : ≅ 11,500					
TRUCK 1 AVERAGE YAW MOMENT : ≅ 200,000 in-lbs					

Table 6.2 Yaw Moment Test Results For Truck 2 (A-end)

DIRECTION	RUN NO.	FORCE 1 (kips)	FORCE 2 (kips)	TOTAL (kips)	MOMENT (in-lbs)
CW	001	2.670	2.570	5.24	188,640
CW	002	3.24	3.08	6.32	227,520
CW	003	3.29	3.11	6.4	230,400
CW	004	3.39	3.2	6.59	237,240
CW AVERAGE : ≅ 221,000 in-lbs					
STANDARD DEVIATION : ≅ 21,900					
CCW	005	4.0	3.91	7.91	284,760
CCW	006	3.45	3.41	6.86	246,960
CCW	007	4.09	4.04	8.13	292,680
CCW	008	3.49	3.45	6.94	249,840
CCW AVERAGE : ≅ 268,600 in-lbs					
STANDARD DEVIATION : ≅ 23,500					
TRUCK 2 AVERAGE YAW MOMENT : ≅ 244,800 in-lbs					

6.1.1.2 Axle Alignment Test

The radial misalignment, as well as the lateral misalignment, between the two axles in each truck were the subject of this investigation. Six measurements, described in Table 6.3, were made during each test (Figure 6.3).

Table 6.3 Axle Alignment Measurements

MEASUREMENT NAME	DESCRIPTION
AS1	Right Side Axle Spacing Measured Directly From Caliper
AS2	Left Side Axle Spacing Measured Directly From Caliper
LA1	Leading Axle Lateral Alignment -- Leading Half of Wheel
LA2	Leading Axle Lateral Alignment -- Trailing Half of Wheel
TA1	Trailing Axle Lateral Alignment -- Leading Half of Wheel
TA2	Trailing Axle Lateral Alignment -- Trailing Half of Wheel

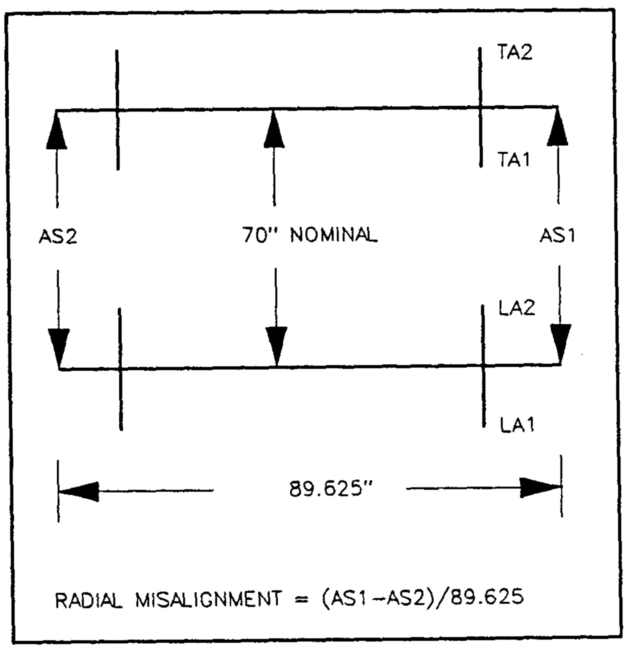


Figure 6.3 Axle Alignment Measurements

Axle radial misalignment was calculated with the axle spacing values AS1 and AS2, as shown in the equation in Figure 6.3. Axle lateral misalignment was calculated with the leading axle and trailing axle measurements LA1, LA2, TA1, and TA2, respectively. Those numbers were measured from rulers with a Brunson Optical Transit. The transit was first rotated until LA1 was equal to TA2. It was then translated so that LA1 and TA2 were on a round number. LA2 and TA1 were then measured. Delta LA and TA were then calculated. It was assumed that the transit was parallel to the side frame when LA1 and TA2 were equal. The lateral misalignment could be implied from the deltas. A tabulation of alignment measurements from both trucks is presented in Table 6.4. The variations were consistent and typical for conventional three-piece truck design.

Table 6.4 Axle Alignment Data

TRUCK NO.	RUN NO.	AS1 L1-L2 (in.)	AS2 R1-R2 (in.)	RADIAL MIS. (Mrad)	DELTA LA1-LA2 (in.)	DELTA TA1-TA2 (in.)
1 (B-end)	045	69.929	70.03	-1.13	0.0	-0.041
1	046	69.870	70.010	-1.56	-0.09	-0.037
1	047	69.857	70.028	-1.91	-0.011	-0.045
1	048	69.885	70.02	-1.51	-0.019	-0.037
2 (A-end)	009	69.77	69.815	-.502	0.0	+0.02
*2	010	69.13	69.832	-7.83	0.0	+0.006
2	011	69.844	69.8	+.491	-0.033	0.0
2	012	69.814	69.820	-0.067	-0.004	+0.012

* Value is inconsistent with the other three runs and will not be used.

6.1.1.3 Longitudinal Stiffness

Axle box longitudinal stiffness is related to the ability of the axles to move longitudinally independently of each other. In most standard three-piece trucks, the longitudinal stiffness is very high once the bearing adapters run up against the side frame stops. In trucks with primary suspension components, there is some stiffness associated with the shearing of the suspension before the bearing adapters run up against the stops. The MC does not have primary suspension. Figure 6.4 shows the longitudinal movement and restraint of the axles in a truck.

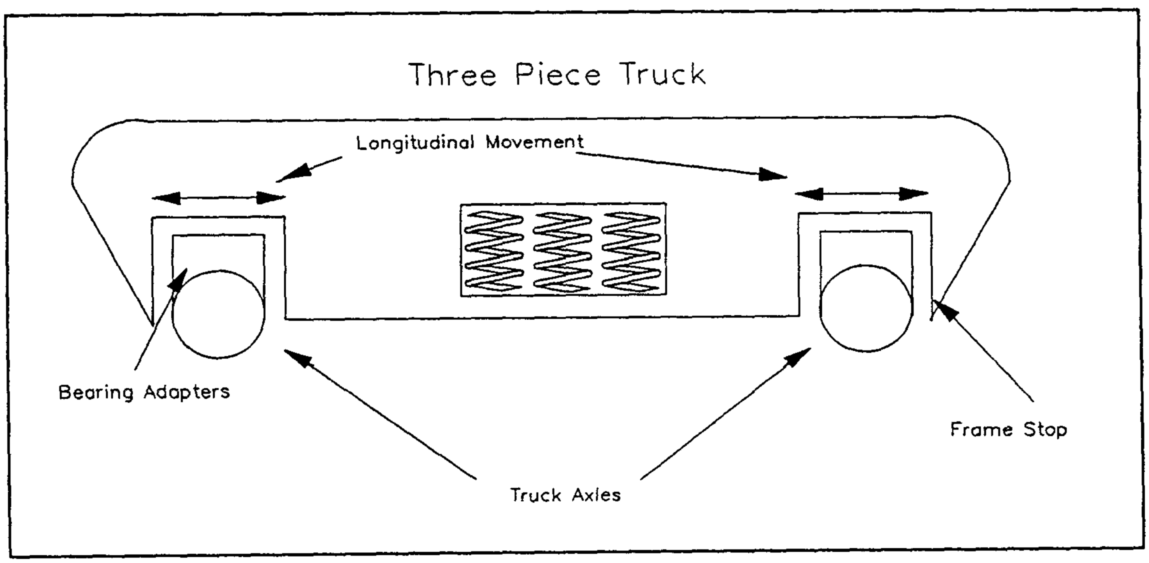


Figure 6.4 Longitudinal Movement and Restraint of Axles

NUCARS requires axle box stiffness rather than truck side stiffness. It was assumed that the truck side was symmetric. Force versus displacement plots were produced for each truck side on all test runs. Typical plots, for run BMCA031 in this case, are shown in Figures 6.5 and 6.6.

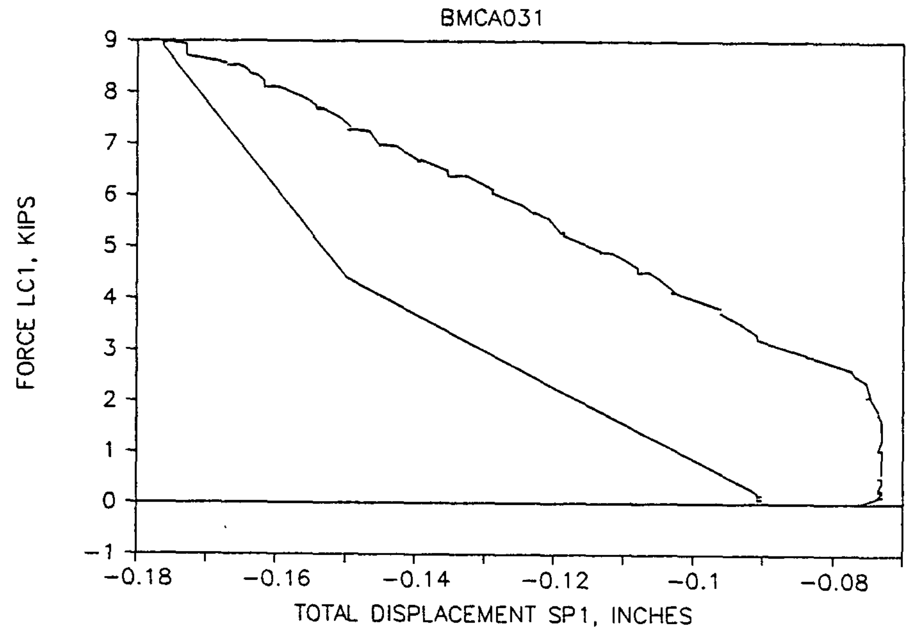


Figure 6.5 Right Truck Side Longitudinal Stiffness Plot

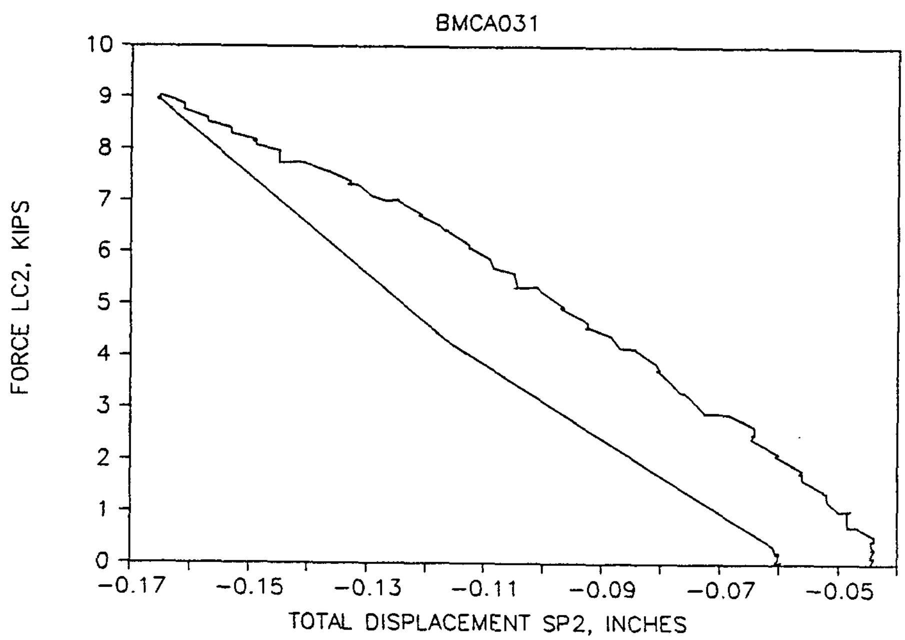


Figure 6.6 Left Truck Side Longitudinal Stiffness Plot

Table 6.5 is a tabulation of the truck side longitudinal stiffness measurements.

Table 6.5 Truck Side Longitudinal Stiffness Measurements

RUN NO.	TRUCK NO.	DIRECTION	RIGHT SIDE SLOPE (kips/inch)	LEFT SIDE SLOPE (kips/inch)
030	1	Pulling	*50.42	*50.93
031	1	Pulling	67.39	65.64
032	1	Pulling	69.31	70.61
033	1	Pulling	70.64	70.29
034	1	Pushing	*41.28	*30.00
035	1	Pushing	70.40	58.58
036	1	Pushing	76.65	59.03
037	1	Pushing	77.76	64.06
		AVERAGE :	72.03	64.70
		STANDARD DEVIATION :	**3.82**	**4.78**
021	2	Pulling	*30.94	*33.26
022	2	Pulling	*35.96	63.85
023	2	Pulling	*37.25	61.05
018	2	Pushing	63.89	69.41
019	2	Pushing	66.67	67.69
020	2	Pushing	63.33	64.29
		AVERAGE :	**64.63**	**65.26**
		STANDARD DEVIATION :	**1.46**	**2.96**
OVERALL AVERAGE : ≅ 66.7 kips/inch				
STANDARD DEVIATION BETWEEN FOUR AVERAGE : ≅ 3.1				

* Not used in calculations

The overall average truck side longitudinal stiffness was 66.7 kips/inch. The first test in each direction yielded a much lower stiffness. Runs 018, 019, and 020 were curve fit by hand due to the same sliding motion affecting the plot at approximately 5,000 pounds force. The right side of the truck yielded a lower stiffness than normal in runs 022 and 023. That data was also rejected, though no cause was found for the discrepancy.

The truck side average was then doubled to give an axle box stiffness of 133 kip/inch. A final stiffness of 1000 kips/inch was expected as the bearing adapter ran up against the stops; however, contact was never made because the actuators were not capable of enough force. The axle needed to displace approximately 1/8 inch before that could happen (see Figure 6.7).

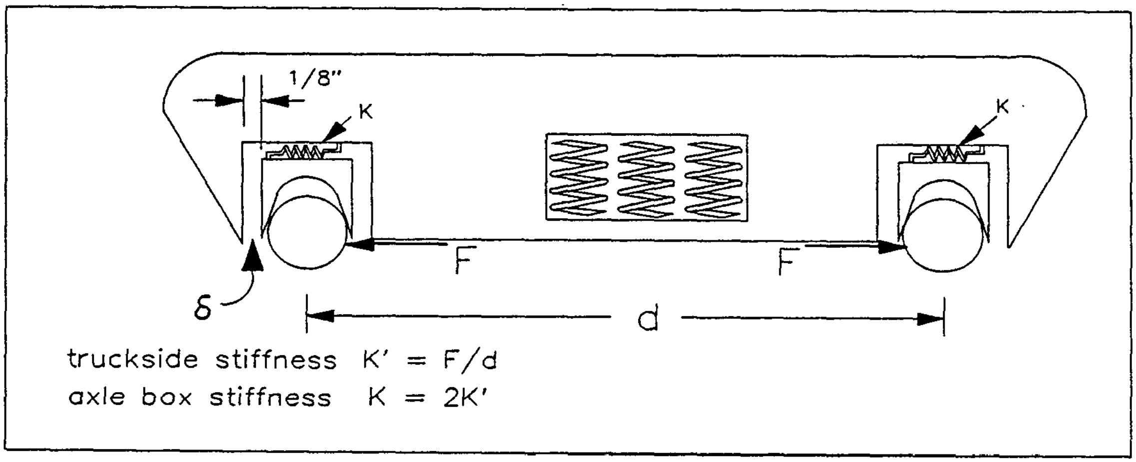


Figure 6.7 Axle Box Stiffness Test Measurements

It was necessary to provide a complete force versus displacement profile in the form of a look-up table for NUCARS. However, the axles were never displaced to the 1/8 inch necessary to reach the second stiffness of 1000 kips/inch. The first slope was extrapolated to a deflection of 1/8 inch, point 2, and the second slope, points 2 to 3, was assumed to be 1000 as shown in Table 6.6 and Figure 6.8.

The equations used to extrapolate the stiffness data are shown below. The sub-scripted numbers represent points on the graph in Figure 6.8.

$$F_2 = K_{(1-2)} * \delta_2$$
$$\delta_3 = F_3 - F_2 / K_{(2-3)} + \delta_2$$

Table 6.6 NUCARS Look-up Table for Axle Box Stiffness

	1	2	3
FORCE (kips)	0	16.7	50.0
DELTA (inches) δ	0	0.125	0.16

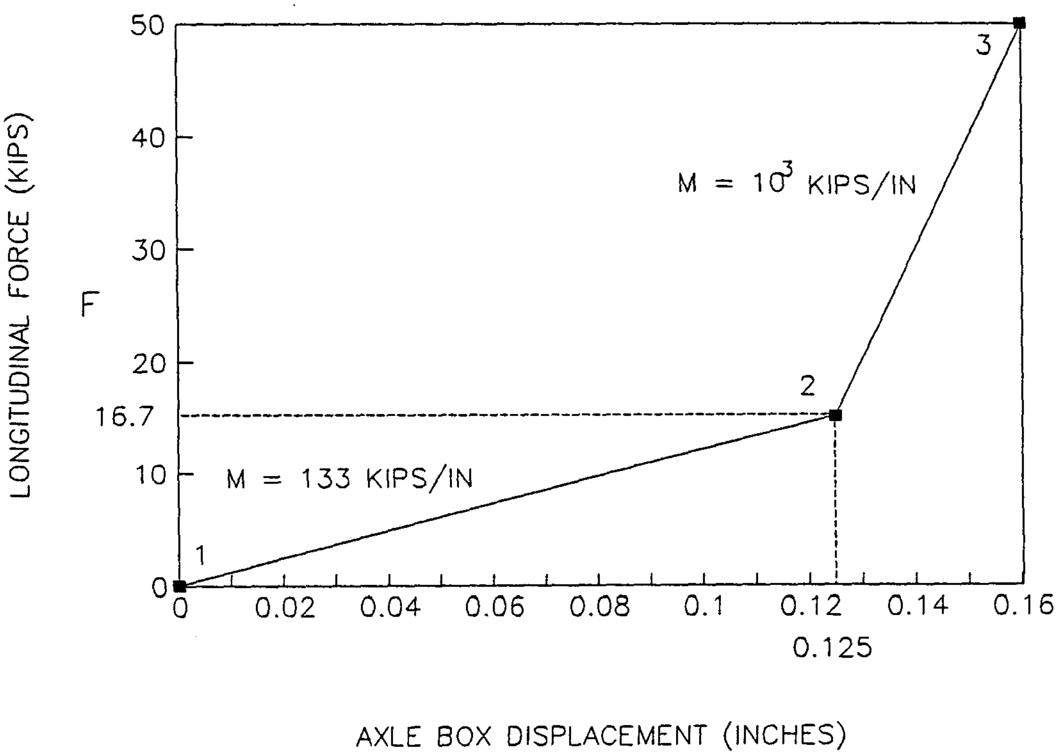


Figure 6.8 Axle Box Stiffness Graph

6.1.1.4 Axle Yaw And Inter-Axle Bending Stiffness

In curving, the axles have a tendency to yaw with respect to each other, as shown in Figure 6.9.

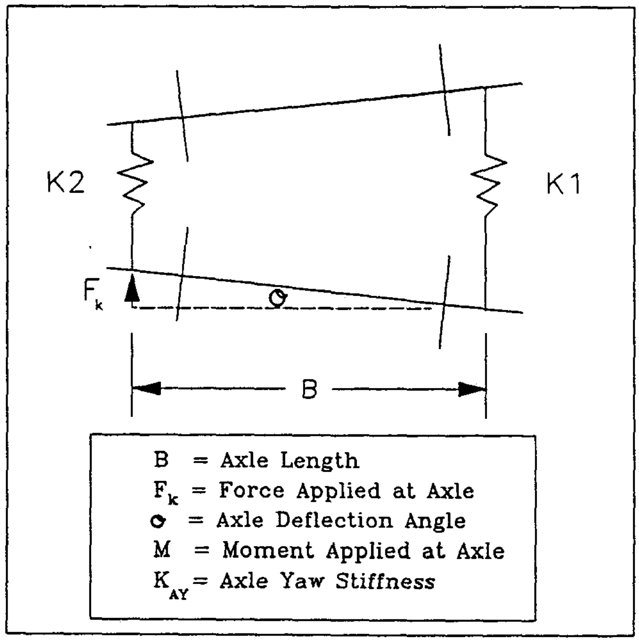


Figure 6.9 Exaggerated Axle Yaw

The first step in finding axle yaw stiffness was to calculate the stiffnesses K1 and K2 in the same manner as longitudinal stiffness. Linear regression was performed on each force versus displacement plot. Table 6.7 shows a summation of stiffness data for each test. Most of the first runs were not used and almost all of the curves had to be fit by hand due to the sliding of the bearing adapter. The calculation for average axle yaw stiffness is shown below the table. Figure 6.9 defines the variables in the equation.

Table 6.7 Axle Yaw Stiffness Summary Sheet

RUN NO.	TRUCK NO.	DIRECTION	RIGHT SIDE (LC1) SLOPE (kips/inch)	LEFT SIDE (LC2) SLOPE (kips/inch)
038	1	LC1 Push/LC2 Pull	65.25	*31.12
039	1	LC1 Push/LC2 Pull	67.29	75.26
040	1	LC1 Push/LC2 Pull	64.57	71.39
041	1	LC1 Pull/LC2 Push	*33.63	*35.93
042	1	LC1 Pull/LC2 Push	68.97	54.29
043	1	LC1 Pull/LC2 Push	71.43	53.12
044	1	LC1 Pull/LC2 Push	67.26	52.73
K1 = TRUCK 1 AVERAGE PUSH STIFFNESS LC1 and LC2 ≅ 59.5 kips/inch				
K2 = TRUCK 1 AVERAGE PULL STIFFNESS LC1 and LC2 ≅ 70.9 kips/inch				
024	2	LC1 Push/LC2 Pull	54.55	61.53
025	2	LC1 Push/LC2 Pull	57.14	70.83
026	2	LC1 Push/LC2 Pull	60.00	62.61
027	2	LC1 Pull/LC2 Push	*29.39	*37.17
028	2	LC1 Pull/LC2 Push	*41.78	55.95
029	2	LC1 Pull/LC2 Push	*37.91	59.98
K1 = TRUCK 2 AVERAGE PUSH STIFFNESS LC1 and LC2 ≅ 57.5 kips/inch				
K2 = TRUCK 2 AVERAGE PULL STIFFNESS LC1 and LC2 ≅ 65.0 kips/inch				

* Not included in calculations

$$F_k = 2(K_1 + K_2)B\theta \qquad M = F_k B = (K_1 + K_2)B^2\theta$$

$$K_{AY} = AXLE \quad YAW STIFFNESS = \frac{M}{\theta} = (K_1 + K_2)B^2$$

$$K^1_{AY} = (59.5 + 70.9)(79)^2 = 813{,}826 inch-kips/rad = 814 inch-kips/mrad$$

$$K^2_{AY} = (57.5 + 65.0)(79)^2 = 764{,}522 inch-kips/rad = 765 inch-kips/mrad$$

The final calculation yielded axle yaw stiffnesses of 814 and 765 in-kips/mrad for trucks one and two respectively. The average of 789 in-kips/mrad could be compared to that calculated by NUCARS from the longitudinal stiffness inputs. The average axle box stiffness of 66.7 kips/inch obtained in the longitudinal stiffness test would yield an axle yaw stiffness of 833 inch-kips/mrad, 6 percent higher than the axle yaw test result.

6.1.2 Quasi-static Truck Characterization Test Results

The test data was reduced using AAR developed software on the HP 370 computer system. Plots were made to display the spring and damping rates in the particular suspension component. The x-axis corresponds to the displacement measurement; the y-axis corresponds to the rail-force measurement or the load cell forces. The upper and lower slopes of the curves correspond to the stiffness in kips/inch for the vertical and lateral tests, and inch-kips/radian for the roll runs. The offset between the upper and lower slopes (hysteresis) corresponds to the damping. Figure 6.10 is a typical hysteresis plot. In this case the right vertical spring displacement versus the sum of the right vertical rail forces is shown.

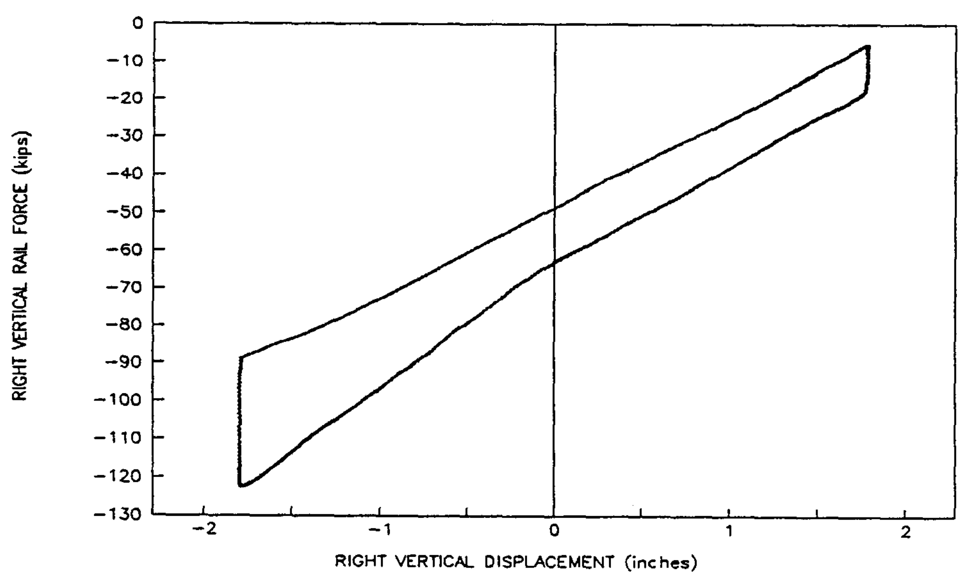


Figure 4.10 Force Versus Displacement Plot Showing Vertical Stiffness and Damping (Hysteresis) for the Right Side Truck Suspension

The secondary suspension stiffnesses for vertical, roll, and lateral tests were determined. The damping was calculated for the vertical and lateral runs. The stiffness and damping values were averaged for each truck in the vertical, roll, and lateral configurations. Table 6.8 gives the vertical secondary suspension average stiffness rates and damping for the vertical test runs at 0.1 Hz and 0.25 Hz, respectively. When observing the vertical data, a consistently higher spring rate (approximate average of 12.0 percent) for the left side over the right side on both trucks can be seen. The overall average vertical stiffness and damping were 27.0 kips/inch and 16.8 kips respectively.

Table 6.8 Secondary Suspension Average Vertical Stiffness and Damping for Test Runs @ 0.1 Hz and .25 Hz

TRUCK	FREQ (Hz)	LEFT SIDE VERTICAL DATA		RIGHT SIDE VERTICAL DATA	
		STIFFNESS	DAMPING	STIFFNESS	DAMPING
1	.1	30.6 kips/in	19.3 kips	26.4 kips/in	18.9 kips
2	.1	26.1 kips/in	11.7 kips	24.4 kips/in	13.7 kips
1	.25	31.8 kips/in	20.3 kips	25.6 kips/in	20.8 kips
2	.25	26.6 kips/in	13.0 kips	24.3 kips/in	16.5 kips

Table 6.9 lists the secondary suspension average roll stiffness for each truck. The overall average roll rate was 93,594 inch-kips/radian.

Table 6.9 Secondary Suspension Average Roll Stiffness

TRUCK	AVERAGE TRUCK ROLL STIFFNESS
1	102,196 inch-kips/radian
2	84,991 inch-kips/radian

Table 6.10 lists the secondary suspension average stiffness and damping for both trucks in the lateral case at 0.1 and 0.25 Hz. The overall average lateral stiffness and damping for a truck were 29.8 kips/inch and 36.4 kips respectively. These were divided by two, yielding 14.9 kips/inch and 18.2 kips per spring nest.

Table 6.10 Secondary Suspension Average Lateral Stiffness and Damping for Test Runs @ 0.1 Hz and .25 Hz

TRUCK	FREQ (Hz)	LEFT SIDE VERTICAL DATA		RIGHT SIDE VERTICAL DATA	
		STIFFNESS	DAMPING	STIFFNESS	DAMPING
1	.1	32.1 kips/in	39.5 kips	30.4 kips/in	38.9 kips
2	.1	28.7 kips/in	33.2 kips	31.1 kips/in	33.5 kips
1	.25	29.8 kips/in	40.1 kips	28.1 kips/in	37.2 kips
2	.25	28.8 kips/in	33.9 kips	29.0 kips/in	34.8 kips

There is little difference between the lateral stiffness and damping of Truck 1 and the lateral stiffness rates and damping of Truck 2 in Table 6.10. No further investigation was considered necessary.

6.1.3 Modal Response Test Results

Modal analysis was performed with Structural Measurements Systems (SMS) modal analysis software. Transfer functions between actuator force and transducers mounted along the car body (Figure 6.11) were created with AAR analysis software and imported into the SMS model.

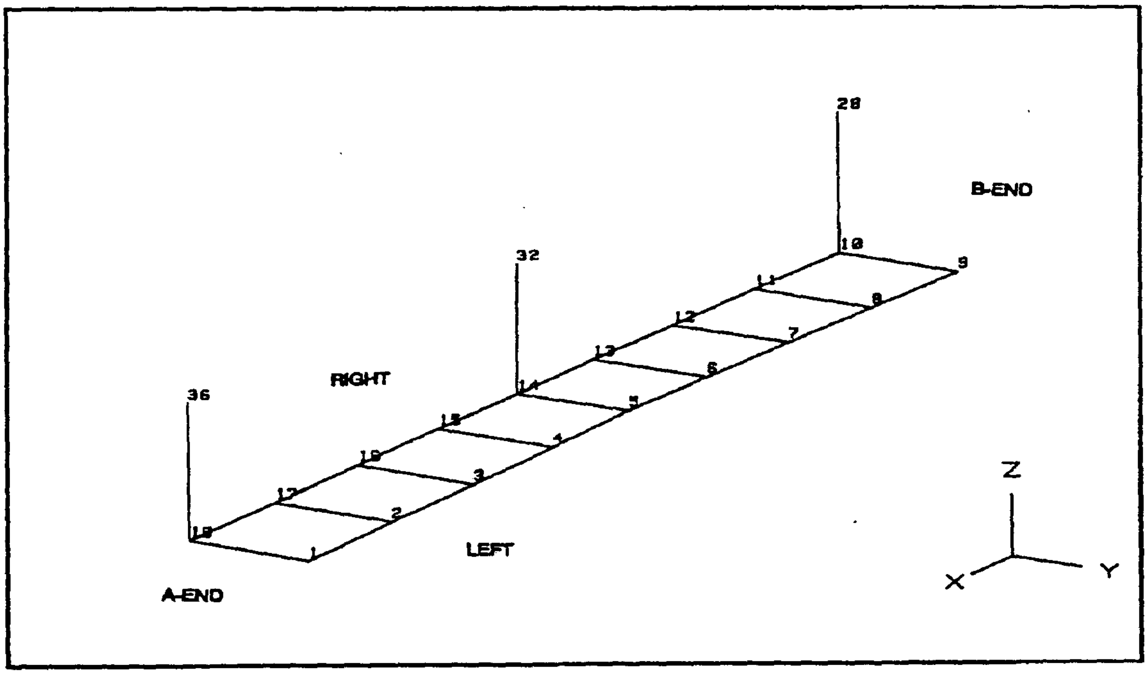


Figure 6.11 SMS Model of Maintenance Car Showing Transducers Around Car Body Center Sill and at Car Body Roof Line

Lateral and vertical accelerations were measured at every location except 3, 4, 6, 7, and 8, where only vertical accelerations were measured. In the model they were constrained laterally to locations on the other side of the car. Transfer functions between the acceleration at each location and the input force at location 2 were supplied to the model. Magnitude and phase were compared at each peak in the transfer functions and a mode shape was calculated and displayed by SMS. The transfer functions and the phase relationships between accelerations along the body were used to differentiate between modes.

6.1.3.1 Rigid Body Vertical

Data from run RN033 was analyzed to obtain pitch and bounce in the loaded configuration. The matrix below shows the pitch and bounce resonant frequencies.

MODE	FREQUENCY
Pitch	6.9 Hz
Bounce	2.5 Hz

A 15-kip sinusoidal load was applied by each actuator in-phase. The sine wave was swept from 0.5 Hz to 10 Hz in steps of 0.1 Hz. A transfer function of the actuator input force and a vertical accelerometer near the B-end (location 7) is shown in Figure 6.12

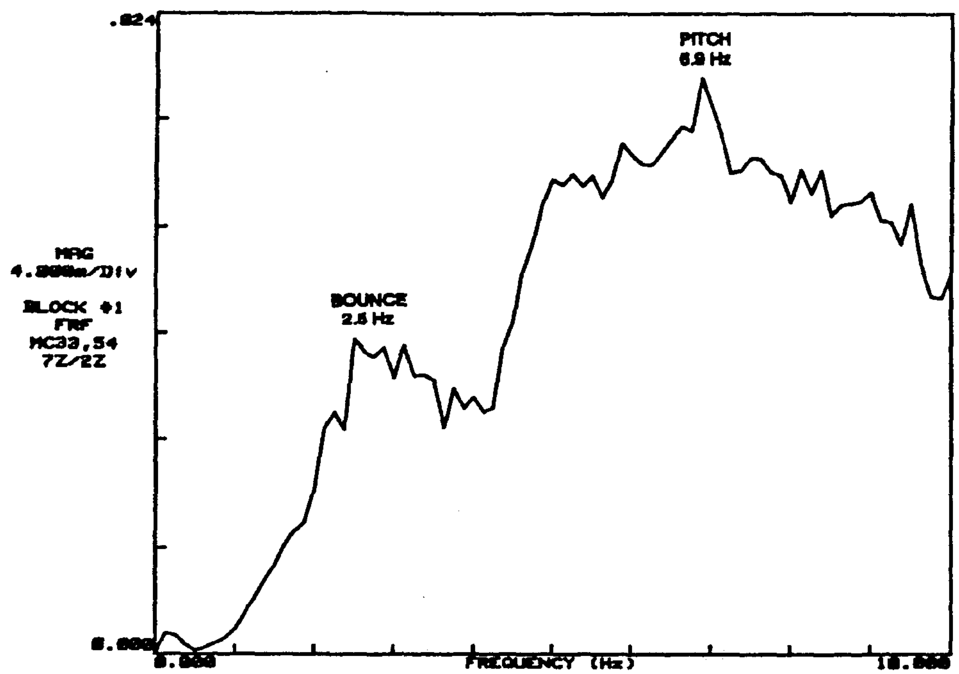


Figure 6.12 Transfer Function Plot of Vertical Acceleration Near B-end of Car versus Input Force

Pitch and bounce were determined by the phase relationship between the A- and B-end vertical accelerations. Figure 6.13 shows the phase of a location (3) near the A-end and a location (7) near the B-end. The two vertical lines represent the frequency at which the two peaks were found on the previous transfer function plot.

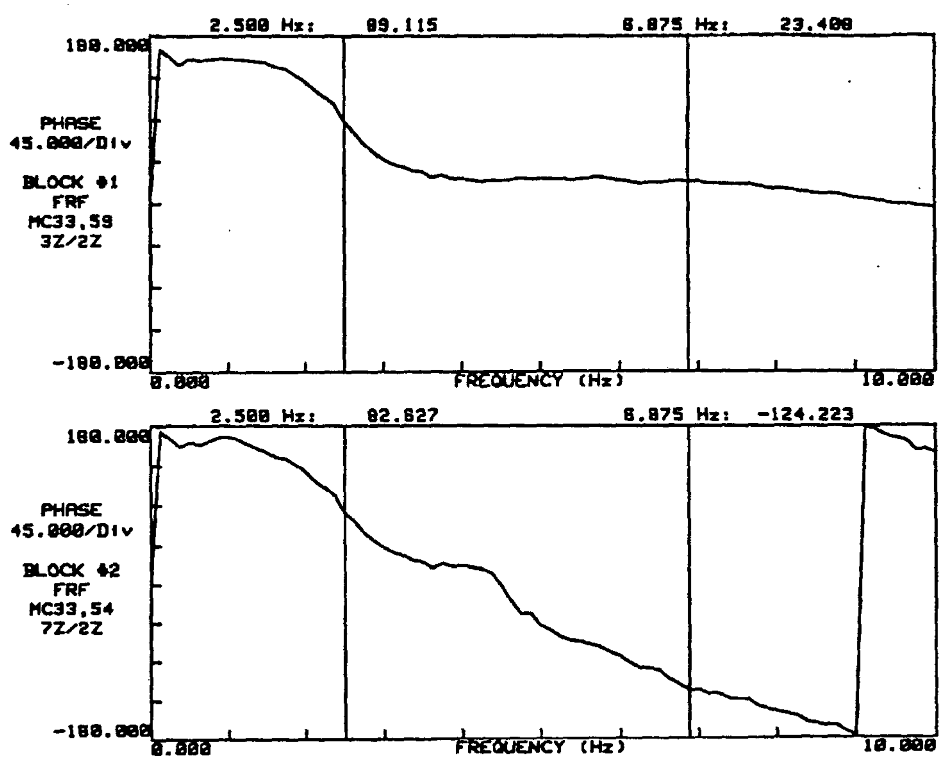


Figure 6.13 A- and B-End Vertical Phase Relationship

At 2.5 Hz, location 3 was at 89 degrees and location 7 was at 83 degrees, essentially in-phase, indicating bounce. At 6.9 Hz, location 3 was at 23 degrees and location 7 was at -124 degrees. The difference between the two was 147 degrees, slightly less than 180 degrees but close enough to indicate a pitching motion.

Figure 6.14 indicates the bounce mode shape as displayed by SMS. The base of the car is shown. Locations 28, 32, and 36 were removed for clarity. The smaller lines are vectors indicating relative amplitude and direction of motion for each location.

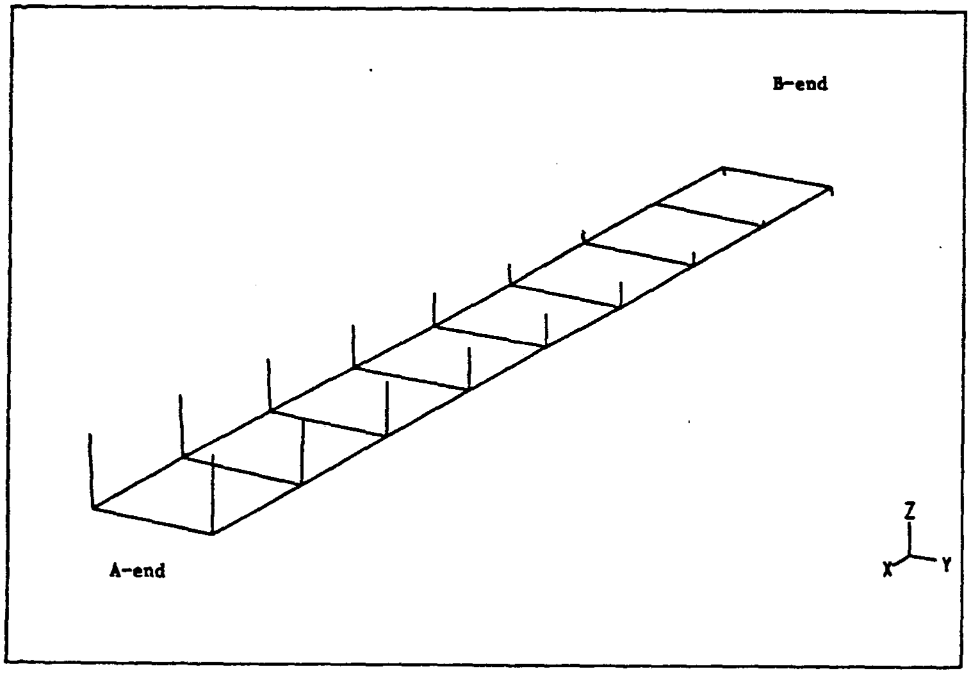


Figure 6.14 Bounce Mode Shape Plot with Vectors

A pure bounce mode would show all locations along the car moving vertically in the same direction with the same amplitude. The case shown here is typical of that excited by the MSU. The car was actually pitching about the trailing truck. It is difficult to excite bounce at the rear of the car with actuators at the front only. Figure 6.15 shows the same type vector plot for pitch.

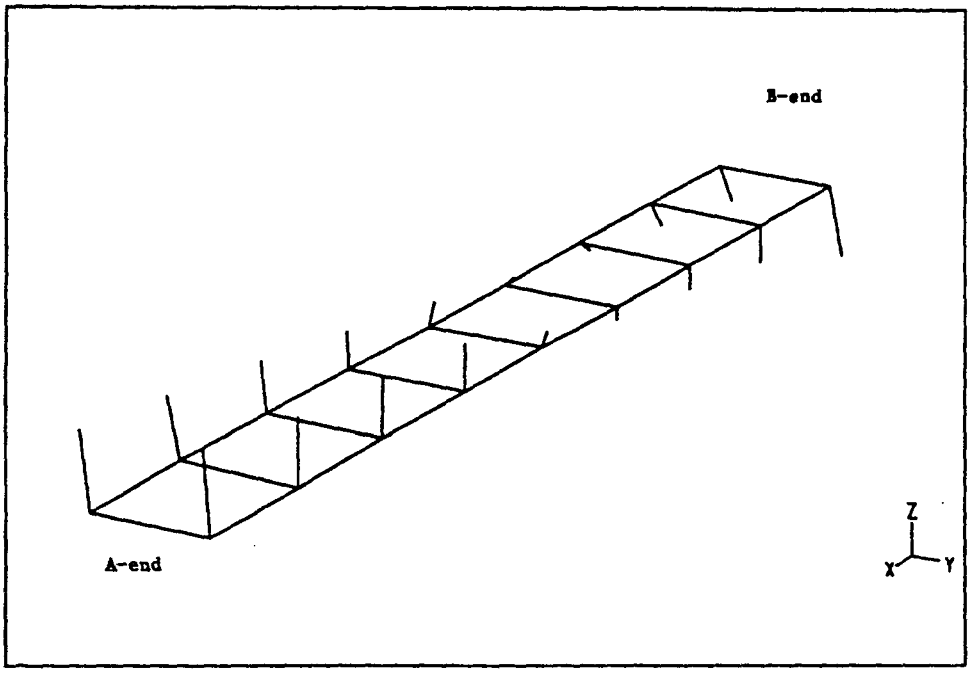


Figure 6.15 Pitch Mode Shape Plot with Vectors

The relative magnitude of motion decreases towards the center of the car then increases again towards the B-end. At the A-end the direction of motion is in the positive z direction, while the B-end is moving in the negative z direction. The center is hardly moving at all. This indicates vertical pitch about the center of the car. Ideally the vectors would have no y component. However, this plot indicates that the car was moving laterally at the same time, probably rolling slightly. This coupling of rigid body modes is common due to their closeness in resonant frequencies.

6.1.3.2 Rigid Body Roll

Data from run RN036 was analyzed to obtain lower and upper center roll in the loaded configuration. The matrix below shows upper and lower center roll resonant frequencies.

MODE	FREQUENCY
Lower Center Roll	.75 Hz
Upper Center Roll	5.0 Hz

A 15 kip sinusoidal load was applied by each actuator out of phase. The frequency sweep was from 0.2 Hz to 5 Hz in steps of 0.1 Hz. A transfer function of one actuator input force and an A-end vertical accelerometer is shown in Figure 6.16. There are two very distinct peaks.

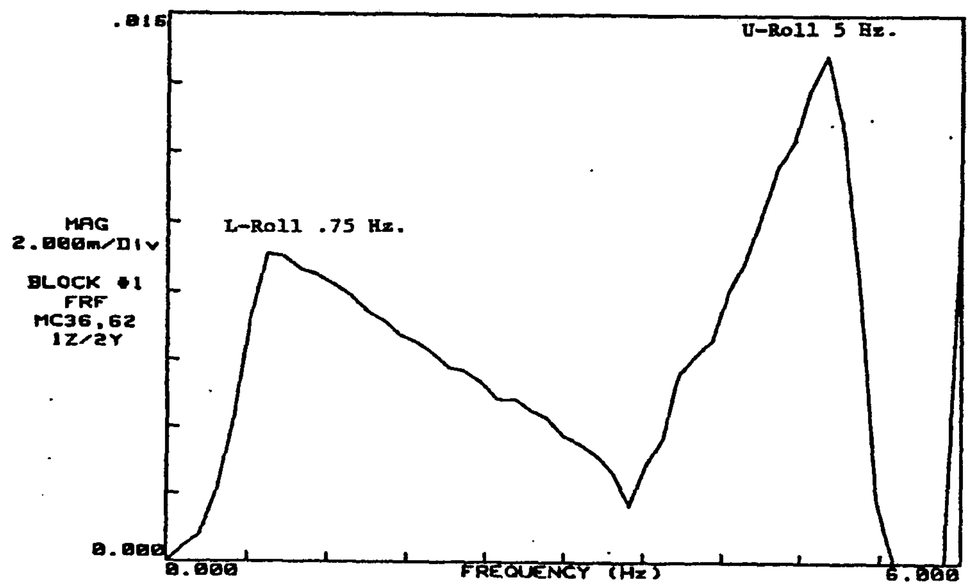


Figure 6.16 Transfer Function Plot of Vertical Acceleration Versus Actuator Force

The phase relationship between right and left side vertical accelerations was examined to determine if roll was present. Figure 6.17 shows the phase of the A-end (18) right vertical accelerometer (top plot) and the A-end left (1) vertical accelerometer (bottom plot). The vertical lines indicate the two frequencies at which the peaks were observed on the previous transfer function.

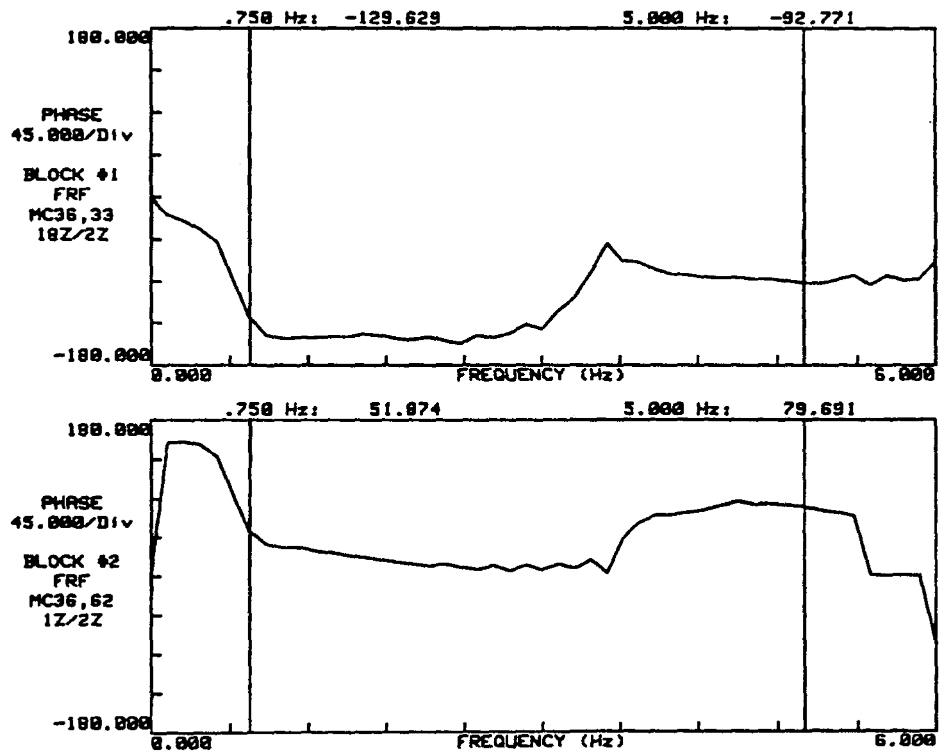


Figure 6.17 Right and Left Side Vertical Phase Showing Roll

At 0.75 Hz, the right side phase was -130 degrees and the left was 52 degrees. The difference was 182 degrees. This indicates that while the left side was moving down the right side was moving up, indicating roll. At 5.0 Hz, the phase difference was 173 degrees, again indicating roll or twist. The B-end phases were also examined to ensure that the phase relationship was similar to the A-end. If the car was twisting, the A- and B-end right side accelerometers would have been out-of-phase. The A-end would have been rolling right while the B-end was rolling left. In this case the A- and B-ends were in-phase. Figure 6.18 shows the A-end of the MC with vectors indicating the relative magnitude and direction of motion at 0.75 Hz.

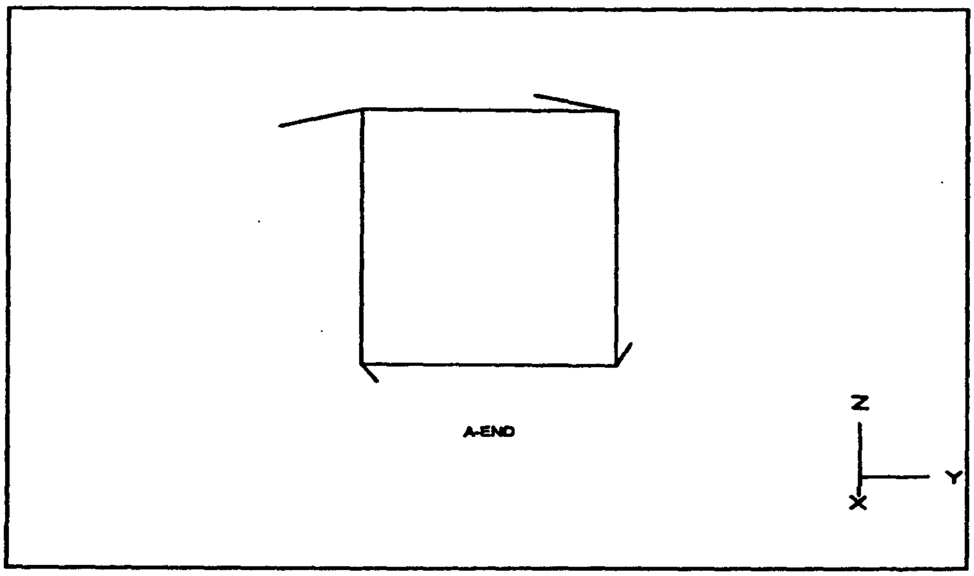


Figure 6.18 Lower Center Roll at A-End

The bottom of the car experienced very little lateral motion compared to the top of the car. The vertical displacements were nearly equal from top to bottom. The difference laterally would indicate that the car was rolling about some point near the bottom of the car. The top of the car, at a longer radius, swept a larger arc than the bottom of the car at a shorter radius. The upper center roll mode is shown in a similar fashion in Figure 6.19.

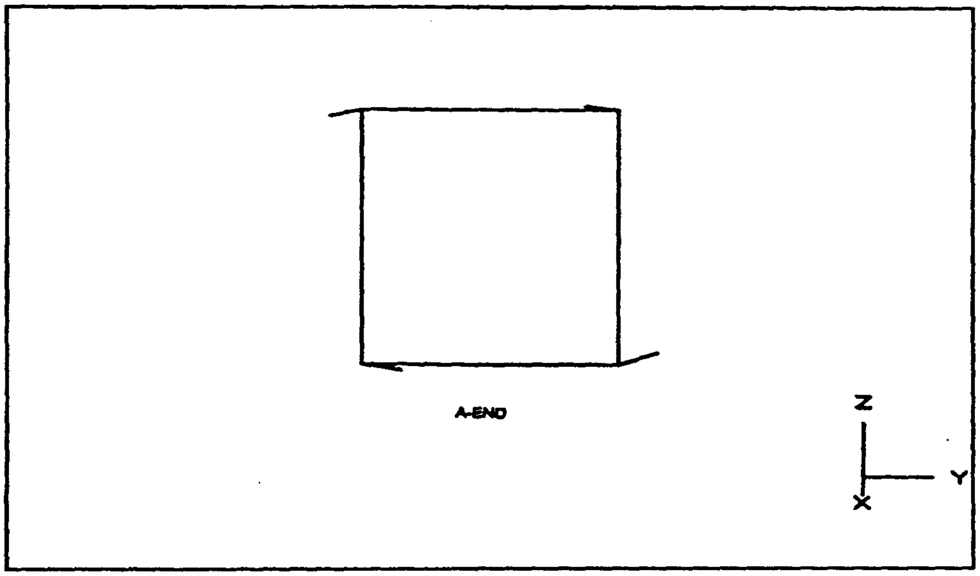


Figure 6.19 A-End Upper Center Roll Mode Shape

The primary difference between lower and upper center roll is the point which the car rolls about. The magnitudes of the vectors were nearly equal in both lateral and vertical directions. This indicates that the car was rolling about some point within the car body. Since the lower vectors are slightly longer, this indicates that the roll center was somewhere between the halfway point and the top of the car.

6.1.3.3 Flexible Body Vertical

Data from run RN040 was analyzed to obtain first vertical bending in the loaded configuration. The matrix below shows the first vertical bending resonant frequency.

MODE	FREQUENCY
First Vertical Bending	20.2 Hz

A constant acceleration of 0.4 g was used in a frequency sweep which started at 5 Hz and finished at 30 Hz. Figure 6.20 is a transfer function between input force and car center vertical acceleration. Large peaks are present at 15.0 and 20.2 Hz.

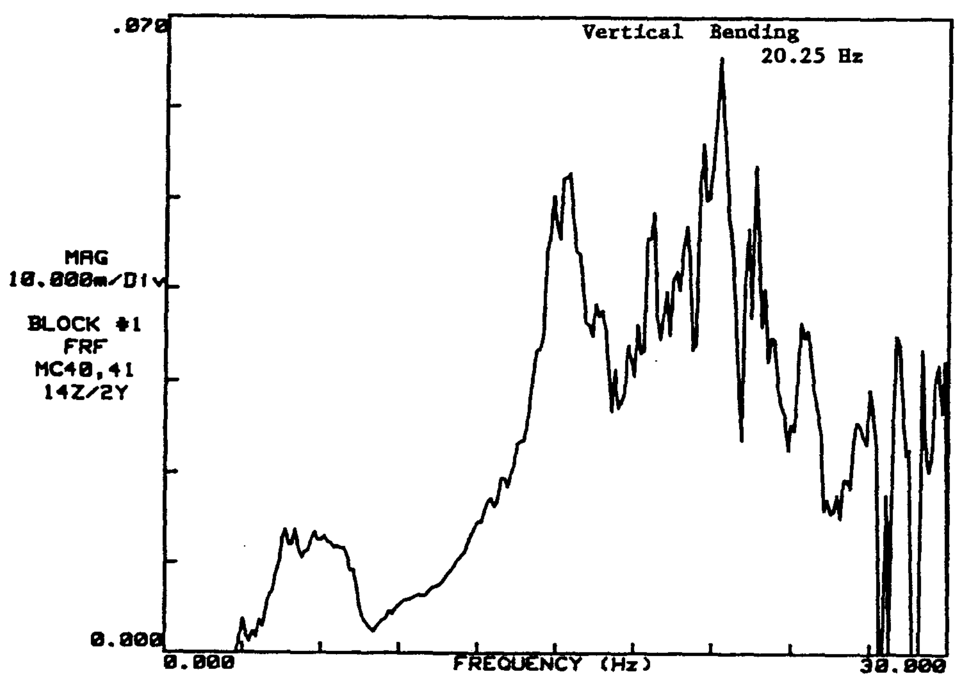


Figure 6.20 Transfer Function Plot of Car Center Acceleration Versus Actuator Force

Other frequencies were examined for vertical bending including 15 Hz, which is also prevalent in Figure 6.20. Twenty hertz seemed like a high vertical bending frequency but no other frequency exhibited the shape shown in Figure 6.21.

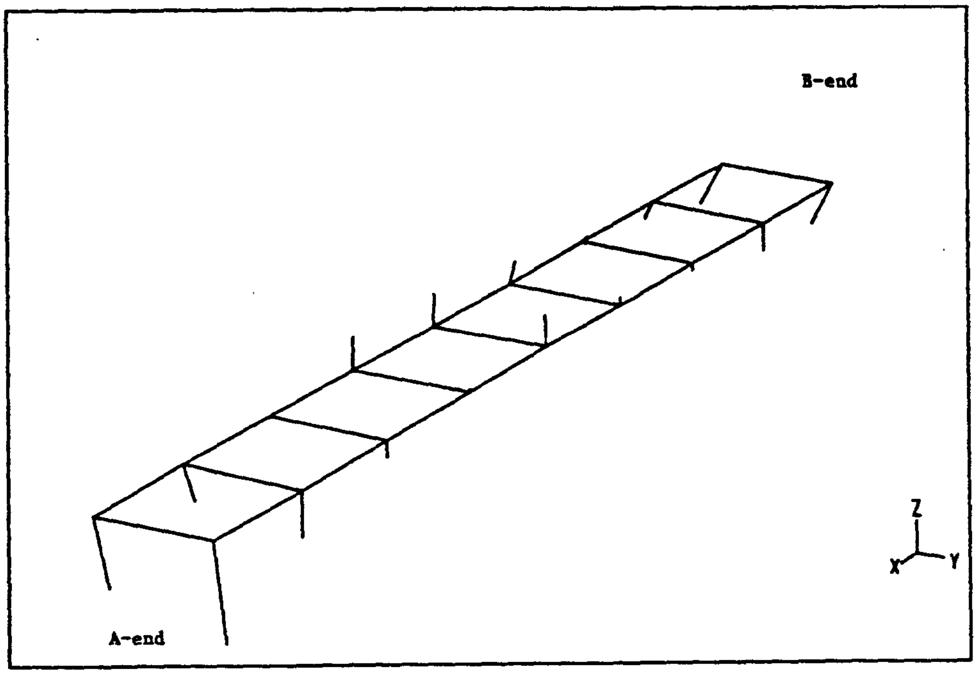


Figure 6.21 Vertical Bending Mode Shape Plot with Vectors

The vectors near the ends of the car point down where the vectors near the center of the car point up. This implies that the center of the car was out-of-phase with the ends, indicating first vertical bending.

6.1.3.4 Flexible Body Torsion

Data from run RN044 was analyzed to obtain first twist in the loaded configuration. The matrix below shows the first twist resonant frequency.

MODE	FREQUENCY
First Twist	8.1 Hz

A constant acceleration of 0.4 g was used in a frequency sweep from 5 Hz to 30 Hz. Figure 6.22 shows twist in a transfer function between input force and left side A-end vertical acceleration.

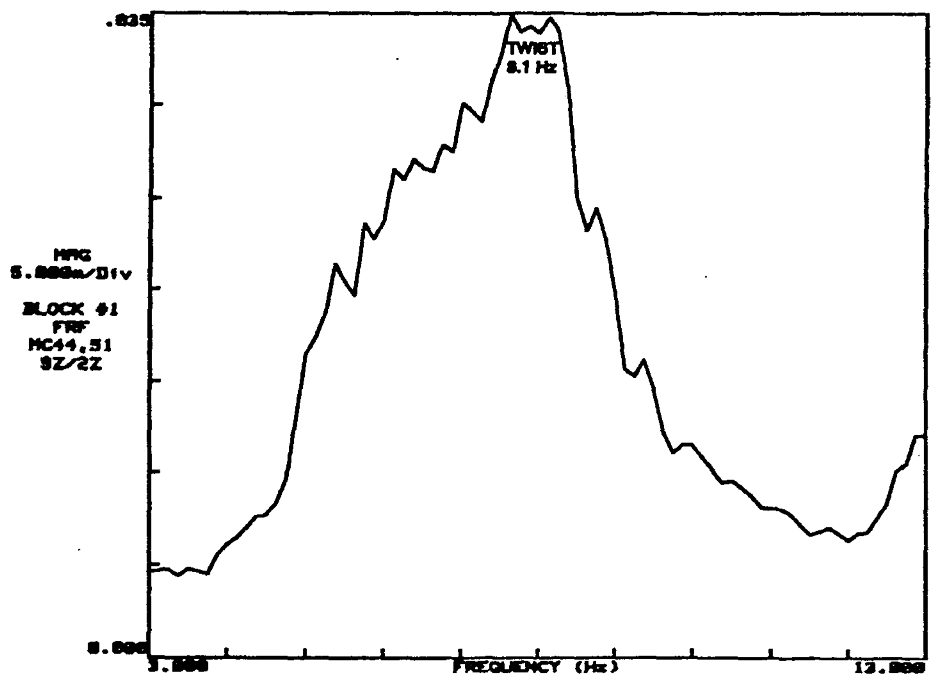


Figure 6.22 Transfer Function Plot of Left Side A-End Acceleration Versus Actuator Force

As discussed in the roll section, the first longitudinal torsion or twist mode can be described as the A-end rolling right while the B-end is rolling left. The 8.1 Hz peak in Figure 6.22 was verified as twist by checking the phase of locations 1 and 10. They are in opposite corners of the car diagonally. In twist they should be in-phase. Figure 6.23 shows the relationship between those locations. The upper plot is location 1 and the lower plot is location 10.

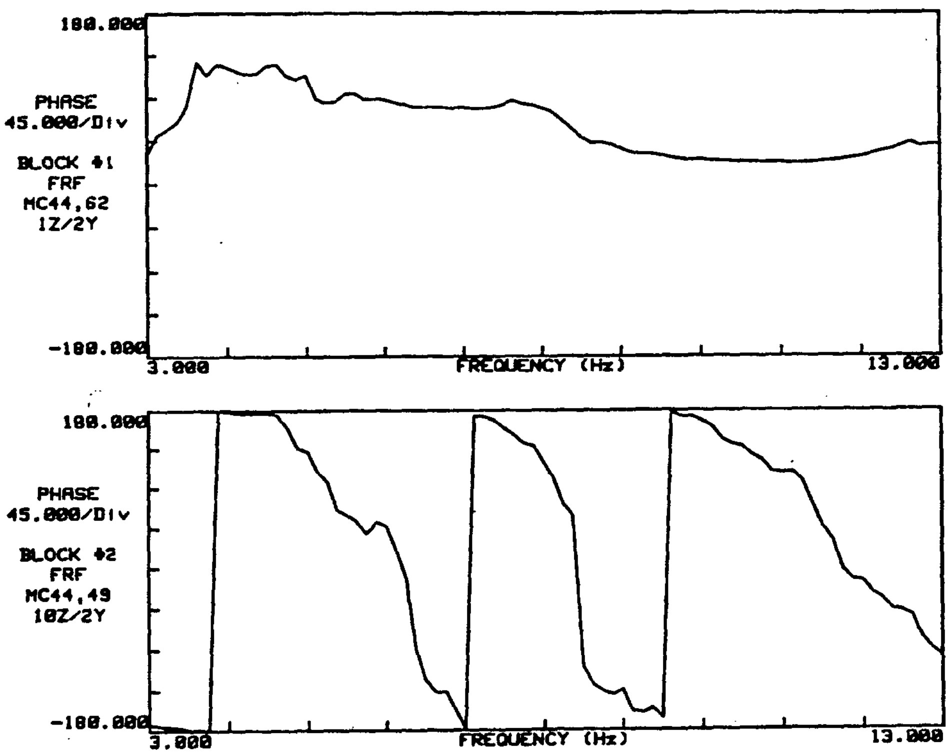


Figure 6.23 Phase Relationship of Two Locations Showing Twist

The phase of the two locations was equal at 8.1 Hz. The lines at 4, 7, and 10 Hz are data rollover. When the phase passes -180 degrees the next point is placed at 180 degrees so that the scale of the plot stays within reasonable bounds. Since the difference is 360 degrees it can also be called 0 degrees.

The mode shape was examined at 8.1 Hz in more detail. Figure 6.24 shows the mode shape of the four corners of the car in relative deflection rather than vector format.

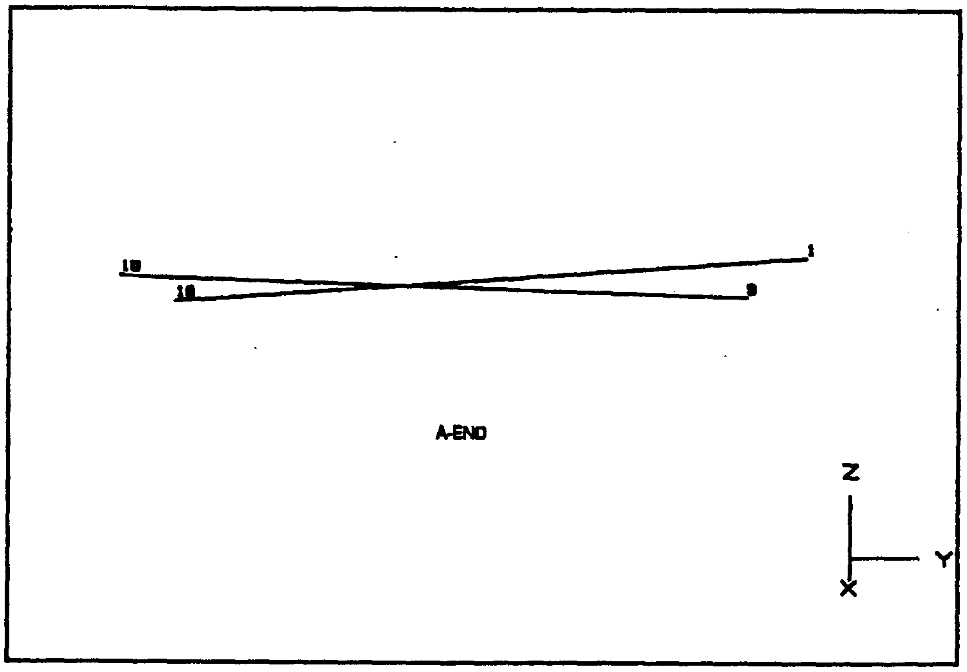


Figure 6.24 Relative Deflection of MC Corners in Twist

Locations 1 and 18 represent the A-end corners and locations 9 and 10 represent the B-end corners. The B-end was rolling right while the A-end rolled left. The ends of the car were also displaced laterally as seen in Figure 6.24. This was probably due to a small yaw influence while twisting.

Twist was further confirmed by performing a curve fitting analysis with SMS at that frequency. The motion simulated by SMS at that frequency was twist.

6.1.3.5 Rigid Body Lateral

Data from run RN078 was analyzed to obtain yaw and sway in the loaded configuration. The matrix below shows the yaw and sway resonant frequencies.

MODE	FREQUENCY
Sway	Undetermined
Yaw	4.7 Hz

A 15 kip sinusoidal load was applied laterally with one actuator. The sine wave was swept from 0.5 Hz to 10 Hz in 0.1 Hz increments. Figure 6.25 shows the yaw frequency and what was initially believed to be sway in a transfer function of right side lateral acceleration near the A-end versus input force. Peaks at 1.2 and 4.7 Hz were evident.

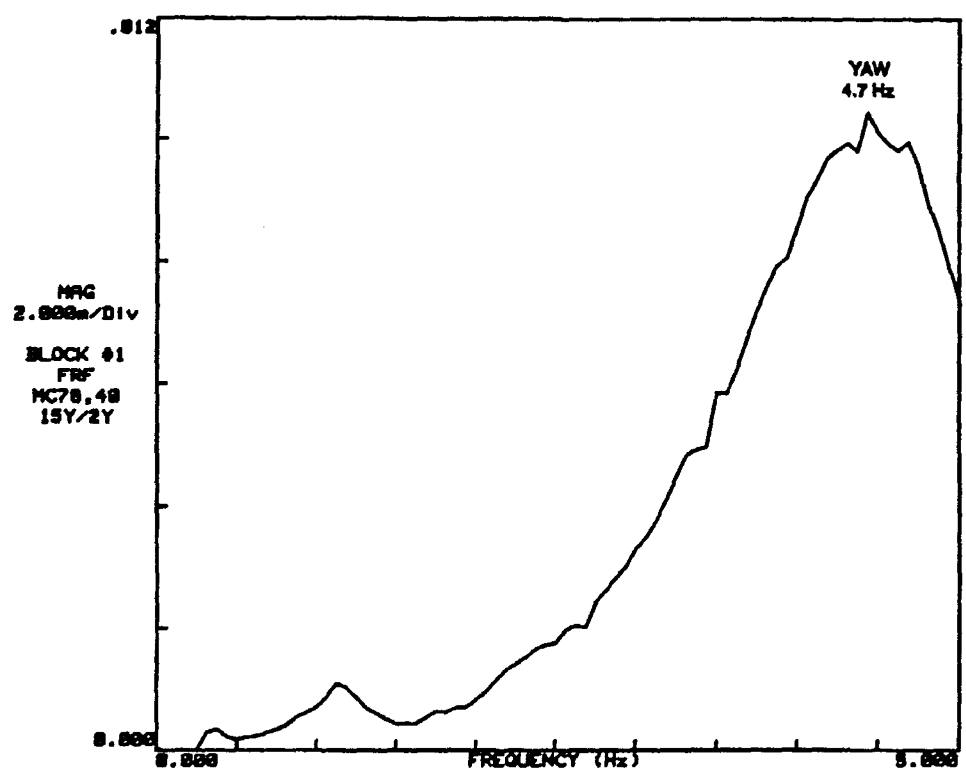


Figure 6.25 Transfer Function Plot of A-End Right Acceleration versus Input Force

Figure 6.26 shows the mode shape at 1.2 Hz with vectors at the A-end of the car. It looks more like lower center roll than sway. Sway should be pure lateral motion rather than roll. Lower center roll was previously found at 0.75 Hz.

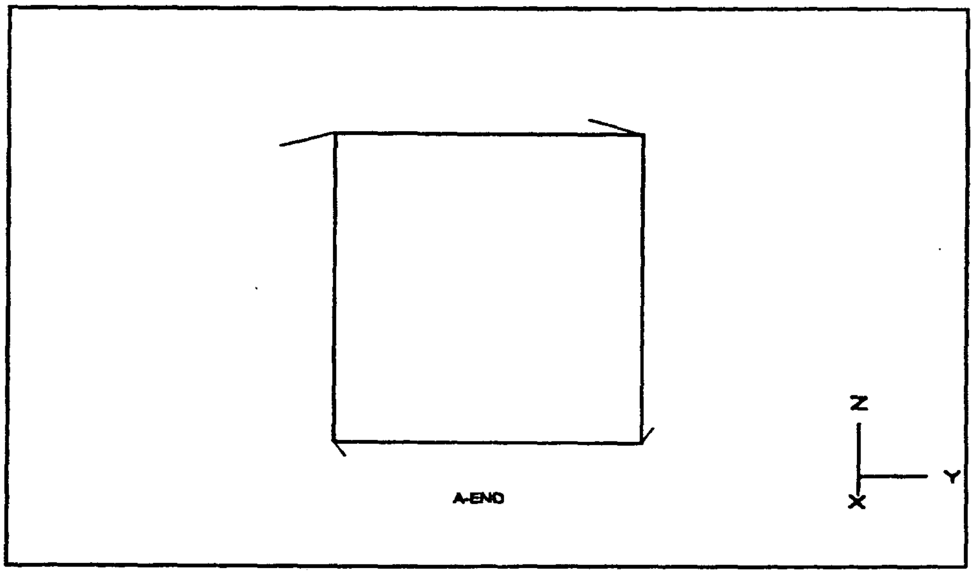


Figure 4.26 Roll Mode Shape Plot From A-End

The mode shown was not a pure lateral or sway motion. The resonant frequency for sway was not determined.

The yaw mode was verified at 4.7 Hz by comparing the phase of the A- and B-End lateral accelerations as in Figure 6.27.

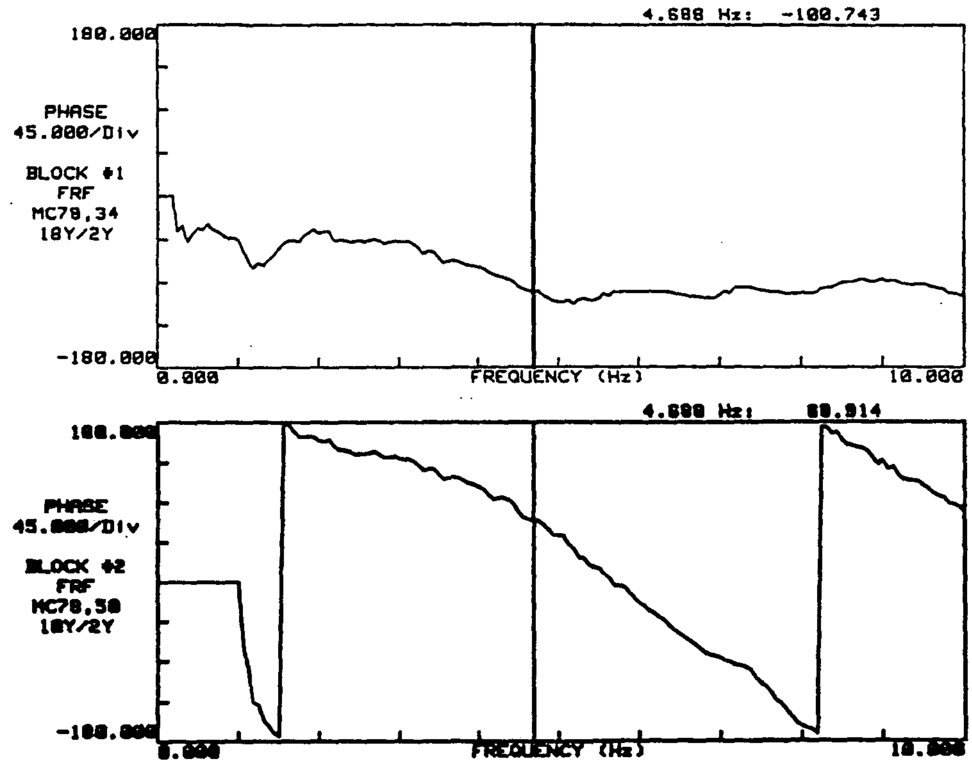


Figure 6.27 A- and B-End Phase Comparison Showing Yaw

The top graph represents the phase at the A-end (location 18) and the bottom graph represents the phase at the B-end (location 10). The vertical line through both graphs is at 4.7 Hz. The difference between the two phases of -101 and 69 degrees is 170 degrees, indicating yaw. Figure 6.28 is a top view of the car floor showing yaw in a relative displacement format. The single line shows the normal orientation of the car.

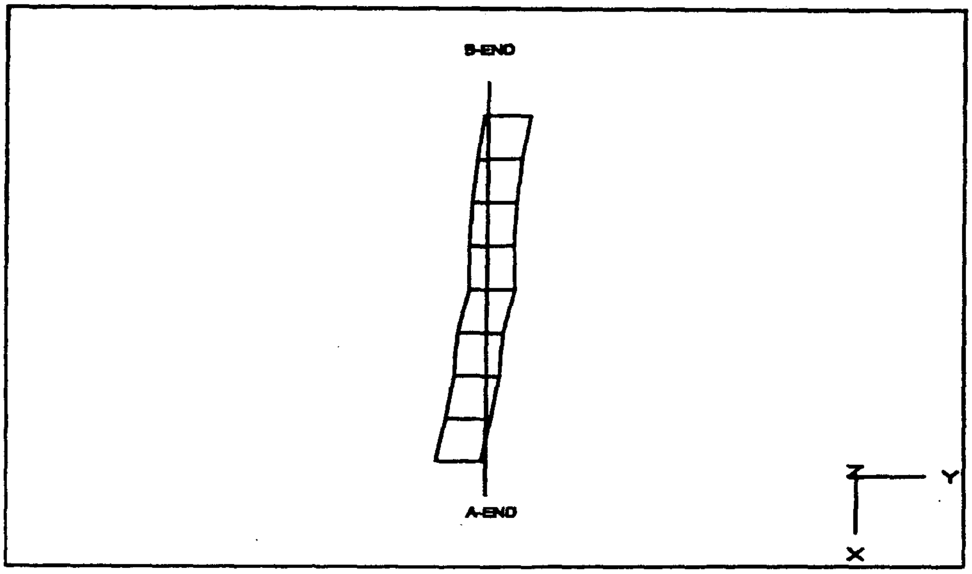


Figure 6.28 Yaw Mode Shape in Top View

Ideally the car should still look rigid, but it may have been twisting slightly while yawing.

6.1.3.6 Flexible Body Lateral

Data from run RN083 was analyzed to obtain first lateral bending in the loaded configuration. The matrix below shows the first lateral bending resonant frequency.

MODE	FREQUENCY
First Lateral Bending	14.9 Hz

A constant acceleration of 0.4 g was used in a frequency sweep from 5 Hz to 30 Hz. Figure 6.29 shows the transfer function between car center lateral acceleration and lateral actuator force.

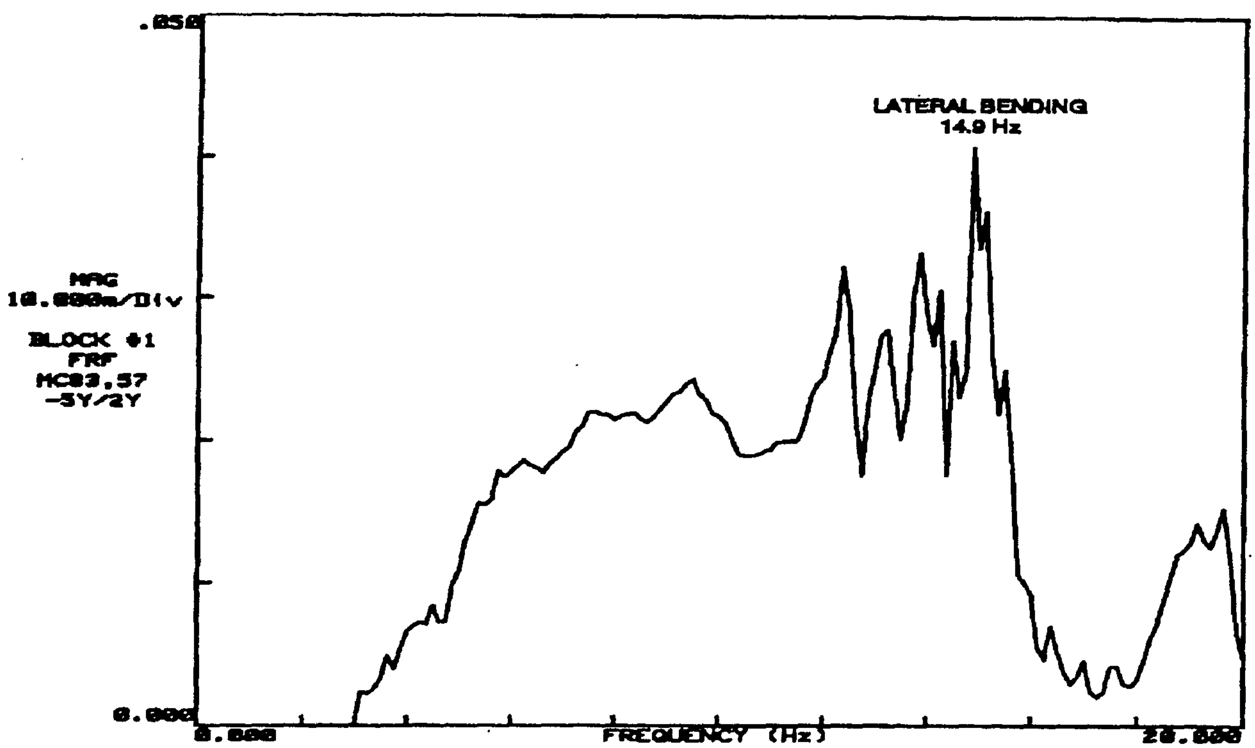


Figure 6.29 Transfer Function Plot of Car Center Lateral Acceleration versus Actuator Force

The highest peak in the transfer function at the center of the car usually indicates first bending. Several other frequencies were examined, but none yielded the shape shown in Figure 6.30.

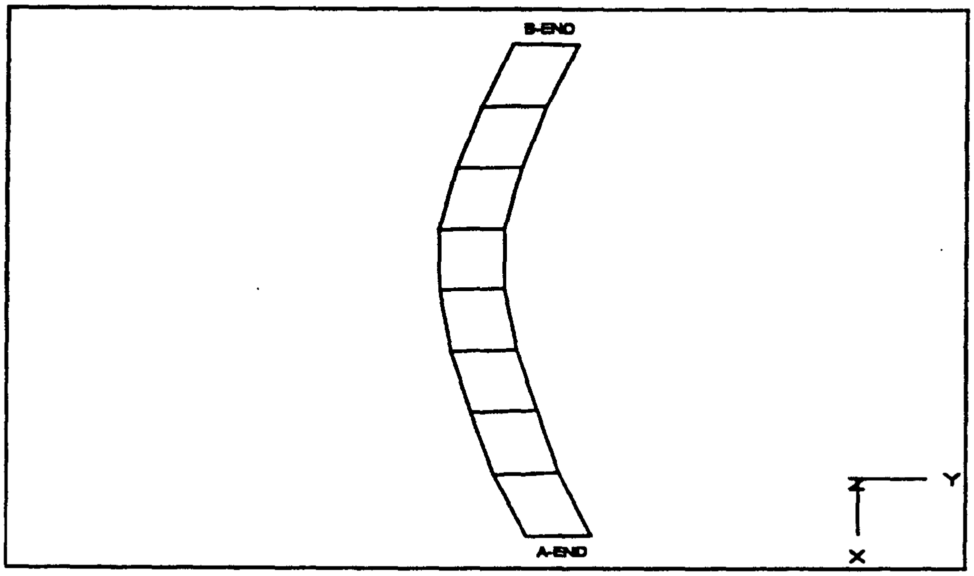


Figure 6.30 Exaggerated Lateral Bending Shape in Top View

The shape shown in Figure 6.30 is exaggerated. It is nearly symmetric, as it should be. The actuator location may have had some affect on the mode shape, as with all modes.

6.1.4 Vehicle Characterization Results Summary

Tables 6.11 and 6.12 present a summary of characterization data which was provided for NUCARS.

Table 6.11 Truck Characterization Summary

PARAMETER	VALUE	
	STIFFNESS	DAMPING
Secondary Suspension Vertical	27.0 kips/in	16.8 kips
Secondary Suspension Lateral	14.9 kips/in	18.2 kips
Truck Roll Rate	93,594 in-kips/rad	
Truck Yaw Moment	222,000 in-lbs	
Axle Alignment	No Effect	
Axle Box Longitudinal Stiffness	133 kips/inch	
Inter Axle Yaw and Bending Stiffness	789 in-kips/mrad	

Table 6.12 Modal Summary

PARAMETER	FREQUENCY
Bounce Frequency	2.5 Hz
Pitch Frequency	6.9 Hz
Roll Frequency Lower Center	0.75 Hz
Roll Frequency Upper Center	5.0 Hz
Sway Frequency	Undetermined
Yaw Frequency	4.7 Hz
First Vertical Bending Frequency	20.2 Hz
First Torsional Frequency	8.1 Hz
First Lateral Bending Frequency	14.9 Hz

6.2 SERVICE WORTHINESS

6.2.1 Curve Stability Results

The MC was placed in a consist on a 10-degree curve with less than 1/2 inch super-elevation. A 200,000 pound buff and draft (compression and tension) force was applied to the MC in the consist. The MC was monitored for wheel lift with the wheel lift gage on the inside of the curve for buff and the outside of the curve for draft. No wheel lift occurred on the MC during the Curve Stability Test.

6.3 TRACK WORTHINESS

Chapter XI criteria were used as a guideline to measure the performance of the MC and to indicate safe conduct of each test. **The tests were not performed to certify the MC.** The criteria were not used as pass/fail.

6.3.1 Yaw and Sway Results

Chapter XI specified two limiting criteria for yaw and sway testing. The first criterion was a maximum absolute axle sum L/V of 1.3 sustained for 50 milliseconds. The second limiting criterion was a maximum instantaneous truck side sum L/V of 0.6 sustained for 50 milliseconds or 6 feet. In order to obtain truck side L/V's, the leading truck had two instrumented wheel sets.

The measured values were within the limiting criteria. Note that the perturbations on the actual track were approximately 1.0 inches, somewhat less than the Chapter XI specified 1.25 inches. Table 6.13 is a tabulation of yaw and sway results.

Table 6.13 Yaw and Sway Results

SPEED	MAXIMUM AXLE SUM L/V AXLE 1	MAXIMUM AXLE SUM L/V AXLE 2	MAXIMUM TRUCK SIDE L/V LEFT	MAXIMUM TRUCK SIDE L/V RIGHT
30	0.43	0.24	0.08	0.20
40	0.47	0.28	0.12	0.21
50	0.80	0.52	0.35	0.40
55	1.02	0.89	0.48	0.58
57	1.06	0.93	0.57	0.62
60	1.03	0.94	0.60	0.65

Right truck side L/V's exceeded 0.60 at 57 and 60 mph but for less than 50 milliseconds. At 60 mph, 6 feet equates to 68 milliseconds so the 50 millisecond criteria is more restrictive. Either way the car's performance was within Chapter XI at speeds of 60 mph and less. Axle sum L/V's were substantially lower than Chapter XI limiting criteria. Figures 6.31 and 6.32 contain test results compared to the Chapter XI limiting criteria.

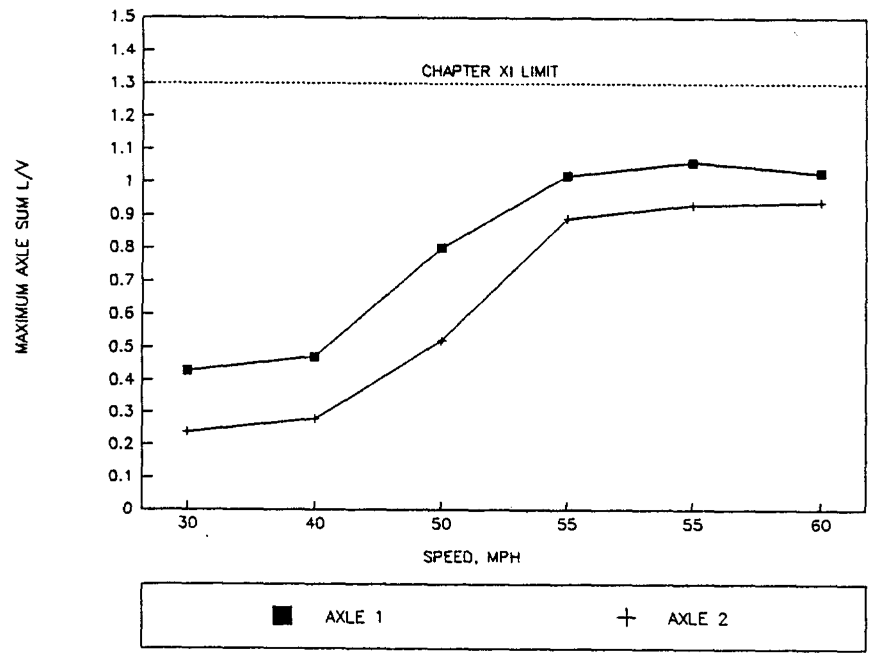


Figure 6.31 Yaw and Sway Axle Sum L/V Results

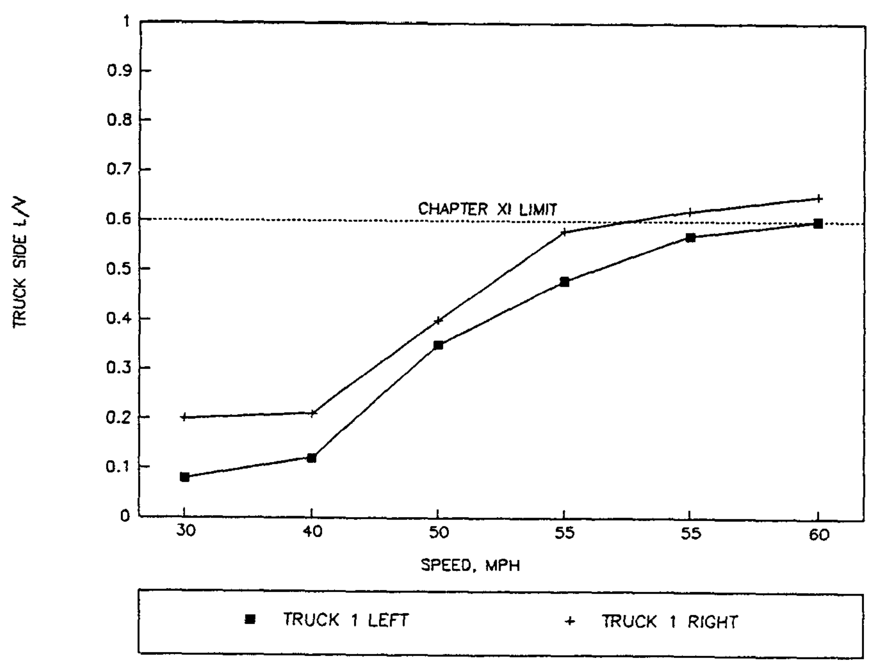


Figure 6.32 Yaw and Sway Truck Side L/V Results

6.3.2 Hunting Results

There were three limiting criteria for the Hunting Test: (1) maximum axle sum L/V of 1.3 sustained for 50 milliseconds, (2) maximum peak-to-peak lateral acceleration of 1.0 g sustained for 20 seconds, and (3) maximum single occurrence acceleration of 1.5 g peak-to-peak. The trailing instrumented wheel set was moved from the lead truck to the leading position on the trailing truck for the rest of the track worthiness tests. The maximum test speed was 70 mph. Table 6.14 is a tabulation of the hunting results.

Table 6.14 MC Hunting Results

SPEED (mph)	MAXIMUM PEAK-TO-PEAK LATERAL ACCELERATION (g)				MAXIMUM AXLE ABSOLUTE SUM L/V	
	A-END	ABOVE TRUCK 1	B-END	ABOVE TRUCK 2	AXLE 1	AXLE 3
30	0.26	0.18	0.23	0.33	0.26	0.24
40	0.29	0.25	0.29	0.37	0.25	0.36
50	0.36	0.26	0.28	0.34	0.26	0.24
60	0.41	0.28	0.36	0.40	0.29	0.25
70	0.47	0.36	0.43	0.06	0.29	0.28

Car body acceleration was measured at the ends of the car and on the deck above each center plate. At 70 mph, the maximum lateral peak-to-peak car body acceleration was 0.47 g. This value, which was not sustained for 20 seconds, was half the Chapter XI limit of 1.0 g. The stability of the car could be attributed to: (1) new trucks with close tolerances, (2) constant contact side bearings, and (3) a loaded configuration. The hunting test is usually performed on an empty car.

6.3.3 Pitch and Bounce Results

The performance criterion listed in Chapter XI for pitch and bounce was in reference to minimum vertical wheel load. The limit was 10 percent of the static wheel load.

The first step in data analysis was to determine the static wheel load for each instrumented wheel. Low speed twist and roll test runs were analyzed to determine the rolling unperturbed vertical wheel load. The entrance zone to twist and roll is tangent track and is well maintained. Runs of 14 mph to 22 mph were analyzed. The average wheel load for each wheel was recorded. The vehicle weight was also estimated from those loads. Table 6.15 shows that tabulation.

Table 6.15 MC Rolling Unperturbed Wheel Loads

RUN	SPEED (mph)	AXLE 1 LEFT	AXLE 1 RIGHT	AXLE 3 LEFT	AXLE 3 RIGHT	ESTIMATED VEHICLE WEIGHT
16	14	24.88	27.72	24.94	25.71	206.50
17	16	24.86	27.62	25.95	25.70	206.26
18	20	24.58	28.25	24.51	26.38	207.44
19	22	24.92	27.71	24.97	25.77	206.74
AVERAGE:		24.81	27.83	24.84	25.89	206.74
STD DEV:		0.13	0.25	0.19	0.28	--

NOTE: All loads in kips

Axle 1 was the leading axle on the A-end of the car. Left and right were referenced by standing at the B-end facing toward the A-end. The estimated vehicle weight was 206,740 pounds. This was slightly higher than the 205,300 pound weight as measured on the TTC scale. The estimated weight is used to check the instrumented wheel sets.

Table 6.16 is a tabulation of the minimum vertical wheel loads for the pitch and bounce test runs. Figure 6.33 shows a comparison of actual and limiting values. The lowest vertical wheel load was 68 percent at 60 mph. This was much higher than the limiting criterion of 10 percent.

Table 6.16 Pitch and Bounce Test Results Summary

SPEED (mph)	MINIMUM VERTICAL WHEEL LOAD (%)			
	AXLE 1 LEFT	AXLE 1 RIGHT	AXLE 3 LEFT	AXLE 3 RIGHT
30	75	74	70	76
40	72	77	71	74
50	76	72	71	71
60	71	68	62	72

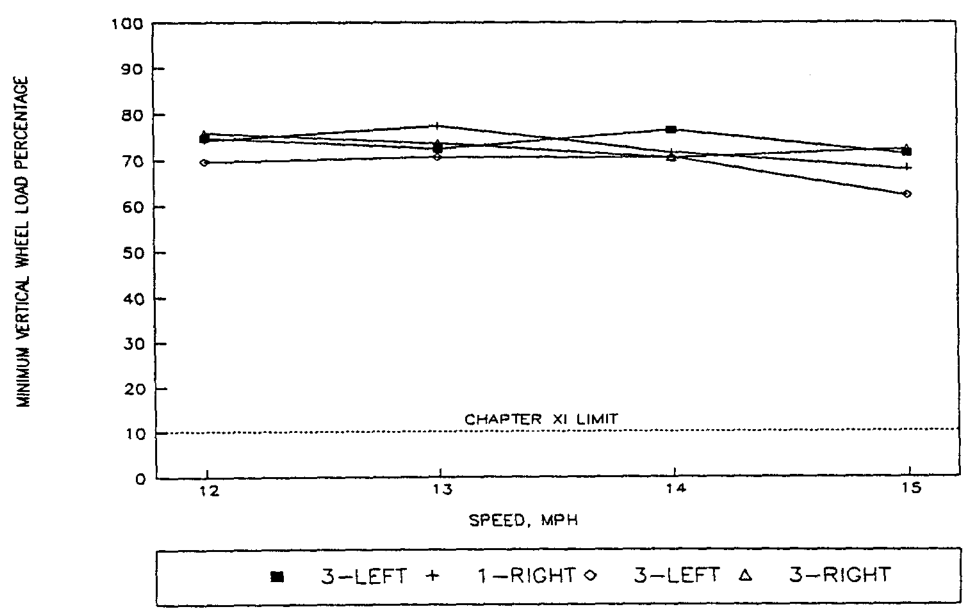


Figure 6.33 Pitch and Bounce Test Results

6.3.4 Twist and Roll Results

Chapter XI specified three limiting criteria for twist and roll. The first criterion was a 10 percent minimum wheel load. The second criterion was a maximum axle sum L/V of 1.3, and the third criterion was a maximum car body roll angle of 6 degrees peak-to-peak. Table 6.17 is a summary of the test data for each of the three criteria.

Table 6.17 Twist and Roll Results Summary

SPEED (mph)	MIN VERT WHEEL LOAD (%)	A-END ROLL ANGLE (Degrees) (p-p)	B-END ROLL ANGLE (Degrees) (p-p)	AXLE SUM L/V AXLE 1	AXLE SUM L/V AXLE 3
14	59	1.32	1.14	0.24	0.22
16	53	1.53	1.50	0.24	0.23
20	46	2.22	1.98	0.35	0.27
22	50	1.95	1.83	0.34	0.24
24	50	1.62	1.35	0.34	0.20
30	56	1.62	1.53	0.33	0.25
35	59	1.44	1.26	0.32	0.29
40	62	1.14	0.99	0.32	0.34
50	58	0.66	0.66	0.39	0.39
60	59	0.54	0.51	0.48	0.51

The minimum vertical wheel load was lowest at 20 mph (46 percent) and much higher than the 10 percent limit. This test speed equates to a frequency of 0.75 Hz with 39-foot wavelength perturbations, which agrees with the modal analysis of lower center roll at 0.75 Hz. Figure 6.34 shows minimum vertical wheel loads for twist and roll testing.

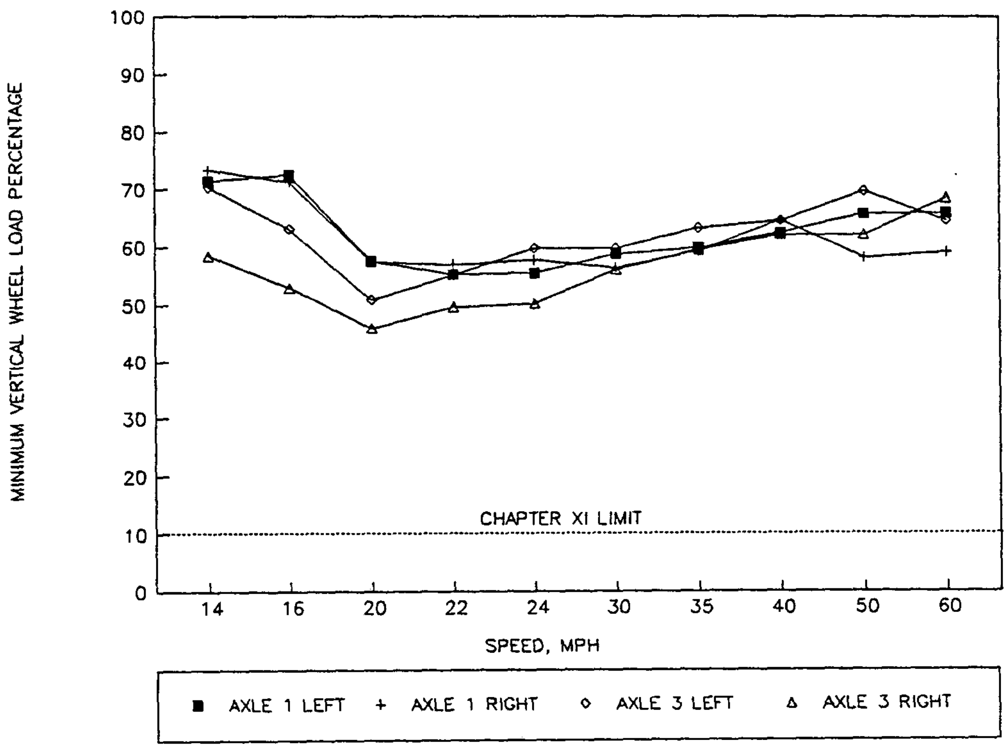


Figure 6.34 Twist and Roll Minimum Vertical Wheel Loads

The largest peak-to-peak roll angle was reached at 20 mph, which was 2.22 degrees peak-to-peak. This was well below the 6-degree limit. Figure 6.35 shows a comparison of actual roll angles versus Chapter XI Limit.

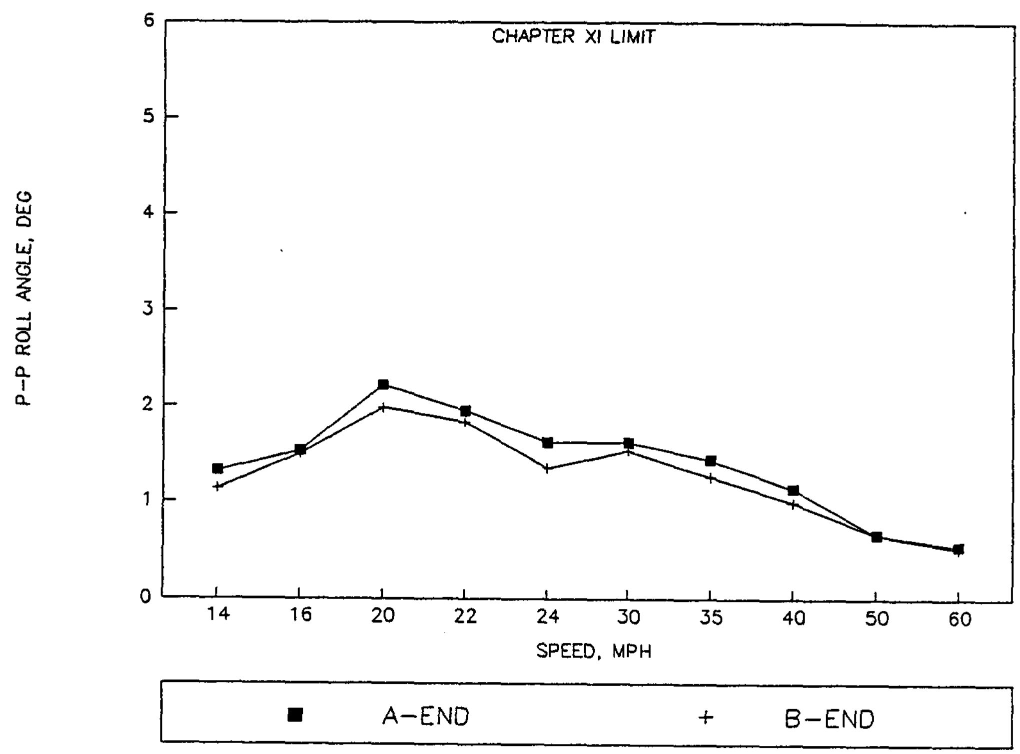


Figure 6.35 Roll Angle Results from Twist and Roll Test

The highest axle sum L/V was reached at 60 mph. It was 0.51 and well within the 1.3 guideline value. The L/V measured at 60 mph was not directly related to a vehicle roll mode. The roll resonance was 20 mph. The increase in axle sum L/V at higher speeds was due to the perturbations exciting the lateral vehicle instability. Figure 6.36 shows axle sum L/V's.

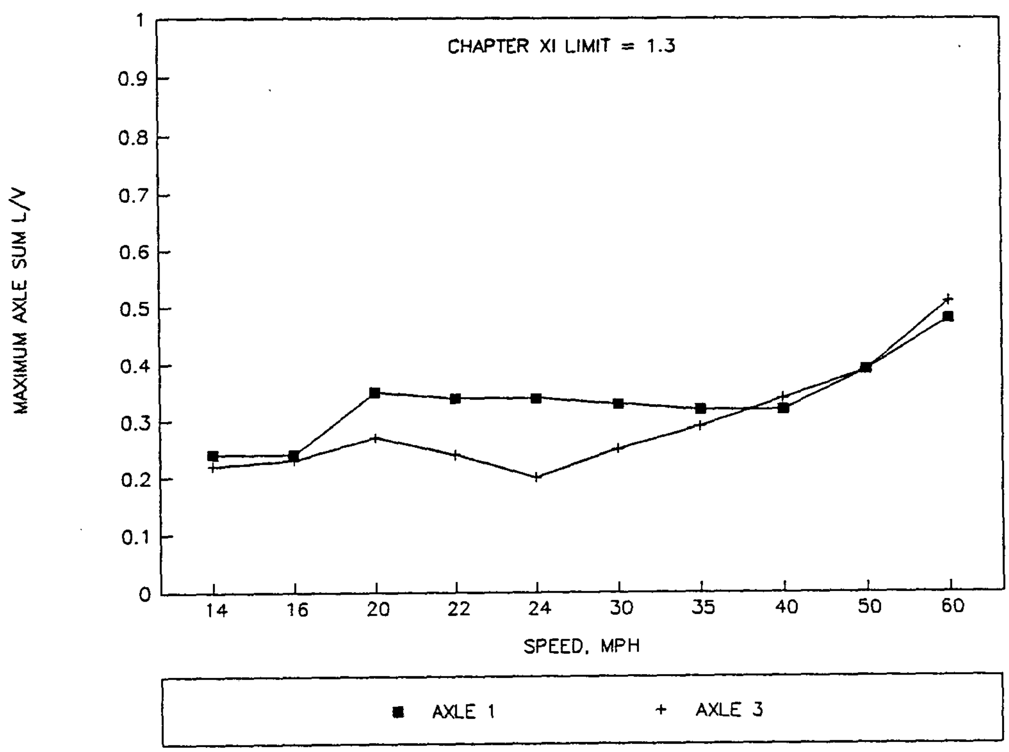


Figure 6.36 Twist and Roll Axle Sum L/V

6.3.5 Turnout and Crossover Results

Since turnout and crossover testing was not an official Chapter XI test, there were no official limiting criteria. The wheel L/V of 0.8 and axle sum L/V of 1.3 were used as guidelines. Table 6.18 summarizes these results.

Table 6.18 Turnout and Crossover Results

TEST	SPEED	MAXIMUM WHEEL L/V	MAXIMUM AXLE SUM L/V
Turnout	10	0.71	1.26
Turnout	15	0.70	1.25
Crossover	15	0.43	0.79
Crossover	25	0.48	0.84
Crossover	35	0.51	0.85

The turnout L/V results were nearer the limiting value than the crossover results. The wheel and axle L/V's, 0.71 and 1.26, were very near the limits of 0.8 and 1.3, respectively. The highest crossover L/V was 0.51 for one wheel at 35 mph.

6.3.6 Dynamic Curving Results

Dynamic curving imparts twist and roll excitation, lateral track misalignment input, and curving input. Chapter XI specified limiting values for the following four parameters: (1) maximum wheel L/V of 0.8, (2) maximum axle sum L/V of 1.3, (3) maximum roll angle of 6 degrees peak-to-peak, and (4) a minimum vertical wheel load of 10 percent. The test was performed in the clockwise and counterclockwise directions. Dynamic curving performance, especially in counterclockwise operation, did not meet the Chapter XI guidelines. Tables 6.19 and 6.20 summarize the dynamic curving results for both directions.

Table 6.19 Clockwise Dynamic Curving Results

SPEED (mph)	MIN VERT WHEEL LOAD (%)	ROLL ANGLE (deg) (p-p)	AXLE SUM L/V	TIME AXLE SUM OVER 1.3 (msec)	WHEEL L/V	TIME WHEEL L/V OVER 0.8 (msec)
10	51	1.20	0.70	--	0.48	--
12	48	1.29	0.65	--	0.44	--
14	48	1.37	0.69	--	0.47	--
16	45	1.32	0.78	--	0.55	--
18	33	1.84	1.10	--	0.71	--
20	33	1.98	1.19	--	0.73	--
22	36	1.86	1.25	--	0.79	--
24	32	1.89	1.27	--	0.81	12
26	26	2.10	1.03	--	0.70	--
28	21	1.57	1.19	--	0.78	--
30	31	1.66	1.33	17	0.88	59
32	26	1.83	1.22	--	0.81	10

Table 6.20 Counterclockwise Dynamic Curving Results

SPEED (mph)	MIN VERT WHEEL LOAD (%)	ROLL ANGLE (deg) (p-p)	AXLE SUM L/V	TIME AXLE SUM OVER 1.3 (msec)	WHEEL L/V	TIME WHEEL L/V OVER 0.8 (msec)
10	57	0.68	1.22	--	0.76	--
12	62	0.72	1.23	--	0.76	--
14	61	0.89	1.24	--	0.77	--
16	63	1.18	1.31	10	0.83	28
18	56	1.80	1.34	74	0.87	90
20	47	2.50	1.31	10	0.86	58
22	41	2.50	1.46	84	0.95	113
24	43	2.23	1.44	60	0.90	69
26	40	1.96	1.42	40	0.90	48
28	35	1.82	1.33	19	0.88	56
30	31	1.82	1.29	--	0.82	25
32	19	1.46	1.21	--	0.74	--

The counterclockwise direction was clearly the most severe, as shown in the comparison of maximum axle sum L/V for both directions in Figure 6.37.

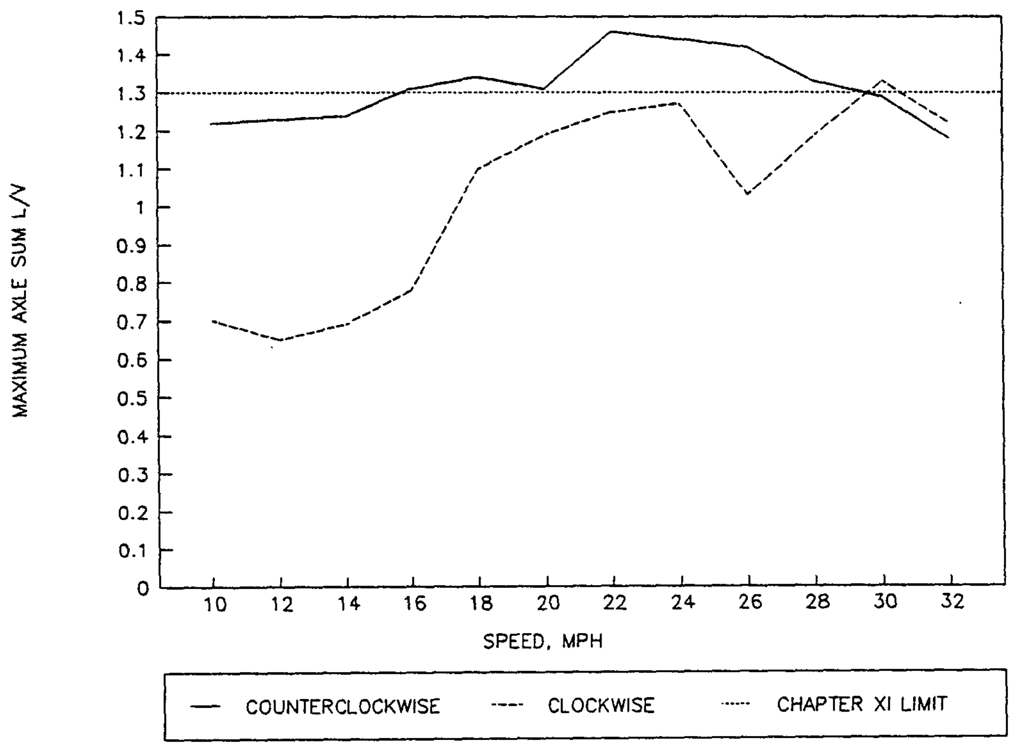


Figure 6.37 Axle Sum Dynamic Curving Results

The leading axle of the trailing truck had consistently higher L/V's than the leading axle of the leading truck. The worst case was at 22 mph, near the roll resonance speed, in the counterclockwise direction. Figure 6.38 is a wheel L/V time history of the 22 mph run for the period in which it exceeded 0.8.

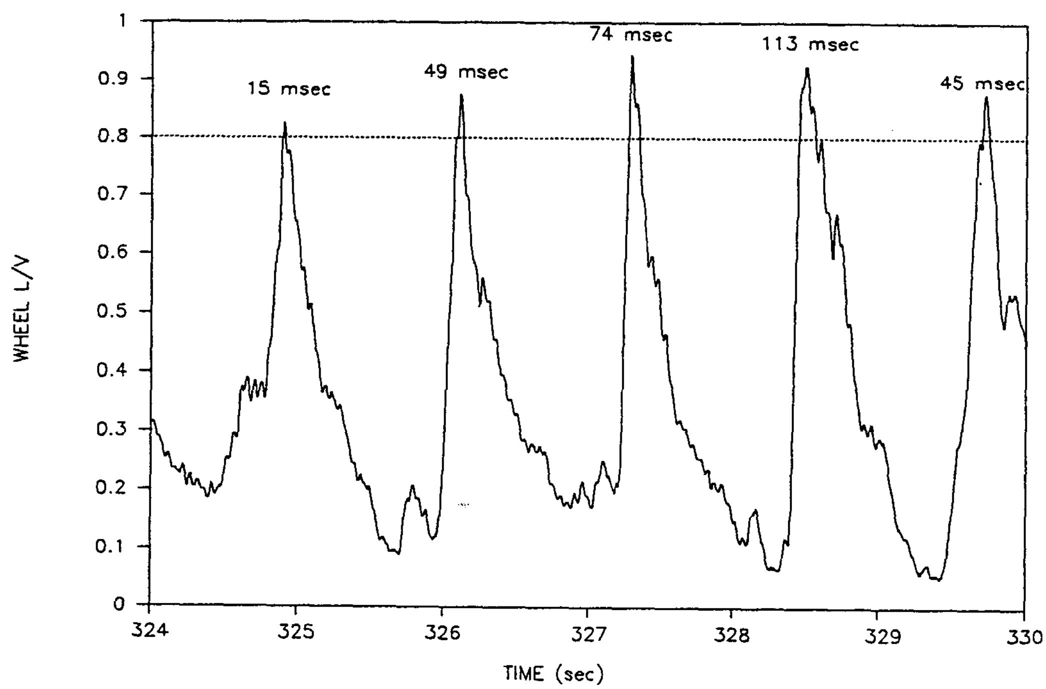


Figure 6.38 Wheel L/V Time History for Dynamic Curving

The fact that the L/V's were much higher in the counterclockwise direction may be due to two things. First, the car may tend to curve better in one direction due to some misalignment of the trucks. Second, the dynamic curving section is at one end of the 10-degree curve. The results may have been due to the fact that the car had just entered the curve in the counterclockwise direction whereas the car was in a steady state curving mode before entering the test zone in the clockwise direction.

6.3.7 Constant Curving Results

Tests were performed in the clockwise and counterclockwise directions on the 7.5-degree curve at speeds of 14, 24, and 32 mph and the 12-degree curve at speeds of 16, 25, and 32 mph. The test criteria were wheel and axle L/V's of 0.8 and 1.3 respectively. The 50 millisecond criterion did not apply to the Constant Curving Test per Chapter XI. In constant curving only, Chapter XI specifies a 95th percentile criterion. This criterion states that the value can exceed the limit for 5 percent of the total test time. Table 6.21 is a summary of results for the 7.5-degree test in both directions.

Table 6.21 Constant 7.5-Degree Curving Results

SPEED (mph)	WHEEL L/V CW		WHEEL L/V CCW		AXLE L/V CW		AXLE L/V CCW	
	PEAK	95%	PEAK	95%	PEAK	95%	PEAK	95%
14	.65	.46	.66	.53	1.10	.85	1.12	.94
24	.58	.40	.64	.51	.97	.74	1.05	.92
32	--	--	.58	.43	--	--	1.00	.83

The 32 mph run was not performed in the clockwise direction due to the car's performance in the bunched spiral at 25 mph. Those results are discussed in Section 5.8. Since the 32 mph run was not performed, it is difficult to determine whether the car curved better in one direction or the other at overbalance speeds. The 12 and 24 mph 95th percentile results would imply poorer but acceptable performance in the counterclockwise direction.

Table 6.22 is a summary of results for the 12-degree constant curve test.

Table 6.22 Steady State 12-Degree Curving Results

SPEED (mph)	DIR	WHEEL L/V				AXLE L/V			
		PEAK	DUR. (msec)	% OF TOTAL	95%	PEAK	DUR. (msec)	% OF TOTAL	95%
16	CW	.83	20	0.1	.70	1.39	149	0.7	1.22
25	CW	.72	--	--	.61	1.21	--	--	1.12
32	CW	--	--	--	--	--	--	--	--
16	CCW	.93	249	1.2	.73	1.42	340	1.6	1.23
25	CCW	.83	48	0.4	.65	1.34	201	1.5	1.15
32	CCW	.78	--	--	.59	1.32	31	0.3	1.09

The 16 mph counterclockwise run yielded high wheel and axle L/V's but didn't exceed the 95th percentile criteria. An example of the steady state analysis may be seen in the 16 mph counterclockwise data. The automatic locating device (ALD) triggered when the car entered and exited the curve. The length of the car was subtracted from the entry ALD to allow the car to be completely into the curve before steady state analysis. That block of data was added to the curve entry analysis with the spiral. For the 16 mph run, steady state curving ran from 689.196 to 711.041 seconds.

Figure 6.39 shows half of that time block, from 700 to 709 seconds for axle sum L/V on the lead axle.

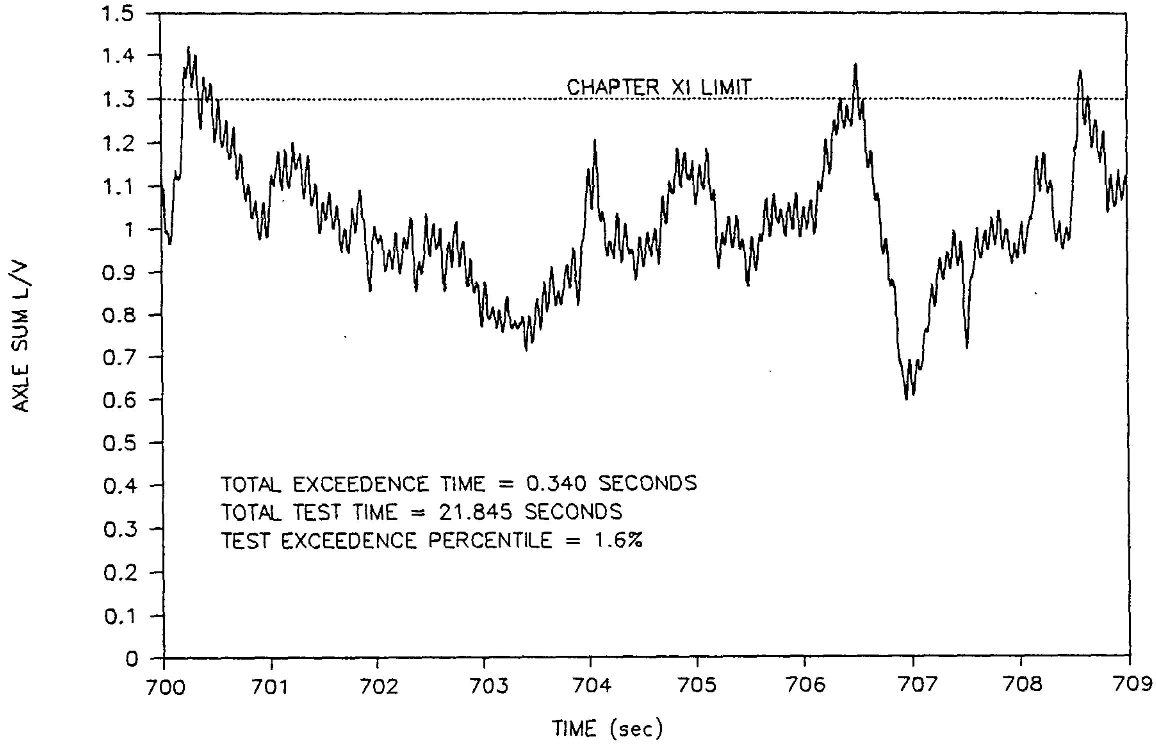


Figure 6.39 Axle Sum Time History for 16 mph in 12-Degree Curve

Since the limit of 1.3 was exceeded for a total of 0.340 seconds and the total test time was 21.845 seconds, the limit was exceeded for 1.6 percent of the time. The Chapter XI limit is 5 percent. The 95th percentile value for that axle was 1.23.

Figure 6.40 shows the 95th percentile wheel L/V's for both clockwise and counter-clockwise directions on the 12-degree curve.

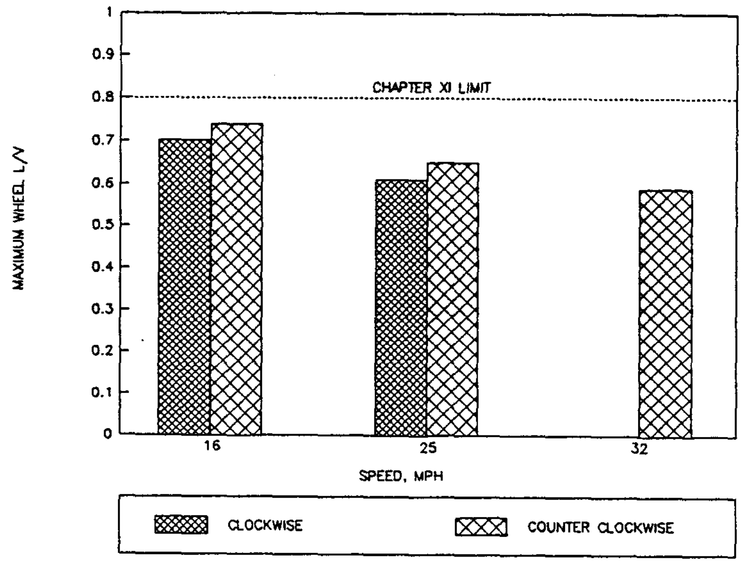


Figure 6.40 95th Percentile Wheel L/V's for 12-Degree Curve Steady State

The steady state curve performance was slightly more severe in the counterclockwise direction at balance and underbalance speeds. The curving performance was also speed dependent. The L/V's decreased as the speed increased. The fact that the wheel L/V trends were very similar to the axle sum L/V trends indicates a good test for a three-piece truck performance. On dry curved track, the L/V for the wheel that is not flanging should not exceed the static coefficient of friction of that interface barring any large longitudinal wheel loads. The data yielded an average coefficient of friction of 0.5 indicating dry track. The vehicle performance was within Chapter XI criteria.

It should be noted, however, that Chapter XI criteria for the other track tests only allow exceedence for less than 50 milliseconds. It is an inconsistency that the constant curving test limit is a 95 percent value which can allow exceedences greater than 50 milli-seconds as shown here.

6.3.8 Spiral Negotiation Results

Curve entry and exit performance was measured during the constant curving tests. Analysis was performed from the time the car entered the spiral to the time at which the car was completely in the curve. The 7.5-degree curve had conventional spirals at each end. A spiral is the section of track which makes the transition from tangent to curve with constant changes in curvature and superelevation at the same time. The 12-degree curve had a bunched spiral at one end. The bunched spiral was curve-exit for clockwise runs and curve-entry for counterclockwise runs. Chapter XI only specifies the bunched spiral for this test. The Chapter XI bunched spiral makes the usual change in curvature but has concentrated change in superelevation in the middle of the spiral. The limiting criteria for spiral negotiation were 10 percent minimum vertical wheel load and a maximum wheel L/V of 0.8 sustained for 50 milliseconds. Axle sum L/V was also provided for analysis. Table 6.23 shows a summary of the 7.5-degree curve nominal spiral entry and exit results. Curve exit was more severe than entry, but both were well within Chapter XI.

Table 6.23 7.5-Degree Curve Entry and Exit Results

SPEED (mph)	CURVE ENTRY			CURVE EXIT		
	MAX WHEEL L/V	MAX AXLE SUM L/V	MIN VERT WHEEL LOAD (%)	MAX WHEEL L/V	MAX AXLE SUM L/V	MIN VERT WHEEL LOAD (%)
14	.46	.78	49	.67	1.06	51
24	.38	.73	54	.57	.91	50
32	.39	.76	60	.42	.71	47

Table 6.24 is a summary of the 12-degree curve entry and exit results. Results are shown for both operational directions. Bunched spiral and conventional spiral results are so noted in the table footnote.

Table 6.24 12-Degree Spiral Negotiation Summary

DIR	SPEED (mph)	CURVE ENTRY					CURVE EXIT				
		MAX WHEEL L/V	TIME OVER 0.8 (msec)	MAX AXLE L/V	TIME OVER 1.3 (msec)	MIN WHEEL LOAD %	MAX WHEEL L/V	TIME OVER 0.8 (msec)	MAX AXLE L/V	TIME OVER 1.3 (msec)	MIN WHEEL LOAD %
CW	16	.78	--	1.31	5	50	.85	45	1.42	118	26
CW	25	.85	46	1.32	10	43	.92	12	1.32	8	<1
CW	32	--	--	--	--	--	--	--	--	--	--
CCW	16	.81	11	1.32	18	23	.82	19	1.32	5	48
CCW	25	.69	--	1.18	--	39	.77	--	1.29	--	51
CCW	32	.64	--	1.15	--	26	.99	19	1.63	24	14

Footnote:

1. **Curve entry clockwise (CW) - conventional spiral**
2. **Curve entry counterclockwise (CCW) - bunched spiral**
3. **Curve exit clockwise - bunched spiral**
4. **Curve exit counterclockwise - conventional spiral**

Chapter XI limits for wheel and axle sum L/V were exceeded for most conditions but were sustained for less than 50 milliseconds with one exception. At 16 mph in the bunched spiral curve exit the axle sum criteria of 1.3 was exceeded for 118 milliseconds. The minimum vertical wheel load criteria was exceeded once. At 25 mph in the bunched spiral exit, the vertical wheel load was less than 10 percent for 107 and 96 milliseconds. Figure 6.41 is a time history of the left wheel on the third axle (trailing inside wheel) during that event.

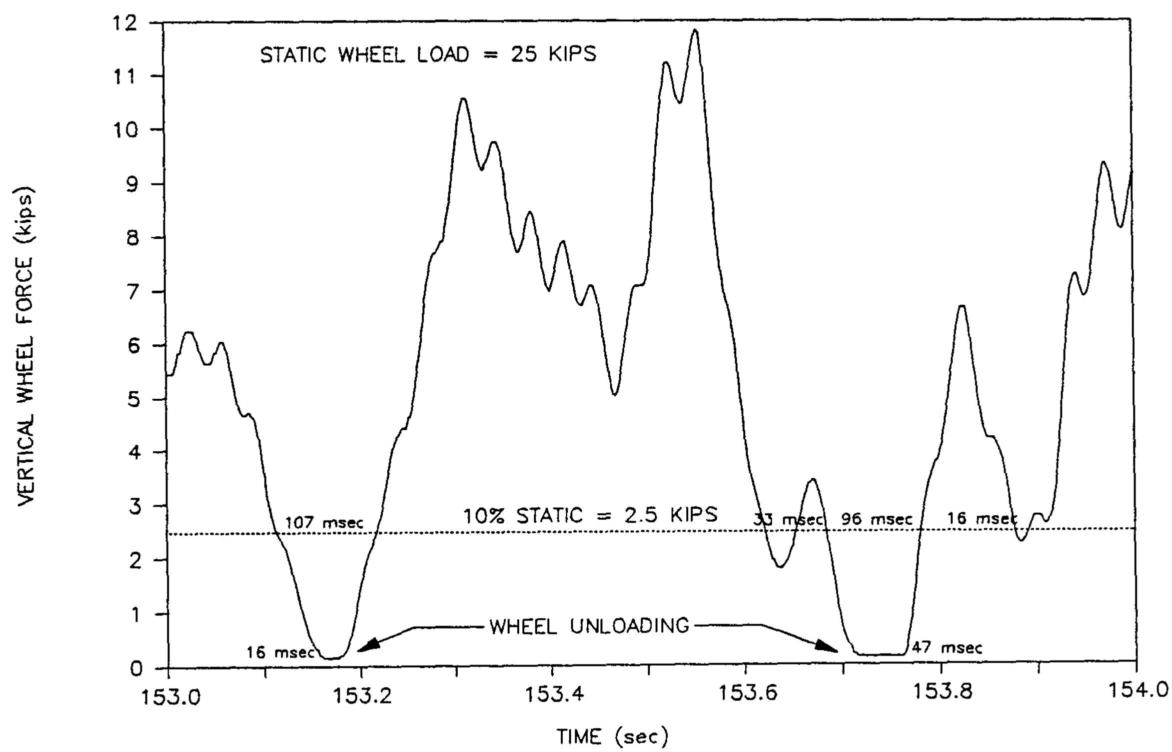


Figure 6.41 Trailing Inside Wheel Load Time History for Bunched Spiral Curve Exit

The 10 percent wheel load criteria was exceeded four times. The two most severe cases were for 107 and 96 milliseconds. In both cases the wheel force seemed to bottom out at 150 pounds. Most likely there was a 150 pound electronic offset in the wheel set system. The actual wheel force was probably zero. This zero force or wheel lift was sustained for 47 milliseconds in one case. Due to such severe results, the 32 mph test run was suspended.

Figure 6.42 is a comparison of maximum wheel L/V's from all 12-degree curve entry and exit conditions.

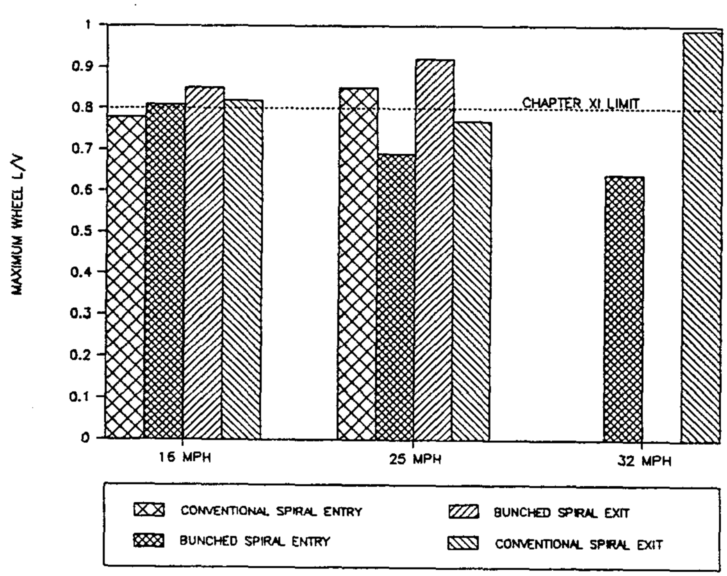


Figure 6.42 12-Degree Spiral Negotiation Results for Both Spirals

The high L/V's and low wheel loads during spiral negotiation would indicate that the car is not twisting sufficiently. The car is forced to twist in a spiral because the leading truck is at a different superelevation than the trailing truck. If the car is unable to twist and conform, a wheel will lift.

6.4 STATIC BRAKE TEST

The Static Brake Test was performed on the MC, which consisted of a single car test and a net shoe force test. The tests were performed with the assistance of Blaine Consulting Services.

Instrumented brake shoes were installed in place of each set of brakes on each truck. Data was obtained with the brake rigging tapped and untapped. Since the tapped readings are closer to the dynamics of a car rolling over the railroad, these values were used for the following analysis. Figure 6.43 shows the sum of the eight shoe forces on the car for each test.

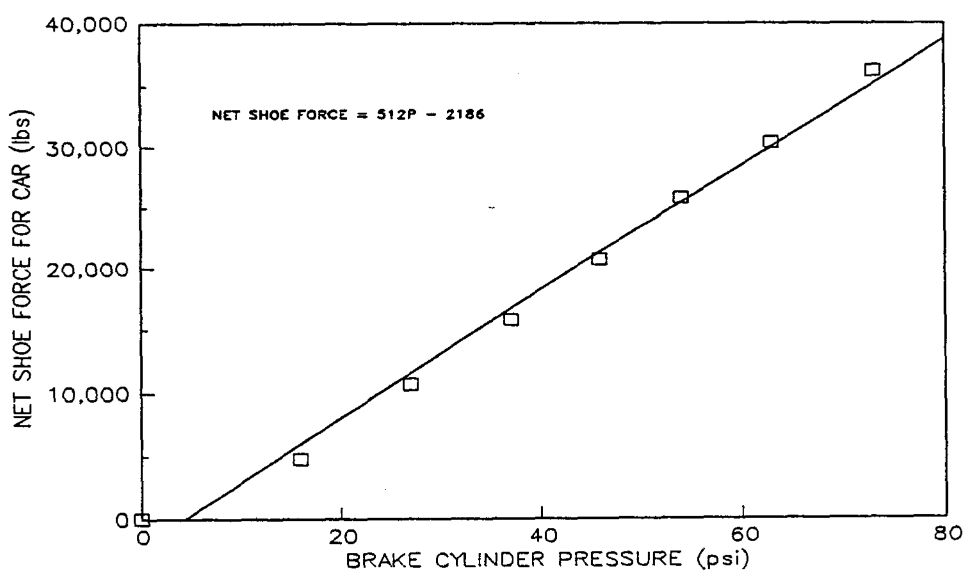


Figure 6.43 Static Brake Test Results

The linear regression yielded the following equation for total car brake force.

$$TotalCar\,BrakeForce = 512 * BrakeCylinder\,Pressure - 2186$$

Braking performance is based on brake shoe force and car weight. The net braking ratio, which is car brake force divided by car weight, is the parameter regulated by the AAR. The net braking ratio must be between 6.5 and 10 percent at a brake cylinder pressure of 50 pounds per square inch (psi) according to AAR Standard S-486. The net braking ratio for the MC was calculated with the following equation.

$$Net Braking Ratio = Total Car Brake Force / Car Weight$$

The specification recommends a loaded braking ratio of 6.5 percent to 10 percent at maximum gross rail load; 263,000 pounds for this car. That would equate to 8.9 percent, which is within specification. Brake cylinder pressure is dependent on the train line pressure and the amount of pressure bled off (reduction). Since the operational train line pressure could be between 70 and 110 psi and the operational weight of the MC will be 205,000 pounds, Table 6.25 was developed to show brake ratios for various brake cylinder pressures using 205,000 pounds for the loaded car weight rather than gross rail load.

Table 6.25 Loaded MC Net Braking Ratio Summary

EXPLANATION	BRAKE CYLINDER PRESSURE (psi)	NET BRAKING RATIO (%)
Full Service Reduction at 70 psi Train Line	50	11.4
Full Service Reduction at 90 psi Train Line	64	14.9
Full Service Reduction at 110 psi Train Line	78	18.4
Emergency at 70 psi Train Line	60	13.9
Emergency at 90 psi Train Line	77	18.1
Emergency at 110 psi Train Line	93	22.1

The Hand Brake Net Shoe Force Test was also performed. Hand brake chain forces from 2,900 pounds to 6,150 pounds were applied in 1,000 pound increments. Operationally, a 4,250-pound force could be obtained in the horizontal chain after the second sheave with a 125-pound application at the hand brake wheel. Figure 6.44 shows plotted points for both trucks and the linear regression and associated equation.

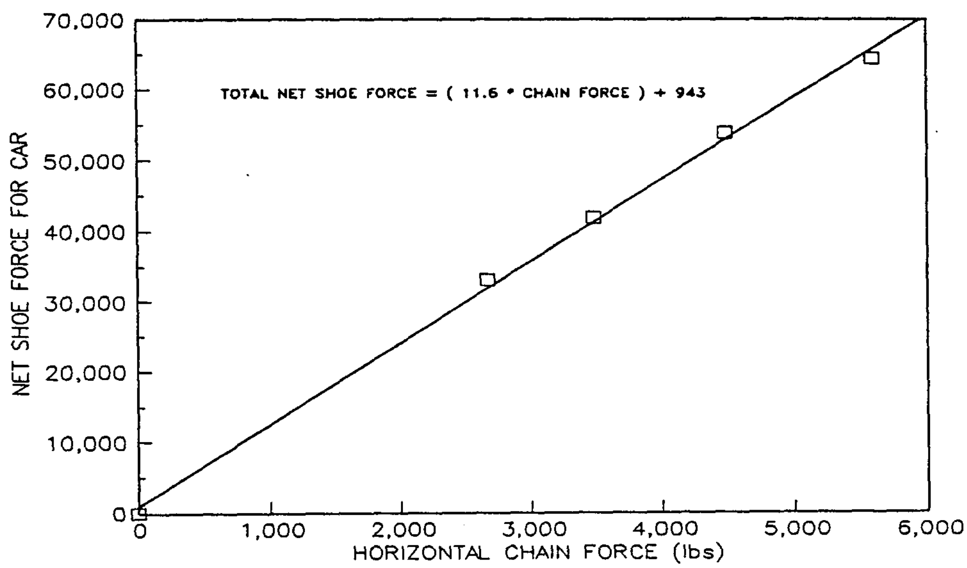


Figure 6.44 Hand Brake Test Results

$$TotalHandbrakeForce = 11.6 * HorizontalChainForce + 943$$

A horizontal chain force of 4,250 pounds would have yielded a total hand brake force of 50,243 pounds and a net braking ratio of 24.5 percent. This value is much higher than the AAR minimum of 11 percent and meets the standard.

7.0 CONCLUSIONS

1. Chapter XI states that values better than the criteria outlined in this report are regarded as indicating the likelihood of safe car performance. With the exception of dynamic curving and spiral negotiation, the loaded MC performed within the Chapter XI criteria. The main reason for acceptable MC performance was the truck spacing. Twist and roll, yaw and sway, and pitch and bounce contain track perturbations of a 39-foot wavelength. It would be likely that a car with 39-foot truck spacing would be most sensitive to such perturbations or multiples of that wavelength. The truck spacing on the MC was 64 feet. A wavelength of 39-feet is the most typical of excitation expected from the track. Perturbations of other wavelengths are possible but less likely. Multiples of 64 feet will provide more input to this car than the Chapter XI, 39-foot wavelengths.

2. The MC barely stayed within Chapter XI guidelines in yaw and sway. However, the actual lateral perturbations over which the car was tested were 0.25 inches lower in amplitude than those specified in Chapter XI and used in the model. The car may exceed Chapter XI specifications if the perturbation amplitudes are 1.25 inches.

3. The MC exhibited no lateral instability at speeds up to 70 mph. This was due, primarily, to the use of constant contact side bearings, to the fact that the trucks were new and had tighter tolerances than worn trucks, and to the fact that the car was loaded rather than in the Chapter XI specified unloaded configuration.

4. The MC negotiated the turnout and crossover within the standard Chapter XI limits for wheel and axle L/V. This was a very important test, because switching is a frequent operation in a railroad environment.

5. The MC exceeded Chapter XI criteria in dynamic curving primarily in the counterclockwise direction. Exceedences in excess of 50 milliseconds were measured at speeds from 18 mph through 28 mph. In constant curving tests the car performed better in the clockwise direction at speeds under 25 mph. This may have played a role in the dynamic curving directional dependency. The car also had some difficulty in curve entry. This may have played a role in the high L/V's in the counterclockwise direction as well. Soon after the car entered the 10-degree curve in that direction it encountered the dynamic curving perturbations. Even though the values were in excess of Chapter XI criteria, they were still low enough to allow the completion of the test at all speeds.

6. The car negotiated the 7.5- and 12-degree curves within Chapter XI limits with the exception of the 12-degree clockwise run at 32 mph, which was not performed due to poor spiral negotiation performance at 25 mph.

7. The most severe test for the MC was the bunched spiral. At 25 mph the car experienced wheel lift. This was likely due to the inability of the car to twist.

8. The unloaded MC successfully completed the service worthiness curve stability test by exhibiting no suspension separation or wheel lift.

9. Braking ratios for the MC were within AAR specification.

8.0 RECOMMENDATIONS

1. Post test modeling should be performed to develop a fine tuned computer simulation using measured car performance. This will validate the MC model for use in train modeling and future modeling for possible design changes or extension of results to regimes not tested.

2. Post test modeling should be performed to examine car performance in yaw and sway. The yaw and sway test should be modeled with the actual amplitudes in perturbations. If the model predictions match the test results, then predictions should be made with the Chapter XI specified perturbation amplitudes. This may indicate a possible need for design changes.

3. Post test modeling should also include possible design changes for an improvement in spiral negotiation performance. If a solution is found, the OM car could be retrofitted to allow a re-test.

4. If significant suspension changes are made, the car should be subjected to a limited re-test to include dynamic and constant curving, spiral negotiation, and hunting.

5. Some subtle changes in the design and operation of the brake system should be made. The operational scenario for the PKRG train is more similar to a passenger than freight train. A train line pressure of 110 psi should be considered to increase the overall braking ratio for the train, improving stop distance and grade handling.

APPENDIX A

MC EM-1 TRACK WORTHINESS
TEST CONFIGURATION DATA SHEETS

PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

TEST NAME M.C. TRACK WORTHINESS DATE 10-5-90 W.O. 87559 LOC. ______

INSTR. ENGR./TECH. ______ TEST ENGR. BRENT WHITSITT QA ______

SOFTWARE/VERSION ______ RECORDER I.D. NO. ______ SET-UP FILE ______

SAMPLE RATE ______ ENCODER/DIGITIZER I.D. NO. ______

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER					AMPLIFIER						FILTER				SYSTEM			RECORDER		COMMENTS
				MFG.	S.N.	SENS.	LOC.	CAL VOID DATE	CH. NO.	EXC.-L/R	GAIN FIX/VAR	R-CAL RES.	CAL. E.U. & VOLTS	S.N. CAL VOID DATE	NO.	FREQ	GAIN	CAL VOID DATE	AO (E.U.)	AI (E.U./VOLT)	ENGR. UNITS	CH. NO.	SENS. (E.U./DIV.)	
ALD	0	0	XBX001A	WARNER		20MS PULSE	X= Y= Z=												0	1 EVENT/ 10 V	EVENT			ALD
TSPD	1	1	RDW001A	AIRPAX		64 PULSES/ REV	X= Y= Z=												0	10 MPH/ VOLT	MPH	8		SPEED
FVIL	2	2	LBW001A	IITRI	21A	CALC	X= 964 Y= 28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLT	KIPS	1		IWS TRK-1 LEAD AXLE, VERT LEFT
FVIR	3	3	LBW002A	IITRI	21B	CALC	X= 964 Y= -28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLT	KIPS	2		IWS TRK-1 LEAD AXLE, VERT RIGHT
FLIL	4	4	LBW003A	IITRI	21A	CALC	X= 964 Y= 28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLT	KIPS	3		IWS TRK-1 LEAD AXLE, LAT LEFT
FLIR	5	5	LBW004A	IITRI	21B	CALC	X= 964 Y= -28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLT	KIPS	4		IWS TRK-1 LEAD AXLE, LAT RIGHT
LVIL	6	6	LBW005A	IITRI	21A	CALC	X= 964 Y= 28.5 Z= 18									15 HZ	1		0	.5 L/V / VOLT	L/V	5		IWS TRK-1 LEAD AXLE, L/V LEFT
LVIR	7	7	LBW006A	IITRI	21B	CALC	X= 964 Y= -28.5 Z= 18									15 HZ	1		0	.5 L/V / VOLT	L/V	6		IWS TRK-1 LEAD AXLE, L/V RIGHT
FT1	8	8	LBW007A	IITRI	21	CALC	X= 964 Y= 0 Z= 18									15 HZ	1		0	5.0 K/ VOLT	KIPS	7		IWS TRK-1 LEAD AXLE, TORQUE
FV3L	9	9	LBW008A	IITRI	22A	CALC	X= 1732 Y= 28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLT	KIPS	9		IWS TRK-2 LEAD AXLE, VERT LEFT

NOTES:

ACAD FILE: MCTW01.DWG

PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

TEST NAME M.C. TRACK WORTHINESS DATE ________ W.O. 87559 LOC. ________

INSTR. ENGR./TECH. ________ TEST ENGR. BRENT WHITSITT QA ________

SOFTWARE/VERSION ________ RECORDER I.D. NO. ________ SET-UP FILE ________

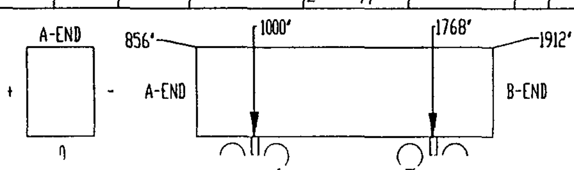
SAMPLE RATE ________ ENCODER/DIGITIZER I.D. NO. ________

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER MFG.	S.N.	SENS.	LOC.	CAL VOID DATE	AMPLIFIER CH. NO.	EXC.-L/R	GAIN FIX/VAR	R-CAL RES.	CAL. E.U. & VOLTS	S.N. CAL VOID DATE	FILTER NO.	FREQ.	GAIN	CAL VOID DATE	SYSTEM A0 (E.U.)	A1 (E.U./VOLT)	ENGR. UNITS	RECORDER CH. NO.	SENS. (E.U./DIV.)	COMMENTS
FV3R	10	10	LBW009A	IITRI	22B	CALC	X= 1732 Y= -28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLTS	KIPS	10		IWS TRK-2 LEAD AXLE, VERT RIGHT
FL3L	11	11	LBW010A	IITRI	22A	CALC	X= 1732 Y= 28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLTS	KIPS	11		IWS TRK-2 LEAD AXLE, LAT LEFT
FL3R	12	12	LBW011A	IITRI	22B	CALC	X= 1732 Y= -28.5 Z= 18									15 HZ	1		0	10.246 K/ VOLTS	KIPS	12		IWS TRK-2 LEAD AXLE, LAT RIGHT
LV3L	13	13	LBW012A	IITRI	22A	CALC	X= 1732 Y= 28.5 Z= 18									15 HZ	1		0	.5 L/V / VOLTS	L/V	13		IWS TRK-2 LEAD AXLE, L/V LEFT
LV3R	14	14	LBW013A	IITRI	22B	CALC	X= 1732 Y= -28.5 Z= 18									15 HZ	1		0	.5 L/V / VOLTS	L/V	14		IWS TRK-2 LEAD AXLE, L/V RIGHT
FT3	15	15	LBW014A	IITRI	22	CALC	X= 1732 Y= 0 Z= 18									15 HZ	1		0	5.0 K/ VOLTS	KIPS	15		IWS TRK-2 LEAD AXLE, TORQUE
AY1A	16	16	Y001A	ENDEVLO 7290-10	AE 74	198.4 MV/G	X= 856 Y= 0 Z= 50		K	15 V	2 FIX	V SUB	3.35			15 HZ	1		0	2.5458 G/V	G	17		ACCEL LATERAL A-END CAR BODY
AY2A	17	17	Y002A	ENDEVLO 7290-10	AC 58	197.2 MV/G	X= 1912 Y= 0 Z= 50		K	15 V	2 FIX	V SUB	3.35			15 HZ	1		0	2.4765 G/V	G	18		ACCEL LATERAL B-END CAR BODY
AFF 250A AZ1A	18	18	Z001A	ENDEVLO 7265HS	BM 54	22.61 MV/G	X= 1384 Y= 0 Z= 44		H	10 V	100 FIX	29.8 K	5.12			15 HZ	1		0	.442 G/V	G	19		ACCEL VERTICAL CNTR OF CAR, CNTR OF CAR FLOOR
AFF 251A AY3A	19	19	Y003A	ENDEVLO 7265HS	BD 08	31.22 MV/G	X= 1384 Y= 0 Z= 44		H	10 V	100 FIX	29.8 K	4.81			15 HZ	1		0	.320 G/V	G	20		ACCEL LATERAL CNTR OF CAR, CNTR OF CAR FLOOR

NOTES:

ACAD FILE: MCTV02.DWG

CAR PULLED BY 'A-END'



PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

PAGE 3 OF ____

TEST NAME MAINTENANCE CAR (TW) DATE ________ W.O. 87559 LOC. ____________

INSTR. ENGR./TECH. ____________ TEST ENGR. BRENT WHITSITT QA ____________

SOFTWARE/VERSION ____________ RECORDER I.D. NO. ____________ SET-UP FILE

SAMPLE RATE ________ ENCODER/DIGITIZER I.D. NO. ____________ ____________

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER					AMPLIFIER						FILTER				SYSTEM			RECORDER		COMMENTS
				MFG.	S.N.	SENS.	LOC.	CAL VOID DATE	CH. NO.	EXC. - L/R	GAIN FIX/VAR	R-CAL RES.	CAL. E.U. & VOLTS	S.N. CAL VOID DATE	NO.	FREQ	GAIN	CAL VOID DATE	A0 (E.U.)	A1 (E.U./VOLT)	ENGR. UNITS	CH. NO.	SENS. (E.U./DIV)	
AFF 252A AZ2A	20	20	Z002A	ENDEVCO 7265-HS	BT 14	23.90 MV/G	X= 1000 Y= 0 Z= 44		H	10 V	100 FIX	29.88 K	4.80		1	15 Hz			0	.4184 G/V	G	21		ACCEL VERT A-END ABOVE TRK-1 CTR PLATE
AFF 253A AY4A	21	21	Y004A	ENDEVCO 7265-HS	BR 08	23.60 MV/G	X= 1000 Y= 0 Z= 44		H	10 V	100 FIX	29.88 K	4.79		1	15 Hz			0	.4237 G/V	G	22		ACCEL LAT A-END ABOVE TRK-1 CTR PLATE
AFF 254A AX1A	22	22	X001A	ENDEVCO 7265-HS	BP 51	25.92 MV/G	X= 1000 Y= 0 Z= 44		H	10 V	100 FIX	29.88 K	4.78		1	15 Hz			0	.3858 G/V	G	23		ACCEL LONG A-END ABOVE TRK-1 CTR PLATE
AFF 255A AZ3A	23	23	Z003A	ENDEVCO 7265-HS	BP 64	27.82 MV/G	X= 1768 Y= 0 Z= 44		H	10 V	100 FIX	29.88 K	5.00		1	15 Hz			0	.3594 G/V	G	24		ACCEL VERT B-END ABOVE TRK-2 CTR PLATE
AFF 256A AY5A	24	24	Y005A	ENDEVCO 7265-HS	BM 52	21.63 MV/G	X= 1768 Y= 0 Z= 44		H	10 V	100 FIX	29.88 K	4.965		1	15 Hz			0	.4623 G/V	G	25		ACCEL LAT ABOVE TRK-2 CTR PLATE
AFF261A AZ4A	25	25	Z004A	ENDEVCO 7265-HS	BM 62	26.21 MV/G	X= 1905 Y= 0 Z= 168		H	10 V	100 FIX	29.88 K	4.84		1	15 Hz			0	.3815 G/V	G	26		ACCEL VERT CTR OF CAR A-END WALL TOP
AFF 262A AY6A	26	26	Y006A	ENDEVCO 7265-HS	BP 23	25.69 MV/G	X= 1905 Y= 0 Z= 168		H	10 V	100 FIX	29.88 K	4.77		1	15 Hz			0	.3892 G/V	G	27		ACCEL LAT CTR OF CAR A-END WALL TOP
AFF 263A AX2A	27	27	X002A	ENDEVCO 7265-HS	BE 46	23.47 MV/G	X= 1905 Y= 0 Z= 168		H	10 V	100 FIX	29.88 K	4.89		1	15 Hz			0	.4260 G/V	G	28		ACCEL LONG CTR OF A-END WALL TOP
AFW 264A AY7A	28	28	Y007A	ENDEVCO 7265-HS	BP 81	26.31 MV/G	X= 1768 Y= -41 Z= 24		H	10 V	100 FIX	29.88 K	4.97		1	15 Hz			0	.3800 G/V	G	29		ACCEL LAT B-END TRK-2 LF SIDE FRAME ABOVE BLST CTR
AFW 265A AY8A	29	29	Y008A	ENDEVCO 7265-HS	BG 66	25.74 MV/G	X= 100 Y= -41 Z= 24		H	10 V	100 FIX	29.88 K	5.00		1	15 Hz			0	.3885 G/V	G	30		ACCEL LAT A-END TRK-1 LF SIDE FRAME ABOVE BLST CTR

NOTES:

ACAD FILE: MCTV03.DWG

PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

PAGE 4 OF ____

TEST NAME MAINTENANCE CAR (TW) DATE ________ W.O. 87559 LOC. ________

INSTR. ENGR./TECH. ________ TEST ENGR. BRENT WHITSETT QA ________

SOFTWARE/VERSION ________ RECORDER I.D. NO. ________ SET-UP FILE ________

SAMPLE RATE ________ ENCODER/DIGITIZER I.D. NO. ________

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER MFG.	S.N.	SENS.	LOC.	CAL VOID DATE	AMPLIFIER CH. NO.	EXC.-L/R	GAIN FIX/VAR	R-CAL RES.	CAL. E.U. & VOLTS	S.N. CAL VOID DATE	FILTER NO.	FREQ	GAIN	CAL VOID DATE	SYSTEM A0 (E.U.)	A1 (E.U./VOLT)	ENGR. UNITS	RECORDER CH. NO.	SENS. (E.U./DIV.)	COMMENTS
DFF 001A JBX1A	30	30	JBX 001A	HUMPHREY	105	4.052 DEG/V	X= 856 Y= 0 Z= 44		K	10 V	1 FIX	VSUB				15 Hz	1		0	1.008 DEG/V	DEG	31		C/L A-END CARBODY ROLL ANGLE 8/8
DFF 002A JBX2A	31	31	JBX 002A	HUMPHREY	102	4.087 DEG/V	X= 1912 Y= 0 Z= 44		K	10 V	1 FIX	VSUB				15 Hz	1		0	1.018 DEG/V	DEG	32		C/L B-END CARBODY ROLL ANGLE 8/8
AFW 266A AZ5A	32	32	Z005A	ENDEVCO 7265-HS	BP 39	23.94 MV/G	X= 965 Y= 41 Z= 24		H	10 V	100 FIX	29.88 K	5.404			15 Hz	1		0	.4177 G/V	G			ACCEL VERT TRK-1 LEAD AXLE RT WHEEL ADAPTER
AFW 269A AZ6A	33	33	Z006A	ENDEVCO 7265-HS	BN 58	24.62 MV/G	X= 965 Y= - 41 Z= 24		H	10 V	100 FIX	29.88 K	4.799			15 Hz	1		0	.4061 G/V	G			ACCEL VERT TRK-1 LEAD AXLE LF WHEEL ADAPTER
AFW 270A AY9A	34	34	Y009A	ENDEVCO 7265-HS	BR 09	25.83 MV/G	X= 965 Y= - 41 Z= 24		H	10 V	100 FIX	29.88 K	4675			15 Hz	1		0	.3871 G/V	G			ACCEL LAT TRK-1 LEAD AXLE LF WHEEL ADAPTER
AFW 271A AY10A	35	35	Y0A0A	ENDEVCO 7265-HS	BP 37	24.55 MV/G	X= 965 Y= - 41 Z= 24		H	10 V	100 FIX	29.88 K	4.14			15 Hz	1		0	.4073 G/V	G			ACCEL LAT TRK-2 LEAD AXLE LF WHEEL ADAPTER
DFW 100A D27A	36	36	Z007A	CELESCO	A 45607	622.6 MV/IN	X= 1000 Y= -41 Z= 20		K	10 V	2 FIX	VSUB				15 Hz	1		0	.803 IN/V	INCH			DISPLACEMENT TRK-1 LF SIDE FRAME TO TRK BLST
DFW 101A DZ8A	37	37	Z008A	CELESCO	A 45608	623.0 MV/IN	X= 100 Y= 41 Z= 20		K	10 V	2 FIX	VSUB				15 Hz	1		0	.8025 IN/V	INCH			DISPLACEMENT TRK-1 RT SIDE FRAME TO TRK BLST
DZ9A	38	38	Z009A	CELESCO	A 45609	622.4 MV/IN	X= 1768 Y= -41 Z= 20		K	10 V	2 FIX	VSUB				15 Hz	1		0	.8033 IN/V	INCH			DISPLACEMENT TRK-2 LF SIDE FRAME TO TRK BLST
DZ10A	39	39	Z010A	CELESCO	45610	622.4 MV/IN	X= 1768 Y= 41 Z= 20		K	10 V	2 FIX	VSUB				15 Hz	1		0	.8033 IN/V	INCH			DISPLACEMENT TRK-2 RT SIDE FRAME TO TRK BLST

NOTES:

ACAD FILE: MCTW04.DVG

PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

PAGE 5 OF ____

TEST NAME M.C. TRACK WORTHINESS DATE ____ W.O. 87559 LOC. ____

INSTR. ENGR./TECH. ____ TEST ENGR. B. WHITSETT QA ____

SOFTWARE/VERSION ____ RECORDER I.D. NO. ____ SET-UP FILE ____

SAMPLE RATE ____ ENCODER/DIGITIZER I.D. NO. ____

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER					AMPLIFIER						FILTER				SYSTEM			RECORDER		COMMENTS
				MFG.	S.N.	SENS.	LOC.	CAL VOID DATE	CH. NO.	EXC.-L/R	GAIN FIX/VAR	R-CAL RES.	CAL. E.U. & VOLTS	S.N. CAL VOID DATE	NO.	FREQ	GAIN	CAL VOID DATE	A0 (E.U.)	A1 (E.U./VOLT)	ENGR. UNITS	CH. NO.	SENS. (E.U./DIV.)	
	40	40	IRIG				X= Y= Z=																	
V21A	41	41	V21AA				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
V21B	42	42	V21AB				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
V21C	43	43	V21AC				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
L21A	44	44	L21AA				X= Y= Z=			15	200	118 K	1.535			200	1			1				
L21B	45	45	L21AB				X= Y= Z=			15	200	118 K	1.535			200	1			1				
VA21	46	46	V21BA				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
VB21	47	47	V21BB				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
VC21	48	48	V21BC				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
LA21	49	49	L21BA				X= Y= Z=			15	200	118 K	1.535			200	1			1				

NOTES:

ACAD FILE: MCTW05.DWG

PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

TEST NAME M.C. TRACK WORTHINESS DATE ________ W.O. 87559 LOC. ____________

INSTR. ENGR./TECH. ____________ TEST ENGR. B. WHITSETT QA ____________

SOFTWARE/VERSION ____________ RECORDER I.D. NO. ____________ SET-UP FILE ____________

SAMPLE RATE ________ ENCODER/DIGITIZER I.D. NO. ____________

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER MFG.	TRANSDUCER S.N.	TRANSDUCER SENS.	TRANSDUCER LOC.	TRANSDUCER CAL VOID DATE	AMPLIFIER CH NO.	AMPLIFIER EXC.-L/R	AMPLIFIER GAIN FIX/VAR	AMPLIFIER R-CAL RES.	AMPLIFIER CAL. E.U. & VOLTS	AMPLIFIER S.N. / CAL VOID DATE	FILTER NO.	FILTER FREQ	FILTER GAIN	FILTER CAL VOID DATE	SYSTEM A0 (E.U.)	SYSTEM A1 (E.U./VOLT)	SYSTEM ENGR. UNITS	RECORDER CH. NO	RECORDER SENS. (E.U./DIV)	COMMENTS
LB21	50	50	L21BB				X= Y= Z=			15	200	118 K	1.535			200	1			1				
JBX3	51	51	JBX3A				X= Y= Z=																	ROLL RATE B-END RAV GYRO
P21A	52	52	P21AA				X= Y= Z=			15	1000	815 K	1.095			200	1			1				
P21B	53	53	P21BA				X= Y= Z=			15	1000	815 K	1.095			200	1			1				
V22A	54	54	V22AA				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
V22B	55	55	V22AB				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
V22C	56	56	V22AC				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
L22A	57	57	L22AA				X= Y= Z=			15	200	118 K	1.535			200	1			1				
L22B	58	58	L22AB				X= Y= Z=			15	200	118 K	1.535			200	1			1				
VA22	59	59	V22AB				X= Y= Z=			15	200	681 K	1.980			200	1			1				

NOTES:

ACAD FILE: MCTW06.DWG

PEACEKEEPER RAIL GARRISON
TEST CONFIGURATION DATA SHEET

PAGE 7 OF ____

TEST NAME M.C. TRACK WORTHINESS DATE ________ W.O. 87559 LOC. ____________

INSTR. ENGR./TECH. ____________ TEST ENGR. B. WHITSETT ____________ QA ____________

SOFTWARE/VERSION ____________ RECORDER I.D. NO. ____________ SET-UP FILE ____________

SAMPLE RATE ____________ ENCODER/DIGITIZER I.D. NO. ____________

INST INIT	DAS CH	PP CH	MEAS. CODE	TRANSDUCER					AMPLIFIER						FILTER				SYSTEM			RECORDER		COMMENTS
				MFG.	S.N.	SENS.	LOC.	CAL VOID DATE	CH. NO.	EXC.-L/R	GAIN FIX/VAR	R-CAL RES.	CAL. E.U. & VOLTS	S.N. CAL VOID DATE	NO.	FREQ	GAIN	CAL VOID DATE	AO (E.U.)	AI (E.U./VOLT)	ENGR. UNITS	CH. NO.	SENS. (E.U./DIV.)	
VB22	60	60	V22BB				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
VC22	61	61	V22BC				X= Y= Z=			15	1000	681 K	1.980			200	1			1				
LA22	62	62	L22BA				X= Y= Z=			15	200	118 K	1.535			200	1			1				
LB22	63	63	L22BB				X= Y= Z=			15	200	118 K	1.535			200	1			1				
JBX4A	64	64					X= Y= Z=																	ROLL RATE A-END RAW GYRO
P22A	65	65	P22AA				X= Y= Z=			15	1000	815 K	1.095			200	1			1				
P22B	66	66	P22BA				X= Y= Z=			15	1000	815 K	1.095			200	1			1				
	67	67					X= Y= Z=																	
							X= Y= Z=																	
							X= Y= Z=																	

NOTES:

ACAD FILE: MCTW07.DWG