UMTA-MA-06-0025-83-1
DOT-TSC-UMTA-82-28

Wheel/Rail Force Measurement at the Washington Metropolitan Area Transit Authority - Phase II

Volume I
Analysis Report

J.A. Elkins

The Analytical Sciences Corporation
Reading MA 01867

June 1983
Final Report

This document is available to the public through the National Technical Information Service, Springfield, Virginia 22161.

U.S. Department of Transportation
Urban Mass Transportation Administration

Office of Technical Assistance
Office of Systems Engineering
Washington DC 20590

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
UMTA-MA-06-0025-83-1	PB83 245472	

4. Title and Subtitle	5. Report Date
Wheel/Rail Force Measurement at the Washington Metropolitan Area Transit Authority-Phase II. Volume I: Analysis Report.	June 1983
	6. Performing Organization Code: TSC/DTS-77
7. Author(s): John A. Elkins	8. Performing Organization Report No.: DOT-TSC-UMTA-82-28
9. Performing Organization Name and Address: The Analytical Sciences Corporation* One Jacob Way Reading, Massachusetts 01867	10. Work Unit No. (TRAIS): MA-06-0025(UM204/R2608)
	11. Contract or Grant No.: DTRS-57-80-C-00062
12. Sponsoring Agency Name and Address: **U.S. Department of Transportation Urban Mass Transportation Administration 400 Seventh Street, S.W. Washington, D. C. 20590**	13. Type of Report and Period Covered: Final Report Volume I of II February 1981-March 1982
	14. Sponsoring Agency Code: URT-10

15. Supplementary Notes *Under contract to: U.S. Department of Transportation; Research and Special Programs Administration; Transportation Systems Center; Cambridge, MA 02142. Volume II of this report is the Test Report (UMTA-MA-06-0025-83-2).

16. Abstract

In support of the Office of Rail and Construction Technology of the Urban Mass Transportation Administration, the Transportation Systems Center (TSC) is conducting analytical and experimental studies to relate transit truck design characteristics, wheel/rail forces, and wheel/rail wear rates, in order to provide options for reducing the wear rates of wheels and rails experienced by transit properties and minimizing system life cycle costs of vehicle and track components, while maintaining or improving equipment performance. As part of this work, TSC planned and implemented an expanded Phase II measurements program in order to: 1) obtain onboard wheel/rail force measurements over a representative range of the Washington Metropolitan Area Transit Authority's (WMATA's) operation conditions; 2) obtain data to quantify the load environment of direct fixation fasteners and evaluate the influence of changes in fastener characteristics on performance; 3) evaluate the influence of taper and suspension modification on high speed stability; and 4) assess the feasibility of a retrofit to the WMATA truck to improve curving performance. These tests were conducted in the Fall of 1981. This report, Volume I, describes analytical studies, shop tests, and general field test activities provided in support of the WMATA Phase II truck test project. These tests were aimed at evaluating effects of increased wheel taper and softened primary suspension stiffness on vehicle and truck curving performance and stable speed range. An overview of the test series and a preliminary discussion of principal test results is also included. Shop tests were conducted to provide accurate measurements of selected vehicle and truck parameters for use in engineering analyses to predict vehicle curving and stable speed capability as a function of primary suspension stiffness and wheel taper parameters. These results were used to develop design specifications for fabricating a modified primary suspension element with reduced longitudinal stiffness for field test evaluation. Selected truck configurations and the utilization of existing or modified suspension elements with different wheel tapers, were subsequently evaluated during field tests using typical curves ranging from 2.3 to 7.6 degrees. Measurement indicated that mean curving forces were substantially reduced by softening the longitudinal suspension and providing a slight increase in wheel taper. In addition, measurements were unable to detect any degradation in the vehicle stability as a result of these changes. Uncertain ties associated with experimental and analytical results are also discussed, as well as conclusions and recommendations.

17. Key Words	18. Distribution Statement
Curve Negotiation; Rail Transit; Truck Hunting; Washington, D.C.; Washington Metropolitan Area Transit Authority; Wheel/Rail Force Measurement; Wheel/Rail Wear	**Available to the Public through the National Technical Information Service, Springfield, Virginia 22161.**

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	82	

Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

PREFACE

In support of the Office of Rail and Construction Technology of the Urban Mass Transportation Administration (UMTA), the Transportation Systems Center (TSC) is conducting analytical and experimental studies to relate transit truck design characteristics, wheel/rail forces and wheel/rail wear rates, in order to provide options for reducing the wear rates of wheels and rails experienced by transit properties and minimizing system life cycle costs of vehicle and track components, while maintaining or improving equipment performance.

As part of this work, TSC planned and implemented a measurement program, in order to obtain onboard wheel/rail force measurements over a representative range of Washington Metropolitan Area Transit Authority (WMATA) operating conditions; obtain data to quantify the load environment on direct fixation fasteners and evaluate the influence of changes in fastener characteristics on fastener performance; evaluate the influence of taper and suspension modifications on high speed stability and to assess the feasibility of a retrofit to the WMATA truck to improve curving performance. These tests were conducted in the fall of 1981. The Analytic Sciences Corporation (TASC), under Contract DTRS-57-80-C-00062, provided support to TSC in these activities, by conducting analyses of the tradeoffs between curving performance and high speed stability, by definition and coordination of in-shop measurements to obtain engineering parameters for use in the analysis, by specification and procurement of the retrofit primary suspension element used in the truck tests, by comparison of measured data, with analytic predictions to assess measurement consistency and by recommending test program modifications to improve the accuracy and completeness of the results relating to the truck modifications. This report describes the work performed by TASC under this effort in support of this measurement program.

Vehicle/truck instrumentation and data acquisition support for the truck tests was provided by ENSCO, Inc., while equipment and support personnel, for conducting selected vehicle and truck measurements in the WMATA shop, were provided by the Transportation Test Center (TTC), Pueblo, Colorado under Contract DTFR-53-80-C-00002. The report number is UMTA-MA-06-0025-83-. The modified primary suspension elements used in the (Phase II) test program were developed and fabricated by the BUDD Co., under subcontract to and in accordance with design specifications developed by TASC. Vehicles, operators, track rights, shop facilities and shop test support were provided by WMATA.

Wayside instrumentation development, installation and calibration and data acquisition support for measurements of fastener loads was provided by Battelle Columbus Labs, under contractual arrangement with TSC. Results of these measurements will be documented in subsequent reports.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

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

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

Approximate Conversions from Metric Measures

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

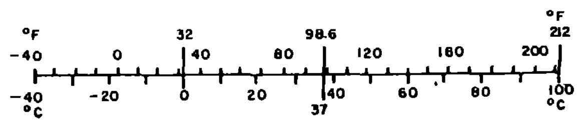


TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

LIST OF FIGURES

LIST OF FIGURES (Continued)

LIST OF FIGURES (Continued)

LIST OF TABLES

SUMMARY

Early experience of the WMATA Rail Rapid Transit System indicated surprisingly high rates of wheel and rail wear, resulting in higher than anticipated maintenance costs. At the request of UMTA, TSC conducted an initial evaluaton of the WMATA wheel rail wear and concluded that some improvements might be effected by changes in wheel profile and track gauge. Accordingly, a limited series of tests was conducted in the summer of 1979, using wayside force measurements at the curve approaching the National Airport Station to evaluate the magnitude of the wheel rail forces being experienced and the effectiveness of a British Rail 1/20 profile and gauge variation in reducing these forces. These tests demonstrated that the British Rail profile and widening of the gauge did produce a reduction in wheel rail force. The limited data obtained, however, did indicate a large fluctuation in forces from location to location but was not sufficient to determine the high speed stability changes that might be produced by the increased wheel taper. Concurrently, WMATA was experiencing a high rate of rail fastener failures at a number of curves in the system and was seeking data to quantify the fastener load environment. At about the same time, tests conducted at the TTC for UMTA, on a Metropolitan Atlanta Rapid Transit Authority (MARTA) car, indicated that sufficient changes could be made in the longitudinal primary stiffness, within the available volume, to produce a significant reduction in curve negotiation forces.

Accordingly, TSC planned and implemented an expanded Phase II measurement program in order to; obtain onboard wheel rail force measurements over a representative range of WMATA operating conditions; obtain data to quantify the load environment on direct fixation fasteners and evaluate the influence of changes in fastener characteristics on fastener performance; evaluate the influence of taper and suspension modifications on high speed stability and to assess the feasibility of a retrofit to the WMATA truck to improve curving performance. These tests were conducted in the fall of 1981. TASC provided support to TSC in the Phase II activities by conducting analyses of the tradeoffs between curving performance and high speed stability, definition and coordination of in-shop measurements to obtain engineering parameters for use in the analysis, by specification and procurement of the retrofit primary suspension element used in the truck tests, comparison of measured data with the analytic predictions to assess measurement consistency and to recommend test program modifications to improve the accuracy and completeness of the results relating to the truck modifications. This report describes the work performed by TASC under this effort in support of the Phase II activities.

The shop measurements of vehicle and truck physical characteristics consisted of vehicle/secondary suspension response tests, primary suspension stiffness measurements of the standard and modified primary bush designs and measurements of axle misalignment.

Analytical studies to define tradeoffs between curving and stable speed capabilities as a function of wheel taper and primary suspension stiffness, were performed. These analyses indicate that for a maximum effective wheel taper of 0.2 (1/5) the longitudinal stiffness of the primary suspension element could be reduced from 115,000 lb/in to 25,000 lb/in without producing

hunting oscillations at speeds below 75 mph. Calculations of curve negotiation forces indicate relatively low sensitivity to wheel taper. The curve negotiation forces are found to monotonically decrease with primary longitudinal stiffness (i.e., the lower the longitudinal stiffness the smaller the curve negotiation forces). Accordingly, a target reduction in longitudinal stiffness from 115,000 lb/in to a value of 25 - 30,000 lb/in was established, to provide a maximum reduction in longitudinal stiffness while maintaining stability over the operating speed range.

The design constraints imposed by the truck configurations and practical considerations such as limits on truck axle displacements were considered. A review of bushing design considerations indicated that an experimental bushing with a longitudinal stiffness of 25,000 lb/in was feasible and a contract was issued to the Budd Company for fabrication and delivery of 10 sets of bushes for the test program. The longitudinal stiffness of the bushes delivered were measured at 29,000 lb/in as installed in the unloaded WMATA car.

A brief description of the truck related portions of the tests is provided and the preliminary results of the test data analysis summarized. For the unmodified suspension, the measured mean value of lateral force in curve 311 (a 6 degree curve) was 5,700 lb at balance speed. Significant fluctuations were observed in the lateral wheel forces, with a peak force of 10,000 lb occurring in the body of the curve. Use of the British Rail 1/20 taper with the unmodified suspension produced a 27% reduction in the mean lateral force and the use of the AAR 1/10 taper produced a 40% reduction in mean lateral force for the sharpest curve measured. During the tests with the AAR 1/10 taper, problems were encountered with the wheel tread impacting the frogs of turnouts. Concern about the effect of these impacts resulted in restrictions on further running with the 1/10 tapered wheels. The modified suspension resulted in a 36% reduction in mean lateral force for the cylindrical wheels and reductions in mean lateral force of 56% with the BR 1/20 profile and 75% with the AAR 1/10 profile. The dynamic fluctuations in lateral force, relative to the mean force of about ±3,500 lb, was not strongly influenced by either the profile or suspension modifications.

The reductions in lateral force that were obtained with the modified suspension were in reasonable agreement with the analytic results. However, the effects of wheel taper were poorly predicted. The disagreement between the analytic results and the test data on the influence of taper variations is believed to be due to the wheel rail contact assumptions used in the analysis. The analysis is currently being extended to include the effect of two point contact to provide a more effective predictive tool.

The measurements of car body response, wheel rail forces and axle acceleration indicated no evidence of hunting at speeds less than the maximum operating speed of 75 mph attained in the tests.

It is recommended that a vehicle equipped with the experimental modified suspension and the British Rail 1/20 profile be placed in service on a trial basis with regular measurements of wheel profile and suspension settling. The purpose of this trial operation would be to provide data on the wear behavior of the modification and supplementary data to assist in design of a retrofit configuration. In addition to the extensions in the analysis currently being conducted to include two point wheel rail contact situations, further analyses of the dynamics of the vehicle track interaction are required to account for the large dynamic force variations and irregular wear patterns observed.

1. BACKGROUND

During the early years of service, WMATA has experienced excessive levels of wheel and rail wear. An early study, which was conducted by De Leuw Cather and Company, indicated that the severe rail wear was confined to curves of less than 1000 feet radius (or greater than 5-7° of curvature) (Ref. 1). In view of the additional cost of maintenance, which was likely to result from the excessive rates of wear and the implications for other Transit Authorities, UMTA sponsored a test program to investigate the problem.

TSC conducted an initial evaluation of the WMATA wheel rail wear and concluded that some improvements might be effected by changes in wheel profile and track gauge. Accordingly, a limited series of tests was conducted in the summer of 1979, using wayside force measurements at the curve approaching the National Airport Station to evaluate the magnitude of the wheel rail forces being experienced. In addition, the effectiveness of a change from the standard AAR cylindrical profile (Fig. 1-1) to a British Rail 1/20 profile (Fig. 1-2) and variations in track gauge were evaluated. These tests demonstrated that the British Rail profile and widening of the gauge did produce a reduction in wheel rail force. The limited data obtained, however, did indicate a large fluctuation in forces from location to location but was not sufficient to determine the high speed stability changes that might be produced by the increased wheel taper. Concurrently, WMATA was experiencing a high rate of rail fastener failures at a number of curves in the system and was seeking data to quantify the fastener load environment.

In addition to the situation described above, MARTA had experienced similar problems during the early service life of their vehicles. As a result, a test program, sponsored by UMTA, had been carried out at the TTC (Refs. 4 and 5). During this test program, wayside measurements of wheel/rail forces and angles-of-attack were made. The effects of a modification to reduce the longitudinal stiffness of the primary suspension bush and a change in wheel profile from the standard AAR 1 in 20 (Fig. 1-3) to a CN-A worn wheel profile (Fig. 1-4) were tested. In addition, the effect of axle misalignments was investigated, as it had been noted that some trucks were experiencing particularly high rates of wheel wear and that this was occurring asymmetrically.

The test results were compared with the predictions from a mathematical model of the truck curving behavior (Refs. 6 and 7). Subsequently, the model was used to examine the likely effect on wheel and rail wear rates of the various parameters discussed above. Results indicated that a substantial reduction in wear rate was likely to result from the reduction in primary longitudinal stiffness. The analysis confirmed by the test data indicated that the stiff longitudinal suspension used in this truck made the truck curving forces particularly sensitive to axle misalignment. It is quite likely that the asymmetries in wheel wear and variations in wear from truck to truck were due to variations in axle alignment of the order of 0.2 degree.

Following the Phase I tests at WMATA and the tests on the MARTA cars at the TTC, TSC planned a Phase II test series at WMATA which included definition and evaluation of the effects which feasible modifications to the truck

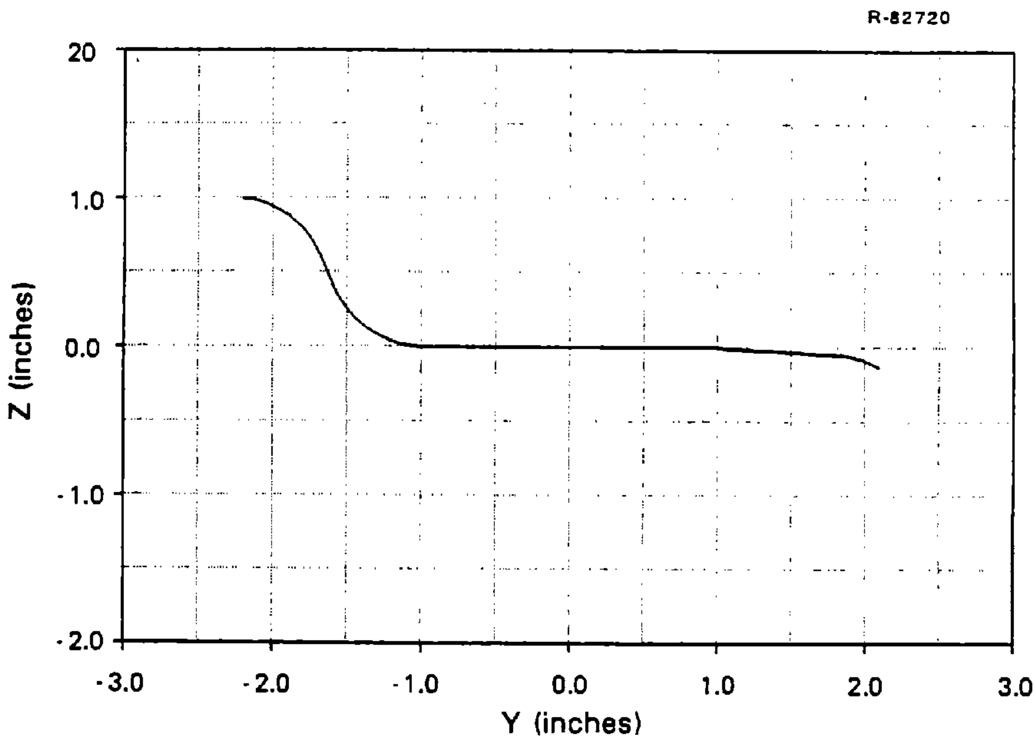


Figure 1-1   AAR Cylindrical Profile

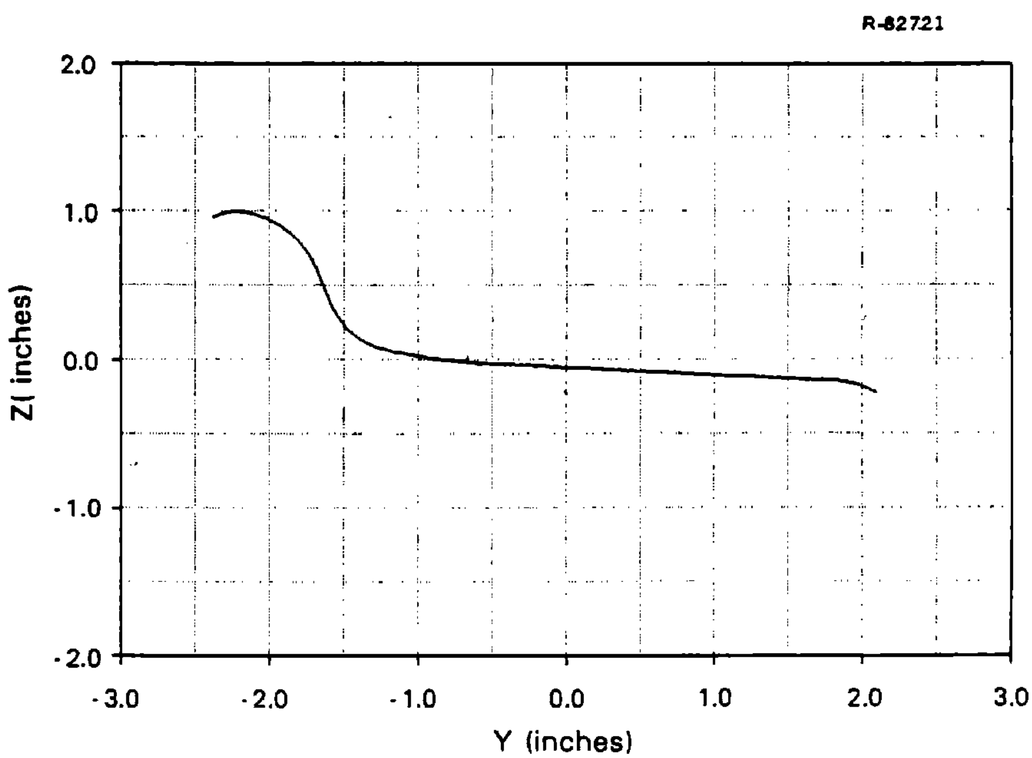


Figure 1-2   British Rail 1 In 20 Profile

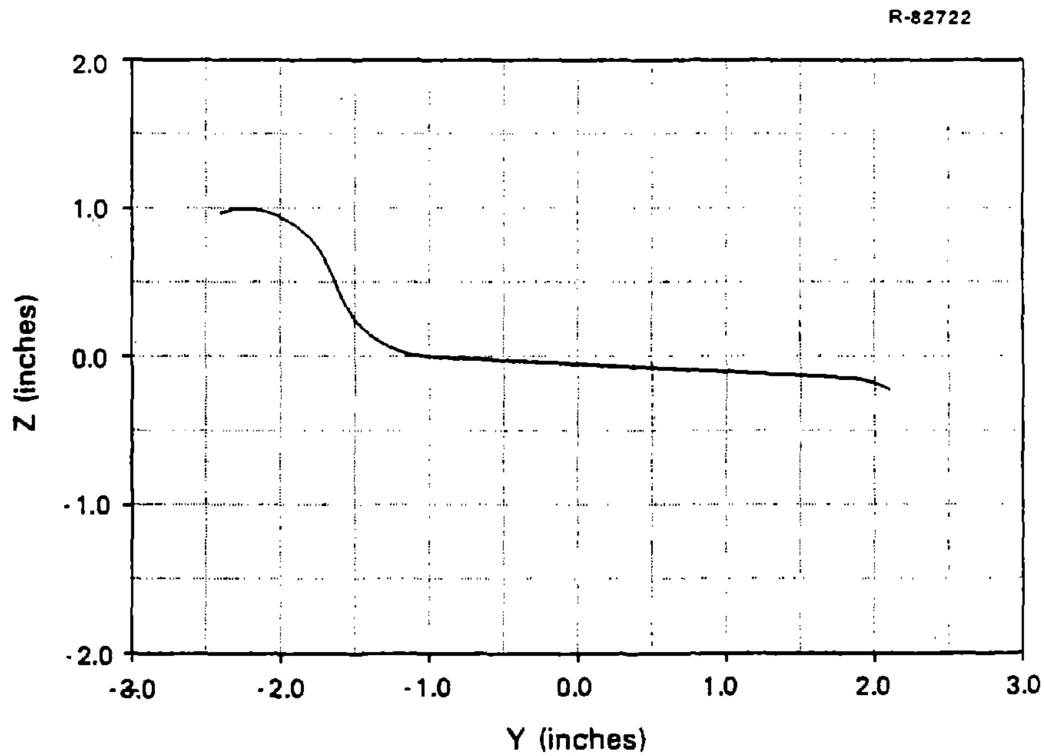


Figure 1-3 AAR 1 In 20 Profile

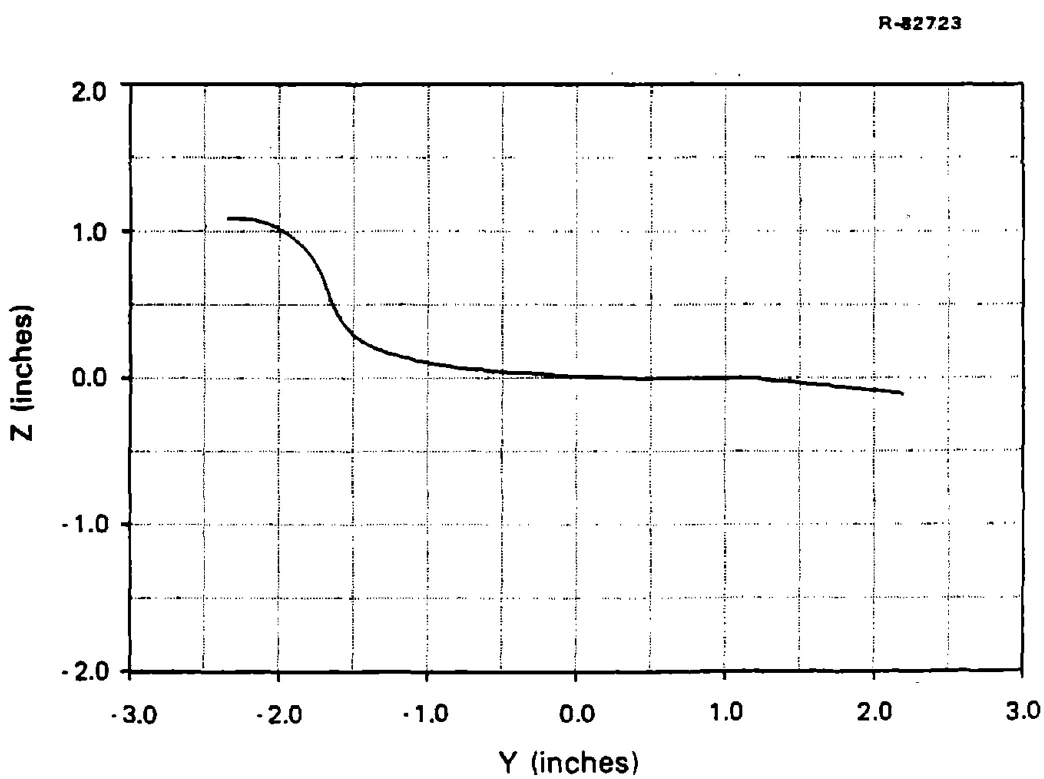


Figure 1-4 CN-A Profile

primary suspension and/or wheel profile would have on curving forces. TASC participated in this activity under a technical task directive from TSC. The purpose of the effort was to provide support in the form of engineering and design studies aimed at characterizing the vehicle and defining the parametric values of the truck modifications, to assist in the planning and implementation of the tests and to compare the test results with the predictions from a mathematical model of the vehicles' curving behavior.

2. OBJECTIVES

The primary objectives of this effort were:

- To establish optimum values for the primary suspension longitudinal and lateral stiffnesses
- To determine the feasibility of producing a primary suspension bush, that achieved the optimum parameters within the space envelope occupied by the existing bush
- To establish, by theory and experiment, the reduction in wheel/rail forces that could be obtained by modifying the primary suspension
- To investigate whether further improvements in curving performance were available from wheel/rail effective conicities higher than the 0.05 used during the Phase I test program
- Evaluation of the influence of suspension and wheel profile changes on hunting stability and speed capability of the WMATA car.

3. VEHICLE CHARACTERIZATION

3.1 DATA FROM MARTA VEHICLE AND FROM SUPPLIERS

In order to estimate the curving and critical speed characteristics of the WMATA vehicle, it was necessary to compile a complete set of mass, damping and stiffness parameters. The Rockwell truck used on the WMATA vehicle is very similar to the same manufacturer's truck on the MARTA vehicle. Accordingly, many of the parameters for the WMATA vehicle are the same as those for MARTA. As a complete set of vehicle parameters had been compiled for the MARTA vehicle (Refs. 8 and 9) this was a convenient source for many of the required parameters.

Rockwell, the truck manufacturer, confirmed that many of the truck and secondary suspension parameters were the same as for MARTA. They also provided data on some of the parameters that were different (Ref. 10). WMATA supplied information on car body dimensions and masses and truck drawings describing the truck frame/primary suspension system interface.

An eigenvalue analysis was performed using this initial set of vehicle parameters. The results of the analysis indicated that all of the car body on secondary suspension modes were more than critically damped. Experience with other vehicle designs indicates that secondary suspension parameters are usually selected to obtain modal damping in the range 20% to 40%. Therefore, the results from this analysis suggested that the estimated secondary damper rates were too high. In addition, experience from riding the WMATA vehicle had indicated the existence of a very low frequency lower center roll mode, which was fairly lightly damped, and a body yaw mode at approximately 1 Hz.

A close examination of the vehicle data suggested that the value being used for the secondary lateral damper rate was too high. Discussions with Houdaille, the damper manufacturer, produced data for both the secondary vertical and lateral damper rates. The new value for the secondary lateral damper rate was considerably lower than that used previously.

A further eigenvalue analysis with this new data produced three oscillatory modes; lower and upper center roll and yaw for the car body on secondary suspension. However, the natural frequencies of these modes did not correspond with values that were apparent from riding the vehicle.

Accordingly, a simple resonance test was conducted to establish values for the carbody on secondary suspension natural frequencies.

3.2 VEHICLE RESONANCE TEST

The secondary vertical and lateral suspensions of the WMATA vehicle are both provided by the air springs. Damping in both the vertical and lateral direction is provided by hydraulic dampers. When these dampers are removed, very little damping remains in the secondary suspension. As a result, it is comparatively easy to excite the vehicle by means of hand excitation when the dampers are removed.

This procedure was carried out on one of the WMATA vehicles. Two people were required to excite the vehicle and different points of excitation were required for each of the five modes that were excited. Once the vehicle was in motion in a particular mode, the natural frequency was determined by measuring the time taken for approximately 50 cycles of oscillation. This was repeated several times, and the results obtained were very consistent. The approximate mode shape associated with each natural frequency was determined by observation while the vehicle was in motion.

Table 3.2-1 lists the natural frequencies for all the modes determined from this experiment.

TABLE 3.2-1
MEASURED CAR BODY ON SECONDARY SUSPENSION NATURAL FREQUENCIES

MODE	NATURAL FREQUENCY (Hz)
Lower Center Roll	0.366
Bounce	1.15
Pitch	1.25
Yaw	1.28
Upper Center Roll	1.39

The results show that all of the modes except lower center roll are very close in natural frequency. When such close spacing of the modes exists, separation has to be obtained by careful selection of the points of excitation. This was the reason for using different points of excitation for each of the modes. Knowledge of the car body mass and all of the car body on secondary suspension frequencies permits calculation of all the secondary suspension stiffnesses and car body inertias. The process by which this was performed is described in detail in Appendix A.

The results of this analysis revealed considerable errors in the data for secondary suspension stiffnesses that had been used previously. The results from this test are contained in Table 3.3-2, which appears later and lists the complete set of vehicle parameter data used subsequently in the theoretical analysis.

3.3 SHOP TESTS ON THE TRUCK

A number of vehicle parameters have an important effect upon the lateral stability and curving performance of the vehicle. These parameters were measured during a static test performed in the WMATA maintenance shop at Brentwood. The tests carried out included:

- Primary longitudinal stiffness test
- Primary lateral stiffness test
- Primary vertical stiffness test
- Truck rotation test
- Axle alignment test.

The three stiffness tests and the axle alignment test were carried out on trucks equipped with the standard and modified primary suspension bushings.

In order to perform the longitudinal stiffness test, the brake discs and calipers were removed from the truck. In addition, the third rail paddle assembly was removed. A longitudinal force was applied between the two axles of the truck by means of two small hydraulic actuators. These were positioned on either side of the truck frame and applied their force through two chains, which were passed through the hollow center of each axle. Load cells were placed in series with each of the actuators to measure the longitudinal force being exerted. Longitudinal displacements of the bushings were measured using dial indicators.

During the test with standard bushings, measurements using the displacement transducers mounted to measure primary longitudinal displacement, were compared with the values being obtained from the dial indicators and found to be in good agreement. Figure 3.3-1 illustrates the test set-up used for performing this test.

For the lateral stiffness test, the third rail paddle assembly was removed. A fixture was attached in its place for the purpose of reacting a lateral force applied through a chain from a single hydraulic actuator. A load cell in series with the actuator measured the applied force. Lateral displacements across the primary suspension were measured using dial indicators. The set-up used for this test is illustrated in Figure 3.3-2.

The vertical stiffness test was performed by jacking underneath the primary suspension journal housing. In this manner, a proportion of the vertical load, normally supported by the primary suspension bush, is reacted by the jack. Increasing the load supported by the jack, decreases the load on the primary bush. The load being applied was measured using a button load cell, placed between the jack and the journal housing. Again displacements were measured using dial indicators.

The truck rotation test measured the break out value of yaw torque between the truck and the car body. The magnitude of this torque may have a significant effect upon the hunting speed of a vehicle. During this test, both axles of the truck were supported on a single air bearing table which allowed the truck to yaw freely with respect to the ground. Equal and opposite lateral forces were applied at diagonally opposite corners of the air bearing table, with the magnitude of the forces again measured using load cells. The angular displacement of the air bearing table with respect to the ground was measured using dial indicators.

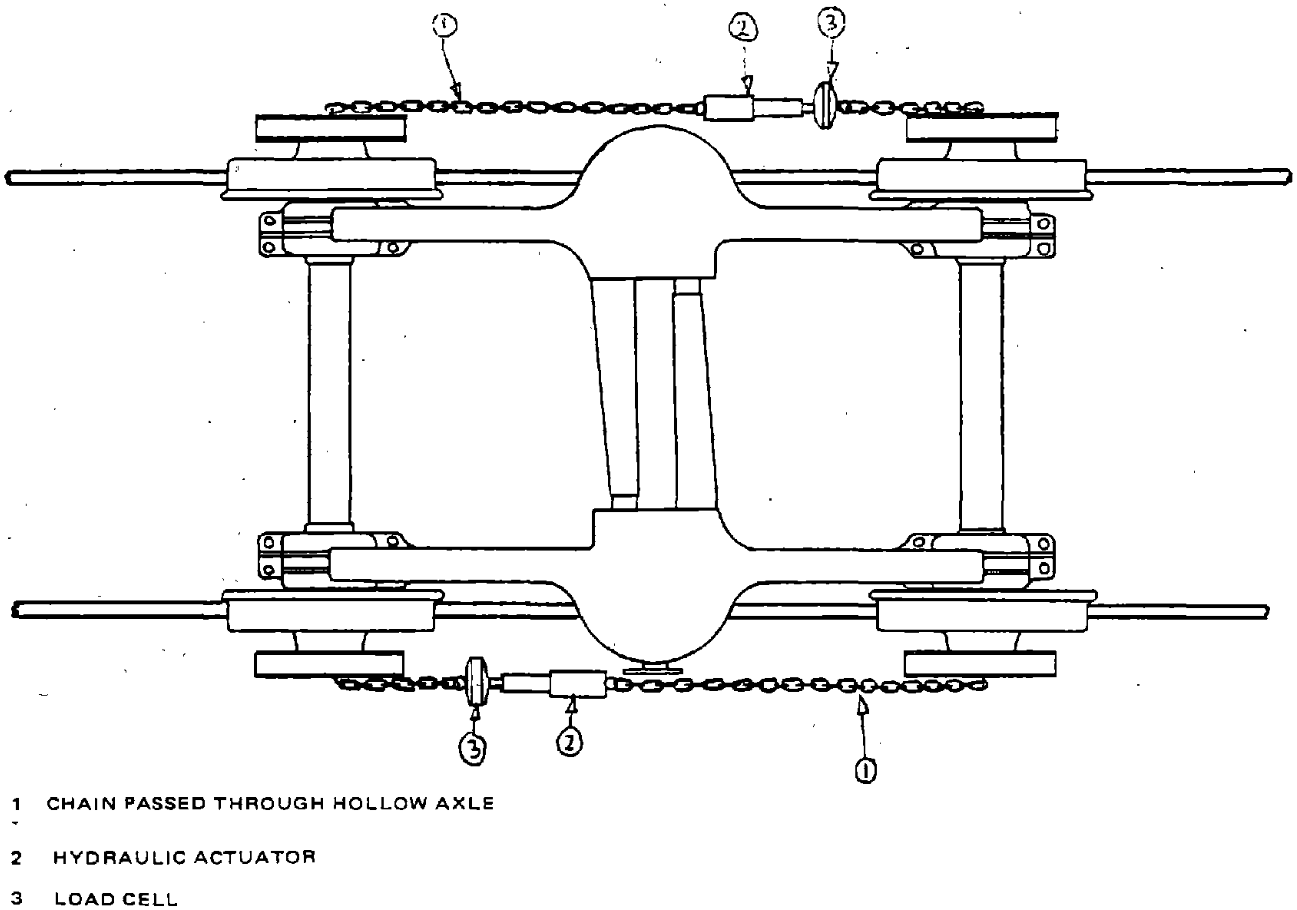


Figure 3.3-1 Primary Longitudinal Stiffness Test Arrangement

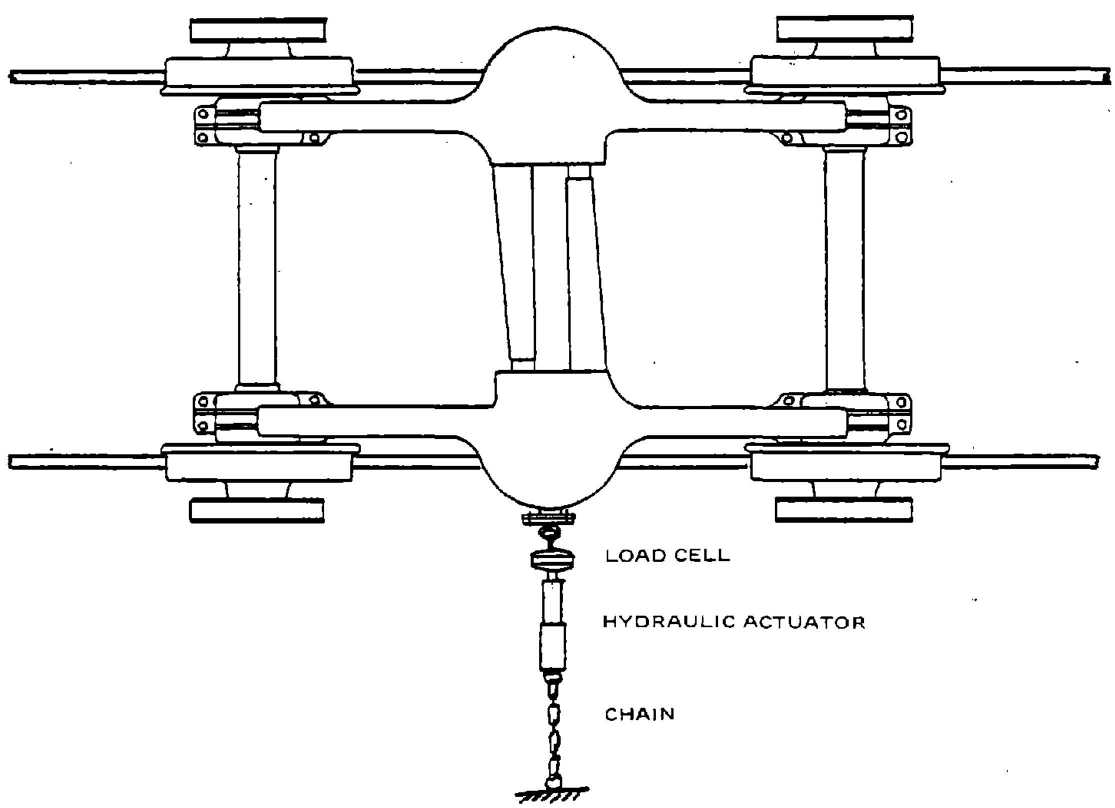


Figure 3.3-2 Primary Lateral Stiffness Test Arrangement

With the truck floating freely on the air bearing table, loads were increased slowly up to the point where gross rotation of the truck took place. The magnitude of the breakaway torque was determined from the levels of the applied loads required to obtain gross rotation.

The truck parameters determined during these tests with standard and modified bushes are presented in Table 3.3-1. It will be noted from Table 3.3-1 that the modified bushes have a somewhat higher vertical stiffness than the standard bushes. Previous information had suggested that the vertical stiffness of the standard bushes would be close to 100,000 inch/in. and this was the value specified for the modified bushes. In the event, the standard bushes turned out to be somewhat softer than expected and the modified bushes were slightly stiffer than the specification.

TABLE 3.3-1
TRUCK PARAMETERS MEASURED IN LABORATORY

PARAMETER	STANDARD BUSH	MODIFIED BUSH
Primary Vertical Stiffness (per wheel)	74,000 lb/in.	116,000 lb/in.
Primary Lateral Stiffness (per wheel)	62,300 lb/in.	32,000 lb/in.
Primary Longitudinal Stiffness (per wheel)	115,000 lb/in.	29,000 lb/in.
Secondary Yaw Pivot Friction (per truck)	90,000 lb-in.	90,000 lb/in.

The final test carried out in the shop was concerned with the alignment of the two axles in a truck. It had been found during the tests on the MARTA vehicle that axle misalignment could be a significant factor in determining the vehicles' curving behavior. In particular, wheelset angles of attack measured on right-hand curves were different from those measured on left-hand curves. For this test, two air bearing tables were used to support each of the two axles of a truck. The air bearing tables allowed the two axles to take up a relative position in which the primary suspension was unstrained. When the air bearing tables were deflated and lowered to the ground, it was assumed that the primary suspension remained unstrained.

The alignment of the axles was then measured using an optical technique (Fig. 3.3-3). An optical transit was used to enable very accurate measurements to be made, relative to an optical line-of-sight (Fig. 3.3-4). Measurements were made of the lateral distance from two points on the outside face of each wheel (Y_{L1}, Y_{L2}, Y_{t1}, Y_{t2}) using precision scribed scales. The two scales on each wheel were placed as far apart longitudinally as was possible (X_L,X_t) and in contact with points on the rim, which were known to lie

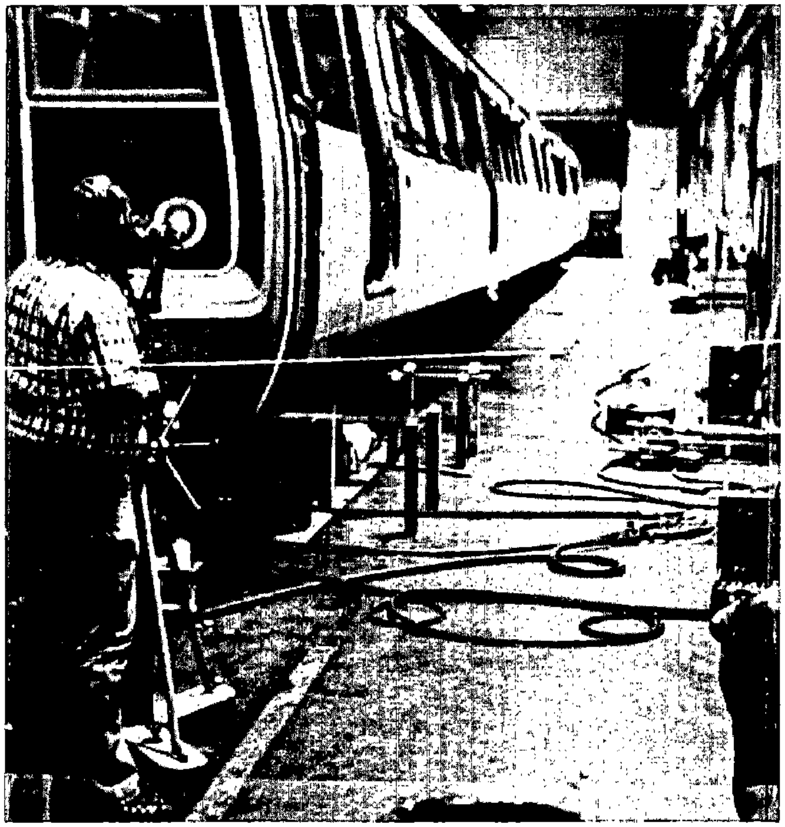


Figure 3.3-3 Axle Alignment Measurement

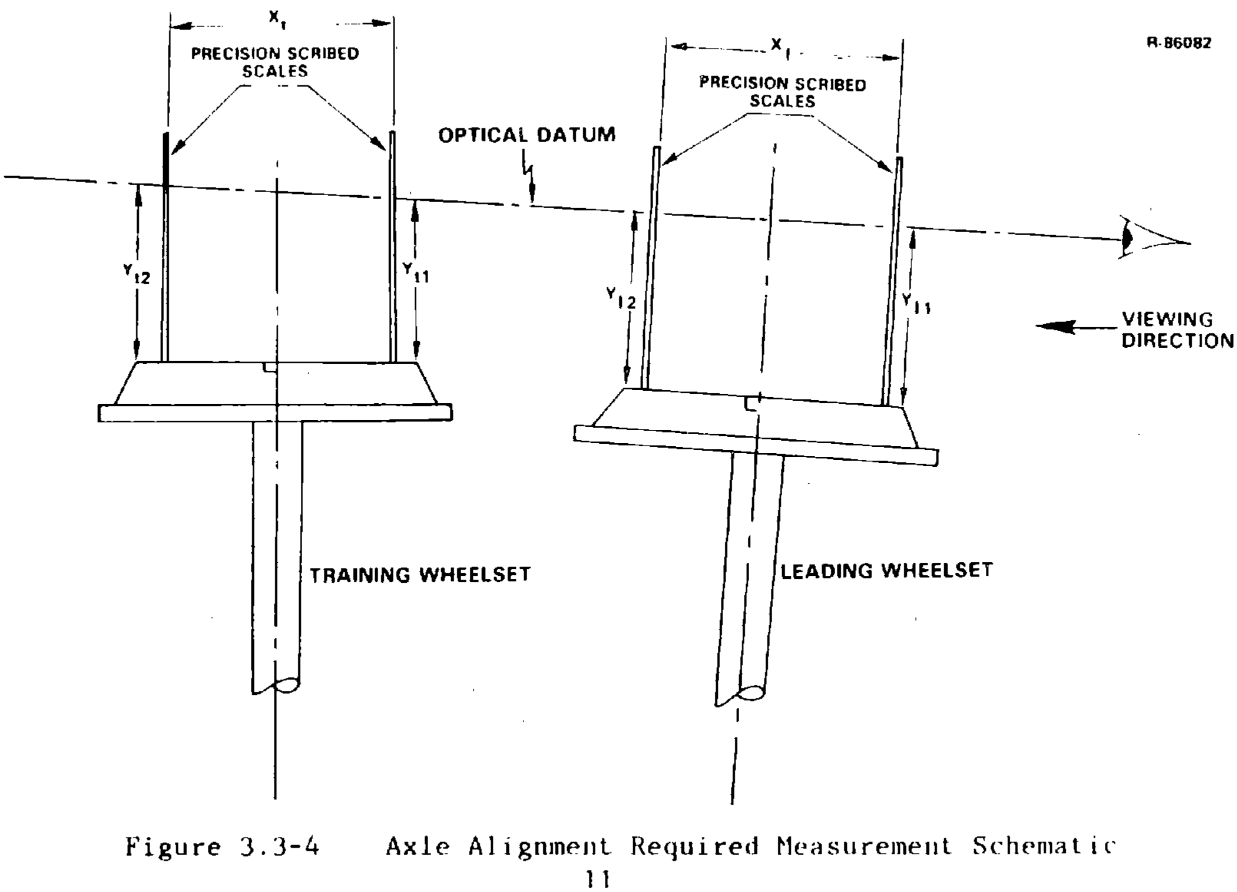


Figure 3.3-4 Axle Alignment Required Measurement Schematic

on a line perpendicular to the axle centerline. This was determined by measuring the run out on the outside face of the wheel rim as the axle was rotated in its bearings. From the four lateral distances between the outside faces of the wheels and the optical line-of-sight, it was possible to compute the angular misalignment of the two axles with respect to one another.

The alignment of the axles for the two vehicle configurations is shown in Fig. 3.3-5. As shown, the misalignments can be considered as having two components -- a radial misalignment, where the axles have equal and opposite angles with respect to the truck centerline, and a lateral misalignment where the axles are offset laterally with respect to one another. The analyses described in Ref. 5 indicate that the radial misalignment component is the more important of the two. The effect of axle misalignment, on wheel/rail forces in curves, is dependent upon the stiffness of the primary suspension, in particular the primary yaw stiffness. For example, a truck with a stiff primary yaw suspension is less tolerant of misalignment than a truck with a soft yaw suspension. Therefore, the increased misalignment, that happened to be obtained with the modified bush in this particular case, will almost certainly have less effect that the original misalignment with the standard bush.

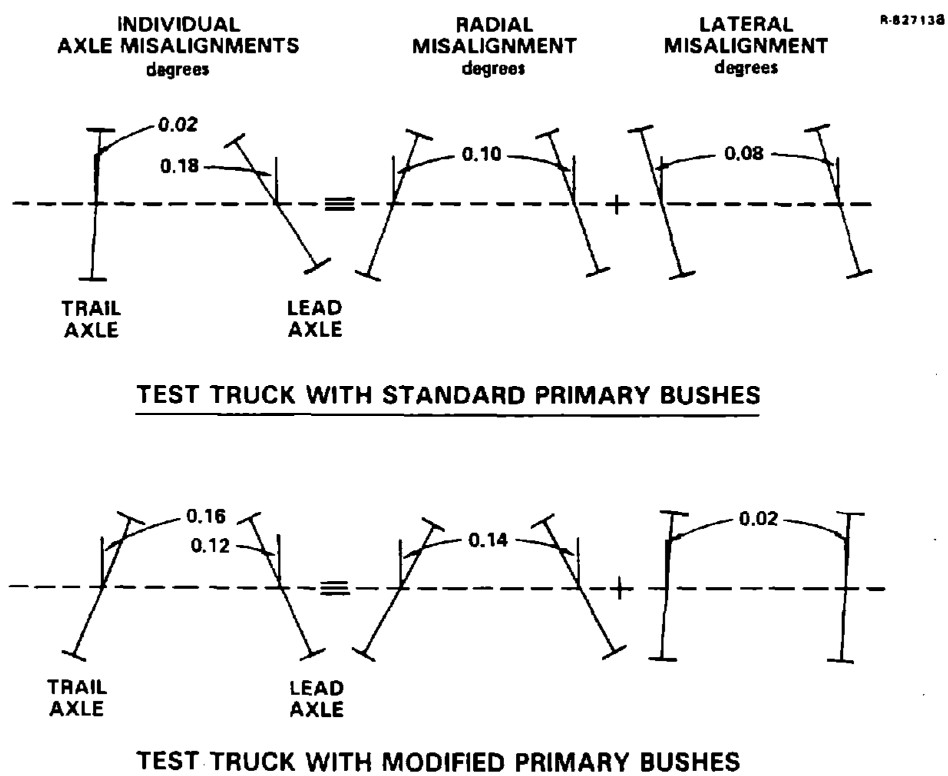


Figure 3.3-5 Measured Axle Misalignments of Test Truck

Results obtained from all the sources and tests discussed in this section of the report are incorporated in the complete set of vehicle parameter data presented in Table 3.3-2.

TABLE 3.3-2
UNLADEN WMATA CAR AND ROCKWELL TRUCK PARAMETERS

	VALUES
MASS PARAMETERS	
Body Mass	130.5 lb-sec^2/in.
(2) Car Body Yaw Inertia	10.75 10^6 lb-in.-sec^2
(2) Car Body Roll Inertia	0.53 10^6 lb-in.-sec^2
Truck Frame Mass	14.25 lb-sec^2/in.
Truck Frame Yaw Inertia	10,500 lb-in.-sec^2
Truck Frame Roll Inertia	4,600 lb-in.-sec^2
Axle Mass	8.16 lb-sec^2/in.
Axle Yaw Inertia	4,700 lb-in.-sec^2
DAMPING PARAMETERS	
Secondary Vertical Damping (per truck)	272 lb-sec/in.
Secondary Lateral Damping (per truck)	236 lb-sec/in.
Secondary Roll Damping (per truck)	0.245 10^6 lb-in.-sec/rad
(1)*Secondary Yaw Pivot Friction (per truck)	90,000 lb-in.
STIFFNESS PARAMETERS	
(2) Secondary Vertical Stiffness (per truck)	3405 lb/in.
(2) Secondary Lateral Stiffness (per truck)	3250 lb/in.
* Secondary Yaw Stiffness (per truck)	31.2 10^6 lb-in./rad
(2) Secondary Roll Stiffness (per truck)	2.17 10^6 lb-in./rad
STANDARD PRIMARY BUSHING	
(1) Primary Vertical Stiffness (per wheel)	74,000 lb/in.
(1) Primary Lateral Stiffness (per wheel)	62,300 lb/in.
(1) Primary Longitudinal Stiffness (per wheel)	115,000 lb/in.
MODIFIED PRIMARY BUSHING	
(1) Primary Vertical Stiffness (per wheel)	116,000 lb/in.
(1) Primary Lateral Stiffness (per wheel)	32,000 lb/in.
(1) Primary Longitudinal Stiffness (per wheel)	29,000 lb/in.
GEOMETRICAL PARAMETERS	
Longitudinal Semi-Spacing of Truck Centers	312 in.
Lateral Semi-Spacing of Air Springs	25.25 in.
Vertical Height of Car C.G Above Rail	55.5 in.
Vertical Height of Secondary Suspension Roll Center Above Rail	11.5 in.
Vertical Height of Truck C.G Above Rail	11.5 in.
GEOMETRICAL PARAMETERS	
Semi-Wheelbase of Truck	43.5 in.
Lateral Semi-Spacing of Primary Bushes	22.63 in.
Lateral Semi-Spacing of Wheel/Rail Contact Patches	30 in.
Wheel Radius	14 in.
OTHER PARAMETERS	
Axle Load	18,500 lbs
Lateral Creep Coefficient	1.33 10^6 lbs
Longitudinal Creep Coefficient	1.46 10^6 lbs

*Linearized values of secondary yaw damping and stiffness were obtained from the secondary yaw pivot friction torque and secondary yaw stiffness using a sinusoidal describing function method which is discussed in Section 4.2.

(1)From shop test measurements.

(2)Computed from measured carbody on secondary suspension modal frequencies.

4. ANALYTIC STUDY TO IDENTIFY OPTIMUM PRIMARY SUSPENSION STIFFNESSES AND WHEEL PROFILE

4.1 GENERAL CONSIDERATIONS OF VEHICLE BEHAVIOR

There are a number of factors concerning the performance of a vehicle which need to be considered when carrying out a study to determine optimum vehicle parameters. These factors are concerned with three principal measures of vehicle behavior:

- Ride quality
- Lateral stability
- Steady state curving performance.

When ride quality is considered, both the vertical and lateral components have to be taken into account. However, the vertical ride of this type of vehicle is affected primarily by the vertical stiffness parameters of the primary and secondary suspensions. It was not the intent of this test program to change the vertical stiffness parameters, therefore, the vertical ride quality of the vehicle was not expected to change as a result of the modifications.

The lateral ride quality of a rail vehicle, as a result of the response to track irregularities, usually consists of two major components. One of these is composed of the response of the vehicle secondary suspension modes (yaw, upper center roll and lower center roll). The other component is due to the response of the truck.

The response of the truck depends upon the effective conicity of the wheel and the rail profiles. Rail car trucks have an oscillatory mode with a natural frequency that varies with speed, which is commonly referred to as the "kinematic frequency".

Vibrations perceived by passengers are controlled primarily by the response characteristics of the secondary suspension system modes, (yaw, upper center roll, lower center roll) the principal effect of the proposed modifications in the primary suspension and wheel profile will be to change the truck kinematic modes. This will generally not influence the body mode natural frequencies but will change the damping in the body modes when the truck kinematic frequency is in the vicinity of the body frequencies. The principal requirement here is for an adequate level of damping in the kinematic mode. Lateral ride quality was not examined directly in this study, only by implication from the results of the lateral stability study.

4.2 LATERAL STABILITY STUDY

This study was carried out using CARHNT, a computer program developed by Battelle Columbus Laboratories in 1974 (Ref. 11). In this model the vehicle is described by seventeen degrees-of-freedom, which comprise the following:

- Car body lateral, yaw and roll displacements
- Truck frame lateral, yaw and roll displacements
- Axle lateral and yaw displacements.

The vehicle parameters used in the study are listed in Table 3.3-2. These parameters were obtained from a number of sources, which include various manufacturers and data obtained for the Rockwell trucks of the MARTA cars by TSC. In addition, parameters obtained from tests on the vehicle (Sections 3.2 and 3.3) were included when they became available.

In order to carry out an eigenvalue/eigenvector analysis for determining stability, as is used in CARHNT, it is necessary that the vehicle equations of motion are linear. Thus, any nonlinear vehicle parameters must be linearized.

The secondary yaw restraint between the truck and car body is a nonlinear suspension element which can have a significant effect on vehicle hunting speed. This element consists of a yaw friction torque in series with a yaw stiffness. The friction torque comes from the side bearers, and the stiffness is provided by the rubber bushes of the two radius rods, which are connected longitudinally between the bolster and the truck frame. Equivalent linear values are obtained using a sinusoidal describing function method (Refs. 12, 13).

The principal variables in this study were the primary longitudinal and lateral stiffnesses, the wheel/rail effective conicity and the creep coefficients.

If the purpose of the study had been to select a set of vehicle parameters for a production modification, then it would have been necessary to ensure that adequate damping remained in the kinematic mode at the maximum operating speed of the vehicle. Eight percent of critical damping is a reasonable value and should be obtained for a range of wheel/rail effective conicities and creep coefficients covering the range expected to occur in service.

However, the modifications being examined here, were for an experimental test program. It would be helpful in validating the analysis, if hunting could be obtained and measured during the track testing. Therefore, values were selected from the analysis with this objective in mind.

Effective conicities ranging from 0.05 to 0.3 were used in the analysis. As wheel profiles with a straight tapered tread were to be used in the test program then these values represent a 1 in 20 and a 1 in 3.33 taper, respectively. The wheel/rail creep coefficients were varied from full to half of Kalkers values.

Low conicity and low creep is a condition that often leads to a body hunting instability at speeds where the kinematic frequency is coincident with a body on secondary suspension frequency. Figure 4.2-1 shows natural frequency and damping against speed for the standard vehicle with 0.05 conicity and half creep coefficients. At a speed of 70 mph where the kinematic frequency is coincident with the body yaw frequency, there is a minimum damping of 6% in a mode which has a combination of car body yaw and truck motions. At higher speeds the damping in this mode increases. Truck hunting eventually occurs at 180 mph.

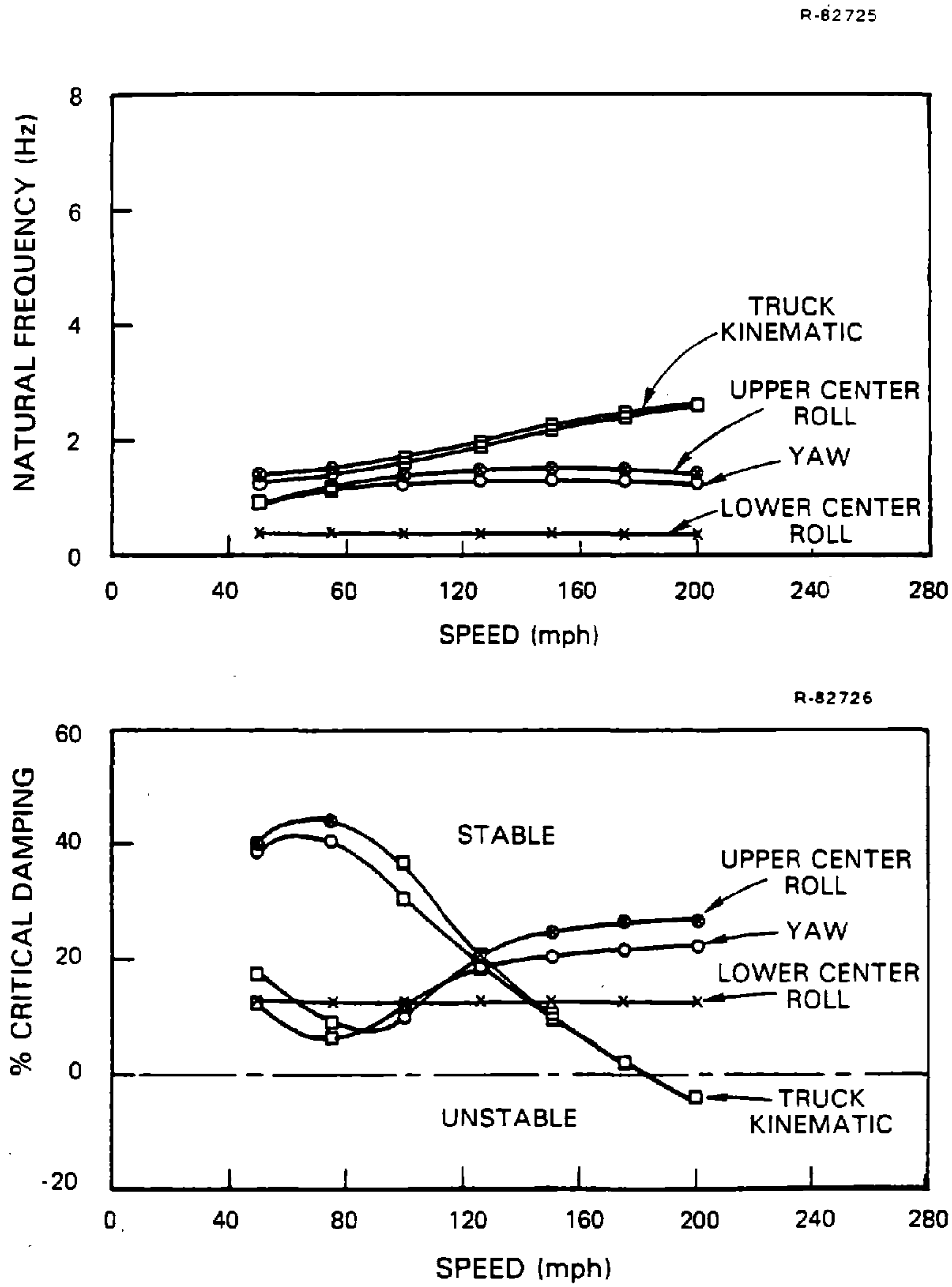


Figure 4.2-1 Natural Frequency and Damping Against Speed for Standard Vehicle (0.05 Conicity; Half Kalker)

With higher conicity the coincidence between kinematic frequency and body frequencies occurs at a lower speed and usually does not cause a very significant reduction in body mode damping. Figure 4.2-2 shows natural frequencies and damping as a function of speed for the standard vehicle with a conicity of 0.3 and full Kalker creep coefficients. Truck hunting is predicted at 108 mph.

The effect of reducing the primary suspension stiffnesses was then investigated. For the main part of the study, it was assumed that the longitudinal and lateral stiffnesses provided by the bushings were the same. The effect of varying the lateral stiffness separately was investigated later. Figure 4.2-3 shows stability boundaries for truck hunting as a function of primary stiffness for various conditions of conicity and creep coefficient. It can be seen that increased conicity reduces the critical speed and in general increased creep coefficients also reduce the critical speed.

Profiled wheels give an effective conicity which depends upon rail profile. Therefore, wheels with a straight taper were to be used during the test program so that the effective conicity for any track condition would be known. It was anticipated that a 1 in 5 taper would be the highest value used, giving a maximum effective conicity of 0.2. Figure 4.2-3 shows, that with 0.2 conicity and full Kalker creep coefficient, hunting would be obtained at the 75 mph maximum operating speed of the vehicle with a primary longitudinal stiffness of 25,000 lb/in.

Figure 4.2-4 shows natural frequency and damping against speed for the vehicle with the proposed modified primary suspension and 0.05 conicity, half Kalker creep coefficients. Comparison with Fig. 4.2-1, which is for the standard vehicle, shows that now there is a much smaller effect on damping when the kinematic frequency is coincident with the car body frequencies.

Figure 4.2-5 shows natural frequency and damping against speed for the proposed modification with a conicity of 0.2 and full Kalker, 0.2 conicity being the highest conicity that was likely to be used in the test program. The critical hunting speed for this case is approximately 78 mph.

Past experience has indicated that Kalker's full creep coefficients are rarely obtained in practice. In fact, it is quite difficult to obtain this condition in the laboratory, where the slightest contamination of the wheel or rail surface results in a lower creep coefficient (Ref. 14). As a reduction in creep coefficient was expected to give an increase in critical speed, a primary longitudinal stiffness of 25,000 lb/in. could be expected to give a small safety margin above the maximum operating speed.

The effect of varying primary lateral stiffness with a primary longitudinal stiffness of 25,000 lb/in. was then investigated. This was done because it is difficult to obtain a specified value for the lateral stiffness of the bush used on this truck at the same time as achieving the desired longitudinal stiffness. Figure 4.2-6 shows the effect on critical speed of varying primary lateral stiffness. It can be seen that for a reasonable range of variation in stiffness there is only a small change in critical speed.

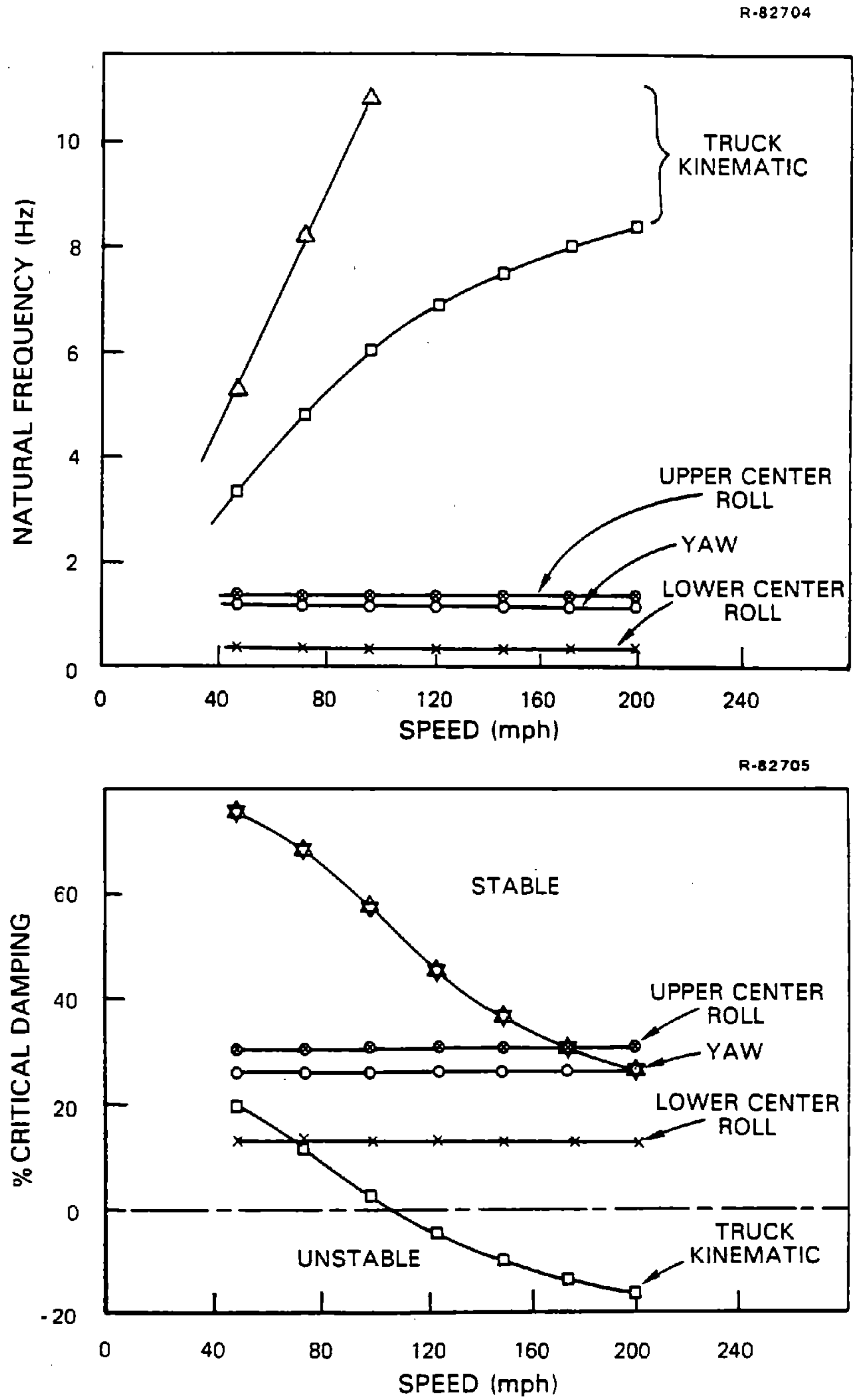
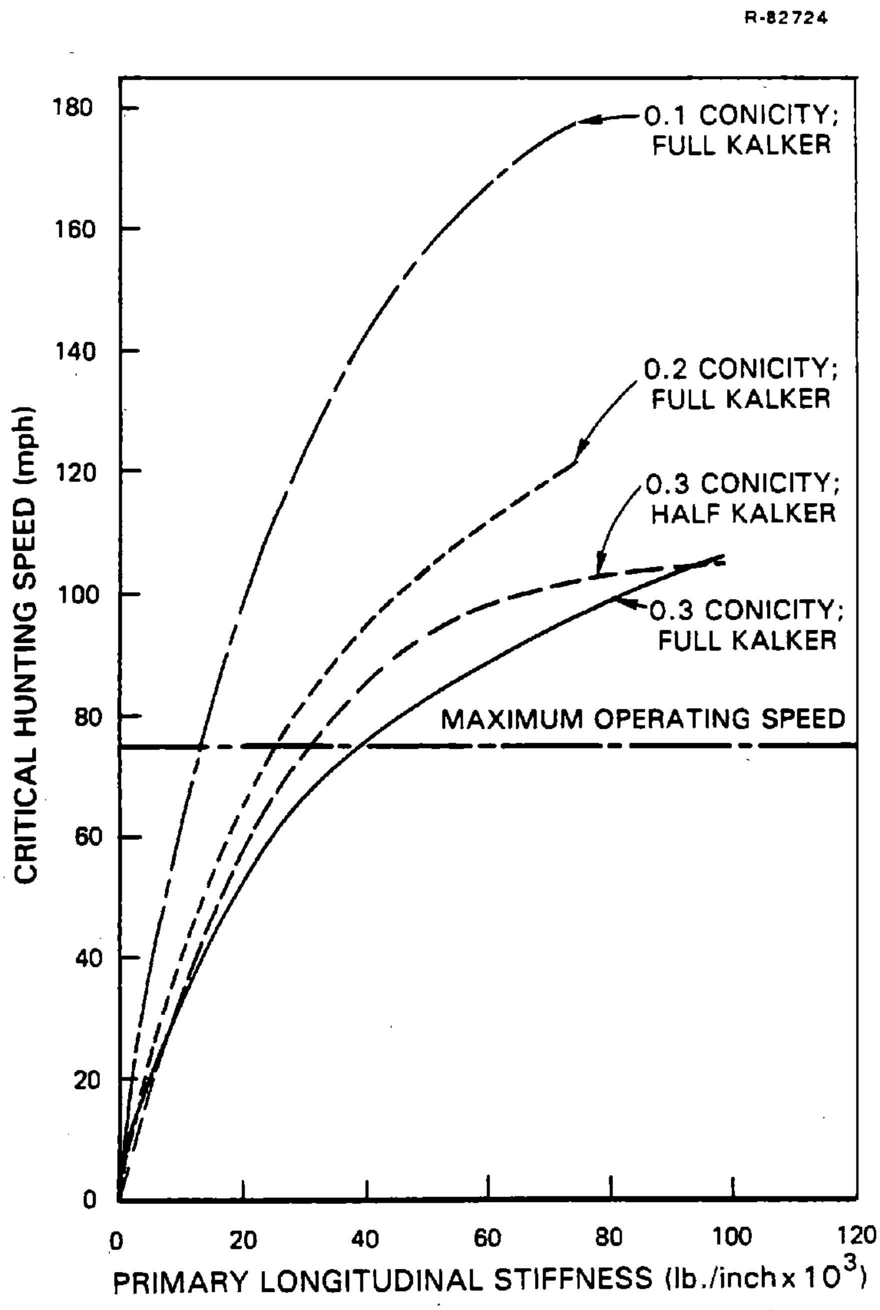


Figure 4.2-2 Natural Frequency and Damping Against Speed for Standard Vehicle (0.3 Conicity; Full Kalker)



(Primary Longitudinal Stiffness Equals Primary Lateral Stiffness)

Figure 4.2-3 Stability Boundaries Against Primary Longitudinal Stiffness

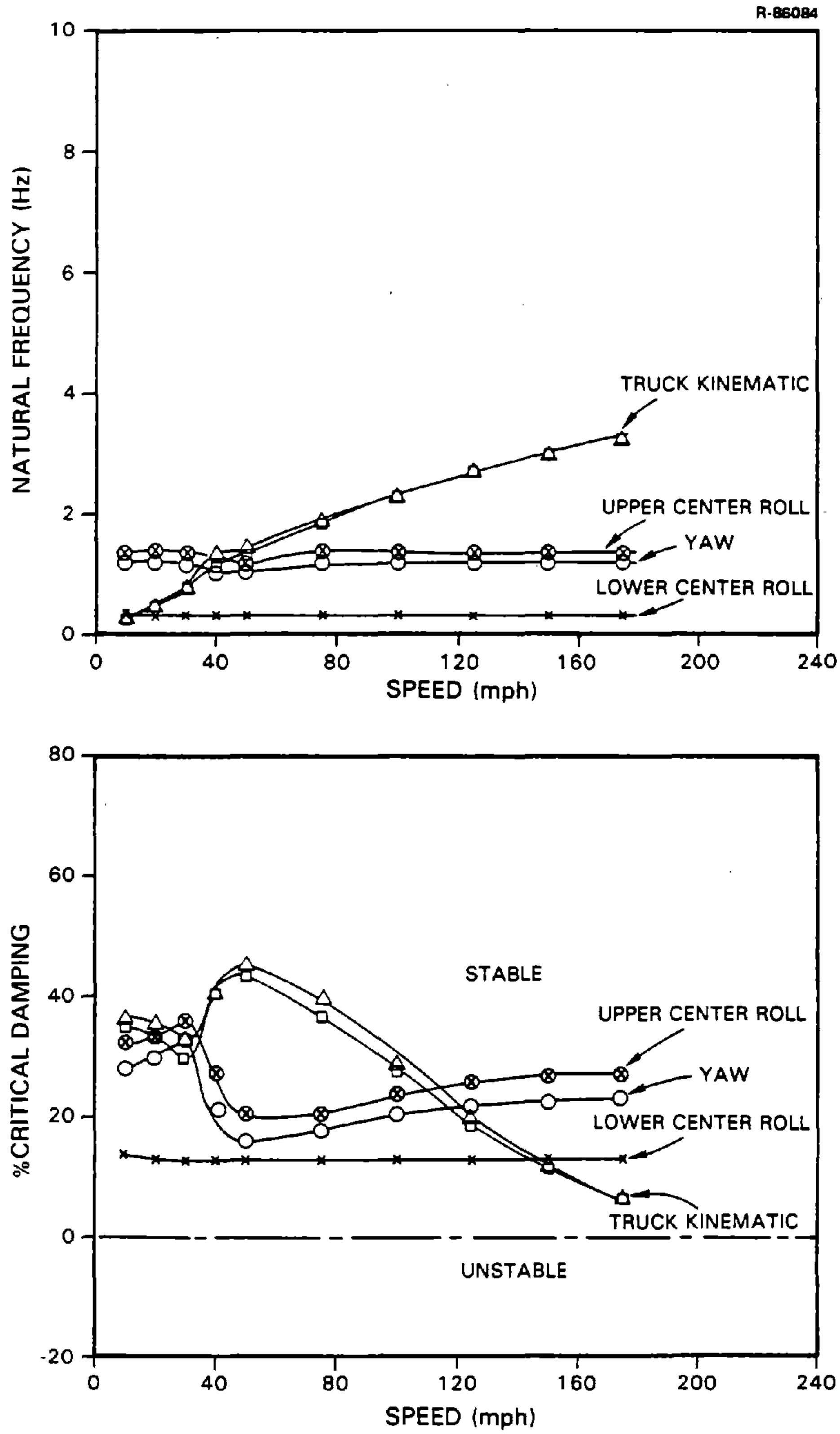


Figure 4.2-4 Natural Frequency and Damping Against Speed for Primary Longitudinal Equals Lateral Stiffness = 25,000 lb/in. (0.05 Conicity; Half Kalker)

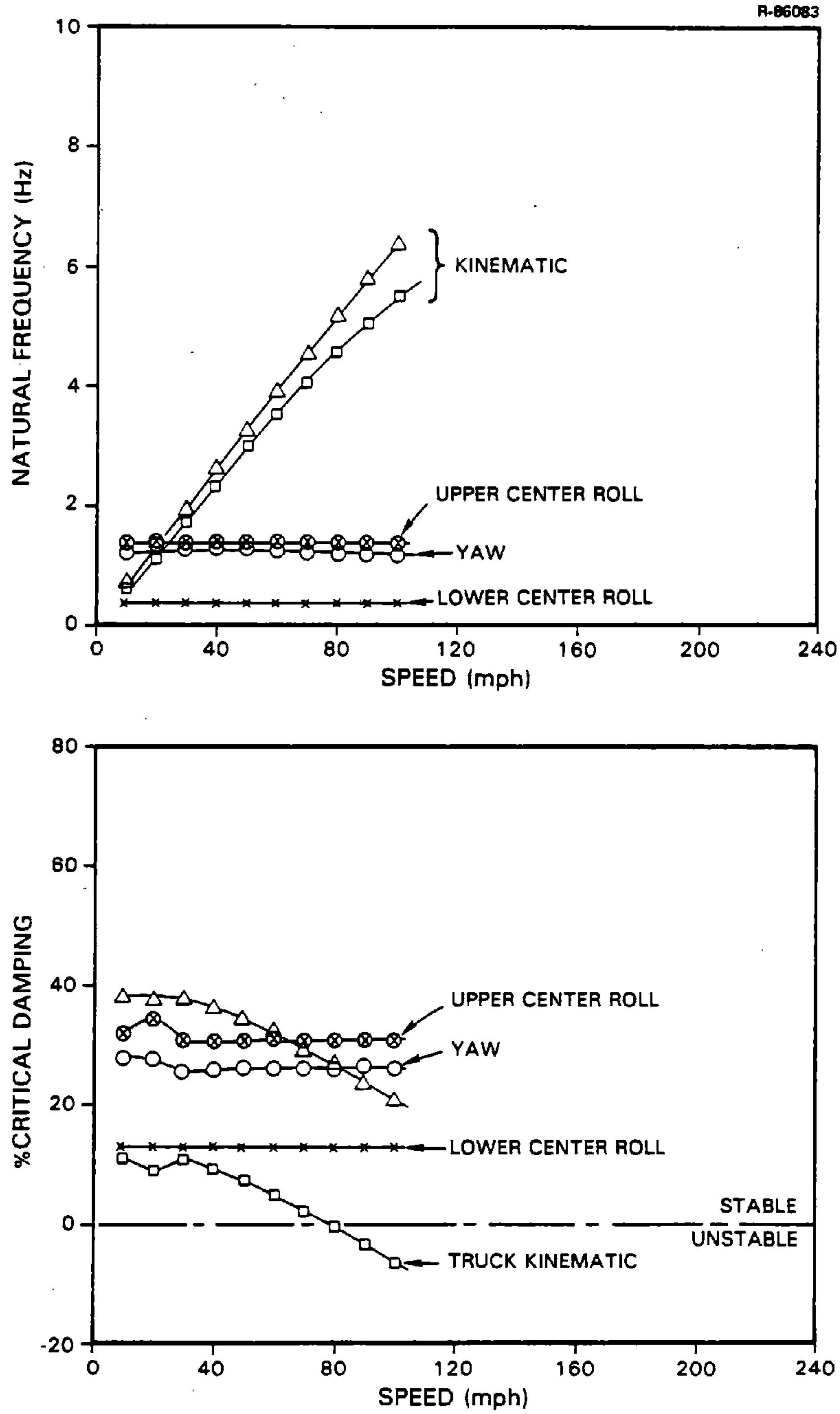


Figure 4.2-5 Natural Frequency and Damping Against Speed for Primary Longitudinal Equals Lateral Stiffness = 25,000 lb/in. (0.2 Conicity; Full Kalker)

R-82710

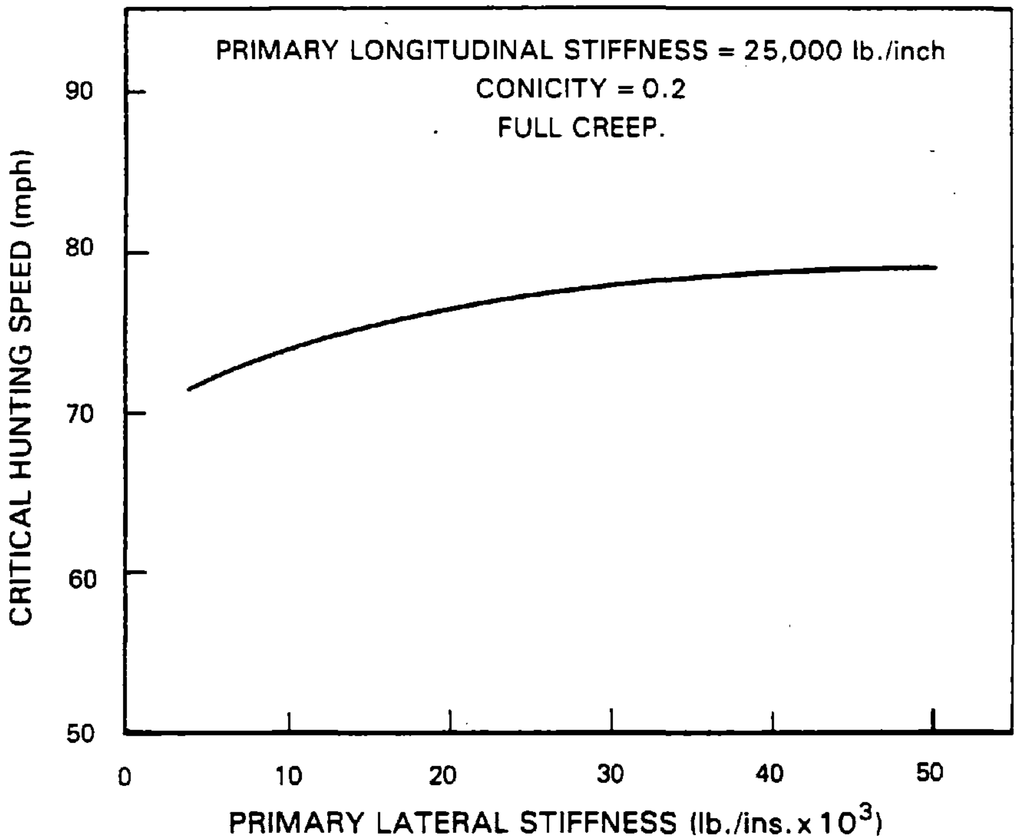


Figure 4.2-6   Effect of Primary Lateral Stiffness on Hunting Speed

4.3 STEADY STATE CURVING STUDY

The mathematical model used for this part of the study is based on work described in detail previously (Refs. 6 and 7). It is an extension of the linear curving theory produced almost simultaneously by Boocock (Ref. 15) and Newland (Ref. 16). Linear curving theory was found to be accurate for most vehicles only on very large radius curves. On smaller radius curves, the nonlinearities arising from two main causes were found to have very significant effects on vehicle curving behavior. The two areas of concern were the wheel/rail contact geometry, which becomes highly nonlinear once contact occurs in the root of, or on the flange, and Kalker's simple linear creep force/creepage relationships, which are valid only for small creepages (Ref. 17).

The vehicle parameters used in this study are contained in Table 3.2-2. Design case profiles for the wheels and rails were used in the study as measured profiles were not available. Six different wheel profiles were used:

- AAR Cylindrical (Fig. 1-1)
- AAR 1 in 20 (Fig. 1-3)
- 1 in 10 taper with same flange as AAR 1 in 20 (Fig. 4.3-1)
- 1 in 5 taper with same flange as AAR 1 in 20 (Fig. 4.3-2)
- 1 in 3 taper with same flange as AAR 1 in 20 (Fig. 4.3-3)
- British Rail 1 in 20 taper (Fig. 1-2).

The British Rail 1 in 20 tapered wheel was included as it had been tested during the Phase I test program conducted at WMATA by TSC. In addition, WMATA has introduced this profile on a small number of their fleet of cars.

The rail profile was a design case AREA 115RE rail, Fig. 4.3-4, which is the rail cross-section used on the WMATA system. The rails were inclined at 1 in 40, which is the inclination provided by the standard baseplates.

Track curvatures in the range 0-10 degrees were used in the study. This covered a range somewhat larger than was experienced during the track tests, where the maximum curvature was 7.6 degrees. The wheel/rail friction coefficient was assumed to be 0.5 throughout the study. Previous experience has shown that a coefficient of friction of 0.5 is typical of the values obtained on sharp curves when the rails are dry (Ref. 5). The majority of sharp curves on the WMATA system are in tunnels and, therefore, are always dry.

Results from the curving study are presented in the form of lead axle high and low rail lateral force, lead and trail axle yaw suspension displacement and lead and trail axle angle-of-attack with respect to the track.

Figures 4.3-5, 4.3-6, and 4.3-7 show predicted results for the standard vehicle with a British Rail 1 in 20 wheel profile as a function of track curvature. The effect of running at, above, and below balance speed are also shown. For the tightest curve on the test route, which is approximately 7.6 degrees of curvature, a lateral force of approximately 5,000 lbs is predicted at balance speed. For the same condition the lead axle angle-of-attack is approximately 6.5 milliradians or 0.37 degrees.

The effect on the standard vehicle of changing the wheel profile from the standard AAR cylindrical wheel to various tapers is shown in Figs. 4.3-8, 4.3-9, and 4.3-10. The predictions indicate that wheel tread conicity will have a relatively small effect upon the vehicles curving behavior. In terms of high rail lateral force, a minimum is obtained for most curvatures with a wheel tread conicity of between 0.1 and 0.2. However, as the figures show there is very little predicted effect due to wheel tread taper. This result agrees with the results obtained during the test program on the MARTA cars at the TTC. During those tests, the effect of a change from the standard AAR 1 in 20 to a CN-A worn wheel profile was tested. Although the CN-A profile gives an effective conicity, when contacting the tread, of approximately 0.15, compared with the 0.05 of the AAR 1 in 20 profile, very little difference in vehicle curving behavior was measured.

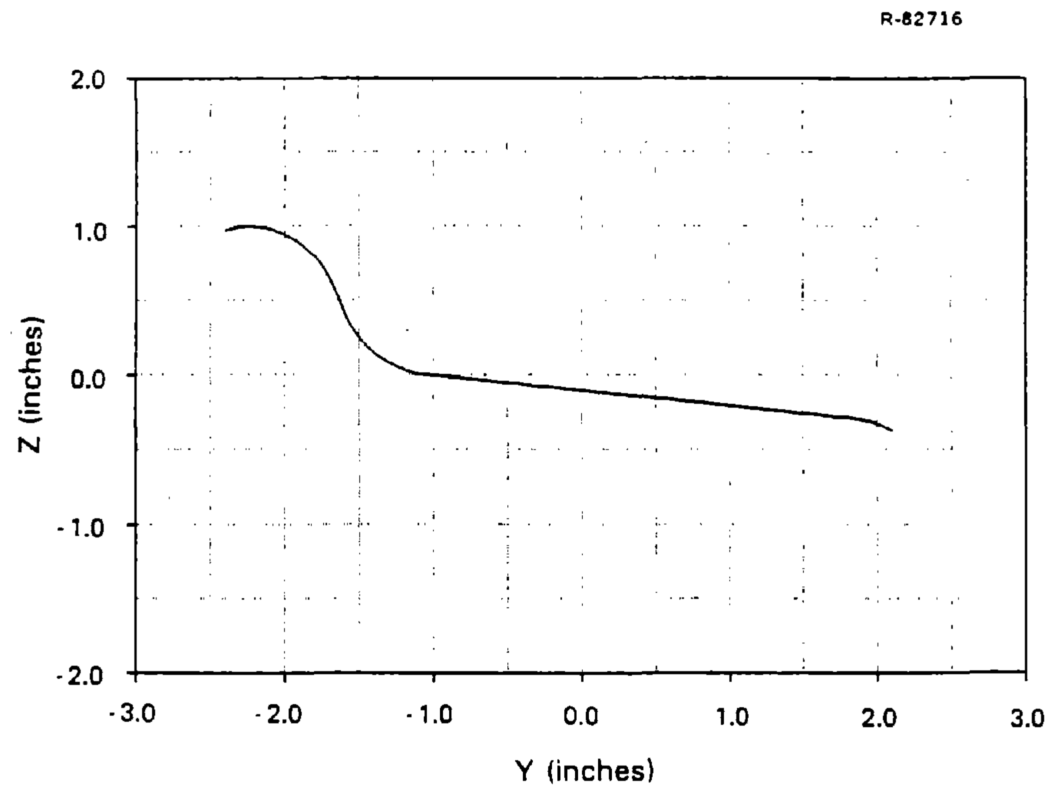


Figure 4.3-1 1 in 10 Profile with AAR Flange

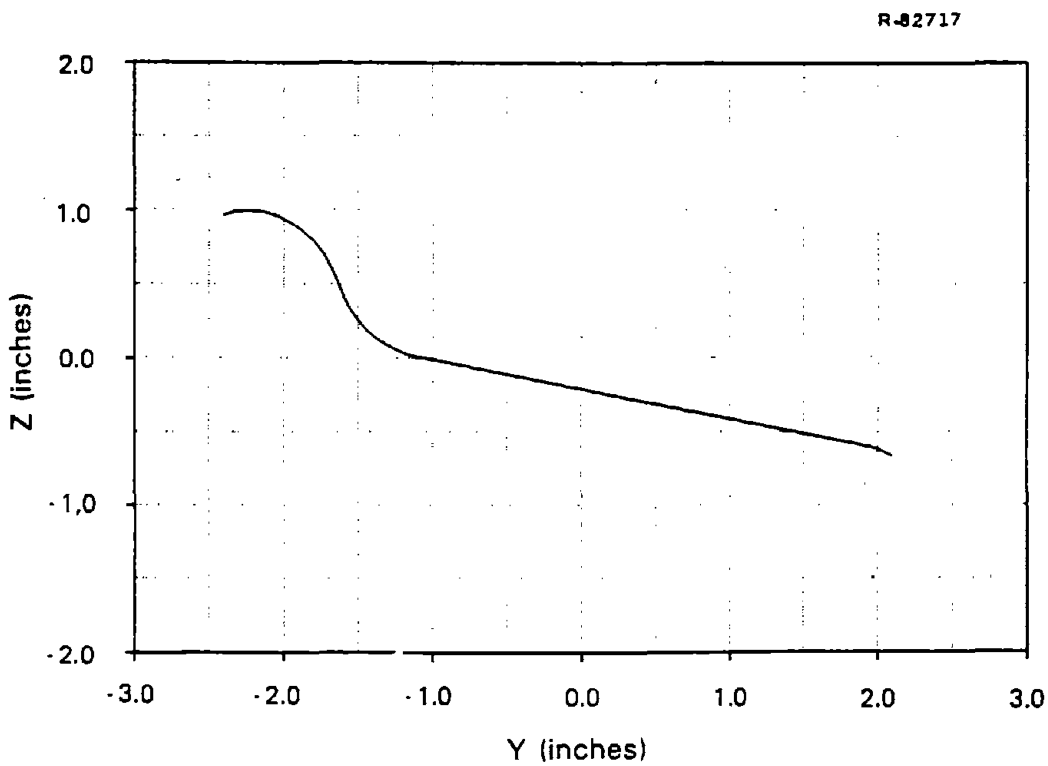


Figure 4.3-2 1 in 5 Profile with AAR Flange

R-82714

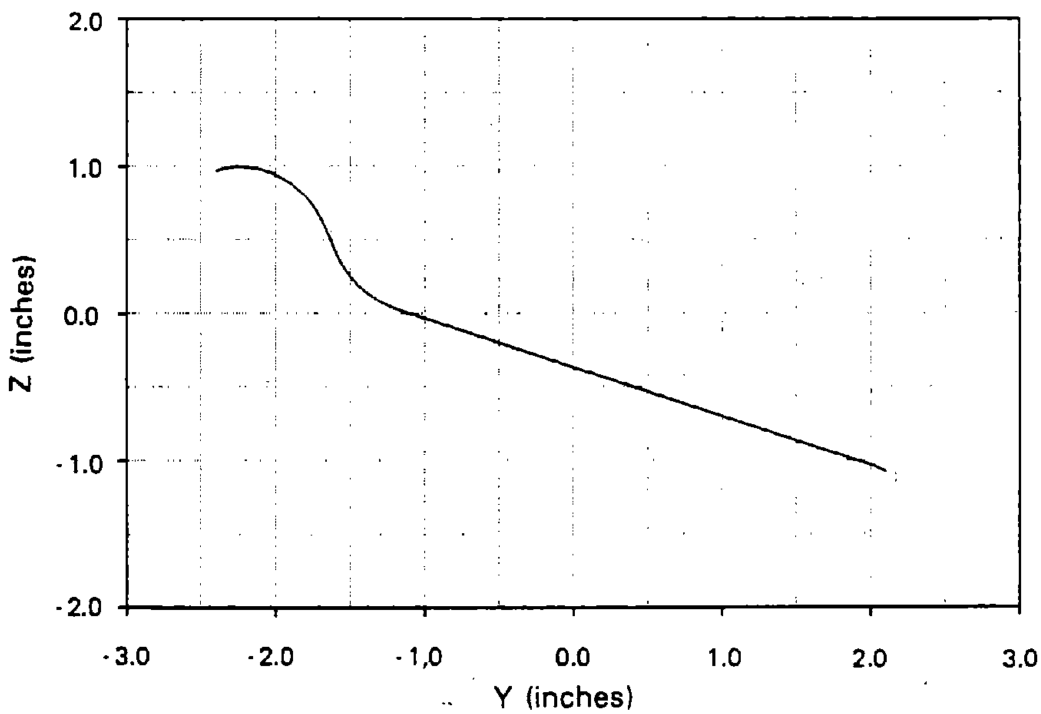


Figure 4.3-3 1 in 3 Profile with AAR Flange

R-82715

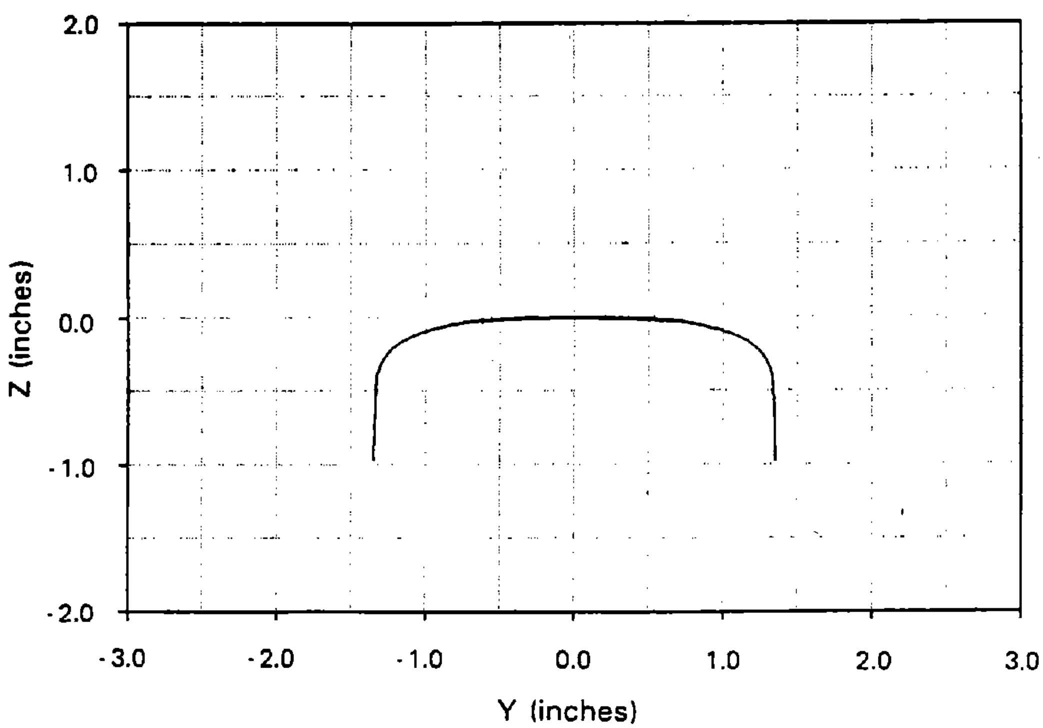
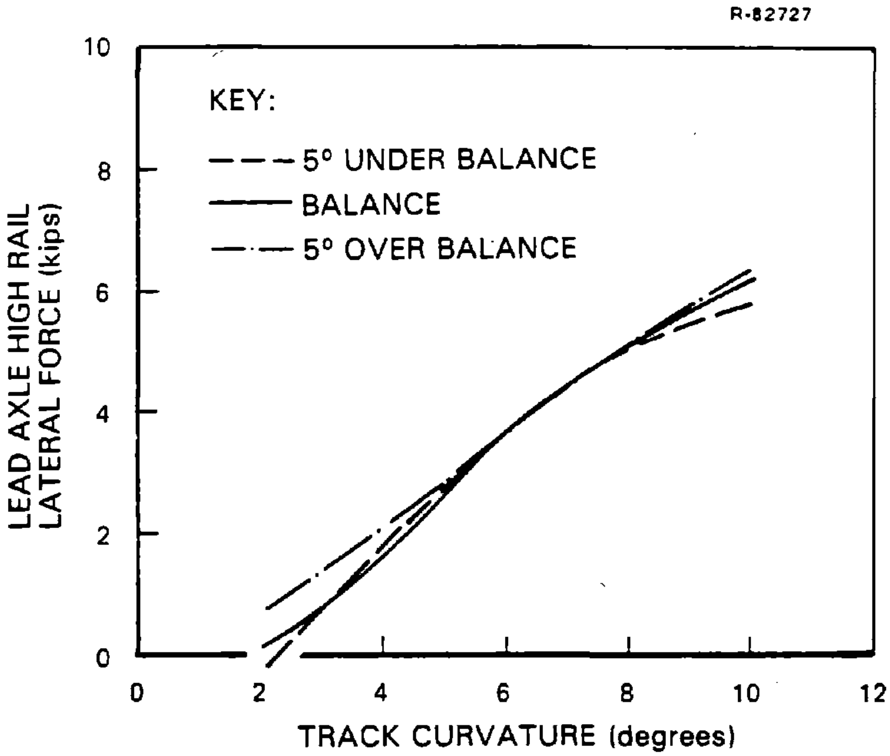


Figure 4.3-4 Area 115RE Rail Profile



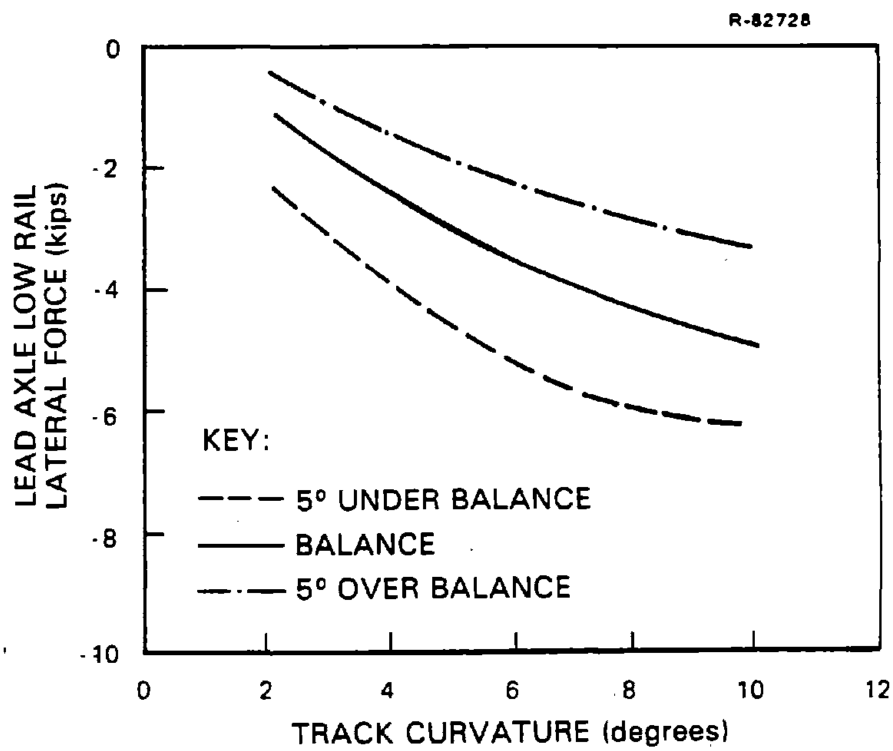


Figure 4.3-5 Lead Axle Wheel/Rail Lateral Forces Against Track Curvature for Standard Vehicle (BR 1 In 20 Profile; $\mu = 0.5$)

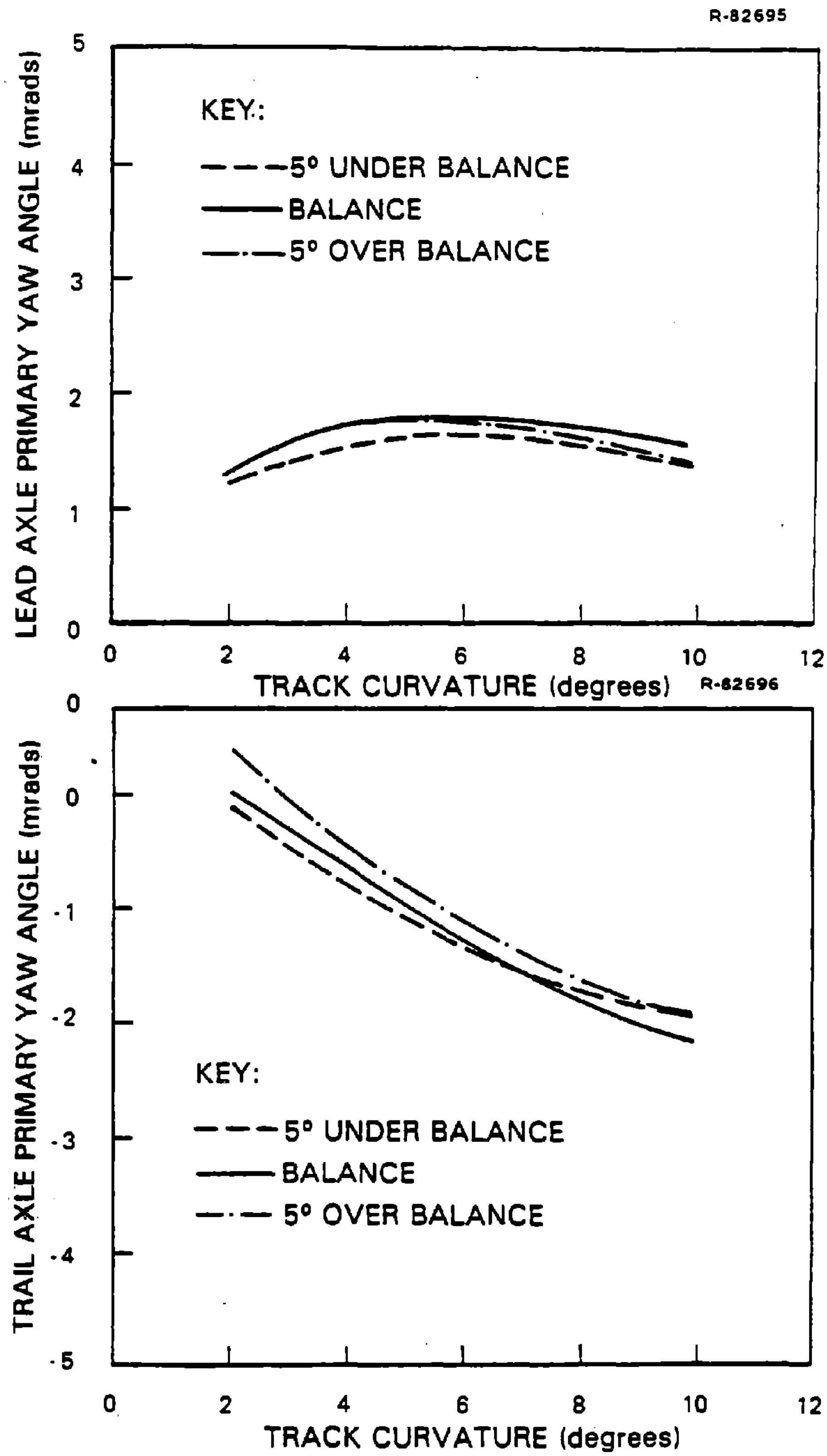


Figure 4.3-6 Primary Suspension Yaw Angles Against Track Curvature for Standard Vehicle (BR 1 In 20 Profile; $\mu = 0.5$)

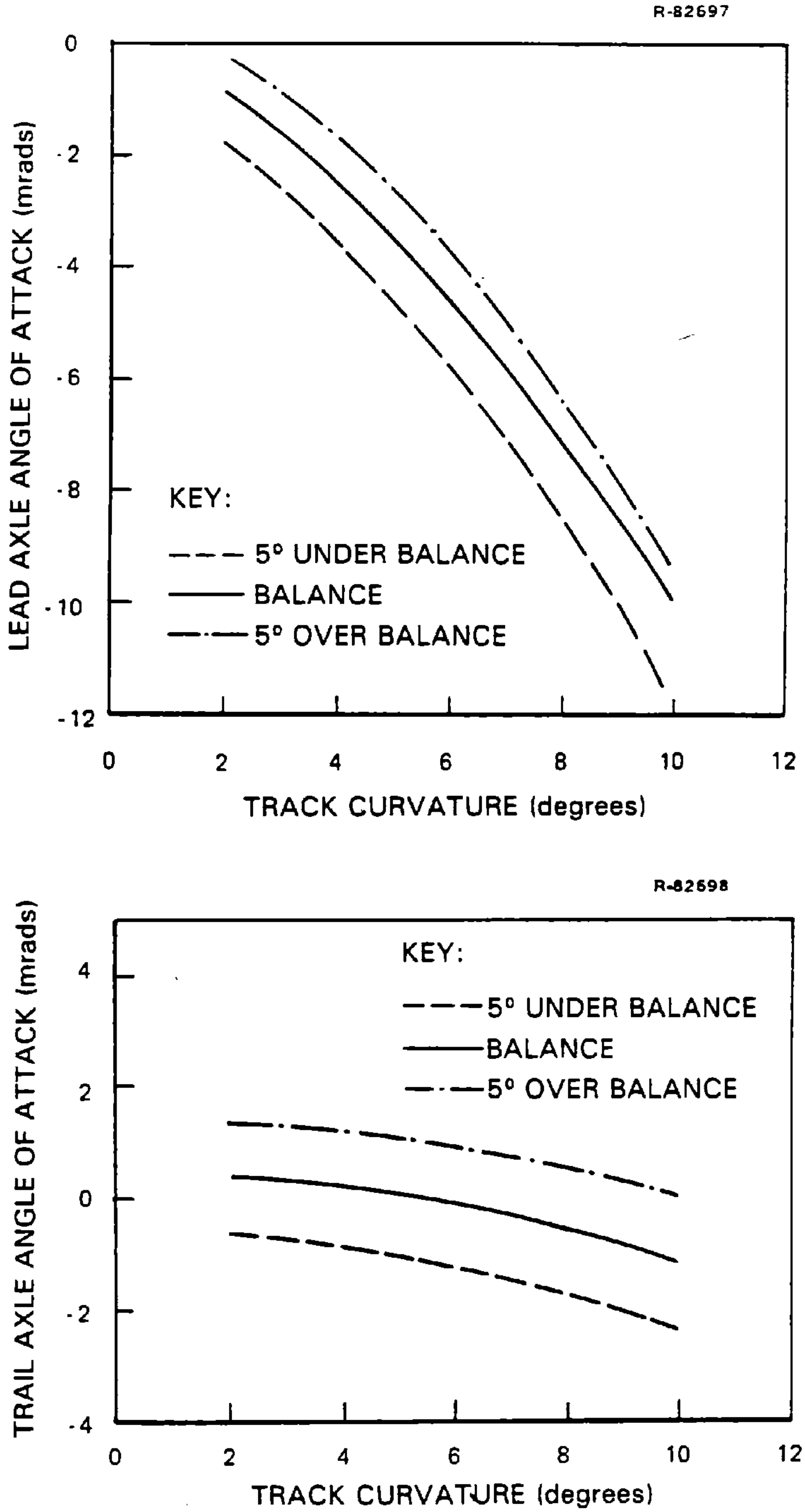


Figure 4.3-7 Axle Angles of Attack Against Track Curvature for Standard Vehicle (BR 1 In 20 Profile; $\mu = 0.5$)

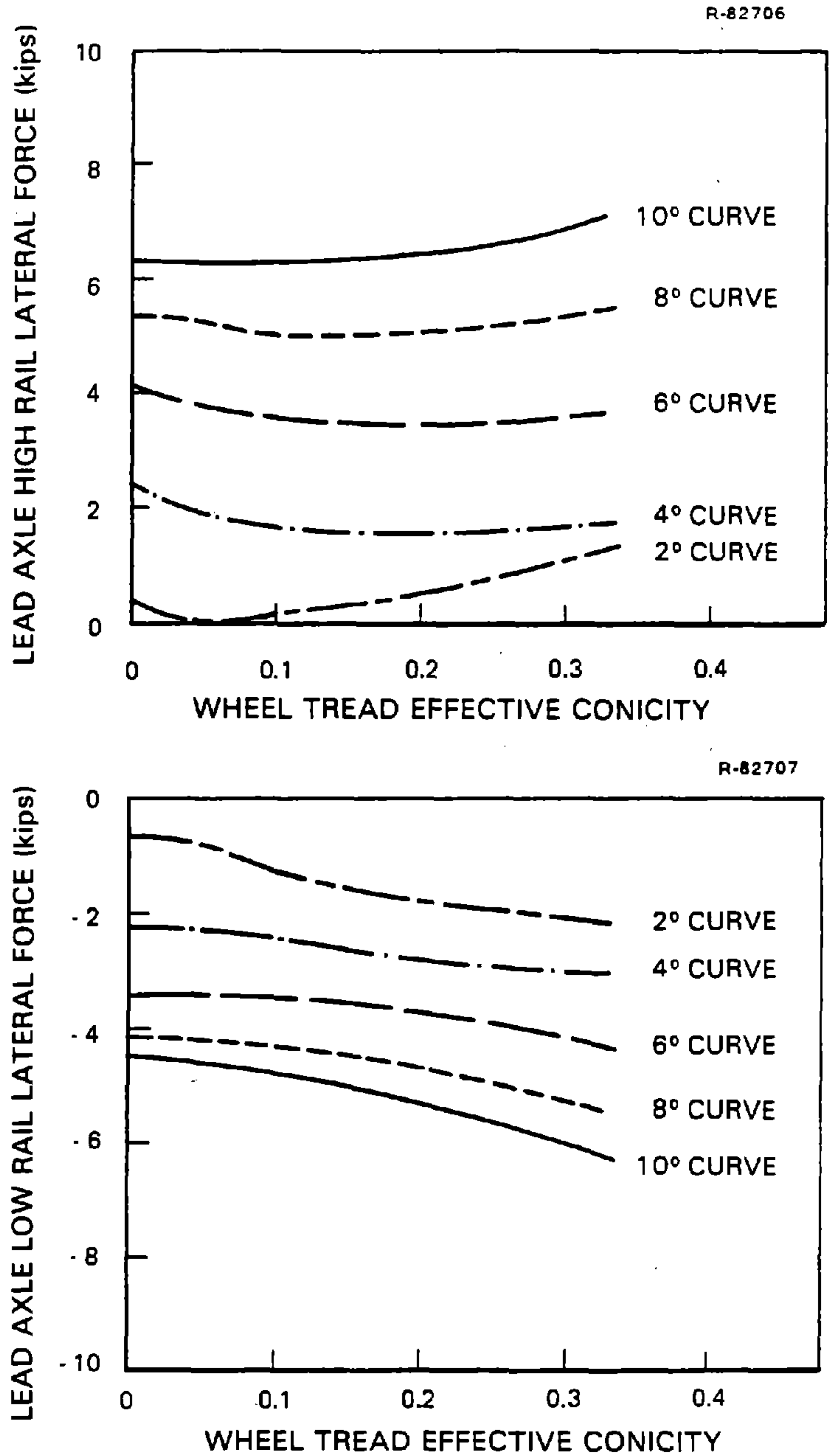
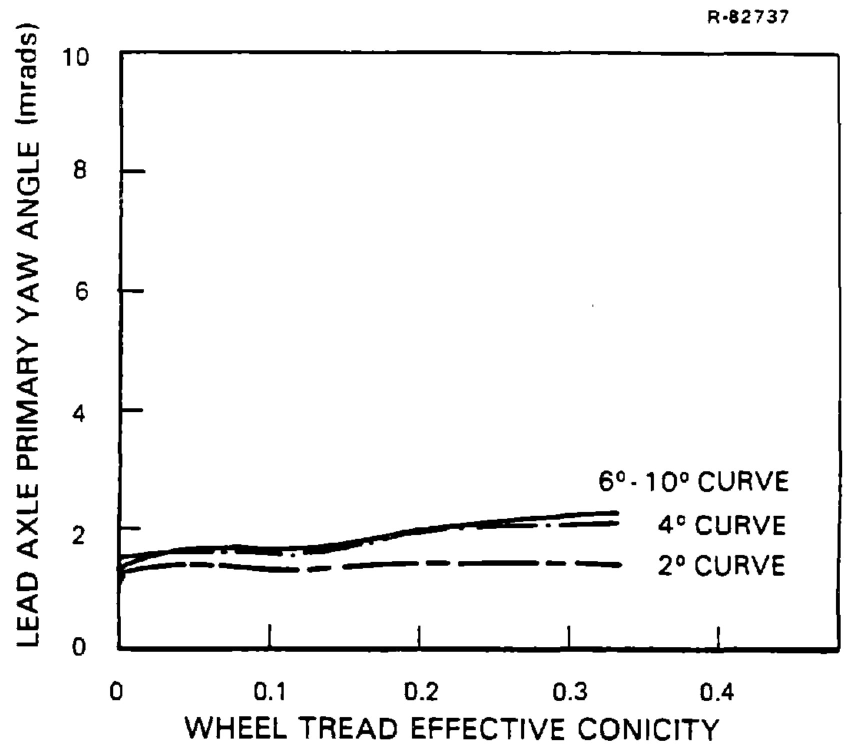


Figure 4.3-8 Lead Axle Wheel/Rail Lateral Forces Against Wheel Tread Effective Conicity for Standard Vehicle (μ = 0.5; Balance Speed)



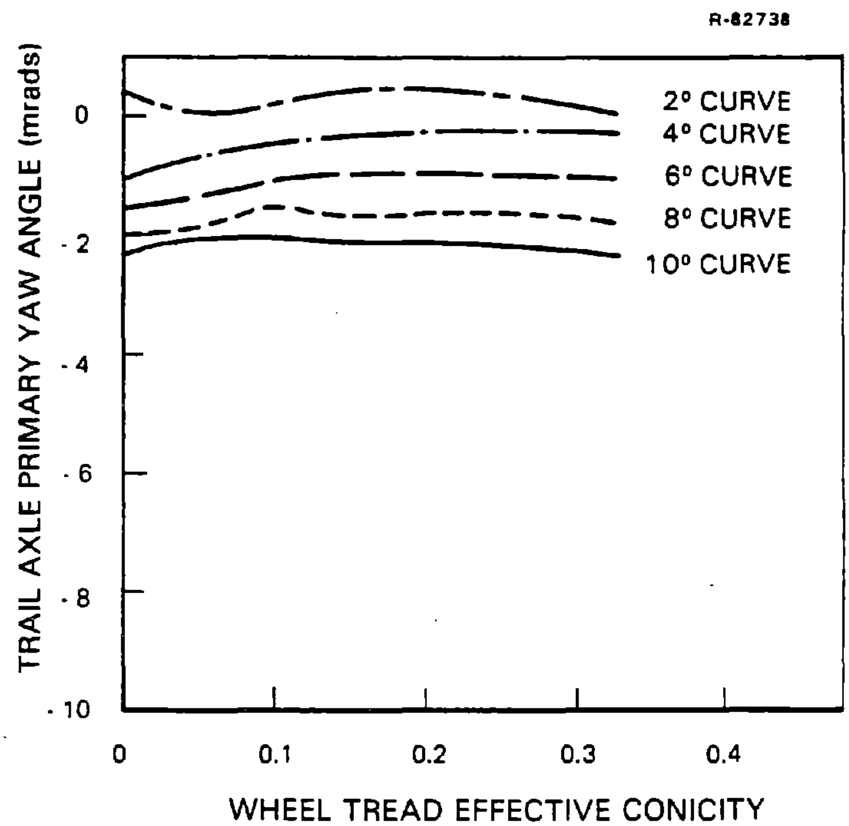


Figure 4.3-9 Primary Suspension Yaw Angles Against Wheel Tread Effective Conicity for Standard Vehicle ($\mu = 0.5$; Balance Speed)

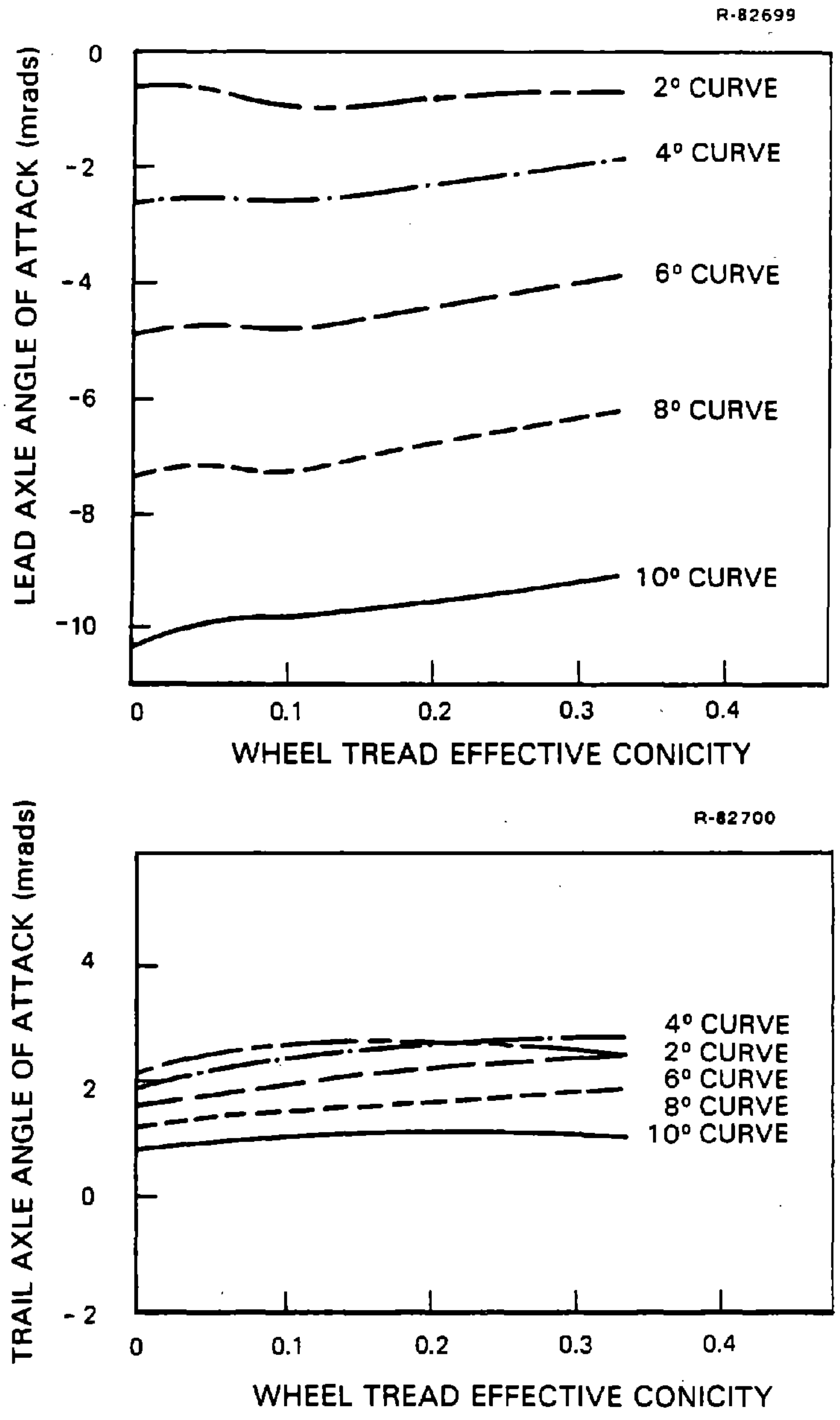


Figure 4.3-10 Axle Angles of Attack Against Wheel Tread Effective Conicity for Standard Vehicle (μ = 0.5; Balance Speed)

Figures 4.3-11, 4.3-12, and 4.3-13 show the effect of changing the primary longitudinal stiffness. For this study, the lateral stiffness of the primary suspension was assumed to be equal to the longitudinal stiffness and a wheel tread conicity of 0.1 was used. This being a value somewhere near the middle of the range that had been planned for the test program. It can be seen that reducing the primary longitudinal stiffness has a very dramatic effect upon the predicted curving behavior.

The lateral stability study reported in Section 4.2 has shown that the longitudinal stiffness may be reduced to approximately 25,000 lb/in. with a modest stability margin. With this value of stiffness the predicted curving results indicate a high rail lateral force of 1,800 lb for the 7.6° curve. For this case the lead axle yaw displacement is 4.8 mrads, which would result in a longitudinal displacement of 0.109 in. across the primary suspension bush.

These predictions agree with the test results obtained with the MARTA vehicle at the TTC, which showed the very substantial reductions in wheel/rail forces and angles-of-attack that could be obtained from a reduction in primary longitudinal stiffness.

Finally, Figs. 4.3-14, 4.3-15, and 4.3-16 show the predicted effect of changing wheel tread taper with a primary longitudinal stiffness of 25,000 lb/in. As with the standard value of longitudinal stiffness, the predictions show very little change in curving behavior with varying wheel tread conicity.

As a result of the lateral stability and curving study, it was decided that a value of 25,000 lb/in. would be specified for the longitudinal stiffness of the modified primary suspension bush. Of course, this specification would be revised if any practical limitations were discovered during the design study which follows.

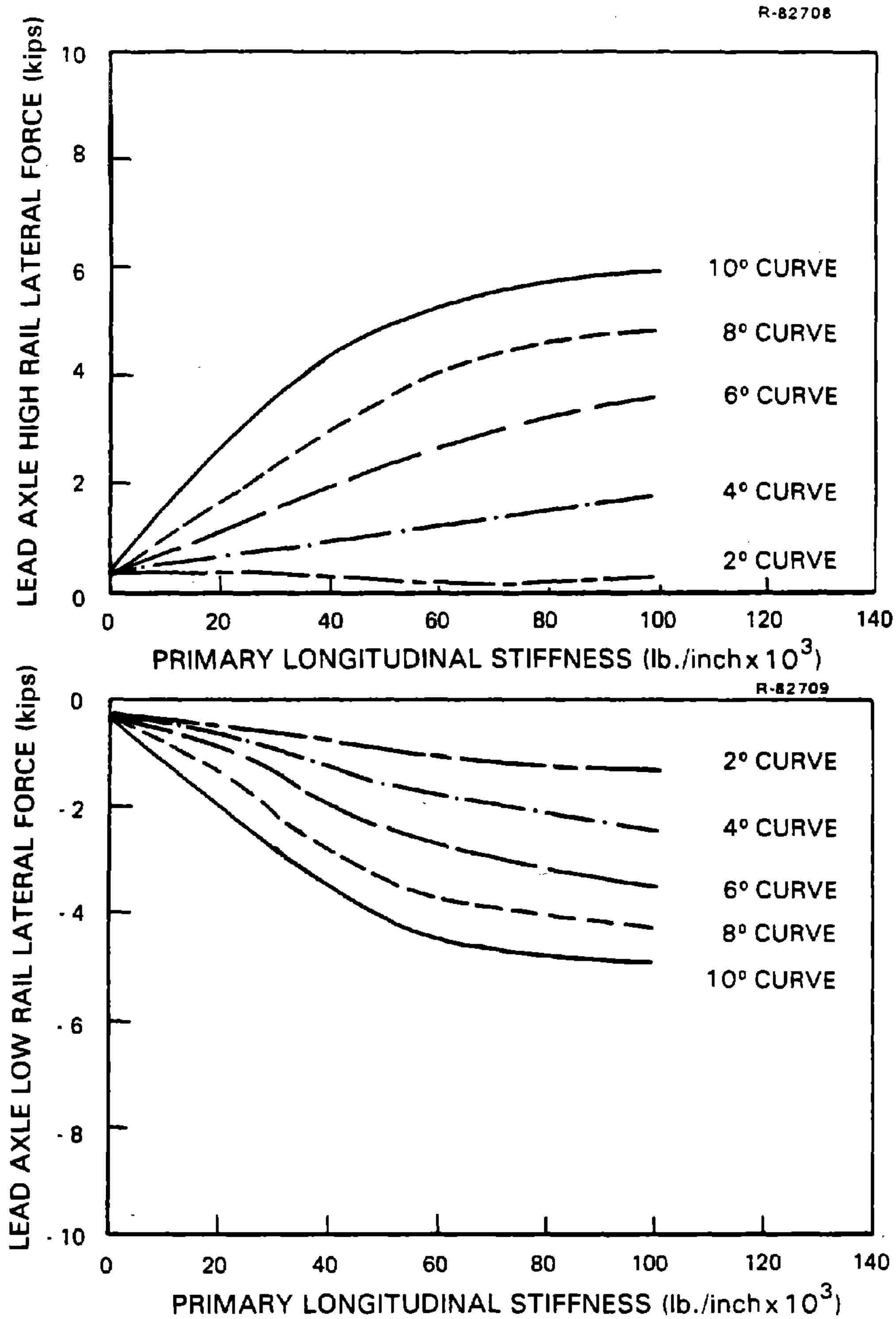
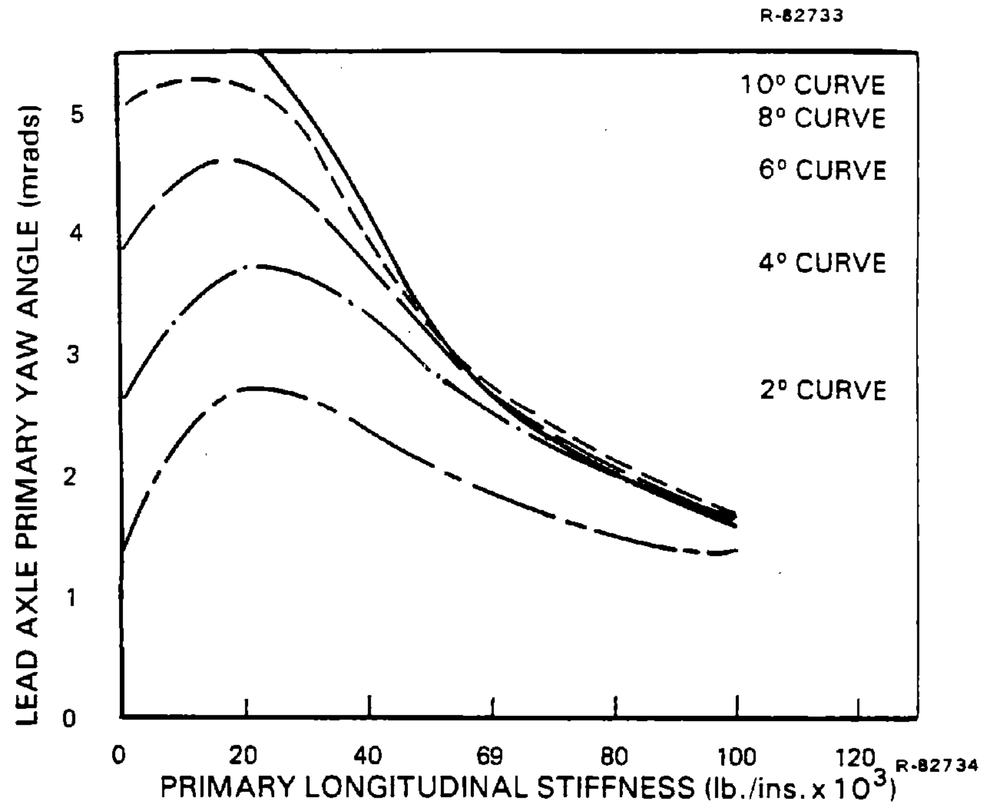


Figure 4.3-11 Lead Axle Wheel/Rail Lateral Forces Against Primary Longitudinal Stiffness for Standard Vehicle (μ = 0.5; Balance Speed)



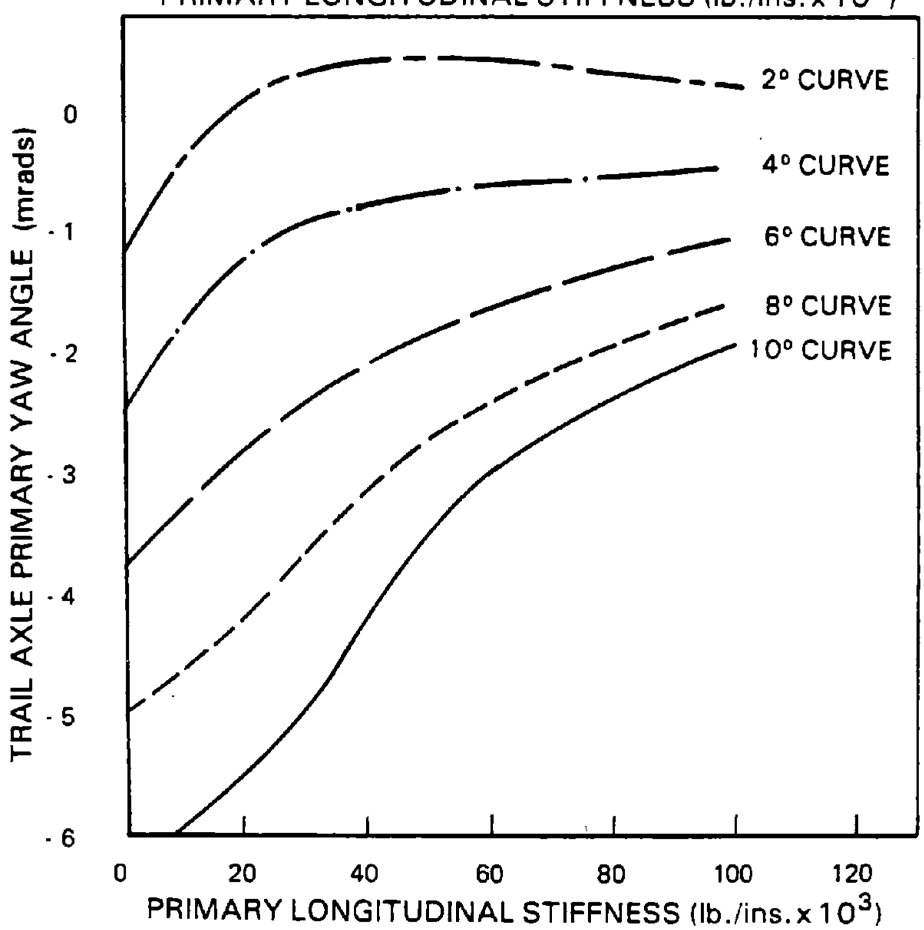
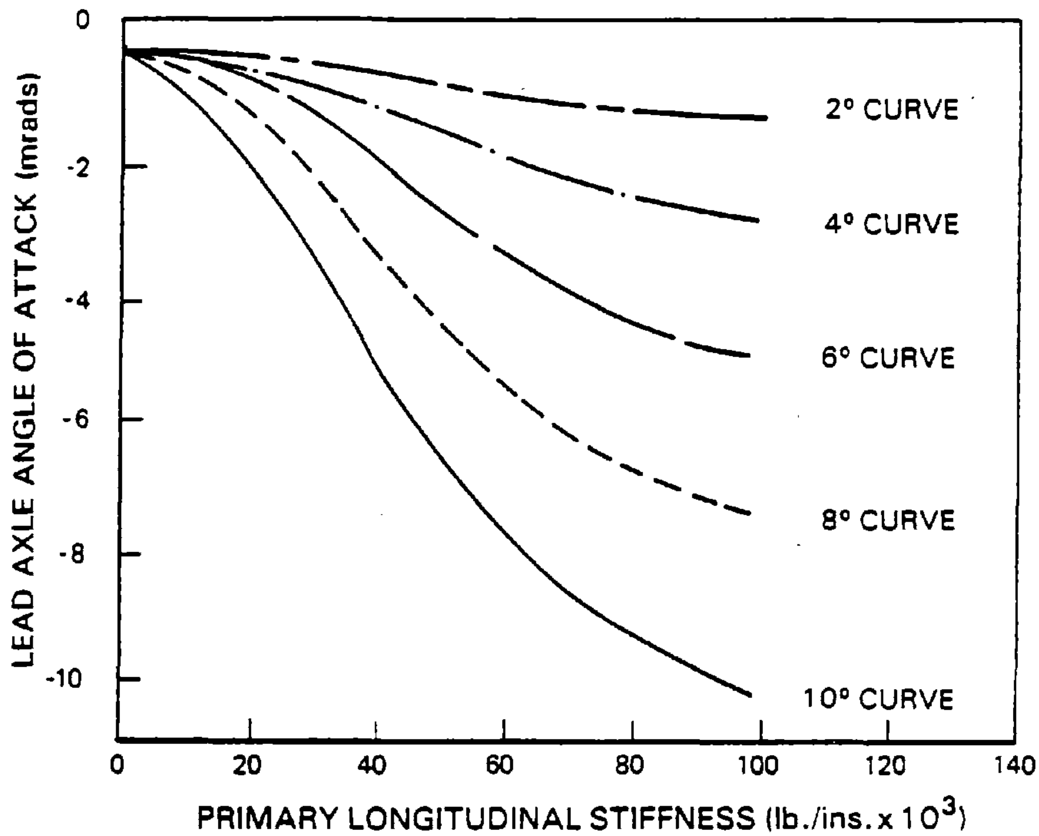


Figure 4.3-12 Primary Suspension Yaw Angles Against Primary Longitudinal Stiffness for Standard Vehicle (μ = 0.5; Balance Speed)

R-82731



R-82732

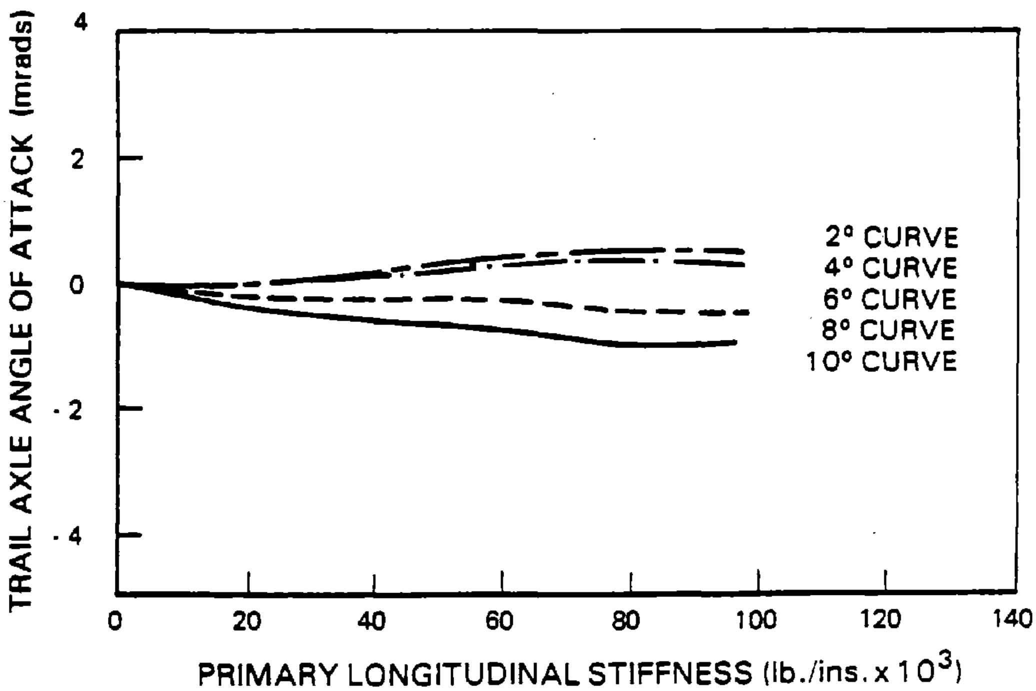


Figure 4.3-13 Axle Angles of Attack Against Primary Longitudinal Stiffness for Standard Vehicle (μ = 0.5; Balance Speed)

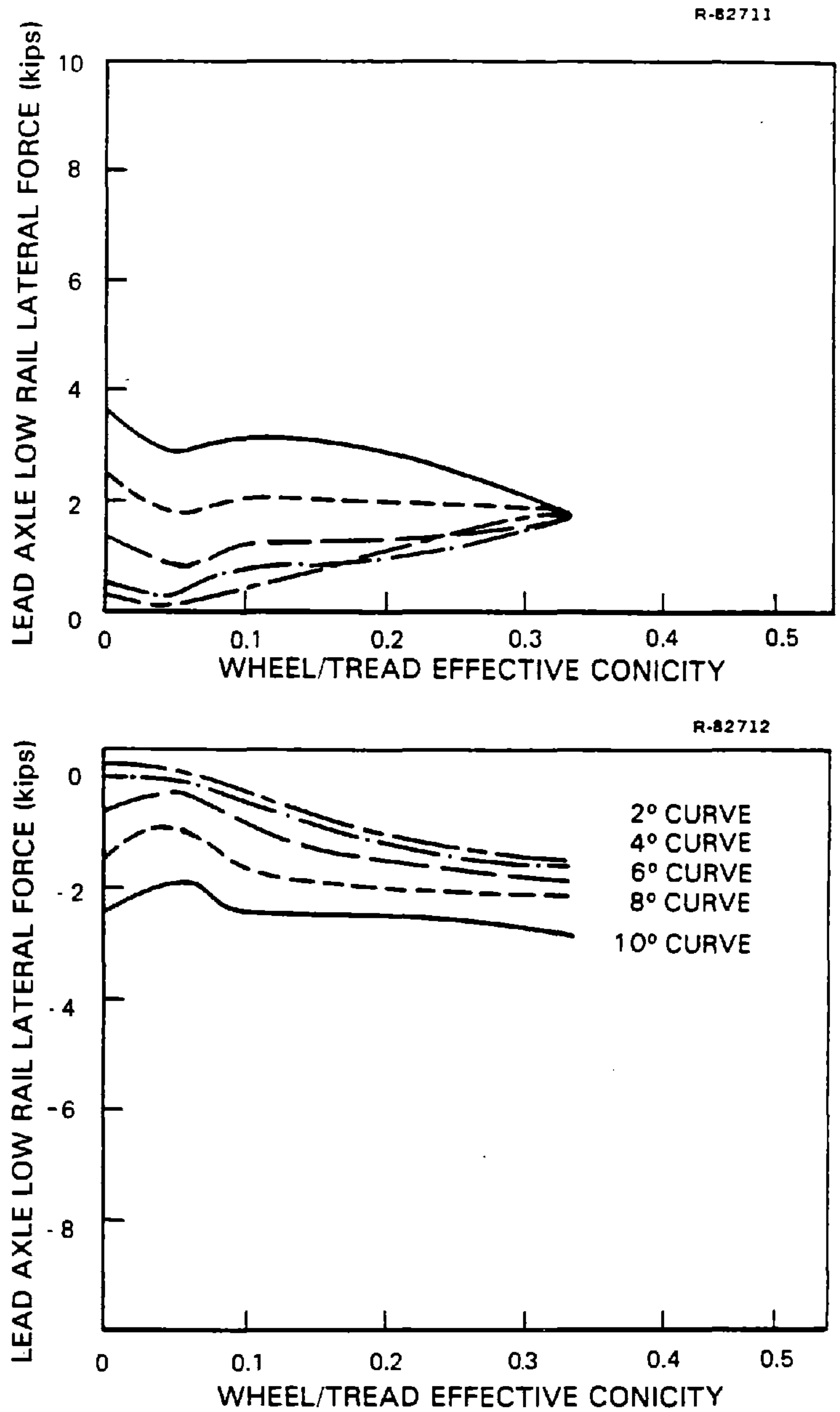


Figure 4.3-14 Lead Axle Wheel/Rail Lateral Forces Against Wheel Tread Effective Conicity (Primary Longitudinal Stiffness = 25,000 lb/in., $\mu = 0.5$; Balance Speed)

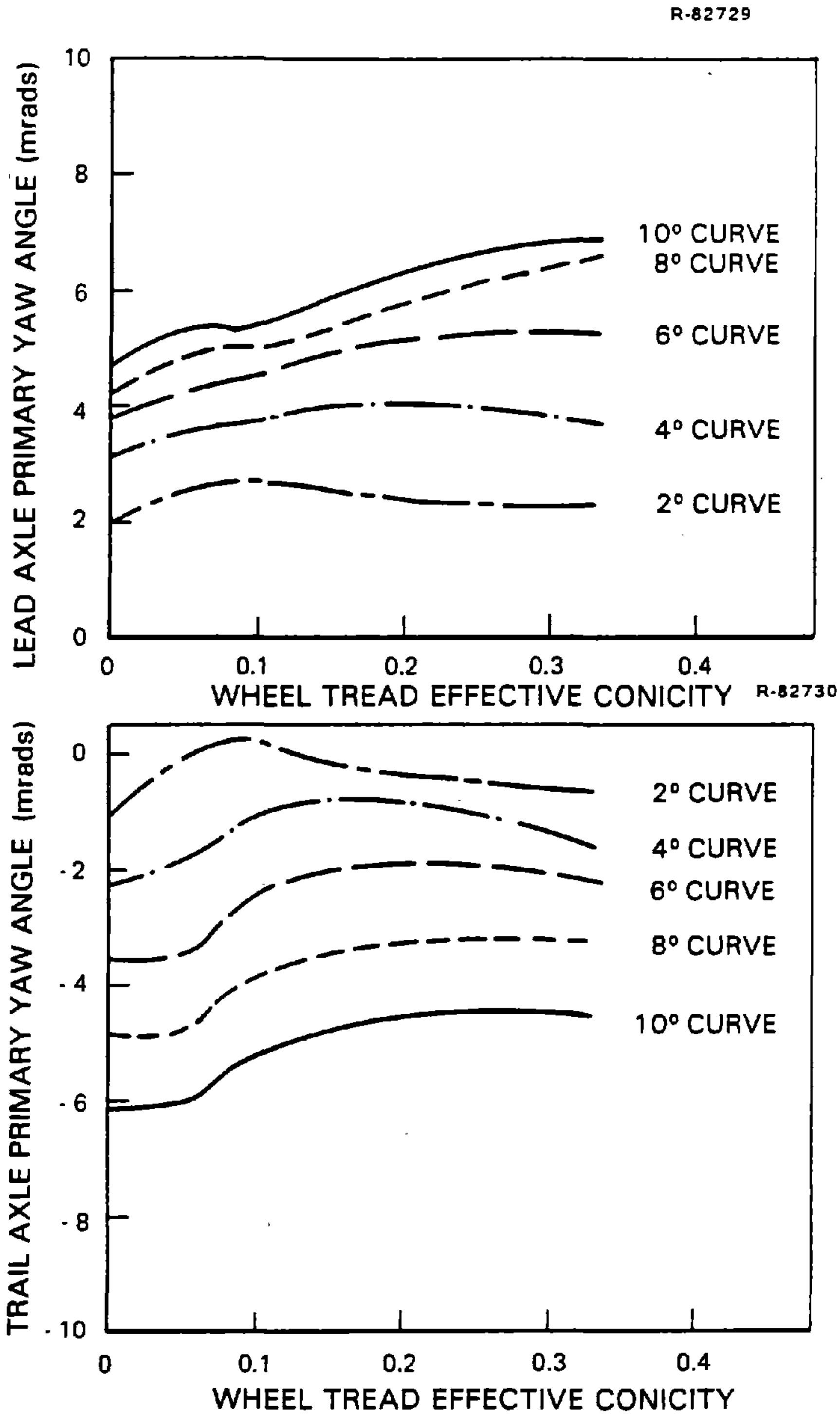
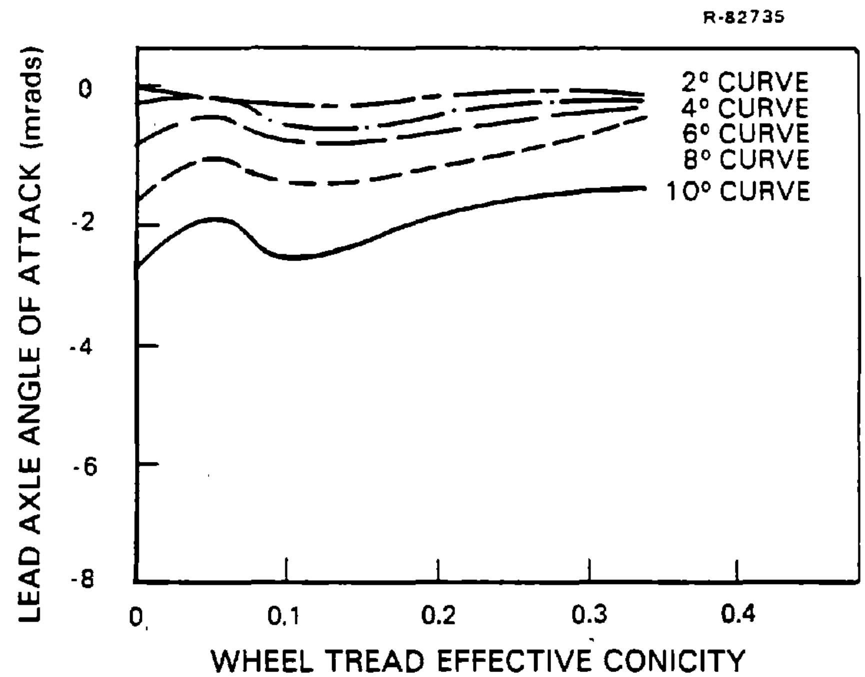


Figure 4.3-15 Primary Suspension Yaw Angles Against Wheel Tread Effective Conicity (Primary Longitudinal Stiffness = 25,000 lb/in.; $\mu = 0.5$, Balance Speed)



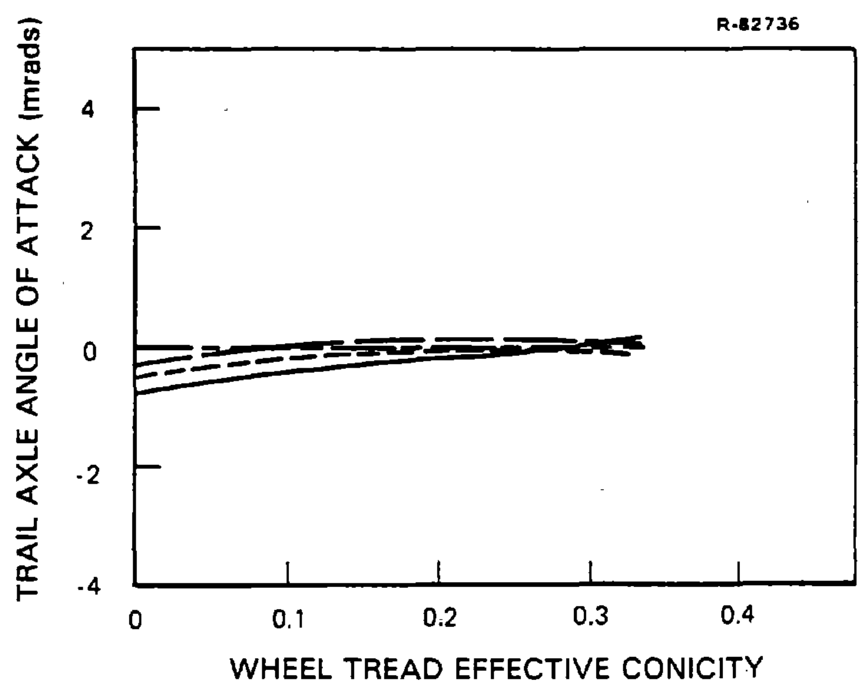


Figure 4.3-16 Axle Angles of Attack Against Wheel Tread Effective Conicity (Primary Longitudinal Stiffness = 25,000 lb/in.; μ = 0.5, Balance Speed)

5. DESIGN STUDY TO IDENTIFY FEASIBLE TRUCK MODIFICATIONS

5.1 PREVIOUS EXPERIENCE WITH MARTA

The Rockwell trucks used on the WMATA vehicles are articulated for wheel load equalization, each truck side frame having a transverse transom arm which terminates in a ball joint assembly in the opposing side frame (Fig. 5.1-1). Primary suspension is provided by two rubber spring halves which surround each journal bearing and are retained by the frame journal housing and a journal cap (Fig. 5.1-2). Secondary suspension is provided by two air spring units positioned between the truck side frames and a bolster. The bolster is restrained longitudinally by two radius rods to the truck frame and laterally by a rubber bumpstop.

The principal modification to the truck, that was envisaged for this test program, was a reduction in stiffness of the primary suspension bushing. The analytic results given in Section 4 indicate that a reduction in the longitudinal stiffness of the primary suspension is very beneficial in reducing curving forces.

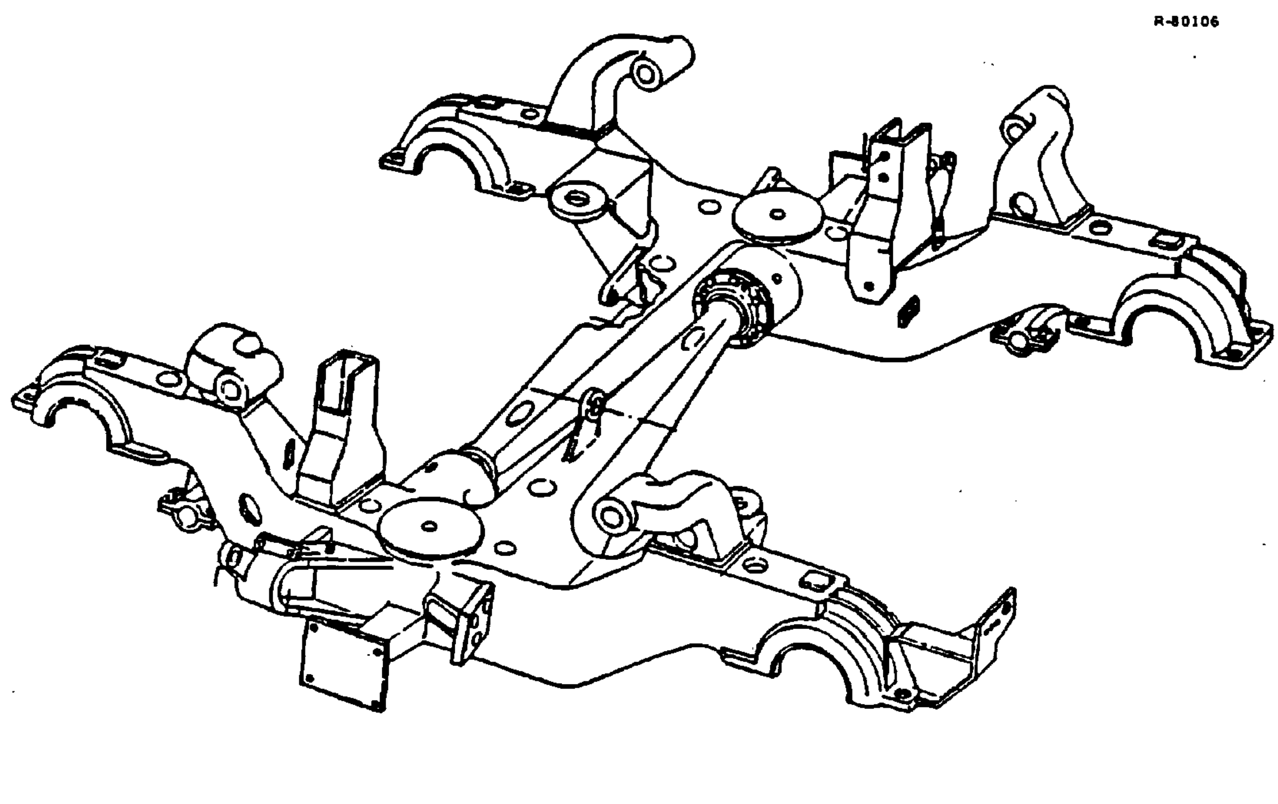


Figure 5.1-1 Rockwell Truck Frame

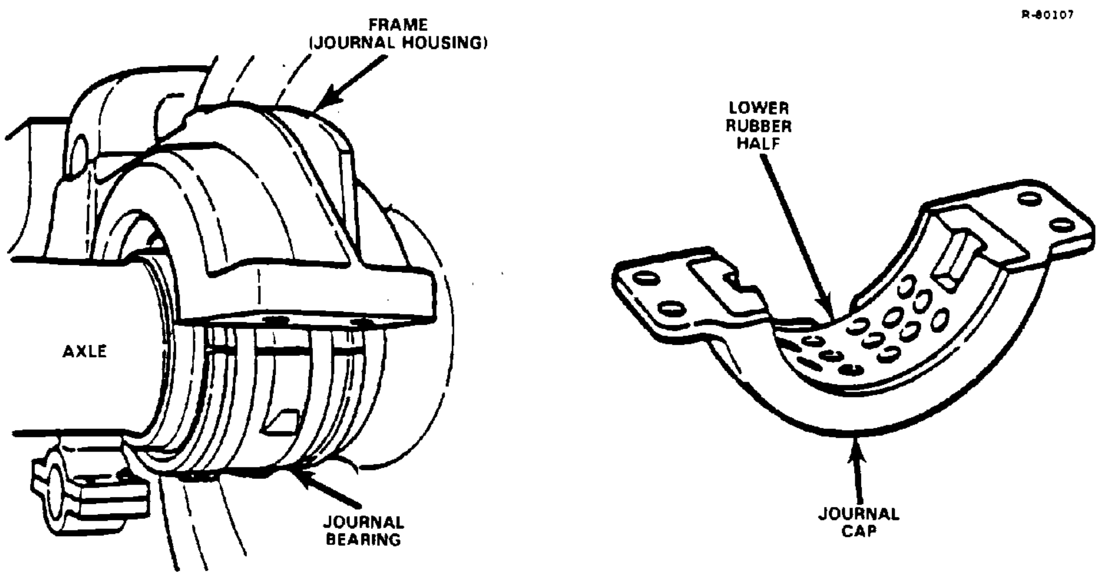


Figure 5.1-2 Axle Journal Housing and Primary Rubber Bushing

There are two limitations on the extent of the reduction in stiffness that is possible. The first of these concerns the braking system on the Rockwell truck of the MARTA vehicles. This consisted of a friction tread brake in which a single brake shoe is pushed against the tread of each wheel by a hydraulic actuator. A brake application results in a longitudinal displacement of the axle and, owing to a limitation on the stroke of the brake actuator, a minimum value for the longitudinal stiffness is implied.

The second limitation is associated with the requirement that the existing bushing should be modified in order to obtain the reduction in stiffness. A completely redesigned bush was considered to be outside the scope of the test program. The reduction in stiffness was achieved by removing rubber from an area near the horizontal axis of the bushing. The value of 50,000 lb/in., that was obtained by this process, appeared to be near the limit of what could be obtained by a modification to the existing bushing.

In the case of the Phase II test program on WMATA, neither of these limitations applied. Although the WMATA truck is also manufactured by Rockwell, and is identical in many respects to the MARTA truck, it has a disc brake system which does not exert a longitudinal force on the axle when the brake is applied. In addition, it was decided from the outset that a new design of bushing, fitting within the existing journal housing, would be considered if it was necessary in order to obtain the desired value of longitudinal stiffness.

5.2 PRACTICAL LIMITATIONS ON PRIMARY LONGITUDINAL STIFFNESS

The results of the analytic study described in Section 4 had indicated that the optimum value of longitudinal stiffness for the WMATA vehicle would be approximately 25,000 lb/in. Previous experience with MARTA had shown that when modifications were confined to the existing bushing, 50,000 lb/in. seemed to be near the practical minimum. Therefore, it appeared that a new design of bushing would be required, in order to achieve the anticipated stiffness.

Accordingly, discussions were held with a number of companies who have experience in the manufacturer of molded rubber components The purpose of these discussions being to determine a practical range for the primary stiffnesses, for a bushing which fitted within the existing journal housing.

The existing primary suspension consists of a pair of molded rubber half cylinders. Each of the halves is cut away, to some extent, in order to effect a reduction in the vertical stiffness. Inspection of bushes that had been in service for some time, indicated that considerable creep of the rubber was taking place (Fig. 5.2-1). This was particularly evident with the upper halves of the bushing, which have a number of holes through the rubber and carry the majority of the static vertical load. It was the opinion of all of the manufacturers consulted, that the standard bushing was rather highly stressed and that this was the reason for the high rate of creep being observed in service. In addition, any attempt to remove rubber from the existing bush in order to reduce the longitudinal stiffness would increase the stress due to the vertical load and result in even higher rates of creep.

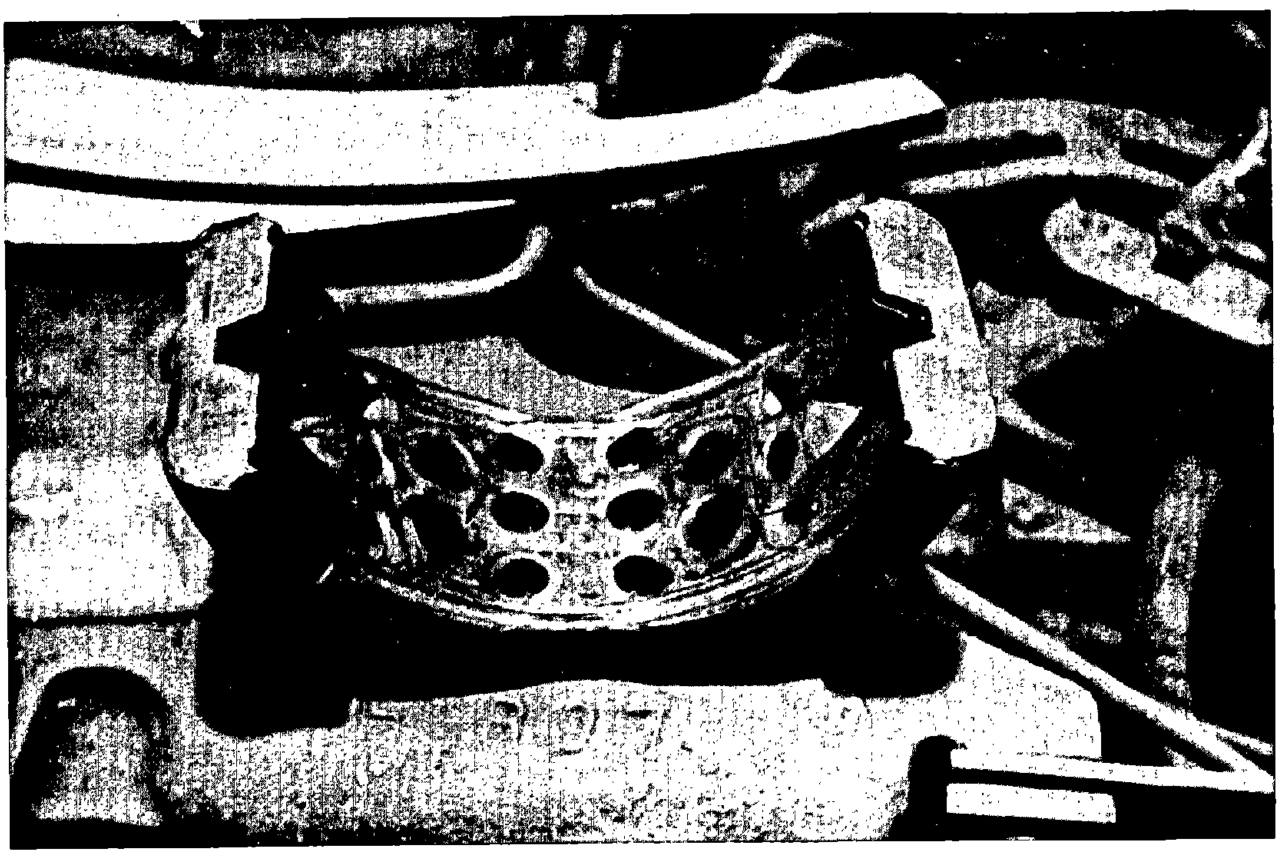


Figure 5.2-1 Upper Half of Primary Suspension Bush Showing Rubber Creep

In order to reduce the level of stress due to the vertical load and, therefore, provide scope for a reduction in longitudinal stiffness, a bonded rubber bushing was proposed. In this arrangement, the rubber is bonded onto a number of steel plates. In its simplest form, this consists of an inner and outer steel shell with the rubber bonded in between. However, one manufacturer suggested that additional steel plates bonded into the rubber might be required.

With the simplest form of bonded bushing, it was anticipated that a minimum longitudinal stiffness of approximately 25,000 lb/in. could be achieved. A vertical stiffness of 100,000 lb/in. was assumed. This being the approximate design value for the standard bushing.

As preliminary estimates had indicated that the optimum value for longitudinal stiff was approximately 25,000 lb/in., it was decided that the simpler form of bonded bushing would be adequate.

5.3 PRACTICAL LIMITATIONS ON AXLE TO TRUCK DISPLACEMENTS

The large reduction in longitudinal stiffness being contemplated was expected to lead to significant increases in displacement between the wheelsets and the truck frame. Even though the primary suspension of the WMATA truck is not subjected to the force of the brake actuator, substantial longitudinal forces are generated at the wheel/rail interface under certain operating conditions and these have to be reacted by the primary bushing.

The maximum longitudinal acceleration experienced by the vehicle will be during emergency braking. Under this condition a maximum longitudinal force of approximately 2000 lb will be exerted on each bushing. However, under conditions of steady curving, the longitudinal steering forces generated at the wheel/rail interface can be sufficiently large for a force of approximately 4000 lb to be exerted on each bushing.

It was necessary to establish whether adequate clearances existed within the truck to accommodate the axle to truck displacements that would occur under these loadings.

Initially, it was thought that a limitation on longitudinal travel would be imposed by clearances between the journal housing and components surrounding the journal bearing. However, a trip was made to the WMATA Maintenance Shop at Brentwood, to check at first hand on available clearances. As a result of this visit, it became apparent that there was adequate clearance in the region of the journal housing but that clearances in the braking system would be the limiting factor.

When new, adequate clearance exists between the brake caliper and the wheel hub on which the brake disc is mounted. However, owing to creep of the primary suspension bushing under load (Fig. 5.3-1) and wear in the linkage that supports the brake caliper, this clearance is considerably reduced in practice. Nevertheless, just sufficient clearance (approximately 0.3 in.) was available and as creep was expected to be less with the modified primary bushings, this provided some additional effective clearance.

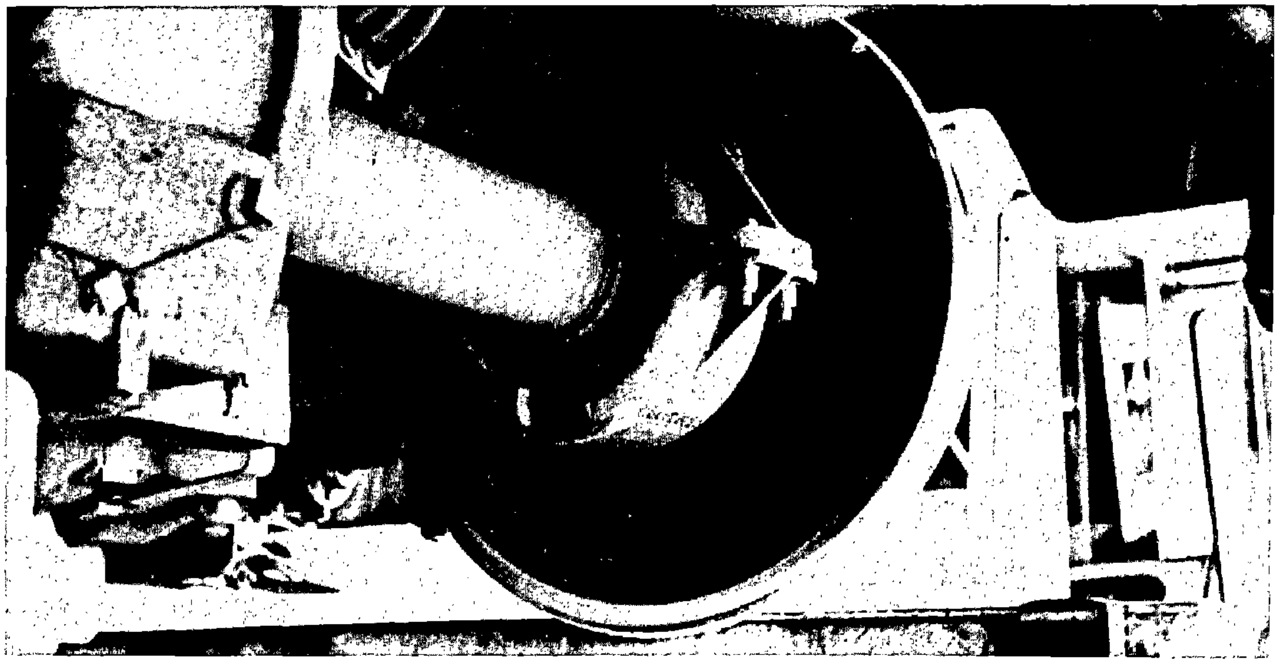


Figure 5.3-1 Side View of Journal Housing Showing Considerable Creep and Deflection of Primary Bush

Vertical creep of the bushings causes the brake caliper to be offset vertically from the centerline on the disc, as can be seen in Fig. 5.3-1. This has the effect of reducing the longitudinal clearance available, as the caliper is concentric with the wheel hub when the primary suspension is unladen.

An additional consideration concerns the angular displacement in yaw, to which the traction motor drive is subjected, during the negotiation of sharp curves. Preliminary calculations had indicated that for a longitudinal stiffness of 25,000 lb/in., the maximum angular displacement of the axle with respect to the motor would be approximately 5 milliradians or 17 minutes of arc. During the negotiation of track that has been maintained to normal standards, angular misalignments of the coupling in roll will occur. These will be of a larger amplitude than the 5 milliradian yaw misaligument that is predicted. Therefore, it seems unlikely that the additional yaw misalignment, occurring as a result of a primary longitudinal stiffness reduction, will pose a problem.

5.4 DEVELOPMENT OF MODIFIED PRIMARY BUSH USED DURING THE TEST PROGRAM

As a result of the design study, a specification was prepared for a new design of primary bush with the following desired parameters:

- Longitudinal stiffness = 25,000 lb/in.
- Lateral stiffness > 25,000 lb/in.
- Vertical stiffness = 100,000 lb/in.
- Creep of the bush, under the normal vertical loading, to be less than 0.1 inches over the 2 month period of the test program.

The Budd Company was selected to supply bushes which met the desired specification. They provided 10 sets of bushes, sufficient to equip the two trucks of the test vehicle plus two spares.

The bush arrangement consisted of a urethane elastomer bonded onto inner and outer steel sheels. The bonded urethane configuration being chosen to provide flexibility in achieving the desired stiffness parameters. The elastomer was cut away, in appropriate places near the vertical and horizontal axes of the bush, in order to achieve the desired values of stiffness in the three planes.

During the development of the modified bushes, stiffness tests were carried out using a special test rig built by TTC and used during the MARTA test program. The rig is illustrated in Section 10 of Ref. 4. With this rig, measurement of stiffness could be made in all three planes, but without the presence of the normal static vertical load. As a result, it was anticipated that stiffnesses would be higher when the bushes were installed in the truck.

Several bush configurations were tried in an attempt to achieve the desired longitudinal stiffness of 25,000 lb/in. The final configuration that was adopted achieved this stiffness without the static vertical load. A stiffness somewhat higher was anticipated when the bush was installed in the truck. However, this seemed to be near the limit of what could be achieved with this configuration of bush.

The values of stiffness obtained for the final configuration, without the static vertical load applied, are given in Table 5.4-1 along with the values that were measured with the bushes in the truck. The arrangement of the final bush is illustrated in Fig. 5.4-1.

TABLE 5.4-1
MODIFIED PRIMARY BUSH STIFFNESSES

PARAMETER	WITHOUT STATIC VERTICAL LOAD	WITH STATIC VERTICAL LOAD
Primary Vertical Stiffness	104,000 lb/in.	116,000 lb/in.
Primary Lateral Stiffness	26,000 lb/in.	32,000 lb/in.
Primary Longitudinal Stiffness	25,000 lb/in.	29,000 lb/in.

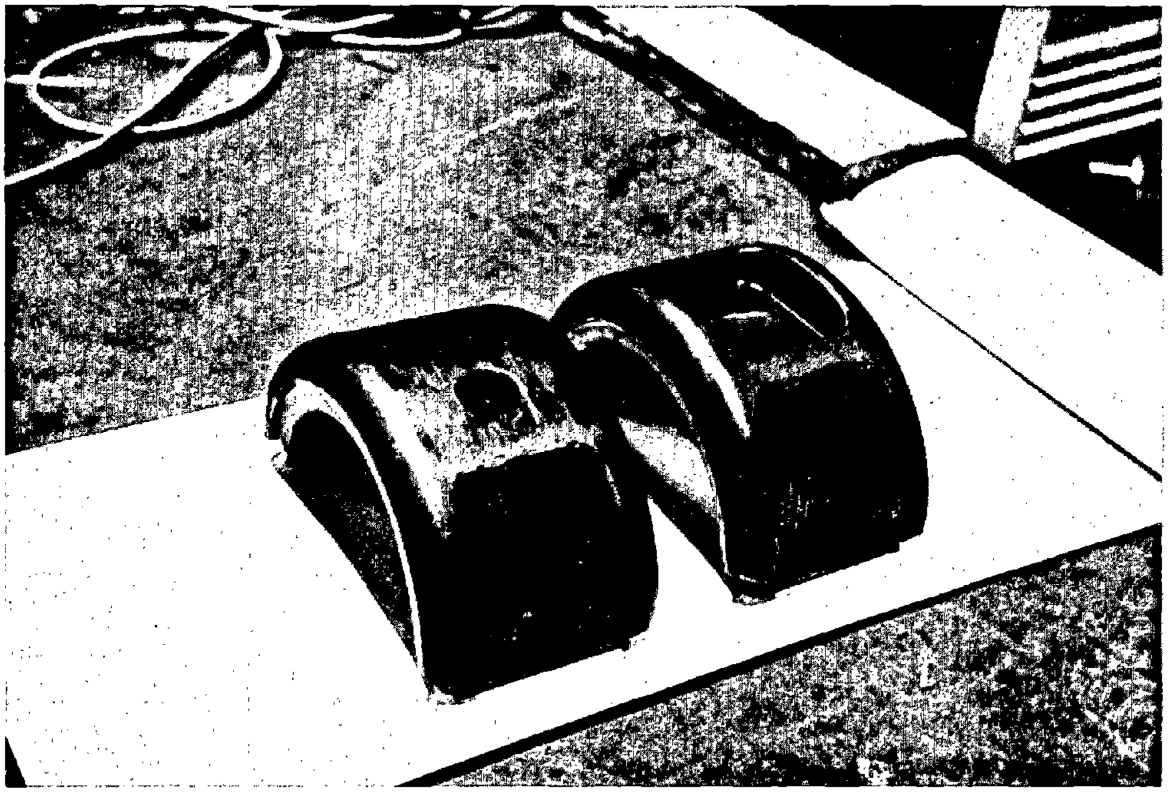


Figure 5.4-1 Modified Primary Bush

6. TRACK TESTS

6.1 GENERAL

The main purpose of the track test program was to assess the lateral stability and curving performance of the WMATA vehicle. This was to be carried out with the standard vehicle and with various modifications to the wheel profile and the primary suspension bush.

The complete instrumentation system and a quick-look analysis of the test results obtained from it will be described by ENSCO in a report covering their contribution to the program. Accordingly, this report will confine itself to a cursary description of the instrumentation used for evaluating curving performance and the data obtained from that instrumentation for the various vehicle configurations that were tested.

One truck was instrumented in order to measure the forces being exerted between wheel and rail on one axle, and to determine the attitude of the truck and axles in each of the test curves. A single instrumented wheelset was used, to give a continuous measure of vertical and lateral force on each of the two wheels. Two string potentiometers measured the lateral displacement between the truck frame and the car body, from which the truck yaw angle and lateral displacement with respect to the car body could be determined. Displacement transducers (LVDT type) were used to measure lateral and longitudinal displacements of the primary suspension. This permitted a determination of the yaw angles and lateral displacements of the axles with respect to the truck frame. The data from these transducers and the rest of the instrumentation was recorded on analogue tape and the output from selected channels was displayed on strip recorders.

The route over which the tests were performed was chosen to be suitable for both curving and lateral stability evaluation. For this purpose the route required a number of curves, with radii evenly distributed throughout the range of interest from 750 feet to 2500 feet. In addition, it was desirable that the circular portion of each of the curves should be as long as possible in order to permit an accurate assessment of the steady-state curving behavior. An additional requirements was for a tangent section of approximately 0.5 miles for carrying out the lateral stability tests.

After a thorough review of track data for the WMATA system, the Red line route from Dupont Circle to Fort Totten was selected. This route contained a good selection of curves with radii in the range of interest and a tangent section of just under 0.5 miles, as can be seen from Table 6.1-1. In addition, the route runs past, and has easy access, to the Brentwood maintenance shops, where the test vehicle was to be stationed throughout the test program.

TABLE 6.1-1
TRACK TEST SITES ON THE RED LINE

METRO STATION	CURVE NUMBER	CURVE DIRECTION	CURVE RADIUS (ft)	CURVATURE (deg)	SUPER-ELEVATION (in.)	SERVICE SPEED (mph)
Farragut North						
	3	Left	1200	4.8	4	50
Metro Center						
Gallery Place						
	311	Right	956	6.0	4	40
Judiciary Square						
	37	Left	755	7.6	4	40
Union Station						
	43	Left	1750	3.3	6	65
(Brentwood Shop)	49	Left	800	7.2	6	45
Rhode Island Avenue						
	157	Left	2508	2.3	6	70
Brookland						
	-	-	∞	0.0	0	75
Fort Totten						

6.2 TEST CONFIGURATIONS AND METHOD OF TESTING

The complete track test program consisted of the following types of test:

- Runs over the Blue Line to the National Airport test site that had been used during the WMATA Phase I tests
- Lateral stability tests
- Constant speed curving tests
- Curving tests under traction
- Route evaluation tests.

A number of truck configurations were used during the course of the test program. These consisted of various combinations of wheel profile and primary suspension. After tests with the 1 in 10 taper wheel profile had been completed, imprints were noted on the wheel tread close to the flange root, which were subsequently attributed to wheel impacts occurring at the frogs of turnouts. Although it seemed probable that a single track location was responsible for the majority of the imprints, concern about the impacts resulted in restrictions on further running with the 1 in 10 taper, and elimination of running with wheel tapers steeper than 1 in 10. As a result, only the following six configurations were used during the test program.

1. Standard primary bush with AAR cylindrical wheel
2. Standard primary bush with BR 1 in 20 wheel
3. Standard primary bush with 1 in 10 wheel
4. Modified primary bush with 1 in 10 wheel
5. Modified primary bush with BR 1 in 20 wheel
6. Modified primary bush with AAR cylindrical wheel.

The test consist was made up of a two car set comprising car numbers 1130 and 1131. Car 1130 was the test car and the leading truck was instrumented. Car 1131 remained as standard throughout the test series. For the majority of the test runs both cars were unladen, apart from the instrumentation equipment and the test personnel.

This report confines itself to a discussion of the constant speed curving tests and a brief mention of the lateral stability tests. A more detailed examination of the test results will be carried out and reported later.

Each of the curved test zones was marked, at the beginning and end, with ALD markers. These indicated the extent of the circular portion of each curve.

The testing was performed by proceeding in a continuous line from Farragut North Station to Fort Totten Station, while maintaining specified values of constant speed over each of the test zones, including the tangent zone used for lateral stability assessment. In this direction, the single instrumented wheelset was in the leading position. Following this procedure, the test consist was then run in the reverse direction over the same track from Fort Totten to Farragut North, again maintaining specified constant speeds over the test zones. In this direction of running, the instrumented wheelset was in the trailing position and this enabled wheel/rail force data to be obtained for a trailing axle. However, owing to axle misalignments within the truck, trailing axle forces in the reverse direction will not be the same as trailing axle forces in the forward direction. Therefore, an accurate force balance for the truck can not be performed. Two instrumented wheelsets would have been necessary for this to be possible.

During the pilot runs, instrumented wheelset force data was examined, in order to determine the location at which the maximum lateral force occurred on the high rail of curve 37, a 7.6° curve. The rails were strain-gaged in this area to measure wheel/rail force, as part of an experiment to examine the effect of modification to the rail fasteners. This experiment was conducted by Battelle Columbus Laboratories for TSC and is reported separately.

Data obtained during the constant speed curving and lateral stability tests was analyzed by ENSCO. The outputs from a selected number of channels, which had been displayed on one of the two strip chart recorder, were analyzed for peak and mean values in each of the test zones.

6.3 PRELIMINARY TEST RESULTS AND COMPARISON WITH THEORETICAL PREDICTIONS

The primary objectives of the test program are outlined in Section 2. However, they will be reiterated here:

- To establish by theory and experiment the reduction in wheel/rail forces that could be obtained by modifying the primary suspension

- To investigate whether further improvements in curving performance were available from wheel tread effective conicities higher than the 0.05 used during the Phase I test program.

Axle yaw torque could have been determined from the primary longitudinal displacement measurements. However, these measurements were found to be inaccurate. The inner race of the journal bearing, to which the measurement was being made, not only translated longitudinally but also rotated in pitch. The measurement of longitudinal displacement was made at a point below the centerline and, therefore, included a component from the rotation. This caused the measurement to be too large.

The measured lateral forces from the instrumented wheelset were reduced to peak and mean values in each of the test zones. These results were plotted against speed for each curve, and the values for the balance speed in each curve extracted. Figure 6.3-1 shows the peak and mean values of high rail lateral force plotted against track curvature, for all three test configurations with the standard primary suspension bush. These results show the existence of a substantial dynamic component of force over and above the mean steady-state curving force. This component exists throughout the range of track curvature and is of a similar magnitude regardless of wheel profile.

The maximum values of mean and peak lateral force at balance speed were 5,700 lbs and 10,000 lbs, respectively. These forces were measured on curve 311 which is a 6 degree right-hand curve in the normal direction of testing. It is in fact the only right-hand curved test zone on the route. The existence of the highest values of force on curve 311, which is not the sharpest curve, can probably be explained in terms of the axle misalignments which existed in the test truck and are illustrated in Fig. 3.3-4. The radial component of misalignment, which is the more important in terms of its effect on curving behavior, (Refs. 4 and 5) is in a direction which causes the truck to prefer left-hand curves. This will result in higher than normal forces on right-hand curves and lower than normal forces on left-hand curves.

The graph of mean curving force indicates the magnitude of improvement in steady curving behavior obtained as a result of wheel profile changes with the standard bush. It can be seen that the BR 1 in 20 offers a reduction of approximately 27% compared with 40% for the 1 in 10 taper.

Theoretical predictions, which were described in Section 4.3, had indicated reasonable agreement with the magnitude of force for the cylindrical wheel, approximately 5,000 lbs on a 7.6° curve with a wheel/rail friction

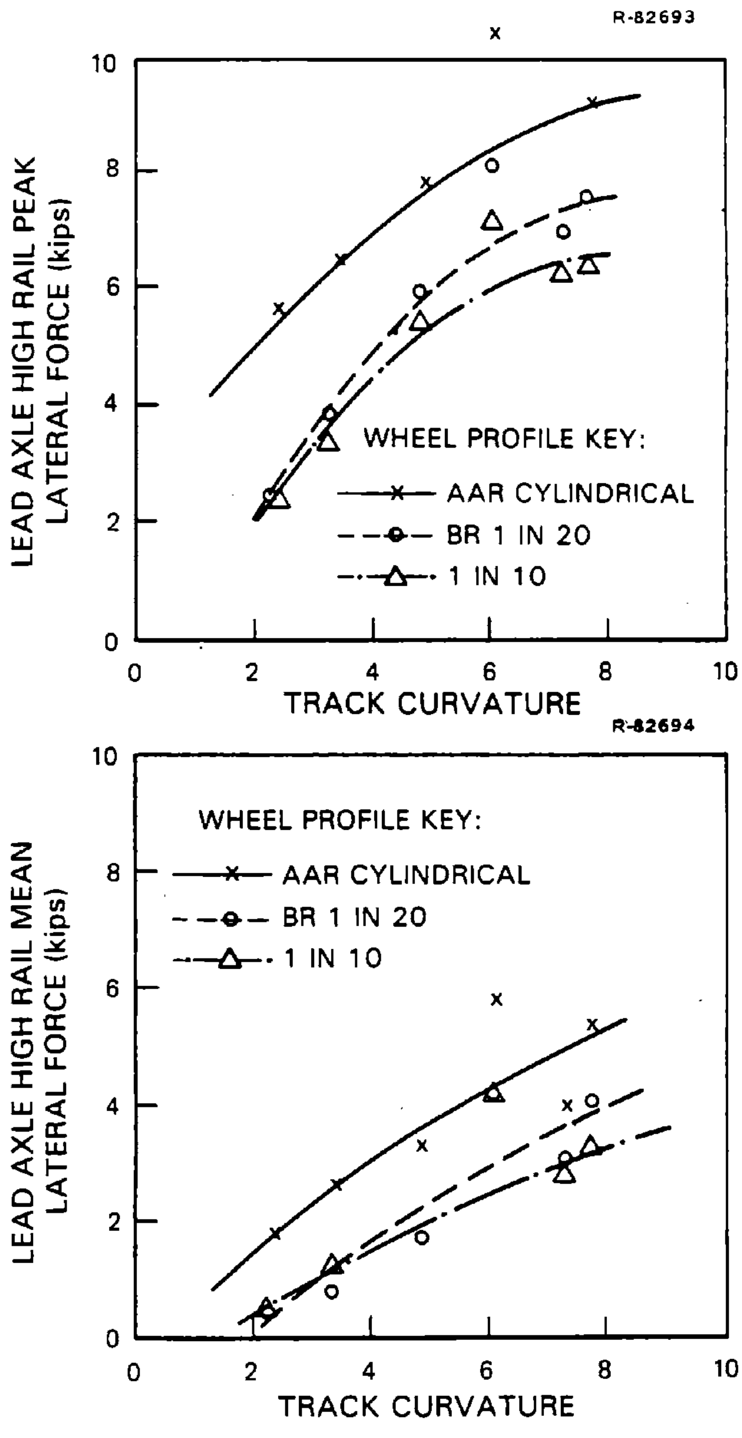


Figure 6.3-1 Track Test Results for Lead Axle Peak and Mean High Rail Lateral Force Against Track Curvature (Standard Primary Bush; Balance Speed)

coefficient of 0.5. However, the predictions had suggested only a minor difference in lateral force with changes in wheel tread effective conicity. Reductions of only 4% and 8% were predicted for BR 1 in 20 and 1 in 10 taper, respectively. Possible reasons for this discrepancy will be discussed later.

Figure 6.3-2 illustrates high rail lateral force as a function of track curvature with the modified primary bush. Again, these are values for the balance speed on each curve. Owing to the restrictions on running with the 1 in 10 tapered wheel, only test results for curve 37 are available with this profile and the modified bush. The test series with the 1 in 10 taper and standard bush had been completed before the restrictions were imposed.

The highest value for mean and peak lateral force at balance speed were 4,400 lbs and 6,900 lbs, respectively. Again, these were recorded on curve 311. Although dynamic force levels of a substantial magnitude are still present with the modified bush, the mean curving forces have been reduced considerably. In fact, the magnitude of the dynamic component seems to be unchanged by either wheel profile or the modified bush. The peak forces are typically 3,000 to 4,000 lbs above the mean forces for all configurations at balance speed.

Mean curving forces, with respect to the standard suspension and cylindrical wheel, have been reduced by 36%, 65%, and 75% in the sharpest curve, by the cylindrical wheel, BR 1 in 20 and 1 in 10 profiles, respectively when used with the modified bush. It can be seen that moderate improvement is obtained by modifying the primary suspension with the cylindrical wheel. However, dramatic reductions in force are obtained with the BR 1 in 20 and 1 in 10 tapered wheels.

The analytical predictions, for the vehicle equipped with the modified bush, had indicated about the right magnitude of force with the BR 1 in 20 and 1 in 10 tapered wheel, but had predicted relatively small changes with wheel profile, and therefore, substantially underpredicted for the case of the cylindrical wheel. Two thousand pounds mean lateral force was predicted compared with 3,300 lbs measured for the sharpest curve.

The large mean curving forces, occurring on the sharpest curves, is causing high rates of wear, particularly on the gage face of the high rail. In addition, the large dynamic component of force is causing the wear to be quite irregular. Figures 6.3-3 and 6.3-4 show cross-sectional measurements of the high rail in curve 37 at two locations which were 8 ties apart. These two figures show the substantial wear that has taken place and also the large variation in wear that is occurring.

It should be said that the analytical predictions of curving behavior were carried out using the design case rail profile, (Fig. 4.3-4) which is significantly different from the measured profiles illustrated in Figs. 6.3-3 and 6.3-4. This would result in different wheel/rail geometric constraint functions. However, for both the design case and measured rail profiles, two points of contact exist between the outer wheel on a curve and the high rail, once flange contact has occurred. The method, used in this study, for steady-state curving predictions assumes only a single point of contact between each wheel and rail.

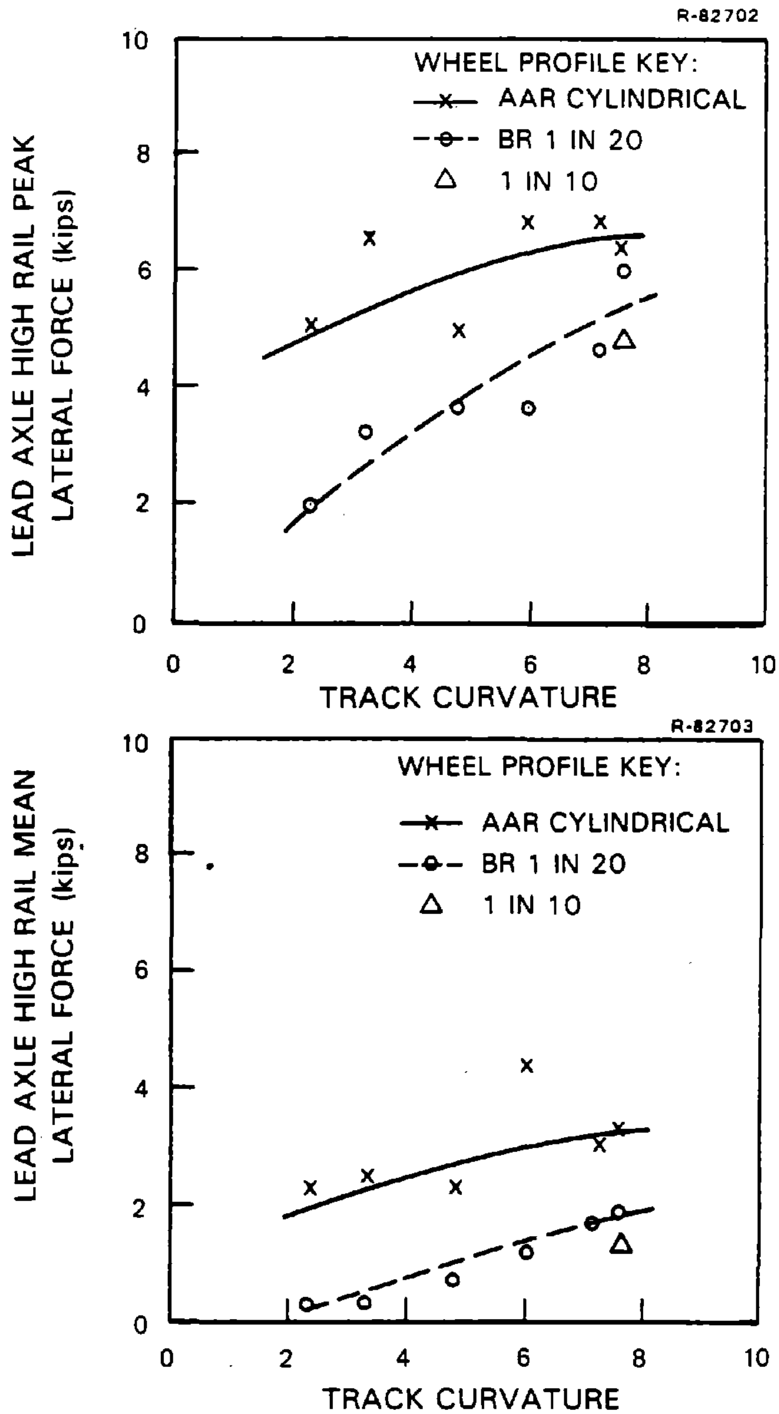


Figure 6.3-2 Track Test Results for Lead Axle Peak and Mean High Rail Lateral Force Against Track Curvature (Soft Primary Bush; Balance Speed)

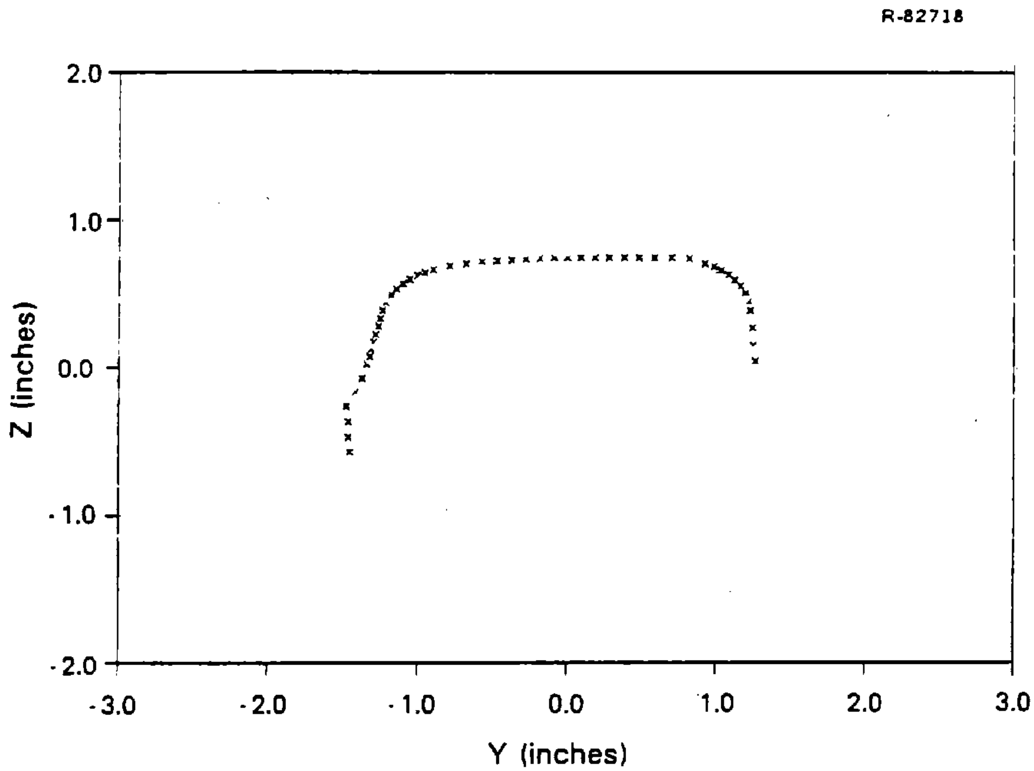


Figure 6.3-3 Curve 37 High Rail (Low Wear Site)

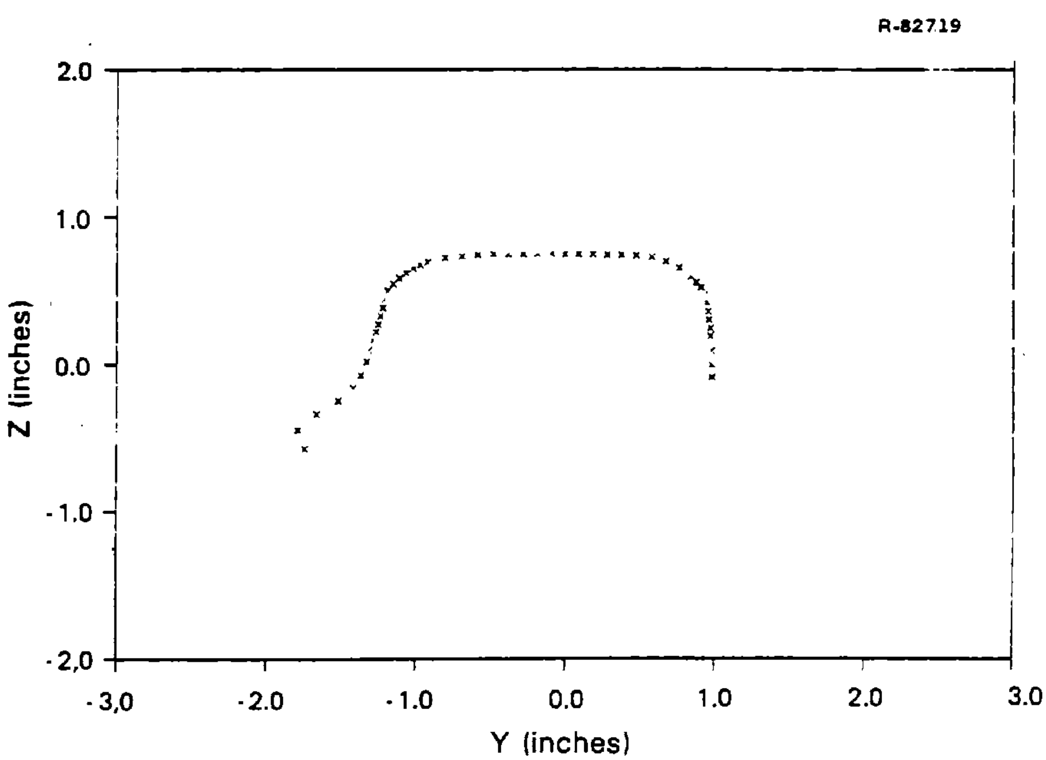


Figure 6.3-4 Curve 37 High Rail (High Wear Site)

Comparisons of the wheel/rail force characteristics for single and two point contact assumptions have shown substantial differences in the axle steering moment for a given lateral force, particularly for small angles-of-attack (Ref. 18). It is thought that the reason for the discrepancy between theory and experiment lies with the single point of contact assumption.

In addition, the measured axle misalignments have not yet been included in the theoretical predictions. As the major source of error lies in the wheel/rail contact assumption, it was felt that more detailed predictions were not justified until a two-point contact prediction was available.

With regard to lateral stability, no evidence of an approach to truck hunting was found for any of the truck configurations that were tested. Truck and car body lateral acceleration and secondary lateral displacement data were reviewed and indicated similar magnitudes for all the configurations tested with tapered wheel profiles. However, maximum speed permitted during testing was 75 mph, which is the normal maximum service speed. It would have been preferable to have tested up to a speed above the maximum service speed, in order to check that a satisfactory margin of stability existed.

7. CONCLUSIONS

Theoretical analysis and previous experience had indicated that a substantial reduction in primary longitudinal stiffness of the Rockwell truck was possible, without causing the onset of hunting within the operating speed range of the WMATA vehicle. In addition, it had shown that such a reduction would lead to a large decrease in wheel/rail lateral force. Also, the Phase I WMATA tests had indicated that a significant reduction in wheel/rail lateral force could be obtained by the use of a 1 in 20 tapered wheel profile rather than the standard AAR cylindrical profile.

This work has shown that it is practical to reduce the primary longitudinal stiffness of the Rockwell truck to a level which is close to the optimum for the duty performed by the WMATA vehicle. Also, that this can be achieved within the space envelope of the standard primary bush and, therefore, can be considered for use as a modification at relatively little expense.

It should be noted, however, that the longitudinal stiffness of the modified bush, used in this test program, is probably lower than would be chosen for a permanent modification to the vehicle. The value of stiffness that was chosen was influenced by the desire to explore the feasible range of modification and to obtain a measured value of hunting speed to confirm the theoretical predictions. In the event, hunting was not obtained, primarily because testing could not be performed with a wheel profile having a high enough effective conicity

If a permanent modification to the vehicle was desired, a primary bush configuration suitable for this purpose should be determined. This process would involve determining an adequate stability margin for the vehicle, for the extremes of wheel/rail effective conicity likely to occur in service as wheels and rails wear and their profiles change. In addition, the durability of the bush, in terms of creep and change in stiffness with time, should be determined.

The results from the track tests indicate that the large improvements in curving behavior, which had been anticipated from the theoretical analysis and previous experience, were realized in practice. Mean lateral force on the high rail of the sharpest curve was reduced by 36% at balance speed by changing to the modified primary bush. Changing the bush and using a 1 in 10 tapered wheel reduced the lateral force for the same case by 75%.

On the evidence of the test results, the BR 1 in 20 wheel profile appears to offer the best practical alternative to the standard AAR cylindrical profile. Although the 1 in 10 taper gave slightly better results, the problem associated with the impacts of the wheel tread at the frogs of turnouts, rules out its adoption, without substantial modifications to the track. However, the consequences of the two points of contact, occurring between the flanging wheel and the high rail, should be carefully considered before making any recommendations with regard to a new wheel profile.

The large dynamic component of lateral force which occurs in most of the curves gives cause for concern. The irregular wear patterns found on the high rail of many curves is a consequence of the large dynamic fluctuations in force. It is important to know the cause of this phenomena for a number of reasons.

- If vehicles were equipped with the modified primary bush, what effect would this have on the development of cyclic wear?

- If vehicles were equipped with the modified primary bush and ran over new curved track, would the irregular wear occur?

- Are there any practical modifications that could be made to vehicle or track that would prevent the occurrence of irregular wear?

8. RECOMMENDATIONS

- One vehicle running in service should be equipped with the modified primary bushes. Regular inspections should be made to check the condition of the bushes. In addition, regular measurements of wheel profile should be made on the vehicle with modified bushes and another vehicle running in the same consist. This would serve two purposes. First, it would provide a direct measure of the reduction in wheel wear rate to be expected from the modification, which was the major objective of the test program. Secondly, it would provide wheel wear data for a vehicle whose curving performance and parameters are known quite accurately. These data could be used for establishing a full-scale wear rate - wear index relationship.

- The effect of two point contact between a single wheel and rail should be investigated, in order to understand properly the consequences of this situation. Preliminary indications suggest that two point contact is undesirable and likely to lead to high initial rates of wheel and rail wear. This is due to the lower steering moment for a given lateral force, which leads to a larger angle-of-attack. In addition, the work done in the contact between wheel and rail can be much larger with two points of contact. An outcome from this study could be a new recommended wheel profile giving a single point of contact when new.

- The cause of the large dynamic force variations and the consequent cyclic wear, which occurs on the high rails of most curves, should be investigated. The necessity for this is discussed in the preceding conclusions.

APPENDIX A

DETERMINATION OF CAR BODY INERTIAS AND SECONDARY SUSPENSION STIFFNESSES FROM CAR RIGID BODY MODE NATURAL FREQUENCIES

Total weight of the WMATA vehicle and the weight of the Rockwell trucks is known quite accurately. However, the truck bolsters move with the car body for all motions, except yaw of the truck with respect to the car body. Therefore, the bolster masses must be added to the car body mass in order to obtain a correct effective mass for the body.

Effective mass of car body,

$$m_b = 130.5 \text{ lb-sec}^2/\text{inch} \tag{A-1}$$

Car body bounce frequency was measured at 1.15 Hz. Therefore, secondary vertical stiffness per truck,

$$k_{zb} = \frac{130.5 \times (2\pi\times 1.15)^2}{2} \tag{A-2}$$

$$= 3405 \text{ lb/inch}$$

Semi-spacing of trucks = 312 inches. Therefore, pitch stiffness,

$$k_{\phi b} = 2 \times 3405 \times 312^2 = 663 \times 10^6 \text{ lb-in./rad} \tag{A-3}$$

Pitch frequency was measured at 1.25 Hz. Therefore, car body pitch inertia,

$$I_{\phi b} = \frac{664 \times 10^6}{(2\pi \times 1.25)^2} = 10.75 \times 10^6 \text{ lb-in.-sec}^2 \tag{A-4}$$

For a long relatively slender body it is reasonable to assume that the yaw inertia is approximately equal to the pitch inertia. Therefore, car body yaw inertia,

$$I_{\psi b} = 10.75 \times 10^6 \text{ lb-in.-sec}^2 \tag{A-5}$$

Yaw frequency was measured at 1.28 Hz. Therefore, yaw stiffness (car body to ground),

$$k_{\psi b} = 10.75 \times 10^6 \times (2\pi \times 1.28)^2$$
$$= 695 \times 10^6 \text{ lb-in./rad} \qquad \text{(A-6)}$$

However, the yaw stiffness between the car body and ground is made up of two components. That due to the secondary lateral stiffness and that due to the yaw stiffness between truck and car body provided by the traction rods. The former is by far the larger component owing to the large spacing between the two trucks.

Yaw stiffness between car body and each truck $\simeq 31 \times 10^6$ lb-in./rad. Therefore, secondary lateral stiffness (per truck),

$$k_{yb} = \frac{(695 \times 10^6 - 2 \times 31 \times 10^6)}{2 \times 312^2} = 3250 \text{ lb/in.} \qquad \text{(A-7)}$$

Roll stiffness between truck and car body is provided by the vertical stiffness of the air springs which are placed at 25.25 inches from the car body centerline. Therefore, secondary roll stiffness (per truck),

$$k_{\theta b} = 3405 \times 25.25^2 = 2.17 \times 10^6 \text{ lb-in./rad} \qquad \text{(A-8)}$$

By considering the end view behavior of the car body, in terms of the lower center roll frequency and the upper center roll frequency, the car body roll inertia and the height of the car body center of gravity above the suspension roll center may be determined.

R-83152

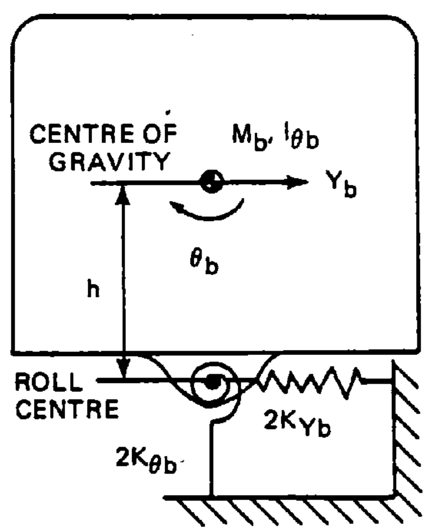


Figure A-1 End View of Car Body

$$m_b \ddot{y}_b + 2k_{yb} y_b - 2k_{yb} h \theta_b = 0 \tag{A-9}$$

$$I_{\theta b} \ddot{\theta}_b + 2(k_{\theta b}+k_{yb} h^2)\theta_b - 2k_{yb} h y_b = 0 \tag{A-10}$$

Assuming sinusoidal motion

$$y_b = y_{b0} \sin \omega t$$

$$\theta_b = \theta_{b0} \sin \omega t$$

$$(-m_b \omega^2+2k_{yb})y_{b0} - (2k_{yb} h)\theta_{b0} = 0 \tag{A-11}$$

$$(-I_{\theta b} \omega^2+2k_{\theta b}+2k_{yb} h^2)\theta_{b0} - (2k_{yb} h)y_{\theta 0} = 0 \tag{A-12}$$

from Eq. A-11

$$\frac{y_{b0}}{\theta_{b0}} = \frac{2k_{yb} h}{-m_b \omega^2 + 2k_{yb}} \tag{A-13}$$

From Eq. A-12

$$\frac{y_{b0}}{\theta_{b0}} = \frac{-I_{\theta b} \omega^2 + 2k_{\theta b} + 2k_{yb} h^2}{2k_{yb} h} \tag{A-14}$$

Equating Eqs. A-13 and A-14 leads to

$$\omega^4 - \omega^2 \frac{2k_{yb}}{m_b} + \frac{2k_{\theta b} + 2k_{yb} h^2}{I_{\theta b}} + \frac{2k_{yb}}{m_b} 2 \frac{(k_{\theta b}+k_{yb} h^2)}{I_{\theta b}}$$

$$- \frac{4k_{yb}^2 h^2}{I_{\theta b} m_b} = 0 \tag{A-15}$$

or

$$\omega^4 - \omega^2(\omega_y^2+\omega_\theta^2) + \omega_y^2 \omega_\theta^2 - \omega_{y\theta}^4 \tag{A-16}$$

where:

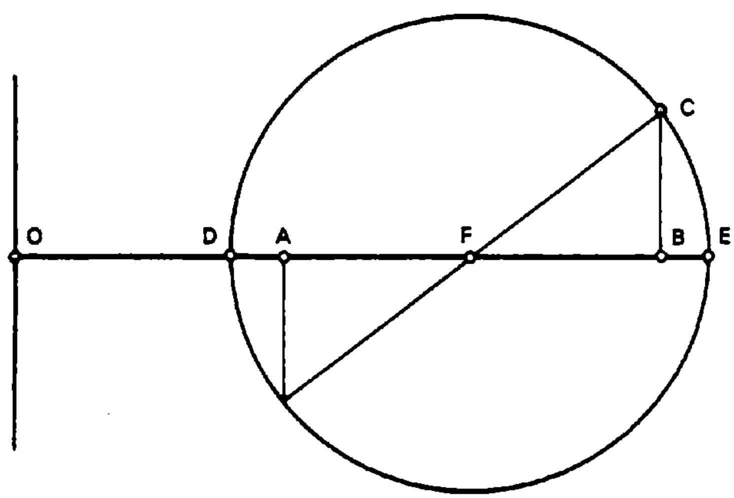
$$\omega_y^2 = \frac{2k_{yb}}{m_b} \tag{A-17}$$

$$\omega_\theta^2 = \frac{2(k_{\theta b}+k_{yb}\ h^2)}{I_{\theta b}} \tag{A-18}$$

$$\omega_{y\theta}^2 = \frac{2k_{yb}\ h}{\sqrt{I_{\theta b}\ m_b}} \tag{A-19}$$

Den Hartog (Ref. 19) shows how Mohr's circle may be used to facilitate the solution of Eq. A-16.

R-82701



where:

$OA = \omega_\theta^2$; $OB = \omega_y^2$; $BC = \omega_{y\theta}^2$

OD then bcomes ω_1^2 and OE ω_2^2

Lower center roll frequency was measured at 0.366 Hz.

$$\omega_1^2 = (2\pi\times 0.366)^2 = 5.3 \tag{A-20}$$

Upper center roll frequency was measured at 1.39 Hz

$$\omega_2^2 = (2\pi \times 1.39)^2 = 76.3 \tag{A-21}$$

From Eq. A-17

$$\omega_y^2 = \frac{2 \times 3250}{130.5} = 49.8 \tag{A-22}$$

Mohr's circle may then be used to yield

$$\omega_\theta^2 = 0A = 0D + DA \tag{A-23}$$

$$DA = BE = 0E - 0B \tag{A-24}$$

$$\omega_\theta^2 = 0D + 0E - 0B = \omega_1^2 + \omega_2^2 - \omega_y^2 \tag{A-25}$$

$$\omega_\theta^2 = 5.3 + 76.3 - 49.8 = 31.8 \tag{A-26}$$

Mohr's circle may again be used to yield

$$\omega_{y\theta}^2 = BC = \sqrt{FC^2 - FB^2} = \sqrt{35.5^2 - 9^2} = 34.3 \tag{A-27}$$

Equations A-18 and A-19 may be combined to yield

$$I_{\theta b} = \frac{2k_{\theta b}}{\omega_\theta^2 \left(1 - \dfrac{\omega_{y\theta}^4 m_b}{2\omega_\theta^2 k_{yb}}\right)} \tag{A-28}$$

$$h = \frac{k_{\theta b}}{k_{yb} \left(\dfrac{2\omega_\theta^2 k_{yb}}{\omega_{y\theta}^4 m_b} - 1\right)} \tag{A-29}$$

Therefore, car body roll inertia

$$I_{\theta b} = 0.53 \times 10^6 \text{ lb-in./sec}^2 \tag{A-30}$$

Height of center of gravity above roll center

$$h = 43.9 \text{ in.} \tag{A-31}$$

APPENDIX B

REPORT OF NEW TECHNOLOGY

This report describes analyses and tests of wheel taper and primary suspension stiffness on wheel-rail interaction forces. The work described has not resulted in the development of any new or unique devices.

REFERENCES

1. Phillips, C., Weinstock, H., Grief, R., and Thompson, III, W.I., "Measurement of Wheel/Rail Forces at the Washington Metropolitan Area Transit Authority, Volume I, Analysis Report," Report Number UMTA-MA-06-0025-80-6, July 1980.

2. Ahlbeck, D.R., Harrison, H.D., and Tuten, J.M., "WMATA Wheel-Rail Force Measurement, Volume II, Test Report," Report Number UMTA-MA-06-0025-80-7, February 1980.

3. Weinstock, H., and Grief, R., "Analysis of Wheel/Rail Force and Flange Force During Steady-State Curving of Rigid Trucks," Report Number UMTA-MA-06-0025-80-8, September 1980.

4. Mutter, H., et al., "Metropolitan Atlanta Rapid Transit Authority Transit Vehicle Engineering Tests," Report Number FRA/TTC-81/05.

5. Elkins, J.A., and Allen, R.A., "Testing a Transit Vehicle for Wheel and Rail Wear," Presented at the 1981 ASME Winter Annual Meeting, Washington, D.C., ASME Paper No. 81-WA/DSC-19.

6. Elkins, J.A., and Gostling, R.J., "A General Quasi-State Curving Theory for Railway Vehicles," Proceedings from 5th VSD 2nd IUTAM Symposium, Vienna, September 1977.

7. Elkins, J.A., and Eickhoff, B.M., "Advances in Non-linear Wheel/Rail Force Prediction Methods and Their Validation," Presented at the ASME Winter Annual Meeting, New York, December 1979, To be published by ASME.

8. Lanza, J., "CARHNT Productions Runs," Memorandum No. M-E239 to H. Weinstock DOT/TSC from SDC Integrated Services, Inc., April 1980.

9. Cooperrider, N.K., and Law, E.H., "Parameters for Calculations of Stability Characteristics of MARTA Vehicle," Acorn Associates, 1215 East Del Rio Drive, Tempe, Arizona, July 1977.

10. Letter to H. Weinstock DOT/TSC from J.J. Williams, Senior Technical Director, Rockwell International, Mass Transit Division, January 1980.

11. Prause, R.H., "User's Manual for Programs CARHNT and TRKHNT for Rail Vehicle Hunting Stability Analysis," Battelle Columbus Laboratories, Columbus, Ohio, January 1974.

12. Cooperrider, N.K., and Law, E.H., "Secondary Yaw Suspension Calculations for MARTA Vehicle, Acorn Associates, 1215 East Del Rio Drive, Tempe, Arizona, July 1977.

13. Hedrick, J.K., Cooperrider, N.K., and Law, E.H., "The Application of Quasi-Linearization Techniques to Rail Vehicle Dynamic Analyses," Report Number DOT-TSC-902.

REFERENCES (Continued)

14. Hobbs, A.E.W., "A Survey of Creep," British Rail Technical Note DYN 52, April 1967.

15. Boocock, D., "The Steady-State Motion of Railway Vehicles on Curved Track," Journal of Mechanical Engineering Science, Volume II, No. 6, 1969.

16. Newland, D., "Steering a Flexible Railway Truck on Curved Track," Journal of Engineering for Industry, August 1969.

17. Kalker, J.J., "On the Rolling Contact of Two Elastic Bodies in the Presence of Dry Friction," (Plus Table Book), Doctoral Thesis, Delft, 1967.

18. Elkins, J.A., and Weinstock, H., "The Effect of Two-Point Contact on the Curving Behavior of Railroad Vehicles," Submitted for presentation at the ASME Winter Annual Meeting, Phoenix, Arizona, November 1982.

19. Den Hartog, Mechanical Vibrations, McGraw-Hill Book Company, Inc., 1956.

100 copies