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Truck Design Optimization Project Phase II

Final Report

Wyle Laboratories
Scientific Services
& Systems Group

Colorado Springs Division
4620 Edison Avenue
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NOTICE

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16. Abstract

The overall objectives of the TDOP Phase II are to provide technological and economic bases for assessing freight car truck performance and design, and to generate uniform testing and performance guidelines for truck/carbody systems. This report summarizes the approaches employed in this study which included the development of a truck evaluation methodology to delineate the relationships between performance and railroad operating costs and profits; establishment of analytic and experimental procedures for relating truck parameters to the economic-based performance indices over the range of in-service train configurations, track conditions, equipment parameters, and speeds; and conduct of analytical and experimental studies to define the performance boundaries of existing and proposed freight car truck configurations. The report discusses the development of performance and test specifications, and the methodology used in defining economic implications. Also included in the report are summary characteristics of Type I and Type II freight car trucks, guideline performance specifications for Type II freight car trucks, a set of guidelines for standardized field and laboratory tests, and conclusions with respect to freight car truck economics, in such areas as maintenance costs, fuel consumption, and rail wear.

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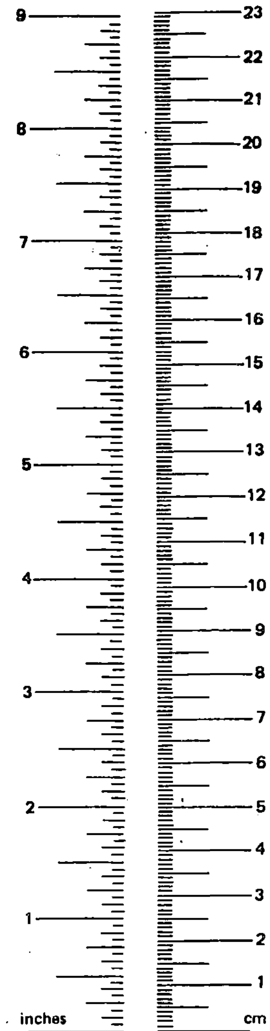
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

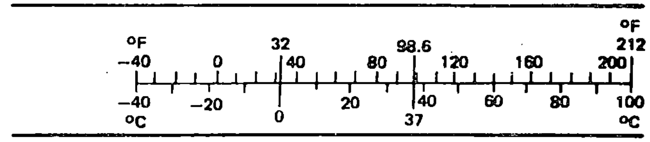
Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2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

*1 in. = 2.54 cm (exactly). For other exact conversions and more detail tables see NBS Misc. Publ. 286. Units of Weight 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	36	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



EXECUTIVE SUMMARY

As the major component between the carbody and the track, the freight car truck performs the essential functions of guidance, support, and vibration absorption for the freight car. In performing these functions in a dynamic environment, the standard three-piece truck has performed remarkably well since its introduction in the early 1940's. However, increasing demands on the rail transportation system, in the form of heavier car weight, higher center of gravity, and increasing speed, coupled with deteriorating maintenance of equipment and track, have brought into focus the need for design improvements in freight car suspension systems.

In response to the railroad industry's requirement for improved suspension systems, the supply industry has developed a variety of add-on devices and retrofit packages for the Type I trucks, as well as completely new suspension systems incorporating innovative design features in the form of Type II trucks. However, there have been no systematic studies or criteria allowing a correlation of the costs and benefits associated with these new design Type II trucks. The Federal Railroad Administration-sponsored Truck Design Optimization Project (TDOP) Phase II is aimed at providing a framework for such an evaluation through which freight car trucks can be studied and the relationship between performance improvements and increased costs can be analyzed. The project's main purpose is to characterize the behavior of existing trucks and to generate performance and test specifications for new truck designs. Using quantitative performance indices defined on the basis of operational and economic considerations, these specifications will not only provide the technical base for an evaluation of design innovations, but also will facilitate a correlation with the cost of such design improvements.

The standard, three-piece freight car truck, or its modified versions with basically similar configurations, is defined in TDOP as the "Type I" truck. The Type II (premium) truck is defined as a truck which utilizes current wheelset and journal bearing assemblies, is compatible with existing air brake systems, and preserves car coupler height, while incorporating engineering innovations in the design of the suspension systems.

Southern Pacific Transportation Company was the contractor for TDOP Phase I. Two standard, three-piece trucks (the American Steel Foundries' Ride Control truck and the Standard Car Truck Company's Barber S-2 truck) were tested under 70- and 100-ton carbodies. The data from Phase I constitute the main basis for characterizing the performance of the Type I truck.

Phase II of the TDOP project, with Wyle Laboratories as the prime contractor and the Union Pacific Railroad as the principal subcontractor, had the objectives of:

- Definition of the performance of both Type I and Type II trucks in quantitative terms, represented by performance indices.
- Establishment of a plan for collecting economic data on the cost of acquiring, operating, and maintaining the standard, Type I truck.
- Determination of a quantitative basis for evaluating the economic benefits to be derived from Type II trucks.
- Development of performance characteristics for Type I trucks and performance specifications for Type II trucks.
- Development of guideline test specifications for freight car trucks.
- Cost/benefit analysis of Type II trucks relative to Type I trucks.

These objectives had been met through several approaches including:

- Road testing several Type I and Type II trucks.
- Mathematical modeling of freight car trucks to augment and complement the comparison of test results.
- Determination of wear of Type I and Type II trucks in unit train service over an extended period of time.
- Collection of economic data on truck maintenance and operation, and correlation of such data with information on truck performance.
- Engineering interpretation including effect on performance of eventual wear and deterioration of truck components.
- Correlation of the costs and benefits associated with incremental changes in the levels of performance as obtained from the results of the engineering evaluation of the trucks.

Based on the technological and economic studies conducted during TDOP Phase II for assessing freight car truck performance and design, the following conclusions may be stated:

- The improved design features in the Type II trucks achieve a degree of qualified success in attaining improved performance from freight car trucks. These successes, however, are limited to some of the domains of the performance rather than comprehensive, all-around improvement in all aspects.
- Performance evaluation of freight car trucks needs to be undertaken under well-defined sets of conditions relating to the state of wear and deterioration of vehicle and track structure in order to address fully all aspects of performance. For example, wheel and rail contact geometry, to which vehicle performance is extremely sensitive needs to be thoroughly documented through the bulk of the wheel and rail life cycle and their representative conditions used in any comprehensive evaluation program.

- On the basis of the analysis of available car maintenance data, costs associated with car maintenance alone do not warrant or justify the levels of increased capital investment demanded by the Type II trucks. On the other hand, improved rolling and curving resistance, and consequent reduction in fuel consumption, seem to be very promising areas, indicating that the additional investments warranted by the Type II trucks could be advantageous. Specific considerations, such as an intermodal scenario, also point to an advantageous outlook with respect to investment in Type II trucks with rigidized frame-primary suspension features.

- Significant economic benefits from the utilization of Type II trucks seem to accrue more in the area of the track structure, in general, and the rail, in particular. Reduced rail wear as well as retardation in rail and track structure deterioration are indicated as a result of improved truck performance. These economic implications, if properly accounted for through a systematic rail wear and track deterioration study, could be significant.

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Mr. Bernard Eck
Director of Product Engineering
Griffin Wheel Company

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Manager of Research & Tests
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Ass't to Vice President-Equipment
Trailer Train Company

Mr. Leo Lombardo
Manager - Research
Consolidated Rail Corporation

Mr. Al Martin
Manager, Product Engineering
Dresser Transportation Equipment Division

Mr. Leonard McLean
Chief of Equipment Engineering
Seaboard Coast Line Railroad

Mr. C.H. Melcher
Chief Engineer R&D
Amcar Division
ACF Industries

Mr. Norm Morella
Manager, International Operations
National Castings Division
Midland-Ross Corporation

Mr. S.K. Punwani
Senior Research Engineer
AAR Technical Center

Mr. Robert Radford
Chief Mechanical & Electrical Engineer
Canadian National Railways

Mr. David Reynolds
Ass't Manager, Research & Tests
Southern Railway System

Mr. William Ruprecht
Director - Engineering
Shippers Car Line Division
ACF Industries, Inc.

Mr. H. Garth Tennikait
Manager - Testing Engineering
American Steel Foundries

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TABLE OF CONTENTS

SECTION 1 - INTRODUCTION

Objectives of the Truck Design Optimization Project (TDOP) Phase II are to quantify freight car truck performance and to establish a formal methodology to evaluate the relationship between performance and economic implications in terms of costs and benefits for freight car trucks. The primary project objectives have been addressed through parallel engineering and economic studies. The engineering effort consisted of field testing of freight car trucks under revenue service conditions to generate both performance and endurance test data; analysis and interpretation of the data; and the establishment of quantitative performance and a set of standard test specifications for freight car trucks. Under the economic studies, maintenance data were obtained from operating railroads and analyzed; a methodology was developed for evaluating truck performance and determining the associated economic implications; and a cost/benefit analysis was performed on premium freight car trucks.

The overall performance of freight car trucks has been compartmentalized into four distinct and non-overlapping performance regimes. These performance regimes are lateral stability, trackability, steady state curve negotiation, and ride quality. Each of these regimes is primarily defined as a set of conditions with predominant features which distinguish one from another (Reference 1). Measurable quantities of truck performance, defined as performance indices, are identified within each regime. The levels of truck performance are represented by a range of performance indices quantified in each of the regimes, and related to a set of specified operating conditions such as speed, track quality, and lading conditions. The performance regimes and associated performance indices are given in Table 1-1. Performance data generated by means of field tests during Phase I and Phase II of TDOP form the basis for quantification of performance indices within each regime.

The bulk of the field test data on Type I (standard) freight car trucks came from data generated during Phase I of TDOP. These data were supplemented with data obtained from selected additional field tests on the Type I truck conducted during Phase II of TDOP. The results from the analysis and interpretation of the data on Type I trucks formed the baseline against which Type II (premium) truck performance was subjected to a comparative evaluation. The analysis of field test data on Type I trucks was enhanced through the use of simulated data from mathematical models. The model simulated data were used primarily in an interpretative mode in quantifying truck performance. The results on Type I trucks have been published in an engineering document entitled "Performance Characterization of Type I Freight Car Trucks" (Reference 2).

From among the population of commercially available Type II freight car trucks, seven candidates were selected (Reference 3) and field tested in TDOP Phase II. Using a set of standard instrumentation including wheel/rail force and angle-of-attack measurement transducers especially developed during the project (References 4 and 5), one Type I and seven Type II trucks were field tested under revenue service conditions on the Union Pacific's track near Las Vegas, Nevada (References 6 and 7). The test data were subsequently reviewed and analyzed. The results from the analysis of the test data led to the quantification of performance of the Type II trucks. A comparative study of the performance of Type II and Type I trucks then led to the specification of performance for the Type II trucks (Reference 8).

TABLE 1-1. TDOP TRUCK PERFORMANCE CLASSIFICATION

PERFORMANCE REGIME		PERFORMANCE INDEX
Lateral Stability		Critical Speed Peak Lateral Acceleration (Zero-to-Peak)
Ride Quality		Transmissibility (Vertical, Lateral, Roll) RMS Accelerations (Vertical, Lateral, Roll) - 0-20 Hz
Steady State Curve Negotiation		Average Lateral Force On Leading Outer Wheel Average L/V Ratio On Leading Outer Wheel Average Angle-of-Attack Of Leading Axle
Trackability	Harmonic Roll	Critical Speed Peak Roll Angle (Zero-to-Peak)
	Bounce	Critical Speed Peak Vertical Acceleration (Zero-to-Peak)
	Track Twist	Wheel Unloading Index* (95th Percentile)
	Curve Entry/Exit	Wheel Unloading Index (95th Percentile)

* Wheel Unloading Index $WUI = 1 - W_L/(W_H/3)$,

where,

W_L is the vertical force on most lightly loaded wheel

W_H is the sum of vertical forces on the three most heavily loaded wheels

A set of standardized field and laboratory test specifications has also been developed so that test programs can be designed to generate performance test data which can be evaluated against the recommended levels of performance developed during the Phase II effort (Reference 9).

In the economic studies, car maintenance data from two major U.S. railroads were systematically analyzed to provide identification of areas of significant

Note: References can be found at the end of each section.

influence on operational economics. Section 5 of this report includes the details and results from this analysis. Following the insight gained from this analysis, efforts were focused on fuel consumption as a major area of economic significance. Rolling resistance/fuel consumption test data obtained from Phase II field tests were then analyzed and used in determining relative improvements in fuel consumption attributable to Type II trucks as compared to Type I trucks. The other significant aspect of the economic study was a cost/benefit analysis on Type II trucks. This effort delineated the costs associated with improved designs incorporated in the Type II trucks and the benefit accruing from them. This analysis attempts to spell out the economic feasibility of the engineering options presented to the railroad industry through the improved design Type II trucks.

An adjunct study on the endurance of the freight car trucks is comprised in the "Wear Data Collection Program." In this program, the trucks have been deployed in unit coal train service and wear data on various truck components periodically collected through field measurements. The program, at present, is still underway. A status report on the program has been published (Reference 10).

The following sections of this report are organized to provide a broad overview of the engineering and economic studies undertaken during the project as well as a summary of the results from these efforts. Section 2 describes the field test program conducted to generate the performance test data; Section 3 describes the efforts of the analytical studies; Section 4 presents the performance characteristics of Type I and Type II trucks, the performance specification for Type II trucks, and the guideline test specifications for freight car trucks; Section 5 discusses the economic methodology developed for the evaluation of costs and benefits associated with improved design freight car trucks; and Section 6 provides some conclusions and recommendations arrived at through the engineering and economic studies under the TDOP project.

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SECTION 2 - FIELD TEST PROGRAM

2.1 INTRODUCTION

One of the objectives of the Truck Design Optimization Project (TDOP) Phase II was to characterize the Type I and Type II trucks. The characterizations of Type I trucks was to be based on test data obtained from TDOP Phase I (Reference 1) and augmented by a TDOP Phase II field test program of the Type I trucks (References 2 through 5). An evaluation of the test data from TDOP Phase I showed that certain omissions in the test matrix and inadequate measurement techniques would require conducting a limited number of tests to complete the Type I truck characterization (References 6 and 7).

The primary objective of the Type I truck test program conducted in TDOP Phase II was to provide adequate lateral and vertical wheel/rail force and wheel/rail angle of attack data during curving. Other objectives were to provide data on truck load equalization performance, on truck rolling resistance for fuel consumption studies, and on wheel/rail forces during hunting.

The TDOP Phase II field test program on Type II trucks was conducted to obtain performance data on several selected Type II freight car trucks in the four performance regimes of lateral stability, trackability, curve negotiation, and ride quality (References 8 through 11). Tests were also conducted to obtain rolling resistance data as part of the fuel consumption study. Data on the longitudinal coupler forces were used to compare the relative ability of various trucks to reduce rolling resistance and flanging forces, thus improving fuel consumption.

Seven Type II trucks were selected for testing (Reference 12). The testing was conducted in four series: Series 1, Dresser DR-1 and National Swing Motion; Series 2, Barber-Scheffel and MTS Maxiride 100; Series 3, Devine-Scales and ACF Fabricated; and Series 4, Alusuisse. Details of each test series are documented in the Type II Truck Test Events Reports, Series 1 through 4 (Reference 10).

The remainder of this section contains an evaluation of the Phase I data (Section 2.2); a description of the instrumentation used (Section 2.3); a description of the test area, track profiles, track geometry measurements, etc. (Section 2.4); the Type I truck testing (Section 2.5), and the Type II truck testing (Section 2.6).

2.2 PHASE I DATA EVALUATION

A brief summary of the equipment tested during TDOP/Phase I is given in Appendix A. A complete description can be found in the TDOP Phase I Final Report (Reference 1).

To determine the usefulness of the Phase I data, the quantity and scope of the data was first evaluated (Reference 7). A data sorting routine revealed that the preponderance of the 273 Type I truck test runs were made with a refrigerator car on ASF 70-ton Ride Control trucks with new wheels (see Table 2-1). This emphasis made the data more difficult to use since the refrigerator car is not a typical freight car because of its uneven weight distribution and very large empty weight.

TABLE 2-1. PERCENT OF TEST RUNS BY BODY, TRUCK, AND WHEEL TYPE

CAR	PERCENT
Refrigerator Car	86%
70-ton Box Car	3%
100-ton Box Car	4.5%
89-ft. Flat Car	3.5%
100-ton Hopper Car	3%
TRUCK	**PERCENT**
ASF 70-ton Ride Control	82%
ASF 100-ton Ride Control	2%
Barber 70-ton	7%
Barber 100-ton	5%
ASF 70-ton Low Level	4%
WHEEL	**PERCENT**
1/20 (new)	72%
1/40 (new)	4%
Cylindrical	12%
Half Worn	2%
Worn	10%

The data sorting routine revealed these other significant omissions:

a. No curving tests were run on 100-ton box cars and hopper cars with the ASF Ride Control truck.

b. No curving tests were run with worn wheels on any car except the refrigerator car.

c. The lateral wheel force at the wheel/rail interface was improperly measured.

d. No high-speed CWR tests were run with the 100-ton box car on an ASF truck, or the 100-ton hopper car with the Barber truck.

e. No tangent track tests were run with worn wheels except for the refrigerator car, and the empty 89-foot flat car.

f. There were no medium-speed jointed rail test runs with a 100-ton box car on an ASF truck, or the 100-ton hopper car with the Barber truck. Since jointed rail exercises the friction snubber, this omission makes it difficult to compare the two types of snubbing systems.

g. Shimmed track tests with other than cylindrical wheels were run only with the refrigerator car.

The next step in the data evaluation process was to determine which measurements taken during Phase I provided useful and accurate representations of the quantity measured. The conclusion was that the measurements were satisfactory except in two areas: the measurement of lateral wheel force at the wheel/rail interface and in the detection of Automatic Location Detector (ALD) targets. The first deficiency is of major significance. The lack of lateral forces at the wheel/rail interface was of critical importance to TDOP Phase II. Without it, little could be done in validating curving models or assessing the curve negoti-

TABLE 2-2. TEST DATA REQUIRED FOR PHASE II ANALYSIS

Performance Regime	Performance Index	Necessary Test Data	Availability of Test Data from Phase I
Lateral Stability	• Critical Speed	Lateral Acceleration of one or more representative points on the truck measured as a function of speed and such variables as: wheel/rail contour, rail surface conditions, car bodies (truck spacing, stiffness), and lading (empty, full, ...)	Lateral acceleration available on axle and car body. Data are taken at constant speeds of 40, 50, 60, 70, and 79 mph. Varying speeds exist between these constant speeds. Variables such as wheel profile, rail surface conditions, car body parameters, and lading is noted in the test header. No rail contour data are available. Tests were not run for a full matrix of variables.
	• Magnitude of Lateral Acceleration	Magnitude of lateral acceleration at or near the hunting speed, for the same set of variables mentioned above.	Lateral acceleration data on axles.
Curve Negotiation	• Lateral force on leading outer wheel per 1000 pounds axle load per degree of curve under, at and over balance speed.	Lateral force on leading outer wheel as a function of lading, degree of curvature at, under, and above balance speed.	No measurements made of lateral force.
	• Wear Index	Angle of attack as a function of lading, and degree of curvature under, at, and above balance speed.	No measurements made of angle of attack.
	• Derailment Potential	L/V ratio as a function of speed, lading, wheel/rail contour.	No measurements made from which to calculate L/V.
Trackability	• Wheel Unloading Index	Simultaneous loads under the wheels as a function of track twist in degrees as a function of lading.	No measurements made of vertical load at wheel. Vertical loads measured at bearing adapters, but cannot be correlated to track geometry.
	• Max. Roll Amplitude	Max. roll amplitude as a function of excitation (amp. and frequency) for different lading conditions.	Roll angle of car body/truck bolster and roll acceleration of car body were measured, however, they cannot be correlated to track geometry.
	• Rate of Energy Dissipation	Level of friction force, displacement (i.e., spring travel), rate of increase of friction level with spring compression, as a function of lading.	No friction snubber force measurements were made.
	• Derailment Potential	L/V ratio as a function of speed, lading, wheel/rail contour.	No measurements made from which to calculate L/V.
Ride Quality	• Transmissibility	Acceleration response, referred to one or more specific locations on the car body, as a function of speed, track quality and lading within the normal operating range of speeds.	Vertical acceleration made on car body. Speed, trackability, and lading were varied, however, a complete matrix of these variables was not tested.
Fuel Consumption	• Differential Force/Grade • Differential Force/Curvature	Coupler force, Coupler angle, actual grade, track curvature	None

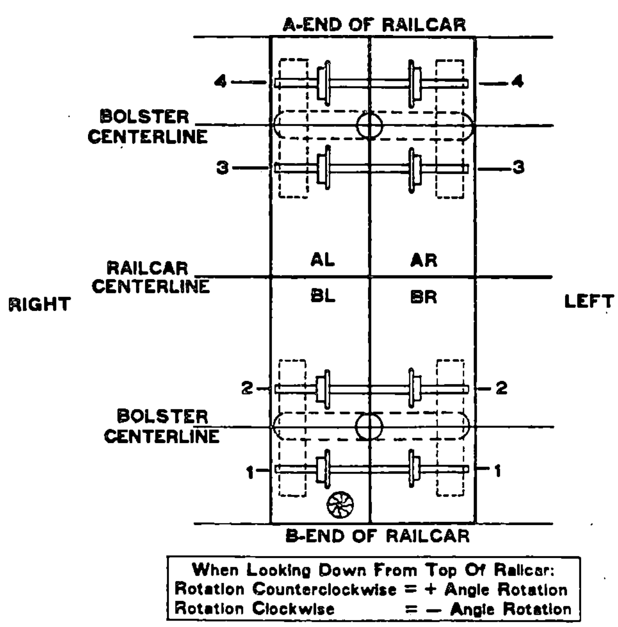


FIGURE 2-1. AAR STANDARD FOR COMPONENT LOCATION

ation performance indices on the Type I truck. The lack of ALD target detection (not being able to correlate ALD targets with response) limits the usefulness of the data for analysis of the trackability regime. The lack of ALD correlation hampers the ride quality evaluation to a lesser degree.

Finally, the TDOP Phase I data were evaluated for their adequacy in the development of Type I truck performance characterization. This evaluation is shown in Table 2-2 which lists the performance index for each of the four regimes and the test data required to quantify the performance index. For the lateral stability and ride quality regimes, the data appeared to be adequate; however, the lack of accurate measurements on the lateral forces at the wheel/rail interface made it difficult to extract meaningful information from the test data in the curve negotiation and trackability regimes.

2.3 INSTRUMENTATION

The primary objectives of the new instrumentation developed in TDOP Phase II were to obtain measurements required to calculate the forces at the wheel/rail interface and the wheel/rail angle of attack. In addition, transducers were installed to measure truck and carbody relative motion, rigid body car motion, and coupler forces.

In order to identify transducers on the carbody and trucks, the AAR standard for component location shown in Figure 2-1 was used. This enabled the exact location of a transducer to be specified. However, the right and left-hand side designation were changed because the B-end was always traveling in the forward position; therefore, the AAR conventional left side becomes the test right side. Thus, the right and left side of the test vehicle were referred to as shown in Figure 2-1 but the AAR standard for component location was used to define transducer mounting locations. For example, the bearing adapter on the right front axle would be BL-1.

The instrumentation varied from truck to truck depending on the data channels required for each truck design. For example, trucks with primary suspension had displacement across the primary springs measured. In particular, the Alusuisse truck had a considerably different set of instrumentation from the other trucks because its radically different design did not allow the deployment of the same instrumentation package used with the more conventional trucks.

Table 2-3 contains a description of all measurements taken for Type I and Type II trucks. This table gives the measurement identification number and a description of the type, frequency response, measurement range, accuracy, and purpose of each measurement.

The following paragraphs describe the measurement taken. However, not all measurements were taken on every truck because of the different truck designs. For this reason, Table 2-4 has been provided to show the exact instrumentation by truck.

2.3.1 Wheel/Rail Force Measurements

The forces at the wheel/rail interface provide the key parameters in the characterization of truck performance during curve negotiation. To accomplish the objectives of TDOP Phase II, it was required that these interface loads be measured with sufficient accuracy to adequately characterize truck performance. After an extensive review of techniques for measuring these forces, the axle-bending technique was chosen (Reference 13). To improve accuracy, additional terms were included in the equations to calculate the lateral and vertical forces. Further, measurements of the point of application of the vertical loads was implemented. This resulted in a mean rms error of 12.6 percent, which was considered acceptable. It should be noted that this error assumes a calibration accuracy of one percent and does not include any errors which may be introduced by the measurement of the vertical loads.

Thus, the approach to the measurement of wheel/rail vertical and lateral forces consisted of:

a. Instrumentation of the axle with strain gauges.

b. Instrumented bearing adapters or displacement measurement across primary spring group to measure vertical loads.

c. Eddy current displacement transducers to measure wheel/rail relative position.

d. Slip rings for the rotating axle transducers.

Strain - Gauged Axle. Each axle on the B-end truck was instrumented with eight, full-bridge strain gauges on each side of the axle. The strain gauges were placed with one-half the bridge at the top and one-half the bridge at the bottom, as shown in Figure 2-2. Thus, there were 16 half bridge strain gages at $22\frac{1}{2}$ degree increments around the axle. The half bridges on opposite side of the axle (1A to 1B, 2A to 2B, etc.) were connected in a full bridge arranged for transmission to the instrumentation car, Mobile Laboratory Car 210. A rotary pulse generator (RPG) was placed on each axle to define the strain gage position as a function of rotation angle.

In addition to measuring the axle bending moments, the torque in each axle was measured using two strain gage measurements at the middle of the axle. This measurement can be used to estimate longitudinal creep forces.

Instrumented Bearing Adapter. One of the prime contributors to the measurement of lateral and vertical forces at the wheel/rail is the vertical forces at the bearing adapter and the line of action of these forces. In previous testing, the vertical force at the bearing adapter was measured using a strain-gauged bearing adapter instrumented with one strain gauge bridge at the center. This approach suffered from two deficiencies. The first was that gage sensitivity changed as the load point changed; the second was that the adapter could not measure the line of action of the vertical force.

TABLE 2-3. INSTRUMENTATION LIST

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY	PURPOSE
S1	Speed	1 Hz	0-120 mph	±.5 mph	Speed from distance counter to time base unit
S2	ALD	256 samples/in.	6-12"	±6"	Location of track test areas. Mounted under Car 210
S3	Brake Cylinder Pressure	10 Hz	0-100 psid	1%	Brake line pressures (to insure inadvertent braking from being mixed with data)
S4	Throttle Setting		1-8		Correlation with draw bar forces in fuel consumption study
C1	"B" End Coupler Force	20 Hz	25000 lb	5%*	Longitudinal draw bar force and correlation to fuel consumption
C2	"B" Coupler Angle	20 Hz	±10°	±.1°	Coupler angle from longitudinal centerline of car
C3	"A" End Coupler Force	20 Hz	25000 lb	5%*	Longitudinal draw bar force and correlation to fuel consumption
C4	"A" Coupler Angle	20 Hz	±10°	±0.1°	Measure coupler angle from longitudinal centerline of car
D1	Rt. Spring Group Vert. Disp. Front	20 Hz	+2" -4"	1%	Right spring group disp. side frame pitch
D2	Rt. Spring Group Vert. Disp. Rear	20 Hz	+2" -4"	1%	Right spring group disp. side frame pitch
D3	Left Spring Group Vert. Disp. Front	20 Hz	+2" -4"	1%	Left spring group disp. side frame pitch
D4	Left Spring Group Vert. Disp. Rear	20 Hz	+2" -4"	1%	Left spring group disp. side frame pitch
D5	Truck Bolster to Side Frame Lat. Disp. Rt. Fr. Side	50 Hz	±0.8"	1%	Side frame relative disp. truck tram
D6	Truck Bolster to Side Frame Lat. Disp. Rt. Rear Side	50 Hz	±0.8"	1%	Side frame relative disp. truck tram
D7	Truck Bolster to Side Frame Right Bottom	50 Hz	±0.8"	1%	Used with D5 & D6 to measure side frame roll
D8	Truck Bolster to Side frame Lat. Disp. Lf. Fr. Side	50 Hz	±0.8"	1%	Side frame relative disp. truck tram
D9	Truck Bolster to Side Frame Lat. Disp. Lf. Rear Side	50 Hz	±0.8"	1%	Side frame relative disp. truck tram
D10	Truck Bolster to Side Frame Lat. Disp. Lf. Rr. Bot.	50 Hz	±0.8"	1%	Used with D8 & D9 to measure side frame roll
D11	Carbody to Truck Bolster Rel. Disp. Rt. Side	20 Hz	±1.5"	1%	Carbody/truck roll angle

*NOTE: Units have hysteresis of ±200 lb but are 1% when either in tension or compression.

TABLE 2-3. INSTRUMENTATION LIST (CONT'D)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY	PURPOSE
D12	Carbody to Truck Bolster Rel. Disp. Left Side	20 Hz	±1.5"	1%	Carbody/truck roll angle
D13	Carbody to Truck Lat. Disp. Forward	100 Hz	±10°	0.1%	Truck/carbody swivel, center plate lateral slip
D14	Carbody to Truck Lat. Disp. Rear	100 Hz	±10°	0.1%	Truck/carbody swivel, center plate lateral slip
D15	Primary Spring Disp. Rt. Fr. Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion
D16	Primary Spring Disp. Rt. Rear Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion
D17	Primary Spring Disp. Lf. Fr. Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion
D18	Primary Spring Disp. Lf. Rear Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion
D19	Axle to Side Frame Long. Disp. Rt. Fr.	20 Hz	±0.8"	1%	Side frame/axle relative motion
D20	Axle to Side Frame Long. Disp. Rt. Rear	20 Hz	±0.8"	1%	Side frame/axle relative motion
D21	Axle to Side Frame Long. Disp. Lf. Front	20 Hz	±0.8"	1%	Side frame/axle relative motion
D22	Axle to Side Frame Long. Disp. Lf. Rear	20 Hz	±0.8"	1%	Side frame/axle relative motion
D23	Steering Arm Disp. Rt. Side	20 Hz	±2"	1%	Right Side Steering Arm Displacement
D24	Steering Arm Disp. Lf. Side	20 Hz	±2"	1%	Left Side Steering Arm Displacement
D25	Spread of Side Frame Legs, Rt. Side	20 Hz	±2"	1%	Right Side Frame Leg Spread
D26	Spread of Side Frame Legs, Lf. Side	20 Hz	±2"	1%	Left Side Frame Leg Spread
R1	Bolster/Side Frame Twist Angle, Front	20 Hz	±2"	1%	Front Bolster/Side Frame Twist Angle
R2	Bolster/Side Frame Twist Angle, Rear	20 Hz	±2"	1%	Rear Bolster/Side Frame Twist Angle
A1	Carbody Vertical Accel. B-End Center	20 Hz	± 5 G	1%	Carbody pitch, bounce & roll accel. over center plate
A2	Carbody Vertical Accel. A-End Right	20 Hz	± 5 G	1%	Carbody pitch, bounce & roll accel. over center plate

TABLE 2-3. INSTRUMENTATION LIST (CONT'D)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY DESIRED	PURPOSE
A3	Carbody Vertical Accel. B-End Right	20 Hz	± 5 G	1%	Carbody pitch, bounce & roll accel. on bolster centerline
A4	Carbody Vertical Accel. A-End Right	20 Hz	± 5 G	1%	Carbody pitch, bounce & roll accel. on bolster centerline
A5	Carbody Lateral Accel. B-End	20 Hz	± 5 G	1%	Carbody lateral & yaw accel. on bolster centerline
A6	Carbody Lateral Accel. A-End	20 Hz	± 5 G	1%	Carbody lateral & yaw accel. on bolster centerline
A7	Carbody Vertical Accel. B-End Left	20 Hz	± 5 G	1%	Carbody center of rotation on bolster centerline
A8	Carbody Long. Accel. B-end Center	20 Hz	± 5 G	1%	Carbody longitudinal accel. to correlate with braking
A9	Fore Axle Brg. Pocket Vert. Accel. B-end Right	20 Hz	± 5 G	1%	Axle vertical & pitch accel. to correlate with profile
A10	Fore Axle Bearing Pocket Lat. Accel. B-end Right	20 Hz	± 5 G	1%	Axle lateral accel. to define hunting and to correlate with alignment
A11	Rear Axle Bearing Pocket Lat. Accel. B-end Right	20 Hz	± 5 G	1%	Axle lateral accel. to define hunting and to correlate with alignment
A12	Fore Axle Bearing Pocket Vert. Accel. B-end Left	20 Hz	± 5 G	1%	Axle vertical & pitch accel. and to correlate with profile
A13	Fore Axle Bearing Pocket Lat. Accel. A-end Right	20 Hz	± 5 G	1%	Axle lateral accel. to define hunting
A14	Rear Axle Bearing Pocket Lat. Accel. A-end Right	20 Hz	± 5 G	1%	Axle lateral accel. to define hunting
A15	Carbody Lateral Accel. B-End Center Roof Level	20 Hz	± 5%	1%	Carbody flexible torsional modes
A16	Carbody Lateral Accel. A-End Center Roof Level	20 Hz	± 5%	1%	Carbody flexible torsional modes
A17	Carbody Lateral Accel. Center of Car at Truck Level	20 Hz	± 5%	1%	Carbody flexible torsional modes
F1	BL-1 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation

TABLE 2-3. INSTRUMENTATION LIST (CONT'D)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY	PURPOSE
F2	BR-1 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation
F3	BL-2 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation
F4	BR-2 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation
F1-1	BL-1 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F1-2	BL-1 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F2-1	BR-1 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F2-2	BR-1 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F3-1	BL-2 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F3-2	BL-2 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F4-1	BR-2 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
F4-2	BR-2 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load
G1	BL-1 Rotary Pulse Generator #1	40 kHz	0-5 V	1/2000 rev.	Rotation of axle #1, used to specify strain gage position
G2	BL-2 Rotary Pulse Generator #2	40 kHz	0-5 V	1/2000 rev.	Rotation of axle #2, used to specify strain gage position
G101 to G116	Axle 1 Bending Measurements, 16 Strain Gages	500 Hz	1,150,000 in-lb	1%	Measure axle bending for L/V calculation
G201 to G216	Axle 2 Bending Measurements, 16 Strain Gages	500 Hz	1,150,000 in-lb	1%	Measure axle bending for L/V calculation
B1	Steering Arm Strain Gage	50 Hz		1%	Dresser truck steering arm forces
B2	Steering Arm Strain Gage	50 Hz		1%	Dresser truck steering arm forces

TABLE 2-3. INSTRUMENTATION LIST (CONT'D)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY DESIRED	PURPOSE
P1	BL-1A Wheel/ Side Frame Position Transducer "A"	50 Hz	2"	1%	Wheel/rail angle of attack
P2	BL-1B Rail/ Side Frame Position Transducer "B"	50 Hz	2"	1%	Wheel/rail angle of attack
P3	BL-1C Wheel/ Side Frame Position Transducer "C"	50 Hz	2"	1%	Wheel/rail angle of attack
P4	BL-1D Rail/ Side Frame Position Transducer "D"	50 Hz	2"	1%	Wheel/rail angle of attack
P5	BL-2A Wheel/ Side Frame Position Transducer "A"	50 Hz	2"	1%	Wheel/rail angle of attack
P6	BL-2B Rail/ Side Frame Position Transducer "B"	50 Hz	2"	1%	Wheel/rail angle of attack
P7	BL-2C Wheel/ Side Frame Position Transducer "C"	50 Hz	2"	1%	Wheel/rail angle of attack
P8	BL-2D Rail/ Side Frame Position Transducer "D"	50 Hz	2"	1%	Wheel/rail angle of attack
T1	B-1 Axle Torque (Gage 1A)	50 Hz	$\pm$30,000 in/lb	1%	Torque on axle due to wheel slip, long. creep forces
T2	B-1 Axle Torque (Gage 1B)	50 Hz	$\pm$30,000 in/lb	1%	Backup for T1
T3	B-2 Axle Torque (Gage 1A)	50 Hz	$\pm$30,000 in/lb	1%	Torque on axle due to wheel slip, long. creep forces
T4	B-2 Axle Torque (Gage 1B)	50 Hz	$\pm$30,000 in/lb	1%	Backup for T3
GR	Filtered Longitudinal Acceleration	1 Hz	$\pm$5 g		Measurement A8 low pass filtered to provide grade information

TABLE 2-4. INSTRUMENTATION BY TRUCK

Instrumentation Channel	ASF Ride Control (Type I Truck)	Dresser DR-1	Barber-Scheffel	Devine-Scales	MTS Maxiride	National Swing Motion	ACF Fabricated	Alusuisse	General Description
S1	●	●	●	●	●	●	●	●	Train parameters
S2	●	●	●	●	●	●	●	●	
S3	●	●	●	●	●	●	●	●	
S4	●	●	●	●	●	●	●	●	
C1	●	●	●	●	●	●	●	●	Coupler force and angle
C2	●	●	●	●	●	●	●	●	
C3	●	●	●	●	●	●	●	●	
C4	●	●	●	●	●	●	●	●	
D1	●	●	●			●	●		Secondary suspension vertical displacement
D2	●	●	●			●	●		
D3	●	●	●			●	●		
D4	●	●	●			●	●		
D5	●	●	●			●			Bolster/side frame
D6	●	●	●			●			
D7	●	●	●			●			
D8		●	●			●			
D9		●	●			●			
D10		●	●			●			
D11	●	●	●	●	●	●	●	●	Centerplate swivel
D12	●	●	●	●	●	●	●	●	
D13	●	●	●	●	●	●	●	●	Carbody roll
D14	●	●	●	●	●	●	●	●	
D15				●	●				Primary suspension vertical displacement
D16				●	●				
D17				●	●				
D18				●	●				
D19		●	●	●		●	●		Primary suspension longitudinal displace ment
D20		●	●	●		●	●		
D21		●	●	●		●	●		
D22		●	●	●		●	●		
D1								●	Spring Group Vertical Displacement
D2								●	
D3								●	
D4								●	
D19								●	Bolster/Leg Lateral Displacement
D20								●	
D21								●	
D22								●	
A1	●	●	●	●	●	●	●	●	Car rigid body modes
A2	●	●	●	●	●	●	●	●	
A3	●	●	●	●	●	●	●	●	
A4	●	●	●	●	●	●	●	●	
A5	●	●	●	●	●	●	●	●	
A6	●	●	●	●	●	●	●	●	
A7	●	●	●	●	●	●	●	●	
A8	●	●	●	●	●	●	●	●	

TABLE 2-4. INSTRUMENTATION BY TRUCK (CONT'D)

Instrumentation Channel	ASF Ride Control (Type I Truck)	Dresser DR-1	Barber-Scheffel	Devine-Scales	MTS Maxiride	National Swing Motion	ACF Fabricated	Alusuisse	General Description
A9	●	●	●	●	●	●	●	●	
A10	●	●	●	●	●	●	●	●	
A11	●	●	●	●	●	●	●	●	
A12	●	●	●	●	●	●	●	●	Axle accelerations
A13	●	●	●	●	●	●	●	●	
A14	●	●	●	●	●	●	●	●	
A15		●	●	●	●	●	●	●	
A16		●	●	●	●	●	●	●	Carbody flexible torsional modes
A17		●	●	●	●	●	●	●	
F1	●	●	●	●		●			
F2	●	●	●	●		●			
F3	●	●	●	●		●			Bearing adapter
F4	●	●	●	●		●			
F1-1	●	●	●			●			
F1-2	●	●	●			●			
F2-1	●	●	●			●			
F2-2	●	●	●			●			Line of action of bearing adapter vertical force
F3-1	●	●	●			●			
F3-2	●	●	●			●			
F4-1	●	●	●			●			
F4-2	●	●	●			●			
G1	●	●	●	●	●	●			Axle position
G2	●	●	●	●	●	●			
G3		●	●	●	●	●			RPG reset pulse
G4		●	●	●	●	●			
G101	●	●	●	●	●	●			
↓	*	*				*			
G116	●	●	●	●	●	●			Axle bending moments
G201	●	●	●	●	●	●			
↓	*	*				*			
G216	●	●	●	●	●	●			
P1	●	●	●	●	●	●	●		
P2	●	●	●	●	●	●	●		
P3	●	●	●	●	●	●	●		Angle of attack of wheel relative to rail on right side
P4	●	●	●	●	●	●	●		
P5	●	●	●	●	●	●	●		
P6	●	●	●	●	●	●	●		
P7	●	●	●	●	●	●	●		
P8	●	●	●	●	●	●	●		
T1	●	●	●	●	●	●			
T2	●			●	●	●			
T3	●	●	●	●	●	●			Axle torsion
T4	●			●	●	●			
GR	●	●	●	●	●	●	●	●	Measure grade
B1		●	●						Strain measurement on steering arms
B2		●	●						

*Channels G104, G108, G110, G114, G206, G207, G211 and G215 not used on this truck

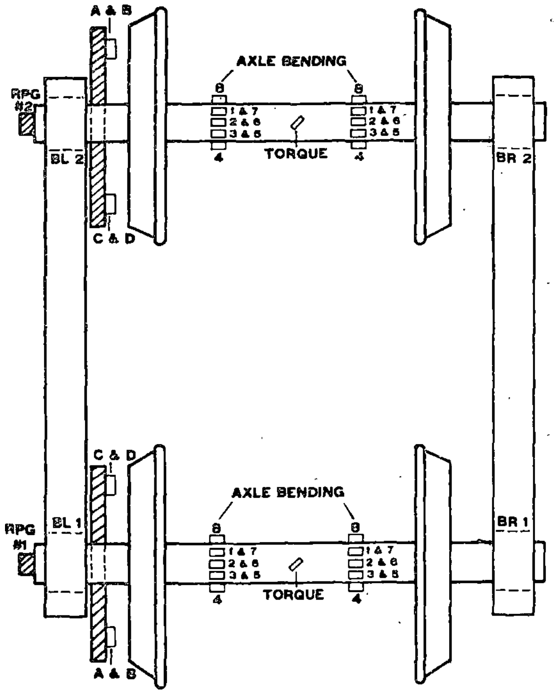


FIGURE 2-2. WHEEL/RAIL MEASUREMENT INSTRUMENTATION

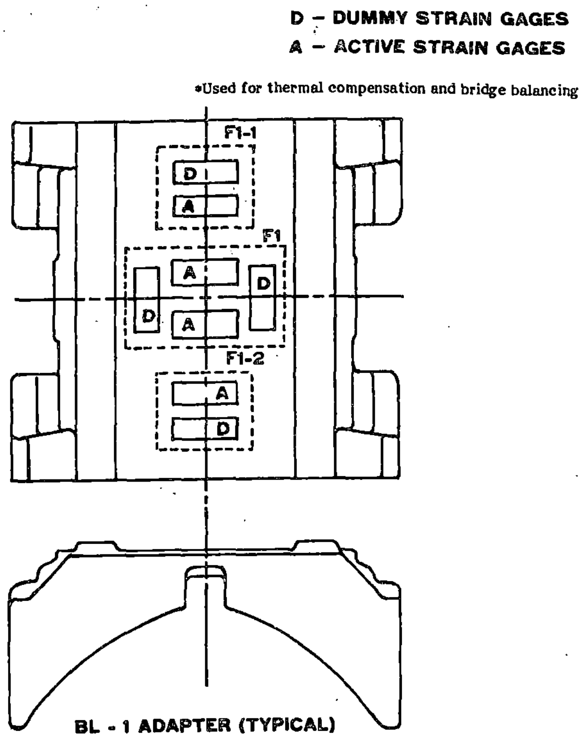


FIGURE 2-3. FORCE TRANSDUCER-BEARING ADAPTER

In Phase II testing, the vertical forces at the bearing adapter were measured by means of a modified strain gaged roller bearing adapter. The modified bearing adapter made use of three sets of strain gages. One full bridge strain gage was applied at the center. The other two strain gages (half bridges) were applied on the outside and inside edge of the bearing adapter as shown in Figure 2-3. A complete calibration was performed on each adapter recording all the three gages. From these data, it was possible to obtain a good estimate of the vertical force and the line of action of that vertical force.

For two of the secondary suspension trucks (Barber-Scheffel and National Swing Motion), the instrumented bearing adapter from the Type I truck test program was used. The bearing adapter on the Dresser DR-1 truck was modified and strain-gaged in a manner similar to the Type I bearing adapter. Both bearing adapters for Type I and Dresser DR-1 trucks were calibrated at the Transportation Test Center (Reference 5 and 11).

2.3.2 Wheel/Rail Position Measurements

Four eddy current transducers were used at one end of each axle of the leading truck to measure the relative position and angle of attack of the wheel relative to the rail. This concept is shown schematically in Figure 2-4. The transducers were mounted on bracketry which was attached to the side frame. Two of the transducers measured the side frame position relative to the rail and two of the transducers measured the side frame position relative to the wheel (see Figure 2-5). The difference between the corresponding two transducers gives the relative angle; the difference between the side frame to wheel and the side frame to rail angles results in the angle of attack.

2.3.3 Truck Measurements

Displacement and acceleration measurements were made on the truck to measure relative motion between truck components and acceleration on the axles. Figures 2-6 shows the accelerometer location on the B-end truck. The instrumentation consists of lateral accelerometers on each axle and vertical accelerometers at each end of the leading axle. The A-end of the truck had only lateral accelerometers on both axles. The displacement measurements are shown in Figures 2-7.

2.3.4 Truck to Carbody Measurements

Relative motions were measured between the truck bolster and carbody as shown in Figure 2-8. Two of the measurements (D11 and D12) were used to measure carbody rocking and the other two (D13 and D14) to measure truck swivel.

2.3.5 Carbody Measurements

The carbody was instrumented with accelerometers at the sill level and roof level to measure the longitudinal, vertical and lateral accelerations at different locations on the carbody. Figure 2-9 illustrates typical locations of these accelerometers.

2.3.6 Instrumented Coupler and Coupler Angle

A pair of instrumented couplers to measure longitudinal draw bar forces, developed and used on the AERO-TOFC II program, were modified at the Transportation Test Center. The modifications were to change measurement transducers to a 25,000-pound precision load cell and recalibration of the coupler. Details of the coupler construction are shown in Figure 2-10.

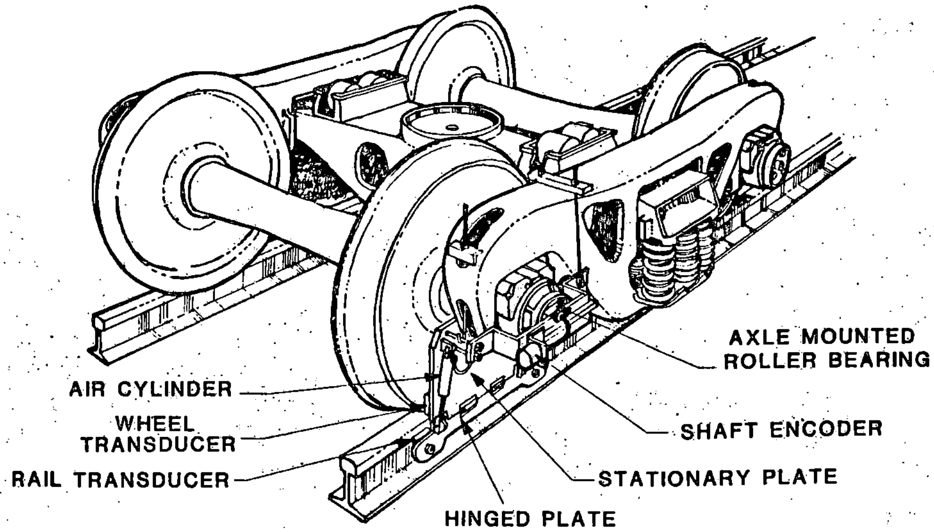
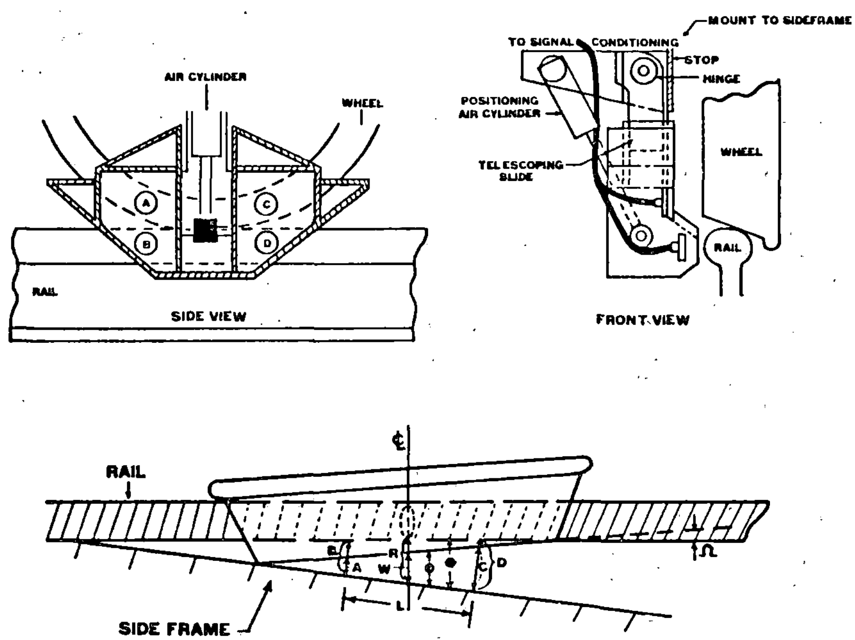


FIGURE 2-4. WHEEL/RAIL DISPLACEMENT MEASUREMENT FIXTURE



Θ = RAIL TO SIDEFRAME $\measuredangle \cong \frac{C-B}{L}$ RADIANS $= \frac{C-B}{L}$ (57.3) DEGREES

Φ = WHEEL TO SIDEFRAME $\measuredangle \cong \frac{D-A}{L}$ RADIANS $= \frac{D-A}{L}$ (57.3) DEGREES

Ω = WHEEL TO RAIL $\measuredangle = \Theta - \Phi$

R = SIDEFRAME TO RAIL DISTANCE $= \frac{B+C}{2}$

W = SIDEFRAME TO WHEEL DISTANCE $= \frac{A+D}{2}$

R/W = RAIL WHEEL POSITION = R-W

FIGURE 2-5. WHEEL/RAIL POSITION MEASUREMENT (BL-1 AXLE)

NOTE: See Table 2-3 for symbol explanation.

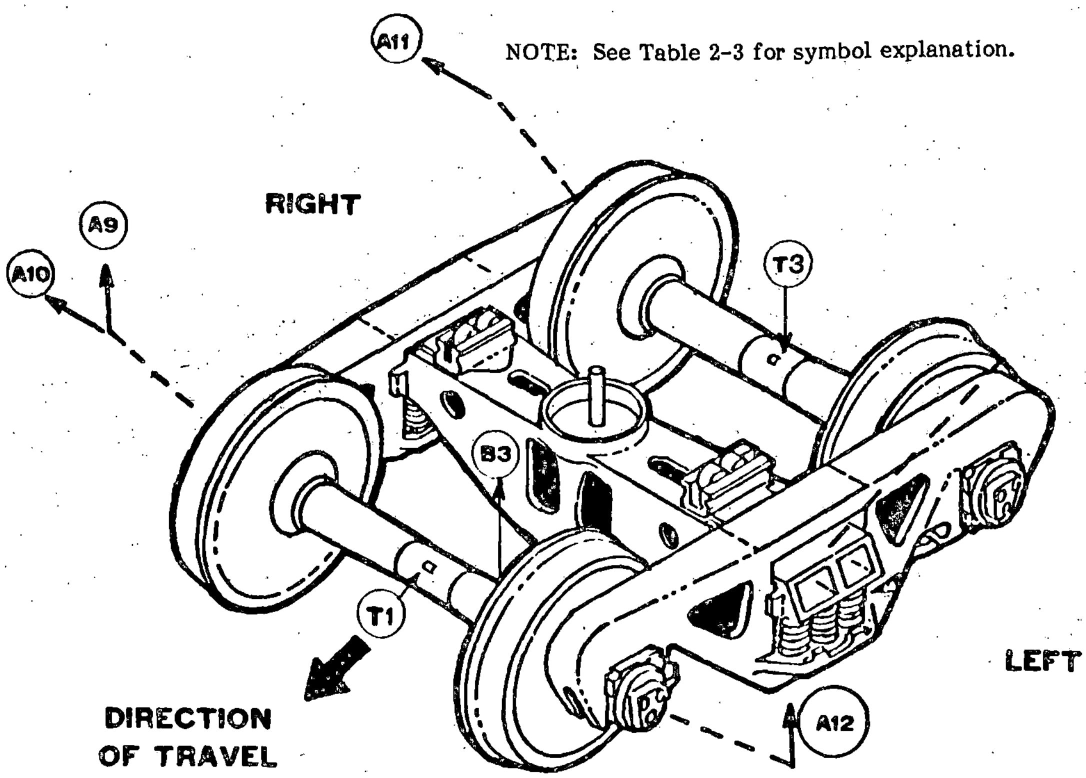


FIGURE 2-6. TRUCK INSTRUMENTATION LOCATION, B-END

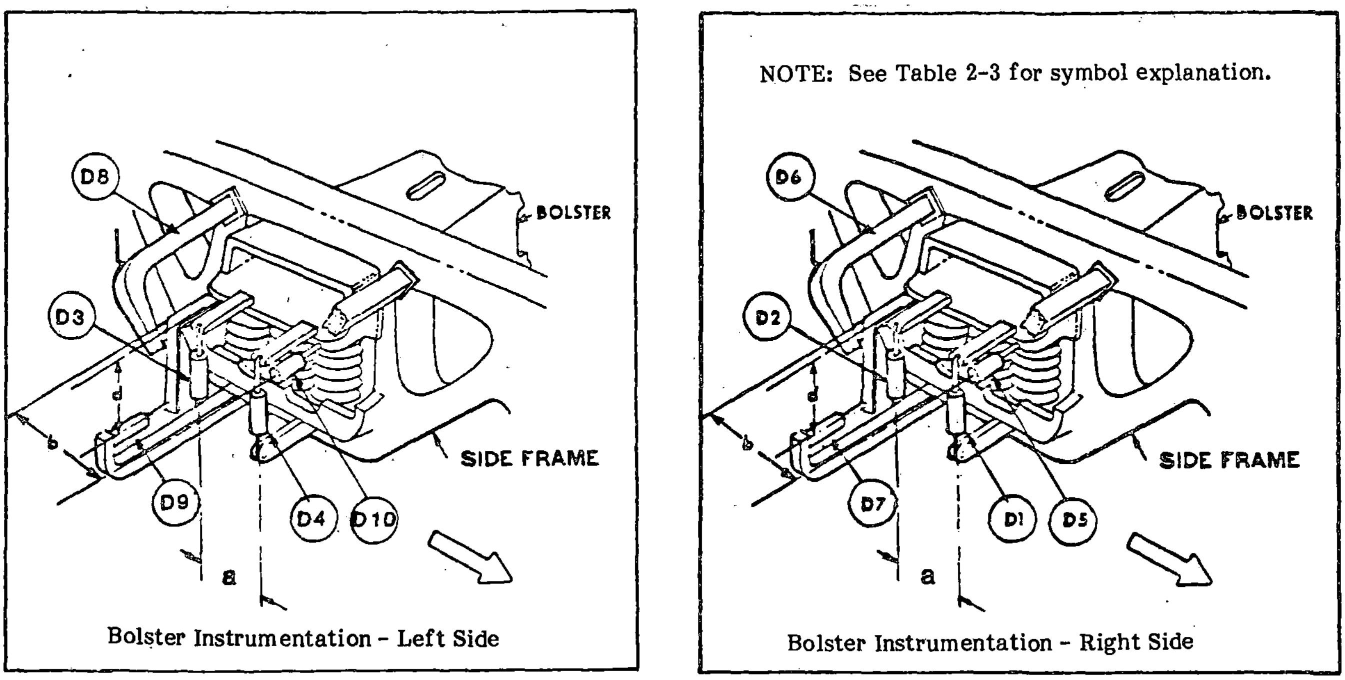


FIGURE 2-7. SPRING DISPLACEMENT, SECONDARY SUSPENSION TRUCKS

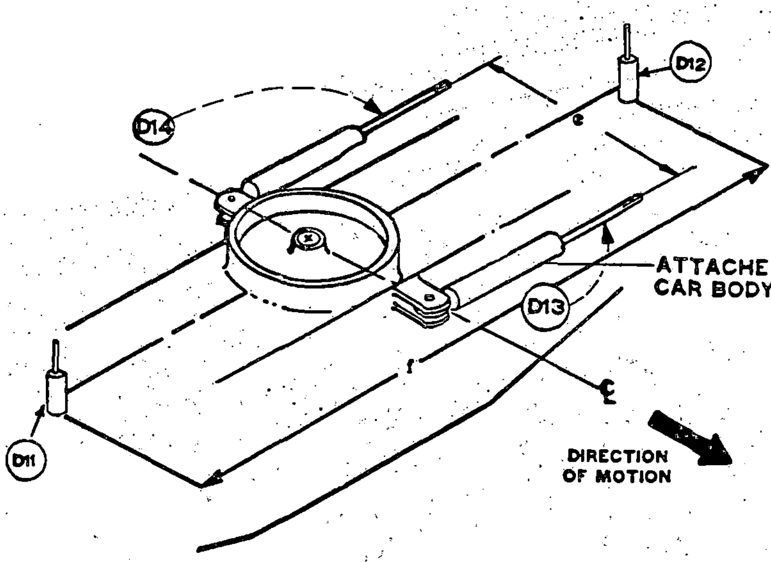


FIGURE 2-8. TRUCK/CARBODY RELATIVE MOTION INSTRUMENTATION

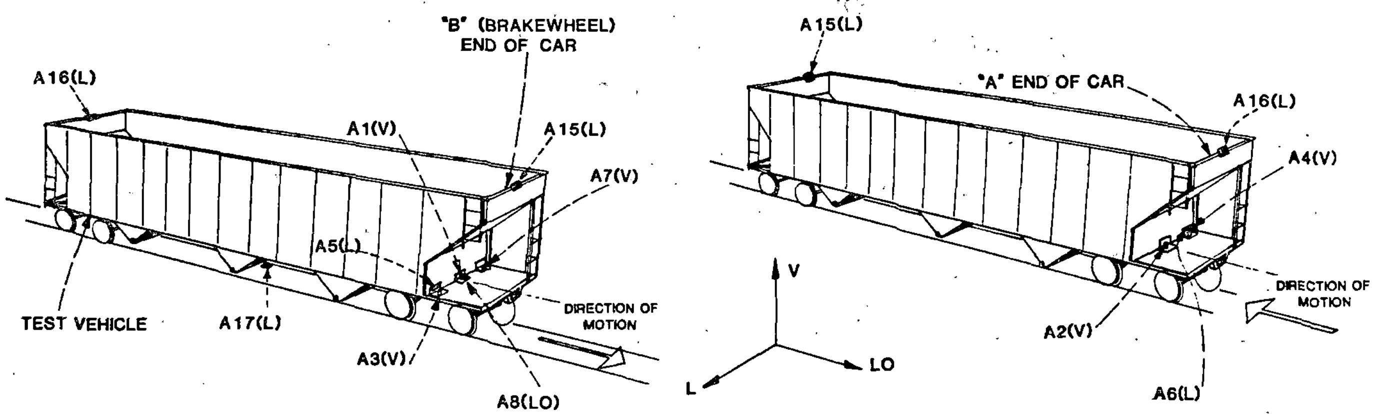


FIGURE 2-9. CARBODY INSTRUMENTATION

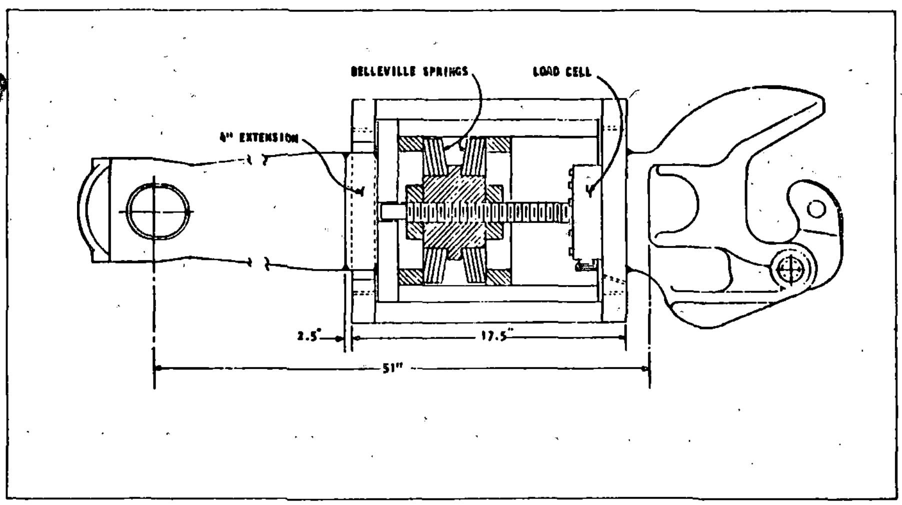


FIGURE 2-10. INSTRUMENTED COUPLER SCHEMATIC

The couplers consist of a pair of concentric cylinders connected by a load cell and belleville springs in series. The concentric cylinders allow the coupler to take torsional or bending loads without affecting the load cell, thus only measuring longitudinal force. Overload stops in the coupler allow it to take transient loads which exceed the capability of the load cells. For this application, the overload stops were set experimentally at 125% of load or approximately 31,000 lbs.

The calibration of the couplers involved standing them in a fixture and applying an incrementally increasing load from 0 to 37,000 lbs. A complete set of calibration curves for the instrumented couplers is contained in the Type I Truck Test Results Report (Reference 5). After completion of the calibration, the couplers were shipped to Las Vegas and installed, one on the leading buffer car and one on the trailing buffer car. Installing them on the buffer cars allows the test car to be switched from empty to loaded without moving instrumented couplers.

In addition to measuring coupler force with the instrumented couplers, bending beams were used to measure coupler angle relative to the carbody longitudinal axis. Two bending beams were placed on a bracket and mounted on the carbody at each coupler. The difference in motion between the two bending beams divided by the distance between them gives the sine of the coupler angle. This technique eliminates any error due to coupler longitudinal motion or roll motion.

2.4 TEST DESCRIPTION

2.4.1 Test Zones

The test sites used for testing freight car trucks consisted of mainline, branch, and yard track of Union Pacific's south central district and California division, (see Figures 2-11 to 2-16). Most test zones were within 20 miles of Las Vegas, Nevada. The five test zones are described in Table 2-5. Test zone 1 consisted of mainline track with curves of 1 to 6 degrees. Test zone 2 provided a section of tangent track, made of bolted jointed rail (BJR) over which high speed (up to 79 mph) tests were conducted. Test zone 3 was a section of yard track over which load equalization (track twist) tests were conducted. Test zone 4 was the Blue Diamond Spur, a section of class 2 branch track containing both tangent and curved track used to conduct harmonic roll tests. Test zone 5 was a section of continuously welded rail (CWR) and was used for lateral stability testing and for comparison with jointed rail tests.

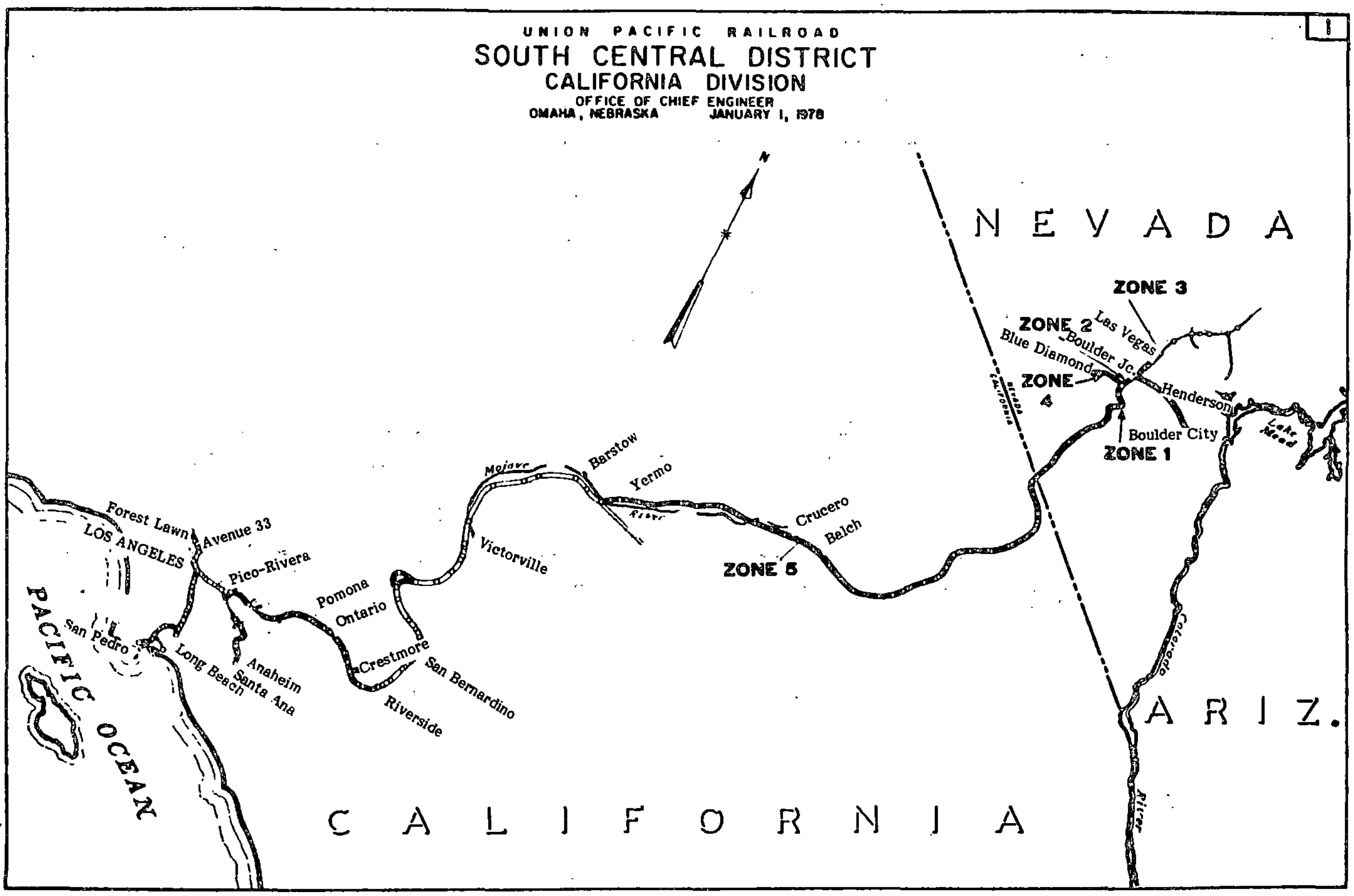


FIGURE 2-11. TEST ZONE LOCATIONS

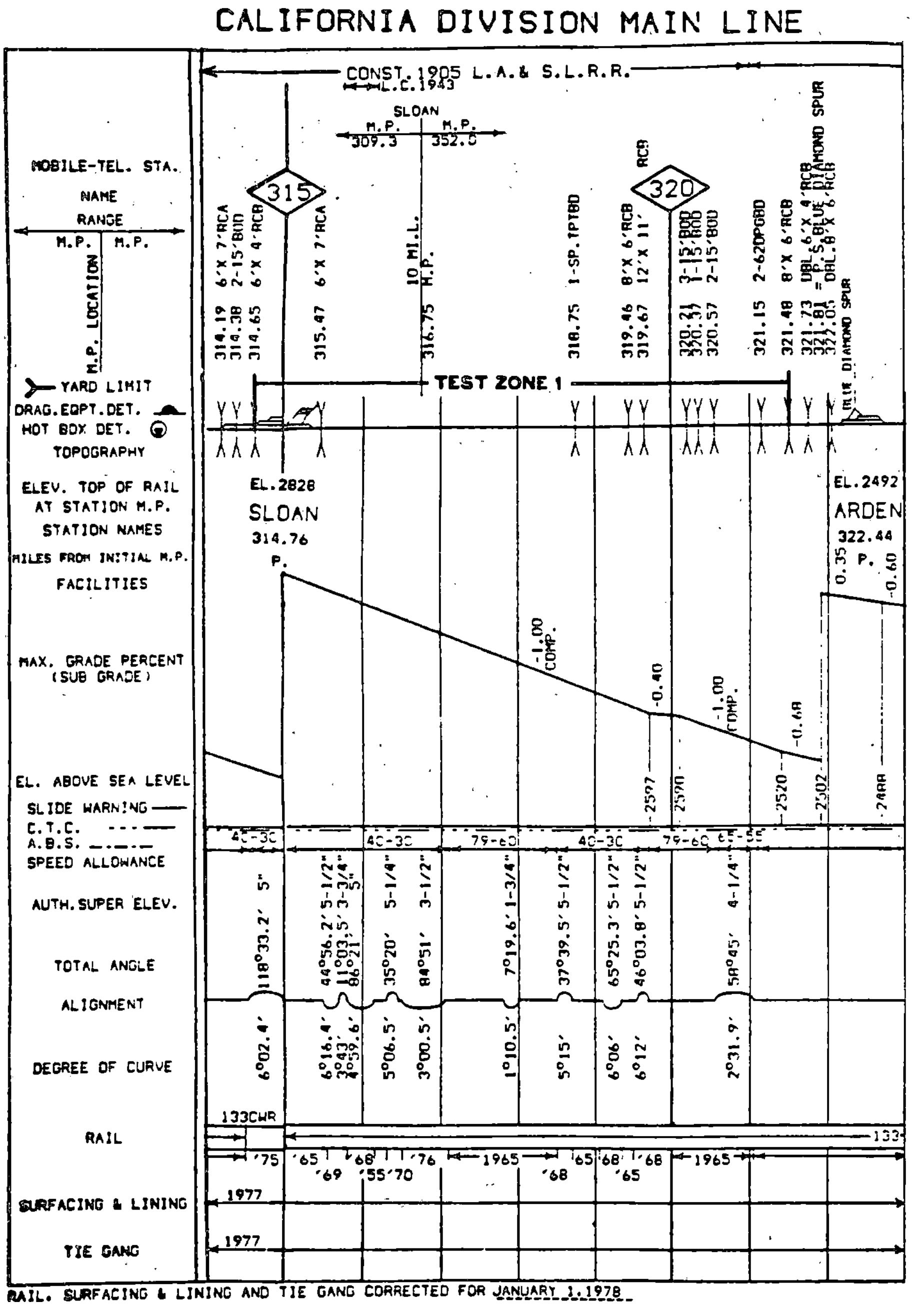


FIGURE 2-12. TRACK PROFILE - TEST ZONE 1

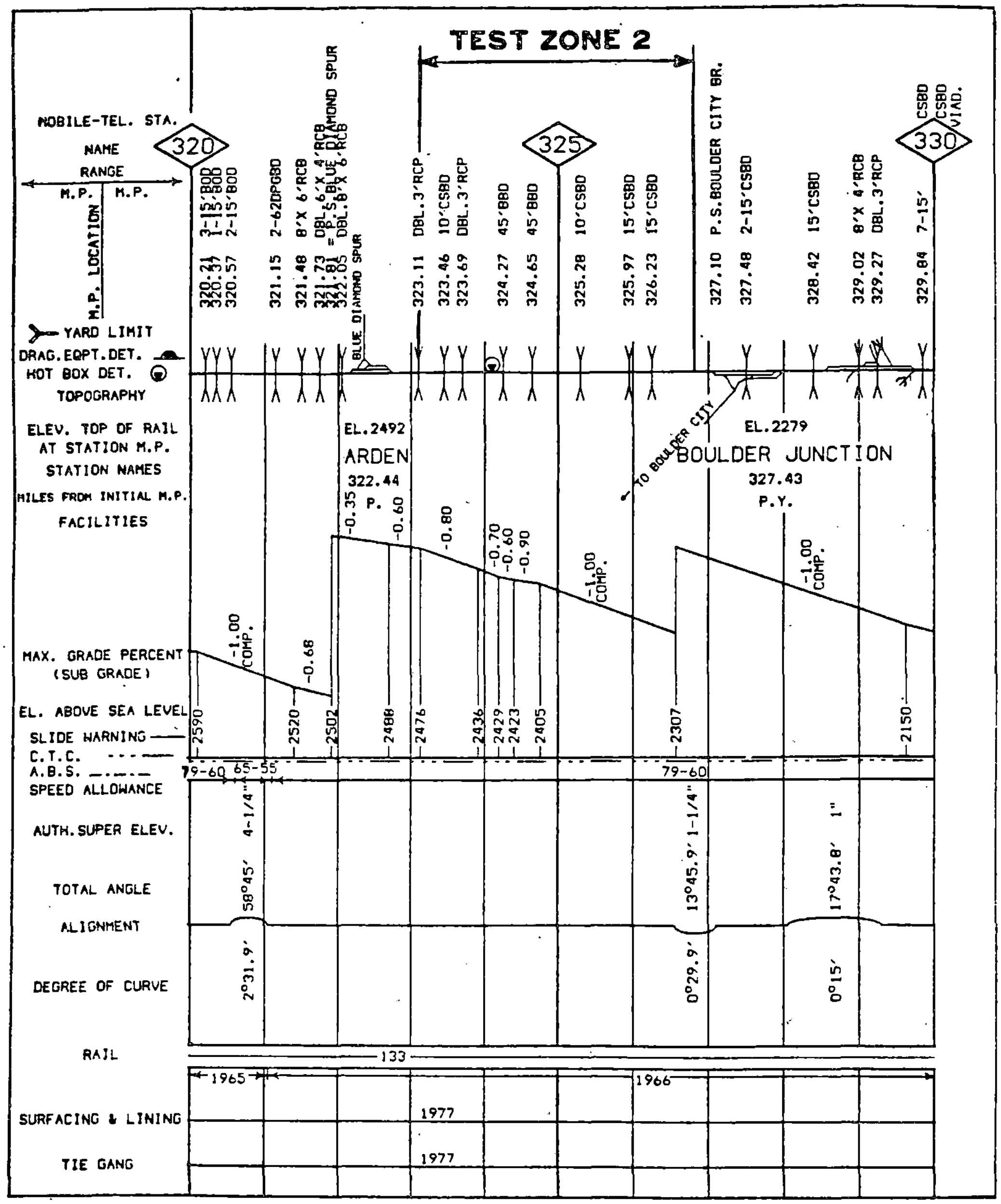


FIGURE 2-13. TRACK PROFILE - TEST ZONE 2

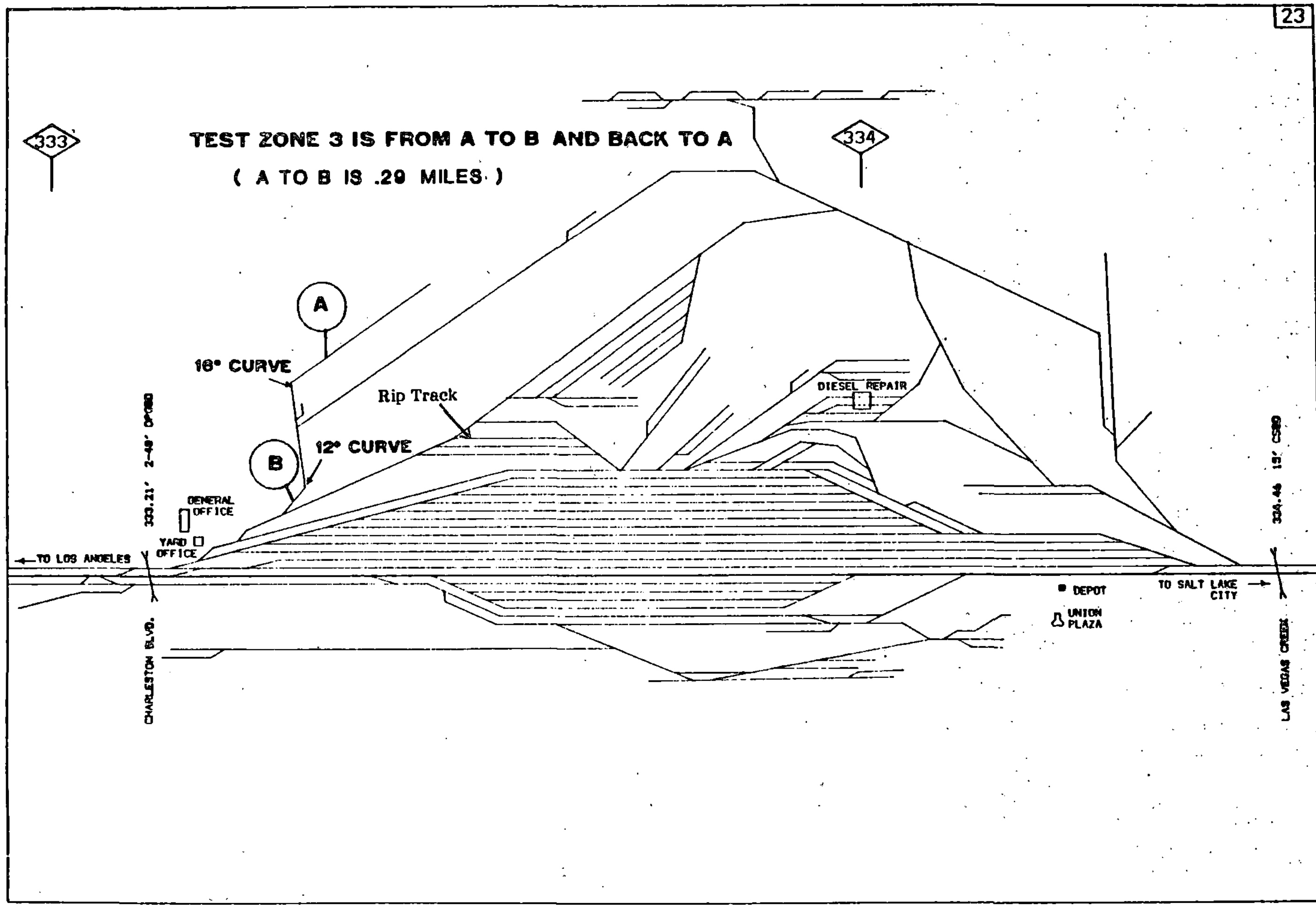


FIGURE 2-14. ZONE 3 - LAS VEGAS YARD

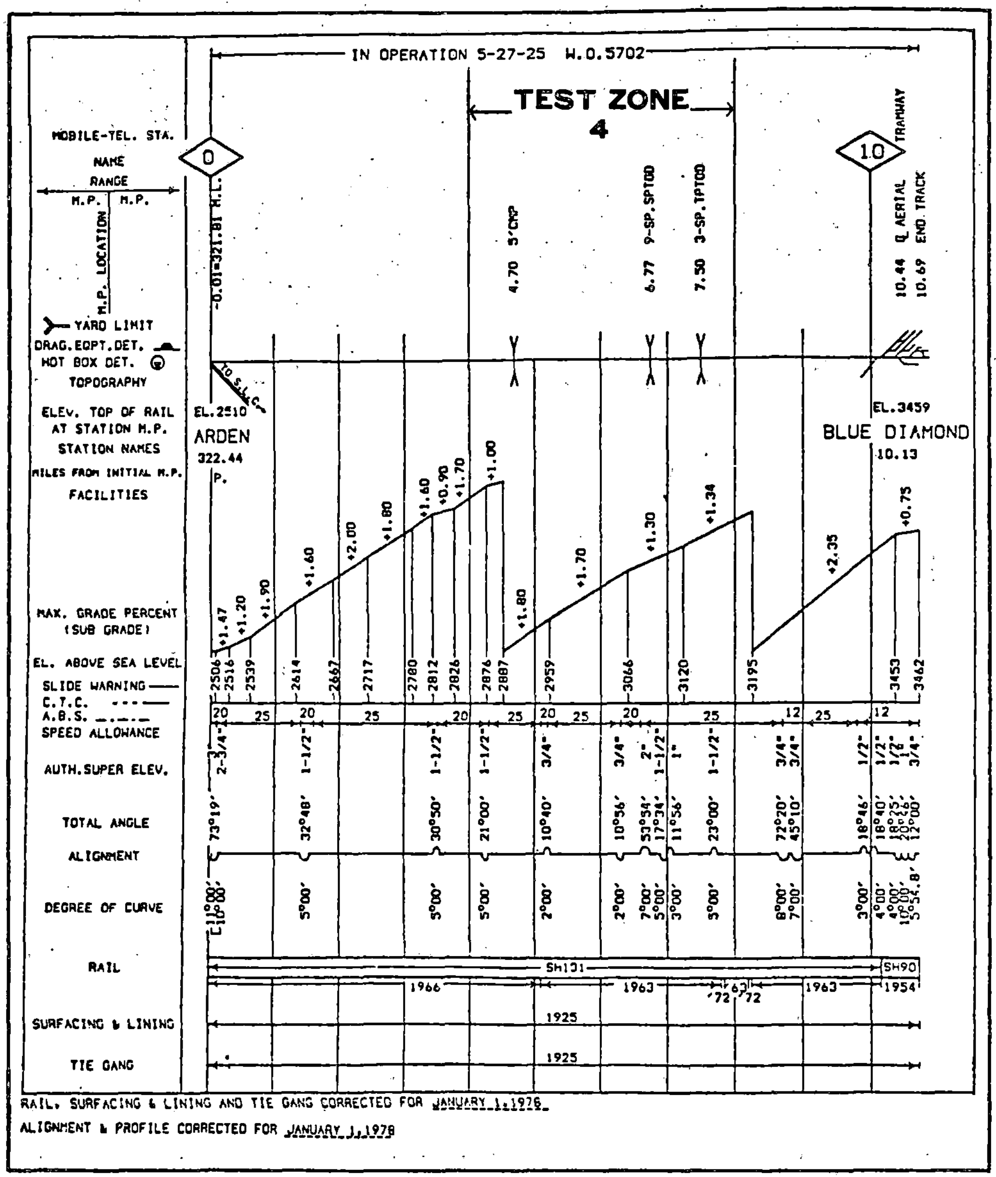
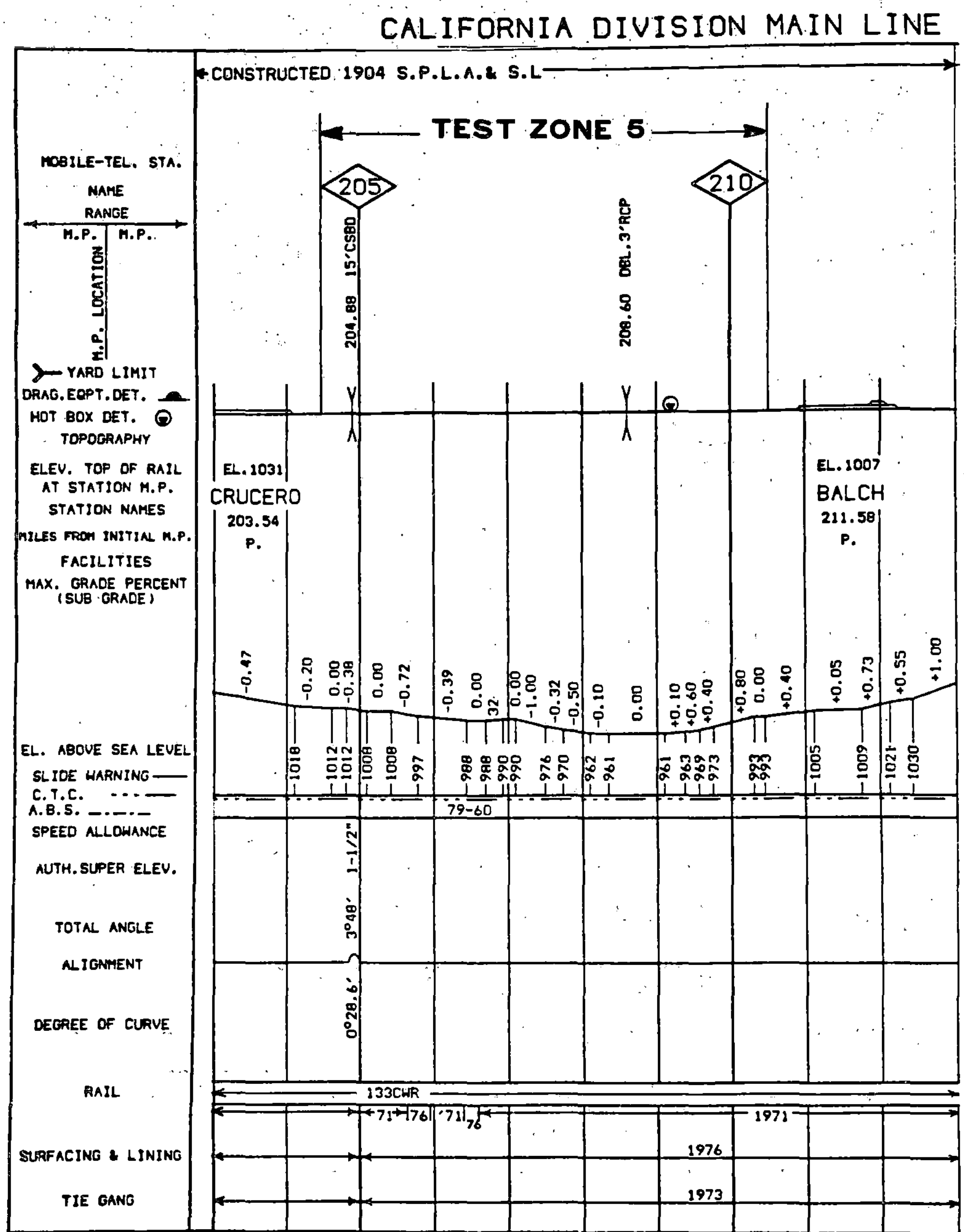


FIGURE 2-15. TRACK PROFILE - TEST ZONE 4



RAIL, SURFACING & LINING AND TIE GANG CORRECTED FOR JANUARY 1,1976

ALIGNMENT & PROFILE CORRECTED FOR JANUARY 1,1978

FIGURE 2-16. TRACK PROFILE - TEST ZONE 5

2.4.2 ALD Placement

An ALD system was developed so that test results obtained from a specific truck could be correlated with measured input track geometry and with test data from other trucks. The ALD system uses magnets imbedded in ties at the center of the track and a detector system on the railcar which senses the magnet as the car passes over it.

2.4.3 Track Geometry Measurements

To be able to correlate response measurements made on test vehicles with a known track input, the track geometry was measured twice during the program. The first set of measurements was taken during the first week in November 1978, and the second set was taken during December 1979. Both tests utilized the T-6 Track Geometry Survey Car (References 14 and 15). The ALD system was utilized during both track surveys.

The survey was conducted over the five test zones. Measurements of each test zone was taken at six-inch sample intervals as the survey car passed through the zone, normally once in each direction.

The track parameters which were reported are: right and left alignment, gauge, right and left profile, cross level, and curvature (degrees per 100 ft.). A digital tape of these parameters (including speed and ALD) had been supplied by the FRA to Wyle Laboratories in the form of both space curve and mid-chord offset. The track properties and the statistical analysis of the geometry parameters are given in Table 2-6. Typical power spectral densities are shown in Figures 2-17 through 2-32.

2.4.4 Test Train Consist

A standard test train consisting of a locomotive, instrumentation car, buffer car, test car, buffer car, and a caboose was established for all test runs and maintained throughout the test program. The buffers were open hopper cars. Prior to the start of testing, each buffer car was loaded with gravel. To provide for easier interchange of test cars, the instrumented coupler was placed on the test car end of each buffer car.

2.4.5 Union Pacific Mobile Laboratory Car 210

The Union Pacific Mobile Laboratory Car 210 was used as the instrumentation car for testing on all trucks. The hardware on the car was modified prior to the start of testing to include additional signal conditioning, magnetic tape, patch panel, and associated wiring to bring the system up to 96-channel capability.

TABLE 2-5. TEST ZONES

Site Number	Site Designation/Description		
1	Location	-	Arden to Sloan, NV
	Mileposts	-	321.5 to 314
	Track Type	-	Class 4 - Curved
	Rail Type	-	133-pound Jointed
	Speed Limit	-	40 mph
2	Location	-	Boulder Junction to Arden, NV
	Mileposts	-	326.5 to 321.5
	Track Type	-	Class 4 - Tangent
	Rail Type	-	133-pound Jointed
	Speed Limit	-	79 mph
3	Location	-	Las Vegas, NV
	Mileposts	-	Las Vegas, Yard
	Track Type	-	Curved, 16 Degrees
	Rail Type	-	Unknown
	Speed Limit	-	10 mph
	Distance	-	0.22 miles
4	Location	-	Blue Diamond Spur, NV
	Mileposts	-	1.5 to 8
	Track Type	-	Class 2 - Curved and Tangent
	Rail Type	-	131-pound Jointed
	Speed Limit	-	20 mph
5	Location	-	Balch to Crucero, CA
	Mileposts	-	210.5 to 204.5
	Track Type	-	Class 4 - Tangent
	Rail Type	-	133-pound CWR
	Speed Limit	-	79 mph

TABLE 2-6. TRACK PROPERTIES AND STANDARD DEVIATIONS OF TRACK ALIGNMENT, GAUGE, PROFILE, AND CROSS-LEVEL

Test Zone	Section	Milepost	Distance Processed foot	Rail Length foot	Alignment Left inch	Alignment Right inch	Alignment Average inch	Gauge inch	Profile Left inch	Profile Right inch	Profile Average inch	Cross Level inch
1	Sloan to Arden	314 to 321.5	39,424	39	0.144	0.145	0.136	0.23	0.115	0.126	0.114	-
	Arden to Sloan	321.5 to 314	39,424	39	0.145	0.147	0.137	0.23	0.106	0.124	0.109	-
2	Arden to Boulder Junction	321.5 to 326.5	26,112	39	0.084	0.083	0.069	0.142	0.114	0.106	0.101	0.172
	Boulder Junction to Arden	326.5 to 321.5	26,112	39	0.09	0.086	0.077	0.134	0.092	0.126	0.100	0.175
3	Las Vegas yard (East Bound)	-	1,536	39	0.936	0.926	0.922	0.414	1.236	1.150	1.183	-
	Las Vegas Yard (West Bound)	-	1,536	39	0.907	0.907	0.900	0.322	1.162	1.249	1.196	-
4	Blue Diamond Spur (East Bound)	1.5 to 8	34,304	33 & 39	0.182	0.179	0.173	0.183	0.132	0.153	0.129	-
	Blue Diamond Spur (West Bound)	8 to 1.5	33,792	33 & 39	0.141	0.144	0.134	0.181	0.127	0.151	0.126	-
5	Crucero to Balch	204.5 to 210.5	31,232	Welded	.084	0.083	0.070	0.136	0.090	0.084	0.082	0.285
	Balch to Crucero	210.5 to 204.5	31,744	Welded	0.077	0.088	0.072	0.133	0.083	0.102	0.088	0.285

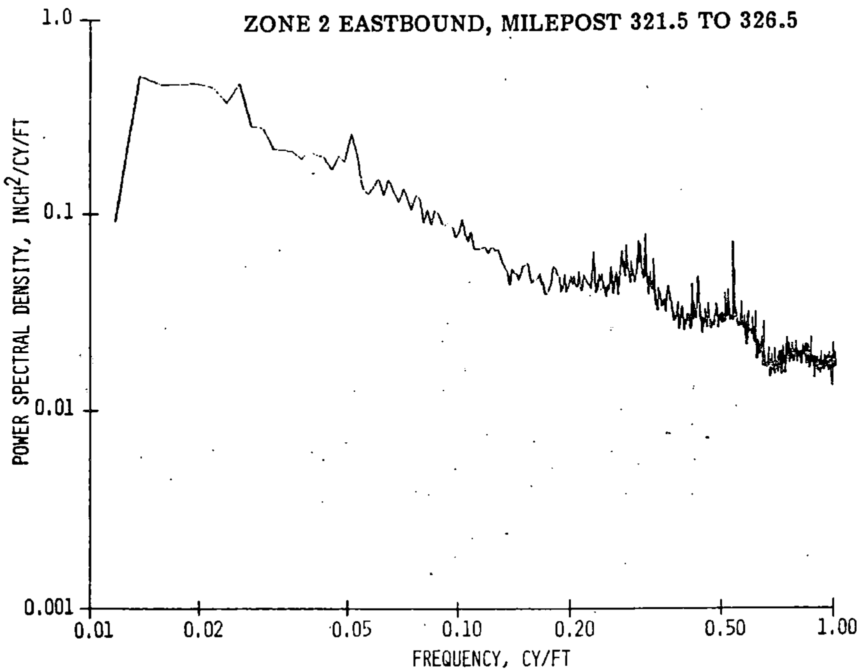


FIGURE 2-17. POWER SPECTRAL DENSITY - ZONE 2, LEFT ALIGNMENT

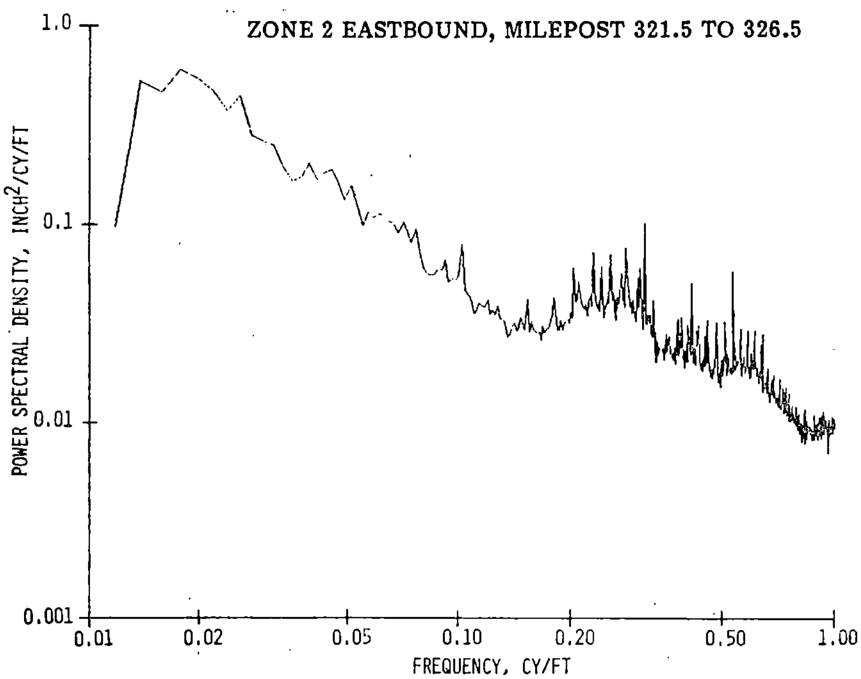


FIGURE 2-18. POWER SPECTRAL DENSITY - ZONE 2, RIGHT ALIGNMENT

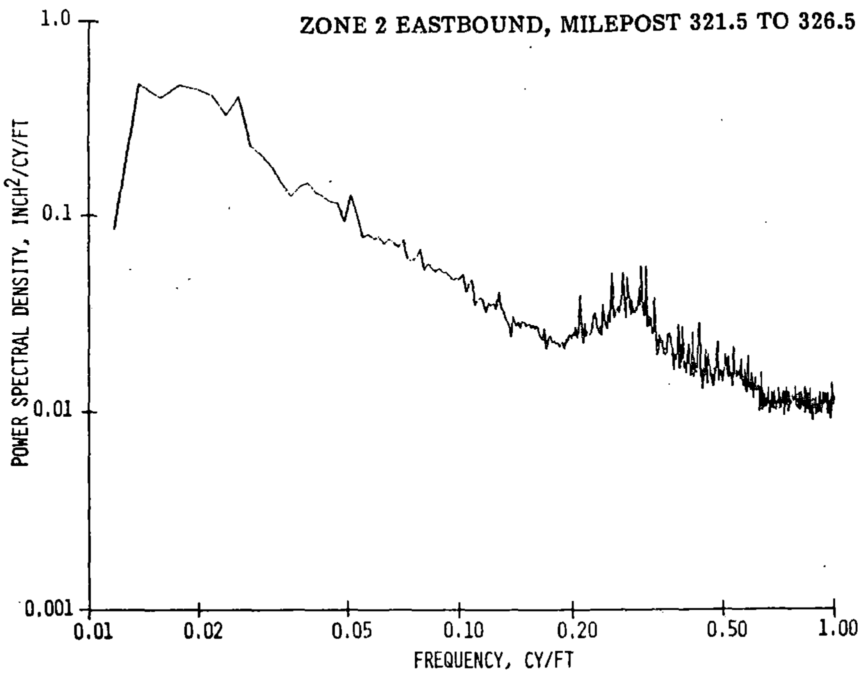


FIGURE 2-19. POWER SPECTRAL DENSITY - ZONE 2, AVERAGE ALIGNMENT

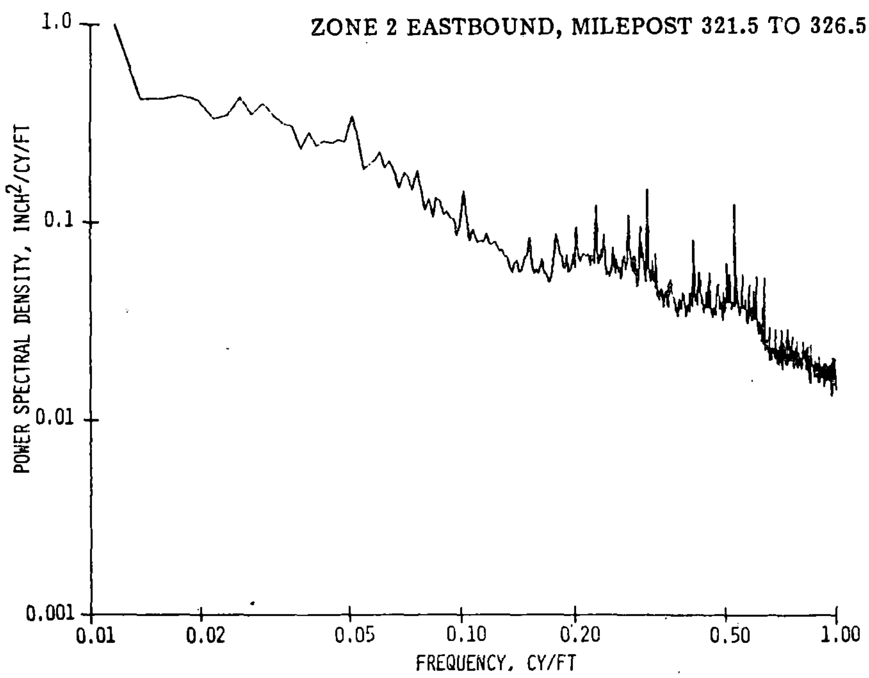


FIGURE 2-20. POWER SPECTRAL DENSITY - ZONE 2, GAUGE

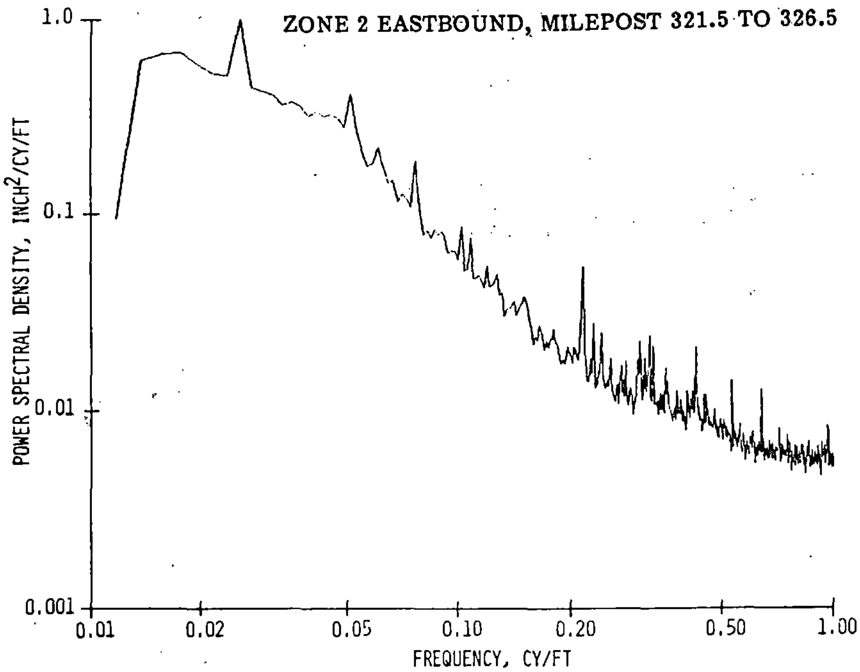


FIGURE 2-21. POWER SPECTRAL DENSITY ZONE 2, LEFT PROFILE

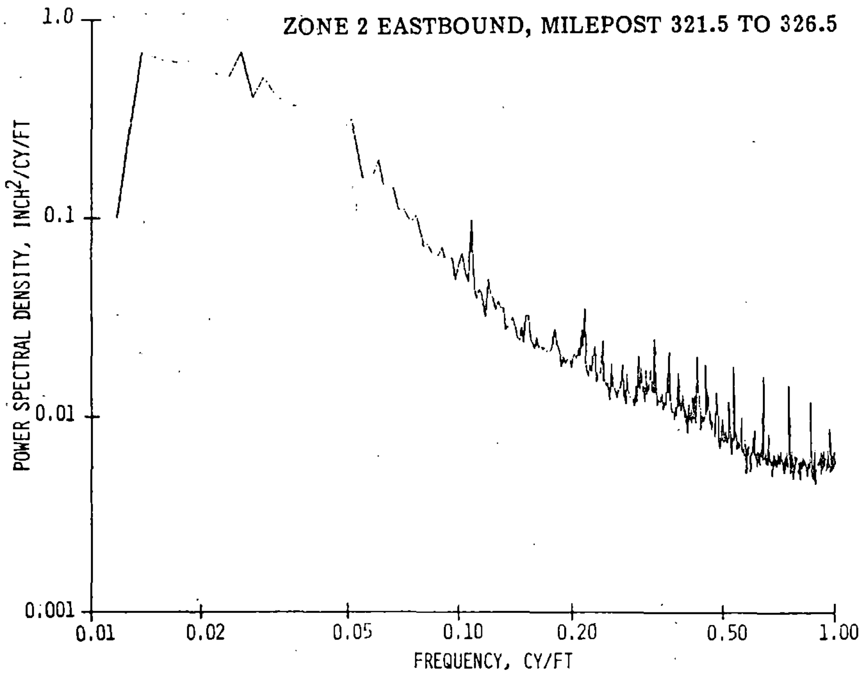


FIGURE 2-22. POWER SPECTRAL DENSITY - ZONE 2, RIGHT PROFILE

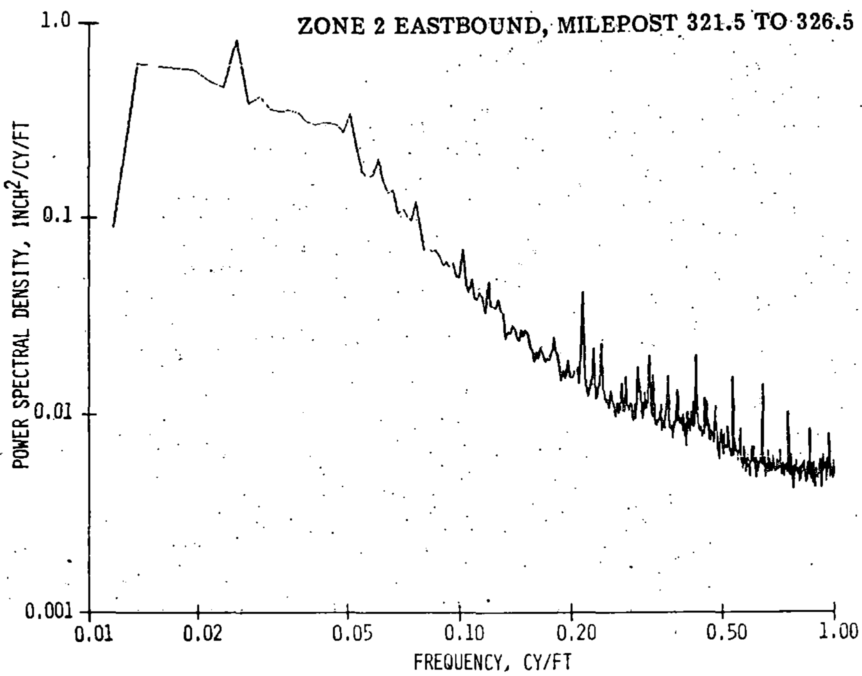


FIGURE 2-23. POWER SPECTRAL DENSITY - ZONE 2, AVERAGE PROFILE

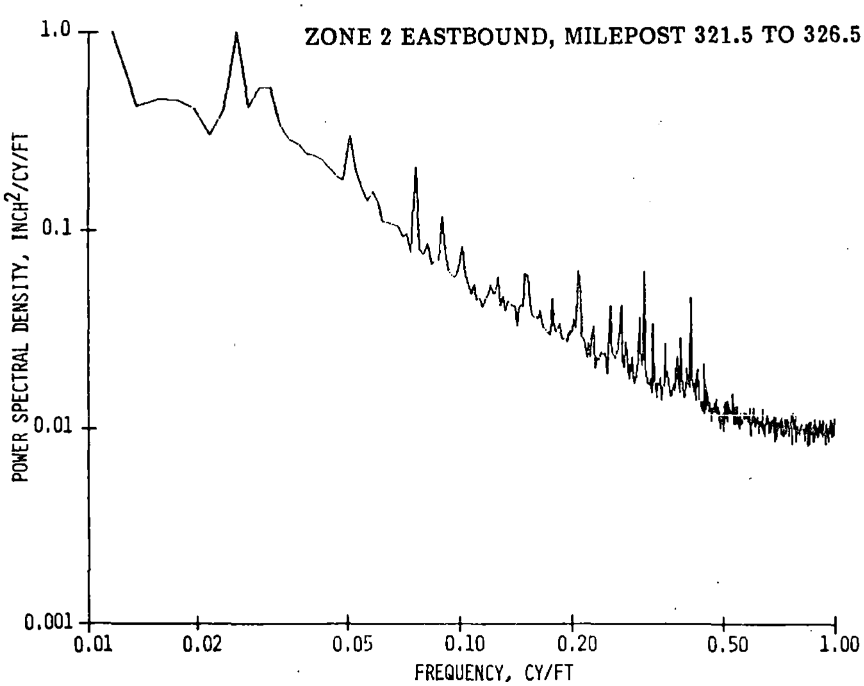


FIGURE 2-24. POWER SPECTRAL DENSITY - ZONE 2, CROSSLEVEL

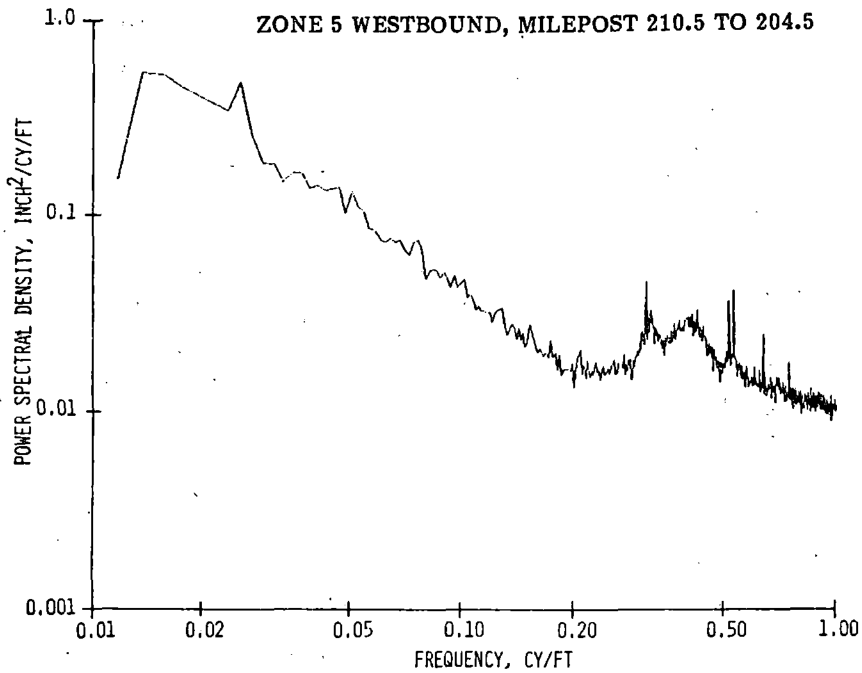


FIGURE 2-25. POWER SPECTRAL DENSITY - ZONE 5, LEFT ALIGNMENT

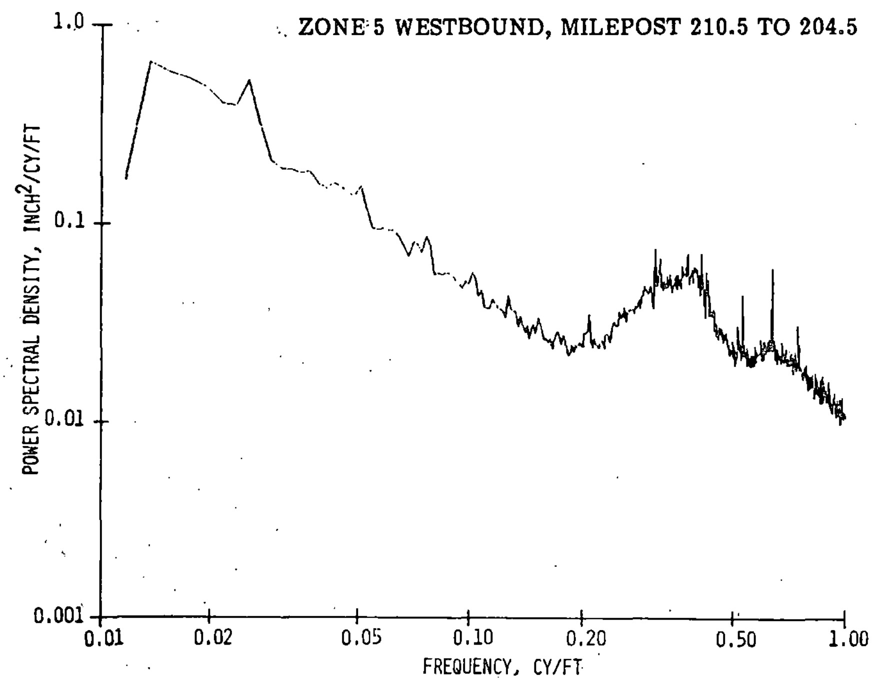


FIGURE 2-26. POWER SPECTRAL DENSITY - ZONE 5, RIGHT ALIGNMENT

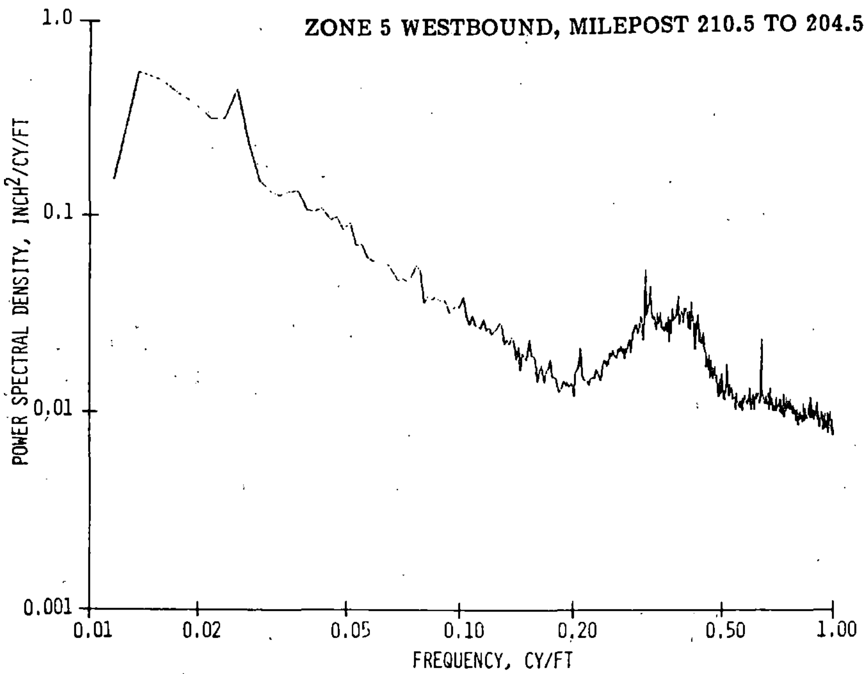


FIGURE 2-27. POWER SPECTRAL DENSITY - ZONE 5, AVERAGE ALIGNMENT

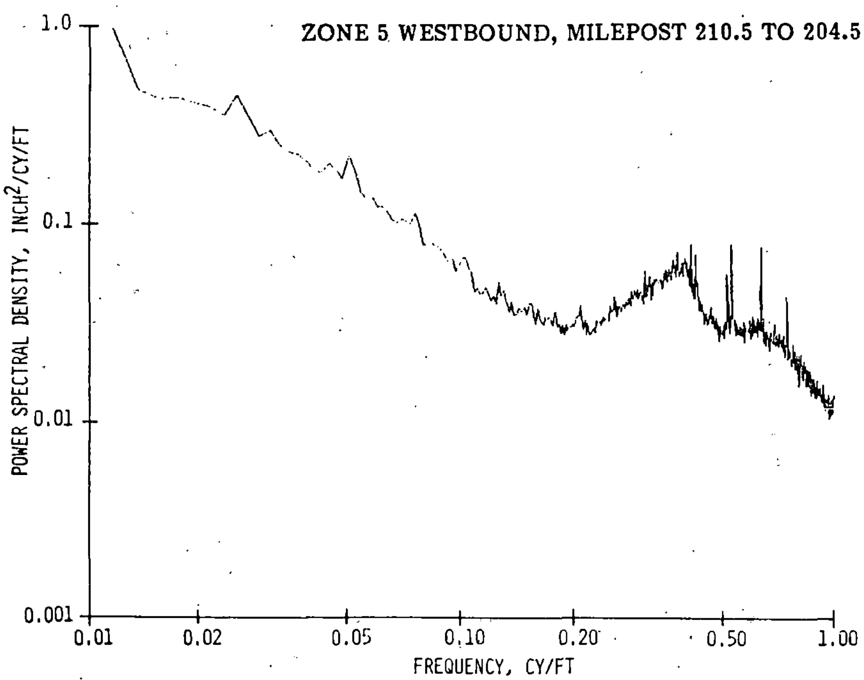


FIGURE 2-28. POWER SPECTRAL DENSITY - ZONE 5, GAUGE

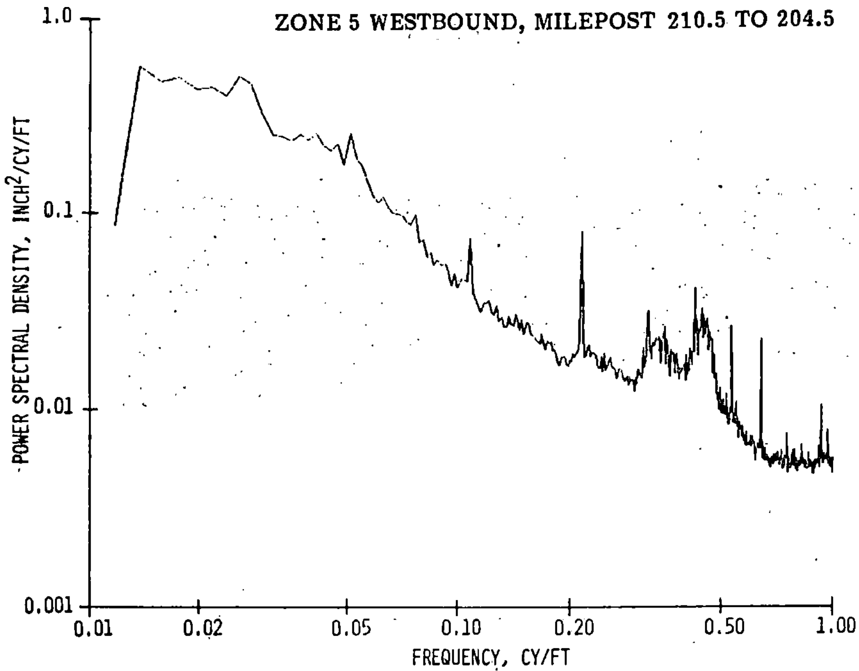


FIGURE 2-29. POWER SPECTRAL DENSITY - ZONE 5, LEFT PROFILE

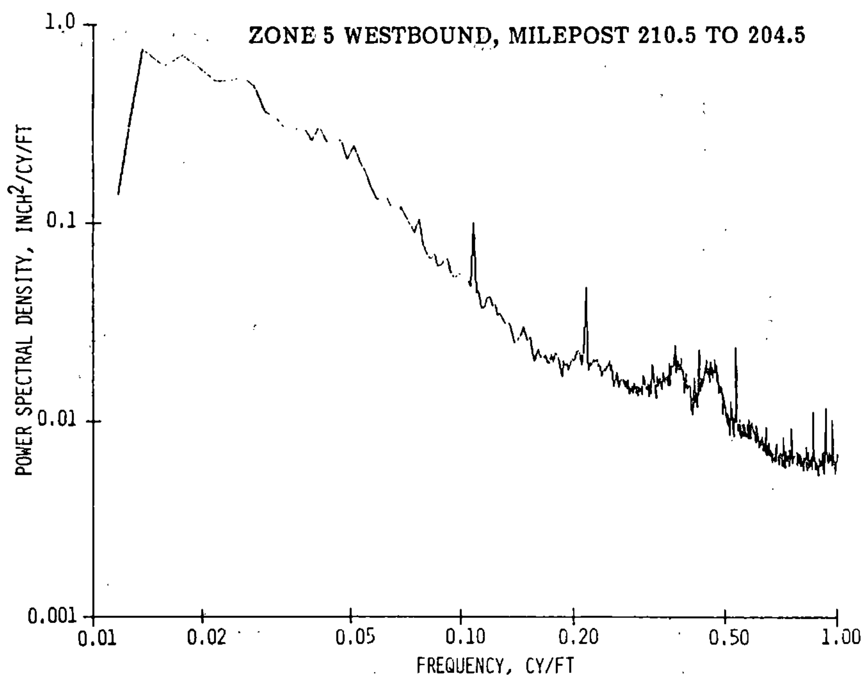


FIGURE 2-30. POWER SPECTRAL DENSITY - ZONE 5, RIGHT PROFILE

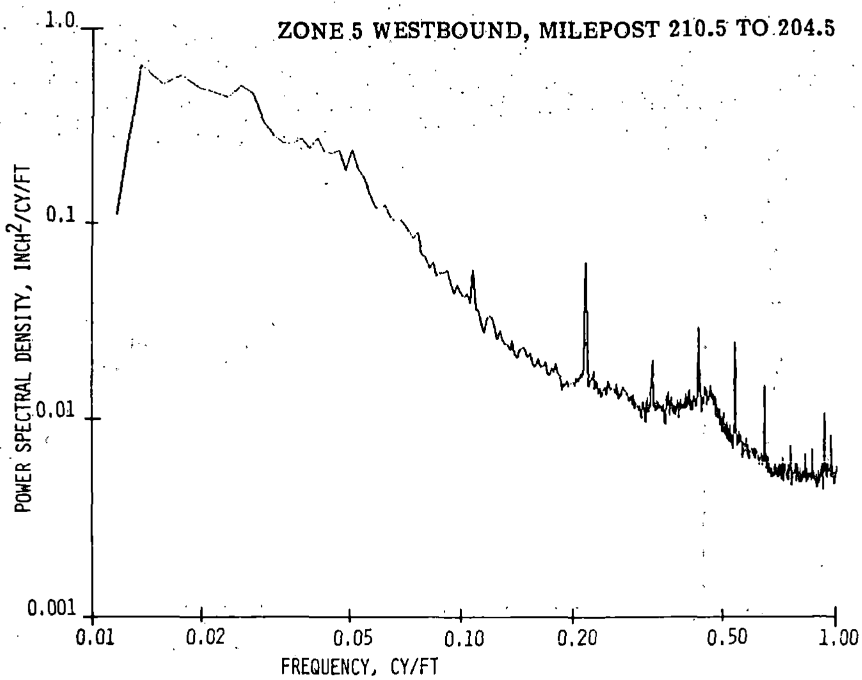


FIGURE 2-31. POWER SPECTRAL DENSITY - ZONE 5, AVERAGE PROFILE

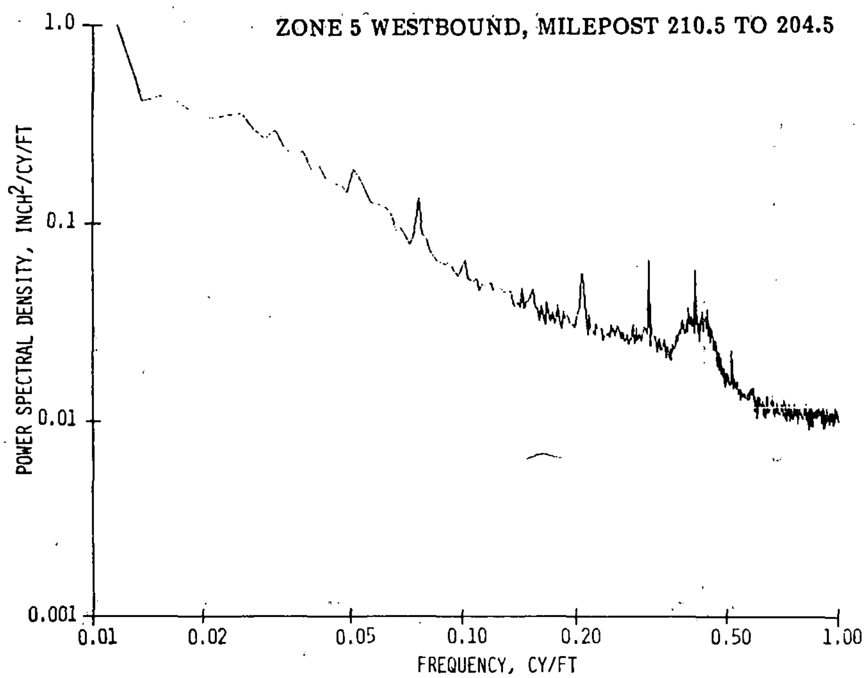


FIGURE 2-32. POWER SPECTRAL DENSITY - ZONE 5, CROSSLEVEL

2.5 TYPE I TRUCK TESTING

Testing on the Type I truck was conducted on two 100-ton ASF Ride Control trucks using AAR Standard 1:20 taper, new wheel profiles. Prior to the start of Type I truck testing, the trucks were in revenue service as part of the TDOP Phase II Wear Data Collection Program. At the completion of testing, the trucks were returned to that program. Before the instrumentation was installed, all bearing seals of the two ASF Ride Control trucks were examined to verify that they were the same on all wheels. The characteristics of the ASF Ride Control truck is given in Table 2-7.

The carbody type used for this test program was the 100-ton open hopper car. The 100-ton hopper car was chosen because it is representative of the higher capacity cars being placed into service today. Two carbodies from the same series were instrumented, one empty and one loaded. Table 2-8 gives the carbody characteristics.

Instrumentation for the test program consisted of 92 data channels. Fifty of the channels were used to obtain data for the computation of lateral and vertical forces (L/V) at the wheel/rail interface. The basic approach taken to measuring L/V was the axle-bending technique. The vertical forces at the bearing adapter were measured using strain-gaged bearing adapters. Forty-two of the 92 channels of data listed in Table 2-4 provided measurements of rigid body car motions, longitudinal coupler forces, truck/carbody relative displacements, and angle of attack. All 92 channels of data were recorded on all test runs in each regime tested.

Most of the testing was conducted over curved track on Union Pacific's main line south of Las Vegas, Nevada. Detailed profile of the curving zone is given in Figure 2-33. However, some high speed runs were made over main line tangent jointed track. In addition, low speed runs were made over the Blue Diamond Spur and on yard track in Las Vegas.

As discussed earlier, the primary objective of the Type I test program was to measure truck performance for the curve negotiation regime. This was accomplished by testing the truck through a series of mainline curves and measuring the response characteristics. The tests were conducted near equilibrium speed in both an uphill and downhill direction. The uphill tests were also conducted at below and above equilibrium speed.

Secondary objectives were to acquire hunting, rolling resistance, and load equalization (track twist) data. The hunting data were acquired from high speed sweep and dwell runs on an empty carbody. The rolling resistance data were obtained by running at several constant speeds in both the uphill and downhill directions and measuring coupler forces, speed, and throttle settings. These test runs were conducted on the fully loaded car only. Load equalization data were acquired by conducting low speed test runs over a section of yard track. For these tests, the locomotive pulled the car through the test zone and then pushed it back through the zone. The test program was run in a sequence of two test series: the first series dealt with the unloaded carbody; the second with the loaded carbody.

2.6 TYPE II TRUCK TESTING

The purpose of the Type II truck test program was to obtain performance data on several Type II (premium) freight car trucks in order to characterize their operational behavior. A set of clearly defined criteria for the selection of Type II trucks for testing was decided upon in consultation with the TDOP Phase II consultants, a group of representatives from the operating railroads and equipment suppliers (Reference 12). A systematic selection process was undertaken to ensure that the trucks selected would be representative of the state-of-the-art in truck design. Of the seven trucks selected for testing, the Dresser DR-1, National Swing Motion, Devine-Scales, and Barber Scheffel were taken from the TDOP Phase II Wear Data Collection Program. The other three, the MTS Maxiride 100, ACF Fabricated, and Alusuisse, were purchased new for performance testing. The new trucks acquired approximately 100 miles of wear during checkout and travel to the test zones. The four trucks acquired from the Wear Data Collection Program had from 59,000 to 90,000 miles of service in unit coal train service.

The same 100-ton hopper cars were used with the 100-ton Type II trucks as were used in the testing of Type I trucks. A 70-ton hopper car was used for the Alusuisse truck. Two carbodies from the same series were instrumented, one empty and one loaded. See Table 2-8 for the characteristics of the carbodies. All trucks except the Alusuisse were tested fully loaded and empty. The Alusuisse was tested with a loaded carbody only. The five track types used for testing were mainline tangent (both jointed and continuous welded rail), mainline curved, class 2 branch line, and yard track (see Table 2-5).

2.6.1 Wheelsets

All Type II trucks tested except the Alusuisse used the new Canadian National (CN) wheel profile. This profile approximates the flange condition of worn wheels and has better steering characteristics than an AAR 1:20 profile. The position of the TDOP rail industry consultants was that the CN wheel would provide the best test data for the program. The radial truck manufacturers also felt that the CN wheel profile would operate satisfactorily on their trucks. A comparison between the AAR and CN profiles is shown in Figure 2-34. The heat treated, class J, 36-inch wheelsets were strain-gaged and calibrated. These wheelsets were used to measure axle bending forces on each truck except ACF Fabricated and the Alusuisse trucks. The wheelsets were used under the ACF truck but axle bending measurements were not taken. The 70-ton Alusuisse truck had its own wheelsets, with a new 33-inch AAR 1:20 taper wheel profile.

2.6.2 Trucks Tested

The seven trucks were categorized according to their characteristics of suspension, i.e., primary, secondary, and others. They were further classified according to whether the connection between their side frames and bolsters was of rigid or radial construction. The trucks were also grouped according to whether they have unique load supporting devices. A summary of the characteristic parameters of each truck is given in Table 2-7. The following paragraphs briefly describe each truck.

TABLE 2-7. TRUCK CHARACTERISTICS

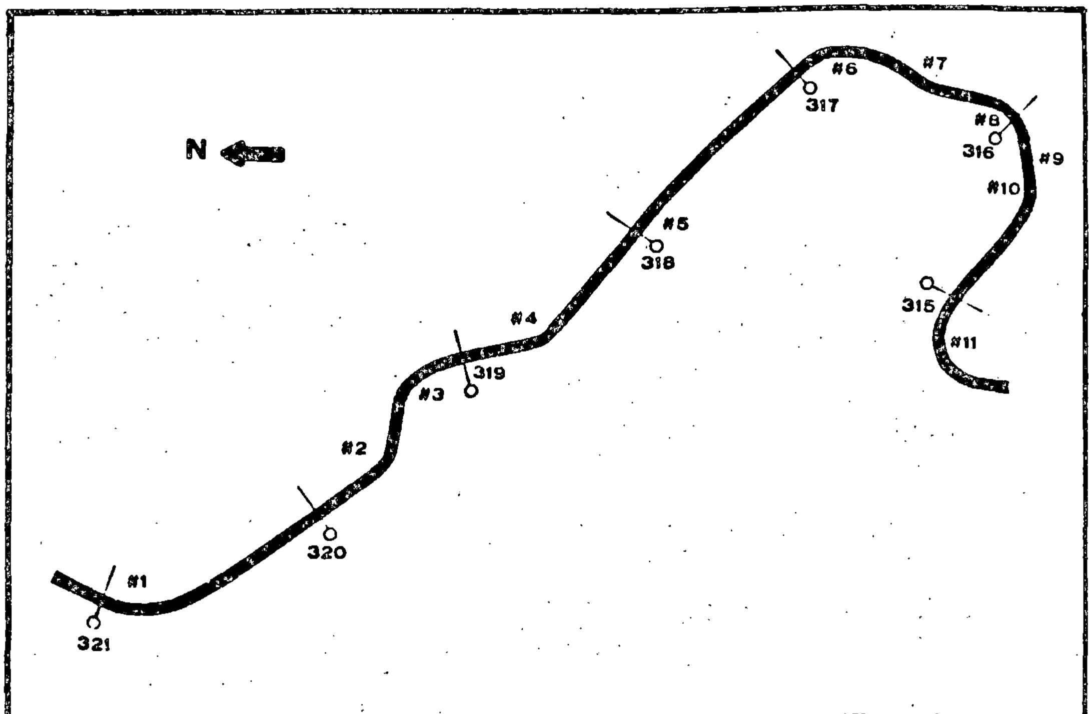
	TYPE I	TYPE II						
	ASF Ride Control	Dresser DR-1**	National Swing Motion**	Barber-Scheffel**	Maxiride*	Devine-Scales*	ACF**	Alusuisse***
Wheel Base, in	70.00	70.00	72.00	75.00	72.44	70.00	70.00	-
Spring Group						-	-	-
In	8-D5	6-D5	6-D6	7-D5	†			
Out	8-D5	7-D5	6-D7	6-D5	†	-	-	-
Center Plate Diameter, in	16.00	16.00	16.00	16.00	14.00 (spherical)	16.00	16.00	13.625 (spherical)
Side Bearer Clearance, in	0.25	0.625	0.28	0.181	None	0.1875	Constant Contact	Constant Contact
Snubbing:	Constant	Load Dependent	Load Dependent	Load Dependent	Load Dependent (non standard)	Load Dependent	Hydraulic Dampers	Load Dependent (Leaf Spring Friction)
Weight, lb	10,540	11,810	11,425	11,500	10,428	12,000	10,600	-
Wheel Diameter, ft	3.00	3.00	3.00	3.00	3.00	3.00	3.00	2.75

* Primary suspension trucks
** Secondary suspension trucks
*** Primary + secondary suspension trucks
† Vertical Spring rate (per car), lb/in
22,100 (empty)
161,100 (loaded)

TABLE 2-8. CARBODY CHARACTERISTICS

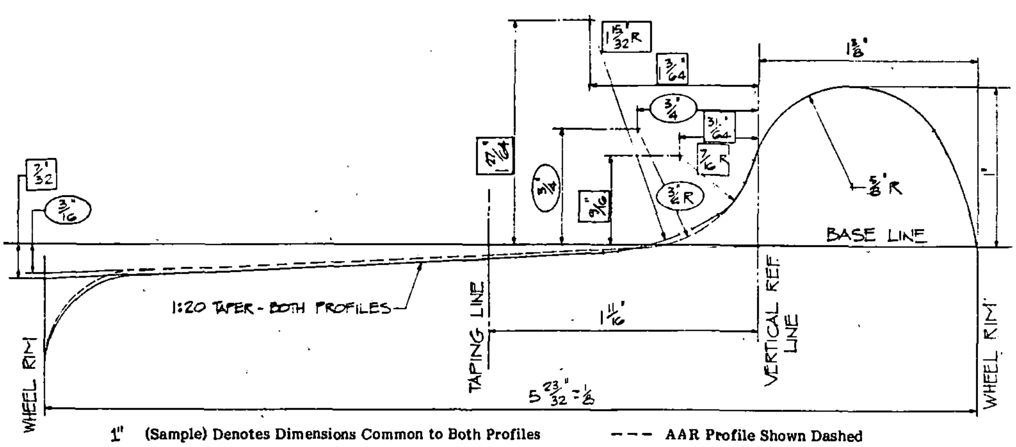
	70-Ton Capacity* Open Hopper Car	100-Ton Capacity Open Hopper Car
Empty (light) weight, lb	44,700	67,300
Loaded weight, lb	167,900	237,300
Capacity, lb	154,000	196,000
Length over pulling face of coupler, ft	46.17	53.04
Truck centers, ft	33.67	40.5
Center of Gravity (above rail):		
Loaded, ft	5.85	7.17
Empty, ft	-	4.38

*Used only on Alusuisse truck testing.



Curve No.	Degree of Curve	Direction Curve	Length (ft.)	Super Elevation (in.)	Start MP	Stop MP	Equilibrium Speed
1	2.5	Left hand	2736	4.09	321.1	320.6	48.3
2	6.2	Left hand	1080	4.98	319.7	319.5	33.9
3	6.1	Right hand	1386	4.90	319.4	319.1	33.9
4	5.2	Left hand	1136	5.08	318.7	318.5	37.4
5	1.1	Right hand	934	1.41	318.0	317.8	42.8
6	3.0	Right hand	3118	3.32	317.1	316.5	39.8
7	5.1	Left hand	961	4.88	316.5	316.3	37.0
8	5.0	Right hand	2070	4.79	316.2	315.8	37.0
9	3.7	Left hand	492	3.72	315.8	315.7	37.9
10	6.2	Right hand	1037	4.94	315.7	315.5	33.7
11	6.1	Left hand	2420	4.92	315.0	314.6	34.1

FIGURE 2-33. CURVE PROFILES - TEST ZONE 1



1" (Sample) Denotes Dimensions Common to Both Profiles
--- AAR Profile Shown Dashed
—— CN Profile
1" AAR Profile Dimensions Only
1" CN Profile Dimensions Only

FIGURE 2-34. WHEEL PROFILE COMPARISON - CN PROFILE VS AAR STANDARD 1:20 PROFILE

Devine-Scales Truck

The Devine-Scales truck consists of a one-piece, fabricated H-shaped frame with suspension assemblies known as subframes positioned in pockets at the corners of the main frame (see Figure 2-35). The subframe can move longitudinally on low-friction slides under the control of a geometric steering linkage connected to the carbody on each side of the truck. On tangent track, the rigid frame and steering linkage keep the wheelsets locked in a straight ahead position to provide lateral stability according to the manufacturer. On curves, the geometric steering linkage adjusts the positions of the subframes, moving them apart on the outside of the curve and together on the inside.

MTS Maxiride 100 Truck

The MTS Maxiride 100 truck is a one-piece, 100-ton capacity fabricated truck derived from a series of European trucks (see Figure 2-36). It features a welded steel frame and bolster unit construction, spring suspended roller bearing, journal bearing boxes, frame-stiffening end transoms and self-lubricating center bowl, and truck-to-carbody locking center pin. The manufacturer claims the truck improves high speed performance and ride quality, and reduces truck hunting and wheel wear.

National Swing Motion Truck

The National Swing Motion truck's conventional side frames and bolster are held in tram and prevented from "parallelogramming" by incorporating a transom connecting the two side frames through special rocker seats (see Figure 2-37). This arrangement permits the side frames to swing laterally in unison as pendulums or "swing hangers." The control of "rock and roll" is accomplished by limiting conventional gibs on truck bolster and providing lateral stops between the bolster and the transom at the height of the side frame spring seat.

Dresser DR-1 Steering Assembly

The Dresser DR-1 Steering Assembly is a retrofit package designed to add self steering and curve negotiation control features to conventional trucks (see Figure 2-38). The steering assembly ties together opposite axle boxes, which are, in turn, connected through one of the bolster openings. An elastomeric pad is provided between the roller bearing adapter and the jaw of the side frame pedestal with adequate clearance longitudinally to allow the wheelsets to move in seeking a radial position. The manufacturer maintains that a truck retrofitted with this device will result in improving curving performance, and reduced wheel wear and fuel consumption.

Barber-Scheffel Truck

The Barber-Scheffel truck consists of cast steel side frame and bolster arranged in a conventional manner (see Figure 2-39). According to the manufacturer, diagonally placed steel cross arms constrain the wheelsets to each other for high speed wheelset stability while, at the same time, allowing the wheelsets to align radially on curved track. Radial alignment is accomplished by using profile wheels having a highly effective conicity and providing a low yaw constraint on each wheelset.

ACF Fabricated Truck

The ACF Fabricated truck is made up of two side frames and a bolster with a secondary spring group in a somewhat conventional arrangement (see Figure 2-40). However, it has a tie between the side frames and is equipped with hydraulic snubbers. There is a flat rectangular plate in a horizontal position that ties the two side frames together, which is designed to hold the truck frame rigid while providing additional equalization by allowing the side frames to rotate relative to each other. The manufacturer claims that holding the truck rigidly in tram is designed to materially reduce hunting.

Alusuisse Truck

A rather radical departure from conventional European or American practice is the truck developed by Swiss Aluminum, Ltd. (see Figure 2-41). The Alusuisse truck frame consists of four hinged arms extending from the bolster to roller bearing "pillow blocks" holding the axles. Longitudinal leaf springs below the hinged arms are shackled to the arms by multiple turns of steel cable. A safety cable is provided to prevent collapse of the scissor arrangement in case of a broken spring.

2.6.3 Test Regimes

One carset each of the selected Type II trucks was tested to generate the performance test data. Field test data on the tested trucks varied in scope in many cases from truck to truck within each of the performance regimes due to the limitations on the deployment of instrumentation imposed by the design features of the trucks themselves. Table 2-9 shows the test matrix and the data available from the field tests of Type II trucks.

Lateral Stability Tests

High speed lateral stability tests were conducted over mainline, bolted jointed rail (BJR), test zone 2, on all seven trucks. High speed lateral stability tests were also conducted over test zone 5 on continuous welded rail (CWR) with the Dresser DR-1 and the MTS Maxiride 100 trucks. The test cars were run at test speeds ranging from 40 to 79 mph with dwells of approximately 60 seconds duration at 5 mph intervals within the test speed range.

Trackability Tests

Harmonic Roll. These tests were conducted over test zone 4 which is a class 2 branch line track. Both loaded and empty cars were run for an operating speed range of 4 mph to 30 mph with dwells approximating 40 seconds in duration at 2 mph increments; the tests were then repeated at 2 mph decrements from 30 mph down to 4 mph.

Track Twist. These tests were run over test zone 3, which consists of 0.22 miles of yard track with 16 degree curves. The test consist was pulled through the test zone in one direction at 10 mph. After the consist had come to a stop, it backed up through the zone at 10 mph with the engine pushing the consist.

Curve Entry/Exit. These tests were run over test zone 1, consisting of mainline jointed track, ranging in curvature from 1.1 to 6.2 degrees and ranging in

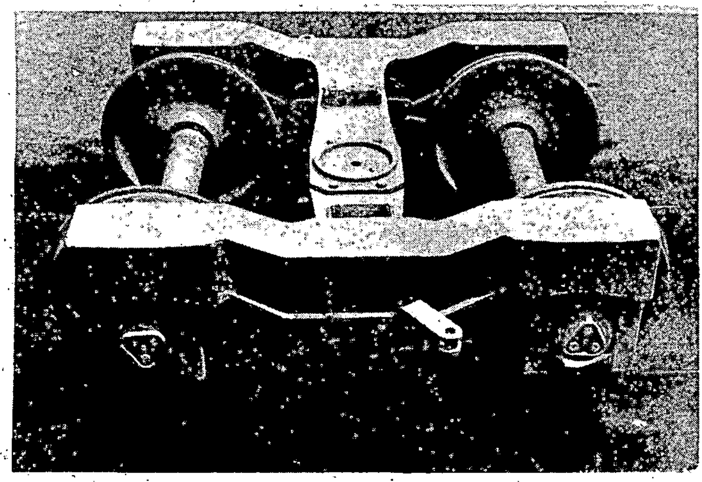


FIGURE 2-35. DEVINE-SCALES TRUCK

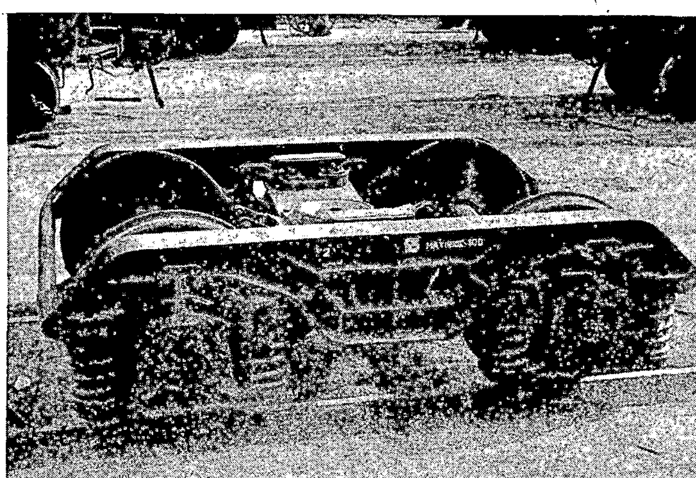


FIGURE 2-36. MTS MAXIRIDE 100 TRUCK

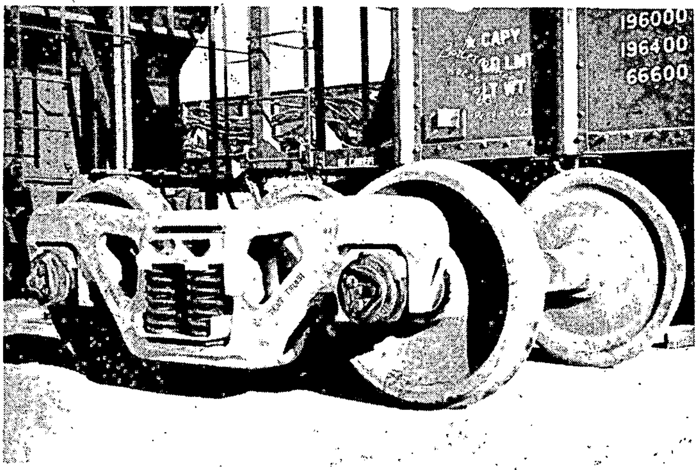


FIGURE 2-37. NATIONAL SWING MOTION TRUCK

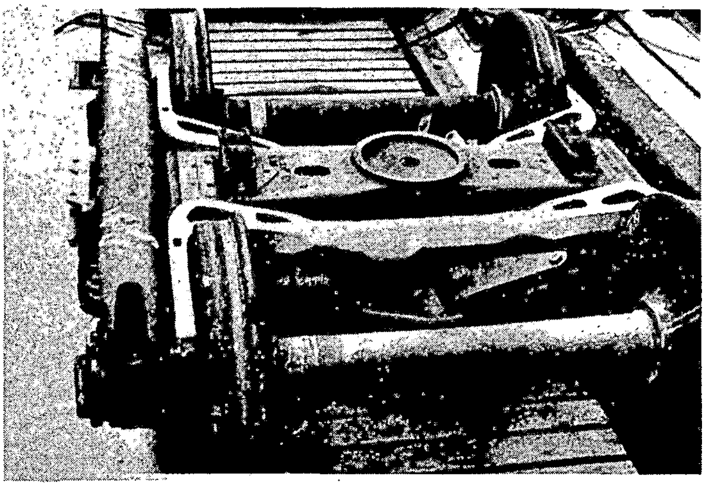


FIGURE 2-38. DRESSER DR-1 STEERING ASSEMBLY

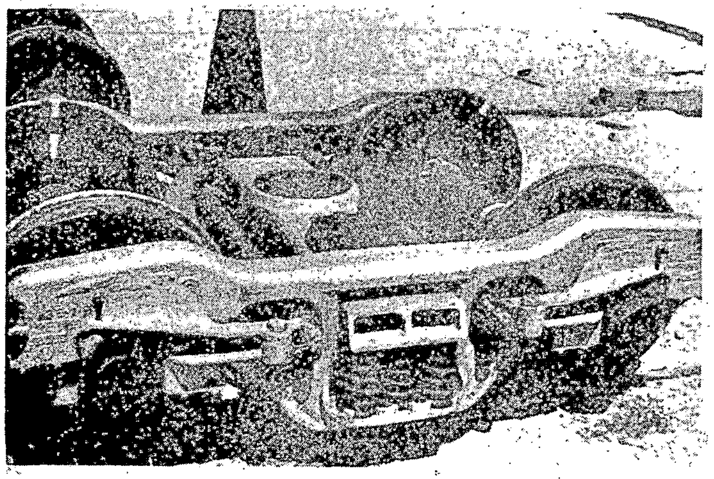


FIGURE 2-39. BARBER-SCHEFFEL TRUCK

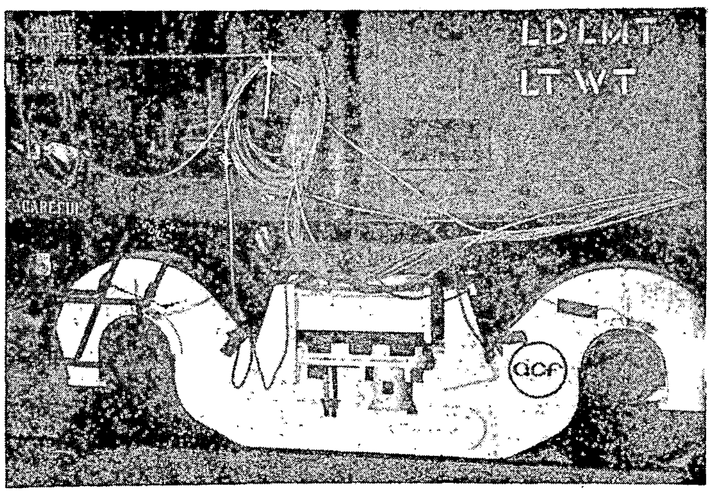


FIGURE 2-40. ACF FABRICATED TRUCK

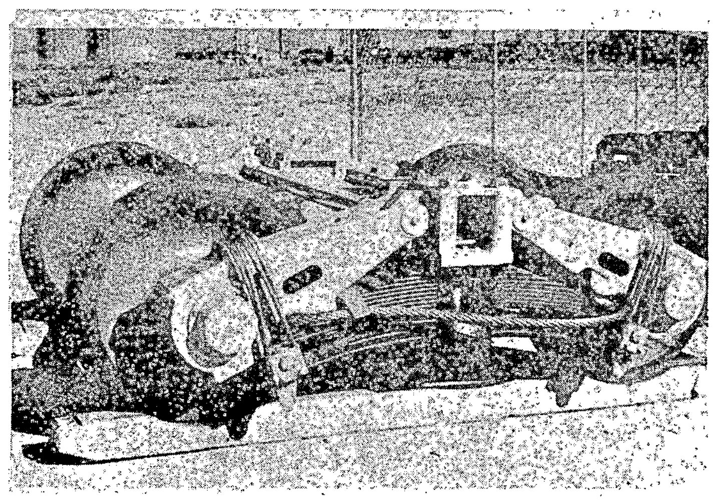


FIGURE 2-41. ALUSUISSE TRUCK

equilibrium speeds from 34 to 48 mph. These tests were the same tests designed to collect data for use in the curve negotiation regime.

Curve Negotiation

These tests were run over test zone 1. A layout of the curved track through test zone 1 is shown in Figure 2-33. Along with the layout is a table giving a detailed description of each curve and the equilibrium speed for that curve based on the degree of curvature and the superelevation. The track data are based on measurements taken during the November 1978 and December 1979 track geometry survey (References 14 and 15).

The test series consisted of four passes over this test zone, three in the uphill (milepost 321.48 to 314.5) and one in the downhill direction. The first test in the uphill direction was run approximately 10 mph below equilibrium speed for the curves; the second near equilibrium speed; and the third at approximately 7 mph over equilibrium. The downhill run was made at a speed near equilibrium for the curves. Tests were conducted using carbodies in the empty and the loaded conditions for all the trucks except Alusuisse which was tested with a loaded carbody only.

Ride Quality

The ride quality test runs were the same runs as those used for the lateral stability tests. These test runs consist of the high speed runs on mainline tangent track. Lateral oscillations indicative of unstable phenomena were excluded from consideration in the ride quality regime.

Fuel Consumption

For the fuel consumption study, the data acquired during the test runs over curving test zone 1 were used. Besides these collected data on the curving zone, two passes were made through test zone 2 with a loaded carbody. One of these passes was made in the downhill direction and the other in the uphill direction. The speed profiles in the test procedure called for a range of speeds from 40 mph to 79 mph and down to a stop. However, it was found that the engine could not pull the test consist in the uphill direction at speeds greater than 50 mph, so the test procedure was revised to make 50 mph the top speed in the uphill direction.

TABLE 2-9. PHASE II TEST MATRIX

Truck	Carbody	Wheel Profile	Lading	Rolling Resistance	Lateral Stability & Ride Quality		Trackability		Curve Negotiation
							Harmonic Roll	Track Twist	
					Class 4 BJR	Class 5 CWR	Class 2 BJR	Yard BJR	Class 4 BJR
Dresser DR-1	100-Ton Open Hopper Car	CN	E	●	●	●	●	●	●
			L	●	●		●	●	●
Barber-Scheffel	100-Ton Open Hopper Car	CN	E, L	●	●		●	●	●
Devine-Scales	100-Ton Open Hopper Car	CN	E, L	●	●		●	●	●
National Swing Motion	100-Ton Open Hopper Car	CN	E, L	●	●		●	●	●
Maxiride 100	100-Ton Open Hopper Car	CN	E	●	●	●	●	●	●
			L	●	●		●	●	●
ACF Fabricated	100-Ton Open Hopper Car	CN	E, L	●	o		●	*	o
Alusuisse	70-Ton Open Hopper Car	AAR 1:20 Taper	L	●	●		●	*	

Legend

● Test Data Available

o Curving Data Consisting of Angle of Attack; No L/V Forces

* No Data Available on L/V Forces

E=Empty

L=Loaded

BJR=Bolted Jointed Rail

CWR=Continuous Welded Rail

CN=Canadian National Profile

2.7 DATA ACQUISITION

2.7.1 Data Recording

Up to 96 data channels were brought from the transducers on the instrumented hopper car through the transfer cables to the signal conditioning on Mobile Laboratory Car 210. As part of the signal conditioning, data were filtered at various filters from 20 to 100 Hz, depending on the transducer type. The axle bending data were recorded directly from the axle using only the 500-Hz filters in the axle-mounted signal conditioning. Real time monitoring of the recorded axle bending signals, via strip chart and oscilloscope, indicated no high frequency data on these signals which would require additional filtering.

From the signal conditioning, each data channel was patched to the appropriate A/D channel. The A/D multiplexer scanned all channels sequentially over a 2.3-millisecond time frame. These scans were filed onto a buffer until 22 scans had been acquired. The buffer was then written to magnetic tape as one data record. The scanning of the channels was repeated at a rate of 200 times per second. The exact data record format is on file at the National Technical Information Service (NTIS) with the data tapes.

A noise floor analysis on the tape of data acquired during Type II truck testing was conducted during the analysis of the test data from the Friction Snubber Force Measurement field test program. This analysis is contained in Reference 16 and is applicable to the Type II truck test data.

2.7.2 Real Time Data

Real time data consisted of a strip chart playout of the analog signal from six selected channels. The exact channels varied from run to run, but always included the automatic location detector to provide a reference for the data. This chart was monitored by the instrumentation engineer during the test run to assure data quality. The data strip charts from all the test runs are on file. Real time oscilloscope monitoring of the analog data as it was multiplexed into the analog to digital converter was done during data recording as a further quality control measure.

2.7.3 Quick Look Test Data

At the completion of each test run, a five-second sample of the data for all channels was played back to verify data quality.

2.7.4 Photographic Documentation

The configuration of all instrumentation was documented by means of 35 mm color slides. Closed circuit television cameras were used to monitor truck and carbody motions, as well as the operation of the angle of attack system. For portions of selected runs, a black and white video recording was made of the television pictures. These video tapes are on file along with the color slides.

2.7.5 Data Printouts

The data tape from each test run has a computer printout which details significant test data and summarizes the actual recorded data by track locations and record number. These printouts are retained and stored with the real time and quick look data.

2.7.6 Data Tapes

Selected test runs of the best or most representative data for each test zone and speed have been submitted to NTIS for public availability. The data tape list is given in Appendix B.

2.8 DATA REDUCTION

2.8.1 Introduction

Test data reduction was performed using the contractor's resident Interdata 8/32 computer system. The system includes a 32-bit computer with 256 kilobytes of 750 nanosecond main memory and includes a high performance single and double precision floating point processor to speed calculations.

Data reduction was accomplished principally with three computer programs. The first, called PHIIBLD, was used to generate a data base on the disc for each test run using the header information and test data from the test tape. Following generation of a disc data based for a particular test run, it was copied to magnetic tape for long-term storage. The second program, called ADARS was used to perform the actual data reduction using the disc data base and command files generated by the analyst. The ADARS computer program is a general-purpose data analysis and reduction software package which provides the capability to perform a variety of analytical calculations. The third program, called TABLER, was used to generate summary tables and plots of statistical data calculated by ADARS and stored on disc. A typical table or plot would be the average lateral wheel/rail force of the leading outer wheel as a function of curvature for all of the curves in the curving test zone. The following sections describe the data reduction that was performed for the four performance regimes of lateral stability, ride quality, steady state curve negotiation, and trackability.

2.8.2 Lateral Stability and Ride Quality

Data reduction for the lateral stability and ride quality regimes included determining carbody and truck accelerations and truck angles. These calculations were performed for tests on the high speed, bolted jointed rail (BJR), and the continuous welded rail (CWR) for each truck. The equations used for the analysis and reduction of the lateral stability and ride quality data are contained in Appendix C.

Output for each run consisted of time history plots of raw accelerometer data, calculated body motions (such as roll, pitch, and lateral), truck and carbody angles, and speed. Additionally, power spectral density (PSD) plots of some of the above quantities were generated. Printouts of statistics included maximum, minimum, average, and standard deviation; probability densities and exceedances; narrow band RMS calculated about the peak in the PSD occurring below 5 Hz; and wide band rms for some of the quantities. Summary tables and plots were also generated.

2.8.3 Steady State Curve Negotiation

Data reduction for the steady state curve negotiation regime included determining wheel/rail forces, angle of attack, and truck swivel and tram angles.

Wheel/Rail Forces. Because of differences in truck design and instrumentation, the equations used to determine wheel/rail forces were not the same for each truck. The equations used for the Type I and Type II trucks are included in Appendix C.

Lateral and vertical forces at the wheel/rail interface and L/V ratio were calculated for each wheel of the B-end truck using bending strain gage data for the leading and trailing axles and data from either the strain-gaged bearing adapters or the primary spring displacements. In addition, the wheel unloading index was calculated for the B-end truck and the net lateral and vertical forces were calculated for each axle of the B-end truck.

The output of the data reduction included time history plots, printouts of statistics (maximum, minimum, average, and standard deviation), summary tables of statistics, and summary plots. Data reduced from the curving tests were generated as time history plots for the steady state portion of the curves for the following measurements (B-end only):

- vertical axle bending moment for each of four locations
- vertical load at the bearing adapters
- bending moment due to the vertical load at each bearing adapter
- vertical wheel/rail force at each of four wheels
- lateral wheel/rail force at each of four wheels
- L/V ratio for each of four wheels
- net lateral wheel/rail force for each of two axles
- net vertical wheel/rail force for each of two axles
- wheel unloading index
- speed.

Statistics and summary tables were printed for all of these except the axle bending moments and the bearing adapter forces and moments. Summary plots of average lateral wheel/rail force and L/V ratio as a function of speed for each curve were generated using results from the under, near, and over equilibrium speeds.

Axle Bending Moments. Strain gage data were acquired for eight strain gage pairs at each of four locations (two on each axle). Originally it was planned to multiplex the eight signals at each location prior to data acquisition and acquire only two signals per location: vertical bending moment and longitudinal bending moment. However, problems with the multiplexer hardware resulted in a maximum of 32 signals being acquired. (In some cases less than 32 strain gage channels were acquired because of constraints in the data acquisition system.)

Prior to using the axle-bending strain gage data, two operations were performed to improve the quality of the data: bias removal and normalization. The data from one gage pair resemble a sine wave with the positive peak occurring when one gage of the pair is at the point of maximum strain and the negative peak occurring when the other gage is at that point. It was reasoned that over a significant time period the negative and positive peaks should, on the average, be the same and that the signal should have an average of zero. Any offset in the signal can be attributed to instrumentation bias. Therefore, the bias was removed from the data before they were used. It was also reasoned that the eight gage pairs at one location all measured the same quantity, only with a phase difference, and that the rms for the eight signals should be the same over a significant period of time. The eight signals at each location were therefore normalized to the average rms for that location. This process corrected any scale factor differences which may have occurred in the signal conditioning.

Two techniques were investigated for determining axle bending moments from the strain gage data. The first method, multiplexing, was a software version of the hardware multiplexer. The Rotary Pulse Generator (RPG) signal was used to determine the rotational position of the axle, and the gage pair nearest the vertical plane was selected for the vertical bending moment. A multiplier of +1 or -1 was used depending on which gage of the pair was up. The second method, referred to as the quadrature method, used the RPG signal to determine the rotational position of the axle and two orthogonal pairs of gages to determine the vertical bending moment. This method assumes that the signal from a gage pair will be sinusoidal if the bending in the axle is constant. By only requiring two gage pairs; this method allows four independent calculations (if all eight signals are good), which can be averaged to provide a better result.

The two methods were compared, and the quadrature method was found to give better results, especially when less than eight gages were functioning. It should be mentioned that it was found that the longitudinal bending of the axle was insignificant when compared to the vertical bending.

Bearing Adapter Forces and Moments. Each strain-gaged bearing adapter has strain gages in three locations. The bearing adapters were calibrated by the Transportation Test Center. From the calibration data, tables were developed for each bearing adapter with the magnitude and location of the vertical force vector, given the sum and difference of the two outer gages. During the data reduction, the sum and difference were determined and then a table look-up was performed to determine the desired vertical force and location. On the Devine-Scales truck, data for the outer gages were not acquired and the data from the center gages were used to determine the vertical force, which was assumed to be at the center of the bearing adapter. The equations used to calculate the wheel/rail forces are given in Appendix C.

Primary Spring Forces. On the Maxiride truck, no data were available for the bearing adapters, and the primary spring displacements were used to determine the vertical forces which were assumed to be at the center of the spring group. A nonlinear spring constant obtained from the truck manufacturer was used for this purpose.

Lateral and Vertical Forces. Following the determination of vertical axle bending moments and the location and magnitude of the vertical bearing adapter forces, the lateral and vertical forces at the wheel/rail interface were calculated using the equations in Appendix C.

Angle of Attack. Wheel/rail angles of attack for the leading and trailing axles of the leading truck were determined from data acquired from eight proximity sensors. The angle of the wheel relative to the side frame and the angle of the rail relative to the side frame were determined by taking the difference of two measurements, dividing by the distance between them, and multiplying by a constant. The wheel/rail angle was determined by subtracting the rail/side frame angle from the wheel/side frame angle. Appendix C shows the angle of attack calculations. In order to eliminate instrumentation biases, the bias was removed from the angle of attack sensors on tangent track.

Output from the angle of attack data reduction included time history plots, printouts of statistics, summary tables of statistics, and summary plots for the steady state portion of each curve in the curving test zone. Time history plots, statistics, and summary tables were generated for each of two axles for the following measurements:

- Rail/side frame displacement
- Wheel/side frame displacement
- Wheel/rail displacement
- Rail/side frame angle
- Wheel/side frame angle
- Wheel/rail angle

Truck Angles. Truck swivel and truck tram were calculated from displacement measurements. The equations used are contained in Appendix C. The bias was removed from the displacement data on tangent track. An output was generated from these angles similar to that generated for angle of attack. On rigid trucks which do not allow training, the tram calculation was omitted.

2.8.4 Trackability

The trackability regime includes harmonic roll, track twist, curve entry and exit. The data reduction for these are described below.

Harmonic Roll. Harmonic roll data reduction was performed on loaded and empty vehicle tests conducted on the class 2 Blue Diamond spur. It consisted of determining vehicle roll angles and accelerations at the leading and trailing ends of the vehicle, and vertical accelerations. Outputs similar to those for the lateral stability regime were generated.

Track Twist. Track twist calculations were performed on data from the yard tests. These calculations included wheel/rail forces and the wheel unloading index (see paragraph 2.8.3).

Curve Entry and Exit. The data reduction for the entry and exit portion of each curve was similar to that for the steady state portion. The only significant difference is that the summary plots concentrated on peak values rather than on average values. See paragraph 2.8.3 for a description of the data reduction for steady state curve negotiation.

2.9 REMARKS

The TDOP Phase II test program provided, by and large, the performance data required for the engineering and economic analysis of the Type I and Type II trucks. However, it is recommended that future noncontacting measurement fixtures (as were used in the angle-of-attack instrumentation) be mounted with larger clearances above the rail head. In addition, the fixtures should have sufficient dynamic range to be mounted from the sprung portions of the truck or carbody. Use of the Ectron signal conditioners should be restricted and rework of the amplifiers is recommended before use on other long-term test programs.

High vibration levels causing mounting and transducer problems on the unsprung portions of the truck may be anticipated on future test programs including primary suspension trucks such as the MTS Maxiride 100. The axle-mounted air gap transformer/power supply proved to be very reliable and rugged. Proven measurement techniques provided data of consistently good quality with a minimum of instrumentation problems. Large volumes of valuable data are contained in the TDOP Phase II data tapes, which can be used to address specific issues related to truck performance.

In addition to those tapes of truck dynamic testing, there are six tapes containing the results of a special test of the friction snubbers of both ASF Ride Control and Barber S-2 70-ton trucks. This test was conducted in various load conditions over sections of the same test tracks near Las Vegas during November and December of 1978. The results are presented in Reference 16.

2.10 REFERENCES

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5. * Gibson, D.W., "Truck Design Optimization Project Phase II - Type I Truck Test Results Report," Federal Railroad Administration Report No. FRA/ORD-81/77, NTIS Accession No. PB 82-15200 January 1982.

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SECTION 3 - ANALYSIS

The objectives of the analytical studies conducted during TDOP Phase II were (a) to assess the available computer models for applications in fulfilling project objectives; (b) to develop criteria for validating the models, (c) to validate the models; and (d) to apply the models in extending and interpreting the results from field test programs.

Both field test data and simulation models have been used to define and interpret the performance levels of Type I (standard) and Type II (premium) freight car truck configurations. The analysis effort and the field testing task have been designed to complement each other in meeting the project objectives. A plan (Reference 1) was developed to define the procedures for meeting the three main objectives of:

- Defining model and test data utilization in each performance regime.
- Determining the analysis requirements needed to extend and extrapolate the field test data for both Type I and Type II trucks.
- Establishing model application requirements which would provide a framework for the model validation criteria.

A number of analytical tools ranging from simple formulae to complex computer simulations were reviewed and assessed (Reference 2). This review and assessment led to the selection of several analytical tools which were applicable to modeling Type I and Type II freight car trucks. A set of validation criteria, for each performance regime against which the analytical tools could be evaluated, was selected (Reference 3). The process of selecting a set of validation criteria took into consideration:

a. Stated capabilities of the models in terms of input requirements, predicted outputs, and modeling assumptions.

b. Quality and adequacy of available experimental data, which serve as the basis against which comparisons could be made.

c. Minimum standards of credibility demanded of the models within the context of freight car truck performance characterization.

The remainder of this section presents an overview of the analysis plan, an assessment of the analytical tools, the development of validation criteria, and the results of the validation effort.

3.1 ANALYSIS PLAN OVERVIEW

An analysis plan (Reference 1) was developed for each of the four performance regimes of lateral stability, trackability, curve negotiation, and ride quality. In each regime, the analysis plan contained a brief review of the analysis requirements, and the model and test data to be used in performing the analysis.

In the lateral stability regime, the plan recommended that field test data and simulation models be used to investigate the influence of the environmental factors as well as operational conditions of truck hunting. It suggested that linear frequency domain modeling techniques be used whenever possible to determine preliminary performance sensitivity to parameter variations. Detailed nonlinear time domain simulations should also be used to calculate motions and forces required for performance specification input not provided by the field tests data.

For the track twist subregime of the trackability regime, the plan recommended that data for analyzing load equalization should be accumulated from field and laboratory testing. Field data would be acquired during the Type I and Type II truck test program. Laboratory data would be obtained from the Vibration Test Unit in the U.S. Department of Transportation Test Center's Rail Dynamics Laboratory. Simple static and kinematic models would be developed from the test data and used to evaluate load equalization capability.

The analysis plan noted that the harmonic roll and bounce subregime analysis would depend heavily upon models rather than field testing because the models would permit the safe investigation of the effects of extreme dynamics.

For curve negotiation, the plan recommended that two types of analytical models be used in predicting the curving performance indices for variations in truck parameters. Steady state models should be used to compare the basic kinematic performance of different trucks and for calculations of wheel wear, fuel consumption, and rail wear in curves. For derailment potential analyses and for rail wear, time domain models would be used.

The ride quality performance index of transmissibility is largely determined by the suspension characteristics. The focus, therefore, would be on the differences exhibited between primary and secondary suspension trucks. Simple linear models would be used to predict the lading vibration environment.

3.2 ASSESSMENT OF ANALYTICAL TOOLS

The term "analytical tool" refers to any analytical method employed to predict and understand the car/truck dynamic behavior. Analytical tools include models which are considered here to be the set of equations describing the car/truck dynamics and the computer program that implements these equations. The analytical tools of most interest to TDOP Phase II are these models and computer programs which have been used in other car/truck modeling research and development projects. The criteria established for assessing the analytical tools is summarized in Table 3-1.

TABLE 3-1. SUMMARY OF ASSESSMENT CRITERIA

- Is the analytical tool applicable to one or more of the TDOP II performance regimes?
- Is the tool useful in studying truck performance analyses that meet TDOP II objectives?
- Is the tool capable of performing or supporting analyses that meet TDOP II objectives?
- Is the tool compatible with the digital computers available to the TDOP Phase II contractor?
- Is the tool capable of analyzing required truck/carbody configurations with minor modifications?
- Is the tool available in terms of the TDOP II schedule?
- What is the validation status of the tool?
- What is the accuracy of the tool?
- What is the precision of the tool?
- Can the tool be verified?
- Is the utility of the tool acceptable?
- Does the tool complement the other tools properly?

3.3 VALIDATION CRITERIA

Validation is the process of determining the ability of an analytical tool to reproduce and/or predict observable behavior. Only the simplest models can be used with confidence without first being validated by comparing results from the model against actual test results. It is not sufficient to merely establish that the model has been formulated with a one-to-one correspondence between the elements of the model and the truck. The validation process verifies that the characterization of the interactions between model elements is sound.

Although there has been a steady growth in the number of rail dynamics models in the last decade, there has been a lack of activity in comparing the results of those models with actual test data. One of the reasons for the lack of model validation is that the modeling activity has tended to take place outside the traditional railroad community. Success of model validation efforts depends not only on the level of experience incorporated within the model, but also on the availability of adequate test data to serve as a basis for comparison. With the large amount of data collected in TDOP Phase I, the opportunity was seen in TDOP Phase II to select a number of models as candidates for validation exercises. The models were selected with regard to the four performance regimes of lateral stability, trackability, curve negotiation, and ride quality. The validation criteria which have been selected reflect the individual performance indices chosen early in TDOP Phase II and the evaluation of test results from Phase I and elsewhere. Phase I data have provided a means for assessing the range and sensitivity of the performance indices. A discussion of the validation criteria for the four performance regimes is given in subsections 3.3.1 through 3.3.4.

3.3.1 Validation Criteria for Lateral Stability Models

The dynamic behavior of a freight car in the regime of lateral stability is complex and difficult to simulate except for highly simplified configurations. This is due to the number of factors that affect stability and the interactions between them. Thus, when only a finite amount of test data is available, as is the case in TDOP, an acceptable model validation procedure includes the requirements not only for a close match between the results of simulation and tests, but also for verification by accepted theory and the results from other test programs.

A detailed review of TDOP Phase I test data (Reference 4) has shown that the transition from lateral stability to fully developed hunting is characterized by several discrete stages, the identification of which is considered helpful both in the development and in the assessment of models. Below critical speed, power spectral densities (PSD's) of lateral carbody accelerations show the presence of all major body modes at their natural frequencies: lateral, yaw, and, in the case of box cars, lower center roll. These oscillations are presumably excited by track irregularities.

The first evidence of self-excited oscillation is the predominance of a single frequency in all degrees of freedom of the carbody. However, the mode of oscillation is not a normal mode, but a combination of lateral displacement, yaw, and upper center roll in such a way that the resulting displacements almost completely cancel out at the trailing truck but add up at the leading truck, thus producing a motion called "nosing." (The reverse of this phenomenon, called "fishtailing," has also been observed but appears to be less common.)

As the speed increases, there is often an abrupt increase in the frequency of oscillation which is close to the natural frequency in yaw of the carbody on its suspension. This was true for the cars tested both by TDOP and the AAR (Reference 5). Since the mass moment of inertia of the carbody about the centerplate is much higher than that about the center of mass (on the order of four times) it is hypothesized, though this remains to be demonstrated, that the circulating energy for the higher frequency and symmetrical mode in pure yaw is lower than that of the system if it were to oscillate about one centerplate at the higher frequency. It has been observed in hunting tests that violent body hunting can co-exist with very small lateral truck displacements, which is an indication of the small amount of energy required to maintain a limit cycle (Reference 6).

These considerations lead to the conclusion that the first appearance of a predominant frequency, indicating the onset of self-excited oscillation, represents a useful criterion for model validation in the hunting regime, although successful simulation of lateral acceleration through the entire critical speed range is considered as valuable evidence supporting model validity.

In setting the validation criteria for prediction of critical speed, the wide range of the critical speed has been considered. A tolerance of ±5 mph has been chosen based on ±10 percent of the 50 mph critical speed range (see Figures 3-1 and 3-2). Although not a performance index, another convenient point of comparison between lateral stability models and test results is the frequency at which the hunting oscillations occur. Figure 3-3 shows the envelope within which Phase I data fell. A tolerance of ±0.3 Hz based on ±10 percent of the maximum observed frequency of approximately 3.0 Hz has been set for comparison of hunting frequencies.

3.3.2 Validation Criteria for Trackability Models

The trackability regime includes several aspects of performance which have in common the ability to maintain loads adequate to provide guidance forces on each of a truck's four wheels. These performance subregimes are harmonic roll and bounce, track twist, and curve entry and exit.

Harmonic roll and bounce are forced response phenomena due to periodic track excitation. Harmonic roll is typically excited by cross level variations arising from half-staggered track at speeds from 10 to 20 mph. Bounce resonance involving pitch and vertical motions

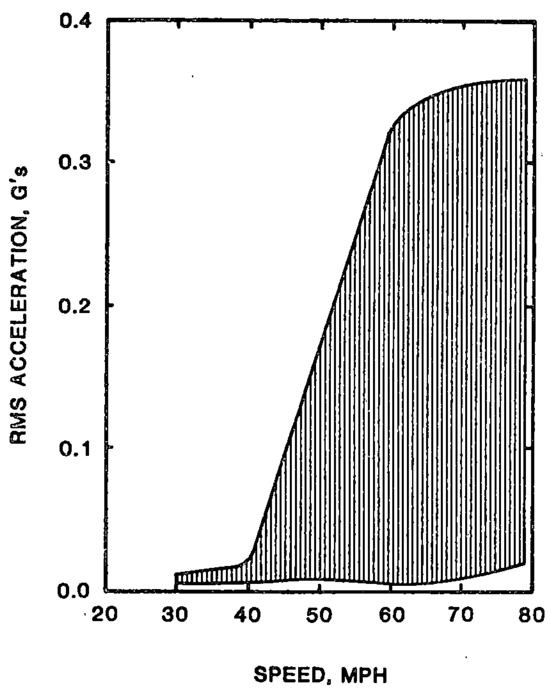

FIGURE 3-1. LATERAL STABILITY PERFORMANCE BOUNDS - RMS LATERAL ACCELERATION VERSUS SPEED (BOX CARS)

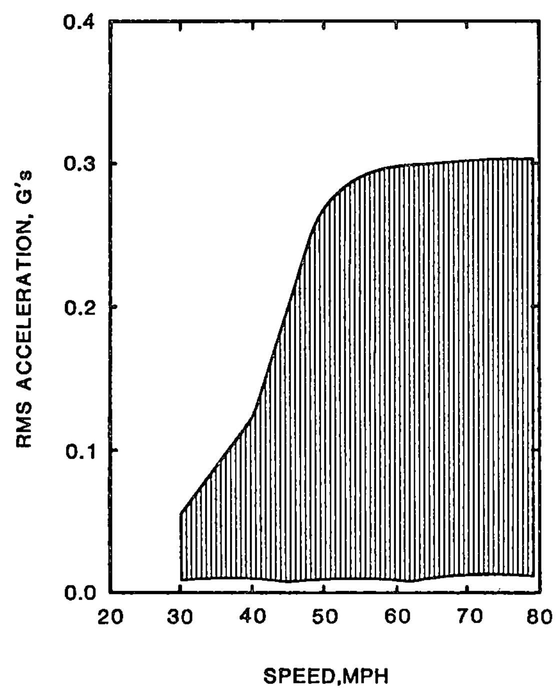

FIGURE 3-2. LATERAL STABILITY PERFORMANCE BOUNDS - RMS LATERAL ACCELERATION VERSUS SPEED (FLAT CARS)

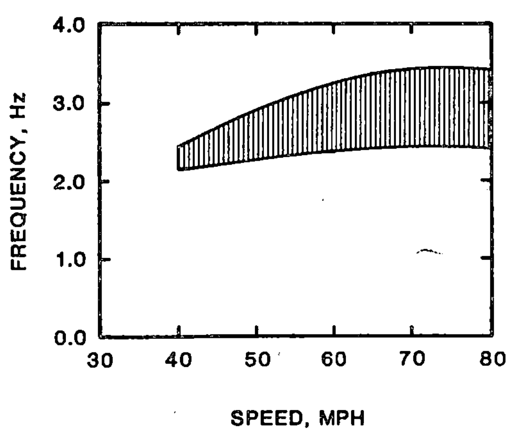

FIGURE 3-3. RANGE OF HUNTING FREQUENCIES VERSUS SPEED FOR THE STANDARD, THREE-PIECE TRUCKS

of the vehicle occurs at higher speeds between 40 and 65 mph staggered or unstaggered track. Harmonic roll and bounce have received a great deal of modeling interest. For harmonic roll, the maximum roll angle has been chosen as a performance index. Load distribution is also an important measure of performance.

Validation of trackability models has been focused on harmonic roll since there are more data available for comparison for that subregime. Figure 3-4 shows results for the case obtained by American Steel Foundries (ASF) of a loaded, 100-ton hopper car from which a measure of the data scatter can be drawn especially near resonance. The validation criteria for peak roll angle has been chosen as ± 1 degree which reflects the variation in test data observed in Figure 3-4. Figure 3-5 shows the variation in spring nest force for the same tests. The spring nest force, though not a performance index, is a relevant point of model comparison. Near resonance the variation is approximately ± 3000 lb or approximately ± 5 percent of the static 60,000 lb spring nest force. A five percent tolerance on spring nest force has, thus, been chosen for the load distribution validation criterion.

A final point of comparison in the validation of harmonic roll and bounce models is the prediction of speed at resonance. A tolerance of ± 1 mph has been selected. However, the difficulties in identifying harmonic roll resonance speed should be noted. The resonance speed has been reported to be dependent on the amplitude of excitation as well as the frequency sweep of the excitation (i.e., entering the resonance speed from above or below).

The track twist/load equalization problem is largely a quasi-static phenomenon (with speeds of 10 mph or less). The accommodation of track twist within the wheel base of the truck is achieved by side frame pitch with flexible trucks and by primary suspension compliance with rigid trucks. With conventional trucks, the problem is aggravated by sticking of the friction snubbing devices when operating at low speeds. The load equalization ability of trucks has not received a great deal of modeling interest since it can be measured relatively simply.

The basic performance index for the load equalization subregime is the wheel unloading index (WUI) which is given by the formula:

$$\mathrm{WUI} = \left[\frac{W_H/3 - W_L}{W_H/3}\right] \div \theta = \left[1 - \frac{W_L}{W_H/3}\right] \div \theta \ \cdot \text{degree}^{-1}$$

Where:

W_L = vertical force on the most lightly loaded wheel

W_H = sum of vertical forces on the three most heavily loaded wheels

θ = angle in degrees of track twist within the wheelbase of the truck.

Although curve entry and exit can lead to loss of trackability, from the modeling standpoint it is logically approached as the dynamic aspect of the overall curving behavior problem. This subregime, therefore, will be discussed in context of the curve negotiation performance regime.

3.3.3 Validation Criteria for Curve Negotiation Models

Although by definition the curve negotiation performance regime consists only of steady state or quasi-static conditions encountered during a negotiation of constant curvature track, modeling efforts covering the transient dynamic response obtained during curve entry and exit are also included for discussion. Steady state models are considered in the simulation of performance under quasi-static conditions; time domain curve negotiation models may be used to address the dynamic response which occurs due to curve entry and exit.

The performance index chosen for constant radius curving is the lateral force on the leading outer wheel per thousand pounds of axle load per degree of curvature at balance speed. In theory, the lateral force should be at a single value in a constant radius, constant speed curve. In practice, however, some variation in the

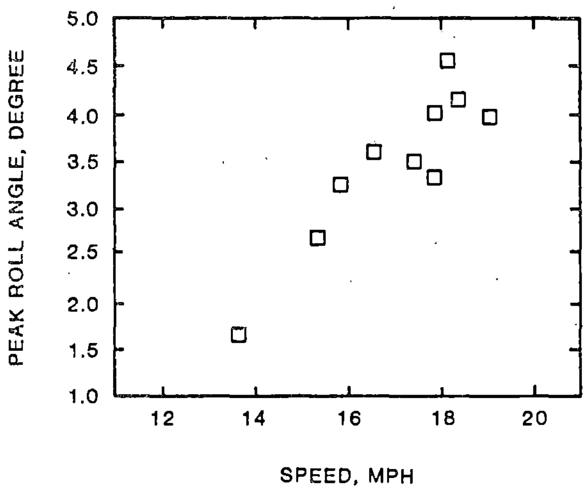

FIGURE 3-4. VARIATION IN HARMONIC ROLL TEST DATA FOR ROLL ANGLE

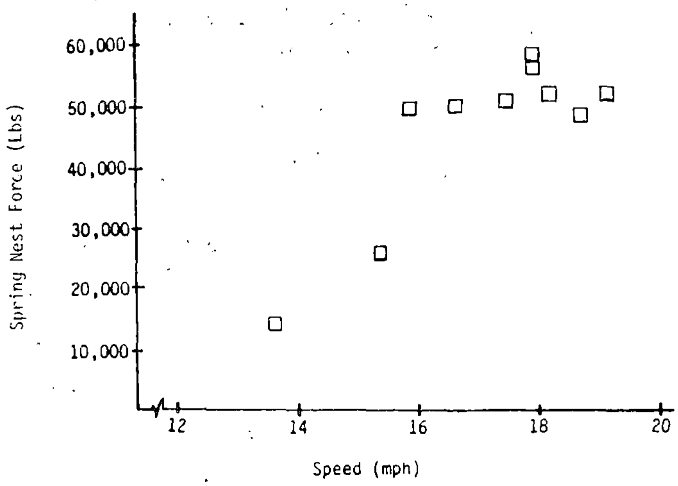

FIGURE 3-5. VARIATION IN HARMONIC ROLL TEST DATA FOR PEAK SPRING NEST FORCE

lateral force is inevitable due to track irregularities which causes the Coulomb friction elements such as the centerplate to take on different "sets." Furthermore, in actual tests, constant speed is only achieved within a finite margin.

Validation of steady state curve negotiation models is among the most difficult to obtain primarily because of limitations in the accuracy of measuring wheel/rail forces. To date, such measurements have been found to be in error by 30 to 50 percent when compared with the theoretical force equilibrium. The wheel/rail force measurement system developed for TDOP Phase II testing improved the measurement accuracy. With improved experimental techniques, a $\pm$ 15 percent tolerance in the prediction of the performance index is considered reasonable.

Curve entry and exit can be characterized by the wheel unloading index. Following the case of steady state curve negotiation, a tolerance of $\pm$15% for predicting wheel unloading index is selected.

3.3.4 Validation Criteria for Ride Quality Models

The ride quality performance regime is comprised of the overall dynamic environment of the carbody response, and the effectiveness of the truck in attenuating track-induced excitations, exclusive of the more extreme dynamics associated with the other performance regimes. The economic impact areas associated with ride quality are lading damage and cost due to component wear.

The principal performance index for ride quality is the transmissibility of the truck. Transmissibility can be measured as a frequency dependent ratio of output-to-input amplitudes at discrete frequencies (i.e., transfer functions) or as ratios of root mean square (rms) output to input over particular frequency bands. The validation criteria selected for ride quality models consider the comparison of the occurrence of principal frequencies and the prediction of magnification/attenuation factors between 0 and 20 Hz. For comparison of principal frequencies, a tolerance of $\pm$0.5 Hz is considered acceptable. The tolerance for reproduction of output/input ratios, either transfer functions or rms, is set at $\pm$10 percent.

3.4 VALIDATION RESULTS

After completing a preliminary survey of 59 tools, 15 of the most promising analytical tools were selected for detailed assessment and validation. While Table 3-2 contains a brief description of these tools, they are discussed more fully in the subsequent paragraphs of this section.

3.4.1 Validation Results of Lateral Stability Models

The models which have been examined in regard to the lateral stability performance regime include a linear frequency domain representation (17 dof Eigenvalue Model), a detailed nonlinear time domain representation (HUNTCT), and a simplified nonlinear time domain representation. The simplified nonlinear time domain modeling work has been done in lieu of validation of the AAR Freight Car Hunting Model (Reference 7) originally selected as a candidate for validation. It was felt that this would be more productive because of the similarity between the Freight Car Hunting Model, HUNTCT, and the 17 dof Eigenvalue Model.

3.4.1.1 17 dof Eigenvalue Model. The model chosen to represent the linear frequency domain family of lateral stability models was the 17 dof Eigenvalue model developed by Law and Cooperrider (Reference 8). The model was selected for validation as one of the most sophisticated linear models of freight car lateral stability. The program provides natural frequencies and mode shapes for the configuration described by the 17 degrees of freedom. Although it is a linearized model, the level of detail is sufficient to allow investigation of the effects of many truck components. The degrees of freedom are lateral and yaw of each wheelset, lateral, yaw and warp (parallelogramming) of each truck (Figure 3-6), and lateral, roll, and yaw of the carbody. Gravitational stiffness, spin creep, and gyroscopic terms are also included. The model can accommodate asymmetrical loading front to rear and nonidentical front and rear wheelsets and suspension parameters. Also included in the model is a provision for modeling, bending, and shear connections between wheelsets such as those implemented in a radial axle truck. The program uses matrix inversion techniques to solve the 17 second order differential equations of motion. Natural frequencies (eigenvalues) and mode shapes (eigenvectors) are produced by the program.

Data consistent with TDOP Phase I testing and the needs of the program was input to the model. The empty mechanical refrigerator car on 70-ton Barber trucks with new wheels was selected as the validation case. This particular combination had exhibited hunting behavior in the Phase I test including curious phenomenon such as occurrences of front truck hunting only (nosing) and intermittent hunting.

An initial comparison of model and test results using a trial set of input data produced results which indicated the onset of instability between 40 and 50 mph at a frequency of 0.75 Hz. Test data indicated the development of hunting between 50 and 70 mph at a frequency of from 2 to 3 Hz.

Re-examination of the input data led to the conclusion that the initial primary suspension stiffnesses were too large, approximating a rigid truck frame. The primary suspension stiffnesses were reduced to values consistent with flexible Type I trucks and a second comparison was made. Again the frequency associated with the unstable mode was quite low with respect to the test results. Variations of parameters which were considered to be the least accurately known were made. It was found that the only parameters which showed a significant sensitivity was the conicity. By artificially

TABLE 3-2. CANDIDATES FOR VALIDATION

Model	Degrees of Freedom	TDOP Areas of Application	Linear/ Nonlinear	Frequency/Time Domain Steady State Equilibrium	Carbody Model
17 dof Eigenvalue*	17	Lateral Stability	Linear	Frequency	Rigid
HUNCT*	21	Lateral Stability, Curve Negotiation	Nonlinear	Time	Rigid or Flexible
Freight Car Hunting	25	Lateral Stability (critical speed, stability margins)	Linear	Frequency	Rigid
FRATE 11*	11	Harmonic Roll, General Vehicle/ Truck Motions	Nonlinear	Time	Rigid or Flexible
FRATE 17	17	Harmonic Roll, General Vehicle/ Truck Motions	Nonlinear	Time	Rigid or Flexible
FRATE	27	Harmonic Roll and Bounce, Ride Quality	Nonlinear	Time	Rigid or Flexible (allows for lumped masses for lading)
Flexible Carbody Vehicle*	20	Harmonic Roll	Nonlinear	Time	Two Lumped
9 dof Steady State Curving	9	Curve Negotiation	Nonlinear	Steady State Equilibrium	Rigid
17 dof Steady State Curving	17	Curve Negotiation	Nonlinear	Steady State Equilibrium	Rigid
CN Curving Model*	10	Curve Negotiation	Nonlinear	Steady State Equilibrium	Rigid
DYNALIST II*	up to 50	Any (depending on particular model definition)	Linear	Frequency and/or Time	Rigid or Flexible
HALF	4	Component Wear, Safety	Linear	Frequency	Rigid
FULL	6	Ride Quality	Linear	Frequency	Rigid
FLEX	6	Ride Quality	Linear	Frequency	Flexible, First Mode Bending Only
LATERAL	15	Ride Quality	Linear	Frequency	Rigid

*Models validated during TDOP Phase II.

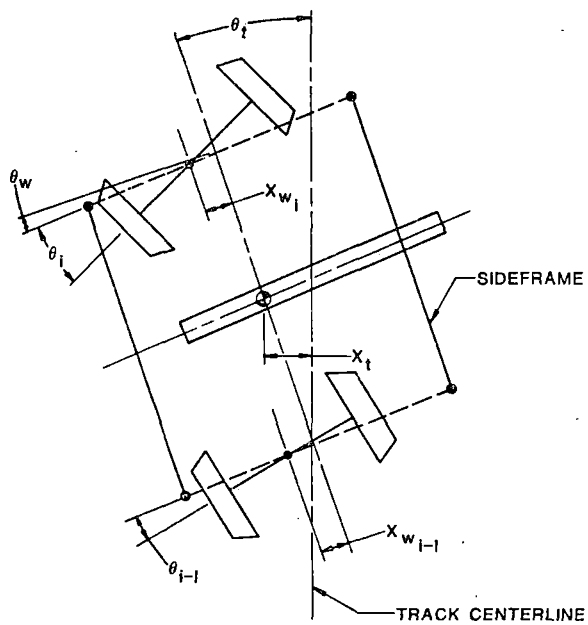

FIGURE 3-6. SEVEN DEGREES OF FREEDOM FREIGHT TRUCK MODEL.

increasing the conicity from the nominal 1/20 new wheel value to 1/15, closer agreement between model and test results was achieved. These results are shown in the complex frequency representation of Figure 3-7. The figure shows the loci which are traced from the kinematic and rigid body modes which are clearly identifiable at low speed. Also shown are actual test points.

From the results presented, it is concluded that the 17 dof Eigenvalue Model is best suited for qualitative comparison. (For instance, will truck A hunt at a lower speed than truck B, all other things being equal?) With care in the choice of wheel conicity, critical speeds within the validation tolerance may be obtained. The accurate prediction of the associated frequency of hunting appear to be beyond the model's capability.

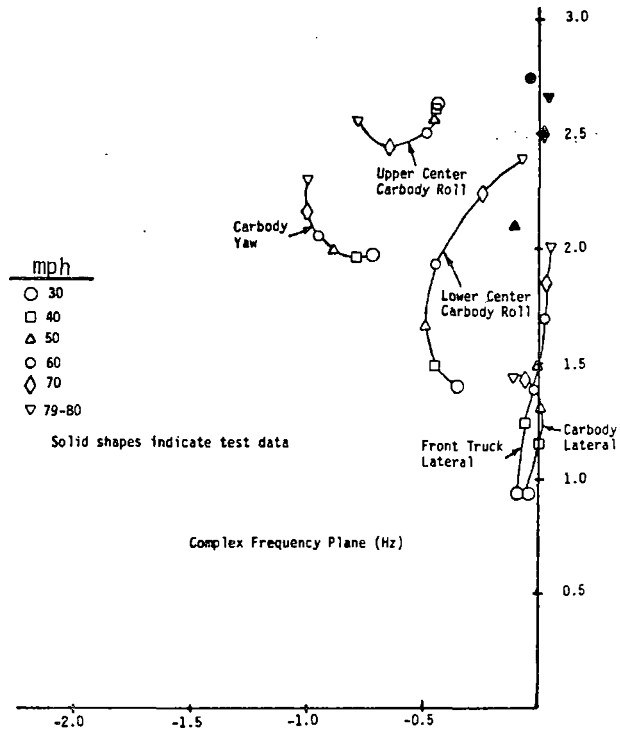

FIGURE 3-7. ROOT LOCI FOR PRINCIPAL MOTIONS OF EMPTY 70-TON REEFER FROM 17 DOF EIGENVALUE MODEL VS. TEST

3.4.1.2 HUNTCT. HUNTCT is a nonlinear time domain model developed by Wyle Laboratories for lateral stability analysis (Figure 3-8). The formulation uses 21 degrees of freedom representing rigid body modes of the units comprising the freight car/truck system. Additional degrees of freedom to represent carbody flexibility can optionally be included. The carbody is allowed to translate in the vertical and lateral direction and to yaw, roll and pitch. The truck is modeled as a single mass with vertical, lateral, yaw, and roll degrees of freedom.

The truck model also provides for coupling between wheelsets in the yaw sense (lozenging stiffness). Each wheelset has two independent degrees of freedom - lateral and yaw. Vertical and roll motions of the wheelset are constrained by the wheel/rail geometry with the assumption of no wheel lift off. Detailed calculations of the wheel/rail interface are carried out for each wheelset. The effective track mass, stiffness, and damping in the vertical sense are lumped with the truck. To simulate actual tests, the model requires track geometry data including left and right rail profile and alignment data. The model makes use of the Symmetric Wheel/Rail Constraint Subroutine (WHRAIL) (References 9 & 10) to relate the track input to wheelset motions.

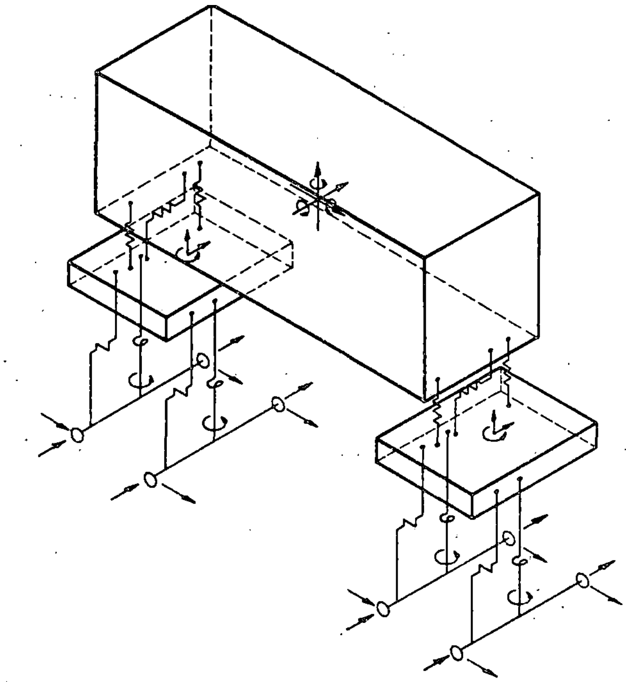

FIGURE 3-8. HUNTCT MODEL

As with the 17 dof Eigenvalue Model, comparisons of model and test results were made for the TDOP Phase I tests of the empty 70-ton refrigerator car with Barber trucks and new wheels. Unfortunately, the Phase I data collection did not include the key measurement of track alignment. Time domain comparisons were thus not feasible. The Friction Snubber Force Measurement System (FSFMS), which was tested on similar class track, included alignment measurements. These track data have been used as model input to obtain statistical characterizations of the response for comparison with Phase I test results. The substitution of the FSFMS data assumes that statistical characterizations of track of the same class were approximately the same.

The results of the comparisons of test and model responses for 50 and 79 mph are shown in Figures 3-9 and 3-10. There is agreement in the location of the principal frequency; however, the overall comparison of response levels in the 1 to 20 Hz range is not good. The results raise the question as to whether the comparison is poor due to the model or the assumptions made about the input data.

In a subsequent exercise, critical speed was determined by simulating the response to an initial lateral displacement on ideal track. The speed at which the response failed to decay was taken as the critical speed. This procedure identified the critical speed to be approximately 65 mph.

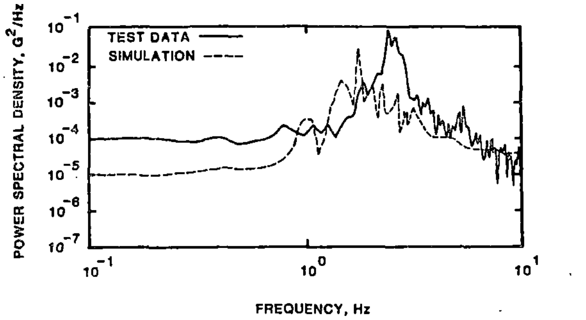

FIGURE 3-9. POWER SPECTRAL DENSITIES OF THE LATERAL ACCELERATION OF THE TRUCK AXLE - 50 MPH

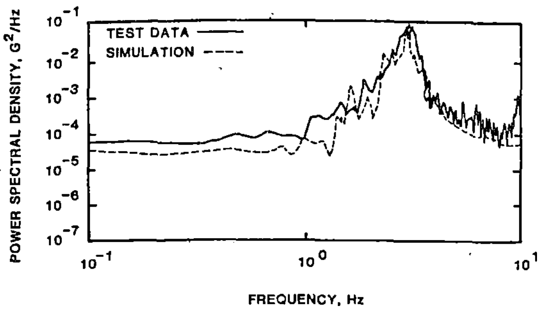

FIGURE 3-10. POWER SPECTRAL DENSITIES OF THE LATERAL ACCELERATION OF THE TRUCK AXLE - 79 MPH

The conclusion from the validation exercise with HUNTCT is that a decision must be reserved until test data can be obtained for which both response and input are recorded. The results thus far obtained indicate some potential to predict principal frequency and critical speed.

3.4.2 Validation Results of Trackability Models

The validation work in the trackability regime has focused on harmonic roll and bounce models. Since the harmonic roll and bounce models can generally be applied to the track twist subregime, validation of the harmonic roll behavior of a model gives a measure of confidence in using it for load equalization analysis as well. Four models are shown in Table 3-2 as being applicable to harmonic roll analysis. These are the Flexible Carbody Vehicle Model and three versions of FRATE. To avoid duplication and based on prior experience, only one of the FRATE versions was validated.

3.4.2.1 Flexible Carbody Vehicle Model. The Flexible Carbody Vehicle Model (FCBVM) was developed by AAR in conjunction with the Track/Train Dynamics Program (Reference 15). The model was selected to complement the FRATE models in the analysis of the harmonic roll and bounce subregimes. Version II of the model was used in the TDOP Phase II validation exercise. The model formulation features 20 degrees of freedom which include the following: vertical, lateral, roll, pitch and yaw of each half-carbody, vertical and roll of each bolster, vertical and lateral displacements and roll of each side frame/wheelset combination as shown in Figure 3-11.

The results of the comparison of roll angle (single amplitude) for the tests and the model are shown in Figure 3-12. There is significant disparity between the two sets of results. The model results indicate a low resonant speed and excessive amplitude at resonance. The input which produced the results shown was discussed with AAR representatives who suggested that proper adjustment of the load spring rates of damping could bring the model results into closer agreement with the test data. Further efforts with this model could not be carried out within the time and resource limitations within the project.

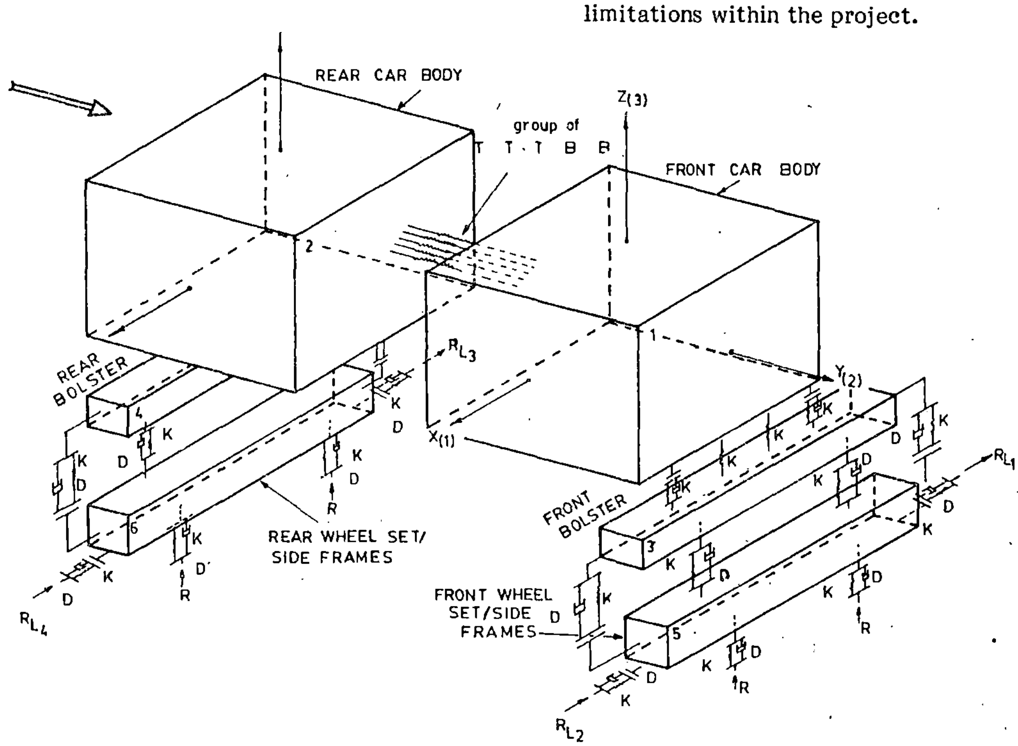

FIGURE 3-11. FLEXIBLE CARBODY VEHICLE MODEL

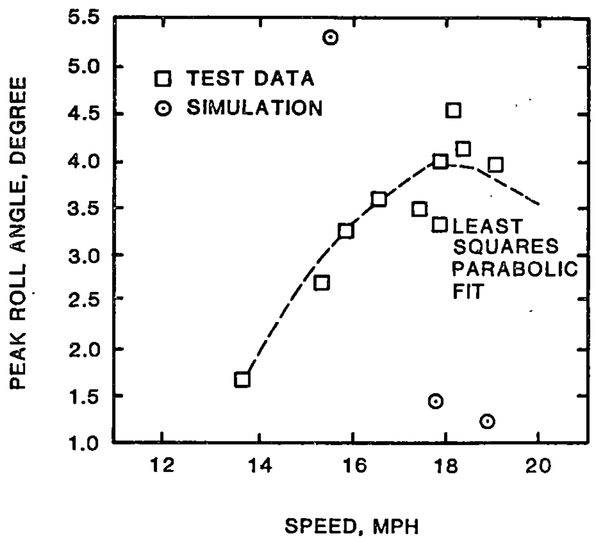

FIGURE 3-12. COMPARISON OF ROLL ANGLE FOR THE TESTS AND THE FLEXIBLE CARBODY VEHICLE MODEL SIMULATION RESULTS

3.4.2.2 FRATE. FRATE 11, FRATE/MITRE, and FRATE 17 (References 16 through 19) are nonlinear, time domain models that can be used to study the harmonic roll and bounce. The most basic of the three is Wyle Laboratories' FRATE 11 (see Figure 3/13). The eleven basic degree of freedom are lateral, vertical and roll of each truck, lateral, vertical, roll, yaw and pitch of the carbody. In addition, one degree of freedom is added for each normal mode of vehicle flexibility included in the carbody representation. Nonlinearities such as dry friction, finite spring travel, clearances, and stops are included. Mass, damping, and stiffness characteristics of the track are included by lumping them with corresponding elements representing the

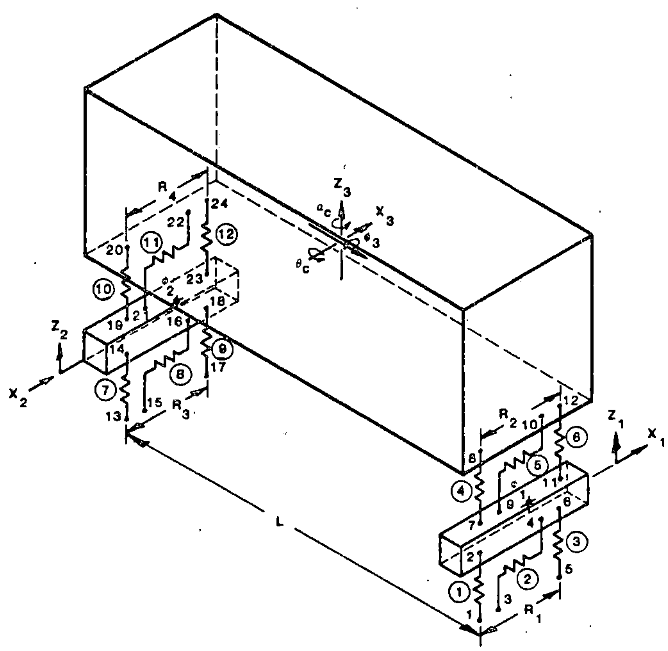

FIGURE 3-13. FRATE 11 MODEL

truck. FRATE/MITRE is an extension of FRATE 11 in which additional lumped elements are added for lading such as trailer on a flatcar. FRATE 17 was developed directly from FRATE 11. The additional six basic degrees of freedom are due to the inclusion of an additional mass for each truck which allows side frames and wheelsets to be treated separately from the bolster. Each additional mass has vertical, lateral, and roll freedom motion. FRATE 11 was selected as the primary tool of the three to be validated. The selection was based on past results showing that FRATE 11 and FRATE 17 produced very similar results regarding carbody motion.

Input to FRATE consists of tabulated track profile data which may be obtained along with the test data or generated from formulae for idealized profiles. Thus, depending on the particular track profiles used, the model can be used to investigate either harmonic roll or bounce response. The FRATE models have been partially validated against test data both by Wyle Laboratories and MITRE.

Prior validation work was performed using the ASF tests of a loaded, 100-ton hopper car on half-staggered shimmed track at Hartford, Illinois, in 1968, the same data used in the Flexible Carbody Vehicle Model Validation exercise. Table 3-3 summarizes the results for the validation exercise with the ASF data against the validation criteria described in paragraph 3.3.2.

Table 3-3. COMPARISON OF FRATE VS TEST RESULTS

Comparison	Criterion	Deviation
Peak Roll Angle at Resonance	$+1^{\circ}$	0.6°
Critical Speed	+1 mph	0.8 mph
Spring Nest Force at Resonance Load	+5% of static	3.5% of static

To extend the validation effort it was decided to compare FRATE 11 results with a different test case from the TDOP Phase I series. The case chosen was test number 191 (Reference 14), which describes the test of a loaded, 100-ton box car with Barber trucks, having standard suspension on half-staggered shimmed track. The truck center spacing was 46 feet. Approximately 400 feet of track were shimmed. The test section was traversed a number of times at constant speed beginning at 12 mph and increasing in increments of approximately 2 mph up to 20 mph. The comparison of the peak roll response versus speed results for the model and tests are shown in Figure 3-14. Figure 3-15 compares time histories of the model and test at the resonant speed. The close agreement is apparent despite the fact that TDOP Phase I test data did not include detailed track profile measurements but only the elevation difference at each rail joint. The input for FRATE in this case had to be idealized between each rail joint.

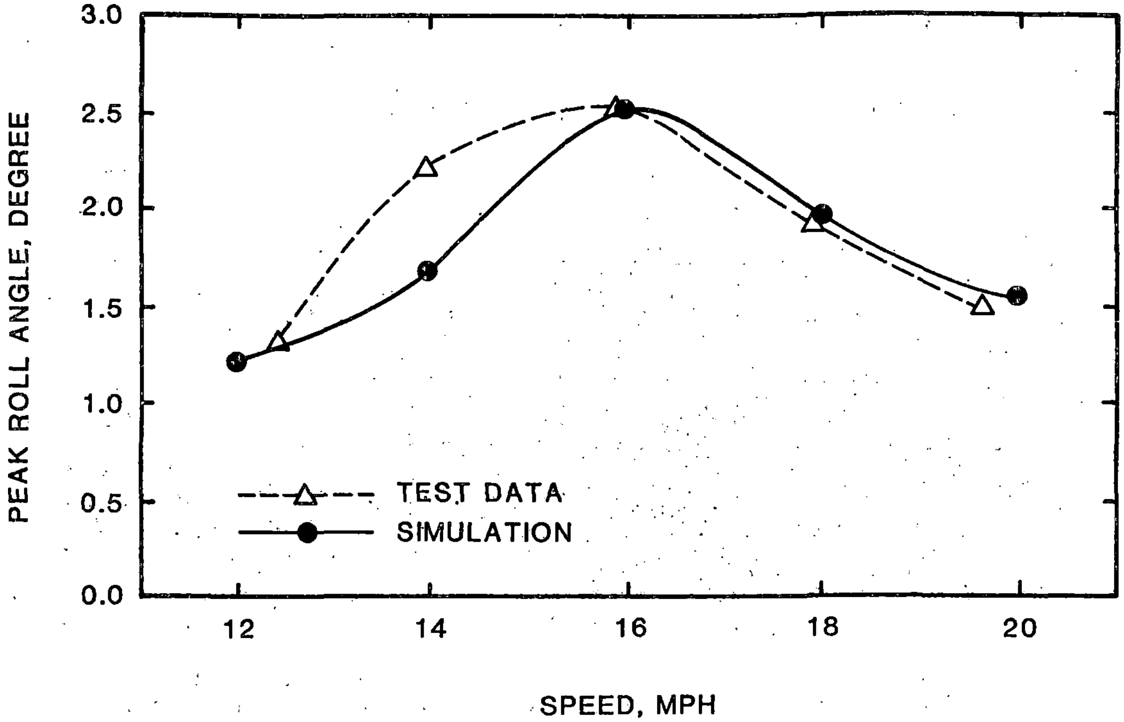

FIGURE 3-14. COMPARISON OF 11 DOF FRATE MODEL WITH SHIMMED TRACK TEST DATA

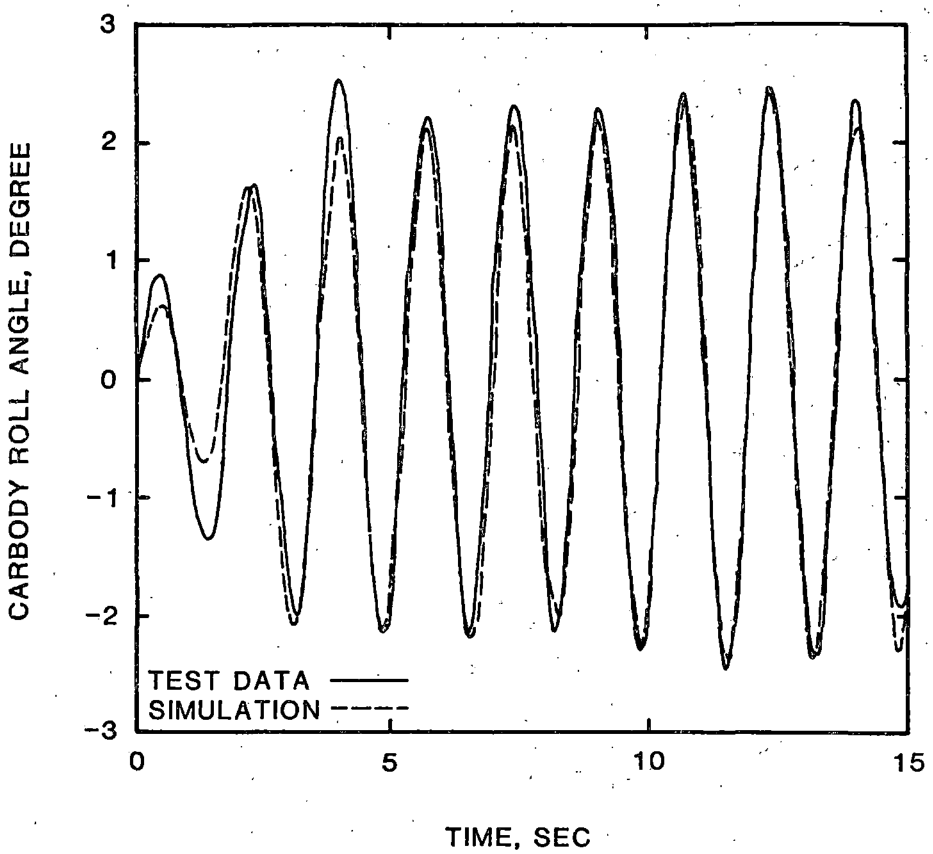

FIGURE 15. COMPARISON OF TIME HISTORIES OF 11 DOF FRATE MODEL AND TEST DATA AT RESONANT SPEED

Although FRATE 11 compared favorably within the validation criteria in the cases discussed, there are limitations to its use. It is known, for instance, from other validation work with FRATE (Reference 19) that the harmonic roll response of flexible flat cars requires a model of greater sophistication. Likewise, a more complex model such as FRATE 17 is recommended in cases where the excitation is sufficiently great to cause centerplate rocking. For the more rigid box type cars at excitation levels below that causing centerplate rocking, the FRATE 11 model produces satisfactory results.

3.4.3 Validation Results of Curve Negotiation Models

Originally, the 9 & 17 dof steady state curving models were selected for validation in the curve negotiation regime. These two models are nonlinear representations of a freight car in steady state curving (Reference 20). Closer scrutiny, however, showed that they were unsuitable for use on the TDOP Phase II project. Primarily, the problem has to do with the inability of these models to treat the nonlinearity associated with flange contact. In the course of talks with the Canadian National (CN) Rail Research Center on the potential use of their test data in TDOP Phase II, a curve negotiation model developed by them was discussed. Upon examining the potential of the model, it was determined that the model indeed accounted for responses including the condition of flange contact at the wheel/rail interface. Hence, it was decided to use the CN Rail Curving model for the steady state curving analysis instead of the 9 & 17 dof models.

The CN Rail Curving Model is an interactive program for the solution of steady-state behavior of trucks in curves. The program is designed to calculate force levels and geometric parameters such as angle of attack describing steady-state behavior on smooth circular arc curves, wholly neglecting dynamic effects.

The model used in the validation exercise represents a two-axle railway truck which is supporting a vehicle body and which is traversing a smooth curved track section having constant curvature and superelevation (Reference 21). Figure 3-16 shows the interfaces between the truck, vehicle body and the track. The vehicle body may be subjected to lateral coupler forces as well as to centrifugal and gravity forces.

The truck model consists of a rigid frame and conventional wheelsets having lateral and longitudinal stiffness characteristics at the primary suspension. Figure 3-17 shows a plan view of the truck. Each wheelset has a degree of freedom in the lateral, longitudinal, yaw and rotational directions, which, for the leading wheelset, are designated as Q_1, Q_2, Q_3 and Q_4 respectively. The truck frame itself has two degrees of freedom which are in the lateral and yaw directions. The model also has the capability of modeling inter-axle steering connections.

The wheel-rail interaction in the model is shown in Figure 3-18. A conical wheel tread is assumed to contact a crowned railhead. Hertizian contact areas, creep coefficients and creep forces are determined from simulated vertical wheel-rail forces using nonlinear curve fitting techniques. Spin creep is neglected, which is justified provided that the tread conicity is small or that the lateral and longitudinal creepages are relatively large.

When the flangeway clearance is exceeded, flange contact occurs. A conical flange face is assumed and point contact occurs between the flange and the gauge-face of the rail. The point of flange contact moves longitudinally depending on the wheelset angle of attack and on the specified flange geometry. The flange force consists of normal and tangential components. The normal force is modeled by a nonlinear stiffness between the flange and the rail. The magnitude of the tangential force is assumed to be a saturated creep force at the point of flange contact. Its direction is calculated from the exact creepages at the point of contact.

The validation efforts consisted of simulating the experimental curving tests of a standard 100-ton freight car truck (Reference 4). The curving tests consisted of running a 100-ton hopper car with Type I trucks over a test section of curved track. The test zone consisted of track ranging in curvature from 2.5 to 6.2 degrees and associated equilibrium speeds ranging from 34 to 48 mph. Three passes were made through the test zone for each condition, one at approximately 10 mph less than the nominal equilibrium speed, another at the nominal equilibrium speed, and a third at approximately 7 mph greater than the nominal equilibrium speed. The forces at the wheel/rail interface were measured using instrumented wheelsets, and the lateral and vertical forces were calculated by means of the axle bending technique.

Comparisons between the theoretical and experimental curving forces were made for the case of the loaded cars at balance speed. Typical results are shown in Figure 3-19. It can be seen that the model and the experiment results both predict a general increase in the lateral force of the leading outer wheel with an increase in track curvature. The model, however, predicts lower force levels than the test data. It should be pointed out here that data for many of the parameters of the actual hardware tested were not available, and that engineering estimates were used to determine their values. In addition, it should be noted that the parameter values were selected prior to the examination of the experimental results.

Figure 3-20 shows the graph of the Force Ratio* R_F for the leading outer wheel versus the superelevation deficiency for the theoretical and experimental results. The latter is given as plots of 27 individual points of the experimental data and by a plot of the least-square fit through these points. Both the theoretical and experimental results show an increase in the force ratio R_F with superelevation deficiency. The figure also shows that the mean slopes of the theoretical and experimental data are closely matched (0.085 for the former and 0.094 for the latter). Further details of the CN model can be found in Reference 21.

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*Force Ratio, $R_F = F_V/F_B$,

where:

F_V = measured lateral force on a given value of superelevation deficiency

F_B = measured lateral force at the balance speed

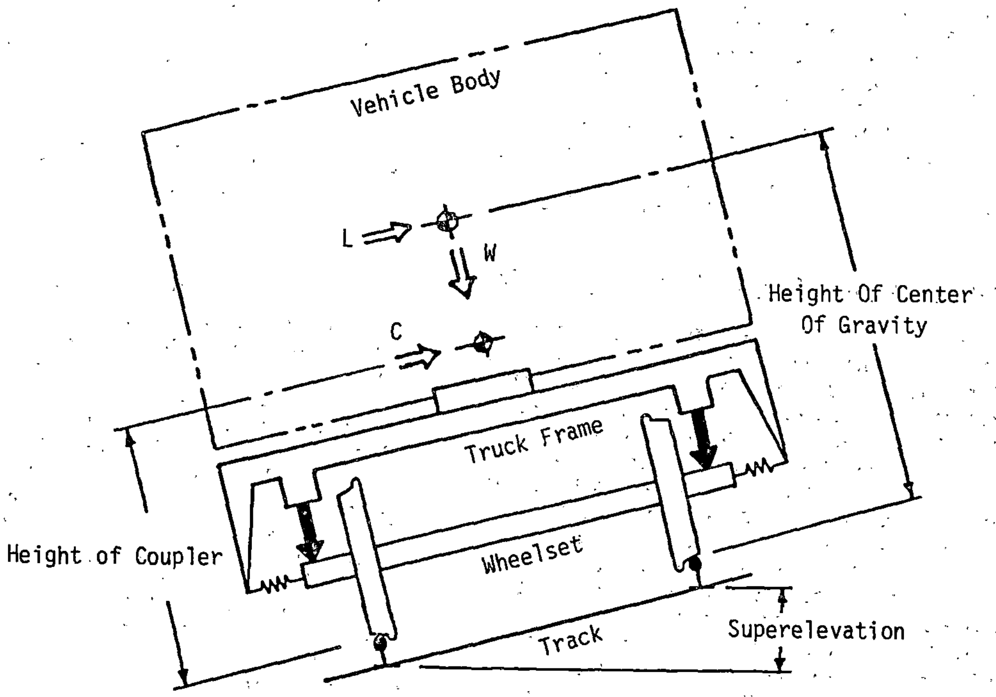


FIGURE 3-16. VEHICLE BODY/TRUCK/TRACK INTERFACES

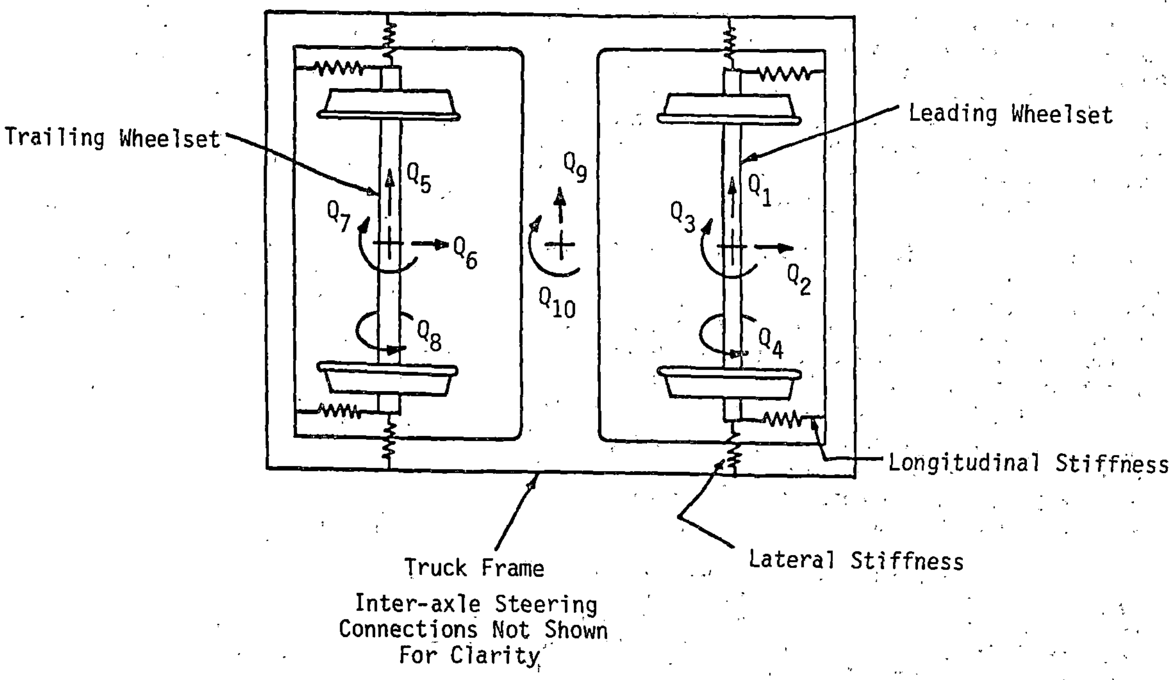
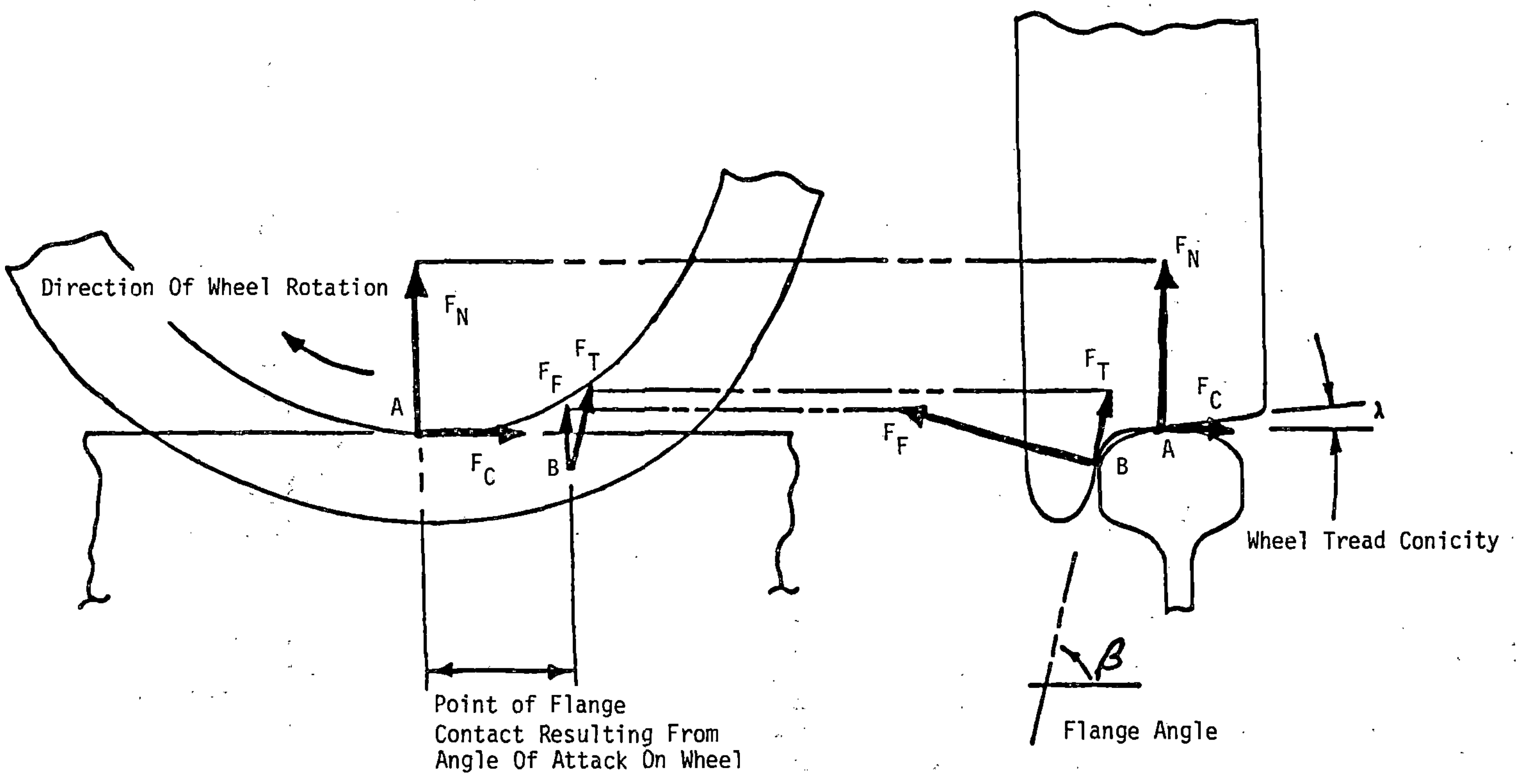


FIGURE 3-17. SCHEMATIC DIAGRAM OF 2-AXLE TRUCK AND ITS DEGREES OF FREEDOM (Q_1 TO Q_{10})



F_T = Tangential Flange Force

F_N = Normal Tread Force

F_F = Normal Flange Force

A = Point of Tread Contact

B = Point Of Flange Contact

FIGURE 18. SCHEMATIC DIAGRAM OF WHEEL/RAIL CONTACT FORCES

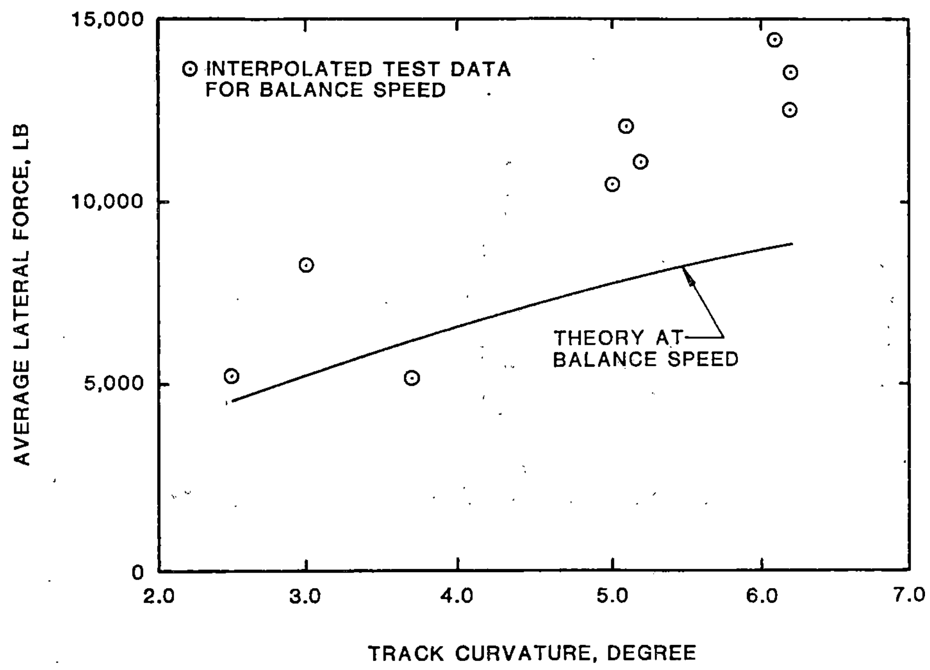


FIGURE 3-19. EFFECT OF CURVATURE ON LATERAL FORCE ON LEADING OUTER WHEEL - TEST AND SIMULATION RESULTS

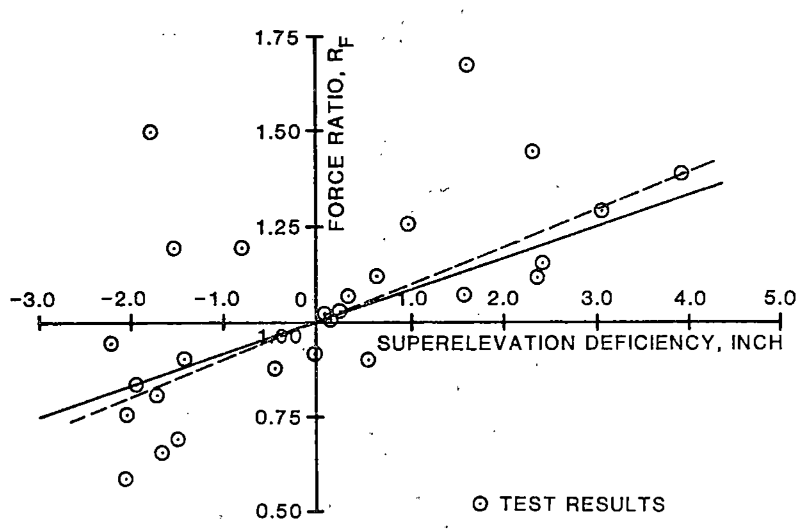


FIGURE 3-20. EFFECT OF SUPERELEVATION AND SPEED ON FORCE RATIO - TEST AND SIMULATION RESULTS

3.4.4 Validation Results of Ride Quality Models

The model selected as a candidate for validation in the ride quality regime was the DYNALIST modeling program (Reference 22). DYNALIST is a general purpose, computer program which solves systems of linear second order differential equations. Dynamic models of freight cars with up to 50 degrees of freedom can be analyzed both in the time and frequency domains. The DYNALIST program has no particular model structure but rather the program allows the user to define the structure by means of the input. The structure may be composed of rigid bodies, wheelsets with lateral degrees of freedom, model mass elements, springs and dampers. Flexible bodies can also be included by using an appropriate modal representation. The forcing function can be harmonic, periodic, or random in character. DYNALIST was selected for TDOP Phase II validation because of its flexibility, the extent of its prior use, its excellent graphics capability, and good documentation. It is particularly useful because of its capability of performing analysis in both the time and frequency domains.

The DYNALIST frequency domain modeling capability was applied to the 70-ton refrigerator car combined with track inputs, represented as spatial PSD's, to produce the response of the vehicle in the form of acceleration PSD's. This work was performed by Wyle Laboratories' subcontractor, the J.H. Wiggins Co. J.H. Wiggins followed the procedure used by other modelers (Reference 23) of separately modeling the dynamics of the vehicle in the vertical and lateral planes, and linearizing all friction mechanisms as well as the kinematics of wheel/rail contact. Typical results are given in Figures 3-21 and 3-22.

In the case of the vertical model it was found that, in the low frequency range, the results of simulation and tests could be made to match only if unrealistically high equivalent viscous damping was assumed. For the roll model, the response predicted by the model in the low frequency range was almost an order of magnitude low.

In attempting to explain the discrepancies in the vertical model, it was found that other modelers have also found it necessary to introduce high damping in order to make the results of simulation and tests agree; see Figures 3-23 and 3-24. The authors (Reference 23) concluded that nonlinearities in the suspension, not considered in the model, are responsible for these discrepancies, and that the flexibility of the carbody, considered rigid in the model, may be a contributing factor.

As mentioned elsewhere in this report, Coulomb friction excites higher frequencies in the carbody than does viscous friction. In addition, Coulomb friction raises the natural frequency of a system. In the case of the Barber S-2 truck, an additional nonlinearity is introduced by load-dependent snubbing, the magnitude of which, moreover, differs in the upward and downward directions.

However, it is believed that the discrepancies between the results of simulation and testing, while undoubtedly influenced by these approximations, are primarily due to a basic feature of the model, i.e., the separation of vertical and lateral dynamics. It may be seen from Figures 3-23 and 3-24 that the major discrepancies are fairly sharply localized in the frequency range between about 2 and 5Hz which contains the frequencies of the main lateral carbody modes, including lateral displacement, yaw and upper center roll.

This leads to the following possible explanation of the discrepancies. In the vertical model, the track inputs due to staggered rail or other irregularities are either assumed to be applied at the center plane of the vehicle, or equivalent, restraining moments are applied to prevent motion of the simulated vehicle out of the vertical center plane. The finite excitation energy supplied by the rail joints and other vertical track irregularities is, thus, entirely channeled into motions in the vertical plane while in the real vehicle a large portion can be expected to be converted into kinetic energy in the lateral modes at their particular natural frequencies. The suppression of the lateral response by means of unrealistically high damping detracts greatly from the validity of the model even if it succeeds in reducing the mismatch between simulated and observed amplitudes. The fact that, in the DYNALIST model, the introducton of carbody bending modes which have higher frequencies than the rigid body modes, did not succeed in improving the simulation, suggests that this feature does not constitute a necessary or fundamental elaboration of the model.

Several explanations are possible for the low response of the lateral model in the low frequency range. First, it appears that the (half) creep coefficients are too low by an order of magnitude. The greater part of carbody motion is due to wheel/rail excitation rather than to lateral wheel irregularities, and the magnitude of these wheel/rail forces is directly proportional to the creep coefficient. In addition, as is discussed elsewhere, a model with linearized damping tends to oscillate in the lateral modes at a frequency slightly higher than the frequency of kinematic hunting of the truck, between 1 and 1.5 Hz, rather than at the natural frequencies of the carbody that include yaw (between 2.5 and 3.1 Hz in the case of the refrigerator car) which can only be excited by the high frequency content of Coulomb friction.

In summary, while the insertion of unrealistic values of some parameters in a mathematical model may succeed in producing results within the specified validation tolerance of the test results, a model distorted to this extent does not appear to have much practical value for such important procedures as suspension design. Thus, while frequency domain simulation may be useful in checking out subsystems, it is doubted whether the complexity of a frequency domain model containing both vertical and lateral degrees of freedom is more economical than even a simplified time domain model. On the basis of the results with DYNALIST, the structured frequency domain models shown in Table 3-2 (HALF, FULL, FLEX, and LATERAL - Reference 24) were not treated since they are also uncoupled configurations.

3.5 MODEL UTILIZATION

With reference to the use of mathematical models in defining and interpreting the performance levels of freight car trucks, the extent of such use has varied from one performance regime to another. In most cases, the use of models turned out not to be feasible for both technical and economic reasons.

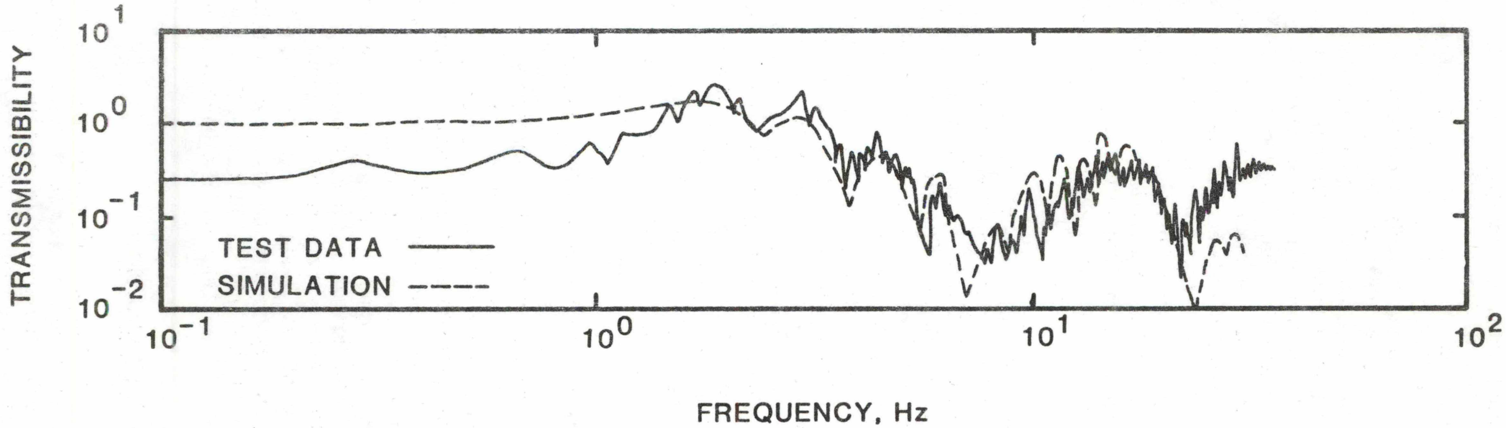


FIGURE 3-21. VERTICAL TRANSMISSIBILITY AT 60 MPH - TEST AND SIMULATION RESULTS

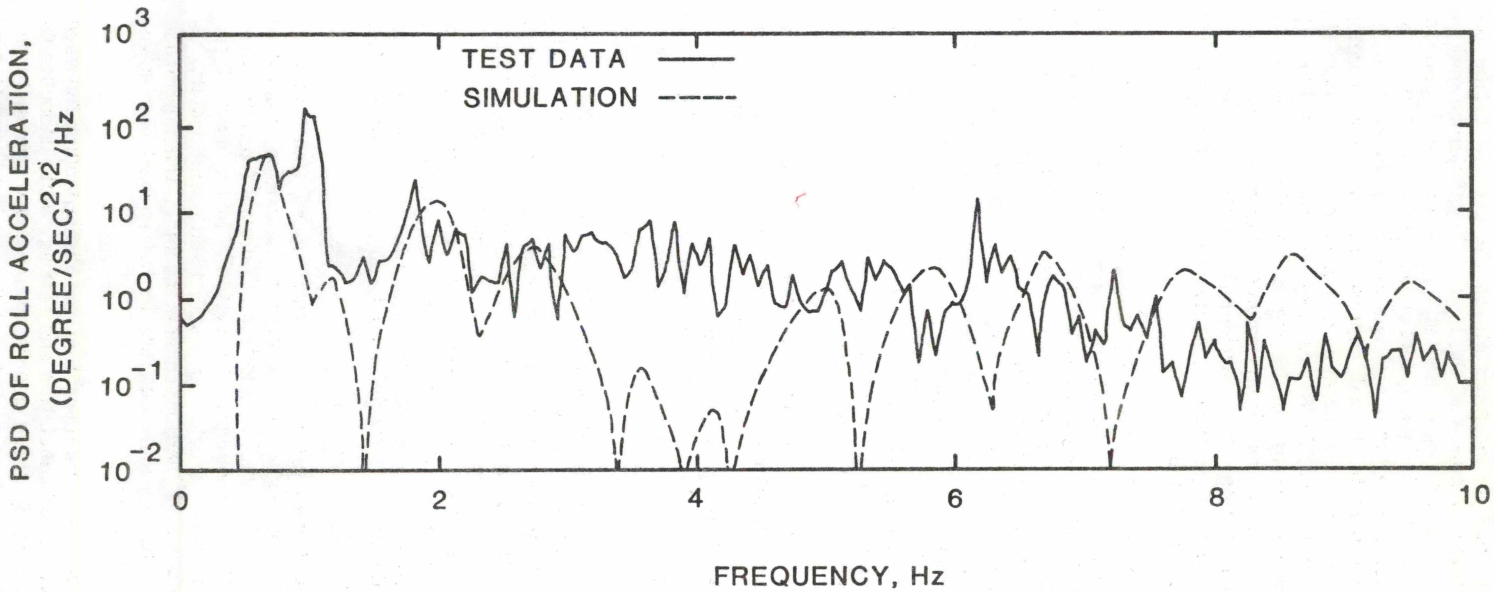


FIGURE 3-22. POWER SPECTRAL DENSITIES OF CARBODY ROLL ACCELERATION - TEST AND SIMULATION RESULTS

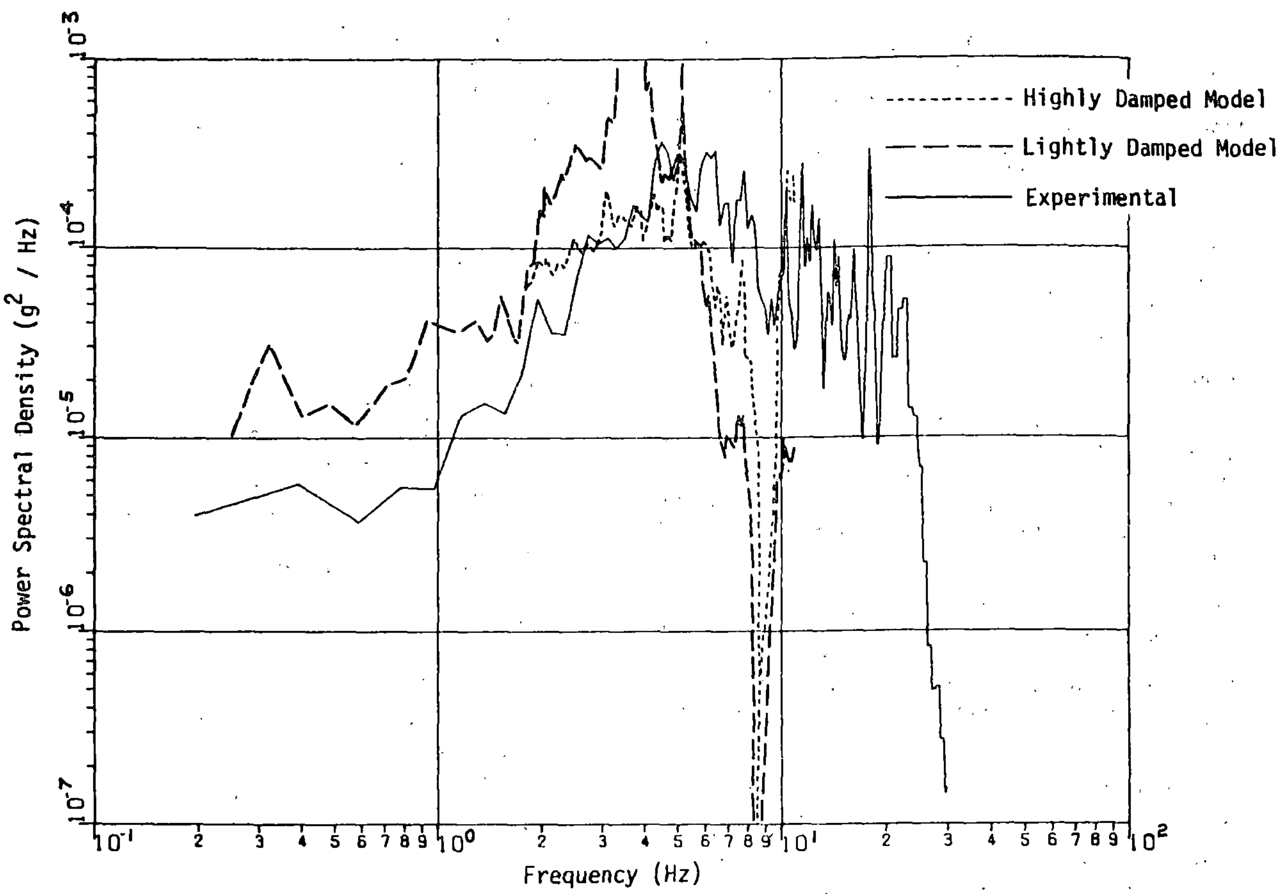


FIGURE 3-23. POWER SPECTRAL DENSITIES OF CARBODY FRONT END VERTICAL ACCELERATION (EMPTY VEHICLE TRAVELING AT 100/FT SEC OVER CWR)

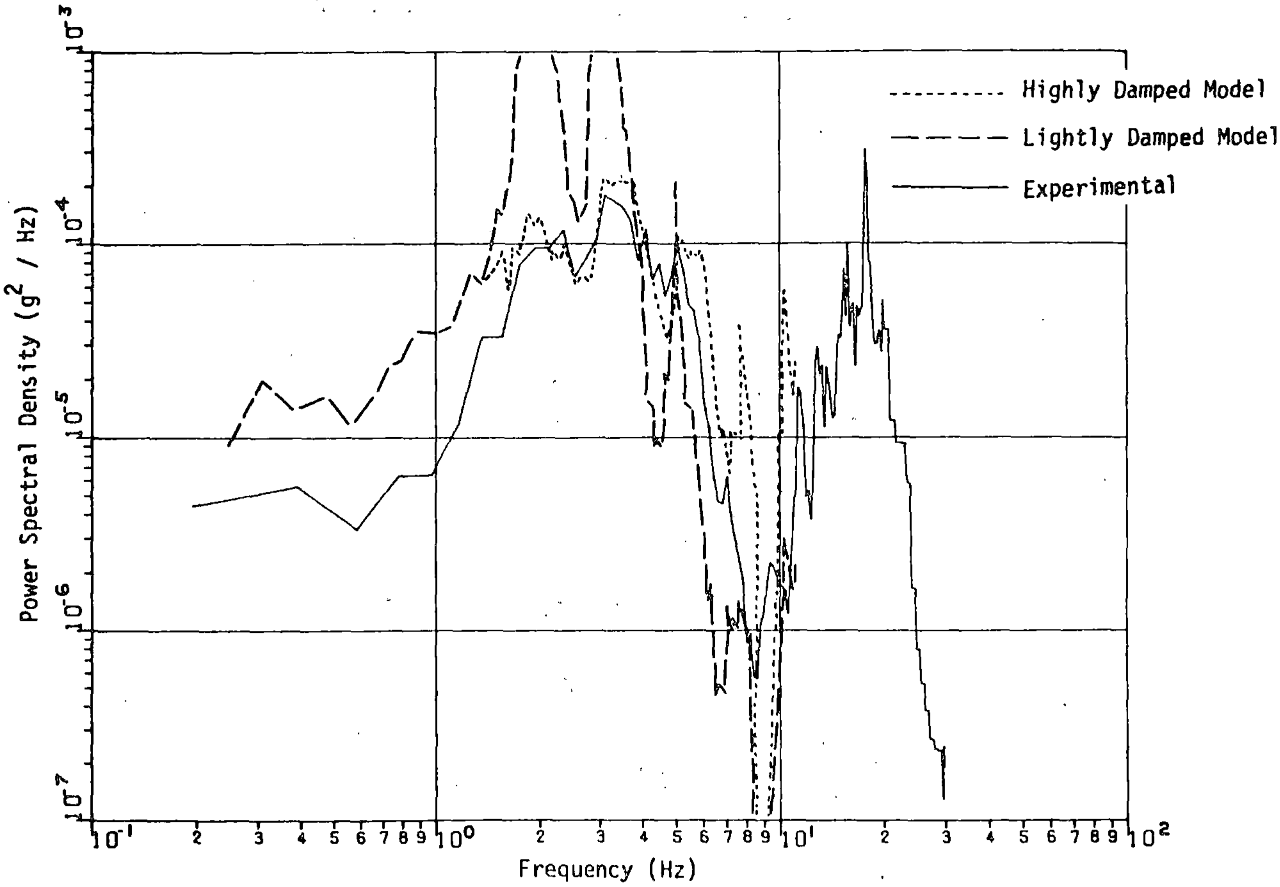


FIGURE 3-24. POWER SPECTRAL DENSITIES OF CARBODY FRONT END VERTICAL ACCELERATION (LOADED VEHICLE TRAVELING AT 100 FT/SEC OVER CWR)

In the lateral stability regime, models were used primarily in an interpretative mode, i.e., addressing specific questions relating to the results from the test data. For example, the test data revealed a consistent tendency on the part of the box type freight cars to initiate the nosing phenomenon prior to the development of hunting. Since no apparent explanation was readily available from the test data or the operating conditions under which they were obtained, analytic simulations using models were used. Additionally, the hunting frequencies associated with results from test data were confirmed through analytic simulations. Analysis also helped address key parameters of influence in freight car hunting, especially as relating to wheel/rail contours and contact geometry.

In the case of harmonic roll, simulations were compared against field test results with good agreement. However, only a minimal amount of field test data were available for this subregime and use of models to extend these sparse results was not considered judicious.

Simulations using a nonlinear curve negotiation model were compared against field test data. The validation exercise proved to be encouraging. Furthermore, Transport Canada has an effort underway to further validate the model against test data from Type II trucks.

In the regime of ride quality, initial efforts centered around simple models in the vertical and roll modes which assumed that these modes were decoupled. Verification against test data proved this assumption unjustified. Restructuring the model to overcome the deficiencies could not be rationalized in light of the abundance of data available for use in the ride quality regime.

Thus, although analytic models were utilized in the simulation of truck behavior, field test data remain the primary basis of the quantified performance characterizations presented in this report. The results from the test data were interpreted and correlated to appropriate operating conditions and parameters of significant influence through the use of analytic simulations, as well as through existing knowledge of the behavioral performance of freight car trucks.

3.6 REMARKS

The results of the analysis tasks carried out thus far have been less than encouraging. With few exceptions, the model results have not agreed with test data within the tolerance of the validation criteria selected. In some cases, faults in the programming and model formulation are to blame. In others, the test data are suspect. The difficulty in obtaining good agreement between model and test data is illustrated by the spread in test results from replicated conditions (e.g., Figure 3-25 taken from Reference 25).

It is recommended that greater emphasis be placed on simple modeling. The aim of such modeling is the interpretation of test results. For some models (such as the CN Curving Model and HUNTCT) additional validation work is justified.

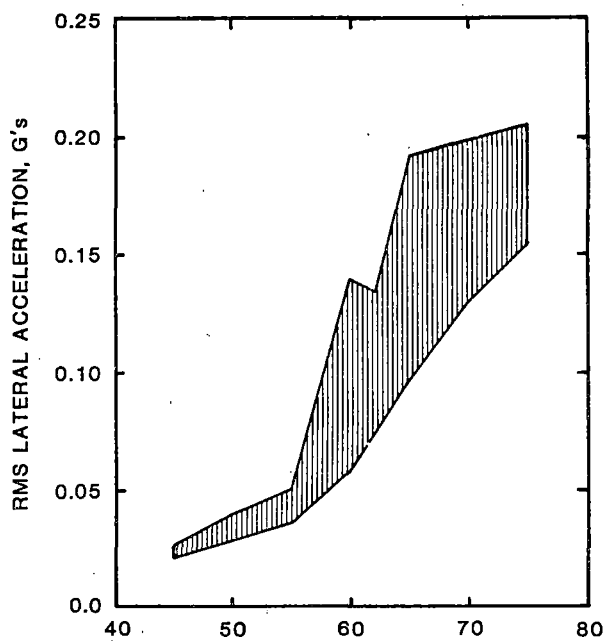


FIGURE 3-25. VARIATION IN PEFORMANCE OF STANDARD TRUCK FOR COMPLETE TEST PROGRAM

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SECTION 4 - ENGINEERING

4.1 INTRODUCTION

One of the primary objectives of the Truck Design Optimization Project (TDOP) Phase II was to define the engineering options available to the railroad industry in order to improve the efficiency and productivity of rail freight transportation. Results from experimental and theoretical investigations were applied, in consultation with the industry, to the development of performance characterizations of Type I (standard, three-piece) and performance specifications for Type II (premium) freight car truck configurations.

Proposed performance guidelines for Type I trucks (Reference 1), as represented by quantified characterizations, were developed principally on the basis of performance test data generated during Phase I of TDOP and supplemented, wherever necessary, by Phase II data. Characterizations and specifications of performance for Type II trucks (Reference 2) were developed from test data acquired during Phase II of the project.

Freight car truck performance has been divided into four distinct regimes which, taken together as inclusive sets of conditions associated with predominant features, identify all aspects of truck behavior. These regimes are identified as lateral stability, trackability (harmonic roll, track twist, and curve entry/exit), steady state curve negotiation, and ride quality. Performance indices, which represent measurable quantities typical of performance, are defined in each of the performance regimes (Reference 3).

Quantitative performance for Type I and Type II trucks presented in this report are defined by ranges of performance indices in each performance regime, specifically related to operating conditions such as speed, track quality, degree of track curvature, and lading. The quantified range of performance indices, developed from field test data, has been interpreted in the light of physical reasoning and tempered by comparative data studied by means of simple analytic and engineering models. Within the domain of statistical significance of the test data upon which the present characterizations and specifications are based, it is expected that tests involving similar equipment and conditions are likely to produce results comparable to the quantified ranges of performance presented in this report.

The results represent a comprehensive characterization of performance of the freight car trucks, embodied in a range of quantified performance indices which are relatable to the economics of railroad operations. Therefore, it is believed that this body of results can be used to provide the basis for a set of performance specifications for freight car trucks which could be useful in railroad procurement and maintenance operations, as well as provide a guideline or basis for equipment manufacturers.

In translating the performance characteristics of Type II trucks into a set of recommended guideline specifications, the test results were interpreted and engineering judgment exercised in correlating factors such as the influence of expected component wear on performance and possible economic implications. The resulting guidelines form a set of recommended "performance specifications" for Type II freight car trucks.

As more information becomes available on such factors as wheel and rail wear, and truck component wear (from the TDOP Phase II Wear Data Collection Program in progress) as well as other sources, the recommended "specifications" may be further refined to reflect factual influences replacing the judgment factor. However, it is believed that the recommended guidelines provide a framework to define a set of improved performance levels associated with design changes.

This section also includes a set of standard test specifications for the performance requirements established in this project. The objective of the test specifications is to provide guidelines for the acquisition and analysis of field test data, so that the results can be evaluated against the recommended guideline specifications.

4.2 METHODOLOGY FOR DEVELOPMENT OF QUANTIFIED PERFORMANCE LEVELS

Establishment of an analytic and experimental methodology for relating truck parameters to the economic-related performance indices defined in each of the performance regimes is a major engineering goal within TDOP Phase II. Applying this methodology, and in coordination with industry, guideline performance characterization for Type I trucks and guideline performance characterizations and specifications for the Type II trucks were developed.

The major elements comprising the methodology for truck evaluation are:

- Field testing of selected trucks to obtain performance test data and a thorough evaluation of the procedures involved in the acquisition of test data.
- Reduction of the data, followed by interpretation, to ensure that the test results are consistent in terms of physical principles as well as of specific characteristics of the vehicle and test environment.
- Simulations utilizing credible mathematical models to augment and complement results from field test data.
- Determination of wear and degradation of freight car trucks under revenue service conditions through a structured program of periodic measurement of various truck components including wheels.
- Correlation of results from analysis of economic data on truck maintenance and operation from operating railroads with results from analysis of performance test data.
- Comparison of test results with results obtained from comparable tests of similar vehicles, to identify and resolve any major discrepancies.
- Establishment of performance boundaries for both Type I and Type II trucks.

A block diagram indicating the flow of elements in this methodological scheme is shown in Figure 4-1.

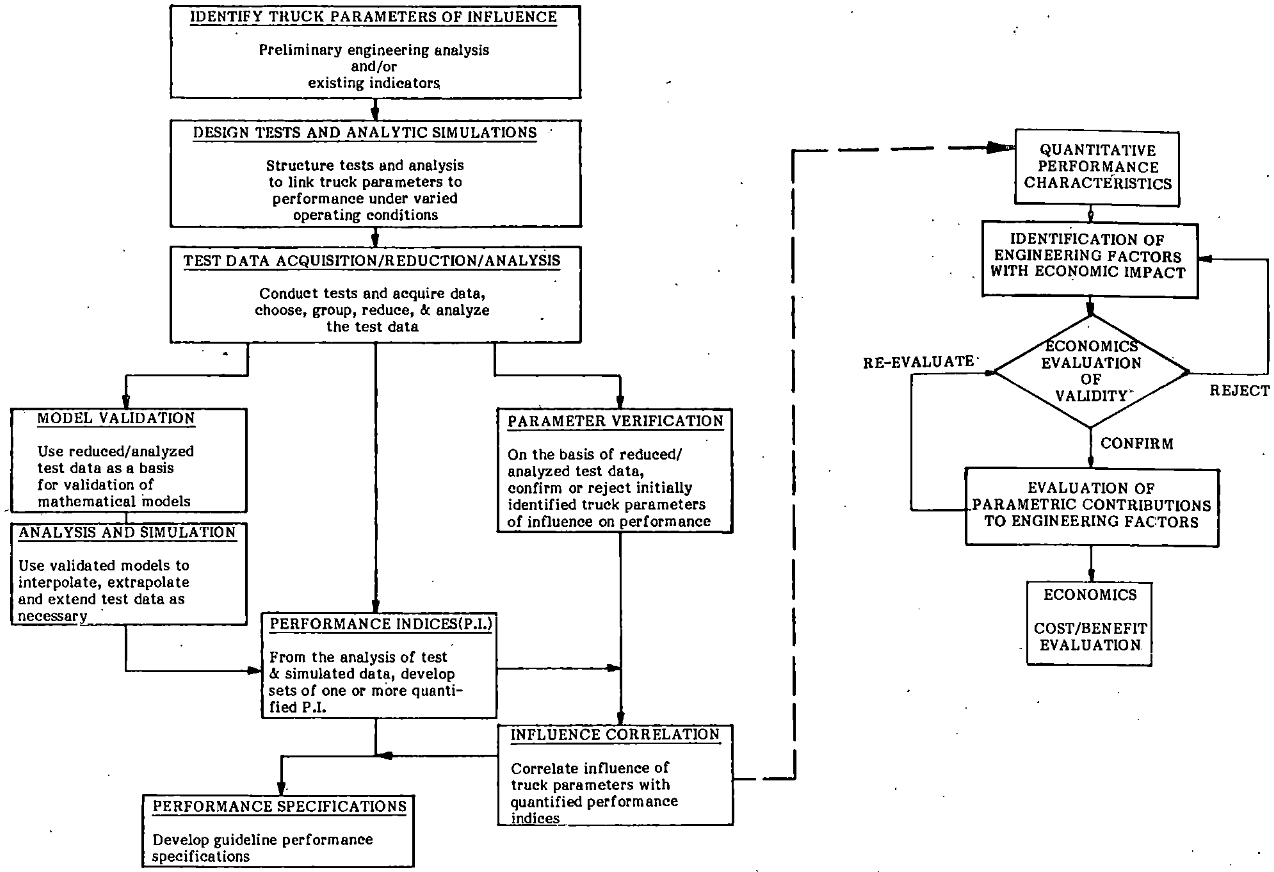

FIGURE 4-1. METHODOLOGY FOR TRUCK EVALUATION

4.3 PERFORMANCE CHARACTERIZATION OF TYPE I TRUCKS

The intent of the guidelines presented in this section is to identify Type I freight car truck performance levels which can be correlated to savings associated with reduced maintenance, longer equipment life, and other tangible benefits in terms of railroad operations. The development of these guidelines has kept in perspective common industry practices and the Association of American Railroad (AAR) requirements for interchange service and regulatory safety requirements.

The quantified levels of performance given under each of the performance regimes represent the results of analysis and interpretation of quantified test data.

4.3.1 Lateral Stability

Characteristic performance levels in the regime of lateral stability are given in terms of rms lateral acceleration and peak lateral acceleration (Reference 1). A summary of results from analysis of test data in the lateral stability regime is given in Table 4-1. The results of the test data analysis have been summarized for the two classes of trucks, namely, 70-ton and 100-ton trucks.

70-Ton Trucks

The empty flat car using worn wheels shows the earliest evidence of instability, at a speed range between 30 and 40 mph. The maximum acceleration level for this case increases sharply to 0.55 g at 40 mph, and to 1.1 g at 79 mph. No data are available on the same configuration in the loaded condition.

For the flat car using new wheels, test data for the loaded and the empty conditions exist. The analysis reveals that the loaded configuration shows no evidence of hunting and the vehicle remains stable through the entire range of operating speeds up to 79 mph. However, in the empty condition with new wheels, hunting is evidenced in the speed range between 70 and 79 mph. In general the critical speed of a flat car varies depending on the operating conditions, and, even more so, on the wheel profiles, with the predominant frequency of hunting being in the range of 2.5 to 2.9 Hz.

The behavior of the mechanical refrigerator car and the box car can be placed into one category since the findings drawn from the performance test data indicate general conformity. Therefore, they will be grouped together and referred to as "box type" cars.

TABLE 4-1. SUMMARY OF RESULTS FROM ANALYSIS OF TEST DATA IN THE LATERAL STABILITY PERFORMANCE REGIME

Vehicle Configuration	Hunting* Yes/No	Critical** Speed Range mph	Hunting** Frequency Hz	RMS Lateral** Acceleration g	Peak Lateral** Acceleration g	Remarks
70-ton Trucks with Box Type Cars						
1. New Wheels/Empty	Yes	40-50	2.5-3.1	0.16-0.36	0.58-1.24	Initiation of nosing at 40-50 mph; fully developed hunting at 60-70 mph.
2. New Wheels/Loaded	No					Nosing initiated in the 60-70 mph speed range with an associated frequency range of 3.3 to 3.5 Hz, RMS acceleration of 0.27 to 0.29 g, and peak acceleration levels of 0.67 to 0.77 g. Nosing continued through the speed range up to 79 mph, the terminal speed for the test runs.
3. Worn Wheels/Empty	Yes	40-50	2.5	0.18-0.40	0.66-1.12	Leading end nosing & trailing end intermittent hunting at 40-50 mph. Both ends hunting with increasing speed.
4. Worn Wheel/Loaded	No	—	—	—	—	No evidence of hunting.
70-ton Trucks with Flat Cars						
1. New Wheels/Empty	Yes	70-79	2.8	0.11	0.59	Fully developed hunting at 70-79 mph.
2. New Wheels/Loaded	No	—	—	—	—	No evidence of hunting.
3. Worn Wheels/Empty	Yes	30-40	2.2-2.9	0.12-0.30	0.55-1.10	Fully developed hunting at 30-40 mph.
4. Worn Wheels/Loaded	—	—	—	—	—	No data available.
100-ton Trucks with Box Type Cars						
1. New Wheels/Empty	Yes	70-79	2.7	0.10-0.25	0.73-0.83	Fully developed hunting.
2. New Wheels/Loaded	No	—	—	—	—	No evidence of hunting.
3. Worn Wheels/Empty	—	—	—	—	—	No data available.
4. Worn Wheels/Loaded	—	—	—	—	—	No data available.

*"Hunting" denotes full-body hunting as differentiated from nosing.

**Includes nosing and full body hunting.

In the case of the empty box type cars with new wheels, hunting begins at a speed between 40 and 50 mph. With the loaded box type cars, there is an indication of "nosing," i.e., hunting restricted to the leading end of the carbody only, initiated in the speed range between 60 and 70 mph, and continuing until 79 mph. To generalize, in this set of configurations, the critical speed increases with increasing loads; with regard to the effect of wheel profiles, the empty cars hunt at a lower frequency with worn wheels than with new wheels. While the empty box type cars hunt at a frequency slightly above 3 Hz with new wheels, those with worn wheels hunt at about 2.5 Hz. The effect of wheel profiles on amplitude response is not significant. For the empty box type cars, maximum acceleration levels range from 0.66 g at 50 mph to 1.25 g at 79 mph. From the very limited test configurations and results, which may not be typical, wheel profiles are seen to have no significant effect on performance in the case of the loaded cars. It is emphasized here that all of these observations regarding the influence of wheel profiles on truck performance are on the basis of limited test data; generalization to all freight car trucks is not intended.

100-Ton Trucks

The test data available in this area cover a 100-ton box car and a 100-ton covered hopper car equipped with new wheels. No data are available with respect to worn wheels. Analysis of reduced data indicates that empty cars on new wheels exhibit hunting in the speed range between 70 and 79 mph. Further, the leading end of the carbody experiences more pronounced motion than the trailing end in the speed range between 60 and 70 mph. The peak lateral acceleration level experienced is about 0.8 g in the 70 to 79 mph speed range. In contrast, the loaded car configuration remains stable through the entire range of operating speeds up to 79 mph.

4.3.2 Trackability

Test data were acquired on harmonic roll, track twist, and curve entry/exit.

Results of the reduction and analysis of the test data on the shimmed track showed that the loaded refrigerator car has the ability to extract energy from the track input excitations, and the carbody reaches a state in which the rocking car exceeds its capability to dampen or absorb rolling motion. The peak roll angle at the leading end of the carbody is 2.9 degrees for the refrigerator car at about 14 mph. The peak roll angle at the leading end of the carbody for the 100-ton box car is 2.4 degrees at about 14.5 mph. The results also showed that the data are quite nonlinear, and contain higher frequency components, particularly when acceleration responses are considered. The carbody is rolling about the lower center; in other words, the mode excited is the lower center roll.

Characterization of performance in the track twist subregime is provided by means of the wheel unloading index (WUI), which is the zero-to-peak value extracted from the time history. Performance characteristics are shown in Tables 4-2 and 4-3.

Results of the reduction and analysis of test data indicate that the dynamic components of the lateral forces and L/V ratios are high. The wheel unloading index for the loaded car has a mean value of 0.138 and a standard deviation of 0.065 on the 16 degree curve. The corresponding values for the 15.75 degree curve are 0.208 and 0.108, respectively. The mean values of the wheel unloading index for the empty car are 0.409 and 0.264 for the 16 degree curve and 15.75 degree curve, respectively, with standard deviations of 0.083 and 0.73, respectively.

The wheel unloading index is substantially higher for the unloaded car than that for the loaded car. This is mainly due to the friction snubber in the suspension which permits little motion between the truck components for the empty car. It may be noted, however, that the field test data considered here included only the constant friction snubber trucks.

Characterization of performance in the curve entry/exit subregime also is provided by means of the wheel unloading index. Analysis of the data indicates that, in general, the peak value of the wheel unloading index increases with increasing degree of curvature. The effect of speed on this index is not clear (i.e., does not have a constant pattern) from the results. This might be due in part to the dependence of this index on just one point extracted from the time history, and in part to the dependence of the car response on the track memory of the truck. Rail contamination and vehicle nonlinearities may also lead to this phenomenon. However, it has been noticed that the empty cars experience a higher wheel unloading index than the loaded cars on all curves tested.

4.3.3 Steady State Curve Negotiation

The results of the reduced data show that the lateral forces and L/V ratios increase with increasing degree of curvature and they tend to have the same characteristics. For the moderate curves of 2.5 degrees and 3 degrees, the lateral forces on the leading outer wheel of the loaded car are comparable. However, these lateral forces show substantial increase in magnitude as the degree of curvature increases, reaching an approximate value of 14,000 lb at the 6.2-degree curve. The ratio of the dynamic lateral forces to the steady state lateral forces are lower for higher degree of curvature. The values of the lateral forces and L/V ratios in both the forward and reverse directions are comparable. By comparing the results for the loaded and empty cars, the following may be stated:

a. The rate of increase of the lateral forces and L/V ratios are more critical for the loaded cars than for the empty cars. This conclusion is based on mean values of L/V ratios at balance speed without considering the associated time duration.

b. The rate of increase of the lateral forces and L/V ratios on the leading outer wheel with increasing degree of curvature is higher for the loaded cars than for the empty cars.

c. The ratio of the dynamic components of the lateral forces and L/V ratios to the steady state components are higher for the empty cars than for the loaded cars. This indicates that the dynamic effect of both curve entry and track irregularities is much higher for empty cars than for loaded cars.

4.3.4 Ride Quality

Characterization of performance in the ride quality regime is provided by means of quantified performance indices identified for the regime. These indices are: transmissibility, and rms response over the wide band spectrum.

Transmissibility, as presented here, is identified as the ratio of the rms value calculated from the response power spectral density within a specified frequency bandwidth to the rms value calculated from the track input power spectral density over a corresponding frequency bandwidth.

Transmissibility has been quantified in both the vertical and the roll directions. Vertical acceleration response at the sill level and roll acceleration response at either end of the carbody in the frequency bandwidths of 0 to 4 Hz and 4 to 10 Hz have been considered. The corresponding input consisted of power spectral densities of track profile in respect to vertical response and track cross level in respect to roll response in the same frequency bandwidths.

The rms values of the response power spectral densities for both the vertical and the roll accelerations were computed over the frequency range of 0-20 Hz as an additional performance index and plotted as a function of speed.

70-Ton Trucks

In general, loaded box type cars on 70-ton trucks indicate increasing values of rms vertical acceleration with increasing speed and a tendency to resonate in the vertical plane at about 50 mph. In the case of empty box type cars, the levels of vertical acceleration response are higher as compared to the response of the loaded cars, the implication being that loaded cars obtain better ride quality than empty ones. However, in one case, the loaded box car indicated higher levels of vertical acceleration above 40 mph as compared to those of the empty car. This case is considered the exception rather than the rule, and one possible explanation for this phenomenon is the coincidence of the natural frequencies of the carbody with those of the excitations from the jointed track, as well as the coincidence of the truck center spacing with the spacing of rail joints.

In the case of the flat cars, only the loaded configuration has been analyzed since the empty configuration was extensively covered by the lateral stability regime by virtue of indications of hunting. The flexural modes of vibration of the car are believed to be significant contributors to the car response.

In the roll mode, the amplitude response of the loaded box type cars is lower than that of the empty cars and the principal reason is considered to be the lower level of friction damping in this mode. Analysis of data on the loaded flat car indicates that the contribution from the torsional mode of vibration is significant. At about 40 mph the response peaks, with the leading end undergoing higher amplitude response than the trailing end.

100-Ton Trucks

Once again, the loaded box type cars on the 100-ton trucks exhibit better vertical ride quality characteristics as compared to those of the empty cars. The difference in the responses between the empty and the loaded cases is attributable, at least in part, to the higher natural frequencies of the empty cars and the effect of friction snubbing.

Among the box type cars in the roll mode, the hopper cars indicate lower levels of amplitude response as compared to those of the box cars. The trailing end of the carbody undergoes higher levels of roll acceleration than the leading end.

TABLE 4-2. STATISTICAL SUMMARY OF WHEEL/RAIL LOAD DATA OF 100-TON TRUCKS (WITH EMPTY BOX TYPE CARS FROM CURVED YARD TRACK TESTS)

	Vertical Load (lb)		Lateral Load (lb)		L/V Ratio		Wheel Unloading Index	
	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.
9.3 mph, 16° Curve								
-.26" Superelevation							0.409	0.083
Leading Outer Wheel	9330	680	1850	1000	0.196	0.103		
Leading Inner Wheel	5440	710	2030	680	0.366	0.135		
Trailing Outer Wheel	8670	970	-15	1030	0.009	0.114		
Trailing Inner Wheel	9740	630	440	740	0.043	0.079		
9.3 mph, 15.75° Curve								
-.3" Superelevation							0.265	0.073
Leading Outer Wheel	7960	920	3210	1510	0.399	0.186		
Leading Inner Wheel	7730	1050	1720	790	0.222	0.105		
Trailing outer Wheel	10080	1280	1710	1780	0.043	0.079		
Trailing Inner Wheel	10430	1860	-1450	1680	-.106	0.130		

TABLE 4-3. STATISTICAL SUMMARY OF WHEEL/RAIL LOAD DATA OF 100-TON TRUCKS (WITH LOADED BOX TYPE CARS FROM CURVED YARD TRACK TESTS)

Test Condition	Vertical Load (lb)		Lateral Load (lb)		L/V Ratio		Wheel Unloading Index	
	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.	Mean	St. Dev.
9.8 mph, 16° Curve -.26" Superelevation							0.138	0.065
Leading Outer Wheel	28300	2880	9790	4430	0.35	0.163		
Leading Inner Wheel	26570	2860	6340	1800	0.239	0.066		
Trailing Outer Wheel	29620	2960	5430	8800	0.175	0.191		
Trailing Inner Wheel	26840	2320	4960	8390	0.317	0.303		
8.55 mph, 15.75° Curve -.30" Superelevation							0.208	0.108
Leading Outer Wheel	24560	3700	8230	2000	0.342	0.0995		
Leading Inner Wheel	28730	3200	12460	5580	0.451	0.234		
Trailing Outer Wheel	25270	2830	9830	2070	0.396	0.103		
Trailing Inner Wheel	26200	5050	12400	3520	0.483	0.148		

4.4 PERFORMANCE CHARACTERIZATION OF TYPE II TRUCKS

Test data acquired from field tests conducted in TDOP Phase II on the seven Type II trucks were analyzed through digital computers and software packages especially tailored to meet the data reduction requirements within the project. The computer outputs of data analyses were arranged in digital printout and plot formats to facilitate analysis and presentation of the results. Data pertaining to each performance regime were first examined for quality; then, the total time history in each of the tests was reviewed in the process of selecting appropriate windows on the data to be analyzed; finally, selected data were analyzed in keeping with specific engineering and analytic requirements for quantitative definition of performance characteristics. The results in each of the performance regimes included digital printouts allowing for statistical analysis, and various forms of plots defining functional relationships of performance characteristics with operational variables included in the test conditions.

As a result of the analysis of field test data on the performance of the Type II trucks, quantified levels of performance could be studied as functions of operational variables. Analysis of test data permitted quantification of the performance indices defined with each of the performance regimes (Reference 2). Table 4-4 listed the names of the seven Type II trucks tested in this program.

In making any comparison with the performance levels associated with Type I trucks described in Section 4.3 and Reference 1, one cautionary note is important to keep in mind. That is, the Type II trucks were tested in conjunction with an open hopper car and the trucks used the CN profile wheels; whereas, the Type I trucks tested under TDOP Phase I were in conjunction with carbodies inclusive of boxcars and covered hopper cars, and the trucks used the AAR Standard 1:20 taper profile wheels. Although the Type I trucks tested in TDOP Phase II were tested in conjunction with an open hopper car, the trucks did use the AAR Standard 1:20 taper wheel profiles. One other cautionary remark is in order; namely, that the Alusuisse truck was a 70-ton truck as compared to the other six Type II trucks which were all 100-ton trucks.

4.4.1 Lateral Stability

A summary of the results is presented in Tables 4-5 and 4-6 on test configurations with empty cars and loaded cars, respectively. Since sustained hunting was observed in only relatively few cases, the analysis considered in some detail the intermittent hunting phenomenon.

4.4.2 Trackability

Quantification of performance characteristics in this performance regime covered the subregimes of harmonic roll, track twist, and curve entry/exit. Performance test data covering the harmonic roll subregime consisted of data from test runs on branch line, Class 2 track. Analysis of the test data indicated that the excitations arising from the track irregularities were not sufficient to cause the rock and roll phenomenon. This phenomenon is characterized by roll angles in the range of 3 to 5 degrees. The test data, however, showed a moderate response with the roll angle being in the range of 0.5 to 1.0 degrees. Therefore, no characterization of performance for the Type II trucks in this subregime is provided. The performance characteristics presented in this section cover only the two subregimes of track twist and curve entry/exit.

The preformance index defined in the subregime of track twist is the wheel unloading index (WUI). To provide some statistical significance associated with the quantitative values presented, the index presented is the 95th percentile; and the average value as well as the standard deviation of the index are given. In descriptive terms, the 95th percentile indicates that the value of the wheel unloading index given is likely to be exceeded only 5 percent of the time during a single passage through the spirals. The results presented represent the performance of trucks as they traverse a left hand, 16-degree, curved yard track at an approximate speed of 10 mph. The superelevation of the curve was -0.26 inch. The results are given in Table 4-7.

The data presented indicate a wide variation in performance between the various trucks tested. The empty cars, in general, experience higher values of wheel unloading index as compared with loaded cars. Although individual Type II trucks seem to attain improved load equalization levels, as a class, the group of vehicles tested cannot make such a claim.

4.4.3 Steady State Curve Negotiation

In the process of analyzing the field test data to quantify the performance indices, some unexpected behavioral trends were observed as they relate to the wheel/rail force measurements. A closer examination of these trends through various test runs as well as examination of the coupler forces data confirmed that the measured lateral forces tended to be asymmetric with respect to the sense of track curvature. In general, the lateral forces tended to be lower on right-hand curves as compared to left-hand curves. Although various hypotheses were formulated to explain the causes of this asymmetric trend, they remain to be verified.

These hypotheses include relating the measurements to well defined wheel/rail contact geometry considerations which may uncover patterns of asymmetry themselves, and influence of truck "set" or "memory" as it travels from one curve to another, among others. A comparison of the lateral forces for Type II trucks as they behaved over right-hand curves and left-hand curves as two distinct groups are given in Figures 4-2 through 4-5 . Figures 4-2 and 4-3 represent the results for the test configurations with empty cars and Figures 4-4 and 4-5 represent results for test configurations with loaded cars. On the basis of conservatism under the circumstances, the characterization of performance of the trucks was determined upon the higher level of forces, namely those obtained over the left-hand curves.

Lateral forces and L/V ratios at each of the four wheel/rail interfaces on the leading truck were examined for the three test speed conditions, namely below, at, and above equilibrium, or "balance" speeds. The algebraic means (average values) of the lateral forces were calculated for each curve over the length of track which could be considered "steady state" or "constant curvature" track. In plotting the characteristics, the absolute values of these algebraic averages were used.

In general, the test data indicate that in all the cases the trailing axle tends to carry the higher net lateral forces for the conditions representing the below balance speed test runs, and the leading axle tends to carry the higher net lateral forces for the conditions representing the balance speed and the above balance speed test runs. The trucks with radial alignment features seem to accomplish their goal of attaining flange free curving in the shallower ranges of track curvature (up to 3.7 degrees), but in the zone with sharper track curvature (5 degrees and above), guidance around the track depends on flange contact. No definitively detrimental degradation in performance was discerned in the case of the rigid trucks relative to the baseline performance of Type I trucks. Of course, any comparative evaluation has to keep in perspective the differences in test conditions, especially as they relate to wheel profiles (i.e., the Type I trucks were tested with AAR Standard 1:20 profile wheels, whereas the Type II trucks were tested with CN profile wheels).

4.4.4 Ride Quality

Only one of the two identified performance indices in this regime was quantified, namely the rms response over the wide band spectrum of 0-20 Hz. The index was analyzed for the vertical, lateral, and roll accelerations on the carbody. Accelerations are measured on both ends of the carbody and the quantitative characteristics presented in this section are the result of studying the vertical, lateral, and roll accelerations at both ends to choose the performance boundaries determined by the maximum levels.

Considering vertical vibrations, trucks with primary suspensions indicate comparable acceleration environments between the empty and loaded conditions, with the rms acceleration levels tending to increase with increasing speeds. On the other hand, secondary suspension trucks indicated a pronounced difference between the empty and loaded carbody responses, with the empty carbody responses being the consistently higher levels. The truck with primary + secondary suspension elements featured in the design was tested only in the loaded condition, and the response levels for this configuration were bordering the lower bounds of performance levels for the whole class of Type II trucks.

In general, for the empty cars equipped with Type II trucks, the rate of increase of the amplitude of vertical oscillations with increasing vehicle speed is small; the response curves level off in the speed range of 40 to 60 mph. Above 60 mph, the rate of increase in the response levels of some trucks indicate possible resonance phenomena at high speeds or, perhaps, a high degree of coupling between the vertical and lateral motions of the vehicle system exciting coupled modes. An examination of the performance of the class of rigid trucks relative to the radial trucks indicate that, for the empty car test conditions, the responses for the radial trucks vary in a range so wide that they form the upper and lower bounds of performance for the whole group of Type II trucks; in the loaded condition, the response of the radial trucks also determines the upper bounds of performance for the whole group of Type II trucks tested.

In lateral motion, the responses of the primary suspension trucks with empty cars indicate levels higher than that for the secondary suspension trucks. In the loaded condition, the differences in the levels of acceleration responses were not significant. Empty cars generally indicated higher levels of lateral acceleration response as compared to loaded cars for the Type II trucks, as a group.

Generalization of performance for groups of Type II trucks in the case of roll motion proved to be difficult. Rather, individual trucks showed the ability of specific design features to influence roll motion. The ability of a given truck to provide the levels of damping required to control the motion was especially demonstrated in the results of the roll response levels.

TABLE 4-4. SYMBOL IDENTIFICATION FOR TYPE II TRUCKS

△	Dresser DR-1	Primary Suspension Trucks	○ ●
□	Barber-Scheffel	Secondary Suspension Trucks	△ □ ▲ ■
○	Devine-Scales	Primary & Secondary Suspension Trucks	◇
●	Maxiride 100		
▲	National Swing Motion	Radial Trucks	△ □ ○
■	ACF Fabricated	Rigid Trucks	● ■ ▲
◇	Alusuisse	Unconventional Suspension Trucks	◇

TABLE 4-5. SUMMARY OF RESULTS FROM ANALYSIS OF TEST DATA IN THE LATERAL STABILITY PERFORMANCE REGIME LADING CONDITIONS: EMPTY CARS

Truck Classification		Phenomenological Behavior	Range of Critical Speed(mph)	Hunting Frequency(Hz)	Track Excitation Frequency(Hz)	RMS Lateral Acceleration (g's)	Peak Lateral Accel. (g's)	Percentage of Time of Occurrence of Observed Phenomenon
Radial Trucks	△	Moderate Amplitude Intermittent Hunting	60-65	2.7-2.8	2.3-2.5	0.05-0.10	0.34-0.43	60-65
	□	Moderate Amplitude Intermittent Hunting	45	2.9	1.70	0.10	0.35-0.43	65-70
		Sustained Hunting	55-60	2.90	2.0-2.30	0.2-0.24	0.65-0.68	100
	○	Moderate Amplitude Intermittent Hunting	60	2.70	2.30	0.12-0.14	0.55-0.60	60-65
		Sustained Hunting	79	3.0	3.0	0.12-0.16	0.87-0.88	100
Rigid Trucks	▲	Moderate Amplitude Intermittent Hunting	60-65	2.70-3.0	2.3-2.5	.07-0.09	0.46-0.48	60-65
	●	Moderate Amplitude Intermittent Hunting	65-70	2.70	2.5-2.6	0.105-0.12	0.63-0.65	60-65
		High Amplitude Intermittent Hunting	79	3.0	3.0	0.12-0.14	0.80-0.84	75-80
	■	Moderate Amplitude Intermittent Hunting	60-65	2.7	2.3-2.5	0.08-0.12	0.50-0.60	60-65

TABLE 4-6. SUMMARY OF RESULTS FROM ANALYSIS OF TEST DATA IN THE LATERAL STABILITY PERFORMANCE REGIME LADING CONDITIONS: LOADED CARS

Truck Classification		Phenomenological Behavior	Range of Critical Speed(mph)	Hunting Frequency(Hz)	Track Excitation Frequency(Hz)	RMS Lateral Acceleration (g's)	Peak Lateral Acceleration (g's)	Percentage of Time of Occurrence of Observed Phenomenon
Radial Trucks	△	Moderate Amplitude Intermittent Hunting	75-79	2.7-3.0	2.8-3.0	0.1-0.14	0.48-0.50	80-85
	□		79	3.0	3.0	0.07-0.09	0.41-0.43	15-20
	○		70-75	2.7	2.6-2.8	.07-0.11	0.60-0.70	40-50
Rigid Trucks	▲	Moderate Amplitude Intermittent Hunting	70-75	2.8	2.6-2.8	0.07-0.08	0.50-0.55	30-35
	●		70-79	2.7	2.6-3.0	0.1-0.14	0.6-0.75	55-60
	■		70	2.7	2.6	0.1-0.14	0.35-0.40	80
Unconventional	◊	None	N/A	N/A	N/A	N/A	N/A	N/A

TABLE 4-7. WHEEL UNLOADING INDEX (WUI) LEVELS

Truck		Empty Car			Loaded Car		
		Average	Standard Deviation	WUI_{95}	Average	Standard Deviation	WUI_{95}
Radial Trucks	△	0.564	0.135	0.783	0.190	0.053	0.281
	□	0.156	0.083	0.343	0.241	0.101	0.400
	○	0.454	0.218	0.744	0.252	0.136	0.512
Rigid Trucks	▲	0.314	0.126	0.553	0.277	0.058	0.368
	●	0.177	0.069	0.297	0.182	0.068	0.307

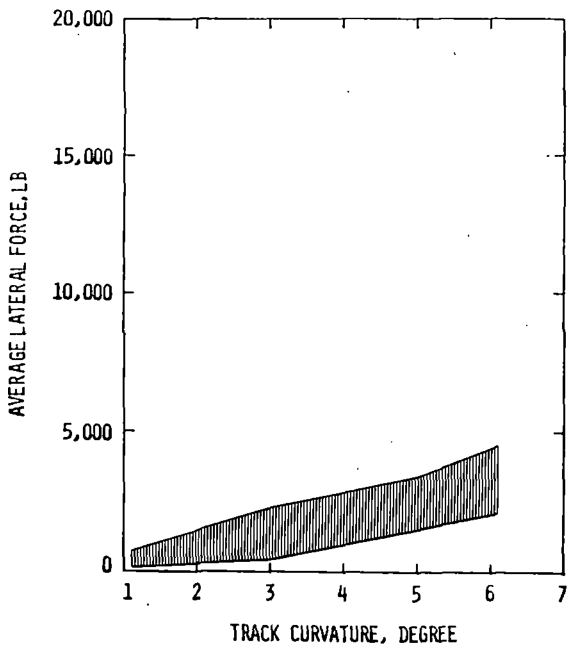

FIGURE 4-2. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS DEGREE OF CURVATURE/NEAR BALANCE SPEED/RIGHT HAND CURVES/TYPE II TRUCKS WITH EMPTY HOPPER CARS

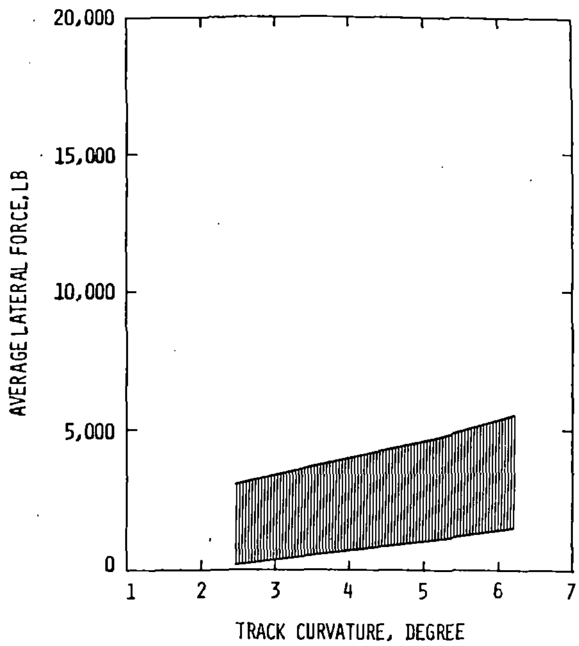

FIGURE 4-3. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS DEGREE OF CURVATURE/NEAR BALANCE SPEED/LEFT HAND CURVES/TYPE II TRUCKS WITH EMPTY HOPPER CARS

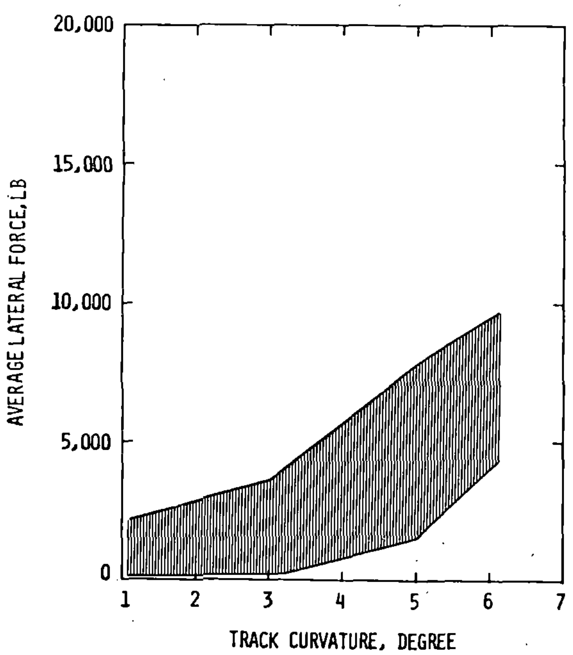

FIGURE 4-4. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS DEGREE OF CURVATURE/NEAR BALANCE SPEED/RIGHT HAND CURVES/TYPE II TRUCKS WITH LOADED HOPPER CARS

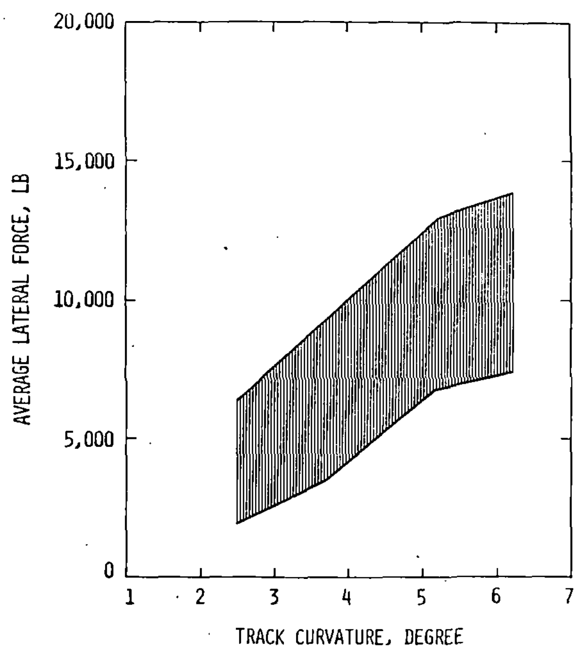

FIGURE 4-5. LATERAL FORCE ON LEADING OUTER WHEEL VERSUS DEGREE OF CURVATURE/NEAR BALANCE SPEED/LEFT HAND CURVES/TYPE II TRUCKS WITH LOADED HOPPER CARS

4.5 PERFORMANCE SPECIFICATIONS FOR TYPE II TRUCKS

One of the major tasks of the Truck Design Optimization Project, Phase II was to prepare performance specifications for Type II trucks; these performance specifications are presented in this section.

4.5.1 Scope

Although it was envisioned that the performance specifications developed on the basis of work performed during the project would be applicable to freight car trucks universally, it is considered essential to keep in perspective the finite frontiers of the effort undertaken during the project when applying the specifications to evaluate freight car trucks. Under conditions comparable to those covered by the project effort, it is indeed believed that the recommended specifications will be applicable. Caution is urged, however, in determining what constitutes a set of comparable conditions for evaluation.

Initially, it was conceived that the development of performance specifications would be on the basis of experimental and analytic investigations of a comprehensive set of freight car truck/carbody configurations that would represent commerically available vehicle systems on the market. Furthermore, the analytic investigations were to be conducted using available analytic tools subject to validation during the project. For various reasons, both technical and economic, compromises had to be made in the course of the project resulting in limitations of these investigations which are reflected in the results.

The recommended performance specifications are organized by performance regimes. In each performance regime, the parametric conditions associated with the recommended guidelines on quantitative performance are outlined. In using the performance specifications, it is advisable to relate them to these parametric conditions to ensure that application of the specifications are to conditions equivalent to, or at least comparable to, the conditions listed.

4.5.2 Development of Performance Specifications

The basis on which the performance specifications were developed was the performance test data acquired during the TDOP Phase II field tests. The field test data were analyzed methodically in each of the performance regimes to yield quantitative measures of performance represented by performance indices. The validity of specific details or trends within each regime was corroborated through physical reasoning, comparison with conventional wisdom in railroad literature, and, whenever possible, through the use of test data from other sources.

Extreme behavior of individual trucks, attributable to specific considerations relatable either to hardware conditions or to test conditions, were excluded from the recommended specifications. Such exclusions were made after careful and deliberate engineering evaluations of associated conditions and also after comparative studies with published results. Nevertheless, they do constitute engineering judgment and contain an element of subjective evaluation.

In the interest of coordinating the results with the industry, the recommended guideline performance specifications were discussed with railroad industry representatives at the TDOP consultants' meetings and at periodic "in progress reviews" to the industry. Final results were subjected to review by industry and government representatives and comments derived from this review process were accounted for in the final specifications presented in this section.

4.5.3 Recommended Quantitative Levels of Performance

This subsection presents the quantitative levels of performance that may be expected of the Type II freight car trucks in each of the performance regimes under the applicable conditions*.

Lateral Stability Performance Specifications

Parametric conditions associated with the guideline performance specifications in this regime are:

Carbodies	-	100-ton open hopper car (with 100-ton Type II trucks)
	-	70-ton open hopper car (with 70-ton Type II trucks)
Wheel Profiles	-	CN Profile (new) (with 100-ton Type II trucks)
	-	AAR Std. 1:20 Taper Profile (new) (with 70-ton Type II trucks)
Track	-	High Speed Tangent Track (Class 4, Mainline, BJR)
Speed	-	40 to 79 mph
Lading	-	Carbodies in empty and fully loaded conditions

Recommended performance specifications are given in Figures 4-6 through 4-9. The given bands of performance levels indicate values that may be reasonably expected to be obtained under the nominal operating conditions and associated reasonable variations. The upper bounds on the bands of quantitative performance levels constitute limiting values on the corresponding parameters.

*With respect to track characteristics, the reader is referred to Table 2-6, Figures 2-16 through 2-31, and References 4 and 5 for more details.

Trackability Performance Specification – Track Twist

Parametric conditions associated with the guideline performance specifications in the subregime of track twist are:

Carbodies	-	100-ton open hopper car
Wheel Profiles	-	CN Profile (new)
Track	-	Yard, BJR, 16° curve (-0.26 inch superelevation)
Speed	-	10 mph
Lading	-	Carbodies in empty and fully loaded conditions

Recommended performance specifications are given in Table 4-8.

TABLE 4-8. WUI_{95} LEVELS FOR TYPE II FREIGHT CAR TRUCKS

Performance Index	Empty Cars	Loaded Cars
WUI_{95}	0.30-0.55	0.28-0.37

Note: 95% level denotes that the given values shall not be exceeded in more than 5% of the time.

Trackability Performance Specifications – Curve Entry/Exit

Parametric conditions associated with the guideline performance specifications in the subregime of curve entry/exit are:

Carbodies	-	100-ton open hopper car
Wheel Profiles	-	CN Profile (new)
Track	-	Class 4, BJR, Curved Track, 1.1° - 6.2°
Speed	-	25-48 mph
Lading	-	Carbodies in empty and loaded conditions

Recommended performance specifications in the curve entry/exit subregime are given in Figures 4-10 through 4-17.

Steady Stage Curve Negotiation Performance Specifications

Parametric conditions associated with the guideline performance specifications in this regime are:

Carbodies	-	100-ton open hopper car
Wheel Profiles	-	CN Profile (new)
Track	-	Class 4, BJR, Curved Track, 1.1° - 6.2°
Speed	-	25-48 mph
Lading	-	Carbodies, in empty and fully loaded conditions

Recommended performance specifications are presented in Figures 4-18 through 4-37. Because of the radical differences between the radial and rigid trucks among the Type II freight car trucks in this performance regime, the limiting performance associated with rigid trucks is indicated separately in the illustrations. The broken lines, shown always at a level higher than the radial truck performance bands, represent the upper limits recommended for the rigid trucks. This exception, in separating the two subclasses of trucks among the Type II designs, is considered warranted since better performance on the part of rigid trucks on curved track is not attainable at this time and imposing such demands is not considered reasonable.

Ride Quality Performance Specifications

Parametric conditions associated with the guideline performance specifications in the regime of ride quality are:

Carbodies	-	100-ton open hopper car (with 100-ton Type II trucks)
	-	70-ton open hopper (with 70-ton Type II trucks)
Wheel Profiles	-	CN profile (new) (with 100-ton Type II trucks)
	-	AAR Sta. 1:20 Taper profile (new)
Track	-	High Speed Tangent Track (Class 4, Mainline, BJR)
Speed	-	40-79 mph
Lading	-	Carbodies in empty and fully loaded conditions

Recommended performance specifications are given in Figure 4-38 through 4-43. The bands of performance levels indicate the values of performance indices likely to be obtained under comparable nominal operating conditions with their associated reasonable levels of variations. The upper boundary of the performance bands represent the limiting levels of performance in each case.

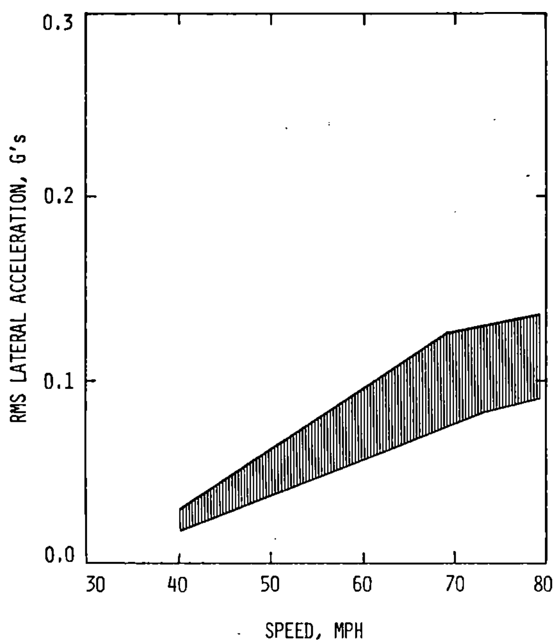

FIGURE 4-6. RMS LATERAL ACCELERATION LEVELS FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS

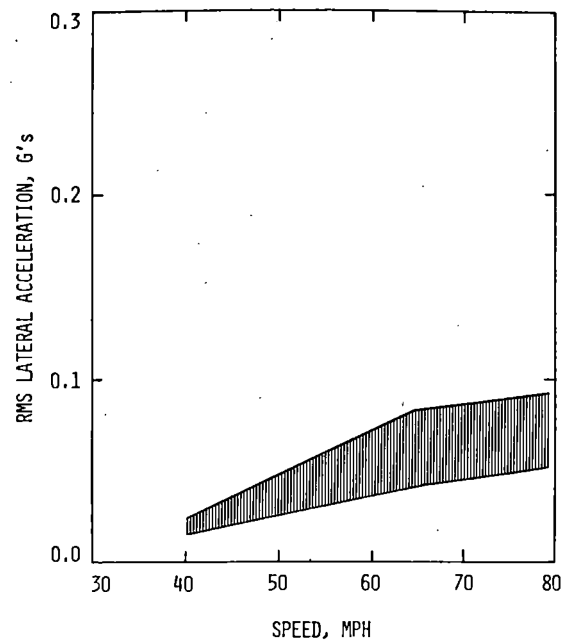

FIGURE 4-7. RMS LATERAL ACCELERATION LEVELS FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS

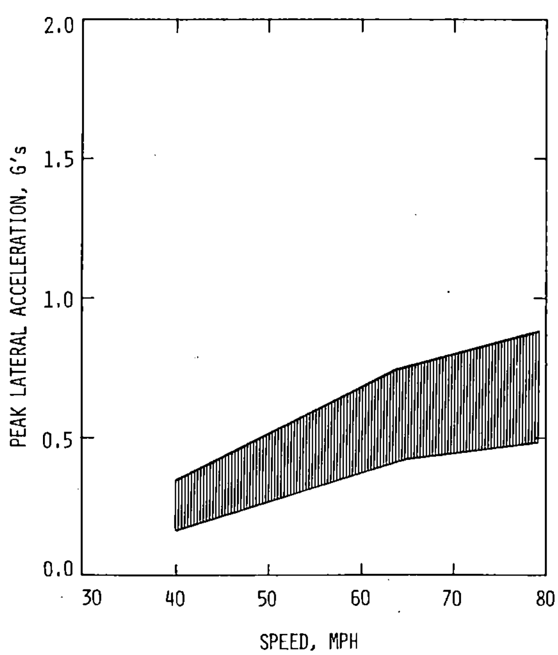

FIGURE 4-8. PEAK LATERAL ACCELERATION LEVELS FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS

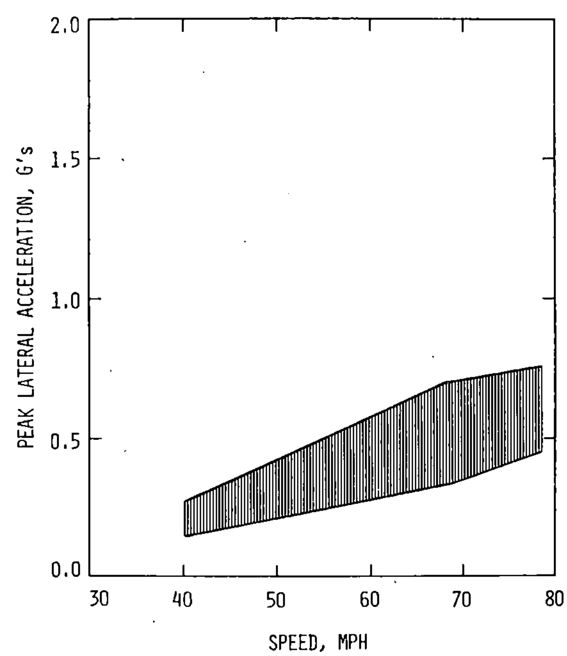

FIGURE 4-9. PEAK LATERAL ACCELERATION LEVELS FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS

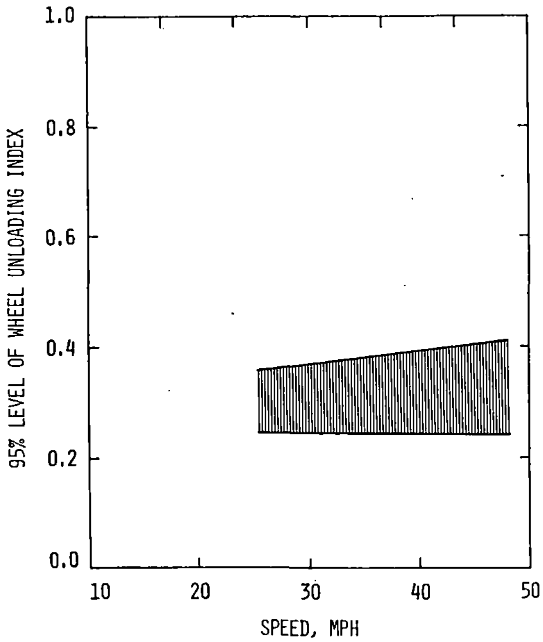

FIGURE 4-10. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH EMPTY OPEN HOPPER CARS - 2.5 DEGREES CURVED ENTRY/EXIT

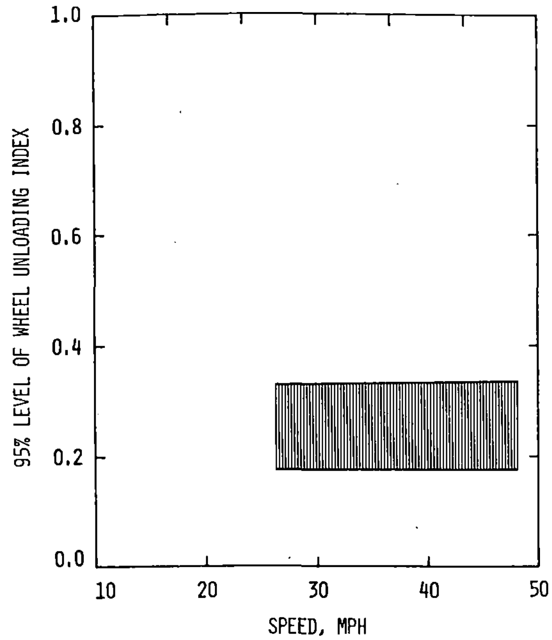

FIGURE 4-11. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH LOADED OPEN HOPPER CARS - 2.5 DEGREES CURVED ENTRY/EXIT

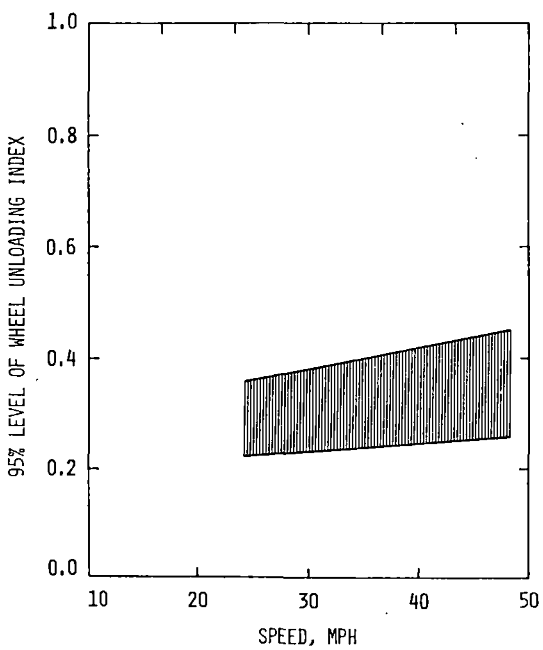

FIGURE 4-12. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH EMPTY OPEN HOPPER CARS - 3.7 DEGREES CURVES ENTRY/EXIT

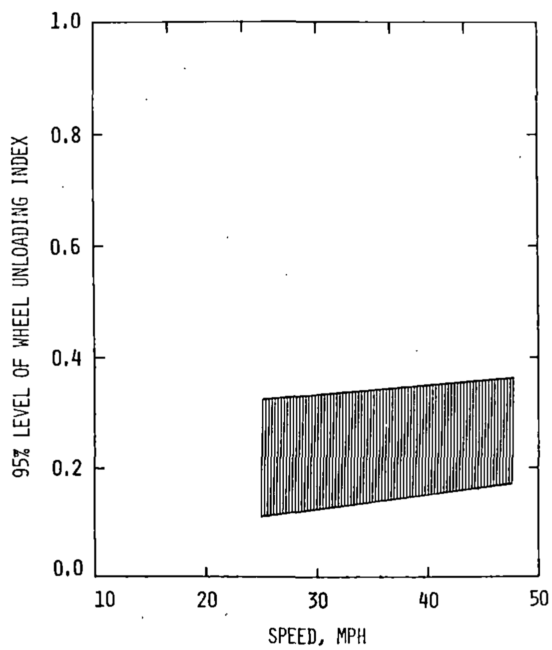

FIGURE 4-13. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH LOADED OPEN HOPPER CARS - 3.7 DEGREES CURVES ENTRY/EXIT

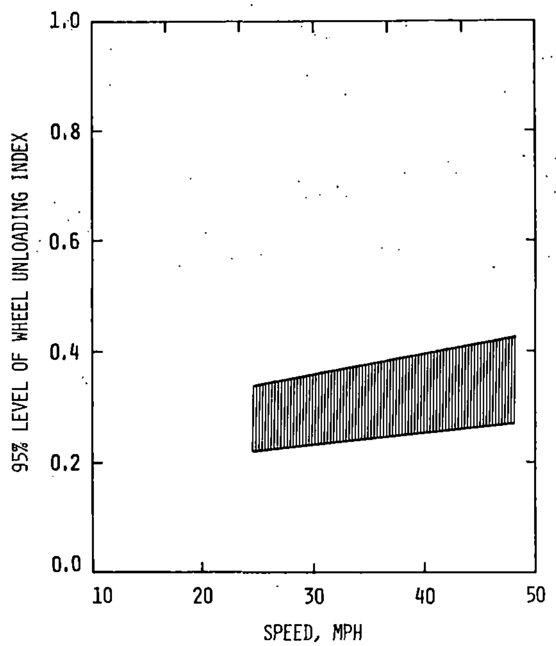

FIGURE 4-14. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH EMPTY OPEN HOPPER CARS - 5.2 DEGREES CURVED ENTRY/EXIT

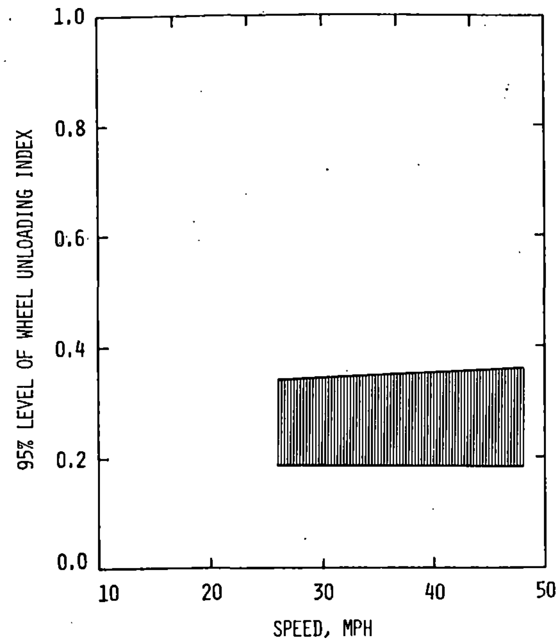

FIGURE 4-15. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH LOADED OPEN HOPPER CARS - 5.2 DEGREES CURVES ENTRY/EXIT

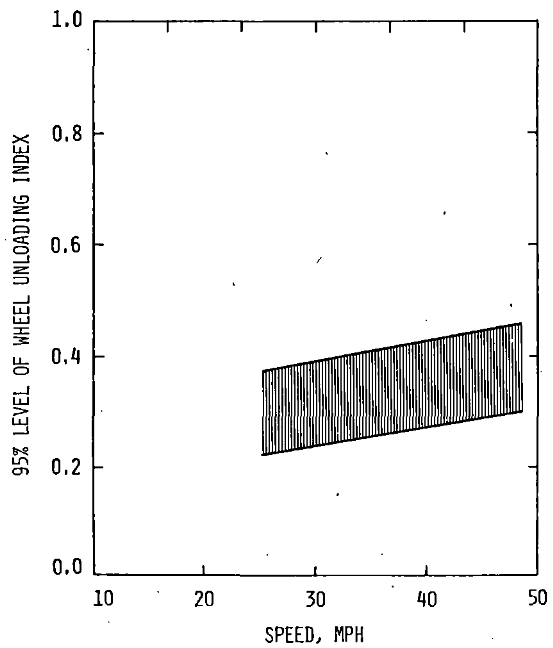

FIGURE 4-16. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH EMPTY OPEN HOPPER CARS - 6.2 DEGREES CURVES ENTRY/EXIT

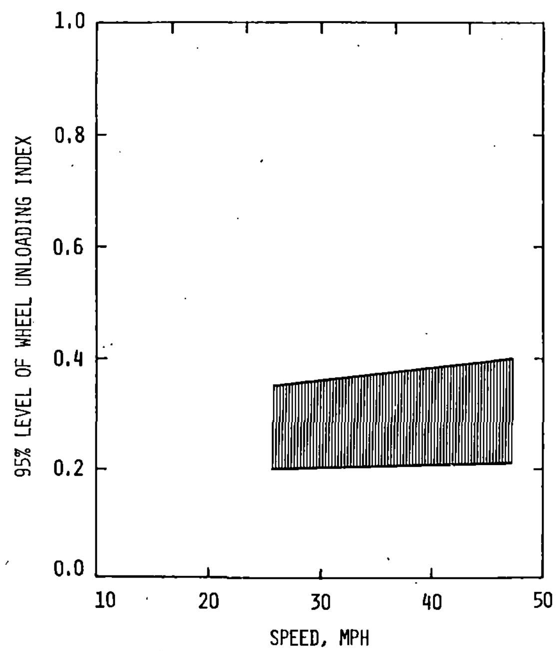

FIGURE 4-17. WUI_{95} LEVELS FOR TYPE II TRUCKS WITH LOADED OPEN HOPPER CARS - 6.2 DEGREES CURVES ENTRY/EXIT

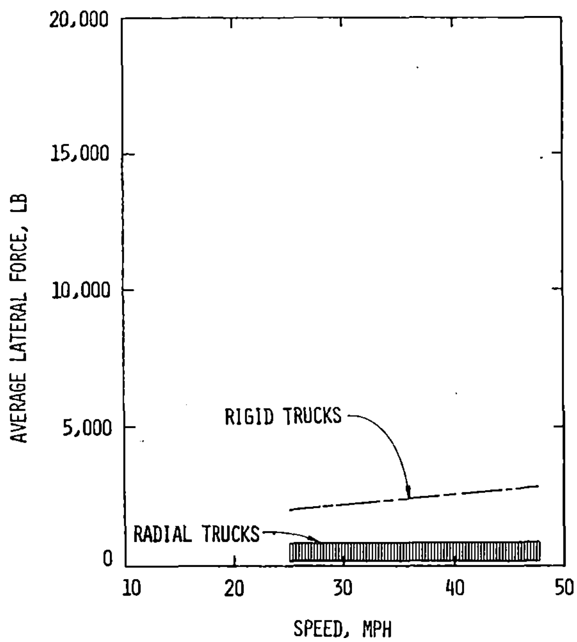

FIGURE 4-18. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 2.5 DEGREE CURVES

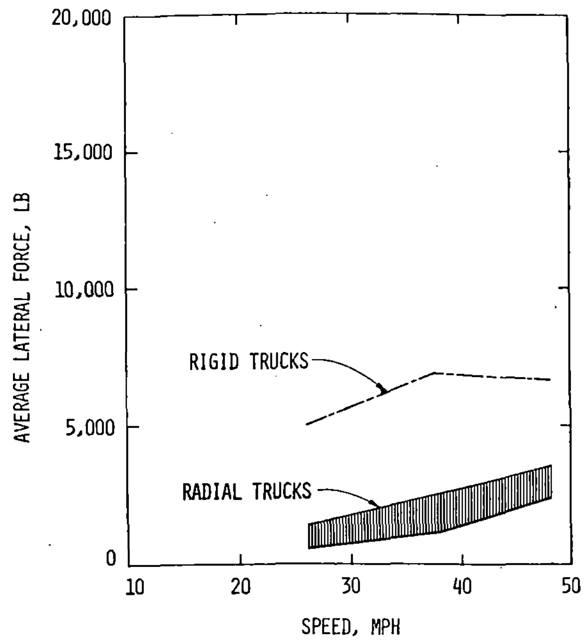

FIGURE 4-19. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS 2.5 DEGREE CURVES

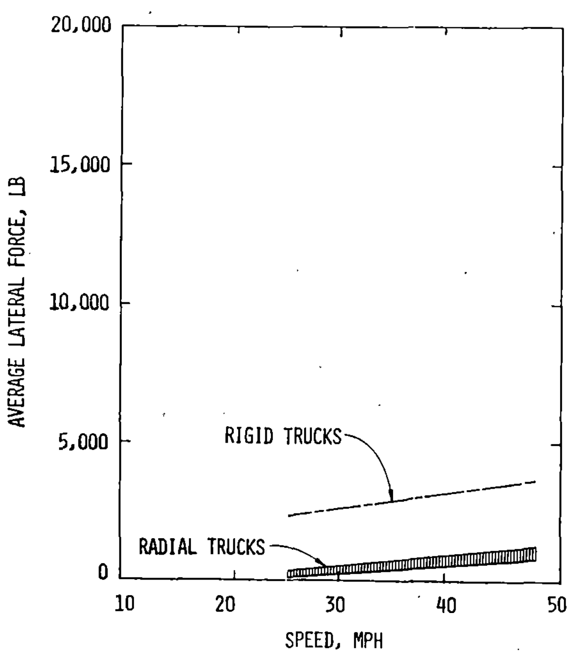

FIGURE 4-20. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 3.7 DEGREE CURVES

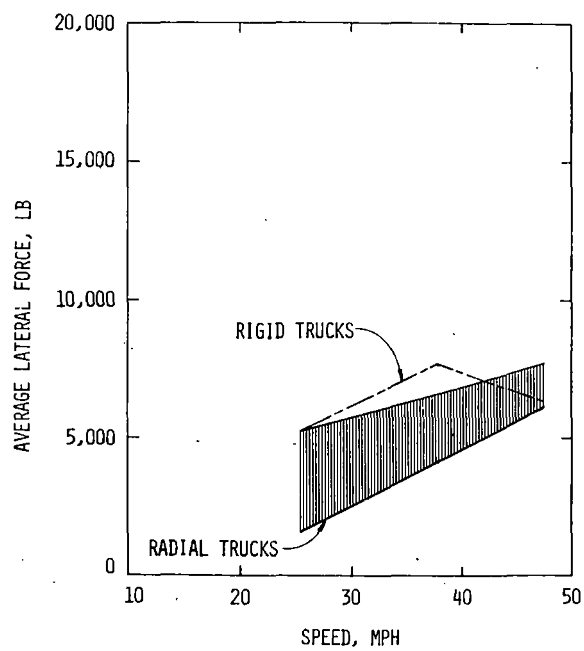

FIGURE 4-21. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS 3.7 DEGREE CURVES

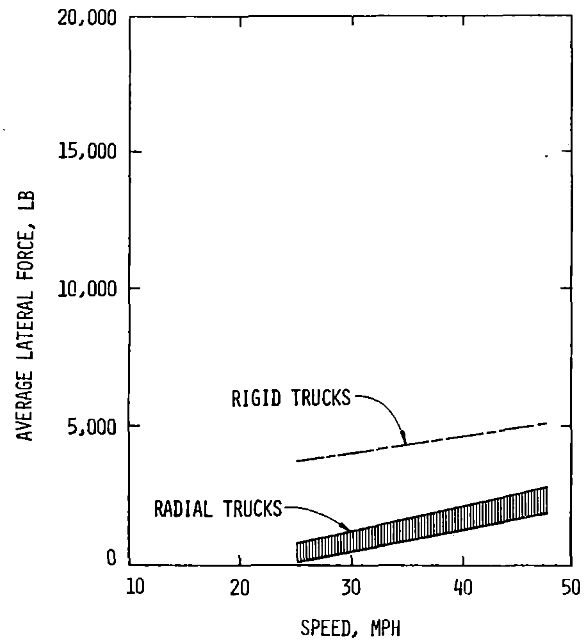

FIGURE 4-22. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 5.2 DEGREE CURVES

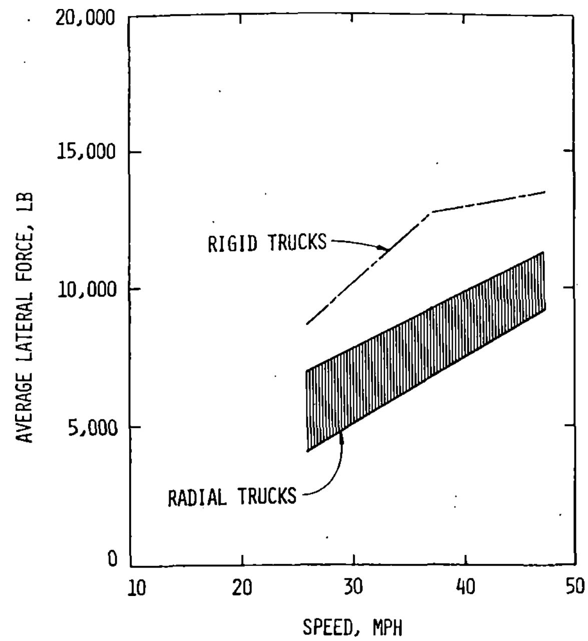

FIGURE 4-23. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS - 5.2 DEGREE CURVES

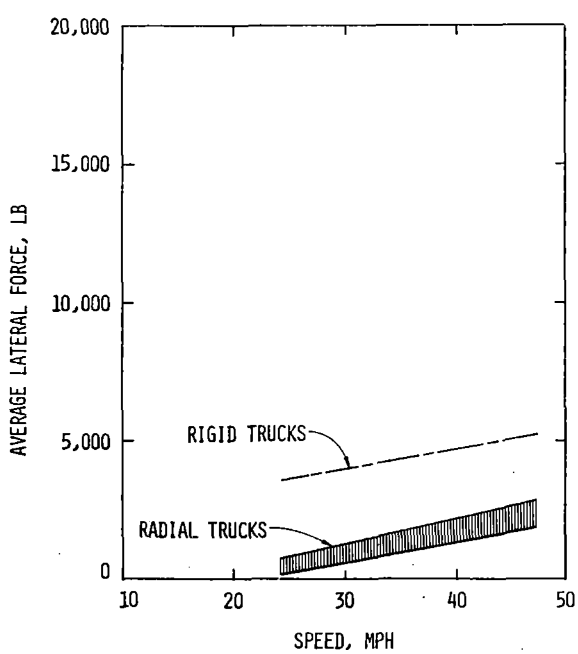

FIGURE 4-24. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 6.2 DEGREE CURVES

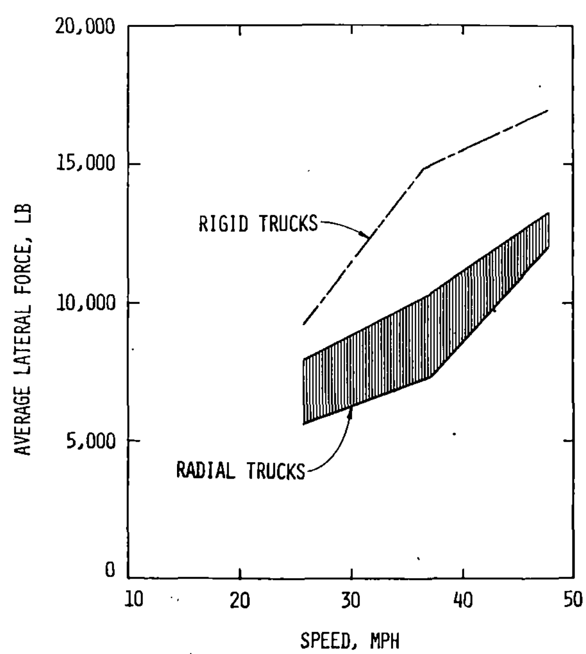

FIGURE 4-25. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS - 6.2 DEGREE CURVES

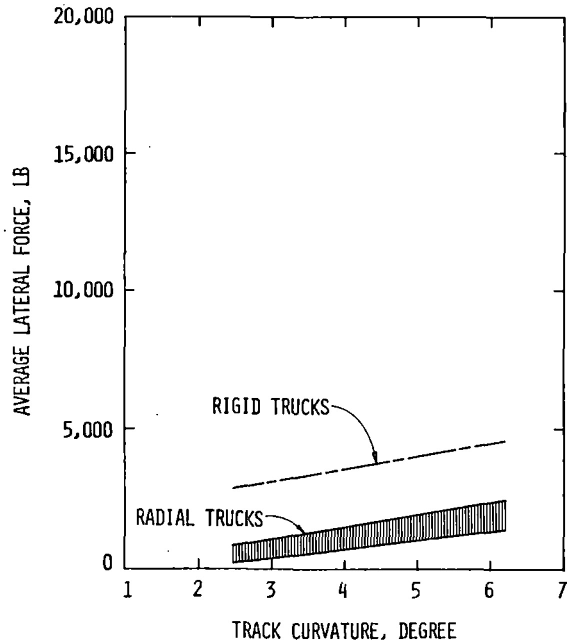

FIGURE 4-26. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCK WITH EMPTY OPEN HOPPER CARS AT BALANCE SPEED (± 2.5 MPH)

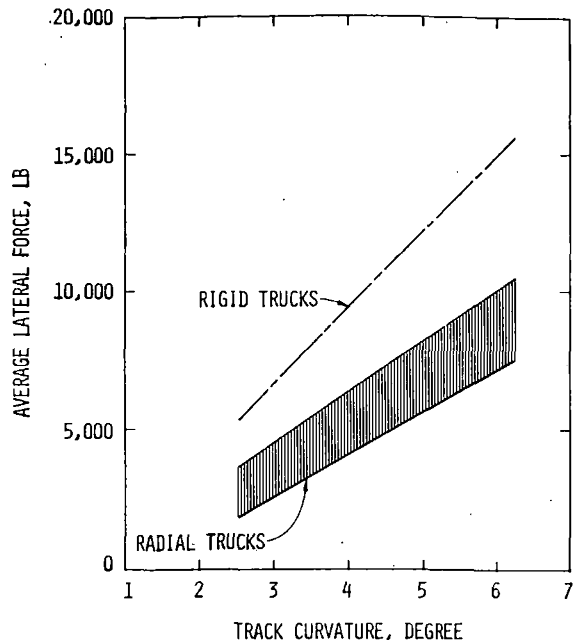

FIGURE 4-27. LATERAL FORCE ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCK WITH LOADED OPEN HOPPER CARS AT BALANCE SPEED (± 2.5 MPH)

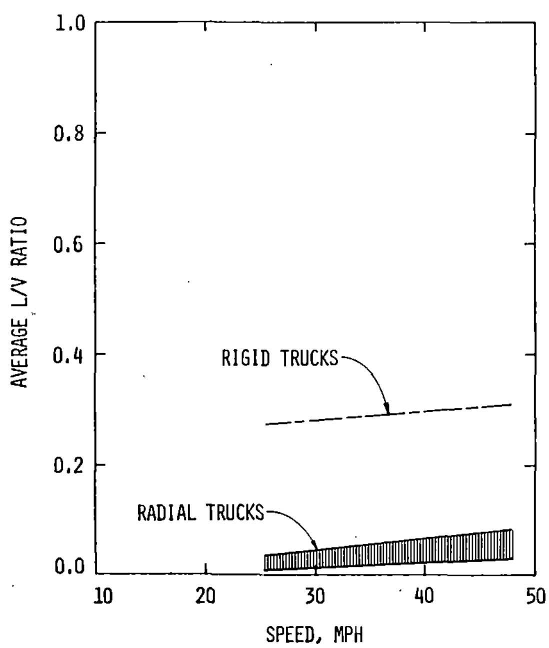

FIGURE 4-28. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 2.5 DEGREE CURVES

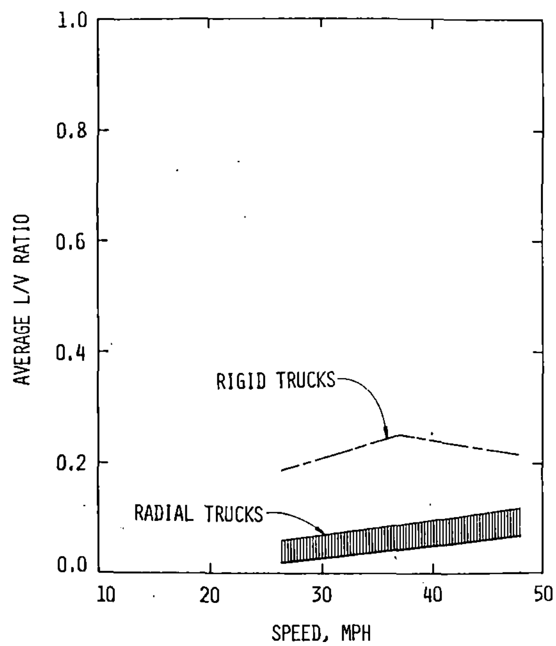

FIGURE 4-29. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS - 2.5 DEGREE CURVES

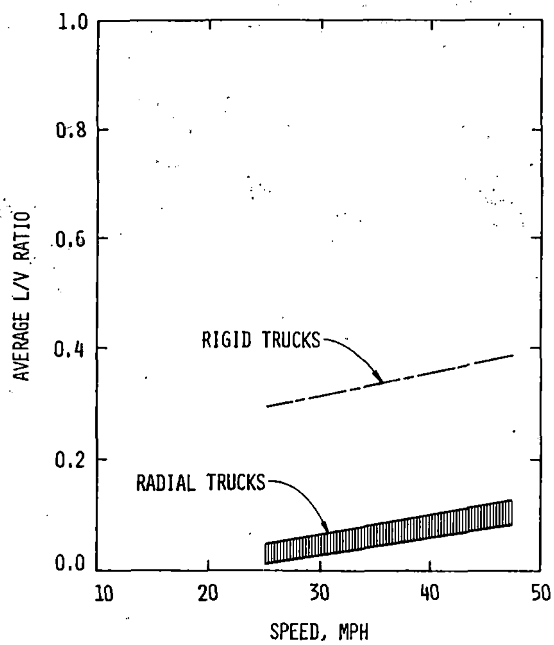

FIGURE 4-30. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 3.7 DEGREE CURVES

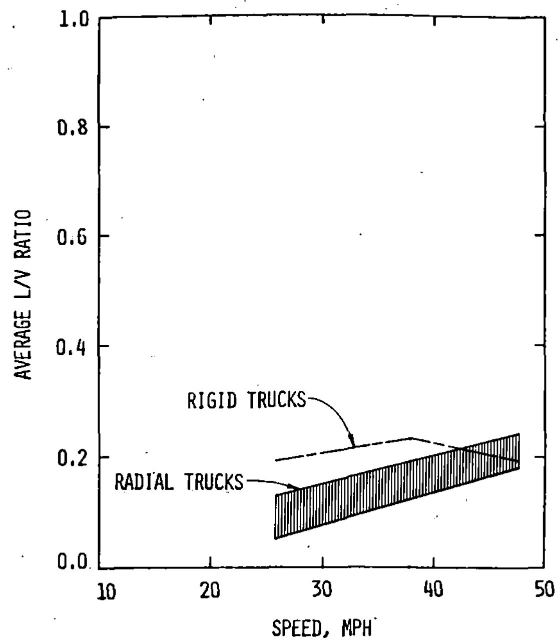

FIGURE 4-31. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS - 3.7 DEGREE CURVES

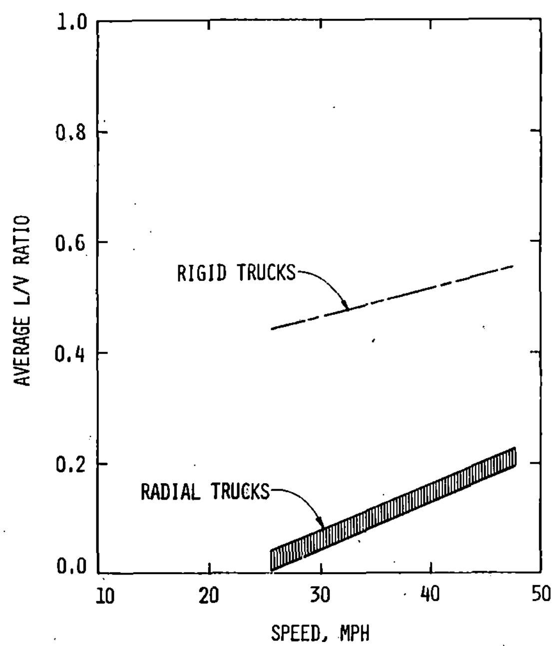

FIGURE 4-32. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS - 5.2 DEGREE CURVES

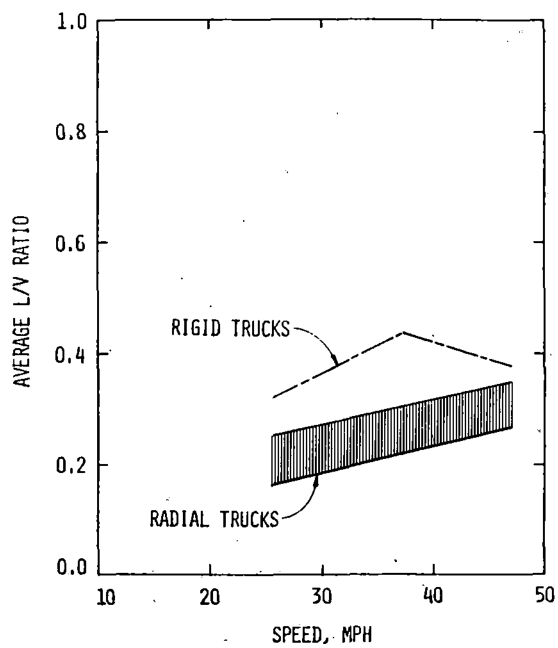

FIGURE 4-33. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS - 5.2 DEGREE CURVES

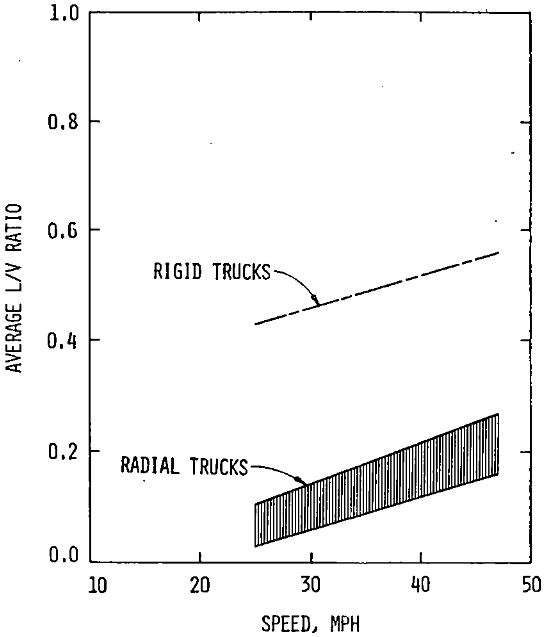

FIGURE 4-34. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CAR - 6.2 DEGREE CURVES

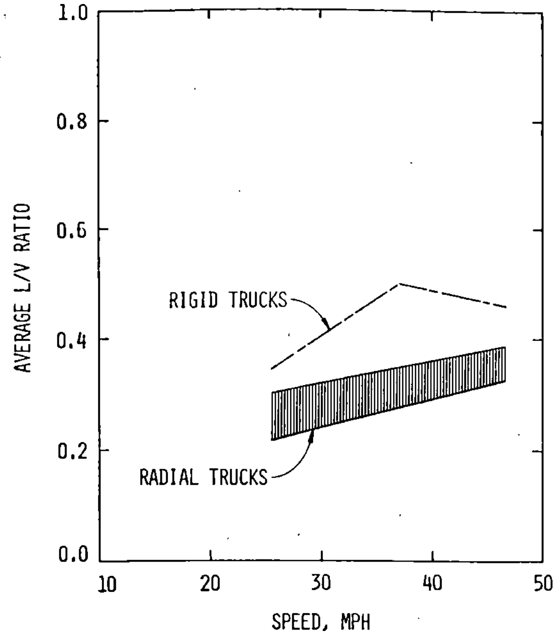

FIGURE 4-35. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CAR - 6.2 DEGREE CURVES

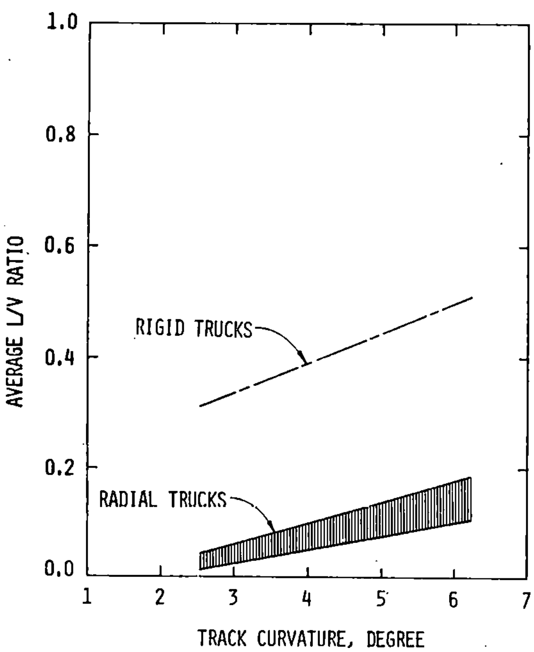

FIGURE 4-36. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS AT BALANCE SPEED (± 2.5 MPH)

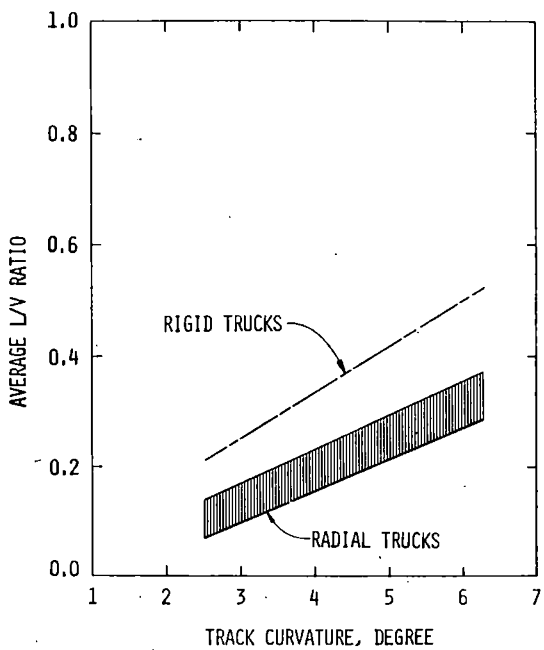

FIGURE 4-37. L/V RATIO ON LEADING OUTER WHEEL FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS AT BALANCE SPEED (± 2.5 MPH)

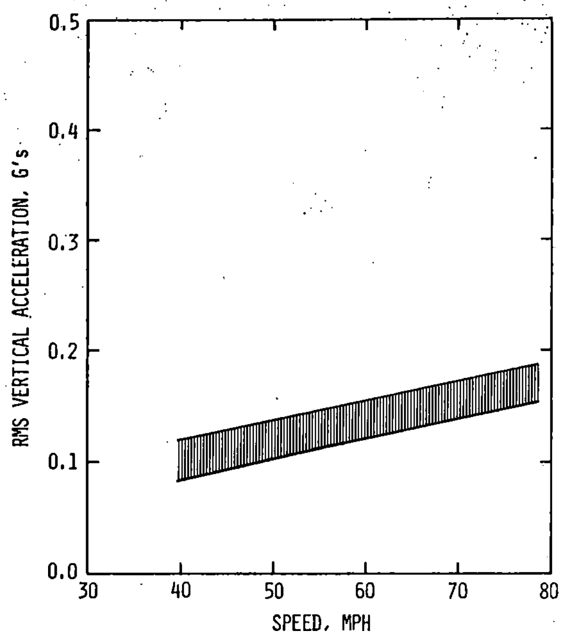

FIGURE 4-38. RMS VERTICAL ACCELERATION LEVELS (0-20 HZ) FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS

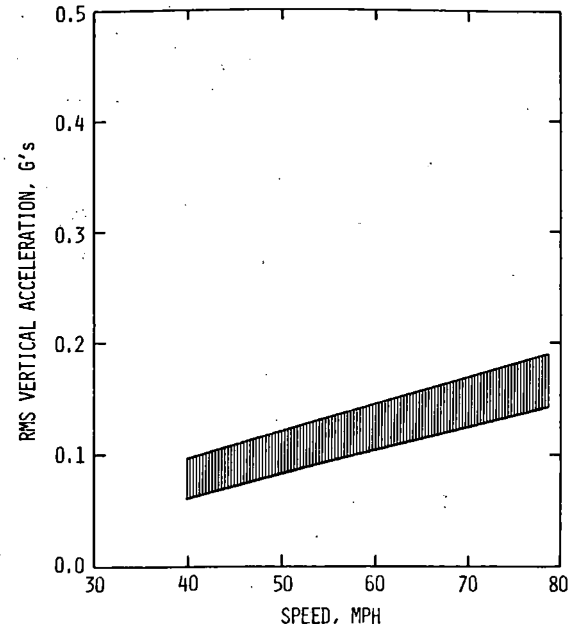

FIGURE 4-39. RMS VERTICAL ACCELERATION LEVELS (0-20 HZ) FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS

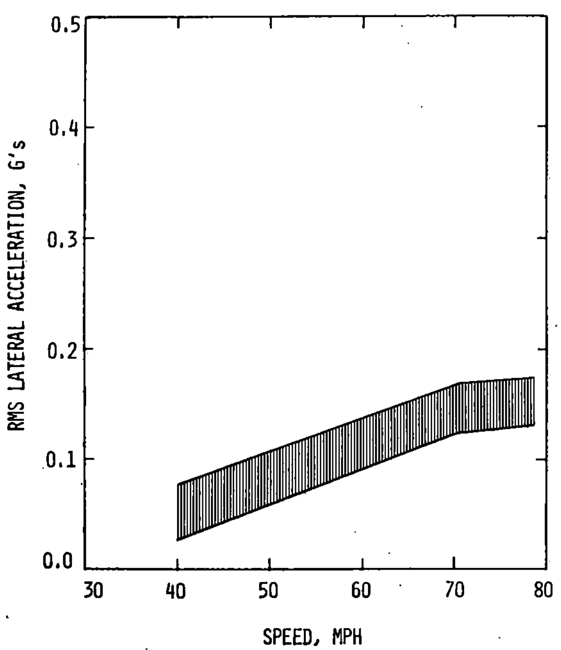

FIGURE 4-40. RMS LATERAL ACCELERATION LEVELS (0-20 HZ) FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS

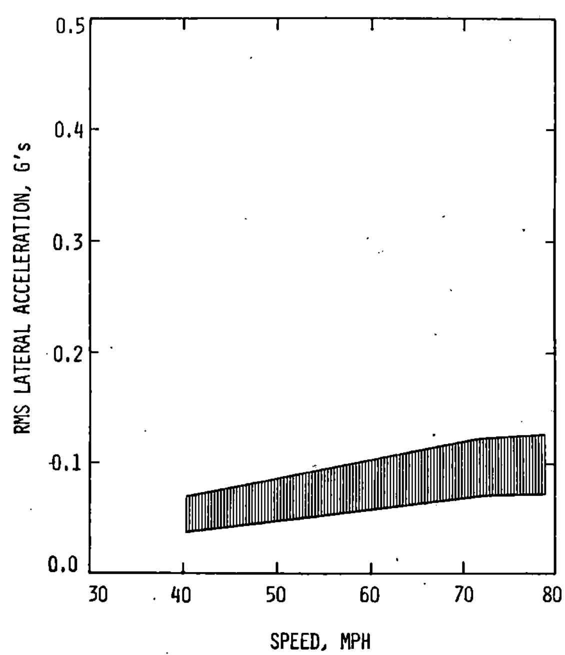

FIGURE 4-41. RMS LATERAL ACCELERATION LEVELS (0-20 HZ) FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS

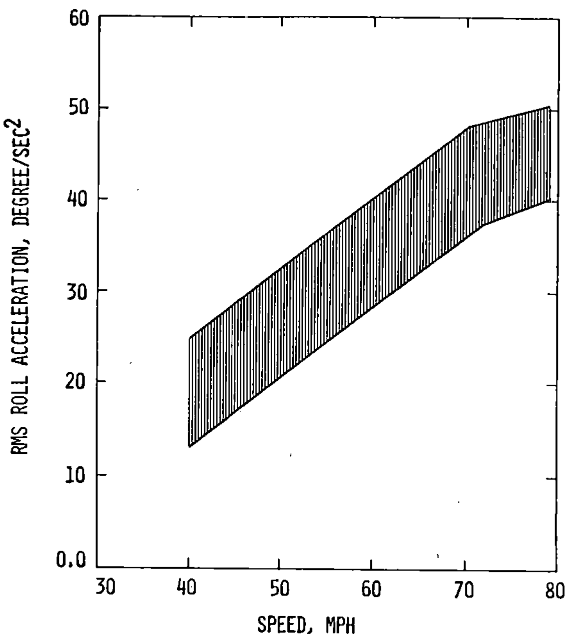

FIGURE 4-42. RMS ROLL ACCELERATION LEVELS (0-20 HZ) FOR TYPE II FREIGHT CAR TRUCKS WITH EMPTY OPEN HOPPER CARS

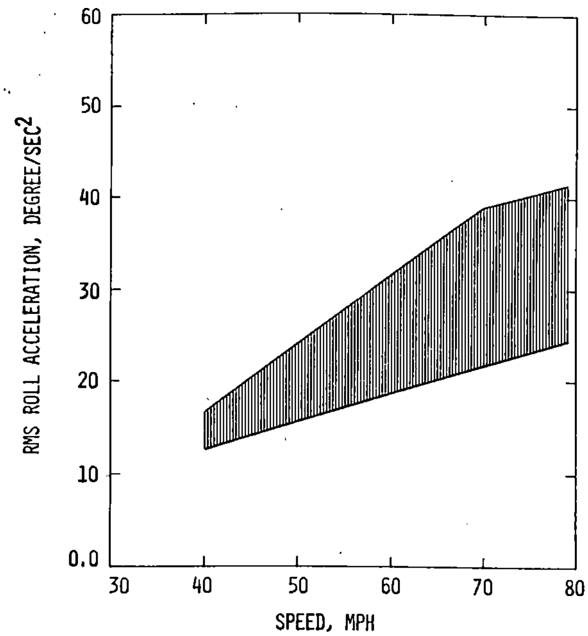

FIGURE 4-43. RMS ROLL ACCELERATION LEVELS (0-20 HZ) FOR TYPE II FREIGHT CAR TRUCKS WITH LOADED OPEN HOPPER CARS

4.6 GUIDELINES FOR TEST SPECIFICATIONS

Specification of performance for freight car trucks developed during TDOP Phase II stipulate quantitative levels of performance characteristics expected of them under a given set of operating conditions. The overall performance of freight car trucks has been compartmentalized into four distinct and non-overlapping performance regimes (lateral stability, trackability, steady state curve negotiation, and ride quality); taken together, these four performance regimes are inclusive of the overall truck performance. In each of the performance regimes, ranges of economics-related engineering performance indices correlated to corresponding sets of operating conditions comprise the specification of performance. A detailed description of the performance regimes and associated performance indices is given in Section 1 of this report.

The guideline test specifications provided in this section set forth the procedures for, and conduct of, field tests (over-the-road tests) as well as laboratory tests for generating the performance test data which will be necessary for the quantification of the performance indices. These indices can then be used to perform a quantitative evaluation of performance of Type I and Type II freight car trucks and a check on their compliance with the performance specification.

A road test represents the rail environment in all its complexity. This tends to lend credibility to the results which may be enhanced by direct observation of the test specimen. However, care should be taken that the road tests planned will be properly conducted, adequately instrumented, and rationally interpreted. The test track is defined in this report so that it can be duplicated. Laboratory tests, on the other hand, are accomplished under a controlled environment to conduct research on the many dynamic factors affecting vehicle performance and safety.

With this in mind, the Rail Dynamics Laboratory (RDL) facility at the Transportation Test Center in Pueblo was designed and constructed. The goal of the RDL is to provide a facility to perform dynamic tests on several configurations of locomotives, cars and trucks under controlled conditions. Such a facility permits the evaluation of various hardware designs in a safe, controlled and reproducible scientific laboratory environment, allowing the performance of a variety of tests. While simulated tests under controlled conditions in the laboratory may not serve as a substitute for field tests, they can be effective and complementary tools used to augment the results from a field test program in a cost effective manner. Thus, the test specifications discussed in this section include both field and laboratory test conditions.

4.6.1 Field Test Specifications

The specifications reported here cover field testing for freight car trucks in the four performance regimes. Under each regime, the data requirements, instrumentation, operating conditions, test procedure, and data reduction and analysis will be specified.

4.6.1.1 Lateral Stability

Data Requirements and Instrumentation. Acceleration data shall be acquired at these locations: lateral accelerations at the B-end, A-end, and the center of the carbody at the sill level; at the B-end and the A-end on the carbody at the roof level; at each of the axles on both trucks under the carbody. The acceleration data acquired through accelerometers at these locations shall meet the following minimum criteria:

Frequency response: 20 Hz
Range of measurements: $\pm$ 10 g's
Accuracy of measurements: 1%

Test Track. Lateral stability data shall be acquired on test runs over tangent track which permits the acquisition of data over a speed range from 30 mph to 79 mph or the operating speed limit, whichever is higher. (Note: the 79 mph limit is chosen on the basis of current legal speed limits on mainline tracks). The tangent track may be bolted, jointed, or continuous welded track, but the jointed track is recommended for testing since it represents a rough roadbed that may excite (initiate) the truck hunting movements.

Test Procedures. Tests shall be conducted on a selected segment of track of sufficient length (recommended length: a minimum of five miles, and more if possible) to permit the acceleration of the test consist from 30 to 79 mph and also to provide dwell times at incremental speeds of 5 mph throughout this range. The dwell times at each incremental speeds, namely 30, 35, ...70, 75, and 79 mph, shall be a minimum of 60 seconds to provide acquisition of quality data at these selected constant speed intervals. If the length of test track does not permit this sequence of data acquisition in one pass, the test run shall be segmented into two, or more passes covering, say, for example 30 to 60 mph, 60 to 70 mph, and 70 to 79 mph as overlapping passes.

Data Reduction and Analysis. The output of the lateral accelerometers shall be examined using time history plots to identify the hunting phenomenon. The rms and the peak values of the collected data shall be determined and plotted as functions of vehicle speed.

4.6.1.2 Trackability

a. Harmonic Roll and Bounce/Pitch

All requirements relating to data acquisition, instrumentation, test conditions, and test procedures shall be in accordance with "Specifications for Testing Special Devices to Control Stability of Freight Cars," Association of American Railroads Standard, adopted, 1968, and revised 1976 (Reference 6).

b. Track Twist

Data Requirements and Instrumentation. Simultaneous measurement of vertical forces at all wheel/rail interfaces on a given truck shall be accomplished through force measurement transducers. Although a combination of strain gaged axles and bearing adapters have been used to arrive at the results presented in the TDOP Phase II reports (References 1 & 2), other acceptable methods of wheel/rail force measurements may be used provided that such methods have been validated to assure that they yield data within acceptable limits of accuracy, namely 5%. Properly calibrated instrumented wheels may be used as force transducers to provide acceptable force measurement data. If only one of the two trucks under a car is instrumented, it shall be the forward truck; preferably, both trucks shall be instrumented to obtain vertical force measurements at all wheel/rail interfaces under the test car. Typical wheel/rail measurement instrumentation is shown in Section 2 of this report (Figure 2-2).

Test Track. Ideally, the tests should be performed on track with known or available information on track twist. Examples may be simulated tracks or perturbed tracks with known measures of track twist introduced into them. Otherwise, tests shall be conducted on existing Class 1 tracks (yards) at speeds of 10 mph or less.

Test Procedures. Tests shall be conducted over the selected test track sections at an operating of 10 mph or less. Data shall be continuously recorded during the test runs.

Data Reduction and Analysis. The data for the vertical forces at the wheels of the truck shall be examined, and the Wheel Unloading Index, defined in Section 1, shall be calculated.

c. Curve Entry and Exit

Test runs and conditions governing the tests for acquisition of data to be used in this performance subregime are discussed below.

4.6.1.3 Curve Negotiation

Data Requirements and Instrumentation. Data requirements under this section, in addition to the steady state curve negotiation performance regime, also covers the curve entry/exit subregime of the trackability performance regime.

Continuous measurement of lateral and vertical forces at the wheel/rail interfaces (preferably, all locations under the test car; at a minimum all locations at the forward truck under the test car) shall be performed. The force measurements may be accomplished by means of instrumented wheelsets where the axles are strain gaged to record axle-bending moments and the bearing adapters are strain gaged to measure vertical forces, with the forces calculated through the axle bending technique (Reference 8); alternately, instrumented wheel plates may be used as force transducers to measure wheel/rail lateral and vertical forces.

Measurement of the wheel/rail angle of attack shall be performed. The angle of attack can be measured using a wayside system or a vehicle-borne (onboard) system. The onboard system is recommended since it provides a continuous measurement of the angle of attack of the wheel with the rail during the negotiation of the curve. The onboard system can be electrical (non-contacting proximity sensors), or mechanical (spring-mass system). However, care should be taken to provide sufficient dynamic range for the system used in measuring the angle of attack. It is recommended that the angle of attack be measured on both sides of the wheelsets of the leading truck.

Measurements of the truck swivel and track tram are recommended since they will help in reducing and analyzing the data.

Test Track. Curve negotiation test runs shall be conducted on mainline (Class 4 or better) test tracks consisting of curves ranging, at a minimum, from 2 to 6 degrees. A larger range of track curvature shall be desirable. The test track shall be selected so as to allow representation of at least one curve each in the classes of approximately 2, 3, 4, 5, and 6 degrees, both right-hand and left-hand curves; the test curves shall be preceeded by a length of tangent track not less than that which permits the test train to accelerate or decelerate and enter the test curves at specified test speeds. It is to be recognized that the test speed will vary from curve to curve.

Test Procedures. A minimum of three test runs shall be conducted in each direction on the test track, representing (a) a test speed at least 5 mph below, but not more than 10 mph below, the equilibrium speed for each curved segment of track represented in the test zone; (b) a test speed equivalent of track represented in the test zone; and (c) a test speed at least 5 mph above, but not more than 10 mph above, the equilibrium speed for each curved segment of track represented in the test zone, with a tolerance of $\pm$ 2 mph on the test speed being permissible. No brake applications are to be made during the test runs. Data generated during the test runs shall be acquired and recorded continuously.

Data Reduction and Analysis. The time history of the data channels shall be examined. Lateral and vertical forces and L/V ratios, as well as angle of attack, shall be calculated, and then plotted as functions of speed (or superelevation deficiency) and the degree of curvature.

4.6.1.4 Ride Quality

Data Requirements and Instrumentation. Lateral and vertical acceleration data shall be acquired at least at the B-end, A-end, and carbody center at sill level. Lateral acceleration data shall be acquired at the B-end and A-end on the carbody at the roof level.

The acceleration data acquired through accelerometers shall meet the following minimum criteria:

Frequency response: 20 Hz
Range of measurements: $\pm$ 10 g's
Accuracy of measurements: 1%

Test Track. Ride quality data shall be acquired on test runs over Class 4, mainline tangent track (jointed welded rail) which permits the acquisition of data over a speed range from 30 to 79 mph. The track geometry data shall be acquired in order to correlate response measurements made on test vehicles with a known track input and to calculate the transmissibility. The track geometry data of interest in the study of ride quality are profile, alignment, gauge, and cross level.

Test Procedures. The test speeds shall range from 30 to 79 mph with 5 mph increment. Sample time of each speed shall be 60 seconds. The data shall be recorded continuously at each speed dwell.

Data Reduction and Analysis. Detailed statistical analysis shall be performed on the test data. The analysis shall include calculations of the frequency content of the data, the rms-values of the output signals and the track input, and the percent of the time a signal amplitude is above a given level as a function of that level. The transmissibility between the output signal and the track excitation will be calculated. This transmissibility may be characterized by a frequency dependent function of amplitude ratios called transfer function, or a sequence of root mean square (rms) ratios of ouput-to-input over selected frequency bands (for example 0-4 Hz, 4-10 Hz, and 10-20 Hz).

4.6.2 Laboratory Test Specifications

The Rail Dynamics Laboratory has been designed to simulate rail vehicle dynamics under laboratory conditions to discover means of reducing the costs and damages currently experienced by railroads. In addition, new vehicles can be tested to assure safety, improved ride quality, stability, and life expectancy prior to actual use.

The Rail Dynamics Laboratory building houses two test rigs (the Roll Dynamics Unit and the Vibration Test Unit) and supporting equipment. The test machines are equipped to accommodate nearly all existing and planned rail vehicles. They have special design features providing for cars varying in weight, length, wheel gauge, and axle and truck spacing. A brief description of the Roll Dynamics Unit and the Vibration Test Unit is given below.

Roll Dynamics Unit: The Roll Dynamics Unit (RDU) is used to study wheel/rail dynamic interaction. The vehicle forward motion is simulated on rollers which are controlled by drive trains consisting of a motor and one or more flywheels.

The RDU provides the capability for driving, or absorbing power from, the wheelsets of four-axle vehicle or locomotive truck. Six- or eight-axle locomotives and cars can be tested with use of auxiliary support stands. Through rotation of the rollers, the RDU simulates tangent track at various vehicle speeds, and permits investigation of dynamic phenomena characteristics of "perfect" tangent track such as truck hunting. A maximum vehicle weight of 400,000 pounds can be accommodated and speeds over 144 mph can be simulated in a steady-state tangent track environment.

Vibration Test Unit: The Vibration Test Unit (VTU) is designed to study suspension characteristics of rail vehicles, component and vehicle natural frequencies, ride comfort, lading responses, component fatigue, as well as rock and roll phenomenon. The VTU provides the capability for subjecting a 320,000-pound rail vehicle equipped with two two-axle trucks, or one truck of a vehicle having three or four axles, to controlled vertical and lateral vibration inputs on the wheels, creating the dynamic effects of irregular track on a vehicle. The VTU has a frequency range of 0.2 to 30 Hz and between 0.2 and 2 Hz motions with displacements up to 2 inches. Computer-generated rail profiles or recordings of actual rail profiles drive hydraulic actuators which can be positioned to accept a variety of truck spacings or axle arrangements.

4.6.2.1 Lateral Stability

Data Requirements and Instrumentation. The Roll Dynamics Unit shall be used to produce the special dynamics caused by wheel/rail interaction by simulating a vehicle's forward motion on rollers. Acceleration data shall be acquired at the following locations:

- Lateral accelerations at the B-end/sill level, A-end/sill level and the center of the carbody at the sill level,
- At the B-end and A-end on the carbody at the roof level,
- At each of the axles on both trucks under the carbody.

The truck bolster yaw angle of the leading and trailing trucks shall be measured using rate gyros. The truck-mounted accelerometers shall have a range of $\pm$ 10 g's. Expected maximum ranges for purposes of scaling and calibrating are $\pm$ 10 g's for the trucks and $\pm$ 5 g's for the body. Actual measurements should be less than these.

The wheel and roller profiles shall be measured using profilometers.

The lateral accelerometers on the trucks and carbody should be recorded as well as their double integrated signals. All data channels signals should be recorded on the analog tape and digitized and recorded on magnetic tape. Analog signals will be filtered by 20 Hz low-pass filter before being digitized.

Test Procedure. A continuous speed sweep shall be conducted from 30 mph to the onset of severe hunting (if it occurs without excitation). Subsequent test runs shall consist of incremental speed sweeps (5 mph increments) up to the onset of truck hunting, followed by a decreasing sweep to zero speed. Due to the RDU simulation of "perfect tangent track," it may be necessary to excite the trucks in order to initiate hunting. If this is necessary, the trucks shall be perturbed laterally during tests for each incremental speed increase. Ten speeds having 1 mph increments shall be selected over the speed range from slightly below the threshold of hunting speed to hard flange contact truck hunting. The threshold of hunting speed is the lowest speed at which sustained oscillation of hunting occurs. A rotary vibrator may be used on the carbody for purposes of overcoming static friction of truck components (Reference 9).

Data Reduction and Analysis. The outputs of the lateral accelerometers and angle rate gyros shall be examined using time history plots to identify the hunting phenomenon. The rms and the peak values of the collected data shall be determined and plotted as a function of vehicle speed. The damping ratios of the hunting mode will be calculated using the log decrement method. It will be plotted versus speed. The effective conicity will be calculated from wheel/roller profile data.

4.6.2.2 Trackability

a. Harmonic Roll

Data Requirements. The Vibration Test Unit (VTU) will be used to provide a suitable environment for the evaluation of vehicle harmonic roll response. The instrumentation transducers shall be comprised of angle rate gyros, displacement transducers and pressure transducers (Reference 10).

The data for roll angles and roll angle rates shall be acquired at the B-end and A-end of the carbody. The suspension deflections (across the spring group) shall be measured at both ends of the carbody. The vertical wheel loads will be obtained from measuring, for example, wheel cradle pressures with pressure transducers. The accuracy of measurements of roll angles, roll angle rates, and spring group deflections should be within 1%. The corresponding accuracy for measurement of wheel load should be within 5%. The data shall be filtered at 20 Hz using low-pass filter and digitized at 200 samples per second.

Excitation Input. The excitation shall be input to the VTU actuators making use of the profile generating system. A rectified sine wave profile shall be used to simulate a 39-foot staggered joint tangent track. Appropriate time delays shall be induced between axles depending on axle spacing for the test car. The rectified sine sweeps will be input with amplitude levels, for example, of 0.125, 0.25, 0.5, and 0.75 inch. (It should be noted here that the VTU does not allow wheel lift.)

Test Procedures. The test speeds shall range from 10 mph to 40 mph. Sample time of each speed (frequency) shall be the time required for ten low joints to be simulated. The speeds shall be simulated by inputting a discrete frequency sweep, data being recorded at each frequency dwell.

Data Reduction and Analysis. The test data shall be previewed through the use of time domain plots. The peak-to-peak values for roll angles, roll angle rates, and suspension deflections shall be extracted from the time history data, tabulated, and then plotted versus speed (frequency). The maximum and minimum values of wheel vertical loads shall be determined. From examining the time history data at different frequency dwells, the resonance speeds (frequencies) will be identified.

b. Bounce/Pitch

Data Requirements and Instrumentation. VTU shall be used to vibrate the rail car to simulate the action of parallel joint tangent track, and consequently examine the bounce/pitch phenomenon. Vertical acceleration data shall be acquired at, as a minimum, the B-end, A-end, and the center of the carbody at the sill level. The spring group deflections at both ends of the carbody shall be measured. The data for wheel vertical loads shall also be acquired. The accuracy of measurements for the accelerometers and the displacement transducers shall be within 1% and the accuracy for pressure transducers used to measure wheel loads shall be within 5%. The data shall be filtered at 20 Hz using low pass filter and digitized at 200 samples per second.

Excitation Input. The profile generating system shall be used to generate a rectified sine wave profile that simulates a 19½-foot parallel joint tangent track. Appropriate time delays shall be induced between axles depending on axle spacing. The amplitude levels of the rectified sine sweeps shall be varied, for example, 0.125, 0.25, 0.5, 0.75 inch.

Test Procedures. Maximum speeds ranging from 35 to 79 mph shall be simulated by inputting a discrete frequency sweep. The data acquired during the test runs shall be continuously recorded at each frequency dwell. Sample time for each speed (frequency) shall be the time required for ten low joints to be simulated.

Data Reduction and Analysis. The time history data for all channels shall be reviewed. The root mean square values of the vertical accelerations shall be determined and plotted versus speed (frequency). Maximum and minimum values of the vertical load at each wheel shall be determined, and the duration of wheel lift, if any, shall be identified. The bounce resonant frequency (critical speed) shall be identified from the time history plots.

c. Track Twist

Data Requirements and Instrumentation. Track twist load equalization includes both the static and quasi-static (very low speed) capabilities of a truck to withstand track irregularities. When the car is perfectly still, unequal wheel loads can exist depending upon the breakout force of the friction snubbers and center of gravity location of the car. For quasi-static case, where the rail car is traveling at a very low speed (less than 10 mph), the unequal wheel loads plus the occurrence of a lateral force can result in derailment.

A thorough investigation of load equalization shall be performed under the controlled laboratory conditions at the Rail Dynamics Laboratory. The Vibration Test Unit shall be used to evaluate static load equalization capability. This machine allows a fully loaded car/truck configuration to be mounted on eight vertical actuators. These vertical actuators can be positioned to cross level differences of up to 5.9 inches between any of the four wheels of a truck.

The vertical loads at the wheel/rail interface shall be determined using, for example, pressure transducers. These values of vertical forces shall be used to calculate the wheel unloading index.

Test Procedure. The VTU will be used to duplicate a full range of actual track twist conditions by varying wheelset roll amplitude and roll center location. This will be accomplished by slowly and continuously varying the actuators to test all possible configurations while simultaneously recording the vertical load at each wheel.

Track twist will be set up by varying the twist amplitude and the center of rotation of wheelsets one at a time and two at a time. During the tests, data shall be recorded for both increasing and decreasing track twist in order to detect any hysteresis. It is possible to have different wheel distributions even for the static load cases depending upon how the friction snubbers lock up when they come to rest.

Data Reduction and Analysis. The measured vertical loads at the wheel/rail interface shall be used to determine the wheel unloading index. The wheel unloading index will be plotted versus the angle of twist within axle spacing of the truck.

Test data analysis shall consist of evaluating the WUI performance index for the full range of track twist and up to the maximum accommodation during laboratory testing. Identification of the worst case conditions will allow correlation with existing track geometries encountered in yards, sidings, and special track work.

4.6.2.3 Ride Quality

Data Requirements and Instrumentation. The VTU will be used to generate the test data required to characterize the ride quality regime. Lateral and vertical acceleration data shall be acquired at least at the B-end and A-end and carbody center at sill level. Lateral acceleration data shall be acquired at the B-end and the A-end on the carobdy at the roof level.

The acceleration data acquired through accelerometers shall meet the following minimum critieria:

Cut-off frequency: 30 Hz
Range of measurements: $\pm$ 10 g's
Accuracy of measurements: 1%

The wheel excitation, whether it is generated using computer or previously recorded of actual rail profiles, shall be recorded continuously and simultaneously with the output response data.

Excitation Input. Computer-generated rail profiles making use of the profile generating system or recording of actual rail profiles shall be used to drive the hydraulic actuators. The track shall be Class 4 or better. The time delay due to the axle spacing will be taken into account when exciting the vehicle system. The track geometry of interest in study of the ride quality regime are profile, alignment, gauge, and cross level.

Test Procedure. The test speeds shall range from 30 to 79 mph. Sample time of each speed shall be 60 seconds. The data shall be recorded continuously at each speed dwell. Each of track input (profile, alignment, gauge, and cross level) will be treated separately and then collectively to study the effect of coupling in the multi-degree of freedom system.

Data Reduction and Analysis. Detailed statistical analysis shall be performed on the test data. The analysis shall include calculations of the frequency content of the data (i.e., the power spectral density functions using Fast Fourier transform technique), the rms values of the output signals and the wheel input, and the percent of the time a signal amplitude is above a given level as a function of that level. The transmissibility between the output signal and the wheel excitation will be calculated (the leading wheel of the leading truck can be used as a reference). This transmissibility may be characterized by a frequency dependent function of amplitude ratios called a transfer function, or a sequence of root mean square (rms) ratios of output-to-input over selected frequency bands (for example 0-4 Hz, 4-10 Hz and 10-20 Hz).

4.7 REMARKS

1. From an examination of the test results concerning the lateral stability regime, the following general conclusions can be drawn:

   a. Increasing vehicle speed is a destabilizing effect. An increase in the speed results in a higher hunting frequency and a decrease in the system damping.

   b. Related to a given truck, an empty car condition causes lower critical speed, i.e., the empty condition has a destabilizing character.

   c. The rail length is a very significant factor in determining the dynamic characteristics of the carbody/truck system. At low speeds, the forced frequencies arising from the rail joints force the vehicle system to oscillate at the jointed frequencies, and do not give the natural modes of oscillation a chance to be fully developed. At higher speeds, if the hunting frequency of the vehicle system is close to the forced oscillations (especially the first and second harmonics of rail joints), a synchronization occurs, i.e., the hunting movements of the vehicle system synchronizes with the forced oscillations. This synchronization makes an autonomous oscillation (hunting) very probable, and may lead to intermittent hunting.

      On the other hand, as the vehicle is experiencing fully developed hunting (sustained oscillation), the vehicle dynamic behaviors on jointed rail and continuous welded rail are comparable.

   d. There are several stages of hunting with increasing vehicle speed. The first stage manifests itself in a form of nosing or fishtailing. The second stage takes place with high probability of the other carbody end undergoing intermittent hunting. The last stage is achieved with violent motion for the whole system of carbody and trucks, indicating fully developed hunting. When the carbody starts hunting, the frequency remains approximately constant.

   e. For Type I trucks, the amplitudes of motion for the 100-ton configurations are much lower than these for the 70-ton configurations.

   f. The greater lozenging stiffness incorporated in the rigid truck designs, in association with the other companion modifications such as primary suspension elements, reduced coupling between the trucks and the carbody and dampening mechanisms, allow these truck designs to achieve improvements in lateral stability performance levels.

2. In the trackability regime the following may be noted:

   a. The loaded cars have better performance in equalizing the vertical load on the wheels than the empty cars.

   b. Primary suspension trucks seem to result in reduced vertical dynamic loads and thus point to potential improvements in freight car truck design.

3. In the steady state curve negotiation the following may be noted:

   a. The leading outer wheel which is the main guiding wheel when entering a curve, experiences and maintains larger lateral forces and L/V ratios above balance speed than does any other wheel. Above balance speed, the leading outer wheel also seems to be more sensitive to track curvature.

b. Track history, i.e., direction and magnitude of preceeding curve, has a significant influence on the curve negotiability. There is a pronounced difference in the level of the lateral forces generated during negotiation of left and right hand curves. This indicates the asymmetric characteristics of the generation of the lateral forces.

c. In general, below balance speed, the major share of the net lateral force applied to the track is due to the trailing axle of the truck. At and above balance speed, however, the leading axle carries the highest net lateral force.

d. The wheelsets of Type I trucks and the rigid trucks in Type II trucks are unable to align themselves with the local normal to the curve, and the guidance depends on the flange forces guiding the truck around the curve. In other words, the wheel flanges perform the primary role in curve negotiability in curves of moderate and large degree of curvature (2.5 degree to 6.2 degree curves).

e. The radial trucks seem to achieve a measured degree of success in attaining their goal of reducing the levels of lateral forces at the wheel/rail interface in curved track, especially in track of moderate curvature (less than 5 degrees).

4. The following are some observations concerning the reduced and analyzed data used to characterize the ride quality regime:

a. The role of train speed on the ride quality response of the carbody is clearly discernible; as the train speed is increased, more track excitation is transferred to the car, resulting in higher amplitude response.

b. Rail joint frequencies and the location of peaks in the power spectra are strongly related, indicating that the input excitation to the car arises mainly from the periodic rail joint spacing, with smaller contributions from the stochastic excitation from the random track irregularities.

c. For Type I trucks, in the roll mode, the loaded 70-ton vehicles exhibit more desirable dynamic characteristics than the loaded 100-ton vehicular combinations. In the vertical mode, the 100-ton trucks are more effective in attenuating the track excitations transmitted to the carbody than the 70-ton trucks.

d. Primary suspension trucks seem to have the ability to attenuate track excitations and thus, provide smooth ride conditions to the lading.

e. The loaded cars for 100-ton Type I and Type II trucks resonate in the speed range of 50 to 60 mph in both the vertical and lateral directions.

5. In general, no single Type II truck tested in the program seems to achieve significantly improved performance in all four performance regimes. The improved performance in specific performance regimes on the part of a given Type II truck can be related to specific design features which have desirable impact on performance in that regime. Improvement in performance in one regime is attained at the cost of degraded performance in another. A thorough evaluation of the specific design features, as compared to evaluation of the truck itself, with a view toward maximizing the potential benefits while at the same time optimizing the trade-off in detrimental effects, should be considered in continuing efforts of the type undertaken in TDOP Phase II. The potential for combining the advantageous technological features into one future truck design cannot be ruled out. The framework for pursuing such an effort is contained in the experimental and analytic methodology developed and used in TDOP Phase II.

6. The classification of truck performance into distinct performance regimes and identification of performance indices typical of each regime is an important first step in a standardized methodology for truck evaluation. Detailed analytic procedures used in reducing, analyzing and interpreting field test data and correlating the results to various service conditions have culminated in a set of recommended truck performance specifications.

4.8 REFERENCES

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SECTION 5 - ECONOMIC ANALYSIS

5.1 INTRODUCTION

The objectives of the TDOP Phase II economic analyses are (1) to identify the major parameters that govern the profitability of the Type II trucks, and (2) to delineate the trade-offs involved in choosing between Type I and Type II trucks. These objectives have been addressed through an analysis of results obtained from field test data, and an analysis of car maintenance data that were acquired from two major operating railroads in the United States. Added to the analyses of these data bases are considerations relating to such areas as expected additional costs that may reasonably be expected to accrue from the use of newer and less conventional equipment. As nearly as possible, such judgmental considerations have been arrived at in consultation with industry sources represented on the TDOP consultants group.

This section of the TDOP Phase II Final Report has been organized to reflect the major elements of analysis. The arrangement of the topics conform to the progression of effort within the economics task. First, the car maintenance analysis is discussed in detail, including procedures, data bases, and results. This is followed by subsections on fuel consumption, roadway maintenance, and lading damage and derailment. Here the various implications of the field test results in terms of freight car truck economics are discussed. Finally, the cost/benefit analysis section reviews the expected cost increases likely to accrue with the choice of Type II trucks as compared with the Type I trucks.

5.2 CAR MAINTENANCE ANALYSIS

An expected benefit with a Type II truck design is reduced car maintenance costs. One of the objectives of the engineering efforts in TDOP was to estimate the changes in performance that can be expected with a Type II truck (e.g., a 30% longer wheelset life with a steering truck). It is the purpose of this analysis to provide the methodology to convert this engineering performance evaluation into a dollar savings that would result from reduced car maintenance.

Conventional economic techniques have been employed to estimate the cost of all future maintenance over the projected life of a new freight car at the time the car is purchased. To build this estimate, complete maintenance records were considered from two railroads (the Union Pacific Railroad and an east coast railroad) over a three-year period of time for all roller bearing-equipped cars. The repair records of approximately 100,000 cars were used.

The principal variable influencing car maintenance was annual mileage. For this reason, the methodology was constructed to control for annual mileage variations among cars. Initially, it was assumed that different railroads would experience different maintenance costs, which is why two railroads were considered. Surprisingly, the overall results were quite similar. Finally, it was suspected that differences in carbody types and car weights were significant. Some effort has been expended along these lines (for example, results showed that 100-ton cars cost slightly less to maintain than 70-ton cars); however, estimates of this type are difficult to make because of a lack of long-time histories for the 100-ton cars. Although roller bearing cars have been used for 16 to 18 years in large numbers, 100-ton cars have considerably shorter histories. In the case of 100-ton open hoppers, there simply is not enough data to draw meaningful conclusions.

The results of the analysis suggest that car maintenance alone will not pay for a Type II truck, at least in the price ranges considered (from \$14,000 to \$21,000 per car set). Since the reduced maintenance costs are not enough on a per car basis to pay the added purchase price, the TDOP economic analysis went on to consider fuel consumption and rail wear. However, much more work could be done with the car maintenance data, for example:

a. Evaluation of equipment performance with the following objectives:
   - Establishment of differential costs between different carbody types for support in setting rates and calculating profitability of service.
   - Timely automatic identification of car series with mechanical problems.
   - Evaluation of individual manufacturer's parts as to their economic performance.

b. Automatic auditing of car repair billing data.

c. Calculation of the correct time for cars to be retired.

5.2.1 Results

Figure 5-1 illustrates the comparison between the eastern and Union Pacific (UP) railroads. The annual car mileages are shown on the X-axis. The Y-axis is the discounted cost of all future maintenance at the time of purchase of the car. This is the amount of money committed to car maintenance when one purchases a new car. The figure assumes a 10% discount rate and a 30-year or 1.2 million miles of car life, whichever comes first.

In examining the UP data, it is obvious that there are significant differences in maintenance costs between normal service cars and high mileage cars. This is an empirical result obtained by comparing data from high mileage cars with normal service cars. Not enough detailed mileage data are available from the eastern railroad to draw a distinction between normal service and high mileage service. As a result, the curve for the eastern railroad falls between the UP lines. High mileage service apparently experiences different repair costs than normal service. For example, unit trains do not couple and uncouple as often, saving wear on the couplers and brake lines. It appears that this type of service is less damaging per mile than conventional service. Furthermore, car types tend to be different between high mileage and normal service cars (e.g., the high mileage cars tend to be newer). Also, high mileage cars on the Union Pacific Railroad have an increased incidence of C-PEP trucks.

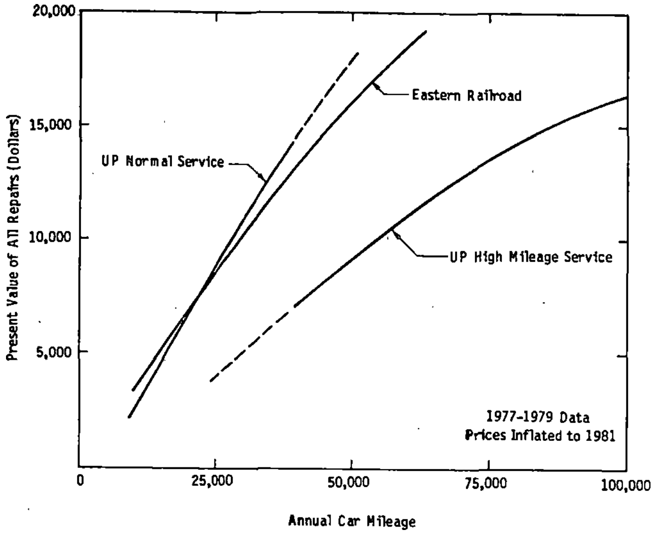


FIGURE 5-1. COMPARISON OF UNION PACIFIC AND EASTERN RAILROAD CAR MAINTENANCE COSTS

On the other extreme there is some evidence that very low mileage service (below 10,000 miles/year) is more expensive. This possibility has not been investigated because the low mileage cars are not very likely candidates for improved trucks, however, it might be an area to explore for other purposes.

Tables 5-1 and 5-2 break out the values from Figure 5-1 into components for the Union Pacific and eastern railroad, respectively. The tables are very heavily weighted toward truck repairs in keeping with the objectives of TDOP; however, all car repairs are covered. The level of aggregation is fairly high in the carbody.

The "present" in the table titles refers to the discounting of future expenditures to equivalent costs at the present time. In the case of Tables 5-1 and 5-2, the "present" is the time of purchase of a new car. The numbers in the tables are the amount of money that would need to be put in a bank at a 10% rate of return to completely pay for the expected cost of repairing the car over its entire life.

Tables 5-2A and 5-2B show the same data reduced to an equivalent annual cost. Timing of repairs has a very significant effect on the value of eliminating a given class of repairs. If a repair typically happens very late in the life of a car, it does not affect purchase decisions as much as the same repair would if it happened earlier. The equivalent annual cost calculation in Table 5-2A and Table 5-2B takes account of this difference; thus, it is not quite the same thing as an average annual cost.

Data in the heavy repair category for the eastern railroad included some very expensive repairs that appear to be a car rebuilding program. Since these costs are not really maintenance costs, any repair costing more than $10,000/car has been eliminated from the eastern railroad heavy repair data. Heavy repair data from UP were based on one year's records (1977) from the Omaha heavy repair facility only. UP currently collects, but does not keypunch, heavy repair data. Because of the non-random method of sampling, anomalies have occurred between 70-ton and 100-ton cars in Appendix D. The UP heavy repair data entry shown in the tables should be viewed with some skepticism.

5.2.2 Use of Table 5-1

The numbers in Table 5-1 are the dollar value per car of totally eliminating a class of repairs. For example, for a car that travels 25,000 miles/year, a braking system that never needed any maintenance would be worth paying $752.57 extra per car. For the same car, a truck braking system that would never wear out would be worth $1404.51 extra per car. If the wheelsets would never wear out, an extra $2307.49 per car would have been realized.

The data shown in Table 5-1 are from 1977 through 1979. The prices have been inflated to reflect 1981 levels. Several rule changes probably have reduced some of the costs. In particular, eliminating the requirement to remove the roller bearings any time there is a minor derailment will reduce the wheelset costs somewhat. About 16 percent of all wheelset replacements were due to derailment. Of these, about 6 percent are for truck set derailments (e.g., only one truck derails). If no wheelsets are replaced because of damaged roller bearings due to minor derailments, the wheelset numbers would be reduced by approximately 6 percent. Under the revised rules, the railroad is expected to inspect the roller bearings and replace them if they are damaged. A 4 to 5 percent reduction in wheelset costs due to this rule change may be expected. Other rule changes also are expected to reduce some of the categories other than trucks.

Finally, it should be pointed out that these are "average" car numbers. Because of the limitations of this type of analysis, the numbers should be regarded as accurate to only one significant figure. The numbers are indicative of the amount spent per car on each repair.

Example 1, Steering Truck

Table 5-1 was developed to evaluate the value of a change in car design. For example, increased wheel life is widely claimed as a benefit of switching to a steering truck. Assume a railroad had the opportunity to buy a steering truck for an extra $2000 per truck and expected a 30% longer wheel life, fuel savings, and reduced rail wear as principal benefits. Data from Table 5-1 can be used to evaluate the increased wheel life part of the benefits as follows:

Car Mileage	Wheelset Cost/Car	30% Savings Per Car	30% Savings Per Truck	Investment Tax Credit
12,500	1060.53	318.16	159.08	174.99
25,000	2307.49	692.25	346.13	380.74
37,500	3464.37	1039.31	519.66	571.63
50,000	3245.84	973.75	486.88	535.56
62,500	3979.21	1193.76	596.88	656.57
75,000	4551.28	1365.38	682.69	750.96
87,500	5052.93	1515.88	757.94	833.73
100,000	5490.52	1647.16	823.58	905.94

As shown from the first column, the savings are strongly dependent on the annual mileage assumed by the cars. The second column is the wheelset cost per car from the summary part of Table 5-1. The third column shows 30% of the wheelset cost/car, i.e., the 30% savings per car. The fourth column is half of the third column (two trucks per car). Finally, the last column is 1.1 times the fourth column, reflecting the 10% investment tax credit on new investment. The result is the nominal dollar value of 30% longer wheel life per truck.

This is not the total amount saved over the life of the car. Since the discount rate is 10%, for the 25,000 mile/year case, (about .10 x $692.25 = $69.23 per year) will be saved for an estimated life of 30 years. So the savings is $2076.74 (about 30 x $69.23) over the life of the car. However, that money is spread out over the life of the car. A railroad should be willing to pay only $380.74/truck today to achieve the projected future savings. Economists regard this as a break-even proposition; money is neither made nor lost. The argument goes that if you have $380.74 today you can invest it and reasonably expect a 10% rate of return. If the 10% return is not realized, the money should be invested in something else.

On the basis of wheelset life alone, this is not a particularly attractive investment. For the normal car (about 25,000 miles/year), a railroad would be willing to pay up to $380, but is asked to pay $2000. An investment like this would cost $1620 per truck. For unit train service (about 75,000 miles/year), a railroad would be willing to pay $750. By itself, this is still not enough to cover the extra purchase price.

Example 2, New Center Bowl Design

Another example would be a new type of center bowl that would never wear out. To contrast this to the use of conventional wear liners, Table 5-1 shows that center plate liners run about $28.75/car for a 25,000 mile/year car.

Savings would also be realized in reduced center plate wear. This item runs about $4.55/car for a 25,000 mile/year car. Perhaps half of the expense is associated with fractures rather than wear.

Adding these costs gives $28.75 + 0.5 x $4.55 = $31.03/year or $15.51/truck. Adjusting for the investment tax credit gives $17.06 as the added cost/center bowl that could be paid for this improvement and still break even.

Example 3, Composite Material Brake Shoes

Assume that a composite material brake shoe would wear out 30% slower than a conventional brake shoe and would cost 25% more than a conventional brake shoe. The cost of replacing brake shoes would probably be 25% higher all through the car's life. This is not a one-time added cost at the time of purchase of a new car.

The table can be used to estimate the value of the saving per new car by calculating what a 5% savings (30% more life - 25% more cost) is worth. For a 25,000 mile/year car, this is 0.05 x $1404.57 = $60.01, illustrating why there is considerable interest in the relative performance of different types of brake shoes. Even small changes in the economic performance of a brake shoe are worth significant amounts of money.

Table 5-1 is not designed for dealing with retrofit issues. It is constructed based on the assumption that any changes will be made at the time of purchase of new cars. However, methodology used to construct the table is directly applicable to retrofit questions. Simple modifications of the procedure are discussed in subsection 5.2.3.

TABLE 5-1. PRESENT VALUE AT TIME OF PURCHASE OF ALL FUTURE REPAIRS FOR UNION PACIFIC ROLLER BEARING CARS

SUMMARY TABLE	NORMAL SERVICE			HIGH MILEAGE SERVICE				
ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	354.80	752.57	1135.20	797.40	1009.55	1171.07	1311.85	1433.38
COUPLERS, YOKES, & DRAFT GEAR	122.58	413.23	737.50	697.10	918.19	1096.97	1263.11	1406.75
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	368.40	1104.62	1754.38	712.48	915.21	1071.52	1214.32	1335.51
OTHER CAR REPAIRS	267.34	701.02	1198.26	1136.02	1524.21	1811.37	2064.67	2282.81
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	640.05	1404.51	2205.92	2153.12	2746.14	3217.04	3625.11	3985.78
WHEELSETS	1060.53	2307.49	3464.37	3245.84	3979.21	4551.28	5052.93	5490.52
OTHER TRUCK REPAIRS	139.19	601.55	1066.28	275.23	355.08	419.97	475.40	523.26
HEAVY REPAIRS	590.59	1529.18	2050.87	153.20	170.17	184.64	200.33	212.32
TOTAL	3543.48	8814.18	13612.79	9170.39	11617.77	13523.86	15207.72	16670.32
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:				ANNUAL MILEAGE				
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	354.80	752.57	1135.20	797.40	1009.55	1171.07	1311.85	1433.38
COT&S	68.83	174.97	275.29	95.11	124.65	147.91	167.43	185.40
IDT&S	179.79	357.87	520.55	431.39	531.50	604.52	666.71	719.84
PRESSURE SYSTEM	84.96	182.70	287.38	244.66	319.43	378.33	431.63	477.14
HAND BRAKES	21.22	37.03	51.99	26.24	33.96	40.32	46.08	51.00
COUPLERS, YOKES, & DRAFT GEAR	122.58	413.23	737.50	697.10	918.19	1096.97	1263.11	1406.75
COUPLER BODIES	43.52	144.21	260.37	342.06	455.85	547.84	632.78	707.42
COUPLER KNUCKLES	21.37	69.87	120.44	94.10	121.11	143.00	163.15	180.39
OTHER COUPLER PARTS	39.19	110.21	179.37	123.27	155.91	181.92	205.67	225.40
YOKES	5.43	27.20	53.44	35.31	47.13	56.57	65.42	73.06
DRAFT GEARS, CARRIERS, AND FOLLOWERS	13.06	61.74	123.88	102.36	138.19	167.64	196.09	220.48
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	368.40	1104.62	1754.38	712.48	915.21	1071.52	1214.32	1335.51
OTHER CAR REPAIRS	267.34	701.02	1198.26	1136.02	1524.21	1811.37	2064.67	2282.81
OTHER CAR REPAIRS	138.38	352.10	627.09	650.75	895.54	1078.20	1241.33	1382.57
WELDING	57.16	176.93	301.44	261.07	343.51	403.61	456.32	500.17
NON BILLABLE INSPECTIONS	71.80	172.00	269.73	224.20	285.16	329.55	367.01	400.07
CAR TOTAL	1113.12	2971.45	4825.34	3343.00	4367.16	5150.93	5853.95	6458.45

TRUCK REPAIRS	NORMAL SERVICE			HIGH MILEAGE SERVICE				
ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	640.05	1404.51	2205.92	2153.12	2746.14	3217.04	3625.11	3985.78
BRAKE BEAMS	61.32	172.66	335.72	279.15	399.97	502.48	591.64	674.35
BRAKE BEAM WEAR PLATES	0.17	0.41	0.57	0.07	0.08	0.09	0.09	0.10
BRAKE BEAM HANGERS	0.02	0.04	0.05					
BRAKE HANGER BRACKET WEAR PLATE	0.00	0.01	0.02					
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.01	0.01	0.02					
BRAKE HANGER OR CONNECTION PIN	2.81	6.22	9.73	8.94	11.22	12.98	14.59	15.93
BOTTOM ROD SAFETY SUPPORT	1.07	2.85	4.92	6.74	9.10	10.91	12.60	14.00
BRAKE BEAM SAFETY SUPPORT	0.32	0.78	1.06	0.09	0.11	0.12	0.12	0.13
BRAKE CONNECTION, BOTTOM	2.38	4.29	5.95	6.11	7.39	8.38	9.29	10.01
BRAKE CONNECTION, TOP	2.44	5.89	9.35	12.93	16.79	19.73	22.25	24.55
BRAKE LEVER	1.68	2.97	4.14	4.12	4.98	5.65	6.29	6.78
BRAKE LEVER GUIDE OR CARRIER	0.05	0.15	0.23	0.08	0.10	0.11	0.13	0.14
DEAD LEVER GUIDE	0.02	0.06	0.22	0.40	0.61	0.78	0.94	1.08
DEAD LEVER GUIDE BRACKET	0.02	0.06	0.15	0.44	0.62	0.75	0.86	0.97
BRAKE SHOES	563.89	1200.27	1822.28	1826.90	2286.24	2644.75	2954.81	3225.23
BRAKE SHOE KEYS	3.86	7.85	11.52	7.15	8.93	10.30	11.48	12.49
WHEELSETS	1060.53	2307.49	3464.37	3245.84	3979.21	4551.28	5052.93	5490.52
LUBRICATE ROLLER BEARINGS	19.85	44.76	63.14	50.93	60.02	66.90	72.72	78.20
ROLLER BEARINGS	148.75	320.44	483.36	461.49	567.51	655.30	733.79	803.10
ROLLER BEARING CAP SCREWS	0.08	0.23	0.36	0.39	0.46	0.51	0.54	0.57
ROLLER BEARING LOCKING PLATES	0.00	0.00	0.01	0.02	0.03	0.03	0.03	0.03
ROLLER BEARING LUBRICATION FITTING	0.02	0.02	0.03	0.02	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	20.16	67.49	114.43	78.10	100.47	118.13	133.75	147.43
WHEELS	356.42	773.29	1170.61	1106.52	1359.56	1569.12	1756.34	1921.56
WHEEL LABOR	509.98	1069.84	1615.23	1531.93	1870.91	2117.91	2329.58	2510.99
AXLES, ROLLER BEARINGS	5.29	11.40	17.21	16.45	20.23	23.35	26.15	28.62
OTHER TRUCK REPAIRS	139.19	601.55	1066.28	275.23	355.08	419.97	475.40	523.26
TRUCK BOLSTERS	38.81	181.09	327.67	71.16	93.09	111.38	126.12	139.67
TRUCK BOLSTERS (REPAIRED)	0.98	6.27	11.48	3.26	4.63	5.66	6.41	7.14
CENTER PINS	2.59	7.43	12.60	7.13	9.07	10.57	11.85	13.03
CENTER PLATES	1.63	4.55	9.13	0.44	0.64	0.79	0.94	1.05
CENTER PLATE LINERS	6.93	28.75	50.60	36.95	47.34	55.08	61.20	67.09
TRUCK SIDE BEARINGS	6.24	18.09	28.54	11.25	14.14	16.30	18.02	19.71
FRICTION CASTINGS	13.19	51.48	86.30	20.46	25.42	29.25	32.51	35.41
SIDE BEARING SHIM	0.29	2.12	3.72	0.73	0.93	1.10	1.26	1.37
SIDE FRAMES	49.53	223.62	405.43	67.27	91.99	112.98	132.32	147.61
SIDE FRAMES (REPAIRED)	0.62	4.69	7.80	0.58	0.77	0.91	1.01	1.10
SPRING PLANKS	0.00	0.02	0.04					
OUTER SPRINGS	7.57	31.13	51.66	17.77	20.52	22.81	24.56	26.00
INNER SPRINGS	4.27	16.37	26.68	10.77	12.41	13.70	14.76	15.57
STABILIZER SPRINGS	3.30	13.25	21.87	5.19	6.13	6.87	7.46	8.01
TRUCK SPRING FRICTION SNUBBER	0.04	0.09	0.12	0.03	0.03	0.03	0.03	0.04
TRUCK SPRING PLATES	0.01	0.02	0.03					
TRUCK SPRING SHIM, WOOD	0.00	0.01	0.01					
STEEL	0.05	0.15	0.28	2.71	3.44	3.87	4.54	4.94
MANUFACTURED MATERIAL (TRUCK)	3.14	12.43	22.32	19.52	24.54	28.65	32.42	35.53
TRUCK TOTAL	1839.77	4313.56	6736.57	5674.20	7080.44	8188.30	9153.44	9999.56

TABLE 5-2. PRESENT VALUE AT TIME OF PURCHASE OF ALL FUTURE REPAIRS FOR EASTERN RAILROAD ROLLER BEARING CARS

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	587.99	1074.81	1526.42	1969.75	2392.25	2706.51	2975.68	3208.45
COUPLERS, YOKES, & DRAFT GEAR	282.57	743.08	1234.30	1736.42	2234.73	2636.13	2985.38	3301.64
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	488.04	1104.55	1635.80	2126.69	2585.72	2934.52	3235.33	3493.61
OTHER CAR REPAIRS	397.89	862.33	1303.38	1703.93	2076.57	2348.17	2581.92	2780.46
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	405.24	828.44	1223.00	1608.43	1973.91	2253.15	2499.63	2713.62
WHEELSETS	945.91	1844.48	2573.33	3227.25	3809.24	4257.48	4632.40	4965.12
OTHER TRUCK REPAIRS	88.08	225.25	347.52	456.77	556.52	630.85	691.90	745.72
HEAVY REPAIRS	856.91	1992.12	2680.01	3151.81	3506.96	3763.61	3964.36	4140.52
TOTAL	4052.63	8695.06	12523.77	15983.04	19135.90	21530.43	23566.60	25349.15
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	587.99	1074.81	1526.42	1969.75	2392.25	2706.51	2975.68	3208.45
COT&S	140.48	262.27	370.91	472.96	569.99	643.08	701.64	753.35
IDT&S	275.99	510.84	732.42	956.72	1170.15	1323.12	1456.86	1570.43
PRESSURE SYSTEM	124.64	226.34	317.55	403.69	485.25	548.35	603.14	650.59
HAND BRAKES	46.88	75.37	105.54	136.38	166.65	191.95	214.04	234.08
COUPLERS, YOKES, & DRAFT GEAR	282.57	743.08	1234.30	1736.42	2234.73	2636.13	2985.38	3301.64
COUPLER BODIES	88.08	231.14	391.43	561.54	732.34	871.26	995.23	1105.65
COUPLER KNUCKLES	43.63	104.85	164.94	222.61	278.94	324.24	363.47	398.71
OTHER COUPLER PARTS	70.83	158.53	247.70	336.97	423.66	492.20	552.66	606.00
YOKES	21.45	65.98	111.98	157.50	201.85	236.44	265.49	292.37
DRAFT GEARS, CARRIERS, AND FOLLOWERS	58.58	182.58	318.25	457.80	597.94	711.99	808.53	898.90
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	488.04	1104.55	1635.80	2128.69	2585.72	2934.52	3235.33	3493.61
OTHER CAR REPAIRS	397.89	882.33	1303.38	1703.93	2076.57	2348.17	2581.92	2780.46
OTHER CAR REPAIRS	300.48	634.07	927.22	1205.30	1465.09	1656.29	1820.19	1960.67
WELDING	97.41	248.26	376.16	498.64	611.47	691.89	761.73	819.79
CAR TOTAL	1756.49	3804.77	5699.91	7538.79	9289.26	10625.33	11778.31	12784.16

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	405.24	828.44	1223.00	1608.43	1973.91	2253.15	2499.63	2713.62
BRAKE BEAMS	70.94	133.58	189.11	241.23	289.63	328.40	361.56	391.30
BRAKE BEAM WEAR PLATES	0.01	0.07	0.11	0.14	0.16	0.17	0.18	0.19
BRAKE BEAM HANGERS	0.01	0.17	0.40	0.68	0.95	1.17	1.36	1.54
BRAKE HANGER BRACKET WEAR PLATE	0.00	0.04	0.12	0.22	0.31	0.40	0.48	0.55
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.00	0.04	0.11	0.20	0.28	0.35	0.41	0.46
BRAKE HANGER OR CONNECTION PIN	4.61	8.81	13.34	18.23	23.30	27.17	30.43	33.40
BOTTOM ROD SAFETY SUPPORT	3.89	7.27	10.45	13.81	16.97	19.52	21.64	23.59
BRAKE BEAM SAFETY SUPPORT	0.51	1.14	1.99	3.01	4.05	4.82	5.55	6.18
BRAKE CONNECTION, BOTTOM	5.60	8.76	11.36	13.90	16.26	18.12	19.75	21.18
BRAKE CONNECTION, TOP	4.10	9.25	15.00	21.54	27.95	32.89	37.71	41.83
BRAKE LEVER	14.87	18.30	20.45	22.28	23.89	25.18	26.20	27.12
BRAKE LEVER GUIDE OR CARRIER	0.28	0.87	1.31	1.65	1.94	2.15	2.33	2.47
DEAD LEVER GUIDE	0.08	0.18	0.28	0.38	0.47	0.55	0.61	0.67
DEAD LEVER GUIDE BRACKET	0.11	0.19	0.23	0.26	0.28	0.29	0.31	0.31
BRAKE SHOES	296.77	633.10	949.28	1258.83	1553.03	1775.72	1973.36	2143.78
BRAKE SHOE KEYS	3.45	6.66	9.47	12.05	14.44	16.22	17.74	19.06
WHEELSETS	945.91	1844.48	2573.33	3227.25	3809.24	4257.48	4632.40	4965.12
LUBRICATE ROLLER BEARINGS	22.93	43.25	61.20	77.84	93.65	105.28	115.45	123.97
ROLLER BEARINGS	140.76	270.85	374.15	461.75	536.52	595.66	645.48	689.42
ROLLER BEARING CAP SCREWS	0.17	0.41	0.77	1.22	1.73	2.18	2.56	2.94
ROLLER BEARING LOCKING PLATES	0.00	0.01	0.02	0.03	0.03	0.04	0.04	0.04
ROLLER BEARING LUBRICATION FITTING	0.01	0.02	0.03	0.03	0.04	0.04	0.04	0.04
PEDESTAL ADAPTERS	25.81	61.69	94.48	125.43	155.35	178.62	198.22	216.03
WHEELS	283.13	562.32	809.62	1039.67	1250.53	1422.45	1569.26	1702.22
WHEEL LABOR	468.21	896.40	1219.76	1504.64	1751.85	1931.12	2077.50	2204.85
AXLES, ROLLER BEARINGS	4.90	9.52	13.30	16.64	19.54	21.88	23.85	25.61
OTHER TRUCK REPAIRS	88.08	225.25	347.52	456.77	556.52	630.85	691.90	745.72
TRUCK BOLSTERS	30.74	76.86	109.18	131.56	149.57	162.23	170.74	178.39
TRUCK BOLSTERS (REPAIRED)	0.61	1.66	2.31	2.79	3.11	3.32	3.51	3.69
CENTER PINS	7.67	16.00	23.55	30.64	37.20	42.05	46.30	50.06
CENTER PLATES	1.12	5.01	8.13	10.70	12.79	14.26	15.57	16.70
CENTER PLATE LINERS	9.55	26.55	45.73	65.90	86.35	101.50	114.09	125.57
TRUCK SIDE BEARINGS	9.22	16.07	22.03	27.58	32.75	36.60	40.00	42.95
FRICTION CASTINGS	4.62	9.71	16.12	22.61	29.16	34.07	38.45	42.26
SIDE BEARING SHIM	0.74	6.59	12.89	20.08	27.16	32.14	36.68	40.41
SIDE FRAMES	6.06	23.10	36.69	47.36	55.18	61.05	65.42	69.03
SIDE FRAMES (REPAIRED)	0.14	1.05	1.66	2.14	2.47	2.67	2.88	3.09
SPRING PLANKS (REPAIRED)	0.00	0.05	0.09	0.12	0.14	0.16	0.18	0.19
OUTER SPRINGS	6.55	19.70	34.90	50.49	65.90	78.27	89.03	98.54
INNER SPRINGS	2.96	8.27	14.97	22.08	29.20	34.80	39.70	44.04
STABILIZER SPRINGS	0.35	0.75	1.08	1.40	1.70	1.92	2.12	2.26
TRUCK SPRING FRICTION SNUBBER	0.40	0.91	1.24	1.45	1.61	1.73	1.83	1.88
TRUCK SPRING PLATES	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
TRUCK SPRING SHIM, WOOD	0.01	0.01	0.06	0.13	0.20	0.24	0.29	0.33
MANUFACTURED MATERIAL (TRUCK)	7.34	12.97	16.88	19.74	22.01	23.61	25.10	26.29
TRUCK TOTAL	1439.23	2898.17	4143.84	5292.45	6339.68	7141.48	7823.93	8424.46

TABLE 5-2A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR UNION PACIFIC ROLLER BEARING CARS

SUMMARY TABLE	NORMAL SERVICE			HIGH MILEAGE SERVICE				
ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	37.64	79.83	120.42	84.59	108.48	130.34	154.09	174.77
COUPLERS, YOKES, & DRAFT GEAR	13.00	43.84	78.23	73.95	98.66	122.09	148.36	171.53
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	39.08	117.18	186.10	75.58	98.34	119.26	142.63	162.84
OTHER CAR REPAIRS	28.36	74.36	127.11	120.51	163.78	201.61	242.52	278.34
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	67.90	148.99	234.00	228.40	295.08	358.06	425.81	485.99
WHEELSETS	112.50	244.78	367.50	344.32	427.57	506.56	593.52	669.46
OTHER TRUCK REPAIRS	14.76	63.81	113.11	29.20	38.15	46.74	55.84	63.80
HEAVY REPAIRS	62.65	162.21	217.56	16.25	18.29	20.55	23.53	25.89
TOTAL	375.89	935.01	1444.04	972.79	1248.35	1505.21	1786.30	2032.62
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	37.64	79.83	120.42	84.59	108.48	130.34	154.09	174.77
COT&S	7.30	18.56	29.20	10.09	13.39	16.46	19.67	22.61
IDT&S	19.07	37.96	55.22	45.76	57.11	67.28	78.31	87.77
PRESSURE SYSTEM	9.01	19.38	30.49	25.95	34.32	42.11	50.70	58.18
HAND BRAKES	2.25	3.93	5.51	2.78	3.65	4.49	5.41	6.22
COUPLERS, YOKES, & DRAFT GEAR	13.00	43.84	78.23	73.95	98.66	122.09	148.36	171.53
COUPLER BODIES	4.62	15.30	27.62	36.29	48.98	60.97	74.33	86.26
COUPLER KNUCKLES	2.27	7.41	12.78	9.98	13.01	15.92	19.16	21.99
OTHER COUPLER PARTS	4.16	11.69	19.03	13.08	16.75	20.25	24.16	27.48
YOKES	0.58	2.89	5.67	3.75	5.06	6.30	7.68	8.91
DRAFT GEARS, CARRIERS, AND FOLLOWERS	1.39	6.55	13.14	10.86	14.85	18.66	23.03	26.88
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	39.08	117.18	186.10	75.58	98.34	119.26	142.63	162.84
OTHER CAR REPAIRS	28.36	74.36	127.11	120.51	163.78	201.61	242.52	278.34
OTHER CAR REPAIRS	14.68	37.35	66.52	69.03	96.23	120.00	145.81	168.58
WELDING	6.06	16.77	31.98	27.69	36.91	44.92	53.60	60.99
NON BILLABLE INSPECTIONS	7.62	18.25	28.61	23.78	30.64	36.68	43.11	48.78
CAR TOTAL	118.08	315.21	511.87	354.62	469.26	573.30	687.60	787.48

TRUCK REPAIRS	NORMAL SERVICE			HIGH MILEAGE SERVICE				
ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	67.90	148.99	234.00	228.40	295.08	358.06	425.81	485.99
BRAKE BEAMS	6.50	18.32	35.61	29.61	42.98	55.93	69.49	82.22
BRAKE BEAM WEAR PLATES	0.02	0.04	0.06	0.01	0.01	0.01	0.01	0.01
BRAKE HANGER OR CONNECTION PIN	0.30	0.66	1.03	0.95	1.21	1.44	1.71	1.94
BOTTOM ROD SAFETY SUPPORT	0.11	0.30	0.52	0.71	0.98	1.21	1.48	1.71
BRAKE BEAM SAFETY SUPPORT	0.03	0.08	0.11	0.01	0.01	0.01	0.01	0.02
BRAKE CONNECTION, BOTTOM	0.25	0.45	0.63	0.65	0.79	0.93	1.09	1.22
BRAKE CONNECTION, TOP	0.26	0.62	0.99	1.37	1.80	2.20	2.61	2.99
BRAKE LEVER	0.18	0.31	0.44	0.44	0.53	0.63	0.74	0.83
BRAKE LEVER GUIDE OR CARRIER	0.01	0.02	0.02	0.01	0.01	0.01	0.01	0.02
DEAD LEVER GUIDE	0.00	0.01	0.02	0.04	0.07	0.09	0.11	0.13
DEAD LEVER GUIDE BRACKET	0.00	0.01	0.02	0.05	0.07	0.08	0.10	0.12
BRAKE SHOES	59.82	127.32	193.31	193.80	245.66	294.36	347.07	393.25
BRAKE SHOE KEYS	0.41	0.83	1.22	0.76	0.96	1.15	1.35	1.52
WHEELSETS	112.50	244.78	367.50	344.32	427.57	506.56	593.52	669.46
LUBRICATE ROLLER BEARINGS	2.11	4.75	6.70	5.40	6.45	7.45	8.54	9.53
ROLLER BEARINGS	15.78	33.99	51.27	48.96	60.98	72.94	86.19	97.92
ROLLER BEARING CAP SCREWS	0.01	0.02	0.04	0.04	0.05	0.06	0.06	0.07
PEDESTAL ADAPTERS	2.14	7.16	12.14	8.28	10.80	13.15	15.71	17.98
WHEELS	37.81	82.03	124.18	117.38	146.09	174.64	206.30	234.30
WHEEL LABOR	54.10	115.61	171.34	162.51	201.03	235.72	273.63	306.17
AXLES, ROLLER BEARINGS	0.56	1.21	1.83	1.74	2.17	2.60	3.07	3.49
OTHER TRUCK REPAIRS	14.76	63.81	113.11	29.20	38.15	46.74	55.84	63.80
TRUCK BOLSTERS	4.12	19.21	34.76	7.55	10.00	12.40	14.81	17.03
TRUCK BOLSTERS (REPAIRED)	0.10	0.67	1.22	0.35	0.50	0.63	0.75	0.87
CENTER PINS	0.27	0.79	1.34	0.76	0.97	1.18	1.39	1.59
CENTER PLATES	0.17	0.48	0.97	0.05	0.07	0.09	0.11	0.13
CENTER PLATE LINERS	0.73	3.05	5.37	3.92	5.09	6.13	7.19	8.18
TRUCK SIDE BEARINGS	0.66	1.92	3.03	1.19	1.52	1.81	2.12	2.40
FRICTION CASTINGS	1.40	5.46	9.16	2.17	2.73	3.26	3.82	4.32
SIDE BEARING SHIM	0.03	0.23	0.39	0.08	0.10	0.12	0.15	0.17
SIDE FRAMES	5.25	23.72	43.01	7.14	9.88	12.57	15.54	18.00
SIDE FRAMES (REPAIRED)	0.07	0.50	0.83	0.06	0.08	0.10	0.12	0.13
SPRING PLANKS	0.00	0.00	0.00					
OUTER SPRINGS	0.80	3.30	5.48	1.89	2.21	2.54	2.89	3.17
INNER SPRINGS	0.45	1.74	2.83	1.14	1.33	1.53	1.73	1.90
STABILIZER SPRINGS	0.35	1.41	2.32	0.55	0.66	0.76	0.88	0.98
TRUCK SPRING FRICTION SNUBBER	0.00	0.01	0.01					
STEEL	0.01	0.02	0.03	0.29	0.37	0.43	0.53	0.60
MANUFACTURED MATERIAL (TRUCK)	0.33	1.32	2.37	2.07	2.64	3.19	3.81	4.33
TRUCK TOTAL	195.16	457.58	714.61	601.92	760.80	911.36	1075.16	1219.25

TABLE 5-2B. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR EASTERN RAILROAD ROLLER BEARING CARS

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	62.37	114.02	161.92	208.95	257.05	301.23	349.52	391.21
COUPLERS, YOKES, & DRAFT GEAR	29.97	78.83	130.93	184.20	240.13	293.40	350.66	402.57
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	51.77	117.17	173.53	225.81	277.84	326.61	380.02	425.98
OTHER CAR REPAIRS	42.21	93.60	138.26	180.75	223.13	261.35	303.27	339.02
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	42.99	87.88	129.74	170.62	212.10	250.78	293.61	330.87
WHEELSETS	100.34	195.66	272.98	342.35	409.31	473.66	544.12	605.40
OTHER TRUCK REPAIRS	9.34	23.89	36.86	48.45	59.80	70.21	81.27	90.93
HEAVY REPAIRS	90.90	211.32	284.29	334.34	376.83	418.89	465.65	504.86
TOTAL	429.90	922.37	1328.52	1695.48	2056.18	2396.34	2768.13	3090.84
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	62.37	114.02	161.92	208.95	257.05	301.23	349.52	391.21
COT&S	14.90	27.82	39.35	50.17	61.25	71.57	82.41	91.86
IDT&S	29.28	54.19	77.69	101.49	125.73	147.26	171.12	191.48
PRESSURE SYSTEM	13.22	24.01	33.69	42.82	52.14	61.03	70.85	79.33
HAND BRAKES	4.97	7.99	11.20	14.47	17.95	21.36	25.14	28.54
COUPLERS, YOKES, & DRAFT GEAR	29.97	78.83	130.93	184.20	240.13	293.40	350.66	402.57
COUPLER BODIES	9.34	24.52	41.52	59.57	78.69	96.97	116.90	134.81
COUPLER KNUCKLES	4.63	11.12	17.50	23.61	29.97	36.09	42.69	48.62
OTHER COUPLER PARTS	7.51	16.82	26.28	35.75	45.52	54.78	64.92	73.89
YOKES	2.28	7.00	11.88	16.71	21.69	26.32	31.18	35.65
DRAFT GEARS, CARRIERS, AND FOLLOWERS	6.21	19.37	33.76	48.56	64.25	79.24	94.97	109.60
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	51.77	117.17	173.53	225.81	277.84	326.61	380.02	425.98
OTHER CAR REPAIRS	42.21	93.60	138.26	180.75	223.13	261.35	303.27	339.02
OTHER CAR REPAIRS	31.87	67.26	98.36	127.86	157.45	184.34	213.80	239.07
WELDING	10.33	26.34	39.90	52.90	65.70	77.01	89.47	99.96
CAR TOTAL	186.33	403.61	604.64	799.71	998.14	1182.60	1383.48	1558.78

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	42.99	87.88	129.74	170.62	212.10	250.78	293.61	330.87
BRAKE BEAMS	7.53	14.17	20.06	25.59	31.12	36.55	42.47	47.71
BRAKE BEAM WEAR PLATES	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
BRAKE BEAM HANGERS	0.00	0.02	0.04	0.07	0.10	0.13	0.16	0.19
BRAKE HANGER BRACKET WEAR PLATE	0.00	0.00	0.01	0.02	0.03	0.04	0.06	0.07
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.00	0.00	0.01	0.02	0.03	0.04	0.05	0.06
BRAKE HANGER OR CONNECTION PIN	0.49	0.93	1.42	1.93	2.50	3.02	3.57	4.07
BOTTOM ROD SAFETY SUPPORT	0.41	0.77	1.11	1.47	1.82	2.17	2.54	2.88
BRAKE BEAM SAFETY SUPPORT	0.05	0.12	0.21	0.32	0.44	0.54	0.65	0.75
BRAKE CONNECTION, BOTTOM	0.59	0.93	1.21	1.47	1.75	2.02	2.32	2.58
BRAKE CONNECTION, TOP	0.44	0.98	1.59	2.29	3.00	3.66	4.43	5.10
BRAKE LEVER	1.58	1.94	2.17	2.36	2.57	2.80	3.08	3.31
BRAKE LEVER GUIDE OR CARRIER	0.03	0.09	0.14	0.18	0.21	0.24	0.27	0.30
DEAD LEVER GUIDE	0.01	0.02	0.03	0.04	0.05	0.06	0.07	0.08
DEAD LEVER GUIDE BRACKET	0.01	0.02	0.02	0.03	0.03	0.03	0.04	0.04
BRAKE SHOES	31.48	67.16	100.70	133.54	166.87	197.64	231.79	261.39
BRAKE SHOE KEYS	0.37	0.71	1.00	1.28	1.55	1.81	2.08	2.32
WHEELSETS	100.34	195.66	272.98	342.35	409.31	473.86	544.12	605.40
LUBRICATE ROLLER BEARINGS	2.43	4.59	6.49	8.26	10.06	11.72	13.56	15.12
ROLLER BEARINGS	14.93	28.73	39.69	48.98	57.65	66.32	75.82	84.06
ROLLER BEARING CAP SCREWS	0.02	0.04	0.08	0.13	0.19	0.24	0.30	0.36
PEDESTAL ADAPTERS	2.74	6.54	10.02	13.31	16.69	19.88	23.28	26.34
WHEELS	30.03	59.65	85.88	110.29	134.37	158.32	184.33	207.55
WHEEL LABOR	49.67	95.09	129.39	159.61	186.24	214.95	244.02	268.84
AXLES, ROLLER BEARINGS	0.52	1.01	1.41	1.76	2.10	2.43	2.80	3.12
OTHER TRUCK REPAIRS	9.34	23.89	36.86	48.45	59.80	70.21	81.27	90.93
TRUCK BOLSTERS	3.26	8.15	11.58	13.96	16.07	18.06	20.05	21.75
TRUCK BOLSTERS (REPAIRED)	0.06	0.18	0.24	0.30	0.33	0.37	0.41	0.45
CENTER PINS	0.81	1.70	2.50	3.25	4.00	4.68	5.44	6.10
CENTER PLATES	0.12	0.53	0.86	1.14	1.37	1.59	1.83	2.04
CENTER PLATE LINERS	1.01	2.82	4.85	6.99	9.28	11.30	13.40	15.31
TRUCK SIDE BEARINGS	0.98	1.70	2.34	2.93	3.52	4.07	4.70	5.24
FRICTION CASTINGS	0.49	1.03	1.71	2.40	3.13	3.79	4.52	5.15
SIDE BEARING SHIM.	0.08	0.70	1.37	2.13	2.92	3.58	4.31	4.93
SIDE FRAMES	0.64	2.45	3.89	5.02	5.93	6.80	7.68	8.42
SIDE FRAMES (REPAIRED)	0.01	0.11	0.18	0.23	0.27	0.30	0.34	0.38
SPRING PLANKS (REPAIRED)	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
OUTER SPRINGS	0.70	2.09	3.70	5.36	7.08	8.71	10.46	12.01
INNER SPRINGS	0.31	0.88	1.59	2.34	3.14	3.67	4.66	5.37
STABILIZER SPRINGS	0.04	0.08	0.11	0.15	0.18	0.21	0.25	0.28
TRUCK SPRING FRICTION SNUBBER	0.04	0.10	0.13	0.15	0.17	0.19	0.21	0.23
TRUCK SPRING SHIM, WOOD	0.00	0.00	0.01	0.01	0.02	0.03	0.03	0.04
MANUFACTURED MATERIAL (TRUCK)	0.78	1.38	1.79	2.09	2.37	2.65	2.95	3.21
TRUCK TOTAL	152.67	307.44	439.58	561.42	681.21	794.85	919.00	1027.20

5.2.3 Data Required

This subsection discusses the data used to calculate the information presented earlier in Tables 5-1 and 5-2. Subsequently, the two-step procedure for these calculations is then described in paragraphs 5.2.4 and 5.2.5.

Car Repair Data

1. Car Repair Billing (CRB) Exchange Tapes -the consolidated repair billings for a railroad's cars off line. These tapes are universally available on a monthly basis through the AAR Mechanical Inspection Department. These records form the basis for billing a railroad for repairs done off line.

2. Car Repair Billing Formatted Tapes - on line repairs for a railroad's cars. These records are not universally available on all railroads. They are often available for the purpose of monitoring one-spot performance or car-type performance. Car lines have no on line repairs and have the necessary data available to them.

3. Heavy or Programmed Repair Records -these records are not universally available on all railroads or car lines. These records contain a detailed description of repairs by car for all those repairs which do not qualify as "light" car repairs under the AAR definition of approximately 20 labor hours maximum. If these records are not available, it is not especially significant. These repairs are normally done relatively late in the life of a car and do not dramatically affect the results. The records are often available for the purpose of monitoring heavy repair facility performance or car type performance.

Car Descriptions

The Universal Machine Language Equipment Register (UMLER) - this file, organized by car initial and number, is available at any railroad or car line. It is required as part of the AAR CRB system. It was used to obtain car age and car type for this analysis.

Car Mileage Data

1. The AAR Per Diem Reporting System Tapes - monthly car mileages by car by railroad for cars off line. However, in the case of car lines, only loaded car movements are reported. They form the basis for paying a railroad for the use of its cars off its line. Usually it is possible to find summaries of these records by AAR car type, apparently for the purpose of reporting them to the AAR.

2. On Line Car Mileage For System Cars -monthly car mileages by car for cars on line. In most cases, these records are available for the purpose of paying state taxes. Usually it is possible to find summaries of these records by AAR car type. This category does not affect car lines.

5.2.4 First Procedure

The first step is to summarize the car repair data by car by year into a more manageable set of data. In doing this, it becomes necessary to "recode" the job codes so that there are less codes. In effect, hundreds of records are compressed into one record indicating numbers of repairs over the course of a year.

Calculating Summary Tables by Car

1. Read in the next car repair record. A typical car repair record format is shown in Figure 5-2.

2. Check if the current summary record (format shown in Figure 5-3) is for the right car and year by comparing the car initial and number from the two records and the year of the repair (column 31 from Figure 5-2) with the year in the summary record (column 11 from Figure 5-3). If they are not the same, find the right summary record.

3. Extract the job code (column 58 from Figure 5-2) and look up the corresponding summary record "row" number from the recode table (Table 5-3). Each summary record "row" number is some group of job codes. For example, row 1 is Clean, Oil, Test and Stencil (COT & S) air brakes. There are a large number of job codes depending on the type and number of air brakes.

4. Extract the quantity field (column 45 from Figure 5-2) or use 1 for the quantity as indicated in Table 5-3. An "R" in the "count" column in Table 5-3 indicates that records were counted (i.e., just use 1 for the quantity). A "Q" for quantity indicates the quantity field should be used.

5. Add the quantity into the summary record for the recoded job code found in step 3.

6. Update the month of repair fields (columns 13 and 15 in Figure 5-3) by extracting the month of repair from the car repair record (column 33 in Figure 5-2).

Calculating Average Prices

1. Extract the responsibility code (column 64 from Figure 5-2). If it is not owner responsible (code 1) or if the repair is for a different year, go back to step 1 in the above paragraph. If the responsibility for the repair is not the owner's, the repair is not billed and there are no charges in the labor and material charge fields.

2. Extract the labor and material charges (column 84 from Figure 5-2). Include the sign of the material charges (column 97 from Figure 5-2).

3. Add the labor and materials charges into the price tables (see the average price shown in Table 5-3). Also, add the quantity into the price tables. Then go back to step 1 in the above paragraph until all input records are processed.

TABLE 5-3. RECODE TABLE AND AVERAGE PRICES

ROW NUM	DESCRIPTION	COUNT	JOB CODES	AVERAGE PRICE
1	COT & S	R	1000-1116	217.06
2	IDT & S	R	1140-1144	26.30
3	Air Brakes & Parts	R	1160-1628	20.07
4	Hand Brakes, Geared & Non-Geared	R	1856-1980	67.62
5	Coupler Body, Type E	R	2000-2049	193.90
6	Coupler Body, Type E/F	R	2180-2189	342.83
7	Coupler Body, Type F	R	2200-2243	377.16
8	Coupler Knuckles	R	2051-2058	44.75
8	Coupler Knuckles	R	2252-2254	44.75
9	Other Coupler Parts	R	2060-2179	13.99
9	Other Coupler Parts	R	2256-2276	13.99
10	Yokes, Type E	R	2300-2328	140.29
11	Yokes, Type E/F and F	R	2350-2366	149.48
12	Draft Gears, Carriers, and Followers	R	2400-2468	287.68
13	TOFC Indication (Lubricate Hitch or Stanchion)	R	2570-2570	64.81
13	TOFC Indicator (Bridge Plates)	R	5600-5628	64.81
14	Other	R	4000-4098	31.46
14	Other	R	4100-4449	31.46
14	Other	R	4489-4598	31.46
14	Other	R	4600-4799	31.46
14	Other	R	4825-4998	31.46
14	Other	R	5000-5599	31.46
14	Other	R	5629-6998	31.46
15	Manufactured Material (Brakes)	R	1999-1999	103.14
15	Manufactured Material (Couplers)	R	2999-2999	103.14
15	Manufactured Materials	R	4099-4099	103.14
15	Miscellaneous Labor	R	4450-4488	103.14
15	Manufactured Materials	R	4559-4559	103.14
15	Manufactured Materials	R	4999-4999	103.14
15	Manufactured Materials	R	6999-6999	103.14
16	Welding	R	4800-4824	25.41
17	Non Billables	R	9900-9999	6.31
21	Brake Beams	R	1640-1676	128.18
22	Brake Head Wear Plates	Q	1692-1692	1.81
23	Brake Beam Wear Plates	Q	1696-1696	10.58
24	Brake Beam Hangers	Q	1708-1708	11.00
25	Brake Hanger Bracket Wear Plate	Q	1720-1720	4.36
26	Brake Hanger Bracket Wear Plate Securement	Q	1724-1724	2.00
27	Brake Hanger or Connection Pin	Q	1742-1742	7.54
28	Bottom Rod Safety Support	Q	1764-1768	26.54
29	Brake Beam Safety Support	Q	1772-1776	40.67
30	Brake Connection, Bottom	Q	1792-1792	24.51
31	Brake Connection, Top	Q	1796-1796	16.78
32	Brake Lever	Q	1800-1800	13.48
33	Brake Lever Guide or Carrier	Q	1804-1804	14.69
34	Dead Lever Guide	Q	1808-1808	17.38
35	Dead Lever Guide Bracket	Q	1812-1812	19.25
36	Brake Shoes	Q	1828-1840	11.87
37	Brake Shoe Keys	Q	1852-1852	1.72
38	Lubricators	R	2500-2508	7.50
39	Repack Journal Box	R	2520-2528	39.19
40	Lubricate Roller Bearings (4-Wheel Trucks)	R	2550-2550	23.16
41	Lubricate Roller Bearings (One Wheelset)	Q	2552-2552	5.79
42	Lubricate Roller Bearings (6 & 8 Wheel Trucks)	R	2554-2558	45.58
43	Journals	R	2600-2652	21.78
44	Journal Wedges	R	2670-2722	11.17
45	Journal Stops	Q	2730-2730	9.69
46	Journal Box Lids	Q	2750-2774	14.99
47	Journal Box Seals	Q	2778-2790	2.83
48	Journal Box Dust Guards	Q	2794-2794	2.16
49	Roller Bearings	R	2800-2853	43.60
50	Roller Bearing Cap Screws	R	2856-2860	3.41
51	Roller Bearing Locking Plates	R	2864-2864	9.72
52	Roller Bearing Lubrication Fitting	R	2868-2868	2.59
53	Pedestal Adapters	R	2870-2878	26.09
54	Wheels	R	3005-3125	96.08
55	Wheel Labor (Plain Bearings)	R	3150-3150	173.51
56	Wheel Labor (Roller Bearings)	R	3160-3160	216.95
57	Wheel Labor (Turned Under Truck)	R	3170-3170	0.00
58	Wheel Labor (Assembly by Owner)	R	3180-3180	80.03
59	Axles, Plain Bearings	R	3200-3242	0.20
60	Axles, Roller Bearings	R	3250-3288	6.61
61	Truck Boslters	R	3500-3554	978.53
62	Truck Bolsters (Repaired)	R	3556-3556	285.15
63	Center Pins	R	3560-3560	32.00
64	Center Plates	R	3564-3564	148.81
65	Center Plate Liners	R	3568-3568	24.41
66	Truck Side Bearings	Q	3572-3580	21.36
67	Stabilizer Friction Casting (Ride Control Truck)	Q	3582-3582	68.14
68	Stabilizer Friction Casting (Stabilized Truck)	Q	3584-3584	35.56
69	Side Bearing Shim	R	3588-3588	34.33
70	Side Frames	R	3700-3768	683.00
71	Side Frames (Repaired)	R	3772-3772	223.23
72	Journal Boxes	R	3776-3796	60.40
73	Spring Planks	R	3850-3858	66.68
74	Spring Planks (Repaired)	R	3862-3862	46.78
75	Outer Springs	Q	3900-3914	11.63
76	Inner Springs	Q	3916-3934	7.60
77	Stabilizing Spring	Q	3940-3940	18.05
78	Truck Spring Package	Q	3948-3948	0.00
79	Truck Spring Friction Snubber	Q	3952-3952	51.50
80	Truck Spring Plate	Q	3956-3956	6.15
81	Truck Spring Shim, Wood	Q	3960-3960	7.56
82	Steel	R	3964-3968	26.45
83	Truck Manufactured Materials	R	3999-3999	14.09

Notes: Count R = Records, Q = Quantity, M = Minutes

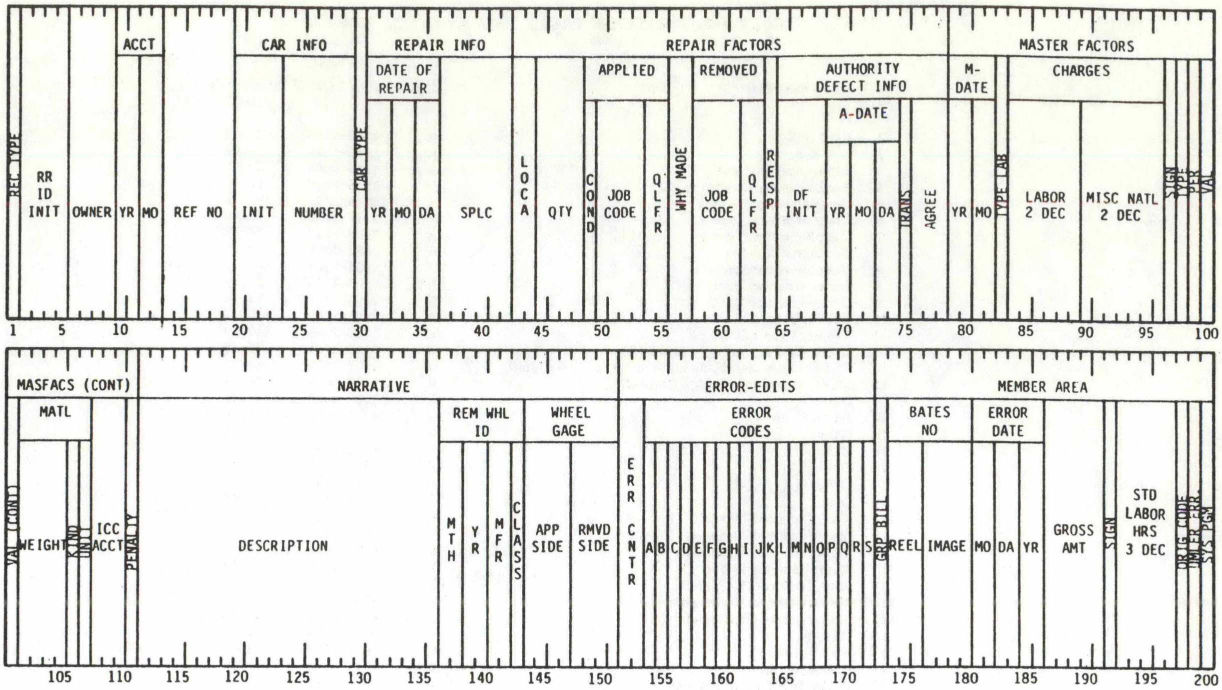
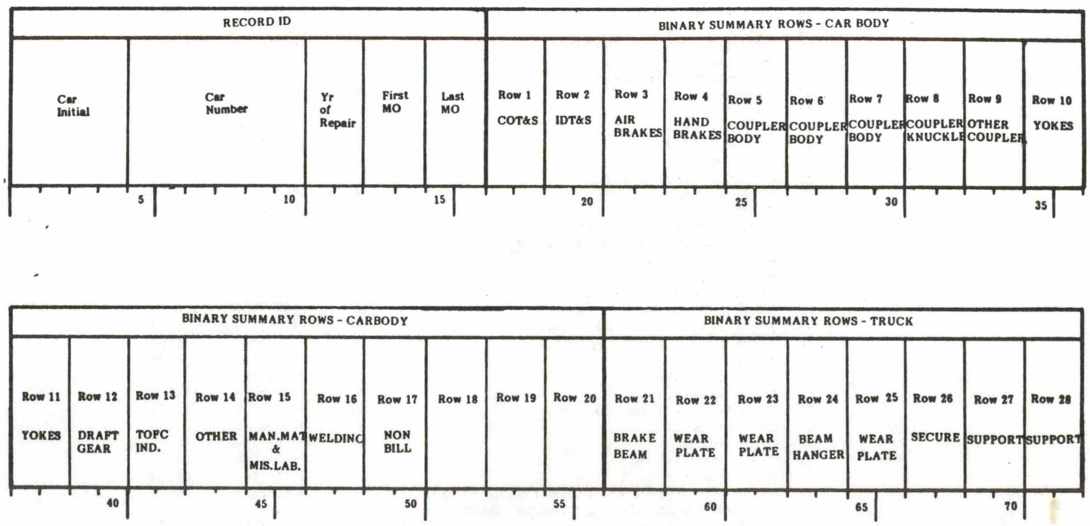


FIGURE 5-2. REPAIR RECORD FORMAT



and so on, per Table 5-3

FIGURE 5-3. SUMMARY RECORD FORMAT

5.2.5 Second Procedure

Having summarized the records so there is one record per car per year, a present-value economic analysis of the cost of car maintenance can be done as described below.

Calculating Cost by Cumulative Mileage Tables

1. Read in the UMLER record for the next car. A typical UMLER record format is shown in Figure 5-4.

2. Check the format of the record by extracting the record code (column 1 in Figure 5-4). The UMLER record formats change for different types of cars. For the subsequent discussion, code 1 formats will be used.

3. Extract the bearing type for the car from column 59, second row in Figure 5-4. If the car is not equipped with roller bearings (i.e., not "R"), go back to step 1.

At this point, cars can be selected on a number of criteria. For example, only 100-ton cars could be used. Other selections might be made on the basis of AAR code, nominal capacity, bearing type, truck center spacing, mechanical class, and annual mileage.

4. Extract the year and month the car was built (columns 22 and 28 in Figure 5-4) for use later at step 7. Also extract the car's annual mileage (column 73, fifth row in Figure 5-4) for use later at steps 7 and 8. Annual mileage is not usually in the UMLER record. It must be estimated and merged into the UMLER record if it is not available.

5. Read in the next year of car repair summary data for the car selected (Figure 5-3).

6. Check to see if the car was in service for this year by checking the month of repair fields (columns 13 and 15 in Figure 5-3). If there were no records for the car (i.e., the dates were not filled in), chances are it was not in service. If this is the case go back to step 5.

7. Extract the year of repair from the summary record (column 11 in Figure 5-3). Assume the repairs were done about midyear and calculate the age of the cars at the time of repairs (care should be taken to avoid a negative number, as could happen if repair were assumed to have been done in July for a car introduced in September). Multiply by the annual mileage to estimate the cummulative miles that the car has traveled.

8. Find the row in Table 5-4 (actually only a small piece of the table is shown; there is a column for each recoded job code) corresponding to this cummulative mileage. Add all the repairs in Table 5-4 for the particular car and year. Also, add the car's annual mileage into the miles column. Go back to step 5 until all years of data are processed. Then go back to step 1 until all cars are processed.

9. Read in the average price of each recoded job code. Multiply the price by the number of repairs to get the total expenditure by all cars of that cummulative mileage. Divide by the miles column in Table 5-4 and multiply by 100 (dollars to cents) to get the cents/mile spent on each class of repair.

10. Extrapolate the data to 1.2 million miles total life by averaging the last 10 non-zero rows. If there are not many cars to work with, it may be necessary to interpolate some of the lower cumulative mileage categories (wherever 0 miles are seen in the cumulative mileage category).

11. Build new recoded job codes by adding the existing "rows" together per the instructions in Table 5-5.

Calculating Present Cost Tables

To convert a column of Table 5-4 to present cost, proceed as follows:

1. Assume an annual mileage (e.g., 25,000 miles/year).

2. For this annual mileage, each row in Table 5-4 corresponds to one year's service. Multiply all the numbers in the column from Table 5-4 by the annual mileage (25,000 miles/year) and divide by 100 to convert from cents to dollars. This gives the estimated cost of that class of repairs for each year of the car's life.

3. Divide by $1/(1+r)^n$ where r is the discount rate (0.10) and n is the year for which the repair is being estimated (the number of the row in the table). This gives the discounted present cost of the future repairs at the time of purchase.

4. Add all the discounted present costs for each year. Stop either when you reach the end of the table (the car has gone 1.2 million miles) or when the car reaches 30 years of age, whichever comes first. In the case of the example of 25,000 miles/year, stop at row 30 with 750,000 cummulative miles. Table 5-6 illustrates the relationship between summary "rows" in the final table and the recoded job codes in Table 5-4.

Go back to step 1 until all mileage numbers and all columns in Table 5-4 are converted.

5.2.6 Problems Encountered

Not all repairs are costed in the CRB records. For example, repairs for which the handling line is responsible are reported but not charged. This was addressed as indicated in the first procedure by only computing average prices for owner responsible repairs.

Cars tend to be renumbered. This problem caused significant numbers of records to be ignored, although there is no reason to believe they would be significantly different than the records that were used. Step 6 in the second procedure tends to alleviate the problem.

The records for a given month contain repairs done in previous months. This is handled by using the date of repair field in step 2 of the first procedure and making more than one year's summary file (Figure 5-3) available to the first routine.

Certain rebilling records in the CRB tapes are not identified by car number. These were dropped from this analysis when car number 0 could not be found. This could be handled by creating a dummy car number 0 so that the records could keep track of it. Inspection of these records suggests that not a large sum of money is involved.

5.2.7 Remarks

The volume of data handled in performing these calculations is extremely large. For this reason, it is absolutely essential that some orderly procedure for doing the I/O be adopted. In this study, the UMLER file and car-by-car summary files have been kept in the same order. The files were then accessed using a hashing procedure in the order that the data appeared on the CRB tapes. An equivalent procedure would be to sort the CRB tapes into car number order and merge it with the UMLER data.

The interpolation routine used at step 10 of the second procedure was as follows. The table for all available cars was built, which did not require any interpolation. Then these tables (see Table 5-4) were saved on disk and used to interpolate cases where there was a problem. The table currently being built was called A and the one for all cars B. At each column, all the rows that have non-zero mileage in A were added for both A and B. The ratio of these two sums was multiplied times the value in B to interpolate A.

Certain car repairs (e.g., COT&S) are done based on dates rather than mileage. Theoretically, the results would be more accurate if these repairs were tracked on the basis of car age rather than cumulative mileage. Since this does not affect truck results, it has been ignored here.

There is clearly a difference in relative price growth of labor and materials. This can be accounted for by keeping average prices for labor and materials separately and using a slightly different discount rate for them. In the tables reported here, a 3 percent difference between the rates for labor and materials has been used.

The presentation would be helped by calculating an equivalent yearly savings. This is done by summing $1/(1+r)^n$ over the life of the car and dividing the sum into the present value. Engineers find this number more intuitive than the present cost, thus Tables 5-2A and 5-2B were adjusted this way.

Throughout this discussion, it should be apparent that all mileages of cars are used to calculate repair costs. The car's annual mileage is used to determine where the car is in its life cycle. There is an implicit assumption that cars wear out due to mileage and that all cars wear out in similar ways. Appendix D illustrates results for 70-ton and 100-ton normal service and high mileage cars.

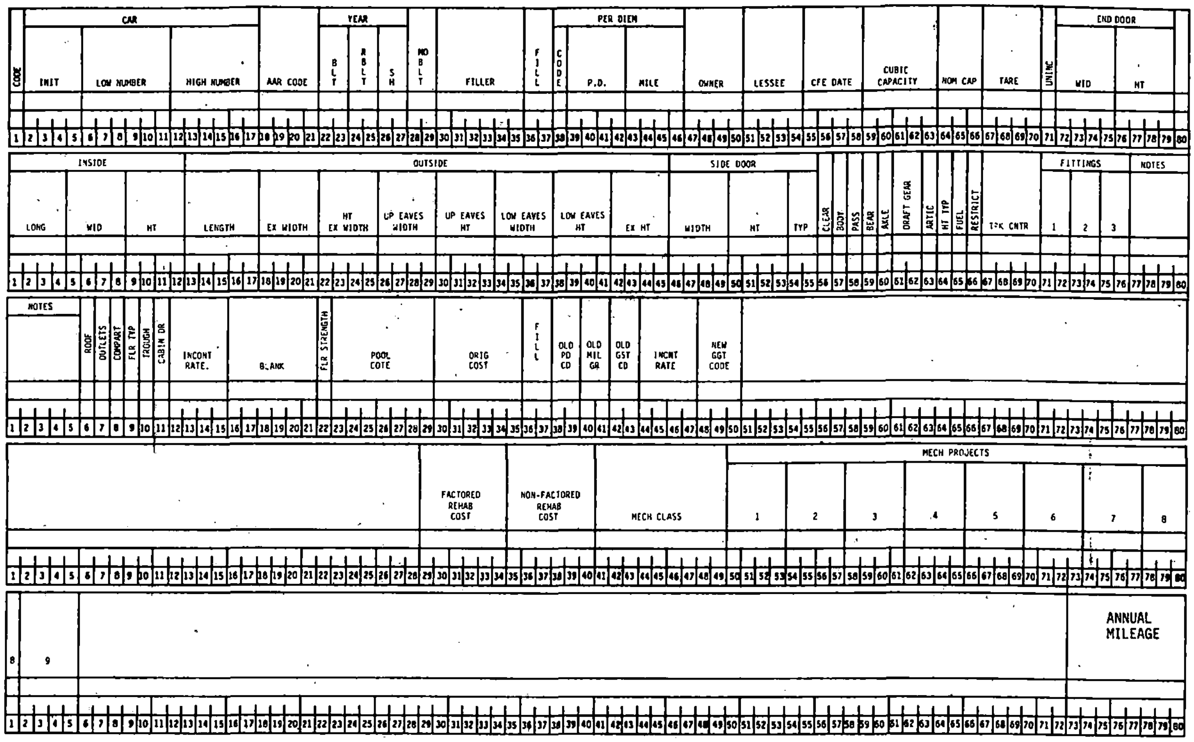


FIGURE 5-4. UMLER RECORD FORMAT, CODES 1 AND 4

TABLE 5-4. COST BY CUMULATIVE MILEAGE

CALCULATION FOR ALL ROLLER BEARING CARS

CUMULATIVE CAR MILEAGE	MILES	CAR BRAKES (CENTS/MILE)	COUPLERS YOKES DRAFT GEARS (CENTS/MILE)	MISC. LABOR MANU. MAT. (CENTS/MILE)	OTHER CAR REPAIRS (CENTS/MILE)	TRUCK BRAKES (CENTS/MILE)	WHEELSETS (CENTS/MILE)	OTHER TRUCK REPAIRS (CENTS/MILE)	TOTAL (CENTS/MILE)
1 TO 25000	7045877.	0.37887228	0.17309469	0.20200920	0.21190649	0.57969618	0.91649050	0.00773403	2.26779461
25001 TO 50000	74108288.	0.10660648	0.01766719	0.06457704	0.06325251	0.24008775	0.45164359	0.00999283	0.88925034
50001 TO 75000	88157632.	0.18045586	0.12678939	0.15010458	0.09254545	0.38842732	0.51800555	0.28722388	1.59344769
75001 TO 100000	169114320.	0.16674483	0.09288263	0.13081998	0.10310096	0.37853652	0.54976219	0.10814571	1.39917278
100001 TO 125000	194589984.	0.18737042	0.10491836	0.18964744	0.15377086	0.39154118	0.52321547	0.29069865	1.65151501
125001 TO 150000	234004656.	0.20967042	0.28826368	0.22386223	0.15520203	0.46559781	0.57074237	0.47207659	2.16155338
150001 TO 175000	196155216.	0.25806427	0.20550108	0.34582400	0.22573382	0.44616359	0.71145976	0.11893171	1.96585464
175001 TO 200000	208663744.	0.29994333	0.32638150	0.65769970	0.28914434	0.61707151	0.83955228	0.40961832	2.78171062
200001 TO 225000	206990000.	0.35616195	0.28896630	0.86746430	0.31912112	0.61351365	0.89287287	0.43569010	2.90632534
225001 TO 250000	194223072.	0.36184341	0.31471366	0.78370720	0.30221063	0.71010023	1.01766014	0.44145411	3.14798164
250001 TO 275000	193567408.	0.32503653	0.39514816	0.61894417	0.27036500	0.70999980	1.06949234	0.51781034	3.28785229
275001 TO 300000	211916336.	0.37698370	0.56212658	0.58511257	0.32038045	0.81667364	1.14392185	0.72937149	3.94945717
300001 TO 325000	181479792.	0.35089380	0.46856266	0.68369830	0.33530235	0.80346304	1.17238617	0.87429434	4.00490189
325001 TO 350000	173915328.	0.34562719	0.57792461	0.60437435	0.31138813	0.85276586	1.12841415	1.01812935	4.23424911
350001 TO 375000	124888144.	0.31473166	0.42309350	0.61782515	0.33989942	0.85555285	1.09031773	0.97650379	4.00009823
375001 TO 400000	82785520.	0.39795125	0.99661171	0.57496893	0.32062960	1.15512180	1.14609051	2.58458519	6.60099030
400001 TO 425000	55305840.	0.31967133	0.48704571	1.06411362	0.30721837	0.78259891	0.91863644	1.54778099	4.36295128
425001 TO 450000	58106768.	0.29478818	0.40578955	1.01193237	0.28005981	0.59661096	0.79741770	0.89747202	3.27213860
450001 TO 475000	64216768.	0.26620853	0.46565729	0.73608589	0.26130635	0.71387625	1.02401447	1.07463932	3.80570221
475001 TO 500000	56726912.	0.23157048	0.33886695	0.92763788	0.28738946	0.53279519	0.94021606	0.36630863	2.69714642
500001 TO 525000	33748832.	0.26772243	0.29189903	1.26400566	0.41122502	0.57979286	0.89699131	1.15551758	3.60314846
525001 TO 550000	28798944.	0.20626128	0.31431502	0.62674189	0.42115313	0.52649558	0.75296003	0.88142610	3.10261154
550001 TO 575000	14098053.	0.38583171	0.29552335	0.50772375	0.59688860	0.34894991	0.53393155	0.04899961	2.21012592
575001 TO 600000	22456416.	0.25929570	0.23958313	0.26317298	0.46792829	0.29876280	0.60139912	0.02692955	1.89389896
600001 TO 625000	4782975.	0.32113570	0.41113597	0.42049772	0.64176232	0.30284959	0.73889875	0.01789179	2.43367481
625001 TO 650000	19694864.	0.30533671	0.11251181	0.25660801	0.31791848	0.29789662	0.61166388	0.01818322	1.66351032
650001 TO 675000	8624454.	0.24923092	0.15613538	0.28940821	0.32082826	0.33732152	0.59157777	0.05390578	1.70899868
675001 TO 700000	5954129.	0.20476943	0.14066023	0.59762394	0.30179161	0.40588254	0.46465689	0.01261964	1.53038025
700001 TO 725000	16944544.	0.14032841	0.22926468	0.28121549	0.19897294	0.34983981	0.83827704	0.12571329	1.88239670
725001 TO 750000	10112060.	0.21934503	0.24390858	0.17237496	0.24093646	0.34782988	0.78116006	0.13561612	1.96879673
750001 TO 775000	19631568.	0.30459869	0.28638309	0.22381115	0.39695787	0.46414417	0.59099150	0.10522860	2.14830399
775001 TO 800000	10105596.	0.25067228	0.41058737	0.32864046	0.56156164	0.63135850	0.69080919	0.61864430	3.16363335
800001 TO 825000	3974560.	0.15876627	0.02062291	0.19462532	0.33542204	1.47351933	0.66877717	0.43057150	3.08767986
825001 TO 850000	2409353.	0.21256250	0.26453900	0.27397233	0.52724952	0.75323200	0.36248904	0.50955939	2.62963104
850001 TO 875000	EXTRAPOLATION BASED ON PREVIOUS TEN ENTRIES 0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
875001 TO 900000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
900001 TO 925000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
925001 TO 950000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
950001 TO 975000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
975001 TO 1000000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1000001 TO 1025000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1025001 TO 1050000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1050001 TO 1075000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1075001 TO 1100000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1100001 TO 1125000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1125001 TO 1150000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1150001 TO 1175000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067
1175001 TO 1200000	0.	0.23667467	0.22757494	0.30387783	0.38434017	0.53638750	0.63393021	0.20279348	2.22170067

TABLE 5-5. DEFINITIONS OF ADDITIONAL ROWS

DEFINITION OF ADDITIONAL ROWS

86	=	01+02+03+04
87	=	05+06+07+08+09+10+11+12
88	=	13+14+16+17
89	=	21+22+23+24+25+26+27+28+29+30+31+32+33+34+35+36+37
90	=	40+41+42+49+50+51+52+53+54+56+57+58+60
91	=	61+62+63+64+65+66+67+68+69+70+71+73+74+75+76+77+78+79+80+81+82+83
92	=	05+06+07
93	=	10+11
94	=	13+14
95	=	40+41+42
96	=	56+57+58
97	=	67+68
98	=	86+87+88+89+90+91

For Example: New Job Code 86 is Formed by Adding Job Codes 1 through 4.

TABLE 5-6. CORRESPONDENCE BETWEEN ROW NUMBERS AND TABLE ENTRIES

```
ROW       LABEL
NUM

00
00
-3           SUMMARY TABLE
-4
00
86 BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)
87 COUPLERS,  YOKES, & DRAFT GEAR
15 MISCELLANEOUS LABOR & MANUFACTURED MATERIAL
88 OTHER CAR REPAIRS
89 TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)
90 WHEELSETS
91 OTHER TRUCK REPAIRS
00
-1                                                  TOTAL
00
-5                                   ASSUMED CAR LIFE IN YEARS
00
00
00
-3           CAR REPAIRS:
-4
00
86 BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)
00
01      COT&S
02      IDT&S
03      PRESSURE SYSTEM
04      HAND BRAKES
00
87 COUPLERS, YOKES, & DRAFT GEAR
00
92      COUPLER BODIES
06      COUPLER KNUCKLES
09      OTHER COUPLER PARTS
93      YOKES
12      DRAFT GEARS, CARRIERS, AND FOLLOWERS
00
15 MISCELLANEOUS LABOR & MANUFACTURED MATERIAL
00
88 OTHER CAR REPAIRS
00
94      OTHER CAR REPAIRS
16      WELDING
17      NON BILLABLE INSPECTIONS
00
-1                                              CAR TOTAL
```

```
ROW       LABEL
NUM

-2
-3           TRUCK REPAIRS
-4
00
89 TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)
00
21      BRAKE BEAMS
22      BRAKE HEAD WEAR PLATES
23      BRAKE BEAM WEAR PLATES
24      BRAKE BEAM HANGERS
25      BRAKE HANGER BRACKET WEAR PLATE
26      BRAKE HANGER BRACKET WEAR PLATE SECUREMENT
27      BRAKE HANGER OR CONNECTION PIN
28      BOTTOM ROD SAFETY SUPPORT
29      BRAKE BEAM SAFETY SUPPORT
30      BRAKE CONNECTION, BOTTOM
31      BRAKE CONNECTION, TOP
32      BRAKE LEVER
33      BRAKE LEVER GUIDE OR CARRIER
34      DEAD LEVER GUIDE
35      DEAD LEVER GUIDE BRACKET
36      BRAKE SHOES
37      BRAKE SHOE KEYS
00
90 WHEELSETS
00
95      LUBRICATE ROLLER BEARINGS
49      ROLLER BEARINGS
50      ROLLER BEARING CAP SCREWS
51      ROLLER BEARING LOCKING PLATES
52      ROLLER BEARING LUBRICATION FITTING
53      PEDESTAL ADAPTERS
54      WHEELS
96      WHEEL LABOR
60      AXLES, ROLLER BEARINGS
00
00
91 OTHER TRUCK REPAIRS
00
61      TRUCK BOLSTERS
62      TRUCK BOLSTERS (REPAIRED)
63      CENTER PINS
64      CENTER PLATES
65      CENTER PLATE LINERS
66      TRUCK SIDE BEARINGS
97      FRICTION CASTINGS
69      SIDE BEARING SHIM
70      SIDE FRAMES
71      SIDE FRAMES (REPAIRED)
73      SPRING PLANKS
74      SPRING PLANKS (REPAIRED)
75      OUTER SPRINGS
76      INNER SPRINGS
77      STABILIZER SPRINGS
78      TRUCK SPRING PACKAGE
79      TRUCK SPRING FRICTION SNUBBER
80      TRUCK SPRING PLATES
81      TRUCK SPRING SHIM, WOOD
82      STEEL
83      MANUFACTURED MATERIAL (TRUCK)
00
-1                                                  TRUCK TOTAL
00

NOTES:  ROW -1 PRINTS AND CLEARS TOTALS
        ROW -2 PAGE EJECTS
        ROW -3 PRINTS "ANNUAL MILAGE"
        ROW -4 PRINTS VALUE OF MILAGE
        ROW -5 PRINTS VALUE OF CAR LIFE
```

5.3 FUEL CONSUMPTION ANALYSIS

Fuel consumption is an area where savings are possible from a Type II truck. Because a radial truck steers around curves instead of being dragged on its leading outer flange, a radial (or steering) truck significantly reduces the forces experienced during curving. This produces significant fuel savings on routes with high curve-to-tangent ratios. On the other hand, the increased weight of most of these trucks produces fuel losses on routes with low curve-to-tangent ratios. While the savings/losses involved are relatively small (on the order of 0.001 gallons/car mile), significant amounts of money can be involved for high mileage cars with today's rapidly increasing fuel costs.

The approach to measuring the rolling resistance involved the use of instrumented couplers on both ends of the test car. The effects of grade and acceleration were removed from the rolling resistance using a specially filtered, DC-coupled, longitudinal accelerometer. The train speed was maintained relatively constant during each test run, thus Davis's formulation of the rolling resistance was essentially constant over the course of a test run. The only remaining variable that was correlated with curvature was the curving force. This measured curving force was fit to a theoretical equation expressing the effect of varying curvature and off balance speed performance. Next a fuel consumption simulator was used to estimate the fuel savings per car mile expected as a function of curve-to-tangent track ratio. Finally these savings were converted to dollar savings under a variety of assumptions about the annual car mileage and real rate of increase of fuel prices (i.e., relative to the rate of inflation).

The conventional thinking is that the added rolling resistance due to curving is approximately 0.8 pound/ton/degree of curvature. This value is built into most railroads in the form of grade compensation of curves (i.e., wherever practical, the grade is reduced to compensate for the curve). This value is almost exactly the result obtained from the TDOP Phase II testing of Type I trucks. Figure 5-5 illustrates the added rolling resistance per ton at balance speed as a function of track curvature for the conventional result of 0.8 pound/ton/degree and the TDOP result of 0.68 pound/ton/degree plus .08 pound/ton/degree squared.

Fuel consumption savings are possible from Type II trucks. This is not only true of steering trucks but also true of at least one rigid truck in the test program. However, meaningful fuel savings should only be expected with routes of high curve-to-tangent track ratios using high mileage cars where the entire consist has Type II trucks.

The value of the fuel savings alone that can reasonably be expected is not high enough to warrant the added purchase price of any of these trucks under any reasonable conditions. However, if taken in conjunction with savings in car maintenance and especially in rail wear, it is possible that one or more of these trucks might be profitable.

5.3.1 Results

TDOP Phase II fuel consumption results come in two parts: measured curving resistance results and computer simulation results that estimate fuel consumption savings. The measured curving resistance data are much harder to obtain than the simuated fuel savings. It appears that previous estimates of fuel savings have been based on very limited curving resistance data. Estimates have often been made based on zero curving resistance for a steering truck.

Curving Resistance

Three steering trucks were tested as part of the program: the Barber-Scheffel, the Devine-Scales, and the Dresser DR-1. All three steering trucks significantly reduced the curving resistance of the test car compared to the Type I truck. Figure 5-6 illustrates the measured curving resistance as a function of curvature obtained for each of the three trucks and also for the Type I truck shown in Figure 5-5. As can be seen from Figure 5-6, each of the steering trucks experienced lower curving forces at every curvature in the range tested. The results for the other four (non-steering) Type II trucks tested are much closer to Type I trucks than to the steering trucks, as illustrated in Figure 5-7.

Off balance speed performance of these trucks was also estimated. The term used to estimate this performance was $Wd\ (v^2 - v_b^2)$, where W is the car weight, d is the degrees of curvature, v is the train speed, and v_b is the balance speed. Off balance speed results for the steering trucks as opposed to Type I trucks are illustrated in Figure 5-8 for a 6-degree curve with a balance speed of 35 mph. This situation is typical of the test data. Similarly, the off balance speed behavior of the other trucks in the program is illustrated in Figure 5-9. As can be seen from Figures 5-8 and 5-9, the steering trucks experienced substantially more variation than the non-steering trucks.

Tabular results for the curving resistance of the trucks tested are shown in Table 5-7. The curving resistance results shown are based on a least squares curve fit of the equation shown above the numbers. The numbers in parentheses below the estimated coefficients are standard error estimates for each coefficient. The column entitled standard error is an overall estimate for the entire equation. The column entitled R^2 is an overall "goodness of fit" estimate for the equation. An R^2 of 100 percent is a perfect fit.

While the R^2 results are not good by conventional engineering standards, this is not a conventional engineering curve fit. Normally one would expect an R^2 of at least 70 percent on a curve fit of a valid engineering equation. This curve fit is based on a statistical technique known as a "staged regression." The coefficients are fitted to the residual of a first stage regression that eliminates $F = ma$ terms from the data. Using this technique, the R^2 results obtained are rather good (e.g., 30 to 40 percent), in general.

Simulated Fuel Savings

The second set of TDOP fuel consumption results are computer simulations indicating what the changes in curving resistance mean in terms of the economic performance of freight cars. It is very difficult to do this in a generally applicable format. In the first place, there are several types of train simulators in use today. Second, there are very many possible routes that might be simulated. Each route would be expected to give different answers. Third, there are a very large number of consists that might be simulated. Each consist would be expected to give different answers.

Under these circumstances, the best that can be achieved is an indication of the savings that might reasonably be expected and an insight into the situations which might warrant further investigation on an individual basis. To accomplish this, data as a function of curve-to-tangent track ratio (defined as miles of curved track divided by miles of tangent track) and annual car mileage were used. These are both very important parameters in determining the fuel savings to be expected from the use of a premium truck. The time during which curving forces are present is directly proportional to the curve-to-tangent ratio of the route, and the annual savings associated with a given truck is in direct proportion to the annual car mileage.

The estimated savings were calculated in two parts: first, the fuel consumption simulator was used to calculate fuel consumption for the Type I and Type II trucks under each curve-to-tangent track ratio. The difference between each Type II truck and the Type I truck divided by the number of car miles represented in the simulation yields an estimate for the gallons saved per mile. These results are shown in Figure 5-10 for the steering trucks and in Figure 5-11 for the non-steering trucks. No Type I truck result is shown since the numbers are relative to a Type I truck. The savings shown are a very small percentage of the overall fuel consumption, which was in the range of 0.1 gallons/car mile. The car mile shown includes both empty and loaded miles and is for an "average" car (as opposed to 70-ton or 100-ton) at this point.

The tendency of many of the results to be negative (i.e., showing losses) at low curve-to-tangent ratios reflects the fact that most of these trucks weigh more than a conventional truck. This results in more fuel consumption when there are very few curves.

The second step taken to convert to fuel savings was to multiply by an annual mileage to convert the savings to gallons saved per year and to multiply by the price of diesel fuel per gallon to convert to dollar savings per year. The price of diesel fuel was taken as 85¢/gallon and annual mileages in 12,500 miles/year increments were estimated up to 100,000 miles/year. This result is tabulated in Tables 5-8 through 5-13 at the top of each table.

Finally, to estimate the value of this savings, it is necessary to choose a rate of return and a rate at which the price of fuel will increase relative to all other goods. A reasonable choice would be a 10 percent rate of return and a 4 percent inflation rate. In other words, it is postulated that at least a 10 percent return should be realized on an investment and that the price of diesel fuel should increase at around 16 percent next year. Further, suppose that diesel fuel will increase in price by about 4 percent more than the consumer price index for the foreseeable future. Under these conditions, the dollar value that should be used is the one under the 6 percent discount rate in Tables 5-8 through 5-13. It should be 6 percent because the price of fuel in the numerator will increase at 4 percent and the rate of return in the denominator is 10 percent, (see subsection 5.3.5, Use of Tables). Part of the two terms cancel and it is the same as a 6 percent discount rate.

Obviously this result is almost impossible to predict (since it depends strongly on the price of diesel fuel), so other discount rates are available in the tables. To help visualize these results, constant investment lines on a plot of curve-to-tangent ratio versus annual mileage are shown for the three steering trucks and one rigid truck in Figures 5-12 through 5-15. These figures are contour plots of the results in Tables 5-8 through 5-13 using the 6 percent discount rate.

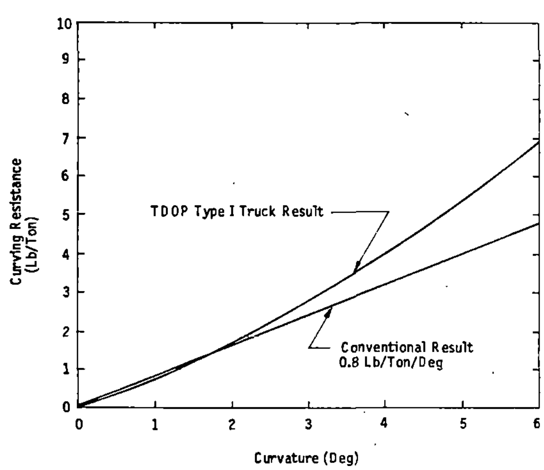


FIGURE 5-5. TYPE I TRUCK CURVING RESISTANCE VS CURVATURE

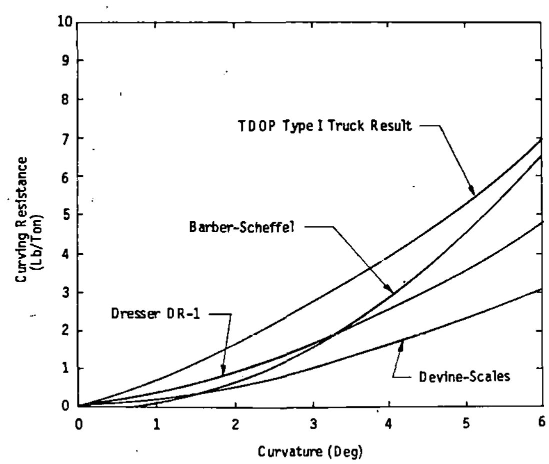


FIGURE 5-6. STEERING TRUCK CURVING RESISTANCE VS CURVATURE

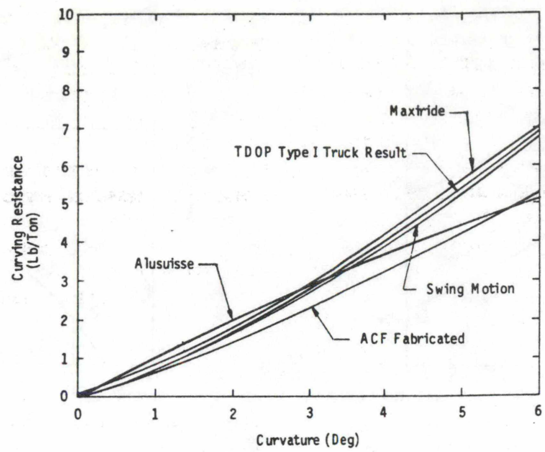


FIGURE 5-7. NON-STEERING TRUCK CURVING RESISTANCE VS CURVATURE

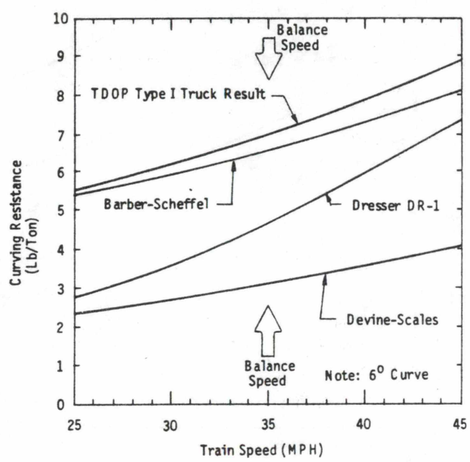


FIGURE 5-8. STEERING TRUCK CURVING RESISTANCE VS TRAIN SPEED

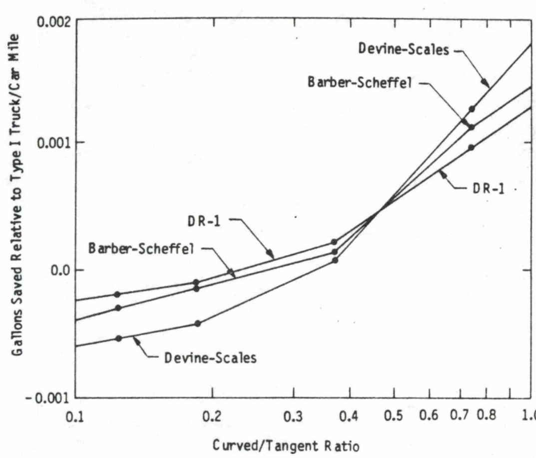


FIGURE 5-10. STEERING TRUCK FUEL SAVINGS VS CURVED/TANGENT RATIO

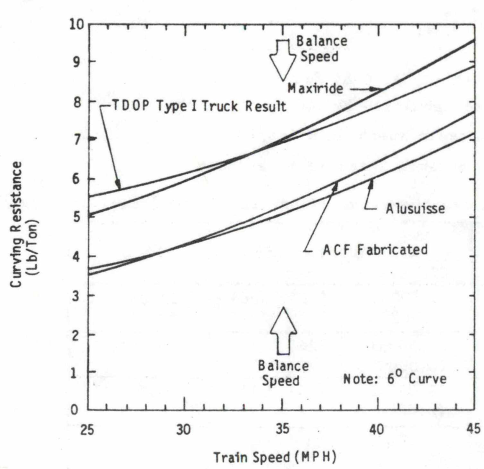


FIGURE 5-9. NON-STEERING TRUCK CURVING RESISTANCE VS TRAIN SPEED

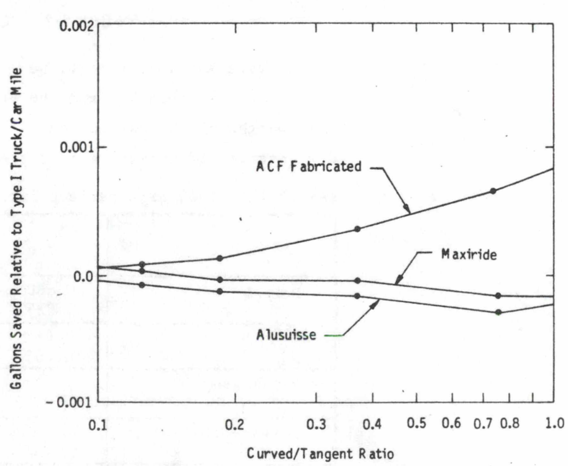


FIGURE 5-11. NON-STEERING TRUCK FUEL SAVINGS VS CURVED/TANGENT RATIO

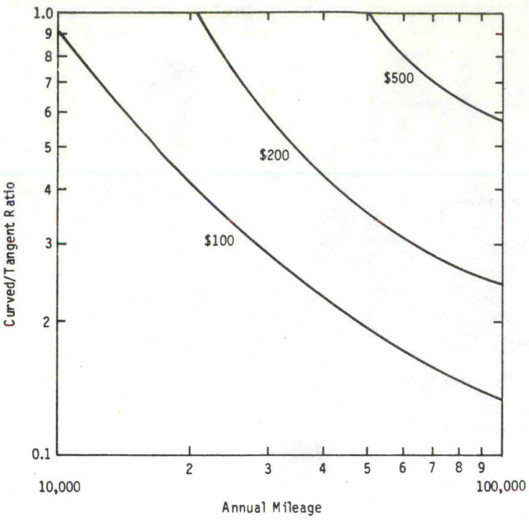


FIGURE 5-12. FUEL SAVING CONTOURS FOR THE ACF FABRICATED TRUCK

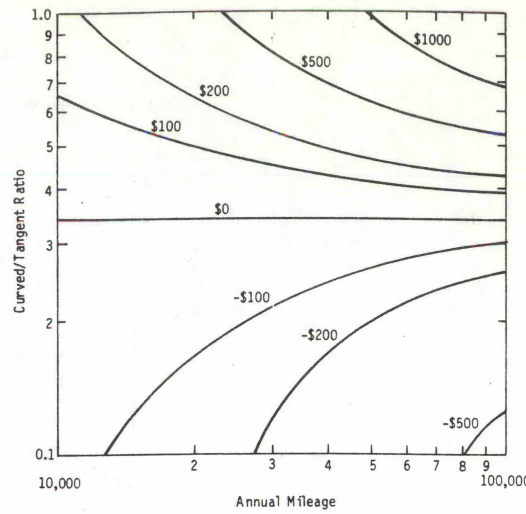


FIGURE 5-14. FUEL SAVINGS CONTOURS FOR THE DEVINE-SCALES TRUCK

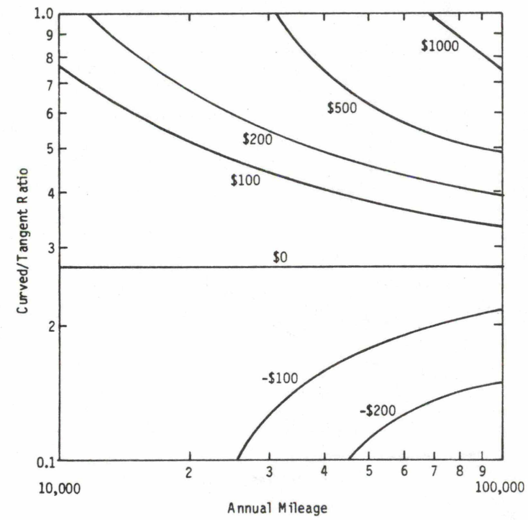


FIGURE 5-13. FUEL SAVING CONTOURS FOR THE BARBER-SCHEFFEL TRUCK

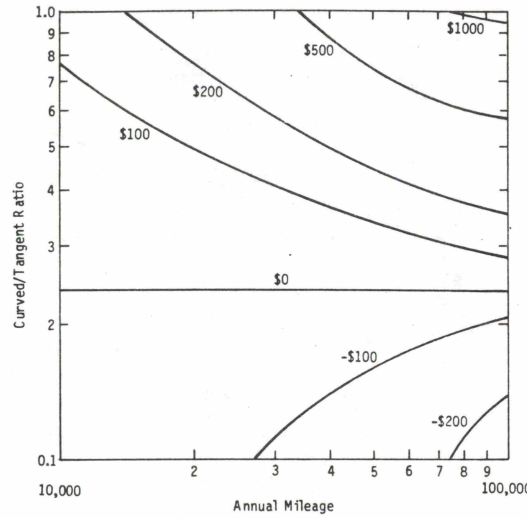


FIGURE 5-15. FUEL SAVINGS CONTOURS FOR THE DR-1 TRUCK

TABLE 5-7. CURVING COEFFICIENTS

Curving Resistance = $[b_1 Wd + b_2 Wd^2 + b_3 Wd (v^2 - v_b^2)] \pm$ St Err

Where: b_1, b_2, b_3, and the standard error are tabulated below

W = weight of the car in tons — v = train speed in miles/hour

d = degrees of curvature in degrees — v_b = balance speed in miles/hour

R^2 is a measure of goodness of fit, 100% is a perfect fit. Numbers in () are the standard errors of the individual coefficients.

	b_1 Curve	b_2 Curve Sq	b_3 Off Bal	Standard Error	R^2
Type I	0.680 (0.025)	0.081 (0.004)	0.00041 (0.00001)	(195.7 Lb)	53.0%
ACF	0.632 (0.033)	0.044 (0.006)	0.00051 (0.00001)	(244.2 Lb)	34.3%
Alusuisse	1.051 (0.043)	-0.031 (0.007)	0.00042 (0.00002)	(241.7 Lb)	34.9%
Barber-Scheffel	-0.040* (0.045)	0.191 (0.009)	0.00032 (0.00002)	(386.2 Lb)	23.0%
Devine-Scales	0.182 (0.039)	0.056 (0.007)	0.00022 (0.00002)	(295.2 Lb)	10.1%
DR-1	0.327 (0.028)	0.077 (0.005)	0.00056 (0.00001)	(228.0 Lb)	41.6%
Maxiride	0.772 (0.032)	0.067 (0.006)	0.00056 (0.00001)	(276.4 Lb)	39.9%
Swing Motion	0.673 (0.044)	0.077 (0.008)	-0.00006 (0.00001)	(297.6 Lb)	34.4%

* not statistically different from zero

TABLE 5-8. FUEL SAVINGS FOR ACF FABRICATED TRUCK

DOLLARS/YEAR BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	1.05	2.10	3.15	4.21	5.26	6.31	7.36	8.41
CURVE/TANGENT= 0.185	1.69	3.38	5.07	6.76	8.45	10.14	11.83	13.52
CURVE/TANGENT= 0.368	4.00	8.01	12.01	16.02	20.02	24.02	28.03	32.03
CURVE/TANGENT= 0.737	7.30	14.60	21.90	29.20	36.50	43.80	51.10	58.40
CURVE/TANGENT= 1.105	9.13	18.26	27.39	36.52	45.65	54.78	63.91	73.04

DOLLAR VALUE AT 10% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	9.91	19.82	29.73	39.64	49.25	56.67	63.65	68.98
CURVE/TANGENT= 0.185	15.93	31.85	47.78	63.70	79.15	91.07	102.28	110.85
CURVE/TANGENT= 0.368	37.74	75.49	113.23	150.98	187.58	215.84	242.40	262.70
CURVE/TANGENT= 0.737	68.82	137.63	206.45	275.27	341.99	393.53	441.95	478.96
CURVE/TANGENT= 1.105	86.07	172.14	258.21	344.28	427.73	492.20	552.75	599.05

DOLLAR VALUE AT 8% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	11.84	23.67	35.51	47.34	58.65	66.41	73.71	78.82
CURVE/TANGENT= 0.185	19.02	38.04	57.06	76.08	94.26	106.73	118.46	126.67
CURVE/TANGENT= 0.368	45.08	90.15	135.23	180.30	223.39	252.94	280.75	300.19
CURVE/TANGENT= 0.737	82.18	164.36	246.55	328.73	407.28	461.16	511.86	547.32
CURVE/TANGENT= 1.105	102.79	205.57	308.36	411.15	509.40	576.78	640.19	684.54

DOLLAR VALUE AT 6% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	14.47	28.94	43.41	57.88	71.44	79.16	86.57	91.06
CURVE/TANGENT= 0.185	23.25	46.51	69.76	93.02	114.80	127.22	139.12	146.34
CURVE/TANGENT= 0.368	55.11	110.23	165.34	220.45	272.08	301.50	329.72	346.82
CURVE/TANGENT= 0.737	100.48	200.97	301.45	401.93	496.06	549.71	601.15	632.33
CURVE/TANGENT= 1.105	125.68	251.35	377.03	502.70	620.43	687.53	751.86	790.87

DOLLAR VALUE AT 4% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	18.18	36.36	54.54	72.71	89.27	96.17	103.24	106.47
CURVE/TANGENT= 0.185	29.21	58.43	87.64	116.85	143.46	154.55	165.91	171.10
CURVE/TANGENT= 0.368	69.24	138.47	207.71	276.94	340.01	366.29	393.20	405.49
CURVE/TANGENT= 0.737	126.23	252.46	378.70	504.93	619.91	667.82	716.89	739.31
CURVE/TANGENT= 1.105	157.88	315.76	473.64	631.52	775.33	835.25	896.63	924.66

DOLLAR VALUE AT 2% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	23.54	47.09	70.63	94.18	114.82	119.30	125.18	126.09
CURVE/TANGENT= 0.185	37.84	75.67	113.51	151.35	184.52	191.72	201.17	202.62
CURVE/TANGENT= 0.368	89.67	179.35	269.02	358.69	437.32	454.38	476.78	480.22
CURVE/TANGENT= 0.737	163.49	326.99	490.48	653.98	797.32	828.43	869.28	875.54
CURVE/TANGENT= 1.105	204.49	408.97	613.46	817.94	997.22	1036.13	1087.22	1095.05

DOLLAR VALUE AT 0% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	31.54	63.06	94.61	126.15	152.44	151.38	154.54	151.38
CURVE/TANGENT= 0.185	50.68	101.37	152.05	202.73	244.97	243.28	248.35	243.28
CURVE/TANGENT= 0.368	120.12	240.24	360.35	480.47	580.57	576.57	588.58	576.57
CURVE/TANGENT= 0.737	219.00	438.00	657.00	876.00	1058.51	1051.21	1073.11	1051.21
CURVE/TANGENT= 1.105	273.91	547.82	821.72	1095.63	1323.89	1314.76	1342.15	1314.76
ASSUMED CAR LIFE	30.0	30.0	30.0	30.0	28.8	24.0	20.6	18.0

TABLE 5-9. FUEL SAVINGS FOR ALUSUISSE TRUCK

DOLLARS/YEAR BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-0.89	-1.79	-2.68	-3.58	-4.47	-5.36	-6.26	-7.15
CURVE/TANGENT= 0.185	-1.58	-3.16	-4.74	-6.32	-7.90	-9.48	-11.06	-12.64
CURVE/TANGENT= 0.368	-1.97	-3.95	-5.92	-7.89	-9.87	-11.84	-13.81	-15.79
CURVE/TANGENT= 0.737	-3.21	-6.41	-9.62	-12.83	-16.03	-19.24	-22.45	-25.66
CURVE/TANGENT= 1.105	-2.60	-5.20	-7.80	-10.40	-13.00	-15.59	-18.19	-20.79

DOLLAR VALUE AT 10% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-8.43	-16.85	-25.28	-33.70	-41.87	-48.18	-54.11	-58.64
CURVE/TANGENT= 0.185	-14.90	-29.80	-44.69	-59.59	-74.04	-85.20	-95.68	-103.69
CURVE/TANGENT= 0.368	-18.60	-37.20	-55.80	-74.40	-92.44	-106.37	-119.46	-129.46
CURVE/TANGENT= 0.737	-30.23	-60.46	-90.69	-120.93	-150.24	-172.88	-194.15	-210.41
CURVE/TANGENT= 1.105	-24.50	-49.00	-73.50	-98.00	-121.76	-140.11	-157.35	-170.53

DOLLAR VALUE AT 8% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-10.06	-20.12	-30.19	-40.25	-49.87	-56.46	-62.67	-67.01
CURVE/TANGENT= 0.185	-17.79	-35.58	-53.38	-71.17	-88.17	-99.84	-110.81	-118.49
CURVE/TANGENT= 0.368	-22.21	-44.43	-66.64	-88.85	-110.09	-124.65	-138.35	-147.94
CURVE/TANGENT= 0.737	-36.10	-72.21	-108.31	-144.41	-178.92	-202.59	-224.86	-240.44
CURVE/TANGENT= 1.105	-29.26	-58.52	-87.78	-117.04	-145.01	-164.19	-182.24	-194.86

DOLLAR VALUE AT 6% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-12.30	-24.61	-36.91	-49.21	-60.74	-67.30	-73.60	-77.42
CURVE/TANGENT= 0.185	-21.75	-43.51	-65.26	-87.02	-107.39	-119.01	-130.14	-136.90
CURVE/TANGENT= 0.368	-27.16	-54.32	-81.48	-108.64	-134.08	-148.58	-162.49	-170.92
CURVE/TANGENT= 0.737	-44.14	-88.29	-132.43	-176.57	-217.92	-241.49	-264.08	-277.79
CURVE/TANGENT= 1.105	-35.78	-71.55	-107.33	-143.10	-176.61	-195.71	-214.03	-225.13

DOLLAR VALUE AT 4% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-15.46	-30.91	-46.37	-61.82	-75.90	-81.77	-87.77	-90.52
CURVE/TANGENT= 0.185	-27.33	-54.66	-81.98	-109.31	-134.21	-144.58	-155.20	-160.05
CURVE/TANGENT= 0.368	-34.12	-68.24	-102.36	-136.48	-167.56	-180.51	-193.77	-199.83
CURVE/TANGENT= 0.737	-55.45	-110.91	-166.36	-221.82	-272.33	-293.37	-314.93	-324.78
CURVE/TANGENT= 1.105	-44.94	-89.89	-134.83	-179.77	-220.71	-237.77	-255.24	-263.22

DOLLAR VALUE AT 2% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-20.02	-40.04	-60.05	-80.07	-97.62	-101.43	-106.43	-107.20
CURVE/TANGENT= 0.185	-35.40	-70.79	-106.19	-141.58	-172.61	-179.35	-188.19	-189.55
CURVE/TANGENT= 0.368	-44.19	-88.38	-132.57	-176.77	-215.51	-223.92	-234.96	-236.65
CURVE/TANGENT= 0.737	-71.82	-143.65	-215.47	-287.29	-350.27	-363.93	-381.87	-384.63
CURVE/TANGENT= 1.105	-58.21	-116.42	-174.63	-232.84	-283.87	-294.95	-309.49	-311.72

DOLLAR VALUE AT 0% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-26.81	-53.63	-80.44	-107.26	-129.60	-128.71	-131.39	-128.71
CURVE/TANGENT= 0.185	-47.41	-94.82	-142.24	-189.65	-229.16	-227.58	-232.32	-227.58
CURVE/TANGENT= 0.368	-59.19	-118.39	-177.58	-236.78	-286.11	-284.13	-290.05	-284.13
CURVE/TANGENT= 0.737	-96.21	-192.42	-288.62	-384.83	-465.00	-461.80	-471.42	-461.80
CURVE/TANGENT= 1.105	-77.97	-155.94	-233.92	-311.89	-376.86	-374.26	-382.06	-374.26
ASSUMED CAR LIFE	30.0	30.0	30.0	30.0	28.8	24.0	20.6	18.0

TABLE 5-10. FUEL SAVINGS FOR BARBER-SCHEFFEL TRUCK

DOLLARS/YEAR BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-3.34	-6.69	-10.03	-13.38	-16.72	-20.07	-23.41	-26.76
CURVE/TANGENT= 0.185	-1.79	-3.58	-5.37	-7.16	-8.95	-10.74	-12.53	-14.32
CURVE/TANGENT= 0.368	1.64	3.28	4.92	6.55	8.19	9.83	11.47	13.11
CURVE/TANGENT= 0.737	11.98	23.96	35.94	47.92	59.90	71.88	83.86	95.84
CURVE/TANGENT= 1.105	16.72	33.44	50.15	66.87	83.59	100.31	117.02	133.74

DOLLAR VALUE AT 10% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-31.53	-63.06	-94.60	-126.13	-156.70	-180.32	-202.50	-219.46
CURVE/TANGENT= 0.185	-16.87	-33.74	-50.62	-67.49	-83.85	-96.49	-108.36	-117.43
CURVE/TANGENT= 0.368	15.45	30.89	46.34	61.79	76.77	88.34	99.21	107.51
CURVE/TANGENT= 0.737	112.93	225.87	338.80	451.73	561.23	645.82	725.27	786.02
CURVE/TANGENT= 1.105	157.59	315.19	472.78	630.38	783.18	901.21	1012.09	1096.85

DOLLAR VALUE AT 8% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-37.66	-75.31	-112.97	-150.62	-186.62	-211.30	-234.53	-250.78
CURVE/TANGENT= 0.185	-20.15	-40.30	-60.45	-80.60	-99.86	-113.07	-125.50	-134.19
CURVE/TANGENT= 0.368	18.45	36.90	55.34	73.79	91.42	103.52	114.90	122.86
CURVE/TANGENT= 0.737	134.87	269.73	404.60	539.47	668.38	756.80	840.00	898.20
CURVE/TANGENT= 1.105	188.20	376.40	564.61	752.81	932.70	1056.09	1172.19	1253.40

DOLLAR VALUE AT 6% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-46.04	-92.08	-138.12	-184.17	-227.30	-251.88	-275.44	-289.74
CURVE/TANGENT= 0.185	-24.64	-49.27	-73.91	-98.55	-121.62	-134.78	-147.39	-155.03
CURVE/TANGENT= 0.368	22.56	45.11	67.67	90.22	111.35	123.39	134.94	141.94
CURVE/TANGENT= 0.737	164.90	329.80	494.70	659.60	814.07	902.11	986.53	1037.71
CURVE/TANGENT= 1.105	230.11	460.23	690.34	920.45	1136.01	1258.86	1376.66	1448.08

DOLLAR VALUE AT 4% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-57.84	-115.68	-173.52	-231.36	-284.04	-305.99	-328.48	-338.75
CURVE/TANGENT= 0.185	-30.95	-61.90	-92.85	-123.80	-151.99	-163.73	-175.77	-181.26
CURVE/TANGENT= 0.368	28.34	56.67	85.01	113.34	139.15	149.91	160.92	165.95
CURVE/TANGENT= 0.737	207.16	414.31	621.47	828.63	1017.32	1095.94	1176.48	1213.26
CURVE/TANGENT= 1.105	289.08	578.16	867.24	1156.32	1419.63	1529.35	1641.73	1693.06

DOLLAR VALUE AT 2% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-74.91	-149.83	-224.74	-299.65	-365.33	-379.59	-398.30	-401.17
CURVE/TANGENT= 0.185	-40.09	-80.17	-120.26	-160.34	-195.49	-203.11	-213.13	-214.66
CURVE/TANGENT= 0.368	36.70	73.40	110.10	146.80	178.98	185.96	195.13	196.53
CURVE/TANGENT= 0.737	268.31	536.61	804.92	1073.23	1308.47	1359.52	1426.55	1436.82
CURVE/TANGENT= 1.105	374.41	748.82	1123.24	1497.65	1825.92	1897.16	1990.69	2005.03

DOLLAR VALUE AT 0% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-100.35	-200.69	-301.04	-401.39	-485.01	-481.66	-491.70	-481.66
CURVE/TANGENT= 0.185	-53.69	-107.39	-161.08	-214.78	-259.52	-257.73	-263.10	-257.73
CURVE/TANGENT= 0.368	49.16	98.32	147.48	196.64	237.61	235.97	240.88	235.97
CURVE/TANGENT= 0.737	359.40	718.79	1078.19	1437.59	1737.09	1725.11	1761.05	1725.11
CURVE/TANGENT= 1.105	501.53	1003.05	1504.58	2006.10	2424.04	2407.32	2457.48	2407.32
ASSUMED CAR LIFE	30.0	30.0	30.0	30.0	28.8	24.0	20.6	18.0

TABLE 5-11. FUEL SAVINGS FOR DEVINE-SCALES TRUCK

DOLLARS/YEAR BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-5.80	-11.59	-17.39	-23.18	-28.98	-34.78	-40.57	-46.37
CURVE/TANGENT= 0.185	-4.31	-8.62	-12.93	-17.24	-21.55	-25.86	-30.17	-34.48
CURVE/TANGENT= 0.368	0.97	1.93	2.90	3.87	4.83	5.80	6.76	7.73
CURVE/TANGENT= 0.737	13.35	26.69	40.04	53.39	66.73	80.08	93.42	106.77
CURVE/TANGENT= 1.105	20.79	41.57	62.36	83.14	103.93	124.72	145.50	166.29

DOLLAR VALUE AT 10% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-54.64	-109.28	-163.92	-218.56	-271.53	-312.46	-350.90	-380.29
CURVE/TANGENT= 0.185	-40.63	-81.25	-121.88	-162.50	-201.90	-232.32	-260.91	-282.76
CURVE/TANGENT= 0.368	9.11	18.22	27.33	36.44	45.27	52.09	58.50	63.40
CURVE/TANGENT= 0.737	125.81	251.63	377.44	503.26	625.25	719.48	808.00	875.67
CURVE/TANGENT= 1.105	195.95	391.89	587.84	783.78	973.77	1120.53	1258.39	1363.78

DOLLAR VALUE AT 8% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-65.25	-130.50	-195.75	-261.00	-323.37	-366.15	-406.41	-434.56
CURVE/TANGENT= 0.185	-48.52	-97.03	-145.55	-194.07	-240.44	-272.25	-302.18	-323.11
CURVE/TANGENT= 0.368	10.88	21.76	32.64	43.51	53.91	61.04	67.76	72.45
CURVE/TANGENT= 0.737	150.25	300.50	450.75	601.00	744.62	843.12	935.82	1000.64
CURVE/TANGENT= 1.105	234.00	468.00	702.01	936.01	1159.68	1313.09	1457.45	1558.42

DOLLAR VALUE AT 6% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-79.78	-159.56	-239.34	-319.13	-393.86	-436.45	-477.30	-502.06
CURVE/TANGENT= 0.185	-59.32	-118.64	-177.96	-237.28	-292.85	-324.52	-354.89	-373.30
CURVE/TANGENT= 0.368	13.30	26.60	39.90	53.20	65.66	72.76	79.57	83.70
CURVE/TANGENT= 0.737	183.71	367.42	551.13	734.84	906.93	1005.01	1099.05	1156.07
CURVE/TANGENT= 1.105	286.11	572.22	858.34	1144.45	1412.47	1565.21	1711.68	1800.48

DOLLAR VALUE AT 4% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-100.23	-200.45	-300.68	-400.90	-492.19	-530.23	-569.20	-586.99
CURVE/TANGENT= 0.185	-74.52	-149.04	-223.57	-298.09	-365.96	-394.25	-423.22	-436.45
CURVE/TANGENT= 0.368	16.71	33.42	50.13	66.84	82.06	88.40	94.90	97.86
CURVE/TANGENT= 0.737	230.79	461.57	692.36	923.14	1133.35	1220.95	1310.67	1351.64
CURVE/TANGENT= 1.105	359.43	718.86	1078.29	1437.72	1765.10	1901.52	2041.25	2105.07

DOLLAR VALUE AT 2% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-129.81	-259.62	-389.43	-519.24	-633.06	-657.76	-690.19	-695.16
CURVE/TANGENT= 0.185	-96.52	-193.04	-289.56	-386.08	-470.70	-489.07	-513.18	-516.88
CURVE/TANGENT= 0.368	21.64	43.28	64.93	86.57	105.54	109.66	115.07	115.90
CURVE/TANGENT= 0.737	298.91	597.82	896.73	1195.64	1457.71	1514.59	1589.26	1600.71
CURVE/TANGENT= 1.105	465.53	931.06	1396.58	1862.11	2270.26	2358.85	2475.14	2492.97

DOLLAR VALUE AT 0% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-173.88	-347.76	-521.65	-695.53	-840.43	-834.63	-852.02	-834.63
CURVE/TANGENT= 0.185	-129.29	-258.58	-387.86	-517.15	-624.89	-620.58	-633.51	-620.58
CURVE/TANGENT= 0.368	28.99	57.98	86.97	115.96	140.11	139.15	142.05	139.15
CURVE/TANGENT= 0.737	400.39	800.78	1201.17	1601.56	1935.22	1921.88	1961.92	1921.88
CURVE/TANGENT= 1.105	623.58	1247.15	1870.73	2494.30	3013.95	2993.16	3055.52	2993.16
ASSUMED CAR LIFE	30.0	30.0	30.0	30.0	28.8	24.0	20.6	18.0

TABLE 5-12. FUEL SAVINGS FOR DRESSER DR-1 TRUCK

DOLLARS/YEAR BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-2.46	-4.93	-7.39	-9.86	-12.32	-14.78	-17.25	-19.71
CURVE/TANGENT= 0.185	-1.35	-2.70	-4.05	-5.39	-6.74	-8.09	-9.44	-10.79
CURVE/TANGENT= 0.368	2.52	5.05	7.57	10.09	12.61	15.14	17.66	20.18
CURVE/TANGENT= 0.737	10.00	20.00	30.00	40.00	50.00	60.00	70.00	80.00
CURVE/TANGENT= 1.105	14.88	29.76	44.64	59.53	74.41	89.29	104.17	119.05

DOLLAR VALUE AT 10% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-23.23	-46.45	-69.68	-92.91	-115.43	-132.82	-149.16	-161.66
CURVE/TANGENT= 0.185	-12.71	-25.43	-38.14	-50.86	-63.18	-72.71	-81.65	-88.49
CURVE/TANGENT= 0.368	23.78	47.57	71.35	95.13	118.20	136.01	152.74	165.53
CURVE/TANGENT= 0.737	94.27	188.54	282.81	377.08	468.48	539.09	605.41	656.12
CURVE/TANGENT= 1.105	140.28	280.57	420.85	561.14	697.16	802.23	900.93	976.38

DOLLAR VALUE AT 8% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-27.74	-55.48	-83.21	-110.95	-137.46	-155.65	-172.76	-184.73
CURVE/TANGENT= 0.185	-15.18	-30.37	-45.55	-60.73	-75.25	-85.20	-94.57	-101.12
CURVE/TANGENT= 0.368	28.40	56.81	85.21	113.61	140.76	159.38	176.90	189.16
CURVE/TANGENT= 0.737	112.58	225.16	337.74	450.32	557.93	631.73	701.18	749.76
CURVE/TANGENT= 1.105	167.53	335.06	502.59	670.13	830.26	940.09	1043.45	1115.73

DOLLAR VALUE AT 6% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-33.91	-67.83	-101.74	-135.66	-167.43	-185.53	-202.89	-213.42
CURVE/TANGENT= 0.185	-18.56	-37.13	-55.69	-74.26	-91.65	-101.56	-111.06	-116.83
CURVE/TANGENT= 0.368	34.73	69.46	104.18	138.91	171.44	189.98	207.76	218.54
CURVE/TANGENT= 0.737	137.65	275.30	412.95	550.60	679.54	753.03	823.49	866.22
CURVE/TANGENT= 1.105	204.84	409.68	614.52	819.35	1011.24	1120.60	1225.46	1289.03

DOLLAR VALUE AT 4% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-42.60	-85.21	-127.81	-170.42	-209.23	-225.40	-241.96	-249.52
CURVE/TANGENT= 0.185	-23.32	-46.64	-69.97	-93.29	-114.53	-123.38	-132.45	-136.59
CURVE/TANGENT= 0.368	43.63	87.25	130.88	174.51	214.25	230.80	247.76	255.51
CURVE/TANGENT= 0.737	172.92	345.84	518.77	691.69	849.19	914.83	982.05	1012.75
CURVE/TANGENT= 1.105	257.33	514.66	771.99	1029.32	1263.70	1361.37	1461.41	1507.10

DOLLAR VALUE AT 2% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-55.18	-110.36	-165.54	-220.73	-269.11	-279.61	-293.39	-295.50
CURVE/TANGENT= 0.185	-30.21	-60.41	-90.62	-120.83	-147.31	-153.06	-160.60	-161.76
CURVE/TANGENT= 0.368	56.51	113.01	169.52	226.02	275.56	286.31	300.43	302.59
CURVE/TANGENT= 0.737	223.97	447.93	671.90	895.87	1092.23	1134.85	1190.80	1199.37
CURVE/TANGENT= 1.105	333.29	666.58	999.87	1333.16	1625.37	1688.79	1772.05	1784.81

DOLLAR VALUE AT 0% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	-73.92	-147.83	-221.75	-295.66	-357.26	-354.79	-362.19	-354.79
CURVE/TANGENT= 0.185	-40.46	-80.92	-121.38	-161.85	-195.56	-194.22	-198.26	-194.22
CURVE/TANGENT= 0.368	75.69	151.38	227.07	302.76	365.83	363.31	370.88	363.31
CURVE/TANGENT= 0.737	300.00	600.01	900.01	1200.01	1450.02	1440.02	1470.02	1440.02
CURVE/TANGENT= 1.105	446.44	892.88	1339.32	1785.76	2157.60	2142.92	2187.56	2142.92
ASSUMED CAR LIFE	30.0	30.0	30.0	30.0	28.8	24.0	20.6	18.0

TABLE 5-13. FUEL SAVINGS FOR MAXIRIDE TRUCK

DOLLARS/YEAR BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	0.41	0.81	1.22	1.62	2.03	2.44	2.84	3.25
CURVE/TANGENT= 0.185	-0.44	-0.88	-1.32	-1.75	-2.19	-2.63	-3.07	-3.51
CURVE/TANGENT= 0.368	-0.42	-0.85	-1.27	-1.70	-2.12	-2.55	-2.97	-3.40
CURVE/TANGENT= 0.737	-1.99	-3.97	-5.96	-7.95	-9.93	-11.92	-13.91	-15.89
CURVE/TANGENT= 1.105	-2.16	-4.33	-6.49	-8.65	-10.81	-12.98	-15.14	-17.30

DOLLAR VALUE AT 10% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	3.83	7.65	11.48	15.31	19.02	21.88	24.58	26.63
CURVE/TANGENT= 0.185	-4.13	-8.27	-12.40	-16.53	-20.54	-23.64	-26.54	-28.77
CURVE/TANGENT= 0.368	-4.01	-8.01	-12.02	-16.02	-19.91	-22.91	-25.73	-27.88
CURVE/TANGENT= 0.737	-18.73	-37.45	-56.18	-74.90	-93.06	-107.09	-120.26	-130.33
CURVE/TANGENT= 1.105	-20.39	-40.78	-61.17	-81.56	-101.33	-116.60	-130.95	-141.91

DOLLAR VALUE AT 8% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	4.57	9.14	13.71	18.28	22.65	25.64	28.46	30.44
CURVE/TANGENT= 0.185	-4.94	-9.87	-14.81	-19.74	-24.46	-27.70	-30.74	-32.87
CURVE/TANGENT= 0.368	-4.78	-9.57	-14.35	-19.14	-23.71	-26.84	-29.80	-31.86
CURVE/TANGENT= 0.737	-22.36	-44.73	-67.09	-89.45	-110.83	-125.49	-139.29	-148.93
CURVE/TANGENT= 1.105	-24.35	-48.70	-73.05	-97.40	-120.67	-136.64	-151.66	-162.17

DOLLAR VALUE AT 6% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	5.59	11.18	16.76	22.35	27.59	30.57	33.43	35.16
CURVE/TANGENT= 0.185	-6.04	-12.07	-18.11	-24.14	-29.79	-33.02	-36.11	-37.98
CURVE/TANGENT= 0.368	-5.85	-11.70	-17.55	-23.40	-28.88	-32.00	-34.99	-36.81
CURVE/TANGENT= 0.737	-27.34	-54.69	-82.03	-109.37	-134.99	-149.58	-163.58	-172.07
CURVE/TANGENT= 1.105	-29.77	-59.54	-89.32	-119.09	-146.98	-162.87	-178.11	-187.35

DOLLAR VALUE AT 4% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	7.02	14.04	21.06	28.08	34.47	37.14	39.87	41.11
CURVE/TANGENT= 0.185	-7.58	-15.16	-22.75	-30.33	-37.23	-40.11	-43.06	-44.40
CURVE/TANGENT= 0.368	-7.35	-14.70	-22.04	-29.39	-36.09	-38.87	-41.73	-43.04
CURVE/TANGENT= 0.737	-34.35	-68.70	-103.05	-137.40	-168.69	-181.72	-195.08	-201.18
CURVE/TANGENT= 1.105	-37.40	-74.80	-112.20	-149.61	-183.67	-197.87	-212.41	-219.05

DOLLAR VALUE AT 2% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	9.09	18.18	27.28	36.37	44.34	46.07	48.34	48.69
CURVE/TANGENT= 0.185	-9.82	-19.64	-29.46	-39.28	-47.89	-49.76	-52.21	-52.59
CURVE/TANGENT= 0.368	-9.52	-19.03	-28.55	-38.07	-46.41	-48.22	-50.60	-50.97
CURVE/TANGENT= 0.737	-44.49	-88.98	-133.47	-177.96	-216.96	-225.43	-236.54	-238.25
CURVE/TANGENT= 1.105	-48.44	-96.88	-145.33	-193.77	-236.24	-245.46	-257.56	-259.41

DOLLAR VALUE AT 0% DISCOUNT RATE BY ANNUAL MILEAGE

ANNUAL MILEAGE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
CURVE/TANGENT= 0.124	12.18	24.36	36.54	48.71	58.86	58.46	59.67	58.46
CURVE/TANGENT= 0.185	-13.15	-26.31	-39.46	-52.61	-63.58	-63.14	-64.45	-63.14
CURVE/TANGENT= 0.368	-12.75	-25.50	-38.24	-50.99	-61.62	-61.19	-62.47	-61.19
CURVE/TANGENT= 0.737	-59.59	-119.19	-178.78	-238.37	-288.04	-286.05	-292.01	-286.05
CURVE/TANGENT= 1.105	-64.89	-129.78	-194.66	-259.55	-313.62	-311.46	-317.95	-311.46
ASSUMED CAR LIFE	30.0	30.0	30.0	30.0	28.8	24.0	20.6	18.0

5.3.2 Instrumentation and Testing

The instrumented coupler technique was used to evaluate the change in curving resistance associated with each truck tested. Two instrumented couplers were adapted from the Full Scale Aerodynamic Test Program conducted at the Transportation Test Center in 1978 under FRA sponsorship (Reference 1). The couplers were upgraded to use load cells capable of $\pm$ 25,000 pounds with a nominal accuracy of about $\pm$ 25 pounds. Hysteresis in the couplers was dramatically reduced by changing the packing of the coupler and draft gear from grease to a mixture of synthetic lubricant and oil. The couplers were mounted on the two loaded buffer cars located on either side of the test car.

Grade and acceleration effects were measured using a Bell & Howell strain gage accelerometer oriented in the longitudinal direction on the forward end of the test car. The need for accuracy in the measurement of the train acceleration was identified before testing began. The accuracy problem in measuring train acceleration can most clearly be seen by considering the influence of grade. The grade in the test zone is 1 percent and dominates the rolling resistance. Grade plus train acceleration is measured directly by the accelerometer because the longitudinal accelerometer is tipped and reads a component of the 1 g gravitational field when the grade is not zero.

For a 1 percent grade, the accelerometor reads 0.01 g. When the train acceleration signal is low pass filtered at .25 Hz, it rarely exceeds $\pm$0.02 g so the desired signal is of the order of $\pm$0.03 g. The full scale longitudinal acceleration without filtering is of the order of $\pm$10 g. If the accelerometer signal was digitized using a 12 bit A to D converter with full scale at $\pm$20 g, each digital step would be $40/2^{12}$ = 0.01 g and the data would be spread over six digital steps. This resolution is inadequate.

To solve this problem, the accelerometer signal was analog filtered at about 1 Hz and run through an amplifier to get full scale to be about $\pm$0.1 g at the digitizer. Although this procedure worked, problems were encountered with thermal drifts affecting the data. During the course of a test run, the wind and sun angles caused the accelerometer to heat or cool, generating a very low frequency extraneous signal. The accelerometer was packed in insulating material but the data indicate that thermal problems have not been eliminated altogether.

In the future, it is recommended that two further refinements be used to measure acceleration with the instrumented coupler technique. First, an accelerometer should position along the track from run to run. Permanent magnets were installed on the roadbed in holes drilled in the ties. A sensor located on the instrumentation car detected the magnets and a channel was recorded indicating when the magnet was passed. This system provided good discrimination between ALD targets (the magnets) and the normal background.

Train speed was measured at the instrumentation car (the Union Pacific's mobile laboratory) using a system developed by the Union Pacific Railroad. This speed was compared to that obtained from two rotary pulse generators on the test car and the results suggest that the two measurements (i.e., the instrumentation car speed and test car speed) were in excellent agreement after being filtered at .25 Hz. Brake line pressure deviation and notch setting of the throttle were also provided as part of the standard instrumentation package.

Curvature was estimated from two displacement transducers located around the center bowl to measure truck to carbody bolster rotation. Initially, curvature was based on track geometry data measured during a separate test run using the FRA is T-6 track geometry car. During the course of data reduction, it was found that the results were more sensitive to small errors in aligning the two sets of data (i.e., the track geometry run and each test run) than to small errors in estimating the local curvature. As a result, truck swivel was scaled to estimate curvature.

An automatic location detection system (ALD) was used to establish position along the track from run to run. Permanent magnets were installed on the roadbed in holes drilled in the ties. A sensor located on the instrumentation car detected the magnets and a channel was recorded indicating when the magnet was passed. This system provided good discrimination between ALD targets (the magnets) and the normal background.

The tests used in calculating the curving resistance were those tests run in the curved test zone. Typically runs were made under, near, and over balance speed in the uphill direction both empty and loaded. In addition, the loaded carbody was run in the downhill direction near balance speed.

5.3.3 Derivation of the Curving Resistance Formula

The conventional formulation of the curving force (i.e., 0.8 pound/ton/degree) assumes the car travels through the curve near balance speed. Usually no term is included to represent off balance speed behavior. Since tests were run below, near, and above balance speed, a term was needed to represent this behavior. The term adopted is based on the theory used to establish balance speed (i.e., at balance speed the centripetal force is cancelled by the superelevation).

Centripetal force is mv^2/r. Since $1/r$ is proportional to curvature, this is proportional to the car weight times the degrees of curvature times the speed squared. This force is balanced by the elevation of the high rail. The amount of superelevation is just enough to cancel the centripetal force at balance speed. Thus, the off balance speed term used is proportional to $Wd\,(v^2-v_b^2)$. When the train speed is equal to balance speed ($v=v_b$), the term is zero. If the train speed is higher than balance speed, the term is positive. If the train speed is lower than balance speed, the term is negative.

During the early formulations of this equation, the negative value below balance speed proved confusing. It did not seem possible that the curving resistance would be reduced by running below balance speed. The curving resistance will increase irrespective of whether the train is pushing on the high or low rail. A slightly more general formulation of the curving resistance was considered:

$$\text{curving resistance} = W\, f\,(d,v)$$

where $f(d,v)$ is some unknown function. The curving resistance is proportional to car weight, this has been confirmed by the TDOP Phase II test results. Expansion of f(d,v) in a two dimensional power series gives:

$$f(d,v) = a_{10}d + a_{20}d^2 + \dots + a_{01}v + a_{02}v^2 + \dots + a_{11}dv + a_{12}dv^2 + \dots$$

Terms that depend only on train speed can hardly be called curving terms since they are not zero when the curvature is zero. Terms that are proportional to v instead of v^2 should be small from the reasoning offered earlier to derive the off balance speed term (i.e., the term should represent centripetal force). Thus, we see that the leading terms of the power series would be:

$$f(d,v) = b_1 d + b_2 d^2 + b_3 d\,(v^2 - v_b^2)$$

If f(d,v) were ever to become negative, this would signify that the truck was pushing on the low rail. The absolute value signs belong around the whole expression:

$$\text{Curving resistance} = W\left|b_1 d + b_2 d^2 + b_3 d(v^2 - v_b^2)\right|$$

In a way this result is somewhat paradoxical. There may be a speed at which the curving resistance goes to zero, however it is dramatically below balance speed. This is consistent with 0.8 pounds/ton/degree of curving resistance at balance speed. It is also consistent with the data collected during field testing from the L/V and angle of attack systems on TDOP.

One interpretation is that the wheelset is displaced at balance speed so that the differences in rolling radii of the two wheels are just sufficient to take the truck around the curve. This ensures minimum slipping of the wheels on the rail.

Rolling Resistance Equation. The complete equation for the coupler forces of a freight car as modeled in TDOP is as follows:

$$CF_f \cos(0.01745\, C_f) - CF_r \cos(0.01745\, C_r) =$$
$$(2000\, W + 8\, W_w)\, a + 20\, Wg + 88\, b + 1.3b^2 +$$
$$29 + 1.3W + 0.045\, Wv + 0.0006\, v^2 +$$
$$\left|\, 0.68\, Wd + 0.08\, Wd^2 + 0.00041\, Wd\,(v^2 - v_b^2)\right|$$

Where:

CF_f = coupler force on the forward end (pounds)

C_f = coupler angle on the forward end (degrees)

CF_r = coupler force on the rear end (pounds)

C_r = coupler angle on the rear end (degrees)

W = total car weight including trucks and wheels (tons)

W_w = weight of a single wheel (pounds)

a = longitudinal car acceleration (in gravity acceleration

g = grade ratio (percentage)

b = brake pressure deficit (psi)

v = car speed (mph)

d = absolute value of 100 ft chord curvature (degrees)

V_b = balance speed (mph)

The first line of terms, $CF_f \cos(0.01745\, C_f) - CF_r \cos(0.01745\, C_r)$, represents the difference of the longitudinal component of the coupler forces. The coefficient 0.01745 converts degrees to radians.

The first term on the second line, $(2000\, W + 8\, W_w)\, a$, is the *inertial term*. The coefficient 2000 converts tons to pounds. The term 8 W_w is a simple way of estimating the rotary inertia of a wheelset. Theoretically, the term should be $4\, I_{rr}/r_w^2$ where I_{rr} is the rotational inertia of the wheelset and r_w is the radius of the wheel. However, ignoring the axle and treating the wheel like a hoop gives $I_{rr} = 2W_w r_w^2$. Thus the term 8 W_w is a close approximation to the real rotary inertia term.

The next term on the second line, 20 Wg, reflects the effect of grade. The coefficient 20 converts tons to pounds and percentage grade to grade ratio.

The last two terms, $88\, b + 1.3b^2$, represent the effect of applying the air brakes. The reason for including two terms is that it would be desirable for the first 5 psi of brake line deficit to have a smaller effect than the next 5 psi. This would give the engineer better control when using air brakes to stretch out the train. The two terms were included to see if this was really done. The positive sign of the second coefficient suggests it was.

The third line of the equation, $29 + 1.3W + 0.045\, Wv + 0.0006\, v^2$, is the original Davis equation. It represents rolling resistance on level, tangent track at a constant speed. This comes from roller bearing drag, aerodynamic drag, resistance at the wheel/rail interface, energy dissipated in the friction snubbers and center bowl, etc. These terms appear virtually as a constant in the TDOP Phase II curving runs. As a result, the terms are not fitted accurately enough to provide useful information for choosing between the various formulations of the equation (Reference 2).

The last line of the equation is the curving resistance formulation described previously.

Curve fit. The rolling resistance equation presented is rather difficult to fit. In the test zone used for TDOP Phase II curving tests, the second line of the equation (i.e., the inertial effects, grade, and braking) completely dominated all other terms. Any error in the gain or bias of either the accelerometer or the coupler forces obscured the curving forces.

In principle, the following procedure could be used:

a. Form the rolling resistance by taking the difference in the coupler forces.

b. Remove grade and acceleration effects by multiplying the filtered longitudinal accelerometer by the car weight and subtracting it from the coupler forces.

c. Remove the rotational inertia of the wheels by differentiating train speed, multiplying by the rotary inertia of the wheels, and subtracting it from the result of step b.

d. Estimate the Davis equation from the train speed and car weight and subtract it from the result of step c.

e. Fit the remainder to the curving terms using a least squares curve fit.

In practice, this procedure does not work very well because the accumulation of small errors in gain and bias of the data are too great.

To solve this difficulty, a modification of a statistical procedure known as a staged regression was used. Basically, the acceleration, grade, and rotary inertia of the wheels are removed using a least squares curve-fit to get them scaled as accurately as possible. Next, the bias is removed from the part of the data that is left over to get rid of the Davis equation, and the curving forces are fit to this residual. Unfortunately, the curving forces are highly correlated with the grade compensation in the curves, and the curving forces are always positive so they can not have a mean of zero. For these reasons, a straightforward staged regression will not work. Instead, what was done was to estimate the curving resistance values and remove them. Then the other terms were fitted and subtracted from the original data, and the curving terms were fitted to the residual. This process was repeated until the estimate at the curving resistance terms was equal to the fitted data. The basic process is illustrated below:

a. Form an estimate for the local curvature based on the ALD milepost and the track charts. Call this d_{est}. Least squares curve fit the equation

$$d_{est} = C_0 + C_1 \text{ D13} + C_2\text{D14}$$

to build the best available curvature channel. Use $C_0 + C_1$ D13 + C_2 D14 as curvature. (D13 and D14 are the test identifications of the truck swivel measurements). Multiply by zero whenever d_{est} is zero to force the curvature to zero when not in a curve.

b. Estimate the balance speed based on the track charts. Multiply by zero whenever d_{est} is zero to force the balance speed to zero when not in a curve.

c. Estimate the reading of the filtered accelerometer (test identification GR) by differentiating the train speed and adding it to the track chart grade. Call this GR_{est}. Least squares curvefit the equation:

$$GR - GR_{est} = C_0 + C_1 \text{ time} + C_2 \text{ time}^2 + C_3 \text{ time}^3 + C_4 \text{ time}^4 + C_5 \text{ time}^5$$

This estimates the bias of GR and the size of the thermal drifts experienced. It is useful if time is scaled to be in the range of -1 to 1. Remove the estimated bias and thermal drifts from GR.

d. Estimate the rolling resistance by taking the difference of the two coupler forces.

e. Estimate the Davis equation using the train speed and car weight based on the equation shown in the previous section.

f. Estimate the curving force term based on the equation shown previously using the best available estimate for the three coefficients for the current truck. Start with Type I truck coefficents as the first estimate.

g. Form the rotary inertia term for the wheels by differentiating train speed to get the necessary acceleration.

h. Subtract the terms formed in steps e through g from the rolling resistance formed at step d. Call this result Ma_{est}. Least squares curve fit the equation

$$Ma_{est} = C_0 + C_1 \text{ GR} + C_2 \text{ distance} + C_3 \text{ distance}^2 + C_4 \text{ distance}^3$$

to fix any gain or bias errors that remain. The polynomial with distance should be scaled to be in the range -1 to 1 and helps to remove any remaining thermal drift. Distance is based on the ALD system and shifted to have 0 at the center of the test zone.

i. Remove this result from the measured rolling resistance (i.e., subtract C_0 + C_1GR+...). Subtract the Davis equation and the rotary inertia terms. This is the residual.

j. Fit the curving terms to this residual. If the resulting coefficients are different than those assumed at step f, go back to step f and use these new results until the process converges.

This procedure is time consuming. It has the advantage that the results compare well with conventional results for the Type I truck and the results for the Type II trucks make sense. It also is very insensitive to gain and bias errors in the test data.

5.3.4 Track, Consist, and Fuel Consumption Simulator

In order to estimate savings/costs associated with the measured changes in curving resistance, it is necessary to make some assumptions about the track and train. Obviously, a train that runs over straight track will not experience any benefit from a reduction in curving resistance. Similarly, a 70-ton car would be expected to have less fuel savings than a 100-ton car. Other parameters probably are not as important as weight and curve-to-tangent track ratio. For example, a TOFC (trailer on flat car) configuration might experience less percentage change because increased aerodynamic drag causes the fuel consumption to increase. However, the savings per mile due to reduced curving coefficients would be expected to remain fairly constant. What is important is the integral of force over distance (i.e., work). For a constant change in force, one would expect to get the same change in work.

Initially the plan for fuel consumption calculations called for using an "average" consist and an "average" track. However, the TDOP consultants pointed out that the average was not necessarily representative of any railroad. Two procedures were proposed to deal with this: 1) scenarios, and 2) parameterization. Since the problem is relatively easy to parameterize (i.e., curve-to-tangent track ratio and car weight are the important parameters), it was decided to proceed this way.

A statistically "average" track was used as illustrated in Table 5-14. To vary curve-to-tangent ratio, each curved section of track was combined with a second identical record, effectively doubling the length of the curves. Also the train was run out and back so that it returned to the same place it left (to avoid any differences in altitude). Each curved segment was repeated three times and the tangent segments were repeated two or three times. This procedure provided track at a variety of curve-to-tangent ratios with no changes in altitude to complicate the data.

TABLE 5-14. STATISTICS FOR AVERAGE TRACK

GRADE	% of Track
0.12	52.60
0.38	24.85
0.63	13.60
0.88	7.73
1.50	0.50
2.00	0.72
CURVATURE	**% of Track**
0.0	73.07
0.5	9.88
1.5	8.57
2.5	5.82
3.5	0.84
4.5	0.47
5.5	0.64
6.5	0.71
SPEED	**% of Track**
0	0.69
10	4.59
20	5.02
25	1.29
35	4.98
40	2.46
45	15.10
55	26.48
65	36.76
79	2.61

Also an "average" consist composed of both empty and loaded cars was used (see Table 5-15). The measured rolling resistance for each truck was used for all the cars in the consist (except the locomotives) and the consist was run over each curve-to-tangent ratio of track. Next the number of locomotives was increased (from three to four) and the runs were repeated. The number of locomotives was increased again (from four to five) and the runs were repeated once again. Finally, grade compensation was added to the track models and all three locomotive arrangements were rerun for all track conditions. All of these modifications produced relatively small variations in the savings (i.e., the difference between Type I and each of the Type II trucks), although substantial differences were observed in overall fuel consumption. All the results were averaged and were presented earlier as Figures 5-10 and 5-11. Table 5-16 shows the numerical results.

The average track, average train, and fuel consumption simulator were all developed as part of an earlier FRA-sponsored study to develop and calibrate a fuel consumption simulator (as opposed to a train simulator). References 3 through 6 document this effort. Data used in this project were as specified in the references, except that the average track model was modified to add a separate tangent track as opposed to track of the lowest curvature class. Also the fuel consumption simulator was modified to allow the curving resistance equation and modified truck weight to be entered externally rather than being built into the routine. Finally, it proved necessary to reduce the step size of the numerical integration routine by a factor of 20 to get consistent results for the fuel savings.

5.3.5 Use of Tables

The average train used in the study carries 61 tons for each 32 loaded cars and also has 35 empty cars. A variety of carbody types are used. The gross train weight exclusive of locomotives and caboose is 4,318 tons. The net train weight is 1,952 tons. Thus the average car used in the study runs 48 percent loaded, has a light weight of 35 tons, and carries an average of 61 tons loaded. The average car weight would be 64 tons.

Since the fuel savings are roughly proportional to car weight, it is easy to re-scale the calculated savings for a specific car. For example, suppose one wanted to know the savings for a 100-ton hopper car that runs 50 percent loaded - 50 percent empty, has a light weight of 33 tons, and carries an average of 99 tons loaded.

Then the average car weight would be calculated as follows:

.5 x 33 tons + .5 x (99 tons + 33 tons) = 83 tons

And the scale factor for this 100-ton car would be 83/64 = 1.3 times the savings reported in the tables for an "average" car.

Given the scale factor, one can proceed to calculate the gallons saved per mile for a specific truck from Table 5-15. For example, suppose the route has a curve-to-tangent track ratio of 0.70 and the truck is the Dresser DR-1. Going to the table, for a curve-to-tangent ratio of 0.737, there is a savings of 0.00094 gal/mile. For a curve-to-tangent ratio of 0.368, there is a savings of 0.00024 gal/mile. By interpolating, the savings is 0.00087 gal/mile for a curve-to-tangent ratio of 0.070. Multiplying by the scale factor of 1.3 gives 0.00113 gal/mile.

If the price of diesel fuel is 85¢/gallon, this is a savings of \$0.00096/mile. If the car goes 50,000 miles/year, the savings is \$48/year. Assuming the price of diesel fuel will inflate at 4 percent above the average rate of inflation, that a 10 percent rate of return is a break even proposition, and that the projected car life is 30 years, then the value of the savings is as follows:

$$\sum_{n=1}^{30} \frac{\$48/\text{year} \times (1.04)^n}{(1.10)^n} = \$660$$

Given the 10 percent investment tax credit, one would break even by paying \$660 x 1.1 = \$726 per car for these savings. Of course, this value must be added to any savings from rail wear, car maintenance, etc., to calculate the warranted investment in a Type II (premium) truck.

TABLE 5-15. STATISTICS FOR AVERAGE TRAIN*

Cars	Number Loaded	Number Empty	Total
Box Cars	11	12	23
Hopper Cars	12	12	24
Flat Cars	3	3	6
Gondola Cars	3	4	7
Tank Cars	3	4	7
	32	35	67

*Refrigerator cars are placed into the boxcar category for these purposes, and the small portion (2%) of the remaining categories is ignored.

TABLE 5-16. GALLONS SAVED/CAR MILE BY CURVE/TANGENT RATIO

TRUCK	CURVE TO TANGENT 1.105	0.737	0.368	0.185	0.124
ACF FABRICATED	0.00086	0.00069	0.00038	0.00016	0.00010
ALUSUISSE (70 TONS)	-0.00024	-0.00030	-0.00019	-0.00015	-0.00008
BARBER-SCHEFFEL	0.00157	0.00113	0.00015	-0.00017	-0.00055
DEVINE-SCALES	0.00196	0.00126	0.00009	-0.00041	-0.00055
DRESSER DR-1 TRUCK	0.00140	0.00094	0.00024	-0.00013	-0.00023
MAXIRIDE TRUCK	-0.00020	-0.00019	-0.00004	-0.00004	0.00004

5.4 ROADWAY MAINTENANCE ANALYSIS

Reduced roadway maintenance is an area where significant savings might be achieved with an improved truck design. The Canadian Institute of Guided Ground Transport (CIGGT) under sponsorship of Transport Canada has agreed to undertake such a study of the TDOP Phase II test data and to publish a report available through National Technical Information Service later in the year. CIGGT has done several similar studies in the past. The most recent of these studies was done for the Track-Train Dynamics Program (References 7 and 8) and is the basis for the material that follows.

In the CIGGT analysis, rail of tangent and low curvature classes tends to wear out due to fatigue or rail end batter; in high curvature classes, it tends to wear out due to plastic flow at the high rail gauge face and low rail head. There is potential for savings due to improved truck design in both failure modes. The tangent and low curvature track life might be increased by reduction in the dynamic vertical loads, which could be achieved by reducing the unsprung mass acting on the rail (e.g., by using a primary suspension system). Also the axle spacing, spring rates, and snubbing would have some affect on the vertical dynamic load. On the other hand, the steady state vertical load would tend to be slightly higher due to the increased weight of these trucks. Of course, the exception would be when the car is fully loaded. Under these conditions, the car would have to carry slightly less lading to avoid exceeding the total rail weight limits.

Based on TDOP Phase II test results, the case for lengthening rail life on tangent and low curvature track does not appear good. In general, the test data show that dynamic vertical loads are higher rather than lower. Only one primary suspension truck showed major decreases in the dynamic vertical load. It appears that increased rail life on tangent track due to reduced dynamic vertical loads will not be realized with most Type II trucks.

In the high curvature classes, one would expect significant savings due to the decrease in lateral curving stresses associated with radial (steering) trucks and, possibly, rigid trucks. Stress level, not simply lateral force, wears out both the wheels and the rail. For this reason, the product of angle of attack and lateral force is often suggested as an index for wear. If the angle of attack is higher, it implies that the contact area is smaller and that the stresses are higher for the same lateral force. This mechanism is modeled in the CIGGT analysis for multiple contact patches on the rail head and gauge faces.

The test data show that a steering truck can achieve a significant reduction in the steady state lateral load on intermediate curvature track. Potentially large savings can be achieved with high annual mileage cars operating on routes with high curve-to-tangent ratios.

Finally, there is a potential for savings in the area of lessened gauge widening from a truck that does not hunt. Results from performance testing in TDOP have provided lateral forces and durations during hunting for a number of circumstances. These results have the potential to be further developed into a gauge widening model. However, a significant effort is needed to calibrate these results to the real world situation. If gauge widening on tangent track is caused primarily by hunting, the tendency of the gauge to widen should increase with increased speed. Data showing the incidence of gauge widening with posted speed need to be gathered to build a model of this type. The savings could be potentially significant not only in the area of rail maintenance but also in reduction of number of spike-killed ties.

5.4.1 Savings on Tangent Track

Reduction in dynamic vertical loads due to reduction in the unsprung mass of the truck, changes in axle spacing, spring rates, damping, etc., offer a potential for savings in the tangent and low curvature track classes. TDOP Phase II used strain gaged pedestal adapters to measure vertical loads for the standard Type I, Barber-Scheffel, Devine-Scales, Dresser DR-1, and National Swing Motion trucks. Since the Maxiride-100 truck did not have conventional adapters, spring group displacement was used to measure vertical loads in it. In the case of the DR-1, modified pedestal adapters with an elastomeric pad (part of the truck design) had to be used instead of the conventional, instrumented adapters. The Devine-Scales truck used different strain gage instrumentation on the adapter. Details of the instrumentation and associated data reduction can be found in Section 2 of this report.

While the trends in the mean vertical loads are probably fairly representative (i.e., the data suggest how the vertical load is redistributed as the car goes through a curve), care needs to be taken that the standard deviations are not over-interpreted. For example, there is a tendency for the standard deviation to decrease as the load on the wheel increases with the Type I truck in the loaded car configuration. This probably has more to do with the nonlinear calibration of the adapter than the Type I truck dynamics. The simplest thing to do is to average all the standard deviations, calling that typical of the truck. Using this procedure, it is possible to compare the Type I, National Swing Motion, and Barber-Scheffel trucks (all of which use the same measurement system). Differences between the Type I truck and the Devine-Scales, Maxiride-100, or Dresser DR-1 must be evaluated in the context of the differences in the measurement system.

Table 5-17 and Figure 5-16 illustrate these results for loaded cars. As expected, the result for the Maxiride-100 truck is lower. The Devine-Scales result is somewhat surprising in that it has primary suspension and reduced unsprung mass. However, the pockets that hold the suspension seized up during testing and very little vertical motion was observed.

Comparison of results from the Barber-Scheffel and National Swing Motion trucks with results from the Type I truck is consistent with the idea that the unsprung mass affects the dynamic vertical loads. Each of these Type II trucks mount extra equipment on the side frame, thus increasing the unsprung mass. Each of them seems to have a slightly greater dynamic vertical load than a Type I truck. In the case of the Barber-Scheffel truck, there are elastomeric pads between the shear pad housing and the side frame pedestal roof. These might be expected to provide some primary suspension action, but do not appear to do so.

The Dresser DR-1 truck result shows a slightly reduced dynamic vertical load compared to the Type I truck, which is counter to the idea that an increased unsprung mass will increase the dynamic vertical load (the DR-1 has slightly greater unsprung mass). However, the DR-1 does feature an elastomeric pad at the bearing adapter which may perform the functions of a primary suspension element, thus, reflecting the influence of reduced vertical dynamic loads.

Turning to the economic issue of how much a change in dynamic loading will save, estimates in Appendix E of Reference 8 are reproduced here as Table 5-18. For a traffic density of 15 MGT/year with a 100-ton car, there is a very significant annual savings from a reduction of 25 percent in the vertical dynamic loads. The equivalent annual benefit due to increased rail life is $231/mile for track below 2 degrees curvature. However, a Type II truck traffic of 15 MGT/year seems extremely high.

TABLE 5-17. STATIC AND DYNAMIC VERTICAL LOADS FOR LOADED CARS

Truck Type	Static Loads, Average Weight/Axle	Dynamic Loads, Standard Deviation	% Relative* to Type I
Type I	33,000 Lb	+2,144 Lb	100%
Barber-Scheffel	33,250 Lb	+2,577 Lb	120%
Devine-Scales	33,375 Lb	+2,979 Lb	139%
DR-1	33,156 Lb	+1,943 Lb	91%
Maxiride	32,982 Lb	+1,312 Lb	61%
Swing Motion	33,250 Lb	+2,513 Lb	117%

* For example, % relative to Type I for the Barber-Scheffel = (2577/2144) x 100.

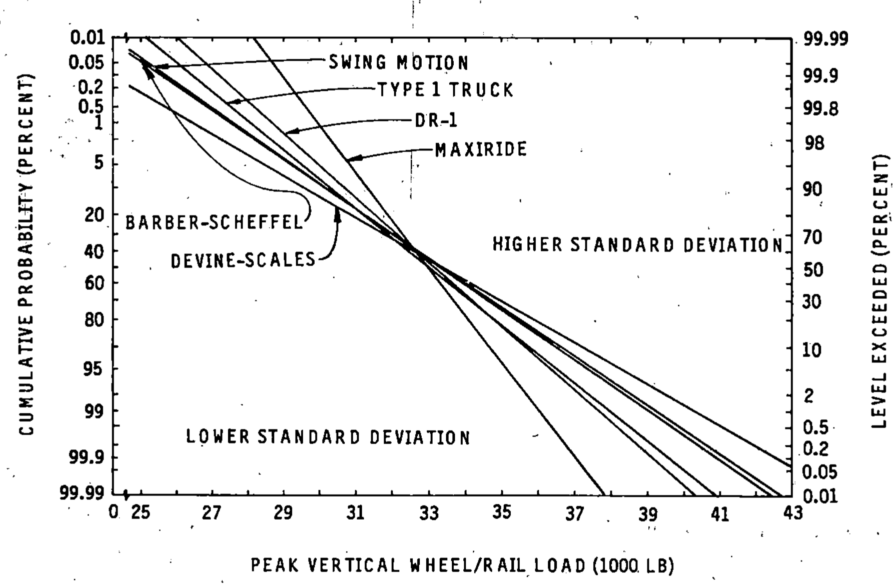


FIGURE 5-16. CUMULATIVE PROBABILITY DENSITY FUNCTIONS FOR VERTICAL LOADS FOR LOADED CARS

For example, a 75,000 mile/year unit coal train with 100 cars averaging one trip out and back each week would have a route 721 miles long (75,000 miles/104 trips). If the gross load on the rails was 132 tons, this would be 0.86 MGT/year (100 cars x 132 tons x 52 loaded trips + 100 cars x 32 tons x 52 empty trips). Scaling the savings/mile down gives $13.2/mile ($231 x 0.86/15). If a curve-to-tangent ratio of about 1 to 3 is assumed, then there are 541 miles of track in this category (.75 x 721 miles) and a total annual benefit of $71/car/year (i.e., 541 miles x $13.2/100 cars). Assuming the remainder of the track is predominately in the 2 to 5-degree category, carrying out the same calculation for this category gives a savings of $47/car/year. Adding these results, the total (rail) savings from a 25 percent reduction in vertical load would be $118/car/year.

While the differences here are rather dramatic in terms of dollars/car/year it should be pointed out that there were procedural problems with the CIGGT analysis and it seems likely these numbers will be subject to change based on additional analysis of the TDOP Phase II performance data being conducted by CIGGT.

5.4.2 Savings on Curved Track

Reduction in lateral curving stresses at the wheel/rail interface due to reduced lateral force or a reduced angle of attack offers the potential for significant rail life increases on curved track. The savings come from a net reduction in the steady state lateral curving force or steady state angle of attack as a function of curvature (see Figures 5-17 and 5-18). There is also a dynamic lateral force and dynamic angle of attack. This is the random variation about the steady state force and can be represented by a standard deviation as in the case of the dynamic vertical force. The lateral dynamic environment is quite large compared to the steady state environment for low degrees of curvature and should not be ignored.

Lateral load, L/V, and angle of attack data for the Type II trucks have been sent to CIGGT in the form of means and standard deviations. The test data for Type II trucks were taken with the CN profile. This profile is intended to cause single point contact between the wheel and rail. The contact patch assumptions in the CIGGT model might need to be reconsidered for these trucks. The Type I truck was tested with a new AAR profile. Based on the results already shown in Figure 5-18, the prospects for reducing the steady state lateral loads by factors of about 50 percent appear excellent for steering trucks. The appropriate data are reproduced in Table 5-19 from the economic analysis in Reference 8.

Considering the case developed earlier for vertical dynamic loads (i.e., a 100-car unit coal train, operating over 721 miles of track, etc.), the scaled annual benefits due to the change in traffic density is $82.3/mile for the two degree to five degree track category ($1435 x 0.86/15). Under the assumptions given earlier (i.e., curved/tangent = 1/3), there are 180 miles of track in this category, thus, the total annual benefit for rail wear is $148/car/year ($82.3 x 180 miles/100 cars).

5.4.3 Savings Other Than Rail Life

Savings other than in rail life have been classified in two categories in the analysis in Reference 8. These are savings in tie life and "other" savings. The values projected for these savings were shown in Tables 5-18 and 5-19 and are often larger than the savings projected for increases in rail life. Although not originally planned for inclusion in TDOP Phase II, these savings have been estimated based on Reference 8 and included in the cost/benefit analysis.

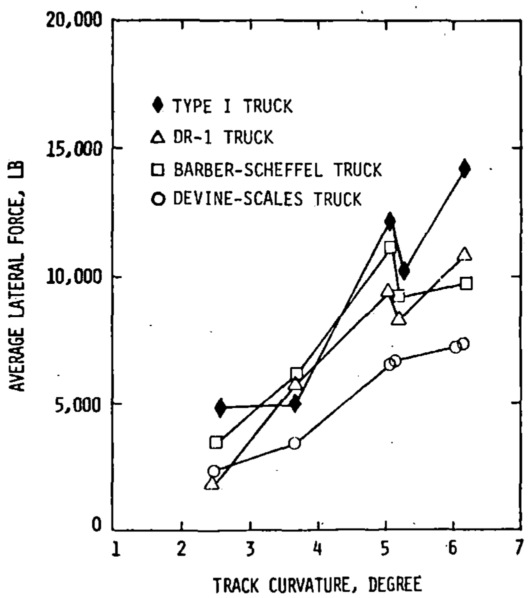


FIGURE 5-17. AVERAGE LATERAL FORCE VS CURVATURE FOR LEFT HAND CURVES FOR LOADED STEERING TRUCKS

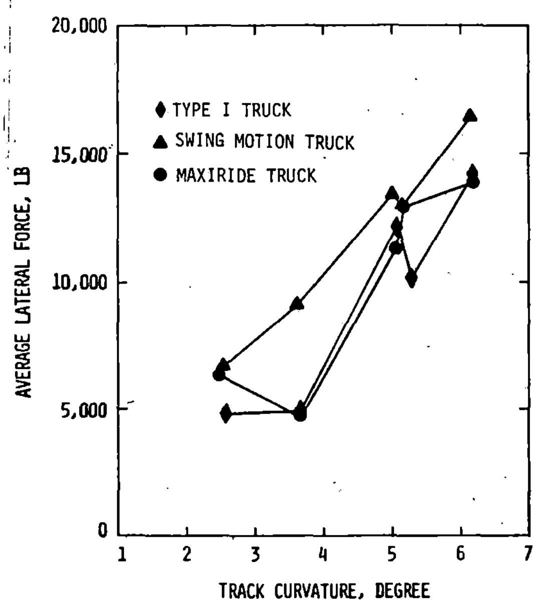


FIGURE 5-18. AVERAGE LATERAL FORCE VS CURVATURE FOR LEFT HAND CURVES FOR LOADED NON-STEERING TRUCKS

TABLE 5-18. ECONOMIC IMPACT OF VARIATIONS IN VERTICAL DYNAMIC LOADS

TRAFFIC - 15 MGT/YR.

BASE CAR - 100 TON (263,000 lb. gross load on rails)

TRIAL CAR - Base car with reductions in static and dynamic loads as listed below:

a) vertical dynamic reduced by 25 percent

b) lateral dynamic reduce by 0 percent

c) static load reduced by 0 percent

BENEFITS OF TRIAL CAR OVER BASE CAR

CURVATURE		TAN	$0^{\circ}-2^{\circ}$	$2^{\circ}-5^{\circ}$	$5^{\circ}-8^{\circ}$
Rail Life (MGT)	Base Car	347.9	347.9	193.6	128.1
	Trial Car	399.2	399.2	225.4	147.7
Advantage $/Mile Present Value		2,570	2,570	5,070	7,525
Equivalent Annual Benefit $/Mile - Rail		231	231	456	677
Tie Life (Yrs)	Base Car	24.8	22.7	16.8	15.9
	Trial Car	25.0	22.9	17.1	16.2
Advantage $/Mile Present Value		141	169	458	510
Equivalent Annual Benefit $/Mile - Ties		13	15	41	46
Other Cost Relative Factor	Base Car	1.235	1.342	1.643	1.689
	Trial Car	1.190	1.294	1.582	1.626
Advantage $/Mile Present Value		3,455	3,689	4,688	4,833
Equivalent Annual Benefit $/Mile - Other		311	332	422	435
Equivalent Annual Benefit $/Mile - TOTAL		555	578	919	1,153

TABLE 5-19. ECONOMIC IMPACT OF VARIATIONS IN LATERAL LOADS

TRAFFIC	-	15 MGT/YR.
BASE CAR	-	100 TON (263,000 lb. gross load on rails)
TRIAL CAR	-	Base car with reductions in static and dynamic loads as listed below:

a) vertical dynamic reduced by 0 percent

b) lateral dynamic reduced by 50 percent

c) static load reduced by 0 percent

BENEFITS OF TRIAL CAR OVER BASE CAR

CURVATURE		TAN	0^o-2^o	2^o-5^o	5^o-8^o
Rail Life (MGT)	Base Car	347.9	347.9	193.6	128.1
	Trial Car	347.9	347.9	347.9	249.5
Advantage $/Mile Present Value		0	0	15,939	26,428
Equivalent Annual Benefit $/Mile - Rail		0	0	1,435	2,379
Tie Life (Yrs)	Base Car	24.8	22.7	16.8	15.9
	Trial Car	24.8	24.6	19.9	19.1
Advantage $/Mile Present Value		0	1,491	4,064	4,618
Equivalent Annual Benefit $/Mile - Ties		0	134	366	415
Other Cost Relative Factor	Base Car	1.235	1.342	1.643	1.689
	Trial Car	1.235	1.245	1.485	1.526
Advantage $/Mile Present Value		0	7,444	12,132	12,510
Equivalent Annual Benefit $/Mile - Other		0	670	1,092	1,126
Equivalent Annual Benefit $/Mile - TOTAL		0	804	2,892	3,920

5.5 LADING DAMAGE & DERAILMENT ANALYSIS

5.5.1 Lading Damage

Reduced lading damage is another area where economic savings may be realized from a Type II truck. Several of the trucks have dramatically altered suspension characteristics (either dual spring rates to provide a different ride when empty than when loaded, or primary suspension where the spring nest is over the roller bearings, or in one case, leaf springs as the primary means of suspension). These modifications are intended to attenuate the track excitation and thus provide a better ride.

With regard to the vertical dynamic environment, small variations in rms levels between the trucks were observed (see Figures 5-19 through 5-24). None of the trucks performed dramatically better than Type I trucks.

Considerable savings could be realized through reduced lading damage claims. Claims that might have something to do with trucks appear in one of three "cause" categories in the AAR lading damage reports. These are cause 3 (all damage not otherwise provided for), cause 4 (defective or unfit equipment), and cause 9 (derailment). Of these, cause 3 is the only one that is significant. Comparing the commodity codes from the AAR lading damage reports to the 1 percent waybill sample, it is possible to estimate the damage costs per mile (after making an assumption that the cars travel 50 percent empty and 50 percent loaded). Typical results are shown in Table 5-20.

It is widely reported that longitudinal dynamics, not vertical or lateral dynamics, is the primary cause of lading damage (see Reference 9). There are probably specific cases where vertical and lateral dynamics contribute to lading damage. If Type II trucks could carry the right commodity, the savings might be substantial. Table 5-20 shows that there are a number of commodities that have lading damage costs in the 1¢/mile range. If a Type II truck were to reduce this cost by 50%, assuming the car travelled 25,000 miles/year, this would be an annual savings of $125/year. The savings could be very comparable to the savings from car maintenance, fuel consumption, or rail wear. Of almost equal importance, an improved truck with better ride characteristics might be able to draw new commodities to the railroads.

Reduction of hunting amplitudes and increases in hunting speeds will reduce lading damage for lightly loaded or partially loaded cars. All the Type II trucks tested produced significant improvements in these areas. The potential is there to reduce lading damage due to improved lateral dynamics. However, it is very difficult to be quantitative about these savings in any general way. On specific trains with specific routes and types of commodity, it would be somewhat easier to identify how much might be saved in lading damage.

5.5.2 Derailment

To the extent that Type II truck reduces the L/V ratio, it would tend to derail less often. Thus, it might seem reasonable that a Type II truck would experience lower derailment costs. Unfortunately, this is probably not the case. Most of the cars that are involved in large derailments are mechanically sound and would not have derailed if the car in front of them had not derailed. Since an improved truck will not improve the performance of the cars ahead of it, it seems unlikely that much savings could be realized in this area by the owner of the car.

Rather, a Type II truck will tend to reduce the overall incidence of derailments. Even this effect will be fairly distant. One might typify derailment as the weakest car encountering the weakest track with catastrophic results. It will take a number of years for a car introduced into service now to become the weakest car. Under the circumstances, savings from derailments are probably not a major economic factor to be considered in purchasing a Type II truck.

On the other hand, there are certain car series (e.g., the 100-ton covered hopper) that have statistically significant increases in the incidence of derailment. Under these circumstances, it might make sense to consider a Type II truck as part of an overall design strategy when redesigning such a car series (see Reference 10).

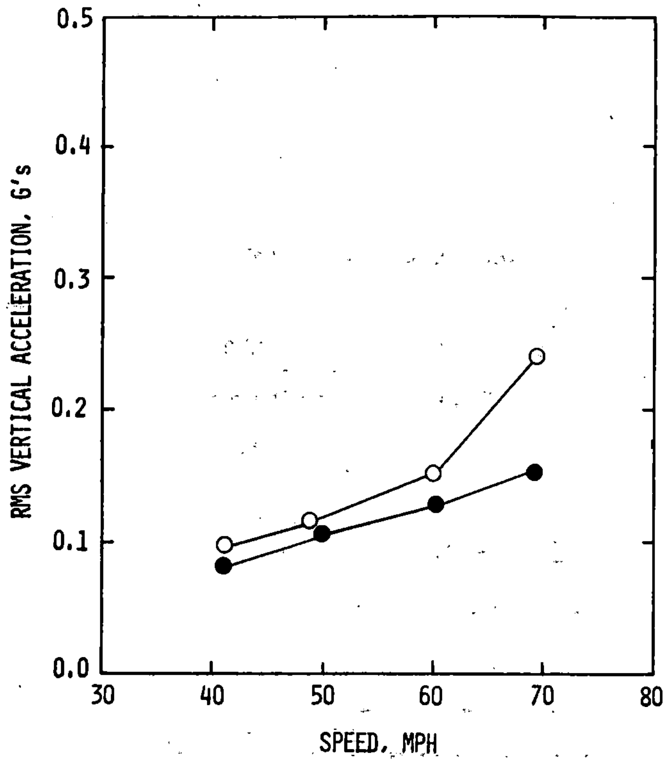


FIGURE 5-19. RMS VERTICAL ACCELERATION VS SPEED - 0-20 HZ FREQUENCY BAND/LOADED CARS/PRIMARY SUSPENSION TRUCKS

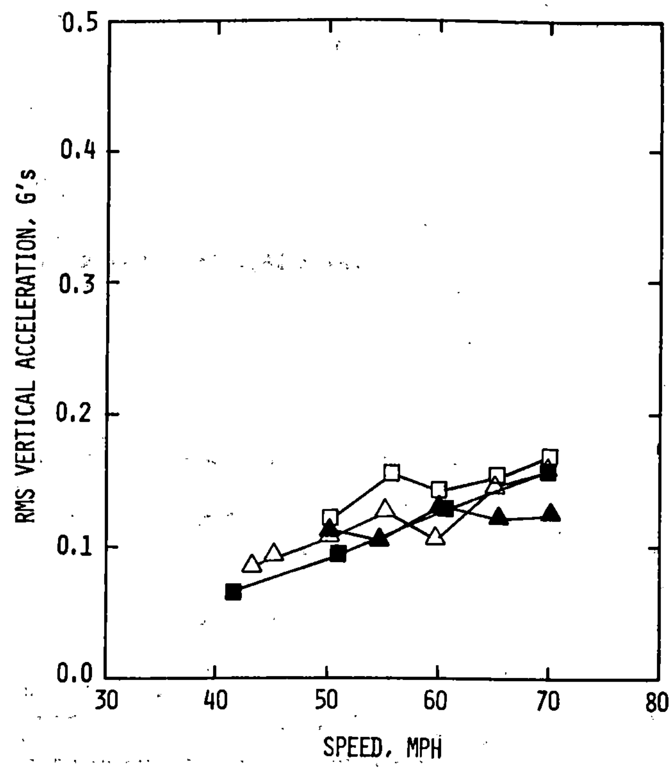


FIGURE 5-20. RMS VERTICAL ACCELERATION VS SPEED - 0-20 HZ FREQUENCY BAND/LOADED CARS/SECONDARY SUSPENSION TRUCKS

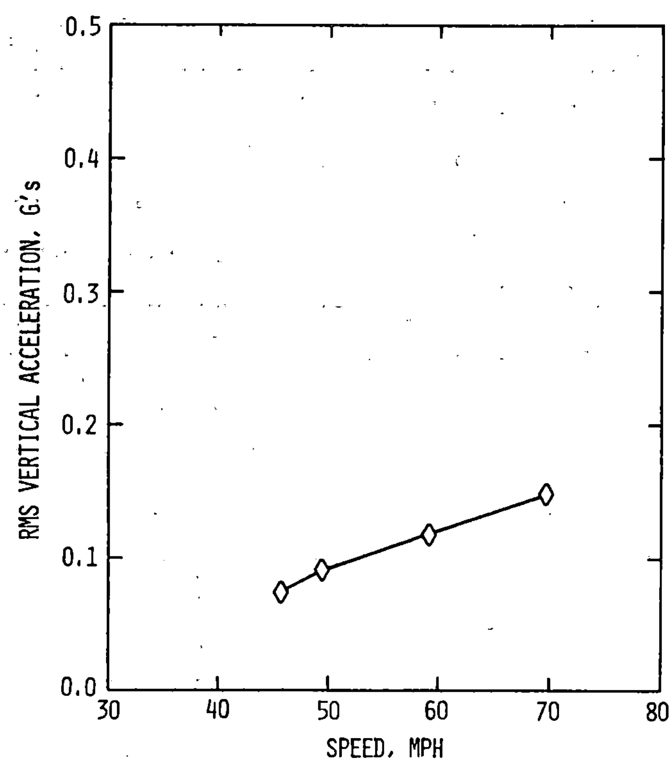


FIGURE 5-21. RMS VERTICAL ACCELERATION VS SPEED - 0-20 HZ FREQUENCY BAND/LOADED CARS/ PRIMARY + SECONDARY SUSPENSION TRUCKS

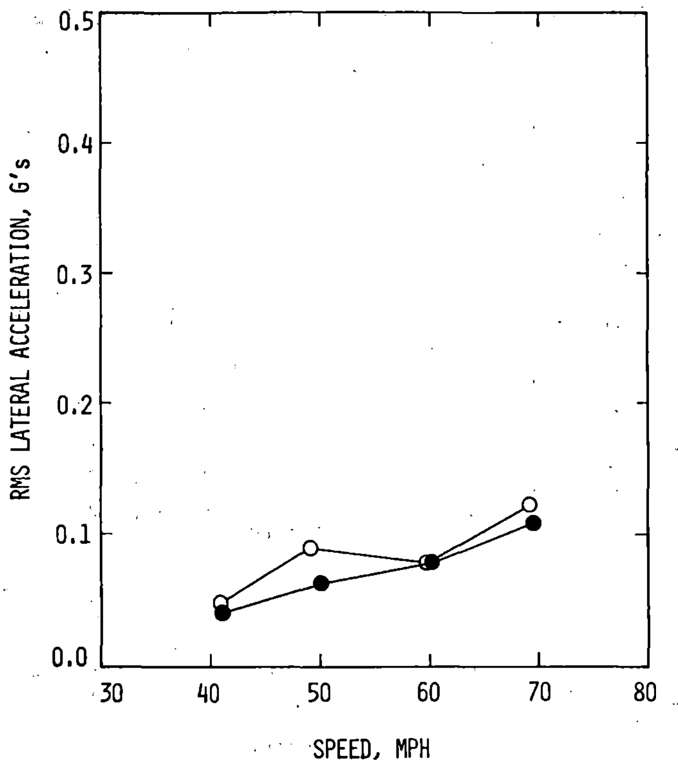


FIGURE 5-22. RMS LATERAL ACCELERATION VS SPEED - 0-20 HZ FREQUENCY BAND/LOADED CARS/ PRIMARY SUSPENSION TRUCKS

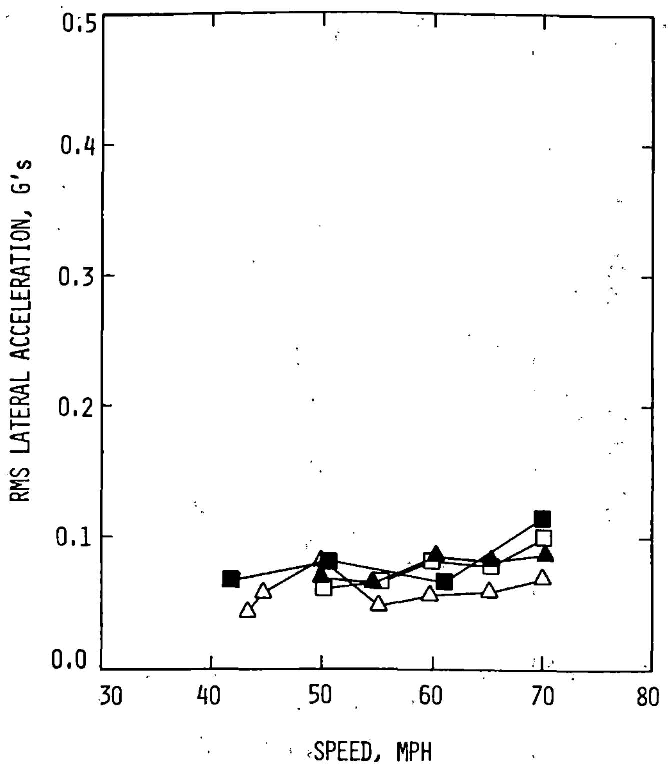


FIGURE 5-23. RMS LATERAL ACCELERATION VS SPEED - 0-20 HZ FREQUENCY BAND/LOADED CARS/ SECONDARY SUSPENSION TRUCKS

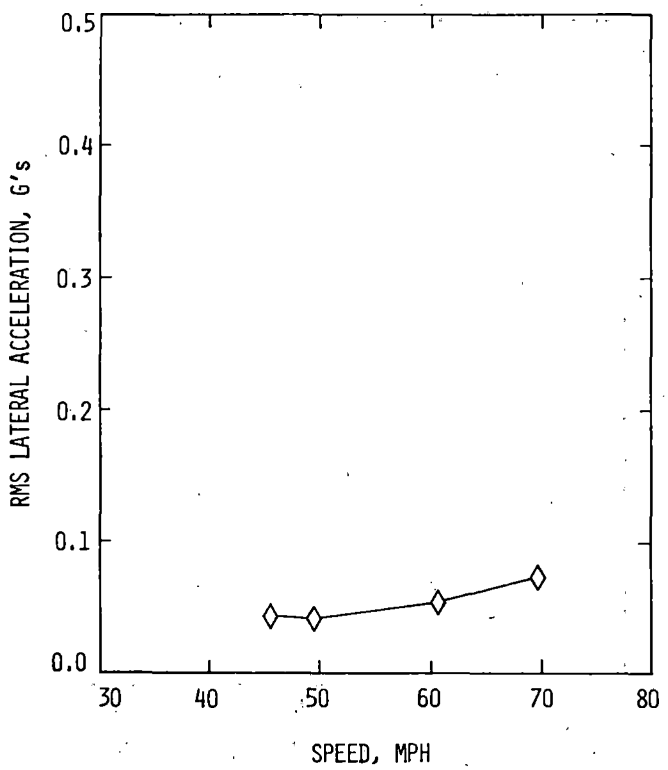


FIGURE 5-24. RMS LATERAL ACCELERATIONS VS SPEED - 0-20 HZ FREQUENCY BAND/LOADED CARS/ PRIMARY + SECONDARY SUSPENSION TRUCKS

TABLE 5-20. LADING DAMAGE PER MILE BY COMMODITY TYPE

		PERCENT ALL CLAIMS	1% WAYBILL 1000 MI	CAUSE SYMBOL 3 1000 $	CAUSE SYMBOL 4 1000 $	CAUSE SYMBOL 9 1000 $	CAUSE SYMBOL 3 CENTS/MI	CAUSE SYMBOL 4 CENTS/MI	CAUSE SYMBOL 9 CENTS/MI
01	FARM PRODUCTS	18.7	8709.	5453.	3202.	10374.	0.31	0.18	0.60
01121	COTTON IN BALES	0.8	405.	16.	14.	1286.	0.02	0.02	1.59
0113	GRAIN	8.4	4996.	72.	7306.	6909.	0.01	0.73	0.69
01144	SOYBEANS	1.1	364.	2.	374.	1376.	0.00	0.51	1.89
01195	POTATOES	0.8	674.	402.	22.	67.	0.30	0.02	0.05
012	FRESH FRUIT & NUTS	2.3	571.	1673.	75.	267.	1.46	0.07	0.23
013	FRESH VEGETABLES	4.8	1130.	2482.	94.	723.	1.10	0.04	0.32
10	METALLIC ORES	0.7	2457.	191.	316.	863.	0.04	0.06	0.18
11	COAL	1.4	11657.	98.	468.	2521.	0.00	0.02	0.11
14	NONMETALLIC MINERALS	0.5	3300.	634.	257.	393.	0.10	0.04	0.06
20	FOOD	20.8	12929.	38812.	2685.	8112.	1.50	0.10	0.31
201	MEAT	1.6	587.	540.	248.	394.	0.46	0.21	0.34
2033	CANNED FRUIT OR VEG	0.7	802.	1611.	21.	255.	1.00	0.01	0.16
2037	FROZEN FRUIT OR VEG	0.5	969.	228.	15.	154.	0.12	0.01	0.08
2039	MIXED CANNED GOODS	0.6	448.	1322.	9.	219.	1.48	0.01	0.24
204	GRAIN MILL PRODUCTS	7.3	4006.	16948.	1119.	2268.	2.12	0.14	0.28
2062	REFINED SUGAR	1.1	594.	2604.	123.	477.	2.19	0.10	0.40
20821	BEER	0.9	1121.	1782.	13.	413.	0.79	0.01	0.18
209	MISC FOOD PREPARATIONS	4.7	2449.	8533.	582.	2197.	1.74	0.12	0.45
21	TOBACCO PRODUCTS	0.8	239.	1214.	71.	251.	2.54	0.15	0.53
24	LUMBER OR WOOD	3.5	9718.	5718.	97.	2748.	0.29	0.00	0.14
2432	PLYWOOD OR VENEER	0.8	2047.	1457.	19.	723.	0.36	0.00	0.18
25	FURNITURE OR FIXTURES	1.8	1273.	3722.	125.	452.	1.46	0.05	0.18
26	PAPER, PULP, ETC.	4.9	7862.	10902.	167.	2069.	0.69	0.01	0.13
26213	PRINTING PAPER	1.1	888.	2724.	65.	419.	1.53	0.04	0.24
28	CHEMICALS	3.9	8452.	4689.	487.	4622.	0.28	0.03	0.27
29	PETROLEUM OR COAL PROD	1.0	3277.	1456.	105.	1051.	0.22	0.02	0.16
32	CLAY, GLASS, STONE	2.6	4271.	5399.	101.	1060.	0.63	0.01	0.12
322	GLASSWARE	0.2	265.	428.	13.	44.	0.81	0.02	0.08
32511	BRICK	0.4	263.	1132.	5.	52.	2.15	0.01	0.10
33	PRIMARY METAL PRODUCTS	2.2	3803.	4321.	213.	981.	0.57	0.03	0.13
34	FAB METAL PRODUCTS	1.5	1240.	3060.	37.	305.	1.23	0.01	0.12
35	MACHINERY (NOT ELECT)	2.4	1010.	4016.	40.	2244.	1.99	0.02	1.11
36	ELECTRICAL MACHINERY	3.1	1366.	4663.	90.	1621.	1.71	0.03	0.59
363	HOUSEHOLD APPLIANCES	1.9	1028.	3354.	17.	3112.	1.63	0.01	1.51
37	TRANSPORTATION EQUIP	20.4	9021.	47090.	54.	8112.	2.61	0.00	0.45
3711	MOTOR VEHICLES	18.5	4667.	44562.	17.	5853.	4.77	0.00	0.63
3714	MOTOR VEHICLE PARTS	1.3	3767.	2108.	29.	1534.	0.28	0.00	0.20

Note: Categories are the same as cause symbols.

5.6 COST/BENEFIT ANALYSIS

The results of the component elements of economic analysis covered in subsections 5.2 through 5.5 are assimilated and discussed in this subsection. The major parameters that govern the profitability of Type II trucks have been identified as:

a. Annual car mileage
b. Curve-to-tangent ratio of the route
c. Number of trucks purchased
d. Car weights
e. Empty-to-loaded ratio
f. Captive versus interchange service
g. Lading sensitivity to damage
h. Added cost of the truck

5.6.1 Results

Reviewing the different areas where savings might reasonably be achieved (car maintenance, roadway maintenance, fuel consumption, lading damage, and derailment), it is clear that with the exception of car maintenance, the handling line receives most of the benefits from improved trucks. The owner (if he is not the handling line) pays for benefits someone else receives. For this reason, Type II trucks can only be profitably used in some type of captive service.

For the same reason, car lines and private owners are not likely to be very interested in Type II trucks. The railroad receives the benefits from a Type II truck, not the private owner. If the railroad is willing to pay higher fees to the car line or charge lower fees to the private owner they might be willing to purchase these trucks, however, there is still the problem of getting nonstandard parts to the truck in a timely way. The owner generally will have to pay for lost car days. In order for a private owner or car line to profitably purchase a Type II truck, the railroad has to actively cooperate in controlling the costs of maintaining nonstandard trucks.

Private owners (especially utility companies and Trailer Train) control a significant portion of the cars (such as these in unit coal trains) that might reasonably be equipped with premium trucks. There are 26 pages of prime candidate cars in the 1977 Official Railway Equipment Register belonging to utility companies and Trailer Train alone. The railroads will have to take the initiative in getting maximum benefits from a Type II truck. If railroads are convinced of such benefits, they must play an active role in equipping appropriate cars with improved trucks.

In order for a Type II truck to be profitable, it must be on a car with high annual mileage because the benefits from such a truck accrue sooner on a high mileage car and are consequently worth more. All the benefits, such as savings from reduced rail wear and fuel consumption, are more or less proportional to annual car mileage. The annual benefits get larger with larger annual mileage.

One of the reasons that Type II trucks cost more than Type I trucks is because Type I trucks are produced in very large quantities. The Type I truck costs 65¢ a pound. A price this low is only possible in very large quantities. The nonstandard parts in a Type II truck cost around \$1 a pound. This is a fairly typical price for iron products produced in normal production quantities. If the truck manufacturer tries to recover his development costs, the price/pound is even higher. Several of the Type II truck manufacturers have taken advantage of these facts. Retrofit kits or trucks that are already in production have distinct cost advantages since they take advantage of the economies of scale in producing a Type I truck.

The advantages of a Type II truck have to be dramatic to overcome the built-in cost advantages of a Type I truck. Also, the advantages of standardization work against a Type II truck. The cost of introducing a set of nonstandard parts to many repair sites throughout the United States is quite significant.

Under the circumstances, it is fairly surprising that any Type II truck has a chance to enter the railroad market. However, it appears that there are two scenarios in which a Type II truck may be profitable. First, a steering (i.e., radial) truck may be able to pay for itself on routes with high curve-to-tangent ratios using high annual mileage cars. The savings come primarily from reduced rail wear of curved track. Doing a worst-case analysis of one of the steering trucks and varying the annual mileage and curve-to-tangent ratio, the boundary (between profitable and marginal) shown in Figure 5-25 was obtained in which the net present value of the benefits was positive.

As can be seen by considering the values of the parameters at which the steering truck is clearly profitable, there are relatively few cars in this situation; however, they do exist. The marginal area is more encouraging. Again, these are not the normal cars being purchased today (the annual mileage is too high). However, significant numbers of cars with these combinations of annual mileage and curve-to-tangent ratio do exist. Also, the requirement for routes with unreasonably high curve-to-tangent ratios has been reduced well into the normal railroad range.

The second scenario with a profitable Type II truck is more questionable. A primary suspension truck reduces the unsprung mass of the truck and may experience lower vertical dynamic loads. This leads to reduced track wear irrespective of curvature. The boundary with curve-to-tangent track ratio is not very meaningful because the savings are insensitive to it (see Figure 5-26). A more interesting case is the boundary with loaded car weight because of the potential for savings in reduced lading damage due to hunting (see Figure 5-27). Normally, loaded cars on Type I trucks hunt at very high speeds, and since they are not run at such speeds, one would not expect lading damage due to hunting. However, particularly in intermodal service, there are a reasonably large number of unit trains that carry relatively light loads.

The boundary with loaded car weight shown in Figure 5-27 is quite encouraging. No savings from lading damage were assumed in computing the boundary and a number of losses were excluded (e.g., small fuel losses) yet there is enough track wear savings to pay for the truck improvement even with a lightly loaded car. While it is virtually impossible to calculate how much savings there would be due to reduced lading damage, substantial savings seem likely.

Depending on the assumptions one makes about car maintenance and truck prices, a very similar scenario can be constructed for rigid trucks in which savings come from car maintenance and reduced lading damage, and losses are due to increased track wear.

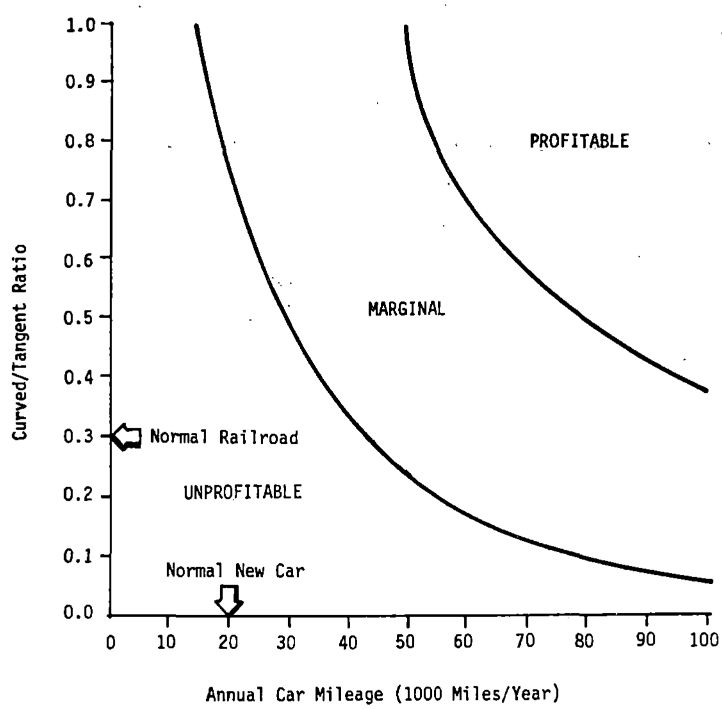


FIGURE 5-25. BOUNDARIES WITH CURVED TO TANGENT RATIO FOR STEERING TRUCK SCENARIO

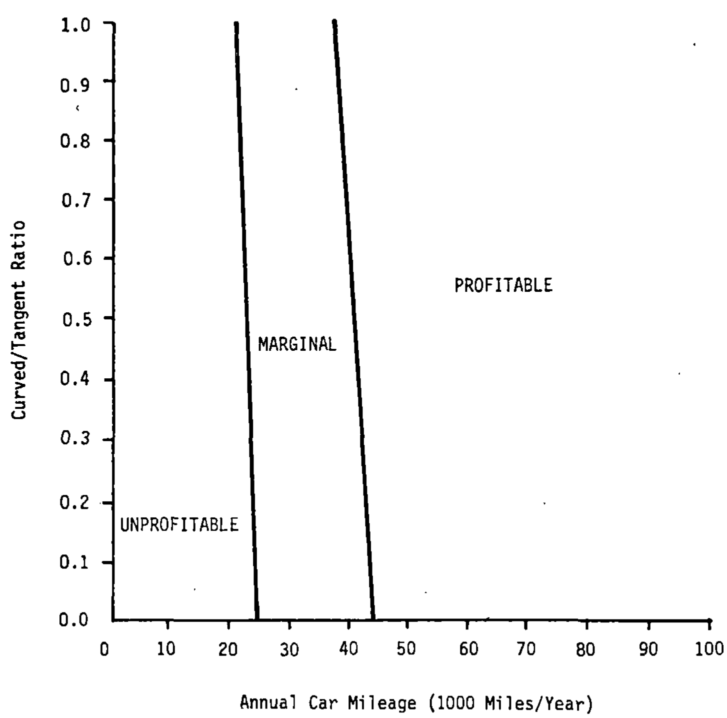


FIGURE 5-26. BOUNDARIES WITH CURVED TO TANGENT RATIO FOR PRIMARY SUSPENSION TRUCK SCENARIO

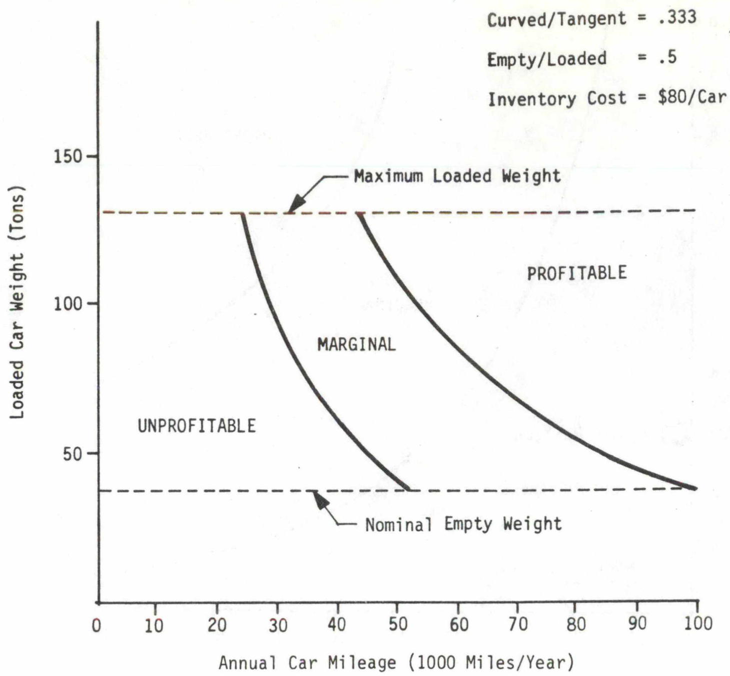


FIGURE 5-27. BOUNDARIES WITH CAR WEIGHT FOR PRIMARY SUSPENSION TRUCK SCENARIO

5.6.2 Procedure

The basic procedure used to assess the economic performance of Type II trucks is illustrated in Table 5-21. The approximate costs and weights of the Type I and Type II trucks are shown in Table 5-22. The data shown are estimated data for a steering truck using the economic methodology developed under Phase I of TDOP (References 11 through 13). The assumptions about the car and route are:

Empty weight	32 tons
Loaded weight	132 tons
Empty to loaded ratio	0.5
Annual mileage	80,000 miles/year
Curved/tangent ratio	0.667

Several other assumptions are made implicitly. Since the rail and fuel benefits have been included, this must be a captive service car (i.e., it only runs on the car owner's own line). No provision has been made to add a lading damage benefit, thus the lading is assumed to be insensitive, as for example, coal.

The definition for empty to loaded ratio is miles traveled empty/total miles traveled. The curve-to-tangent ratio is miles of curved track (any curvature)/miles of tangent track.

In Table 5-21, under the heading, Incremental Net Cash Investment Calculation, the incremental gross cash investment is shown as $3000/car. The next line shows the 10% investment tax credit of $300; the difference ($3000-$300) shown on the following line is the incremental net cash investment of $2700/car. This is the amount of extra money that the buyer would invest in the truck when it was purchased.

5.6.3 Annual Incremental Benefit Calculation

5.6.3.1. Car Maintenance Savings. Under the next heading in Table 5-21, Annual Incremental Net Cash Benefits Calculation, the first category is annual car maintenance savings (subsection 5.2 on car maintenance presented total car maintenance costs by part and by annual mileage). The equivalent annual expenditure data are reproduced here as Table 5-23. Appendix D presents the same results by 70 or 100-ton cars and could be used to further refine the estimates. However, for the purposes of this analysis, that degree of detail is not required.

a. Wheel Life. The largest single maintenance change from a Type II truck is expected to be a significant increase in wheelset life. The evidence from the TDOP Phase II performance testing and wear data collection program supports this conclusion. It would be possible to estimate the changes in wheelset life by using ratios of lateral curving forces from the performance testing; however, in this case better data are available from the wear program. The relevant table from the wear program (Reference 14) is reproduced here as Table 5-24. These estimates are developed by linear extrapolation of measured wear rates. Although better estimates are desirable, these estimates are used in the economic analysis methodology for lack of better available data. Also, since the objective is to demonstrate the methodology, these estimates are used without prejudice.

TABLE 5-21. WORST CASE BENEFIT/COST ANALYSIS OF A STEERING TRUCK

ASSUMPTIONS

Empty Weight	32	Tons
Loaded Weight	132	Tons
Empty to Loaded Ratio	0.500	
Annual Mileage	80,000	Miles/Year
Curved/Tangent Ratio	0.667	

INCREMENTAL NET CASH INVESTMENT CALCULATION

Incremental Gross Cash Investment		3000.00
Less: Investment Tax Credit of 10%		-300.00
Incremental Net Cash Investment		2700.00

ANNUAL INCREMENTAL NET CASH BENEFITS CALCULATION

Car Maintenance Savings:		
Wheel Life	133.71	
Steering Arm	-61.35	
Side Frame	-13.76	
Adapter	-28.35	
Inventory Adjustment	-5.00	
TOTAL	25.25	25.25
Roadway Maintenance Savings:		
Vertical Forces	110.30	
Curving Forces	333.90	
TOTAL	444.20	444.20
Fuel Savings		97.26
Gross Cash Benefits Before Depreciation		566.71
Depreciation ($3000./22.6 Years) (Non Cash Item)		-132.58
Gross Accounting Profit		434.13
Tax at 50%		217.06
Net Accounting Profit		217.06
Gross Cash Benefits Adjusted to Net Cash:		
Gross Cash Benefits	566.71	
Less Tax at 50%	217.06	
Annual Incremental Net Cash Benefits		349.64

NET PRESENT VALUE CALCULATION

Present Value of Benefits	
($349.64 x 8.80 P.V. of $1 at 10% for 22.6 Yr)	3075.67
Less: Incremental Net Cash Investment	-2700.00
Net Present Value	375.67

TABLE 5-22. WEIGHT AND COST OF TYPE II TRUCKS

Trucks	Weight per Car Set	Cost per Car Set
Type I	21,000 lb	$13,350
ACF Fabricated	20,944 lb	N/A
Alusuisse	N/A	N/A
Barber-Scheffel	23,000 lb	$21,300
Devine-Scales	24,000 lb	$21,000
DR-1	23,620 lb	$16,350
Maxiride	20,856 lb	$14,400
Swing Motion	22,850 lb	$15,767

*The weight and cost of a standard 3-piece truck, namely 21,000 lb and $13,350 for a Barber S-2 truck, have been added to the numbers quoted for the DR-1 steering arms in arriving at the given figures.

NOTE: All of the above figures are based on the best information available in TDOP Project files; the information on file has been gathered as submitted by the manufacturers on a voluntary basis. As far as can be determined, all weight and cost figures given above include one car set of two trucks with the associated sets of brake gear and wheelsets and other auxiliaries.

TABLE 5-23. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR UNION PACIFIC ROLLER BEARING CARS

TRUCK REPAIRS	NORMAL SERVICE			HIGH MILLAGE SERVICE				
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	67.90	148.99	234.00	228.40	295.08	358.06	425.81	485.99
BRAKE BEAMS	6.50	18.32	35.61	29.61	42.98	55.93	69.49	82.22
BRAKE BEAM WEAR PLATES	0.02	0.04	0.06	0.01	0.01	0.01	0.01	0.01
BRAKE HANGER OR CONNECTION PIN	0.30	0.66	1.03	0.95	1.21	1.44	1.71	1.94
BOTTOM ROD SAFETY SUPPORT	0.11	0.30	0.52	0.71	0.98	1.21	1.48	1.71
BRAKE BEAM SAFETY SUPPORT	0.03	0.08	0.11	0.01	0.01	0.01	0.01	0.02
BRAKE CONNECTION, BOTTOM	0.25	0.45	0.63	0.65	0.79	0.93	1.09	1.22
BRAKE CONNECTION, TOP	0.26	0.62	0.99	1.37	1.80	2.20	2.61	2.99
BRAKE LEVER	0.18	0.31	0.44	0.44	0.53	0.65	0.74	0.83
BRAKE LEVER GUIDE OR CARRIER	0.01	0.02	0.02	0.01	0.01	0.01	0.01	0.02
DEAD LEVER GUIDE	0.00	0.01	0.02	0.04	0.07	0.09	0.11	0.13
DEAD LEVER GUIDE BRACKET	0.00	0.01	0.02	0.05	0.07	0.08	0.10	0.12
BRAKE SHOES	59.82	127.32	193.31	193.80	245.66	294.36	347.07	398.28
BRAKE SHOE KEYS	0.41	0.83	1.22	0.76	0.96	1.15	1.38	1.52
WHEELSETS	112.50	244.78	367.50	344.32	427.57	506.56	593.82	669.46
LUBRICATE ROLLER BEARINGS	2.11	4.75	6.70	5.40	6.45	7.45	8.54	9.53
ROLLER BEARINGS	15.78	33.99	51.27	48.96	60.98	72.94	86.19	97.92
ROLLER BEARING CAP SCREWS	0.01	0.02	0.04	0.04	0.05	0.06	0.06	0.07
PEDESTAL ADAPTERS	2.14	7.16	12.14	8.28	10.80	13.15	15.71	17.98
WHEELS	37.81	82.03	124.18	117.38	146.09	174.69	206.80	234.30
WHEEL LABOR	54.10	115.61	171.34	162.51	201.03	235.72	273.63	306.17
AXLES, ROLLER BEARINGS	0.56	1.21	1.83	1.74	2.17	2.60	3.07	3.49
OTHER TRUCK REPAIRS	14.76	63.81	113.11	29.20	38.15	46.74	55.84	63.80
TRUCK BOLSTERS	4.12	19.21	34.76	7.55	10.00	12.40	14.81	17.03
TRUCK BOLSTERS (REPAIRED)	0.10	0.67	1.22	0.35	0.50	0.63	0.75	0.87
CENTER PINS	0.27	0.79	1.34	0.76	0.97	1.18	1.39	1.59
CENTER PLATES	0.17	0.48	0.97	0.05	0.07	0.09	0.11	0.13
CENTER PLATE LINERS	0.73	3.05	5.37	3.92	5.09	6.13	7.19	8.18
TRUCK SIDE BEARINGS	0.66	1.92	3.03	1.19	1.52	1.81	2.12	2.40
FRICTION CASTINGS	1.40	5.46	9.16	2.17	2.73	3.26	3.82	4.32
SIDE BEARING SHIM	0.03	0.23	0.39	0.08	0.10	0.12	0.15	0.17
SIDE FRAMES	5.25	23.72	43.01	7.14	9.88	12.57	15.54	18.00
SIDE FRAMES (REPAIRED)	0.07	0.50	0.83	0.06	0.08	0.10	0.12	0.13
SPRING PLANKS	0.00	0.00	0.00					
OUTER SPRINGS	0.80	3.30	5.48	1.89	2.21	2.54	2.89	3.17
INNER SPRINGS	0.45	1.74	2.83	1.14	1.33	1.53	1.73	1.90
STABILIZER SPRINGS	0.35	1.41	2.32	0.55	0.66	0.76	0.88	0.98
TRUCK SPRING FRICTION SNUBBER	0.00	0.01	0.01					
STEEL	0.01	0.02	0.03	0.29	0.37	0.43	0.53	0.60
MANUFACTURED MATERIAL (TRUCK)	0.33	1.32	2.37	2.07	2.64	3.19	3.81	4.33
TRUCK TOTAL	195.16	457.58	714.61	601.92	760.80	911.36	1075.16	1219.25

Based on Table 5-24, wheelset wear can be estimated in three ways: on the basis of high flange, on the basis of thin flange, or on the basis of metal removed. Using the Barber S-2-C truck as the Type I truck and the Dresser DR-1 as the Type II truck, the following comparisons can be made:

DR-1/S-2-C

High flange: 51.81/36.57 = 1.42

Thin flange: 96.23/38.03 = 2.53

Metal removed: .0374/.0208 = 1.80

It appears from Table 5-24 that the DR-1 truck wheelsets will ultimately wear out for high flange at around 518,100 miles. This is just one truck on one specific route; other trucks on other routes can be expected to behave differently. For this reason, the intermediate number of the three estimates was used (i.e., 1.8 based on metal removed).

At this point, it is necessary to estimate the percentage of wheelsets that fail for wear-related reasons. Table 5-25 shows a tabulation of wheel constructed. Other railroads have different distributions of wheelset replacements. To get a reasonably accurate estimate, it is recommended that the equivalent of Table 5-25 for the railroad in question be used to perform the analysis.

Simply tabulating why made codes for wheel replacements will not give an estimate of the percentage of wheels replaced due to wear. For example, most wheels are removed for why made code 11 - removed in good condition on account of associated repairs. TDOP's procedure for doing this tabulation is illustrated in Table 5-26. Individual records from the Car Repair Billing tapes (as shown at the top of the figure) were assembled into the record shown in the middle at the bottom of the figure. Reading across the bottom record, the data transferred is as follows:

- Car initial and number
- Wheel diameter and wheel wear (from the job code)
- L1 roller bearing why made code
- L1 wheel why made code
- 1 axle why made code
- R1 wheel why made code
- R1 roller bearing why made code
- Wear of the second axle's wheels

The repairs shown in Table 5-26 were obviously made for why made code 64 (i.e., high flange). After tabulating wheelsets in this way, causes of wheelset replacement can easily be identified as shown in Table 5-25.

The number of wheelsets that fail in the thin flange, high flange, and the mixed and other categories from the Union Pacific data in Table 5-25 is 41 percent of all wheelsets. Based on this result, the change in overall wheelset life can be calculated as follows:

.41 x 1.8 + .59 x 1 = 1.328 longer life

More sophisticated techniques for forming this estimate can easily be constructed. The chief advantage of this technique is that it is simple. However, one disadvantage is that it may tend to overstate the wheel life somewhat.

The equivalent annual wheelset repair cost shown in Table 5-23 for a Type I truck, is $506.56 at 75,000 miles/year and $593.52 at 87,500 miles/year. Interpolating this to the desired annual mileage of 80,000 miles/year gives $541.34/year. Thus, the reduced expenditure on wheelsets would be as follows: $541.34/1.328 = $407.64, annual saving of $541.34 - $407.64 = $133.70. This is the number shown on the wheel life savings line in Table 5-21.

b. <u>New Repair Categories, e.g., Steering Arms.</u> The next entry in Table 5-21 is an example of a new repair category that has not existed before in the Car Repair Billing system. If there are steering assemblies on a Type II truck, a new repair category for repairing the steering assemblies will have to be created. It is very difficult to estimate the expense associated with this type of change since there are no existing data. The steering arm is geometrically similar to a brake beam and is located in the same part of the truck. The same sorts of disassembly and labor should be involved. Also, brake beams are replaced fairly regularly. Brake shoes wear away and the brake beam grinds on the wheels, destroying itself. It is difficult to imagine that the steering assembly could cause more problems than the brake beams.

Using this logic, the cost of repairing steering assemblies was estimated to be the same as the cost of repairing brake beams. Table 5-23 shows that brake beam repairs average $55.93/year for a 75,000 mile/year car and $69.49/year for an 87,500 mile/year car. Interpolating to 80,000 miles/year one gets a cost of $61.35/year, which is the number shown in Table 5-21.

c. <u>Changes in Repair Frequency, e.g., Side Frames and Pedestal Adapters.</u> The comments on the Dresser DR-1 truck from the TDOP wear measurement report (Reference 14) indicate that the side frame pedestal adapter jaw and pedestal adapters are wearing approximately three times as fast as on the Type I trucks. This would appear to be correct since the DR-1 steering arms are attached to the pedestal adapter and rotate it to accomplish the steering.

Table 5-21 shows repair costs for side frames and pedestal adapters. Again the results were interpolated from Table 5-23. The side frame repairs were doubled and the adapter repairs were tripled; these were taken as cost items and included as negative entries in Table 5-21. The side frame number was doubled, not tripled, because there are other wearing surfaces on the side frame and all of the replacements probably do not come from the interface with the pedestal adapter.

d. Inventory Adjustment. The next extry in Table 5-21 is intended to represent the cost of stockpiling new types of parts along the railroad's line. In this case, DR-1 steering assembly kits would have to be purchased and distributed to the repair sites along the railroad. Each kit costs $3000. Assuming there are 12 places where the parts would be stored, an extra $36,000 would be required to purchase the required stockpile of parts. Since this money could be invested in something else with a 10% rate of return, stockpiling these parts will cost 10% of $36,000 or $3600 per year for as long as the stockpiles of parts are maintained.

At this point, the analysis becomes sensitive to the rate at which Type II trucks are purchased. If small numbers were purchased, the parts could be stored at less sites and the railroad would absorb an additional cost in lost car days waiting for the parts to arrive. Assuming that a large number of cars have been equipped with steering arms, it becomes profitable to distribute the parts to all sites. The cost shown in Table 5-21 (i.e., $5/car) corresponds to about 720 car sets equipped with steering trucks (i.e., $3600/year spread over 720 cars). For 100 cars purchased as an experiment to determine if the modification does pay for itself, a charge of about $36/car would be appropriate.

Calculating this charge correctly is rather complicated. For example, there are other expenses involved besides just the loss in revenue due to the capital tied up in stockpiling parts. It is necessary to actually buy the stockpile (i.e., to spend the $36,000) at some point. Also there are transportation and labor costs associated with moving the parts to the storage sites. Finally, at some point, the stockpiles will need to be increased from one assembly to two or more assemblies due to the demand for the parts. All this has been neglected here in the interests of simplicity. The inventory costs depend strongly on how many storage sites

TABLE 5-24. SUMMARY OF TEST TRUCK WHEEL WEAR

TRUCK	TYPE	SERVICE MILES	ANTICIPATED MILES BEFORE CONDEMNING LIMITS ARE REACHED BASED ON CURRENT TRENDS (X10,000 MILES)		NOTES
			HIGH FLANGE	THIN FLANGE	
National Swing Motion	II	125,701	73.57	43.13	1)
Barber S-2-C	I	100,094 (avg. wheel)	36.57	38.03	2)
Dresser DR-1	II	90,116	51.81	96.23	1)
Barber S-2 Heavy Duty		131,493	75.51	67.06	3)
ASF Ride Control	I	59,813	47.85	41.21	1)
Barber-Scheffel	II	92,709	60.74	111.25	1)
Devine-Scales	II	31,613	N/A	N/A	4)

NOTES:

N/A = NOT AVAILABLE

1) ALL ORIGINAL WHEELS CONTINUE IN SERVICE.

2) FOUR WHEELS REMOVED AT 83,000 MILES FOR SHELLED-OUT TREAD: TWO WHEELS REMOVED AT 116,000 MILES FOR GROOVED TREAD (WITH BRAKE SHOES NOT NORMAL TO CAR): TWO ORIGINAL WHEELS CONTINUE IN SERVICE.

3) ALL ORIGINAL WHEELS IN SERVICE UNTIL FEB. 5, 1981; TWO WHEELS REMOVED FOR CAUSES NOT RELATED TO WHEEL WEAR (LOOSE BEARING SEAL BACKING RINGS) AND REPLACED WITH NEW WHEELS FOR SERVICE CONTINUATION.

4) TRUCK REMOVED FROM PROGRAM AFTER 31,613 MILES.

ALL WHEELS CLASS "U" UNTREATED CAST STEEL, TWO WEAR TYPE CJ36 FOR FREIGHT SERVICE, EXCEPT BARBER-SCHEFFEL WHEELS WHICH ARE CLASS "U" UNTREATED CAST STEEL, TWO WEAR TYPE WITH SPECIAL PROFILE.

TABLE 5-25. UNION PACIFIC WHEELSET REPLACEMENT

	% ALL WHEELSETS	WHEELSET DOLLARS	ALL REPAIRS DOLLARS
• Thin Flange	23.6%	$ 433,154	$ 511,561
• Slid Flat	11.1%	$ 43,380	$ 68,038
• Mixed & Others	10.7%	$ 175,930	$ 209,661
• Car Set Derailment	9.4%	$ 5,146	$ 8,058
• Tread Buildup	9.3%	$ 115,015	$ 128,700
• Bearings & Axle	8.7%	$ 81,422	$ 109,581
• High Flange	8.1%	$ 130,979	$ 158,791
• Truck Set Derailment	5.9%	$ 11,687	$ 19,350
• Tread Shelled	3.7%	$ 37,789	$ 41,905
• Brakes Failed	3.7%	$ 73,679	$ 83,560
	94.2%	$1,108,181	$1,339,205

TABLE 5-26. WHEELSET FAILURE IDENTIFICATION METHODOLOGY

All Repairs to :

Railroad		Car	Date	Site	Loc	Why Made	Job Code	
1	CN	UP 960109	770616	043226	R1	64	3085	WHEEL, 36" 2W STEEL
1	CN	UP 960109	770616	043226	R1	11	2816	ROLLER BEARING, GROU
1	CN	UP 960109	770616	043226	L1	11	2816	ROLLER BEARING, GROU
1	CN	UP 960109	770616	043226	L1	64	3085	WHEEL, 36" 2W STEEL
1	CN	UP 960109	770616	043226	1	09	3160	WHEEL LABOR, ROLLER
1	CN	UP 960109	770616	043226	1	11	3276	AXLE-RWS-ROLLER BRG

		Axle 1		Axle 2		Axle 3		Axle
UP 160406	33 2	11:64-11-11 :11	2	93:64-11-11:11	2	11:64-11-11:11		: - -
UP 960109	36 2	11:64-11-64:11		: - - :		: - - :		:
UP 960195	33 M	11: 11-11-64:11		: - - :		: - - :		: - -

exist along a given railroad. For this reason, inventory costs by railroad are probably highly variable.

e. Total Car Maintenance Savings. The wheel life savings minus the added car maintenance costs gives the \$25.25/year savings shown in Table 5-21. Considerably more sophistication could easily be employed to form this estimate. This is particularly true of the wheelset life estimate and the inventory analysis. During the early stages of TDOP Phase II there were plans to build a statistical wheelset life model based on the large amounts of car maintenance data available. Procedures for doing this were developed and checked out. Also there were plans to build an inventory control model of the Union Pacific Railroad. However, as it became obvious that car maintenance simply was not going to pay for a Type II truck, it was decided to invest the remaining time and resources in fuel and rail savings rather than in further refining the car maintenance methodology.

5.6.3.2. Roadway Maintenance Savings. The methodology for estimating savings in rail life was discussed in subsection 5.4 on roadway maintenance and is summarized here. The Canadian Institute of Guided Ground Transportation performed the analysis on which the estimates here are based for a Track-Train Dynamics study (References 7 and 8). CIGGT has agreed to undertake a similar study based on TDOP Phase II test data. This effort is to be funded by Transport Canada and the results are to be made public through the joint information exchange agreement between the U.S. and Canada. The results should be available through NTIS at the end of 1981.

The CIGGT estimates are in two parts: savings from a reduction in vertical dynamic loads on all track curvatures and savings from a reduction in static lateral loads during curving. In both cases, the dollar savings per mile of track in several curvature categories, including estimated savings from rail, ties, and "other" track costs, are considered. The two relevant tables from the roadway maintenance subsection are Tables 5-18 and 5-19. In order to convert these estimates to an annual savings per car, it is necessary to considerably adjust the assumptions on which the analysis is based. Since the CIGGT analysis is highly nonlinear, the result should be viewed with some skepticism. It would be preferable to refer to the on-going CIGGT analysis of the TDOP performance data when it becomes available later in the year.

a. Vertical Dynamics. The next item in Table 5-21 is savings in rail wear from decreased vertical loads on tangent track. This savings is based on the vertical dynamic load results from CIGGT shown in Table 5-18. Test results from TDOP performance testing were given in the subsection on roadway maintenance (see Table 5-17). Referring to the Dresser DR-1 result, there was 9 percent reduction in vertical dynamic loads measured during TDOP testing.

Referring to the CIGGT results in Table 5-18, the bottom line equivalent annual benefit for tangent track is \$555/mile and the corresponding benefit for the 2 - 5 degree category is \$919/mile. These were the only two track categories used for this analysis.

The procedure for converting the data to a different set of assumptions was illustrated in subection 5.4 with the example of a 75,000 mile/year unit coal train. The train contains 100 cars and averages one trip out and back each week. It can be shown algebraically that the number of trips or cars assumed has no effect on the ultimate savings. In general, the procedure is as follows:

Tangent track savings/year (\$/year) =

$$37 \times 10^{-6}\ sA\ \{W_E e + W_L (1-e)\} / (c + 1)$$

Curved track savings/year (\$/year) =

$$61.3 \times 10^{-6}\ s\ c\ A\ \{W_E e + W_L (1-e)\} / (c + 1)$$

Where:

W_E	=	the empty weight in tons
W_L	=	the loaded weight in tons
e	=	the empty/loaded ratio
A	=	the annual mileage
c	=	the curved/tangent ratio
s	=	the ratio of the reduction of the dynamic vertical loads (i.e., 9 percent as shown in Table 5-18) to the value assuming in calculating the CIGGT data (i.e., 25 percent).

Evaluating these expressions with the assumptions presented at the top of Table 5-21 and adding them together gives the result shown in the table as the vertical dynamic load savings (i.e., \$110.30).

b. Curving Dynamics. The next line in Table 5-21 is the savings due to reduction in the static lateral curving force of a car in a curve. It is calculated in the same way as the savings for the vertical dynamic loads except that the CIGGT data is from Table 5-19. In this case, savings are only experienced on curved track. Using a reduction in lateral force of 1/3 (i.e., the force is 2/3 of what it was for Type I), the result shown in Table 5-21 (\$333.90) is obtained.

c. Total Roadway Maintenance Savings. Adding the two categories, the total annual saving for the track is \$444.20. These results undoubtedly will be revised when

CIGGT finishes its analysis of the TDOP Phase II performance test data. The treatment of the several nonlinear effects in the analysis illustrated above can be improved in a more complex and thorough analysis. For example, there is considerably more effect from varying the empty/loaded ratio than just changing the MGT the car puts on the rail. Separating the rail wear data of empty cars from loaded cars would allow this to be considered.

5.6.3.3 Fuel Savings. Fuel savings were discussed earlier in subsection 5.4. The results are summarized in the table showing fuel savings per mile, reproduced here as Table 5-27. To convert these savings to annual savings, it is necessary to re-scale them to the specific situation involved. In general, the following equation can be used:

$$.85 \ s \ A \ e \ W_E + (1-e) \ W_L / 64$$

Where:

- .85 = price per gallon of diesel fuel
- s = interpolated savings from Table 5-27 based on curved to tangent ratio
- A = car's annual mileage
- e = empty/loaded ratio
- W_E = empty weight of the car
- W_L = loaded weight of the car
- 64 = average tonnage on which the analysis for Table 5-27 was based

Because the price of fuel has been increasing more rapidly than overall inflation, this value should be inflated to reflect the net present value of the increase in fuel prices. Put another way, the price today of diesel fuel probably does not reflect the true savings. Referring to the tables from the fuel consumption subsection, the value of the savings at a 10 percent discount rate and at a 6 percent discount rate are shown. Taking the ratio between the two sets of values, the following table by annual mileage was developed:

Annual Mileage	Scale Factor
12,500	1.46
25,000	1.46
37,500	1.46
50,000	1.46
62,500	1.45
75,000	1.40
87,500	1.36
100,000	1.32

Interpolating from the table and multiplying times the savings from the equation gives the total annual savings ($97.26/year) from fuel shown in Table 5-21.

5.6.3.4 Tax Adjustment. Adding all the benefits from the line items above gives a gross cash benefit of $566.71, which represents the additional money at the end of the year if all the estimated savings actually were realized.

Since the equipment costs more, it will have a higher depreciation value. Taking the same estimated life as was used all through the calculation, and assuming straight line depreciation, one would depreciate the extra $3000 investment at $132.58/year. This tax shield reduces the amount of taxes that have to be paid. Subtracting it from the gross cash benefits gives the accounting profit ($434.13) which is the amount of money on which corporate taxes are paid. At a 50 percent tax rate, $217.06 would appear on the books as profit.

Going back to the $566.71 gross cash benefits, if $217.06 was paid to the government in taxes, then $349.64 must have been retained as cash by the railroad. Thus the annual incremental net cash benefits are $349.64. Every year the Type II truck operates, the railroad is $349.64 ahead.

5.6.4 The Net Present Value Calculation

At this point it is necessary to compare the railroad's cost of the improvement to the annual benefits. The accepted procedure for doing this is to compute the present value of the annual benefits. Calculating the present value of a dollar over 22.6 years at a 10 percent rate, each dollar of annual savings is worth $8.80 of expenditure today. Multiplying the $349.64/year by $8.80 gives a net present value of $3075.67, the amount that could be spent to break even on the investment. In this case, all that was spent was $2700, so the railroad stands to make $375.67 on this investment.

TABLE 5-27. FUEL SAVINGS PER MILE

	GALLONS SAVED/CAR MILE BY CURVE/TANGENT RATIO				
CURVE/TANGENT	1.105	0.737	0.368	0.185	0.124
ACF FABRICATED	0.00086	0.00069	0.00038	0.00016	0.00010
ALUSUISSE (70 TONS)	-0.00024	-0.00030	-0.00019	-0.00015	-0.00008
BARBER-SCHEFFEL	0.00157	0.00113	0.00015	-0.00017	-0.00031
DEVINE-SCALES	0.00196	0.00126	0.00009	-0.00041	-0.00055
DRESSER DR-1 TRUCK	0.00140	0.00094	0.00024	-0.00013	-0.00023
MAXIRIDE TRUCK	-0.00020	-0.00019	-0.00004	-0.00004	0.00004

5.6.5 Remarks

The analysis outlined above is intended as an illustration and is not meant to be an endorsement of the truck involved or even a fair assessment of its relative merits. Some simplifications were made in the interests of automating the results (e.g., using only curved and tangent track categories in the roadway analysis). Further information at higher mileages should be available at a later date from the TDOP wear program which is ongoing under the sponsorship of the FRA. As better information becomes available, this analysis could well need revision.

This economic analysis contains several very conservative assumptions that should be reviewed in light of further data. There is no effect of curve-to-tangent ratio on the car maintenance analysis. However, there is obviously an effect in the case of wheel life. A further refinement might be to assume that thin flange wheel life varies in the same way with curvature as rail wear on curved track (the rail is the other wearing surface).

The assumption that a steering arm will have repair costs similar to a brake beam is obviously a worst-case assumption. Further data from the wear measurement program, or from any of the unit trains equipped with steering trucks that are now in service, will provide additional information on the actual costs. There are obviously going to be repairs done on the nonstandard equipment. The issue is how serious a problem these repairs will be. The assumed rates of wear of the adapter and side frame should similarly be reviewed as data become available.

The inventory analysis offered is probably not as conservative as it ought to be. The inventory costs preclude the profitable operation of a small number of premium trucks. If a railroad is going to buy these trucks, it should have some strategy for controlling the inventory cost of the new parts. For example, parts could be distributed only along the route the cars are intended to take, or the nonstandard parts could be repaired only at one facility on a regular schedule.

Other savings seem possible. In keeping with the worst-case assumption about the steering arm, there is no provision for decreases in the number of lost car days. Data on the performance of these trucks are not available at this time. As it becomes available, some effect from lost car days should be considered. At this point, lost car days could be increased or decreased depending on what kind of maintenance problems are uncovered.

The preceding analysis was set up in an attempt to demonstrate in which situations, (such as specific curve-to-tangent ratios or annual mileages), a specific kind of Type II (premium) truck, such as a steering truck or a primary suspension truck, might pay for itself. A better analysis of an actual situation could be made. For example, fuel savings for an actual, rather than a hypothetical, unit train could be calculated with train performance calculators. The curving resistance data on which to base such an analysis was provided in subsection 5.4 and would provide a much more accurate estimate than the procedure for calculating fuel savings outlined in that section. On the other hand, analysis of an actual situation is tedious and expensive. It is fairly easy to identify the cases in which a detailed analysis might be appropriate by following the steps given in the example. If the result looks promising, for example, if the net present value does not show a loss greater than 50 percent of the added investment cost, it is worthwhile considering a detailed analysis in the larger benefit areas.

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SECTION 6
CONCLUSIONS AND RECOMMENDATIONS

The major results obtained from experimental and analytic studies undertaken during TDOP Phase II are:

- Definition of the performance characteristics of Type I trucks.
- Development of performance specifications for Type II trucks.
- Development of guideline field and laboratory test specifications for freight car trucks.
- Development of a methodology for truck evaluation.
- Development and implementation of a field test program to collect wear data on freight car trucks.
- Establishment of a plan for collecting economic data on costs of acquiring, operating, and maintaining freight car trucks.
- Development of an economic methodology for the evaluation of costs and benefits associated with improved designed freight car trucks.

Some conclusions, on the basis of the results, arrived at through the engineering and economic studies under the program are summarized below.

- The improved design features in the Type II trucks achieve a degree of qualified success in attaining improved performance from freight car trucks. These successes, however, are limited to some of the domains of performance rather than comprehensive, all-around improvement in all aspects.
- One of the significant engineering findings from the field test data related to a definite trend of asymmetry with regard to wheel/rail lateral forces. As the various test trucks traversed a curve in the test zone, they almost uniformly experienced lateral forces consistently higher or lower depending on the orientation of the curve in a left- or right-handed sense. Analysis of data from six independent channels has confirmed a definite trend with respect to such asymmetry. Further detailed studies are necessary to determine the sources which give rise to these asymmetric trends.
- Performance evaluation of freight car trucks needs to be undertaken under well-defined sets of conditions relating to the state of wear and deterioration of vehicle and track structure in order to address fully all aspects of performance. For example, wheel and rail contact geometry, to which vehicle performance is extremely sensitive, needs to be thoroughly documented through the bulk of the wheel and rail life cycles and their representative conditions used in any comprehensive evaluation program.
- On the basis of the analysis of available car maintenance data, costs associated with car maintenance alone do not warrant or justify the levels of increased capital investment demanded by the Type II trucks. On the other hand, improved rolling and curving resistance and consequent reduction in fuel consumption seem to be very promising areas which indicate that the additional investments warranted by the Type II trucks could be advantageous. Specific considerations, such as an intermodal scenario, also point to an advantageous outlook with respect to investment in Type II trucks with rigidized frame/primary suspension features.
- Significant economic benefits from the utilization of Type II trucks seem to accrue more in the area of the track structure, in general, and the rail, in particular. Reduced rail wear as well as retardation in rail and track structure deterioration are potential benefits from improved truck prformance. These economic implications, if properly accounted for through a systematic rail wear and track deterioration study, could be significant.

The project treated all Type II trucks as a single group, in terms of evaluating them relative to the Type I trucks. However, within the Type II truck group, significant design features, such as self-steering and rigid-frames, set some of the trucks apart from the others. Each of these groups, e.g., self-steering, rigid, primary suspension, and secondary suspension, seek to achieve a definite set of objectives with respect to performance improvements. For example, the objective of a rigid frame truck is to eliminate hunting, rather than to improve curve negotiability. Therefore, an evaluation program geared to address specific design features in the context of stated performance improvement objectives would be likely to yield more responsive results. Furthermore, if it can be shown that a specific design feature does succeed in its objectives, then the industry could attempt to incorporate more than one, if not all, of these features into a single design as an integrated suspension system which then could lead to overall improvement in freight car truck performance. The engineering and economic methodologies developed in TDOP Phase II provide a framework for the evaluation of the freight car trucks.

APPENDIX A
PHASE I TEST EQUIPMENT AND CONDITIONS

INTRODUCTION

This appendix contains a brief summary of the equipment tested during TDOP Phase I to generate field test data.

TEST TRAIN

The test train was made up of a locomotive, the SP-250 instrument car, the test car, and a caboose, in that order. This consist reflects the intent to study freight car truck performance with the test car approximating a free body with no extraneously imposed longitudinal forces.

EQUIPMENT TESTED

The cars tested were a 70-ton mechanical refrigerator car, a 70-ton boxcar, a long, low-level "stac-pac" flatcar, a 100-ton boxcar, and a 100-ton covered hopper car. The data on these cars are given in Table A-1.

Trucks tested were 70-ton ASF Ride Control trucks, 70-ton Barber S-2-C trucks, 70-ton ASF low-level Ride Control trucks, and 100-ton Barber S-2-C trucks. The data on these trucks are given in Table A-2.

Wheel profiles used in the Phase I test program, data from which were used in quantifying performance characterizations under the Phase II effort, are listed below.

- CM-33 1/20 taper profile wheels on the 70-ton ASF Ride Control and Barber S-2-C trucks
- CM-33 worn profile wheels on the 70-ton ASF Ride Control trucks
- CJ-36 1/20 taper profile wheels on the 100-ton ASF Ride Control and Barber S-2-C trucks
- CD-28 1/20 taper profile wheels and CB-28 worn profile wheels on the 70-ton ASF Ride Control low-level trucks
- CM-33 TDOP cylindrical profile wheels on the 70-ton ASF Ride Control trucks
- CJ-36 TDOP cylindrical profile wheels on the 100-ton Barber S-2-C trucks

TEST TRACK

High-speed jointed rail test track consisted of a 7.8-mile westbound section of track between Suisun-Fairfield and Bahia (MP 48.5 to 40.7). This track has alternate staggered rail joints of 39-foot, 132-pound per yard rail.

Medium-speed jointed rail test track consisted of a 5-mile section of the Schellville branch beginning near Cordelia and ending near Suisun-Fairfield. This is a section of alternately staggered joints of 39-foot, 132-pound per yard rail (second-hand rail within serviceable limits).

A 3.3 mile section of track in Niles Canyon (MP 30.5 to 33.8) was selected for curve negotiation testing. The test track consisted of 12 curves ranging in curvature from one degree to nine degrees.

A short section of the Schellville branch near Lombard was selected for distortion by instituting 0.75-inch cross level differences at the rail joints.

The track geometry cars were used to measure and record track characteristics at the high-speed and medium-speed test sites. The track geometry measured included profile, alignment, gage, cross level and curvature.

TEST MATRICES

The test matrices for high-speed and medium-speed jointed track used during Phase II in quantifying the performance characteristics of Type I trucks are given in Tables A-3 and A-4. For the shimmed track test, a loaded 70-ton mechanical refrigerator car equipped with a 70-ton ASF Ride Control truck, and a loaded 100-ton boxcar equipped with a 100-ton Barber S-2-C were used. The two test trucks were equipped with cylindrical wheels.

INSTRUMENTATION

The various test cars were instrumented to obtain information for quantifying ride quality, and for measuring track input, track energy transmission through the truck, and movement between truck components. These objectives were accomplished by application of displacement transducers, accelerometers, and force transducers at strategic locations on the trucks. To obtain information on reaction of the carbody, accelerometers were placed at optimum locations to record body movement.

Truck-mounted instrumentation was heavily concentrated on the B-end truck, which was the leading truck in the direction of motion during all tests. A lesser amount of instrumentation was on the trailing truck.

TABLE A-1. CARBODY CHARACTERISTICS

	70-Ton Capacity Mechanical Refrigerator Car	70-Ton Capacity General Service Boxcar	70-Ton Capacity Long Low-Level Flatcar	100-Ton Capacity Auto-Parts Boxcar	100-Ton Capacity Covered Hopper Car
Light Weight, lb	89,100	61,200	56,300	87,300	64,500
Capacity, lb	130,900	154,000	122,000	174,000	197,500
Length Over Pulling Face of Coupler, ft	63.70	55.38	93.67	68.25	54.29
Truck Centers, ft	45.72	40.00	64.00	46.25	40.83
Car Wheel Base, ft	51.39	46.83	69.08	52.08	46.25
Overhang, ft	9.00	7.29	14.83	11.00	7.29
Center of Gravity-Loaded, ft	7.33	7.03	7.17	7.83	7.03
Center of Gravity-Empty, ft	5.55	4.58	1.97	5.17	4.58
Centerplate Diameter, ft	1.17	1.17	1.17	1.33	1.25

TABLE A-2. TRUCK CHARACTERISTICS

	70-Ton ASF Ride Control Truck	70-Ton Barber S-2-C Truck	70-Ton ASF Low-Level Truck	100-Ton ASF Ride Control Truck	100-Ton Barber S-2-C Truck
Wheel Base, ft	5.67	5.67	5.08	5.83	5.83
Wheel Diameter, ft	2.75	2.75	2.33	3.00	3.00
Bolster Centerplate Diameter, ft	1.15	1.17	1.17	1.25	1.33
Centerplate Height, ft	2.15	2.15	1.68	2.07	2.15
Weight, lb	9,080	9,100	7,600	10,540	10,560
Gross Rail Load, lb	220,000	220,000	179,000	263,000	263,000
Vertical Spring Rate (Per Car), lb/in	94,466	89,653	97,450	108,333	109,367
Lateral Spring Rate (Per Spring Nest), lb/in	4,665(at 9.47")* 7,795(at 7.56")*	3,470(at 9.47") 9,080(at 7.56")	4,755(at 9.06") 12,015(at 8.31")	3,655(at 9.47") 9,560(at 7.56")	2,705(at 9.47") 10,285(at 7.56")
Friction Snubber Column Load, lb	3,140	Variable (Load-Dependent)	3,110	4,510	Variable (Load-Dependent)
*Spring Nest Height					

TABLE A-3. HIGH-SPEED JOINTED TRACK TEST MATRIX

TDOP PHASE I TEST MATRIX USED DURING PHASE II ANALYSIS

⊡ **TEST DATA AVAILABLE**
☐ **NO TEST CONDUCTED**

Truck	Carbody	Empty		Loaded	
		Wheel Profile			
		New AAR 1/20	Worn	New AAR 1/20	Worn
70-Ton ASF Ride Control	Refrigerator Car	●	●	●	●
70-Ton Barber S-2-C	Refrigerator Car	●		●	
	70-Ton Boxcar	●		●	
70-Ton Low Level ASF Ride Control	89-ft Flatcar	●	●	●	
100-Ton Barber S-2-C	100-Ton Boxcar	●		●	
100-Ton ASF Ride Control	100-Ton Covered Hopper Car	●		●	

TABLE A-4. MEDIUM-SPEED JOINTED TRACK TEST MATRIX

TDOP PHASE I TEST MATRIX USED DURING PHASE II ANALYSIS

⊡ **TEST DATA AVAILABLE**
☐ **NO TEST CONDUCTED**

Truck	Carbody	Empty		Loaded	
		Wheel Profile			
		New AAR 1/20	Worn	New AAR 1/20	Worn
70-Ton ASF Ride Control	Refrigerator Car	●	●	●	●
70-Ton Barber S-2-C	Refrigerator Car	●		●	
	70-Ton Boxcar	●		●	
70-Ton Low Level ASF Ride Control	89-ft Flatcar	●		●	
100-Ton Barber S-2-C	100-Ton Boxcar	●		●	
100-Ton ASF Ride Control	100-Ton Covered Hopper Car	●		●	

APPENDIX B - TEST TAPES

The following test data tapes are available through the National Technical Information Service.

TDOP Phase I, Type I Truck Test Data Tapes

- NTIS Accession No. PB 250 163 through 345/AS.

TDOP Phase II, Type I Truck Test Data Tapes

- FRA/ORD/MT - 81/12.1 through 81/12.16

- NTIS Accession No. PB 81 181695, 1703, 1711, 1729, 1737, 1745, 1752, 1760, 1778, 1786, 1794, 1802, 1810, 1828, 1836, 1844.

TDOP Phase II, Type II Truck Test Data Tapes - FRA/ORD/MT - 81/38-I through VII

I - Dresser DR-1

II - National Swing Motion

III - Barber-Scheffel

IV - MTS Maxiride 100

V - Devine-Scales

VI - ACF Fabricated

VII - Alusuisse

TDOP Phase II, Friction Snubber Force Measurement System Test Data Tapes

- FRA/ORD/MT-80/72-126

- NTIS Accession No. PB 81 122764, 772, 788, 798, 806, 814

APPENDIX C

DATA REDUCTION EQUATIONS

WHEEL/RAIL FORCES

Typical Vertical Axle Bending Moment Calculations:

$$(VA) = \left\{\left[\left(RMS_{R1}/RMS_{G116}\right) G116\right]^2 + \left[\left(RMS_{R1}/RMS_{G112}\right) G112\right]^2\right\}^{\frac{1}{2}}$$

$$(VB) = \left\{\left[\left(RMS_{R1}/RMS_{G115}\right) G115\right]^2 + \left[\left(RMS_{R1}/RMS_{G111}\right) G111\right]^2\right\}^{\frac{1}{2}}$$

$$(VC) = \left\{\left[\left(RMS_{R1}/RMS_{G113}\right) G113\right]^2 + \left[\left(RMS_{R1}/RMS_{G109}\right) G109\right]^2\right\}^{\frac{1}{2}}$$

$$(RV1) = \left[(VA) + (VB) + (VC)\right] /3$$

Where RMS_{R1} is the average RMS value for the bending moment gages located at the right side of axle 1 (R1) and RMS_{G116} is the RMS value for gage G116.

Similar calculations were made for (LV1), (RV2), and (LV2).

PRIMARY SPRING VERTICAL DISPLACEMENTS (MAXIRIDE)

X_{R1} = 0.8333 D15 + .0834 (D15 + D17)

X_{L1} = 0.8333 D17 + .0834 (D15 + D17)

X_{R2} = 0.8333 D16 + .0834 (D16 + D18)

X_{L2} = 0.8333 D18 + .0834 (D16 + D18)

Using these displacements, vertical forces and moments were calculated using nonlinear spring constants provided by the manufacturer.

LATERAL AND VERTICAL FORCE CALCULATIONS

FVR1 = 1500. + .03333 [(RV1) - (LV1)] + VLA1

FVL1 = 1500. - .03333 [(RV1) - (LV1)] + VLA2

FLR1 = 156.45 - .05556 x BMA1 + .05556 x (LV1) + 0.081944[(RV1) - (LV1)]

FLL1 = 156.45 - .05556 x BMA2 + .05556 x (RV1) - 0.081944 [(RV1) - (LV1)]

QUR1 = FLR1/FVR1

VLA1 is the vertical load on bearing adapter number 1 (R1) and is determined either from an instrument bearing adapter or from the primary spring displacements.

QUL1 = FLL1/FVL1

AXL1 = FLR1 - FLL1

AXV1 = FVR1 + FVL1

Same calculations are repeated for axle 2.

WHEEL UNLOADING INDEX

FVT = FVR1 + FVL1 + FVR2 + FVL2

MINV = Minimum of (FVR1, FVL1, FVR2, FVL2)

WUI = 1 - 3 x MINV/ (FVT - MINV)

ANGLE OF ATTACK

LRS1 = .5 (P2 + P4)

LWS1 = .5 (P1 + P3)

LWR1 = LWS1-LRS1

LRS2 = .5 (P6 + P8)

LWS2 = .5 (P5 + P7)

LWR2 = LWS2-LRS2

ARS1 = C1 x (P2-P4)

AWS1 = C2 x (P1-P3)

AWR1 = AWS1-ARS1

ARS2 = C1 x (P6-P8)

AWS2 = C2 x (P5-P7)

AWR2 = AWS2-ARS2

See Table C-1 for values for C1 & C2.

TRUCK AND TRUCK/CARBODY MOTIONS

SWIV = C3 (D13 - D14)

TRAM = C4 (D6 - D5)

SGVD = (D1 + D2 + D3 + D4)/4.

SGRL = C5 (D1 + D2 - D3 - D4) or C5 (D15 + D16 - D17 - D18)

CBBL = C6 (D11 - D12)

CBSF = SGRL + CBBL

CARBODY MOTIONS

PTCH = C7 (A1 - A2)

VERT = 0.5 (A1 + A2)

AROL = C8 (A2 - A4)

BROL = C9 (A7 - A3)

ROLL = 0.5 (AROL + BROL)

TWST = BROL - AROL

ARLL = C10 x (A16 - A6) + C11 x YAW

BRLL = C12 x (A15 - A5) - C13 x YAW

RLLL = 0.5 (ARLL + BRLL)

LAT = C14 x (A5 + A6) + C15 x (A15 + A16) (EMPTY)

LAT = C16 x (A5 + A6) + C17 x (A15 + A16) (LOADED)

YAWB = C18 x (A5 - A6)

YAWT = C19 x (A15 - A16)

YAW = 0.5 (YAWB + YAWT)

Table C-2 gives the coefficients for carbody motions.

NOMENCLATURE

SWIV	-	Truck swivel rotation (carbody to bolster)
TRAM	-	Truck tram rotation (bolster to side frame)
SGVD	-	Spring group vertical displacement
SGRL	-	Spring group roll angle
CBBL	-	Carbody - bolster roll angle
CBSF	-	Carbody - side frame roll angle
VA	-	Vertical axle bending moment from the first pair of quadrature gages
VB	-	Vertical axle bending moment from the second pair of quadrature gages
VC	-	Same as VA except third pair of gages
RV1	-	Vertical axle bending moment for the gages near the right wheel of axle 1
LV1	-	Same as RV1 except left wheel
RV2	-	Same as RV1 except axle
LV2	-	Same as LV1 except axle 2
VLA1	-	Vertical load on bearing adapter #1 (R1)
VLA2	-	Vertical load on bearing adapter #2 (L1)
VLA3	-	Vertical load on bearing adapter #3 (R2)
VLA4	-	Vertical load on bearing adapter #4 (L2)
BMA1	-	Bending moment due to VLA1
BMA2	-	Bending moment due to VLA2
VMA3	-	Bending moment due to VLA3
BMA4	-	Bending moment to VLA4
FVR1	-	Vertical wheel/rail force - R1
FVL1	-	Vertical wheel/rail force - L1
FVR2	-	Vertical wheel/rail force - R2
FVL2	-	Vertical wheel/rail force - L2
FLR1	-	Lateral wheel/rail force - R1
FLL1	-	Lateral wheel/rail force - L1
FLR2	-	Lateral wheel/rail force - R2
FLL2	-	Lateral wheel/rail force - L2
QUR1	-	L/V ratio - R1
QUL1	-	L/V ratio - L1
QUR2	-	L/V ratio - R2
QUR2	-	L/V ratio - L2
AXL1	-	Total lateral wheel/rail force on axle 1
AXL2	-	Total lateral wheel/rail force on axle 2
AXV1	-	Total vertical wheel/rail force on axle 1
AXV2	-	Total vertical wheel/rail force on axle 2
X_{R1}	-	Primary spring displacement, R1 spring group
X_{L1}	-	Primary spring displacement, L1 spring group
X_{R2}	-	Primary spring displacement R2 spring group
X_{L2}	-	Primary spring displacement, L2 spring group
FTV	-	Total vertical wheel/rail force for B-end truck
MINV	-	Minimum vertical wheel/rail force for four wheels of B-end truck
WUI	-	Wheel unloading index, equal to zero implies all four wheels have equal load, equal to one implies one wheel has no load
LRSi	-	Lateral displacement of rail relative to side frame for axle i, i = 1,2
LWSi	-	Lateral displacement of wheel relative to side frame for axle i, i = 1,2
LWRi	-	Lateral displacement of wheel relative to rail for axle i, i = 1,2
ARSi	-	Angular displacement of rail relative to side frame for axle i, i = 1,2
AWSi	-	Angular displacement of wheel relative to side frame for axle i, i = 1,2
AWRi	-	Angular displacement of wheel relative to rail for axle i, i = 1,2
PTCH	-	Carbody pitch acceleration

VERT	- Carbody vertical acceleration	BRLL	- Carbody B-end roll acceleration (from lateral accelerometers)
AROL	- Carbody A-end roll acceleration (from vertical accelerometers)	RLLL	- Carbody roll acceleration (from lateral accelerometers)
BROL	- Carbody B-end roll acceleration (from vertical accelerometers)	LAT	- Carbody lateral acceleration at CG
ROLL	- Carbody roll acceleration (from vertical accelerometers)	YAWB	- Carbody yaw acceleration near bottom of carbody
TWST	- Carbody twist acceleration (from vertical accelerometers)	YAWT	- Carbody yaw acceleration near top of carbody
ARLL	- Carbody A-end roll acceleration (from lateral accelerometers)	YAW	- Carbody yaw acceleration near center of carbody

TABLE C-1. CALCULATION COEFFICIENTS - TRUCK AND TRUCK/CARBODY MOTIONS

COEFFICIENT	TRUCK							
	ASF-100	DR-1	Swing Motion	Barber-Scheffel	Maxiride 100	Devine-Scales	ACF	Alusuisse
C1	126.17	126.17	126.17	126.17	122.79	122.79	122.79	N/A
C2	171.90	171.90	171.90	171.90	158.07	158.07	125.59	N/A
C3	2.117	2.117	2.491	2.491	2.491	2.451	2.547	2.388
C4	2.12	2.12	2.08	2.08	N/A	N/A	N/A	N/A
C5	0.2812	0.2979	0.2979	0.2979	0.2979	0.2979	0.3247	0.3247
C6	0.5715	0.5715	0.5715	0.5715	0.5715	0.5715	0.5715	0.626

TABLE C-2. CALCULATION COEFFICIENTS - CARBODY MOTIONS

COEFFICIENT	100-TON HOPPER CAR	70-TON HOPPER CAR
C7	45.1	53.9
C8	402.1	402.1
C9	199.0	206.7
C10	207.0	271.0
C11	0.450	0.674
C12	209.0	277.3
C13	0.456	0.690
C14	0.470	N/A
C15	0.0298	N/A
C16	0.305	0.358
C17	0.195	0.142
C18	45.32	54.48
C19	37.81	42.86

APPENDIX D
CAR REPAIRS USING 10% DISCOUNT RATE

TABLE D-1. PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR ALL ROLLER BEARING CARS

				ANNUAL MILEAGE				
SUMMARY TABLE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	354.80	752.57	1135.20	1507.36	1860.20	2124.51	2353.66	2551.77
COUPLERS, YOKES, & DRAFT GEAR	122.58	413.23	737.50	1065.55	1383.71	1636.69	1863.00	2064.71
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	368.40	1104.62	1754.38	2336.07	2862.37	3269.19	3617.06	3902.89
OTHER CAR REPAIRS	267.34	701.02	1198.26	1711.28	2241.00	2577.00	2897.22	3174.14
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	640.05	1404.51	2205.92	2992.85	3754.07	4351.23	4869.21	5329.26
WHEELSETS	1060.53	2307.49	3464.37	4563.70	5590.33	6385.48	7073.03	7676.96
OTHER TRUCK REPAIRS	139.19	601.55	1066.28	1485.00	1866.81	2172.80	2433.04	2654.50
HEAVY REPAIRS	590.59	1529.18	2050.87	2404.85	2613.56	2794.19	2934.60	3050.98
TOTAL	3543.48	8814.18	13612.79	18065.85	22142.06	25311.09	28046.82	30405.21
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00
				ANNUAL MILEAGE				
CAR REPAIRS:	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	354.80	752.57	1135.20	1507.36	1860.20	2124.51	2353.66	2551.77
COTS	68.83	174.97	275.29	370.81	460.81	527.90	585.63	635.15
IDTS	179.79	357.87	520.55	675.65	819.82	924.14	1012.67	1089.26
PRESSURE SYSTEM	84.96	182.70	287.38	393.99	498.35	579.70	652.90	715.18
HAND BRAKES	21.22	37.03	51.99	66.92	81.22	92.78	103.05	112.18
COUPLERS, YOKES, & DRAFT GEAR	122.58	413.23	737.50	1065.55	1383.71	1636.69	1863.00	2064.71
COUPLER BODIES	43.52	144.21	260.37	379.74	496.82	589.90	673.68	747.80
COUPLER KNUCKLES	21.37	69.87	120.44	169.98	217.23	255.37	289.39	319.61
OTHER COUPLER PARTS	39.19	110.21	179.37	245.53	307.81	356.51	399.25	456.97
YOKES	5.43	27.20	53.44	80.63	107.20	127.01	146.05	162.62
DRAFT GEARS, CARRIERS, AND FOLLOWERS	13.06	61.74	123.88	189.67	254.65	307.10	354.63	398.02
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	368.40	1104.62	1754.38	2336.07	2862.37	3269.19	3617.06	3902.89
OTHER CAR REPAIRS	267.34	701.02	1198.26	1711.28	2241.00	2577.00	2897.22	3174.14
OTHER CAR REPAIRS	138.38	352.10	627.09	920.49	1212.70	1429.43	1620.87	1787.20
WELDING	57.16	176.93	301.44	426.32	545.10	630.36	704.40	767.56
NON BILLABLE INSPECTIONS	71.80	172.00	269.73	364.47	453.20	517.21	571.94	619.37
CAR TOTAL	1113.12	2971.45	4825.34	6620.26	8317.29	9607.39	10736.93	11695.51
				ANNUAL MILEAGE				
TRUCK REPAIRS	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	640.05	1404.51	2205.92	2992.85	3754.07	4351.23	4869.21	5329.26
BRAKE BEAMS	61.32	172.66	335.72	509.75	690.83	841.82	972.98	1093.38
BRAKE BEAM WEAR PLATES	0.17	0.41	0.57	0.67	0.74	0.80	0.82	0.88
BRAKE BEAM HANGERS	0.02	0.04	0.05	0.05	0.06	0.06	0.06	0.07
BRAKE HANGER BRACKET WEAR PLATE	0.00	0.01	0.02	0.02	0.02	0.02	0.03	0.03
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.01	0.01	0.02	0.02	0.03	0.03	0.03	0.03
BRAKE HANGER OR CONNECTION PIN	2.81	6.22	9.73	13.19	16.47	18.99	21.22	23.15
BOTTOM ROD SAFETY SUPPORT	1.07	2.85	4.92	7.35	9.81	11.78	13.43	14.92
BRAKE BEAM SAFETY SUPPORT	0.32	0.78	1.06	1.24	1.37	1.47	1.52	1.61
BRAKE CONNECTION, BOTTOM	2.38	4.29	5.95	7.54	9.02	10.18	11.22	12.12
BRAKE CONNECTION, TOP	2.44	5.89	9.35	12.88	16.29	18.90	21.26	23.25
BRAKE LEVER	1.68	2.97	4.14	5.26	6.31	7.13	7.88	8.52
BRAKE LEVER GUIDE OR CARRIER	0.05	0.15	0.23	0.30	0.36	0.40	0.44	0.48
DEAD LEVER GUIDE	0.02	0.06	0.22	0.44	0.66	0.84	1.02	1.17
DEAD LEVER GUIDE BRACKET	0.02	0.06	0.15	0.26	0.37	0.46	0.54	0.61
BRAKE SHOES	563.89	1200.27	1822.28	2418.97	2983.70	3418.04	3794.37	4124.98
BRAKE SHOE KEYS	3.86	7.85	11.52	14.90	18.02	20.37	22.39	24.14
WHEELSETS	1060.53	2307.49	3464.37	4563.70	5590.33	6385.48	7073.03	7676.96
LUBRICATE ROLLER BEARINGS	19.85	44.76	63.14	78.42	91.63	101.40	109.50	116.61
ROLLER BEARINGS	148.75	320.44	483.36	638.58	784.87	905.02	1010.74	1104.85
ROLLER BEARING CAP SCREWS	0.08	0.23	0.36	0.47	0.57	0.65	0.71	0.77
ROLLER BEARING LOCKING PLATES	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
ROLLER BEARING LUBRICATION FITTING	0.02	0.02	0.03	0.03	0.03	0.04	0.04	0.04
PEDESTAL ADAPTERS	20.16	67.49	114.43	158.08	198.83	230.81	258.64	282.61
WHEELS	356.42	773.29	1170.61	1549.25	1908.83	2198.26	2455.61	2685.12
WHEEL LABOR	509.98	1089.84	1615.23	2116.11	2580.60	2917.87	3201.78	3447.59
AXLES, ROLLER BEARINGS	5.29	11.40	17.21	22.74	27.95	32.24	36.01	39.36
OTHER TRUCK REPAIRS	139.19	601.55	1066.28	1485.00	1866.81	2172.80	2433.04	2654.50
TRUCK BOLSTERS	38.81	181.09	327.67	459.17	579.25	677.48	760.71	830.98
TRUCK BOLSTERS (REPAIRED)	0.98	6.27	11.48	18.94	22.19	25.98	29.05	31.94
CENTER PINS	2.59	7.43	12.60	17.61	22.37	26.00	29.11	31.86
CENTER PLATES	1.63	4.55	9.13	14.10	19.23	23.15	26.71	30.12
CENTER PLATE LINERS	6.93	28.75	50.60	72.14	92.25	107.20	119.98	131.40
TRUCK SIDE BEARINGS	6.24	18.09	28.54	37.72	45.86	52.19	57.52	62.09
FRICTION CASTINGS	13.19	51.48	86.30	117.71	145.98	167.60	185.90	201.98
SIDE BEARING SHIM	0.29	2.12	3.72	5.14	6.29	7.06	7.78	8.47
SIDE FRAMES	49.53	223.62	405.43	568.27	717.45	839.68	943.70	1030.53
SIDE FRAMES (REPAIRED)	0.62	4.69	7.80	10.41	12.49	13.99	15.18	16.29
SPRING PLANKS	0.00	0.02	0.04	0.05	0.06	0.07	0.08	0.08
OUTER SPRINGS	7.57	31.13	51.66	68.68	83.11	94.14	103.52	111.46
INNER SPRINGS	4.27	16.37	26.68	35.14	42.40	47.87	52.50	56.36
STABILIZER SPRINGS	3.30	13.25	21.87	29.17	35.47	40.20	44.16	47.51
TRUCK SPRING FRICTION SNUBBER	0.04	0.09	0.12	0.14	0.16	0.17	0.18	0.19
TRUCK SPRING PLATES	0.01	0.02	0.03	0.04	0.04	0.04	0.04	0.05
TRUCK SPRING SHIM, WOOD	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.01
STEEL	0.05	0.15	0.28	0.43	0.59	0.71	0.85	0.96
MANUFACTURED MATERIAL (TRUCK)	3.14	12.43	22.32	32.13	41.60	49.24	56.10	62.21
TRUCK TOTAL	1839.77	4313.56	6736.57	9041.54	11211.21	12909.51	14375.29	15660.71

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-1A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR ALL ROLLER BEARING CARS

SUMMARY TABLE	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	37.64	79.83	120.42	159.90	199.88	236.46	276.46	311.14
COUPLERS, YOKES, & DRAFT GEAR	13.00	43.84	78.23	113.03	148.68	182.16	218.83	251.75
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	39.08	117.18	186.10	247.81	307.57	363.86	424.86	475.88
OTHER CAR REPAIRS	28.36	74.36	127.11	181.55	237.58	286.82	340.31	387.82
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	67.90	148.99	234.00	317.48	403.38	484.29	571.94	649.80
WHEELSETS	112.50	244.78	367.50	484.12	600.69	710.71	830.80	936.06
OTHER TRUCK REPAIRS	14.76	63.81	113.11	157.53	200.59	241.83	285.79	323.66
HEAVY REPAIRS	62.65	162.21	217.56	255.02	280.83	310.99	344.70	372.01
TOTAL	375.89	935.01	1444.04	1916.42	2379.20	2817.13	3293.67	3707.32
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	37.64	79.83	120.42	159.90	199.88	236.46	276.46	311.14
COT&S	7.30	18.56	29.20	39.34	49.51	58.76	68.78	77.44
IDT&S	19.07	37.96	55.22	71.67	88.09	102.86	118.95	132.81
PRESSURE SYSTEM	9.01	19.38	30.49	41.79	53.55	64.52	76.63	87.20
HAND BRAKES	2.25	3.93	5.51	7.10	8.73	10.33	12.10	13.68
COUPLERS, YOKES, & DRAFT GEAR	13.00	43.84	78.23	113.03	148.68	182.16	218.85	251.75
COUPLER BODIES	4.62	15.30	27.62	40.28	53.38	66.66	79.13	91.14
COUPLER KNUCKLES	2.27	7.41	12.78	18.03	23.34	28.92	33.99	38.97
OTHER COUPLER PARTS	4.16	11.69	19.03	26.05	33.07	39.68	46.90	53.26
YOKES	0.58	2.89	5.67	8.55	11.52	14.22	17.16	19.83
DRAFT GEARS, CARRIERS, AND FOLLOWERS	1.39	6.55	13.14	20.12	27.36	34.18	41.65	48.53
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	39.08	117.18	186.10	247.81	307.57	363.86	424.86	475.88
OTHER CAR REPAIRS	28.36	74.36	127.11	181.53	237.58	286.82	340.31	387.02
OTHER CAR REPAIRS	14.68	37.35	66.52	97.65	130.31	159.10	190.39	217.91
WELDING	6.06	18.77	31.98	45.22	58.57	70.16	82.74	93.59
NON BILLABLE INSPECTIONS	7.62	18.25	28.61	38.66	48.70	57.57	67.18	75.52
CAR TOTAL	118.08	315.21	511.87	702.28	893.70	1069.80	1260.46	1425.80

TRUCK REPAIRS	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	67.90	148.99	234.00	317.48	403.38	484.29	571.94	649.80
BRAKE BEAMS	6.50	18.32	35.61	54.07	74.23	93.69	114.29	133.81
BRAKE BEAM WEAR PLATES	0.02	0.04	0.06	0.07	0.08	0.09	0.10	0.11
BRAKE HANGER OR CONNECTION PIN	0.30	0.66	1.03	1.40	1.77	2.11	2.49	2.82
BOTTOM ROD SAFETY SUPPORT	0.11	0.30	0.52	0.78	1.05	1.30	1.58	1.82
BRAKE BEAM SAFETY SUPPORT	0.03	0.08	0.11	0.13	0.15	0.16	0.18	0.20
BRAKE CONNECTION, BOTTOM	0.25	0.45	0.63	0.80	0.97	1.13	1.32	1.48
BRAKE CONNECTION, TOP	0.26	0.62	0.99	1.37	1.75	2.10	2.50	2.83
BRAKE LEVER	0.18	0.31	0.44	0.56	0.68	0.79	0.93	1.04
BRAKE LEVER GUIDE OR CARRIER	0.01	0.02	0.02	0.03	0.04	0.04	0.05	0.06
DEAD LEVER GUIDE	0.00	0.01	0.02	0.05	0.07	0.09	0.12	0.14
DEAD LEVER GUIDE BRACKET	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07
BRAKE SHOES	59.82	127.32	193.31	256.60	320.80	380.43	445.69	502.96
BRAKE SHOE KEYS	0.41	0.83	1.22	1.58	1.94	2.27	2.65	2.94
WHEELSETS	112.50	244.78	367.50	484.12	600.69	710.71	830.80	936.06
LUBRICATE ROLLER BEARINGS	2.11	4.75	6.70	8.32	9.85	11.29	12.86	14.22
ROLLER BEARINGS	15.78	33.99	51.27	67.74	84.34	100.73	118.72	134.72
ROLLER BEARING CAP SCREWS	0.01	0.02	0.04	0.05	0.06	0.07	0.08	0.09
PEDESTAL ADAPTERS	2.14	7.16	12.14	16.77	21.36	25.69	30.38	34.46
WHEELS	37.81	82.03	124.18	164.34	204.78	244.67	288.44	327.40
WHEEL LABOR	54.10	115.61	171.34	224.48	277.29	324.67	376.08	420.37
AXLES, ROLLER BEARINGS	0.56	1.21	1.83	2.41	3.00	3.59	4.23	4.80
OTHER TRUCK REPAIRS	14.76	63.81	113.11	157.53	200.59	241.83	285.79	323.66
TRUCK BOLSTERS	4.12	19.21	34.76	48.71	62.24	75.40	89.35	101.32
TRUCK BOLSTERS (REPAIRED)	0.10	0.67	1.22	1.80	2.38	2.89	3.41	3.89
CENTER PINS	0.27	0.79	1.34	1.87	2.40	2.89	3.42	3.88
CENTER PLATES	0.17	0.48	0.97	1.50	2.07	2.58	3.14	3.67
CENTER PLATE LINERS	0.73	3.05	5.37	7.65	9.91	11.93	14.09	16.02
TRUCK SIDE BEARINGS	0.66	1.92	3.03	4.00	4.93	5.81	6.76	7.57
FRICTION CASTINGS	1.40	5.46	9.16	12.49	15.69	18.65	21.84	24.63
SIDE BEARING SHIM	0.03	0.23	0.39	0.55	0.68	0.79	0.91	1.03
SIDE FRAMES	5.25	23.72	43.01	60.28	77.09	93.46	110.88	125.65
SIDE FRAMES (REPAIRED)	0.07	0.50	0.83	1.10	1.34	1.56	1.78	1.99
SPRING PLANKS	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01
OUTER SPRINGS	0.80	3.30	5.48	7.29	8.93	10.48	12.16	13.59
INNER SPRINGS	0.45	1.74	2.83	3.73	4.56	5.33	6.17	6.87
STABILIZER SPRINGS	0.35	1.41	2.32	3.09	3.81	4.47	5.19	5.79
TRUCK SPRING FRICTION SNUBBER	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
STEEL	0.01	0.02	0.03	0.05	0.06	0.08	0.10	0.12
MANUFACTURED MATERIAL (TRUCK)	0.33	1.32	2.37	3.41	4.47	5.48	6.59	7.58
TRUCK TOTAL	195.16	457.58	714.61	959.12	1200.66	1436.83	1688.52	1909.52

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-2. PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR HIGH MILEAGE ROLLER BEARING CARS

SUMMARY TABLE	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	161.65	352.38	575.56	797.40	1009.55	1171.07	1311.85	1434.38
COUPLERS, YOKES, & DRAFT GEAR	117.67	270.99	473.94	697.10	918.19	1096.97	1263.11	1406.75
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	98.98	271.60	495.40	712.48	915.21	1071.52	1214.32	1335.51
OTHER CAR REPAIRS	118.75	368.32	744.21	1136.02	1524.21	1811.37	2064.67	2282.81
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	449.58	939.56	1550.03	2153.12	2746.14	3217.04	3625.11	3985.78
WHEELSETS	781.56	1628.70	2455.58	3245.84	3979.21	4551.28	5052.93	5490.52
OTHER TRUCK REPAIRS	37.75	100.26	192.52	275.23	355.08	419.97	475.40	523.26
HEAVY REPAIRS	0.00	79.77	121.37	153.20	170.17	184.64	200.33	212.32
TOTAL	1765.83	4011.58	6608.62	9170.39	11617.77	13523.86	15207.72	16670.32
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	161.65	352.38	575.56	797.40	1009.55	1171.07	1311.85	1433.38
COT&S	12.43	32.83	63.96	95.11	124.65	147.91	167.43	185.40
IDT&S	98.20	210.27	323.72	431.39	531.50	604.52	666.71	719.84
PRESSURE SYSTEM	45.59	98.30	169.47	244.66	319.43	378.33	431.63	477.14
HAND BRAKES	5.42	10.98	18.41	26.24	33.96	40.32	46.08	51.00
COUPLERS, YOKES, & DRAFT GEAR	117.67	270.99	473.94	697.10	918.19	1096.97	1263.11	1406.75
COUPLER BODIES	71.83	137.25	232.86	342.06	455.85	547.84	632.78	707.42
COUPLER KNUCKLES	16.27	39.27	66.34	94.10	121.11	143.00	163.15	180.39
OTHER COUPLER PARTS	24.52	54.74	88.92	123.27	155.91	181.92	205.67	225.40
YOKES	1.55	10.80	22.62	35.31	47.13	56.57	65.42	73.06
DRAFT GEARS, CARRIERS, AND FOLLOWERS	3.50	28.92	64.01	102.36	138.19	167.64	196.09	220.48
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	98.98	271.60	495.40	712.48	915.21	1071.52	1214.32	1335.51
OTHER CAR REPAIRS	118.75	368.32	744.21	1136.02	1524.21	1811.37	2064.67	2282.81
OTHER CAR REPAIRS	55.54	184.58	408.82	650.75	895.54	1078.20	1241.33	1382.57
WELDING	25.43	90.04	175.14	261.07	343.51	403.61	456.32	500.17
NON BILLABLE INSPECTIONS	37.78	93.78	160.25	224.20	285.16	329.55	367.01	400.07
CAR TOTAL	497.05	1263.29	2269.11	3343.00	4367.16	5150.93	5853.95	6458.45

TRUCK REPAIRS	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	449.58	939.56	1550.03	2153.12	2746.14	3217.04	3625.11	3985.78
BRAKE BEAMS	30.49	66.53	167.89	279.15	399.97	502.48	591.64	674.35
BRAKE BEAM WEAR PLATES	0.01	0.04	0.06	0.07	0.08	0.09	0.09	0.10
BRAKE HANGER OR CONNECTION PIN	2.19	4.17	6.58	8.94	11.22	12.98	14.59	15.93
BOTTOM ROD SAFETY SUPPORT	0.87	2.45	4.41	6.74	9.10	10.91	12.60	14.00
BRAKE BEAM SAFETY SUPPORT	0.01	0.05	0.08	0.09	0.11	0.12	0.12	0.13
BRAKE CONNECTION, BOTTOM	1.63	3.28	4.73	6.11	7.39	8.38	9.29	10.01
BRAKE CONNECTION, TOP	1.44	5.09	9.01	12.93	16.79	19.73	22.25	24.55
BRAKE LEVER	1.10	2.18	3.17	4.12	4.98	5.65	6.29	6.78
BRAKE LEVER GUIDE OR CARRIER	0.00	0.05	0.05	0.08	0.10	0.11	0.13	0.14
DEAD LEVER GUIDE	0.02	0.05	0.20	0.40	0.61	0.78	0.94	1.08
DEAD LEVER GUIDE BRACKET	0.05	0.14	0.28	0.44	0.62	0.75	0.86	0.97
BRAKE SHOES	410.81	852.33	1348.35	1826.90	2286.24	2644.75	2954.81	3225.23
BRAKE SHOE KEYS	1.45	3.20	5.25	7.15	8.93	10.30	11.48	12.49
WHEELSETS	781.56	1628.70	2455.58	3245.84	3979.21	4551.28	5052.93	5490.52
LUBRICATE ROLLER BEARINGS	9.80	26.88	39.81	50.93	60.02	66.90	72.72	78.20
ROLLER BEARINGS	111.32	231.00	348.32	461.49	567.51	655.30	733.79	803.10
ROLLER BEARING CAP SCREWS	0.09	0.23	0.32	0.39	0.46	0.51	0.54	0.57
ROLLER BEARING LOCKING PLATES	0.00	0.01	0.02	0.02	0.03	0.03	0.03	0.03
ROLLER BEARING LUBRICATION FITTING	0.00	0.01	0.01	0.02	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	8.18	29.18	54.39	78.10	100.47	118.13	133.75	147.43
WHEELS	267.53	554.67	835.69	1106.52	1359.58	1569.12	1756.34	1921.56
WHEEL LABOR	380.87	770.50	1164.40	1531.93	1870.91	2117.91	2329.58	2510.99
AXLES, ROLLER BEARINGS	3.97	8.23	12.41	16.45	20.23	23.35	26.15	28.62
OTHER TRUCK REPAIRS	37.75	100.26	192.52	275.23	355.08	419.97	475.40	523.26
TRUCK BOLSTERS	7.51	23.47	49.03	71.16	98.09	111.38	126.12	139.67
TRUCK BOLSTERS (REPAIRED)	0.00	0.82	2.00	3.26	4.63	5.66	6.41	7.14
CENTER PINS	1.23	2.96	5.08	7.13	9.07	10.57	11.85	13.03
CENTER PLATES	0.00	0.09	0.26	0.44	0.64	0.79	0.94	1.05
CENTER PLATE LINERS	4.80	14.86	26.21	36.95	47.34	55.08	61.20	67.09
TRUCK SIDE BEARINGS	1.68	4.76	8.27	11.25	14.14	16.30	18.02	19.71
FRICTION CASTINGS	4.29	9.02	15.17	20.46	25.42	29.25	32.51	35.41
SIDE BEARING SHIM	0.02	0.18	0.48	0.73	0.93	1.10	1.26	1.37
SIDE FRAMES	6.70	18.00	42.12	67.27	91.99	112.98	132.32	147.61
SIDE FRAMES (REPAIRED)	0.00	0.18	0.48	0.58	0.77	0.91	1.01	1.10
OUTER SPRINGS	5.67	10.37	14.60	17.77	20.52	22.81	24.56	26.00
INNER SPRINGS	3.33	6.30	8.84	10.77	12.41	13.70	14.76	15.57
STABILIZER SPRINGS	1.28	2.62	4.08	5.19	6.13	6.87	7.46	8.01
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.04
STEEL	0.40	1.29	1.95	2.71	3.44	3.87	4.54	4.94
MANUFACTURED MATERIAL (TRUCK)	2.85	8.32	14.04	19.52	24.54	28.65	32.42	35.53
TRUCK TOTAL	1268.89	2668.52	4198.10	5674.20	7080.44	8188.30	9153.44	9999.56

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-2A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR HIGH MILEAGE ROLLER BEARING CARS

SUMMARY TABLE	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	17.15	37.38	61.05	84.59	108.48	130.34	154.09	174.77
COUPLERS, YOKES, & DRAFT GEAR	12.48	28.75	50.28	73.95	98.66	122.09	148.36	171.53
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	10.50	26.81	52.55	75.58	98.34	119.26	142.63	162.84
OTHER CAR REPAIRS	12.60	39.07	76.95	120.51	163.78	201.61	242.52	278.34
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	47.69	99.67	164.43	228.40	295.08	358.06	425.81	485.99
WHEELSETS	82.91	172.77	260.49	344.32	427.57	506.56	593.52	669.46
OTHER TRUCK REPAIRS	4.00	10.64	20.42	29.20	38.15	46.74	55.84	63.80
HEAVY REPAIRS	0.00	8.46	12.87	16.25	18.29	20.55	23.53	25.89
TOTAL	187.33	425.55	[illegible]	972.79	1248.35	1505.21	1786.30	2032.62
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	17.15	37.38	61.05	84.59	108.48	130.34	154.09	174.77
COT&S	1.32	3.48	6.79	10.09	13.39	16.46	19.67	22.61
IDT&S	10.42	22.31	34.34	45.76	57.11	67.28	78.31	87.77
PRESSURE SYSTEM	4.84	10.43	17.98	25.95	34.32	42.11	50.70	58.18
HAND BRAKES	0.58	1.17	1.95	2.78	3.65	4.49	5.41	6.22
COUPLERS, YOKES, & DRAFT GEAR	12.48	28.75	50.28	73.95	98.66	122.09	148.36	171.53
COUPLER BODIES	7.62	14.86	24.62	36.29	48.98	60.97	74.33	86.26
COUPLER KNUCKLES	1.73	4.17	7.04	9.98	13.01	15.92	19.16	21.99
OTHER COUPLER PARTS	2.60	5.84	9.43	13.08	16.75	20.25	24.16	27.48
YOKES	0.16	1.15	2.40	3.75	5.06	6.30	7.68	8.91
DRAFT GEARS, CARRIERS, AND FOLLOWERS	0.37	3.07	6.79	10.86	14.85	18.66	23.03	26.88
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	10.50	26.81	52.55	75.58	98.34	119.26	142.63	162.84
OTHER CAR REPAIRS	12.60	39.07	76.95	120.51	163.78	201.61	242.52	278.34
OTHER CAR REPAIRS	5.89	19.57	43.37	69.03	96.23	120.00	145.81	168.58
WELDING	2.70	9.55	18.58	27.69	36.91	44.92	53.60	60.99
NON BILLABLE INSPECTIONS	4.01	9.95	17.00	23.78	30.64	36.68	43.11	48.78
CAR TOTAL	52.73	134.01	242.83	354.62	469.26	573.30	687.60	787.48

TRUCK REPAIRS	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	47.69	99.67	164.43	228.40	295.08	358.06	425.81	485.99
BRAKE BEAMS	3.23	7.06	17.81	29.61	42.98	55.93	69.49	82.22
BRAKE BEAM WEAR PLATES	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
BRAKE HANGER OR CONNECTION PIN	0.23	0.44	0.70	0.95	1.21	1.44	1.71	1.94
BOTTOM ROD SAFETY SUPPORT	0.09	0.26	0.47	0.71	0.98	1.21	1.48	1.71
BRAKE BEAM SAFETY SUPPORT	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.02
BRAKE CONNECTION, BOTTOM	0.17	0.35	0.50	0.65	0.79	0.93	1.09	1.22
BRAKE CONNECTION, TOP	0.15	0.54	0.96	1.37	1.80	2.20	2.61	2.99
BRAKE LEVER	0.12	0.23	0.34	0.44	0.53	0.63	0.74	0.83
BRAKE LEVER GUIDE OR CARRIER	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.02
DEAD LEVER GUIDE	0.00	0.01	0.02	0.04	0.07	0.09	0.11	0.13
DEAD LEVER GUIDE BRACKET	0.00	0.01	0.03	0.05	0.07	0.08	0.10	0.12
BRAKE SHOES	43.53	90.42	143.03	193.80	245.66	294.36	347.07	393.25
BRAKE SHOE KEYS	0.18	0.34	0.56	0.76	0.96	1.15	1.35	1.52
WHEELSETS	82.91	172.77	260.49	344.32	427.57	506.56	593.52	669.46
LUBRICATE ROLLER BEARINGS	1.04	2.85	4.22	5.40	6.45	7.45	8.54	9.53
ROLLER BEARINGS	11.81	24.50	36.95	48.96	60.98	72.94	86.19	97.92
ROLLER BEARING CAP SCREWS	0.01	0.02	0.03	0.04	0.05	0.06	0.06	0.07
PEDESTAL ADAPTERS	0.87	3.09	5.77	8.28	10.80	13.15	15.71	17.98
WHEELS	28.38	58.84	88.67	117.38	146.09	174.64	206.30	234.30
WHEEL LABOR	40.40	82.58	123.52	162.51	201.03	235.72	273.63	306.17
AXLES, ROLLER BEARINGS	0.42	0.87	1.32	1.74	2.17	2.60	3.07	3.49
OTHER TRUCK REPAIRS	4.00	10.64	20.42	29.20	38.15	46.74	55.84	63.80
TRUCK BOLSTERS	0.80	2.49	5.20	7.55	10.00	12.40	14.81	17.03
TRUCK BOLSTERS (REPAIRED)	0.00	0.09	0.21	0.35	0.50	0.63	0.75	0.87
CENTER PINS	0.13	0.31	0.54	0.76	0.97	1.18	1.39	1.59
CENTER PLATES	0.00	0.01	0.03	0.05	0.07	0.09	0.11	0.13
CENTER PLATE LINERS	0.46	1.58	2.78	3.92	5.09	6.13	7.19	8.18
TRUCK SIDE BEARINGS	0.18	0.50	0.88	1.19	1.52	1.81	2.12	2.40
FRICTION CASTINGS	0.46	0.96	1.61	2.17	2.73	3.26	3.82	4.32
SIDE BEARING SHIM	0.00	0.02	0.05	0.08	0.10	0.12	0.15	0.17
SIDE FRAMES	0.60	1.59	4.47	7.14	9.88	12.57	15.54	18.00
SIDE FRAMES (REPAIRED)	0.00	0.02	0.04	0.06	0.08	0.10	0.12	0.13
OUTER SPRINGS	0.60	1.10	1.55	1.89	2.21	2.54	2.89	3.17
INNER SPRINGS	0.35	0.67	0.94	1.14	1.33	1.55	1.73	1.90
STABILIZER SPRINGS	0.13	0.28	0.43	0.55	0.66	0.76	0.88	0.98
STEEL	0.05	0.14	0.21	0.29	0.37	0.43	0.53	0.60
MANUFACTURED MATERIAL (TRUCK)	0.25	0.88	1.49	2.07	2.64	3.19	3.81	4.33
TRUCK TOTAL	134.60	283.08	445.54	601.92	760.80	911.36	1075.16	1219.25

-DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-3. PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR 100-TON, NORMAL SERVICE, ROLLER BEARING CARS

SUMMARY TABLE

				ANNUAL MILEAGE				
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	325.88	712.36	1093.04	1466.62	1819.66	2089.80	2321.85	2518.95
COUPLERS, YOKES, & DRAFT GEAR	87.45	354.53	650.55	948.18	1232.02	1460.87	1664.49	1841.12
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	244.39	633.39	971.13	1273.97	1542.44	1753.52	1931.25	2081.46
OTHER CAR REPAIRS	238.46	667.63	1133.44	1605.65	2055.32	2386.79	2672.85	2920.67
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	644.92	1452.46	2299.02	3135.17	3936.27	4561.99	5109.44	5595.62
WHEELSETS	981.27	2304.18	3579.65	4804.49	5955.15	6851.24	7632.34	8297.30
OTHER TRUCK REPAIRS	43.65	257.46	476.59	687.08	871.31	1011.52	1134.74	1250.80
HEAVY REPAIRS	426.85	1098.59	1457.25	1710.04	1844.91	1960.24	2066.53	2148.51
TOTAL	2992.87	7480.60	11660.69	5631.21	19257.88	22075.96	24533.51	26654.42
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:

				ANNUAL MILEAGE				
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	325.88	712.36	1093.04	1466.62	1819.66	2089.80	2321.85	2518.95
COTAS	37.11	128.41	221.51	311.11	395.46	462.55	518.25	565.19
IDTAS	177.65	353.72	515.57	670.97	814.96	919.33	1008.47	1085.09
PRESSURE SYSTEM	90.18	199.18	315.45	434.16	549.41	640.46	720.68	789.96
HAND BRAKES	20.93	31.04	40.50	50.38	59.83	67.47	74.45	80.71
COUPLERS, YOKES, & DRAFT GEAR	87.45	354.53	650.55	948.18	1232.02	1460.87	1664.49	1841.12
COUPLER BODIES	27.25	121.28	221.14	319.60	411.88	486.61	552.18	608.47
COUPLER KNUCKLES	15.55	60.83	110.25	159.44	206.35	244.53	278.83	308.54
OTHER COUPLER PARTS	30.14	93.12	156.07	217.19	274.51	319.65	359.64	394.06
YOKES	4.21	26.98	56.07	86.60	116.58	140.26	161.22	179.61
DRAFT GEARS, CARRIERS, AND FOLLOWERS	10.29	52.33	107.02	165.36	222.74	269.82	312.63	350.45
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	244.39	633.39	971.13	1273.97	1542.44	1753.52	1931.25	2081.46
OTHER CAR REPAIRS	238.46	667.63	1133.44	1605.65	2055.32	2386.79	2672.85	2920.67
OTHER CAR REPAIRS	120.62	298.90	510.84	730.20	943.38	1103.75	1242.42	1362.97
WELDING	47.61	187.15	330.99	476.07	612.61	711.49	796.50	869.61
NON BILLABLE INSPECTIONS	70.23	181.58	291.61	399.38	499.33	571.55	633.93	688.08
CAR TOTAL	896.18	2367.90	3848.17	5294.93	6649.44	7690.98	8598.45	9362.19

TRUCK REPAIRS

				ANNUAL MILEAGE				
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	644.92	1452.46	2299.02	3135.17	3936.27	4561.90	5109.44	5595.62
BRAKE BEAMS	54.22	147.82	288.67	441.96	599.93	729.99	844.48	952.17
BRAKE BEAM WEAR PLATES	0.13	0.29	0.40	0.47	0.53	0.56	0.59	0.62
BRAKE BEAM HANGERS	0.01	0.01	0.01	0.01	0.01	0.01	0.02	0.02
BRAKE HANGER OR CONNECTION PIN	2.44	6.02	9.73	13.41	16.90	19.59	21.95	23.95
BOTTOM ROD SAFETY SUPPORT	0.96	3.54	6.53	9.99	13.56	16.26	18.74	20.79
BRAKE BEAM SAFETY SUPPORT	0.04	0.13	0.18	0.23	0.25	0.28	0.29	0.30
BRAKE CONNECTION, BOTTOM	2.56	4.77	6.69	8.53	10.26	11.60	12.82	13.83
BRAKE CONNECTION, TOP	3.02	7.84	12.67	17.57	22.31	25.93	29.22	31.85
BRAKE LEVER	1.85	3.60	5.23	6.81	8.27	9.43	10.47	11.35
BRAKE LEVER GUIDE OR CARRIER	0.07	0.26	0.42	0.56	0.68	0.78	0.85	0.92
DEAD LEVER GUIDE	0.01	0.13	0.61	1.24	1.92	2.48	2.97	3.40
DEAD LEVER GUIDE BRACKET	0.01	0.04	0.08	0.14	0.20	0.25	0.29	0.33
BRAKE SHOES	575.58	1270.38	1956.81	2620.12	3244.44	3725.71	4145.77	4813.37
BRAKE SHOE KEYS	4.01	7.64	10.99	14.12	16.99	19.14	21.01	22.66
WHEELSETS	981.27	2304.18	3579.65	4804.49	5955.15	6851.24	7632.34	8297.30
LUBRICATE ROLLER BEARINGS	19.10	42.55	59.76	74.22	86.79	95.79	103.70	110.61
ROLLER BEARINGS	139.39	319.59	500.85	676.30	843.15	981.14	1103.50	1208.74
ROLLER BEARING CAP SCREWS	0.09	0.26	0.40	0.52	0.62	0.69	0.76	0.83
ROLLER BEARING LUBRICATION FITTING	0.02	0.02	0.03	0.03	0.03	0.03	0.03	0.03
PEDESTAL ADAPTERS	11.70	46.68	82.53	116.94	148.42	172.68	194.77	213.79
WHEELS	333.88	769.75	1208.75	1633.01	2036.33	2369.63	2664.98	2919.15
WHEEL LABOR	472.14	1113.95	1709.48	2279.36	2809.74	3196.56	3525.28	3801.02
AXLES, ROLLER BEARINGS	4.95	11.38	17.85	24.11	30.07	34.99	39.36	43.11
OTHER TRUCK REPAIRS	43.65	257.46	476.59	687.08	871.31	1011.52	1134.70	1250.80
TRUCK BOLSTERS	7.09	47.88	92.62	135.41	172.78	201.70	227.30	251.70
TRUCK BOLSTERS (REPAIRED)	0.48	10.30	20.21	30.96	41.08	48.20	54.19	60.01
CENTER PINS	1.62	5.14	8.94	12.66	16.19	18.86	21.22	23.23
CENTER PLATES	0.55	3.34	8.32	13.83	19.50	23.84	27.77	31.62
CENTER PLATE LINERS	4.94	30.60	57.11	83.67	108.24	126.85	142.41	156.78
TRUCK SIDE BEARINGS	4.19	10.67	16.51	21.82	26.35	29.93	32.94	35.61
FRICTION CASTINGS	6.96	29.12	51.51	73.54	93.27	107.98	121.02	133.32
SIDE BEARING SHIM	0.19	4.54	8.48	12.16	15.05	16.93	18.78	20.61
SIDE FRAMES	8.18	51.43	98.96	144.56	183.11	214.33	240.68	266.54
SIDE FRAMES (REPAIRED)	0.38	8.03	13.97	19.42	23.45	26.04	28.43	31.09
SPRING PLANKS	0.01	0.01	0.01	0.01	0.01	0.02	0.02	0.02
OUTER SPRINGS	3.67	26.58	46.91	64.54	79.04	89.61	98.49	108.22
INNER SPRINGS	1.98	13.18	22.72	30.53	37.57	42.47	46.98	50.80
STABILIZER SPRINGS	1.27	6.61	11.65	16.26	19.98	22.63	24.94	27.26
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
STELL	0.01	0.01	0.02	0.02	0.03	0.03	0.04	0.04
MANUFACTURED MATERIAL (TRUCK)	2.13	9.99	18.63	27.35	35.71	42.35	48.47	53.94
TRUCK TOTAL	1669.84	4014.10	6355.27	8626.74	10762.73	12424.75	13876.53	15143.72

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-3A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR 100-TON, NORMAL SERVICE, ROLLER BEARING CARS

SUMMARY TABLE	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	34.57	75.57	115.95	155.58	195.53	232.60	272.72	307.14
COUPLERS, YOKES, & DRAFT GEAR	9.28	37.61	69.01	100.58	132.38	162.60	195.51	224.49
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	25.92	67.19	103.02	135.14	165.74	195.17	226.84	253.79
OTHER CAR REPAIRS	25.30	70.82	120.24	170.33	220.85	265.65	313.95	356.12
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	68.41	154.08	243.88	332.58	422.96	507.75	600.16	682.28
WHEELSETS	104.09	244.43	379.73	509.66	639.89	762.54	896.50	1011.69
OTHER TRUCK REPAIRS	4.63	27.31	50.56	72.89	93.62	112.58	133.29	152.51
HEAVY REPAIRS	45.28	116.54	154.58	181.40	198.24	218.18	242.74	261.97
TOTAL	317.48	793.54	1236.96	1658.15	2069.20	2457.06	2881.71	3249.99
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	34.57	75.57	115.95	155.58	195.53	232.60	272.72	307.14
COT&S	3.94	13.62	23.50	33.00	42.49	51.48	60.07	68.67
IDT&S	18.85	37.52	54.69	71.18	87.57	102.32	118.45	132.31
PRESSURE SYSTEM	9.57	21.13	33.46	46.06	59.03	71.28	84.65	96.32
HAND BRAKES	2.22	3.29	4.30	5.34	6.43	7.51	8.74	9.84
COUPLERS, YOKES, & DRAFT GEAR	9.28	37.61	69.01	100.58	132.38	162.60	195.51	224.49
COUPLER BODIES	2.89	12.86	23.46	33.90	44.26	54.16	64.86	74.14
COUPLER KNUCKLES	1.65	6.45	11.70	16.91	22.17	27.22	32.75	37.62
OTHER COUPLER PARTS	3.20	9.88	16.56	23.04	29.50	35.58	42.24	48.05
YOKES	0.45	2.86	5.95	9.19	12.52	15.61	18.94	21.90
DRAFT GEARS, CARRIERS, AND FOLLOWERS	1.09	5.55	11.35	17.54	23.93	30.03	36.72	42.73
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	25.92	67.19	103.02	135.14	165.74	195.17	226.84	253.79
OTHER CAR REPAIRS	25.30	70.82	120.24	170.33	220.85	265.65	313.95	356.12
OTHER CAR REPAIRS	12.80	31.71	54.19	77.46	101.37	122.85	145.93	166.19
WELDING	5.05	19.85	35.11	50.50	65.83	79.19	93.56	106.03
NON BILLABLE INSPECTIONS	7.45	19.26	30.93	42.37	53.65	63.61	74.46	83.90
CAR TOTAL	95.07	251.19	408.21	561.63	714.49	856.01	1009.03	1141.54

TRUCK REPAIRS	12500.	25000.	37500.	ANNUAL MILEAGE 50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	68.41	154.08	243.88	332.58	422.96	507.75	600.16	682.28
BRAKE BEAMS	5.75	15.68	30.62	46.88	64.48	81.25	99.19	116.10
BRAKE BEAM WEAR PLATES	0.01	0.03	0.04	0.05	0.06	0.06	0.07	0.08
BRAKE HANGER OR CONNECTION PIN	0.26	0.64	1.03	1.42	1.82	2.18	2.58	2.92
BOTTOM ROD SAFETY SUPPORT	0.10	0.38	0.69	1.06	1.46	1.81	2.20	2.54
BRAKE BEAM SAFETY SUPPORT	0.00	0.01	0.02	0.02	0.03	0.03	0.03	0.04
BRAKE CONNECTION, BOTTOM	0.27	0.51	0.71	0.91	1.10	1.29	1.51	1.69
BRAKE CONNECTION, TOP	0.32	0.83	1.34	1.86	2.40	2.89	3.43	3.89
BRAKE LEVER	0.20	0.38	0.56	0.72	0.89	1.08	1.23	1.38
BRAKE LEVER GUIDE OR CARRIER	0.01	0.03	0.04	0.06	0.07	0.09	0.10	0.11
DEAD LEVER GUIDE	0.00	0.01	0.07	0.13	0.21	0.27	0.35	0.41
DEAD LEVER GUIDE BRACKET	0.00	0.00	0.01	0.02	0.02	0.03	0.03	0.04
BRAKE SHOES	61.06	134.76	207.58	277.94	348.62	416.67	486.96	550.32
BRAKE SHOE KEYS	0.43	0.81	1.17	1.50	1.83	2.13	2.47	2.76
WHEELSETS	104.09	244.43	379.73	509.66	639.89	762.54	896.50	1011.69
LUBRICATE ROLLER BEARINGS	2.03	4.51	6.34	7.87	9.33	10.66	12.18	13.49
ROLLER BEARINGS	14.79	33.90	53.13	71.74	90.60	109.20	129.62	147.38
ROLLER BEARING CAP SCREWS	0.01	0.03	0.04	0.06	0.07	0.08	0.09	0.10
PEDESTAL ADAPTERS	1.24	4.95	8.75	12.40	15.95	19.22	22.88	26.07
WHEELS	35.42	81.66	128.22	173.23	218.81	263.74	313.02	355.93
WHEEL LABOR	50.08	118.17	181.34	241.79	301.91	355.76	414.08	463.46
AXLES, ROLLER BEARINGS	0.53	1.21	1.89	2.56	3.23	3.89	4.62	5.26
OTHER TRUCK REPAIRS	4.63	27.31	50.56	72.89	93.62	112.58	133.29	152.51
TRUCK BOLSTERS	0.75	5.08	9.82	14.36	18.57	22.85	26.70	30.69
TRUCK BOLSTERS (REPAIRED)	0.05	1.09	2.14	3.28	4.41	5.37	6.37	7.32
CENTER PINS	0.17	0.54	0.95	1.34	1.74	2.18	2.49	2.83
CENTER PLATES	0.06	0.35	0.88	1.57	2.10	2.65	3.26	3.66
CENTER PLATE LINERS	0.52	3.25	6.06	8.88	11.63	14.08	16.73	19.11
TRUCK SIDE BEARINGS	0.44	1.13	1.75	2.32	2.88	3.35	3.87	4.34
FRICTION CASTINGS	0.74	3.09	5.46	7.80	10.02	12.02	14.22	16.26
SIDE BEARING SHIM	0.02	0.48	0.90	1.29	1.62	1.88	2.21	2.51
SIDE FRAMES	0.87	5.46	10.50	15.35	19.88	23.85	28.27	32.50
SIDE FRAMES (REPAIRED)	0.04	0.85	1.48	2.06	2.52	2.90	3.34	3.79
OUTER SPRINGS	0.39	2.82	4.98	6.88	8.49	9.97	11.69	13.19
INNER SPRINGS	0.21	1.40	2.41	3.27	4.04	4.73	5.52	6.19
STABILIZER SPRINGS	0.13	0.70	1.24	1.72	2.15	2.52	2.93	3.32
MANUFACTURED MATERIAL (TRUCK)	0.23	1.06	1.98	2.90	3.84	4.74	5.69	6.58
TRUCK TOTAL	177.14	425.81	674.16	915.12	1156.47	1382.88	1629.94	1846.48

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-4. PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR 100-TON, HIGH MILEAGE, ROLLER BEARING CARS

SUMMARY TABLE

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	149.25	309.92	469.00	625.20	771.52	883.41	977.68	1060.69
COUPLERS, YOKES, & DRAFT GEAR	117.69	253.77	416.16	592.30	764.22	902.52	1031.82	1142.91
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	40.39	182.52	332.74	475.51	609.66	712.07	799.58	879.13
OTHER CAR REPAIRS	110.04	270.86	446.12	621.54	791.15	916.45	1022.80	1115.58
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	404.25	904.99	1432.08	1948.32	2450.15	2843.10	3184.03	3482.41
WHEELSETS	764.77	1681.46	2569.05	3430.46	4237.70	4862.80	5408.31	5884.97
OTHER TRUCK REPAIRS	39.79	104.01	161.34	211.00	257.17	291.96	320.75	345.59
HEAVY REPAIRS	0.00	89.79	156.44	172.46	191.06	207.30	224.93	239.00
TOTAL	1626.18	3797.32	5962.93	8076.79	10072.63	11619.62	12969.89	14150.29
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	149.25	309.92	469.00	625.20	771.52	883.41	977.68	1060.69
COT&S	12.54	32.97	58.87	85.67	110.68	131.30	147.29	162.55
IDT&S	88.64	191.47	284.29	372.10	452.64	511.45	561.08	603.72
PRESSURE SYSTEM	42.60	76.57	113.49	151.55	189.01	218.57	244.77	267.68
HAND BRAKES	5.47	8.90	12.35	15.88	19.19	22.09	24.54	26.74
COUPLERS, YOKES, & DRAFT GEAR	117.69	253.77	416.16	592.30	764.22	902.52	1031.82	1142.91
COUPLER BODIES	74.66	151.09	203.82	286.65	371.20	439.23	502.73	557.80
COUPLER KNUCKLES	13.19	31.19	49.96	68.64	86.25	100.91	113.66	125.25
OTHER COUPLER PARTS	24.68	50.44	75.76	99.89	122.57	140.78	157.16	170.84
YOKES	1.58	10.68	21.31	32.55	42.79	51.01	58.66	65.30
DRAFT GEARS, CARRIERS, AND FOLLOWERS	3.59	30.37	65.72	104.58	141.42	170.59	199.61	223.73
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	40.39	182.52	332.74	475.51	609.66	712.07	799.58	879.13
OTHER CAR REPAIRS	110.04	270.86	446.12	621.54	791.15	916.45	1022.80	1115.58
OTHER CAR REPAIRS	52.61	130.13	222.03	316.10	408.50	477.56	537.00	588.55
WELDING	23.92	56.60	89.07	121.52	152.02	174.23	193.32	209.58
NON BILLABLE INSPECTIONS	33.51	84.13	135.02	183.92	230.63	264.66	292.47	317.46
CAR TOTAL	417.37	1017.06	1664.02	2314.56	2936.55	3414.45	3831.88	4198.32

TRUCK REPAIRS

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	404.25	904.99	1432.08	1948.32	2450.15	2843.10	3184.03	3482.41
BRAKE BEAMS	50.71	73.58	150.20	235.67	328.11	404.54	471.43	532.43
BRAKE BEAM WEAR PLATES	0.01	0.03	0.04	0.05	0.05	0.05	0.05	0.06
BRAKE HANGER OR CONNECTION PIN	2.25	3.94	5.51	7.06	8.52	9.62	10.63	11.47
BOTTOM ROD SAFETY SUPPORT	0.90	1.60	2.17	2.80	3.40	3.84	4.30	4.6
BRAKE BEAM SAFETY SUPPORT	0.01	0.05	0.08	0.10	0.12	0.13	0.14	0.14
BRAKE CONNECTION, BOTTOM	1.63	3.02	4.05	4.96	5.79	6.42	7.00	7.4
BRAKE CONNECTION, TOP	1.47	4.72	7.90	10.98	13.99	16.24	18.19	19.9
BRAKE LEVER	1.13	2.12	2.92	3.64	4.29	4.78	5.26	5.6
BRAKE LEVER GUIDE OR CARRIER	0.00	0.02	0.04	0.05	0.07	0.08	0.08	0.0
DEAD LEVER GUIDE	0.03	0.05	0.13	0.24	0.36	0.45	0.55	0.6
DEAD LEVER GUIDE BRACKET	0.05	0.13	0.38	0.49	0.70	0.85	0.99	1.1
BRAKE SHOES	364.70	812.59	1253.41	1675.04	2075.74	2385.77	2653.91	2886.3
BRAKE SHOE KEYS	1.35	3.33	5.33	7.22	9.00	10.32	11.48	12.4
WHEELSETS	764.77	1681.46	2569.05	3430.46	4237.70	4862.80	5408.31	5884.97
LUBRICATE ROLLER BEARINGS	10.38	28.02	41.34	52.90	62.36	69.50	75.60	81.38
ROLLER BEARINGS	108.18	236.53	364.07	488.93	607.44	704.86	791.54	868.21
ROLLER BEARING CAP SCREWS	0.09	0.23	0.33	0.41	0.49	0.54	0.59	0.62
ROLLER BEARING LOCKING PLATES	0.00	0.01	0.01	0.01	0.01	0.01	0.02	0.02
ROLLER BEARING LUBRICATION FITTING	0.00	0.01	0.01	0.02	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	8.33	27.44	45.38	61.65	76.40	87.80	97.85	106.61
WHEELS	259.78	567.86	872.91	1170.78	1452.95	1684.84	1890.97	2073.25
WHEEL LABOR	374.14	812.95	1232.04	1638.34	2016.38	2290.11	2523.53	2723.93
AXLES, ROLLER BEARINGS	3.88	8.43	12.97	17.42	21.64	25.11	28.20	30.93
OTHER TRUCK REPAIRS	39.79	104.01	161.34	211.00	257.17	291.96	320.75	345.59
TRUCK BOLSTERS	8.07	25.09	40.09	52.15	63.59	72.28	78.93	85.2
TRUCK BOLSTERS (REPAIRED)	0.80	1.29	3.14	5.11	7.25	8.86	10.04	11.1
CENTER PINS	1.25	2.96	4.82	6.63	8.34	9.68	10.76	11.8
CENTER PLATES	0.08	0.07	0.16	0.26	0.37	0.46	0.53	0.5
CENTER PLATE LINERS	4.60	14.90	25.47	35.51	45.22	52.51	58.11	63.5
TRUCK SIDE BEARINGS	1.78	4.58	6.99	8.96	10.88	12.21	13.21	14.2
FRICTION CASTINGS	4.50	9.32	13.58	17.34	20.89	23.53	25.70	27.7
SIDE BEARING SHIM	0.02	0.11	0.18	0.24	0.28	0.31	0.35	0.37
SIDE FRAMES	6.49	16.76	24.80	31.57	37.41	41.33	45.74	47.95
SIDE FRAMES (REPAIRED)	0.80	0.29	0.68	0.91	1.20	1.43	1.58	1.73
OUTER SPRINGS	5.84	10.70	14.24	16.87	19.09	20.92	22.26	23.36
INNER SPRINGS	3.46	6.60	8.92	10.68	12.16	13.31	14.23	14.93
STABILIZER SPRINGS	1.26	2.82	4.05	5.06	5.87	6.49	6.98	7.45
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.04
STEEL	0.00	0.00	0.00	0.01	0.01	0.01	0.02	0.02
MANUFACTURED MATERIAL (TRUCK)	2.50	8.59	14.28	19.67	24.58	28.59	32.27	35.33
TRUCK TOTAL	1208.80	2690.46	4162.47	5589.77	6945.02	7997.86	8913.08	9712.97

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-4A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR 100-TON, HIGH MILEAGE, ROLLER BEARING CARS

SUMMARY TABLE

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	15.83	32.88	49.75	66.32	82.90	98.32	114.84	129.33
COUPLERS, YOKES, & DRAFT GEAR	12.48	26.92	44.15	62.83	82.12	100.45	121.20	139.36
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	4.28	19.36	35.30	50.44	65.51	79.25	93.92	107.19
OTHER CAR REPAIRS	11.67	28.73	47.32	65.93	85.01	102.00	120.14	136.02
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	42.88	96.00	151.91	206.68	263.27	316.44	374.00	424.61
WHEELSETS	81.13	178.37	272.52	363.90	455.35	541.23	635.26	717.56
OTHER TRUCK REPAIRS	4.22	11.03	17.11	22.38	27.63	32.50	37.67	42.14
HEAVY REPAIRS	0.00	9.53	14.47	18.29	20.53	23.07	26.42	29.14
TOTAL	172.80	402.82	632.55	856.78	1082.32	1293.27	1523.44	1725.35
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	15.83	32.88	49.75	66.32	82.90	98.32	114.84	129.33
COT&S	1.33	3.56	6.24	9.09	11.89	14.61	17.30	19.82
IOT&S	9.40	20.31	30.16	39.47	48.64	56.92	65.90	73.61
PRESSURE SYSTEM	4.52	8.12	12.04	16.08	20.31	24.33	28.75	32.64
HAND BRAKES	0.58	0.84	1.31	1.68	2.06	2.46	2.88	3.26
COUPLERS, YOKES, & DRAFT GEAR	12.48	26.92	44.15	62.83	82.12	100.45	121.20	139.36
COUPLER BODIES	7.92	15.91	21.82	30.41	39.89	48.89	59.05	68.01
COUPLER KNUCKLES	1.40	3.31	5.30	7.28	9.27	11.23	13.35	15.27
OTHER COUPLER PARTS	2.62	5.35	7.99	10.60	13.17	15.67	18.46	20.83
YOKES	0.17	1.15	2.26	3.45	4.60	5.68	6.89	7.96
DRAFT GEARS, CARRIERS, AND FOLLOWERS	0.38	3.22	6.97	11.09	15.20	18.99	23.45	27.28
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	4.28	19.36	35.30	50.44	65.51	79.25	93.92	107.19
OTHER CAR REPAIRS	11.67	28.73	47.32	65.93	85.01	102.00	120.14	136.02
OTHER CAR REPAIRS	5.58	13.80	23.55	33.53	43.89	53.15	63.08	71.76
WELDING	2.54	6.00	9.45	12.89	16.33	19.39	22.71	25.55
NON BILLABLE INSPECTIONS	3.55	8.92	14.32	19.51	24.78	29.46	34.35	38.71
CAR TOTAL	44.27	107.89	176.52	245.53	315.54	380.03	450.09	511.90

TRUCK REPAIRS

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	42.88	96.00	151.91	206.68	263.27	316.44	374.00	424.61
BRAKE BEAMS	3.26	7.81	15.95	25.00	35.26	45.03	55.37	64.92
BRAKE HANGER OR CONNECTION PIN	0.24	0.42	0.58	0.75	0.92	1.07	1.25	1.40
BOTTOM ROD SAFETY SUPPORT	0.10	0.17	0.23	0.30	0.37	0.43	0.51	0.56
BRAKE BEAM SAFETY SUPPORT	0.00	0.01	0.01	0.01	0.01	0.01	0.02	0.02
BRAKE CONNECTION, BOTTOM	0.17	0.32	0.43	0.53	0.62	0.71	0.82	0.91
BRAKE CONNECTION, TOP	0.16	0.50	0.84	1.16	1.50	1.81	2.14	2.43
BRAKE LEVER	0.12	0.23	0.31	0.39	0.46	0.53	0.62	0.69
BRAKE LEVER GUIDE OR CARRIER	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01
DEAD LEVER GUIDE	0.00	0.01	0.01	0.03	0.04	0.05	0.06	0.08
DEAD LEVER GUIDE BRACKET	0.01	0.01	0.03	0.05	0.08	0.10	0.12	0.14
BRAKE SHOES	38.69	86.18	132.96	177.69	223.04	265.54	311.73	351.94
BRAKE SHOE KEYS	0.14	0.35	0.57	0.77	0.97	1.15	1.35	1.52
WHEELSETS	81.13	178.37	272.52	363.90	455.35	541.23	635.26	717.56
LUBRICATE ROLLER BEARINGS	1.10	2.97	4.38	5.61	6.70	7.74	8.88	9.92
ROLLER BEARINGS	11.48	25.09	38.62	51.87	65.27	78.45	92.97	105.86
ROLLER BEARING CAP SCREWS	0.01	0.02	0.04	0.04	0.05	0.06	0.07	0.08
PEDESTAL ADAPTERS	0.88	2.91	4.81	6.54	8.21	9.77	11.49	13.00
WHEELS	27.86	60.24	92.60	124.20	156.12	187.52	222.11	252.79
WHEEL LABOR	39.69	86.24	130.69	173.79	216.66	254.89	296.41	332.13
AXLES, ROLLER BEARINGS	0.41	0.89	1.38	1.85	2.33	2.79	3.31	3.77
OTHER TRUCK REPAIRS	4.22	11.03	17.11	22.38	27.63	32.50	37.67	42.14
TRUCK BOLSTERS	0.86	2.65	4.25	5.53	6.83	8.04	9.27	10.40
TRUCK BOLSTERS (REPAIRED)	0.00	0.14	0.33	0.54	0.78	0.99	1.18	1.36
CENTER PINS	0.13	0.31	0.51	0.70	0.90	1.08	1.26	1.44
CENTER PLATES	0.00	0.01	0.02	0.03	0.04	0.05	0.06	0.07
CENTER PLATE LINERS	0.49	1.58	2.70	3.77	4.86	5.84	6.83	7.75
TRUCK SIDE BEARINGS	0.19	0.49	0.74	0.95	1.17	1.36	1.55	1.74
FRICTION CASTINGS	0.48	0.99	1.44	1.84	2.25	2.62	3.02	3.38
SIDE BEARING SHIM	0.00	0.01	0.02	0.03	0.03	0.03	0.04	0.04
SIDE FRAMES	0.69	1.78	2.63	3.35	4.02	4.60	5.37	5.85
SIDE FRAMES (REPAIRED)	0.00	0.05	0.07	0.10	0.13	0.16	0.19	0.21
OUTER SPRINGS	0.62	1.13	1.51	1.79	2.05	2.33	2.62	2.85
INNER SPRINGS	0.37	0.70	0.95	1.13	1.31	1.48	1.67	1.82
STABILIZER SPRINGS	0.13	0.30	0.45	0.54	0.63	0.72	0.82	0.91
MANUFACTURED MATERIAL (TRUCK)	0.27	0.91	1.51	2.09	2.64	3.18	3.79	4.31
TRUCK TOTAL	128.23	285.40	441.55	592.96	746.25	890.16	1046.93	1184.31

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-5. PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR 70-TON, NORMAL SERVICE, ROLLER BEARING CARS

SUMMARY TABLE	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	651.74	1238.45	1772.37	2299.36	2792.48	3157.44	3476.25	3748.83
COUPLERS, YOKES, & DRAFT GEAR	212.92	637.47	1099.43	1560.66	2008.68	2357.86	2671.97	2947.65
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	801.44	2063.33	3120.82	4068.87	4921.49	5582.71	6141.32	6595.21
OTHER CAR REPAIRS	610.85	1170.91	1718.26	2290.50	2834.04	3229.21	3577.09	3874.47
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	923.93	1837.14	2693.52	3543.30	4351.71	4978.70	5527.52	6005.85
WHEELSETS	1277.13	2725.38	3989.16	5180.33	6282.29	7143.20	7870.58	8513.07
OTHER TRUCK REPAIRS	419.11	1344.76	2253.48	3082.19	3837.43	4439.12	4957.06	5390.68
HEAVY REPAIRS	1144.47	2597.94	3400.82	3926.58	4241.79	4532.20	4731.07	4909.22
TOTAL	6041.59	13615.38	20047.87	25951.79	31269.92	35420.44	38952.88	41985.80
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	651.74	1238.45	1772.37	2299.36	2792.48	3157.44	3476.25	3745.83
COT&S	214.30	421.73	613.23	808.93	977.39	1107.08	1220.13	1315.63
IDT&S	288.63	529.22	738.86	944.95	1132.54	1267.66	1385.82	1483.54
PRESSURE SYSTEM	127.90	241.40	351.91	464.22	573.13	656.97	732.01	797.57
HAND BRAKES	20.91	46.10	68.37	89.76	109.42	125.74	139.48	152.09
COUPLERS, YOKES, & DRAFT GEAR	212.92	637.47	1099.43	1560.66	2008.68	2357.86	2671.97	2947.65
COUPLER BODIES	50.34	169.19	283.51	387.61	484.02	559.61	624.46	682.35
COUPLER KNUCKLES	49.28	120.35	192.12	262.94	330.17	384.15	432.02	474.93
OTHER COUPLER PARTS	86.32	194.67	299.41	401.50	497.87	572.74	639.29	697.42
YOKES	7.83	47.92	100.21	155.41	211.30	252.70	291.22	323.80
DRAFT GEARS, CARRIERS, AND FOLLOWERS	19.15	105.33	224.19	353.21	485.32	588.66	684.99	769.15
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	801.44	2063.33	3120.82	4068.87	4921.49	5582.71	6141.32	6595.21
OTHER CAR REPAIRS	610.85	1170.91	1718.26	2290.50	2834.04	3229.21	3577.09	3874.47
OTHER CAR REPAIRS	340.55	617.67	897.65	1196.39	1484.06	1696.38	1883.49	2045.58
WELDING	126.13	282.61	436.44	597.12	748.25	855.09	951.79	1031.98
NON BILLABLE INSPECTIONS	144.17	270.62	384.16	496.99	601.72	677.74	741.81	796.90
CAR TOTAL	2276.96	5110.16	7710.88	10219.38	12556.70	14327.22	15866.63	17166.16

TRUCK REPAIRS	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	923.93	1837.14	2693.52	3543.30	4351.71	4978.70	5527.82	6005.89
BRAKE BEAMS	77.42	234.92	397.91	569.55	739.94	880.18	1005.82	1115.85
BRAKE HEAD WEAR PLATES	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.01
BRAKE BEAM WEAR PLATES	0.17	0.51	0.73	0.87	0.97	1.05	1.07	1.16
BRAKE BEAM HANGERS	0.03	0.07	0.09	0.11	0.12	0.12	0.13	0.13
BRAKE HANGER BRACKET WEAR PLATE	0.01	0.02	0.03	0.04	0.04	0.04	0.04	0.05
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.01	0.03	0.04	0.05	0.05	0.06	0.06	0.06
BRAKE HANGER OR CONNECTION PIN	4.86	9.11	13.09	17.02	20.77	23.66	26.18	28.38
BOTTOM ROD SAFETY SUPPORT	0.96	2.09	3.10	4.17	5.23	6.08	6.79	7.45
BRAKE BEAM SAFETY SUPPORT	0.71	1.64	2.21	2.55	2.82	3.01	3.10	3.28
BRAKE CONNECTION, BOTTOM	1.52	3.23	4.64	5.91	7.09	8.02	8.82	9.54
BRAKE CONNECTION, TOP	3.56	6.22	8.40	10.60	12.61	14.19	15.54	16.77
BRAKE LEVER	1.01	2.13	3.15	4.07	4.94	5.62	6.23	6.77
BRAKE LEVER GUIDE OR CARRIER	0.04	0.13	0.20	0.26	0.31	0.35	0.38	0.41
DEAD LEVER GUIDE	0.01	0.04	0.17	0.35	0.54	0.68	0.83	0.95
DEAD LEVER GUIDE BRACKET	0.01	0.07	0.18	0.33	0.47	0.58	0.69	0.78
BRAKE SHOES	829.17	1566.96	2244.76	2908.14	3532.41	4008.58	4422.74	4783.23
BRAKE SHOE KEYS	4.42	9.97	14.80	19.28	23.39	26.46	29.09	31.37
WHEELSETS	1277.13	2725.38	3989.16	5180.33	6282.29	7143.20	7870.58	8513.07
LUBRICATE ROLLER BEARINGS	65.26	99.92	123.86	144.48	161.50	174.37	184.80	193.54
ROLLER BEARINGS	164.05	357.83	527.41	684.28	829.56	950.14	1052.56	1144.32
ROLLER BEARING CAP SCREWS	0.05	0.19	0.35	0.48	0.59	0.69	0.78	0.85
ROLLER BEARING LOCKING PLATES	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
ROLLER BEARING LUBRICATION FITTING	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
PEDESTAL ADAPTERS	70.59	145.62	210.52	278.64	325.60	369.07	406.94	437.82
WHEELS	393.80	902.50	1374.73	1823.49	2243.67	2587.52	2885.59	3152.30
WHEEL LABOR	577.52	1206.54	1733.48	2232.57	2691.76	3027.50	3303.57	3543.40
AXLES, ROLLER BEARINGS	5.84	12.74	18.79	24.38	29.56	33.86	37.52	40.79
OTHER TRUCK REPAIRS	419.11	1344.76	2253.48	3082.19	3837.43	4439.12	4957.06	5390.66
TRUCK BOLSTERS	109.74	408.24	706.21	976.76	1223.17	1422.92	1594.73	1738.31
TRUCK BOLSTERS (REPAIRED)	22.43	32.10	41.61	52.50	62.89	70.55	76.69	82.12
CENTER PINS	5.30	14.61	24.98	35.32	45.33	52.70	59.29	64.86
CENTER PLATES	3.17	7.62	15.70	24.42	33.48	40.46	46.87	52.91
CENTER PLATE LINERS	33.38	65.68	98.03	131.08	161.70	184.48	204.02	221.23
TRUCK SIDE BEARINGS	14.21	36.37	54.93	71.38	85.76	96.88	106.31	114.31
FRICTION CASTINGS	32.68	103.96	167.27	224.14	275.10	314.16	346.99	375.29
SIDE BEARING SHIM	0.59	1.62	2.42	3.04	3.58	3.98	4.32	4.57
SIDE FRAMES	146.46	523.83	903.28	1248.43	1563.59	1818.44	2039.67	2222.26
SIDE FRAMES (REPAIRED)	1.26	5.07	7.93	10.04	11.92	13.54	14.60	15.21
SPRING PLANKS	0.00	0.04	0.07	0.09	0.11	0.12	0.14	0.15
OUTER SPRINGS	17.97	59.40	95.78	126.29	152.42	172.55	189.29	203.24
INNER SPRINGS	10.08	31.63	50.12	65.48	78.87	88.82	97.19	104.08
STABILIZER SPRINGS	9.18	28.36	44.93	58.99	71.32	80.58	88.33	94.68
TRUCK SPRING FRICTION SNUBBER	0.13	0.23	0.30	0.34	0.38	0.41	0.43	0.44
TRUCK SPRING PLATES	0.88	1.06	1.13	1.20	1.20	1.21	1.21	1.22
TRUCK SPRING SHIM, WOOD	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.01
STEEL	0.01	0.03	0.07	0.11	0.16	0.19	0.23	0.26
MANUFACTURED MATERIAL (TRUCK)	11.64	24.89	38.72	52.61	65.03	77.10	86.76	95.50
TRUCK TOTAL	2620.17	5907.28	8936.16	11805.82	14471.43	16561.02	18355.16	19909.82

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-5A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR 70-TON, NORMAL SERVICE, ROLLER BEARING CARS

SUMMARY TABLE

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	69.14	131.37	188.01	245.92	300.86	351.42	408.32	457.10
COUPLERS, YOKES, & DRAFT GEAR	22.59	67.62	116.63	165.55	215.84	262.43	313.85	359.41
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	85.02	218.88	331.06	451.62	528.82	621.36	721.56	804.16
OTHER CAR REPAIRS	64.80	124.21	182.27	242.98	304.52	359.41	420.16	472.42
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	98.01	194.88	285.73	375.87	467.60	554.13	649.26	732.30
WHEELSETS	135.48	289.11	423.17	549.53	675.04	795.04	924.48	1038.00
OTHER TRUCK REPAIRS	44.46	142.65	239.05	326.96	412.34	494.07	582.26	657.29
HEAVY REPAIRS	121.40	275.59	360.76	416.53	455.79	504.43	555.71	598.58
TOTAL	640.89	1444.31	2126.67	2752.96	3360.00	3942.30	4575.40	5119.26
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	69.14	131.37	188.01	245.92	300.86	351.42	408.32	457.10
COTS	22.73	44.74	65.05	84.96	105.02	123.22	143.32	160.42
IDTS	30.62	56.14	78.38	100.19	121.69	141.09	162.54	180.89
PRESSURE SYSTEM	13.57	25.61	37.35	49.24	61.58	73.12	86.08	97.25
HAND BRAKES	2.22	4.89	7.25	9.52	11.76	13.99	16.38	18.54
COUPLERS, YOKES, & DRAFT GEAR	22.59	67.62	116.63	165.55	215.84	262.43	313.85	359.41
COUPLER BODIES	5.34	17.95	30.07	41.12	52.01	62.28	73.35	83.20
COUPLER KNUCKLES	5.23	12.77	20.38	27.89	35.48	42.76	50.75	57.91
OTHER COUPLER PARTS	9.16	20.65	31.76	42.59	53.50	63.75	75.09	85.04
YOKES	0.83	5.08	10.63	16.49	22.70	28.13	34.21	39.48
DRAFT GEARS, CARRIERS, AND FOLLOWERS	2.03	11.17	23.78	37.47	52.15	65.52	80.46	93.78
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	85.02	218.88	331.06	451.62	528.82	621.36	721.56	804.16
OTHER CAR REPAIRS	64.80	124.21	182.27	242.98	304.52	359.41	420.16	472.42
OTHER CAR REPAIRS	36.13	65.52	95.22	126.91	159.46	188.81	221.23	249.42
WELDING	13.38	29.98	46.30	63.34	80.40	95.17	111.80	125.83
NON BILLABLE INSPECTIONS	15.29	28.71	40.75	52.72	64.66	75.43	87.13	97.17
CAR TOTAL	241.54	542.08	817.97	1084.07	1349.23	1594.62	1868.69	2093.08

TRUCK REPAIRS

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	98.01	194.88	285.73	375.87	467.60	554.13	649.26	732.30
BRAKE BEAMS	8.21	24.92	42.21	60.42	79.51	97.98	118.14	136.02
BRAKE BEAM WEAR PLATES	0.02	0.05	0.08	0.09	0.10	0.12	0.13	0.14
BRAKE BEAM HANGERS	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.02
BRAKE HANGER OR CONNECTION PIN	0.52	0.97	1.39	1.81	2.23	2.63	3.07	3.46
BOTTOM ROD SAFETY SUPPORT	0.10	0.22	0.33	0.44	0.56	0.68	0.80	0.91
BRAKE BEAM SAFETY SUPPORT	0.08	0.17	0.23	0.27	0.30	0.34	0.36	0.40
BRAKE CONNECTION, BOTTOM	0.16	0.34	0.49	0.63	0.76	0.89	1.04	1.16
BRAKE CONNECTION, TOP	0.38	0.66	0.89	1.12	1.36	1.58	1.83	2.04
BRAKE LEVER	0.11	0.23	0.33	0.43	0.53	0.63	0.73	0.82
BRAKE LEVER GUIDE OR CARRIER	0.00	0.01	0.02	0.03	0.03	0.04	0.04	0.05
DEAD LEVER GUIDE	0.00	0.00	0.02	0.04	0.06	0.08	0.10	0.12
DEAD LEVER GUIDE BRACKET	0.00	0.01	0.02	0.03	0.05	0.07	0.08	0.10
BRAKE SHOES	87.96	166.22	238.12	308.49	379.56	446.16	519.50	583.22
BRAKE SHOE KEYS	0.47	1.06	1.57	2.04	2.51	2.95	3.42	3.83
WHEELSETS	135.48	289.11	423.17	549.53	675.04	795.04	924.48	1038.00
LUBRICATE ROLLER BEARINGS	6.92	10.60	13.14	15.33	17.35	19.41	21.67	23.60
ROLLER BEARINGS	17.40	37.96	55.95	72.59	89.14	105.75	123.63	139.53
ROLLER BEARING CAP SCREWS	0.01	0.02	0.04	0.05	0.06	0.08	0.09	0.10
PEDESTAL ADAPTERS	7.49	15.45	22.33	28.71	34.99	41.08	47.69	53.38
WHEELS	41.77	95.74	145.83	193.44	241.09	287.99	338.94	384.36
WHEEL LABOR	61.26	127.99	183.89	236.83	289.23	336.96	388.04	432.05
AXLES, ROLLER BEARINGS	0.62	1.35	1.99	2.59	3.18	3.77	4.41	4.97
OTHER TRUCK REPAIRS	44.46	142.65	239.05	326.96	412.34	494.07	582.26	657.29
TRUCK BOLSTERS	11.64	43.31	74.91	103.61	131.43	158.37	187.32	211.95
TRUCK BOLSTERS (REPAIRED)	2.38	3.40	4.41	5.57	6.76	7.85	9.01	10.01
CENTER PINS	0.56	1.55	2.65	3.75	4.87	5.87	6.96	7.91
CENTER PLATES	0.34	0.81	1.66	2.59	3.60	4.50	5.51	6.45
CENTER PLATE LINERS	3.54	6.97	10.40	13.90	17.38	20.53	23.96	26.97
TRUCK SIDE BEARINGS	1.51	3.86	5.83	7.57	9.22	10.78	12.49	13.94
FRICTION CASTINGS	3.47	11.05	17.74	23.78	29.56	34.97	40.76	45.76
SIDE BEARING SHIM	0.06	0.17	0.26	0.32	0.38	0.44	0.51	0.56
SIDE FRAMES	15.54	55.57	95.82	132.45	168.05	202.39	239.98	270.96
SIDE FRAMES (REPAIRED)	0.13	0.54	0.84	1.06	1.28	1.51	1.72	1.85
SPRING PLANKS	0.00	0.00	0.01	0.01	0.01	0.01	0.02	0.02
OUTER SPRINGS	1.91	6.30	10.16	13.40	16.88	19.20	22.23	24.78
INNER SPRINGS	1.07	3.36	5.32	6.95	8.47	9.89	11.42	12.69
STABILIZER SPRINGS	0.97	3.01	4.77	6.26	7.66	8.97	10.38	11.54
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.03	0.04	0.04	0.05	0.05	0.05
TRUCK SPRING PLATES	0.09	0.11	0.12	0.13	0.13	0.13	0.14	0.15
STEEL	0.00	0.00	0.01	0.01	0.02	0.02	0.03	0.03
MANUFACTURED MATERIAL (TRUCK)	1.23	2.64	4.11	5.58	7.09	8.58	10.19	11.64
TRUCK TOTAL	277.95	626.64	947.95	1252.36	1554.98	1843.24	2156.08	2427.59

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-6. PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR 70-TON, HIGH MILEAGE, ROLLER BEARING CARS

SUMMARY TABLE

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	655.40	1039.98	1424.37	1818.09	2196.11	2487.48	2736.81	2950.46
COUPLERS, YOKES, & DRAFT GEAR	171.29	388.65	650.85	927.86	1205.20	1430.14	1630.78	1806.42
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	1968.77	2956.10	3790.60	4572.66	5296.18	5868.25	6335.99	6758.72
OTHER CAR REPAIRS	551.05	1098.49	1754.20	2457.64	3150.31	3669.49	4113.37	4496.73
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	2300.72	3215.88	4142.68	5089.75	6054.67	6808.06	7450.51	8031.73
WHEELSETS	1662.61	2769.01	3775.91	4741.22	5677.66	6382.89	7015.02	7545.78
OTHER TRUCK REPAIRS	19.19	117.88	246.67	366.75	477.95	569.02	649.01	720.04
HEAVY REPAIRS	12.28	25.90	40.41	59.50	79.03	93.34	105.53	116.53
TOTAL	7341.31	11611.90	15825.68	20033.48	24137.10	27308.67	30037.02	32426.41
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	655.40	1039.98	1424.37	1818.09	2196.11	2487.48	2736.81	2950.46
COT&S	10.04	40.83	77.99	113.52	147.03	172.68	194.85	213.83
IDT&S	474.75	678.87	863.22	1050.30	1228.56	1359.44	1469.74	1564.90
PRESSURE SYSTEM	164.50	301.18	449.63	606.07	758.51	881.59	988.44	1079.21
HAND BRAKES	6.12	19.09	33.53	48.20	62.00	73.77	83.78	92.52
COUPLERS, YOKES, & DRAFT GEAR	171.29	388.65	650.85	927.86	1205.20	1430.14	1630.78	1806.42
COUPLER BODIES	18.68	70.94	167.07	275.98	388.17	481.88	564.49	637.13
COUPLER KNUCKLES	100.11	168.19	231.88	295.51	356.89	406.37	452.09	490.78
OTHER COUPLER PARTS	38.34	86.00	137.61	188.96	238.03	278.13	312.38	342.04
YOKES	7.38	18.68	31.92	45.82	60.38	70.95	80.53	89.26
DRAFT GEARS, CARRIERS, AND FOLLOWERS	14.78	44.84	82.38	121.60	161.73	192.81	221.30	247.22
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	1968.77	2956.10	3790.60	4572.66	5296.18	5868.25	6335.99	6758.72
OTHER CAR REPAIRS	551.05	1098.49	1754.20	2457.64	3150.31	3669.49	4113.37	4496.73
OTHER CAR REPAIRS	238.95	510.90	879.84	1288.28	1699.96	2010.94	2280.13	2513.62
WELDING	119.58	293.81	477.37	666.41	846.31	980.51	1092.59	1186.96
NON BILLABLE INSPECTIONS	192.52	293.77	396.99	502.94	604.04	678.04	740.66	796.15
CAR TOTAL	3346.51	5483.21	7620.02	9776.25	11847.79	13485.86	14816.95	16012.34

TRUCK REPAIRS

	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	2300.72	3215.88	4142.68	5089.75	6054.67	6808.06	7450.51	8031.73
BRAKE BEAMS	130.54	266.44	439.63	615.21	815.52	967.02	1099.02	1222.88
BRAKE BEAM WEAR PLATES	0.02	0.11	0.18	0.23	0.27	0.31	0.33	0.34
BRAKE HANGER OR CONNECTION PIN	1.07	5.86	7.23	10.64	13.81	16.39	18.59	20.49
BOTTOM ROD SAFETY SUPPORT	0.71	3.63	7.30	11.65	15.98	19.47	22.48	25.12
BRAKE BEAM SAFETY SUPPORT	0.01	0.02	0.03	0.04	0.06	0.07	0.08	0.09
BRAKE CONNECTION, BOTTOM	1.11	2.99	4.92	6.88	8.62	10.07	11.34	12.40
BRAKE CONNECTION, TOP	0.98	3.86	7.32	11.00	14.49	17.33	19.78	21.92
BRAKE LEVER	0.62	1.60	2.68	3.78	4.77	5.61	6.33	6.95
BRAKE LEVER GUIDE OR CARRIER	0.01	0.05	0.08	0.12	0.15	0.17	0.19	0.21
DEAD LEVER GUIDE	0.01	0.05	0.22	0.45	0.70	0.89	1.08	1.23
DEAD LEVER GUIDE BRACKET	0.01	0.06	0.17	0.30	0.44	0.55	0.64	0.73
BRAKE SHOES	2160.89	2925.89	3662.13	4416.07	5164.06	5752.43	6251.26	6698.57
BRAKE SHOE KEYS	5.54	8.11	18.78	13.38	15.82	17.76	19.37	20.80
WHEELSETS	1662.61	2769.01	3775.91	4741.22	5677.66	6382.89	7015.02	7545.78
LUBRICATE ROLLER BEARINGS	3.63	12.54	20.47	27.33	33.36	38.23	42.25	45.23
ROLLER BEARINGS	247.67	404.92	548.01	685.01	819.57	926.81	1025.24	1108.62
ROLLER BEARING CAP SCREWS	0.01	0.03	0.06	0.08	0.09	0.11	0.12	0.13
ROLLER BEARING LOCKING PLATES	0.01	0.05	0.08	0.11	0.12	0.13	0.15	0.15
ROLLER BEARING LUBRICATION FITTING	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
PEDESTAL ADAPTERS	43.37	88.43	133.32	175.17	216.60	248.43	275.34	299.15
WHEELS	591.17	962.72	1300.67	1623.64	1940.93	2193.28	2425.25	2621.28
WHEEL LABOR	767.92	1285.86	1753.73	2205.42	2637.70	2942.79	3210.05	3431.56
AXLES, ROLLER BEARINGS	8.84	14.46	19.57	24.46	29.27	33.10	36.61	39.59
OTHER TRUCK REPAIRS	19.19	117.88	246.67	366.75	477.95	569.02	649.01	720.04
TRUCK BOLSTERS	2.58	24.73	57.42	87.50	115.54	139.26	159.47	177.77
TRUCK BOLSTERS (REPAIRED)	0.25	0.88	0.75	1.10	1.47	1.73	1.96	2.16
CENTER PINS	0.45	1.81	3.64	5.42	7.08	8.37	9.53	10.50
CENTER PLATES	0.02	0.11	0.30	0.50	0.71	0.89	1.04	1.18
CENTER PLATE LINERS	1.59	13.02	25.03	36.95	47.98	56.40	63.56	69.89
TRUCK SIDE BEARINGS	0.95	4.92	9.33	13.34	16.90	19.84	22.25	24.44
FRICTION CASTINGS	0.92	6.42	13.30	19.24	24.52	28.70	32.33	35.57
SIDE BEARING SHIM	0.04	0.50	1.04	1.51	1.90	2.20	2.45	2.69
SIDE FRAMES	4.11	38.90	98.40	137.87	181.94	219.28	251.49	280.23
SIDE FRAMES (REPAIRED)	0.16	0.34	0.54	0.79	1.05	1.24	1.40	1.54
OUTER SPRINGS	0.61	4.01	7.84	11.01	13.73	15.86	17.78	19.38
INNER SPRINGS	0.34	1.86	3.60	5.02	6.23	7.16	8.01	8.70
STABILIZER SPRINGS	0.10	0.72	1.62	2.35	2.98	3.48	3.92	4.30
STEEL	5.41	13.38	19.35	25.72	31.86	35.98	40.98	45.18
MANUFACTURED MATERIAL (TRUCK)	1.36	4.68	12.51	18.43	24.06	28.64	32.85	36.52
TRUCK TOTAL	3982.51	6102.78	8165.26	10197.72	12210.27	13759.97	15114.54	16297.55

DATA SHOWN USING 10% DISCOUNT RATE

TABLE D-6A. EQUIVALENT ANNUAL EXPENDITURE ON ALL REPAIRS FOR 70-TON, HIGH MILEAGE, ROLLER BEARING CARS

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	69.52	110.32	151.10	192.86	235.97	276.86	321.47	359.75
COUPLERS, YOKES, & DRAFT GEAR	18.17	41.23	69.04	98.43	129.50	159.17	191.55	220.26
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	208.85	313.58	402.11	485.07	569.08	653.14	744.23	824.09
OTHER CAR REPAIRS	58.45	116.53	186.08	260.71	338.51	408.41	483.16	548.29
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	244.06	341.14	439.45	539.92	650.58	757.74	875.14	979.31
WHEELSETS	176.37	293.74	400.55	502.95	610.07	710.42	823.98	920.06
OTHER TRUCK REPAIRS	2.04	12.51	26.17	38.90	51.36	63.33	76.23	87.79
HEAVY REPAIRS	1.30	2.75	4.29	6.31	8.49	10.39	12.40	14.21
TOTAL	778.76	1231.79	1678.78	2125.14	2593.57	3039.46	3528.15	3953.77
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	30.00	28.80	24.00	20.57	18.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	69.82	110.32	151.10	192.86	235.97	276.86	321.47	359.75
COTS	1.06	4.33	8.27	12.04	15.80	19.22	22.89	26.07
IOTS	50.36	72.01	91.57	111.42	132.01	151.31	172.64	190.81
PRESSURE SYSTEM	17.45	31.95	47.70	64.29	81.50	98.12	116.10	131.59
HAND BRAKES	0.05	2.03	3.56	5.11	6.66	8.21	9.84	11.28
COUPLERS, YOKES, & DRAFT GEAR	18.17	41.23	69.04	98.43	129.50	159.17	191.55	220.26
COUPLER BODIES	1.13	7.53	17.72	29.28	41.71	53.63	66.30	77.69
COUPLER KNUCKLES	10.62	17.84	24.68	31.35	38.35	45.23	53.10	59.84
OTHER COUPLER PARTS	4.07	9.12	14.60	20.04	25.58	30.96	36.69	41.70
YOKES	0.78	1.98	5.39	4.86	6.49	7.90	9.46	10.88
DRAFT GEARS, CARRIERS, AND FOLLOWERS	1.57	4.76	8.74	12.90	17.38	21.46	25.99	30.14
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	208.85	313.58	482.11	485.07	569.08	653.14	744.23	824.09
OTHER CAR REPAIRS	58.45	116.53	186.08	260.71	338.51	408.41	483.16	548.29
OTHER CAR REPAIRS	25.35	54.20	93.33	136.66	182.66	223.82	267.82	306.49
WELDING	12.69	31.17	50.64	70.69	90.94	109.13	128.34	144.73
NON BILLABLE INSPECTIONS	20.42	31.16	42.11	53.35	64.91	75.47	87.00	97.08
CAR TOTAL	395.00	561.66	808.35	1037.06	1273.06	1497.58	1740.40	1952.39

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	244.06	341.14	439.45	539.92	650.58	757.74	875.14	979.31
BRAKE BEAMS	13.85	28.26	46.64	65.26	87.63	107.63	129.09	149.11
BRAKE BEAM WEAR PLATES	0.00	0.01	0.02	0.02	0.03	0.03	0.04	0.04
BRAKE HANGER OR CONNECTION PIN	0.11	0.41	0.77	1.13	1.46	1.82	2.18	2.50
BOTTOM ROD SAFETY SUPPORT	0.08	0.39	0.77	1.24	1.72	2.17	2.64	3.06
BRAKE BEAM SAFETY SUPPORT	0.00	0.00	0.00	0.00	0.01	0.01	0.01	0.01
BRAKE CONNECTION, BOTTOM	0.12	0.32	0.52	0.73	0.93	1.12	1.33	1.51
BRAKE CONNECTION, TOP	0.10	0.41	0.76	1.17	1.56	1.93	2.32	2.67
BRAKE LEVER	0.07	0.17	0.28	0.40	0.51	0.62	0.74	0.85
BRAKE LEVER GUIDE OR CARRIER	0.00	0.00	0.01	0.01	0.02	0.02	0.02	0.03
DEAD LEVER GUIDE	0.00	0.01	0.02	0.05	0.07	0.10	0.13	0.15
DEAD LEVER GUIDE BRACKET	0.00	0.01	0.02	0.03	0.05	0.06	0.08	0.09
BRAKE SHOES	229.14	310.29	388.48	468.45	554.89	640.25	734.27	816.76
BRAKE SHOE KEYS	0.59	0.86	1.14	1.42	1.70	1.98	2.28	2.54
WHEELSETS	176.37	293.74	400.55	502.95	610.07	710.42	823.98	920.06
LUBRICATE ROLLER BEARINGS	0.38	1.33	2.17	2.90	3.58	4.25	4.96	5.52
ROLLER BEARINGS	26.27	42.95	58.13	72.67	88.06	103.15	120.42	135.17
ROLLER BEARING CAP SCREWS	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.02
ROLLER BEARING LOCKING PLATES	0.00	0.01	0.01	0.01	0.01	0.01	0.02	0.02
PEDESTAL ADAPTERS	4.60	9.38	14.14	18.58	23.27	27.65	32.34	36.48
WHEELS	62.71	102.12	137.97	172.23	208.56	244.11	284.87	319.61
WHEEL LABOR	81.46	136.40	186.04	233.95	283.42	327.53	377.05	418.41
AXLES, ROLLER BEARINGS	0.94	1.53	2.08	2.59	3.14	3.68	4.30	4.83
OTHER TRUCK REPAIRS	2.04	12.51	26.17	38.90	51.36	63.33	76.23	87.79
TRUCK BOLSTERS	0.27	2.42	6.09	9.28	12.41	15.50	18.73	21.68
TRUCK BOLSTERS (REPAIRED)	0.02	0.05	0.08	0.12	0.16	0.19	0.23	0.26
CENTER PINS	0.05	0.15	0.38	0.57	0.76	0.93	1.12	1.28
CENTER PLATES	0.00	0.01	0.03	0.05	0.08	0.10	0.12	0.14
CENTER PLATE LINERS	0.20	1.38	2.66	3.92	5.16	6.28	7.47	8.52
TRUCK SIDE BEARINGS	0.10	0.52	0.99	1.41	1.82	2.21	2.61	2.98
FRICTION CASTINGS	0.10	0.68	1.41	2.04	2.63	3.19	3.80	4.34
SIDE BEARING SHIM	0.00	0.05	0.11	0.16	0.20	0.24	0.29	0.33
SIDE FRAMES	0.44	4.13	9.59	14.63	19.55	24.41	29.54	34.17
SIDE FRAMES (REPAIRED)	0.02	0.04	0.06	0.08	0.11	0.14	0.16	0.19
OUTER SPRINGS	0.07	0.43	0.83	1.17	1.48	1.77	2.09	2.36
INNER SPRINGS	0.04	0.20	0.78	0.53	0.67	0.80	0.94	1.06
STABILIZER SPRINGS	0.01	0.08	0.17	0.25	0.32	0.39	0.46	0.52
STEEL	0.57	1.42	2.05	2.73	3.42	4.00	4.81	5.51
MANUFACTURED MATERIAL (TRUCK)	0.15	0.71	1.33	1.95	2.58	3.19	3.86	4.45
TRUCK TOTAL	422.46	647.38	866.17	1081.77	1312.01	1531.49	1775.35	1987.17

DATA SHOWN USING 10% DISCOUNT RATE