REPORT NO. FRA/ORD-80/59

TDOP

TRUCK DESIGN OPTIMIZATION PROJECT PHASE II

INTERIM REPORT

WYLE LABORATORIES
SCIENTIFIC SERVICES & SYSTEMS GROUP

Colorado Springs Division
4620 Edison Avenue
Colorado Springs, Colorado 80915

DEPARTMENT OF TRANSPORTATION • UNITED STATES OF AMERICA

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NOTICE

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16. Abstract

The overall objective of the TDOP Phase II project is to characterize the behavior of existing trucks (defined as Type I trucks) and to generate performance and test specifications for new truck designs (referred to as Type II trucks). The purpose of this document is to report on the progress to date of TDOP Phase II. The report summarizes the methodology employed to develop performance and test specifications and the rationale used for selecting seven Type II trucks for field testing. The Interim Report also discusses preliminary conclusions reached in several economic areas, including car and truck maintenance costs, fuel consumption, and rail wear in curves. Further, the report describes on-going field test programs (the truck performance testing and wear measurement programs) and the completed Friction Snubber Force Measurement System test program. In the analysis area, an assessment and validation of 17 computer simulation models of freight cars is discussed in detail. The Interim Report concludes with a description of the work-in-progress for a Type I truck performance characterization.

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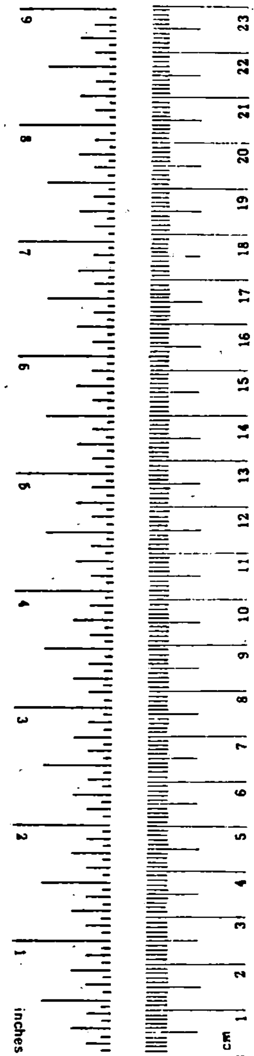
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

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

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



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	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
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m^3	cubic meters	1.3	cubic yards	yd^3
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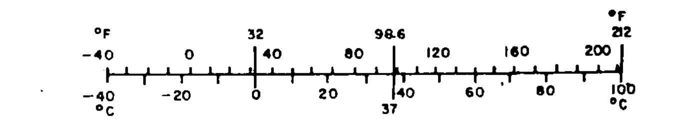
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



EXECUTIVE SUMMARY

As the major component between the lading and the track, the freight car truck performs the essential functions of guidance, support, and vibration absorption to 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 out the deficiencies of the standard truck.

In response to the need of the rail industry for a better freight car truck design, the Federal Railroad Administration (FRA) is sponsoring a broad-based research program: the Truck Design Optimization Project (TDOP). Its 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 by operational and economic information, these specifications will not only provide the technical base for design innovation, but also facilitate its easy 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 "improved" or Type II truck is defined as a truck whose design features bring about functional differences in truck and carbody behavior. In the context of TDOP, the main restriction placed on a Type II truck is that it preserve coupler height, but the method of mounting the wheelsets on the frame and of supporting the carbody are not specified.

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.

The TDOP project is now in Phase II, with Wyle Laboratories as the prime contractor and the Union Pacific Railroad as the principal subcontractor. The objectives of TDOP Phase II are:

- To define the performance of both Type I and Type II trucks in quantitative terms, represented by performance indices.

- To establish a plan for collecting economic data on the cost of acquiring, operating, and maintaining the standard, Type I truck.

- To determine a quantitative basis for evaluating the economic benefits to be derived from Type II trucks.

- To generate performance characterizations for Type I trucks and performance and test specifications for Type II trucks.

These objectives are being 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.

Most of these activities are occurring concurrently; for example, developing and refining a methodology; field testing of the trucks; economic data collection and analysis; and assessment, validation, and use of computer models. Those activities will soon culminate in the establishment of a Type I truck performance characterization document. A Type II performance specification, a test specification for Type I and II trucks, a cost/benefit analysis, and a final report will be produced at the conclusion of the project.

TDOP Phase II's accomplished milestones are described below.

Engineering

In the engineering analysis area, a methodology has been developed to define the four principal performance regimes which, when combined with their associated indices, can be used to quantify truck performance. This methodology also determined how the output from the project's five major tasks (field testing, computer modeling, wear data collection, economic analysis, and engineering interpretation) would be integrated into a performance specification.

A Type II truck selection criteria was established and seven trucks selected for testing as representative of the Type II truck population. The seven Type II trucks are: the NRUC Maxiride, the Dresser DR-1 Steering Assembly, the National Swing Motion, the Devine-Scales, the Barber-Scheffel Radial, the ACF Fabricated, and the Alusuisse. Wyle Laboratories worked closely with the railroad industry during the selection process.

A draft Type I truck performance characterization document has been prepared based on test data from Phase I and will be distributed when the data acquired from the Type I truck field test program of TDOP Phase II are analyzed.

Field Testing

Prior to field testing the Type I and Type II trucks, a determination was first made of the usefulness of the Phase I test data; secondly, an automatic location detection system was installed in the test zones selected in the Union Pacific's South Central District, California Division; and finally, track geometry measurements of the six test zones were made by the FRA's track geometry survey car, T-6. These measurements included alignment, cross level, curvature, profile, and automatic location detection.

The field test program began with a series of tests of the Friction Snubber Force Measurement System. These tests were conducted using two Type I trucks (an ASF Ride Control and Barber S-2) with a 100-ton hopper car in empty, half-loaded, and fully loaded configuration. Forty-eight channels of data were acquired which measured truck motion, carbody rigid modes, carbody/truck relative motion, and friction snubber forces. A report was prepared which described the test, test results, and friction coefficients for each truck. See Section 1 for this report's complete title as well as a list of other documents published to date by TDOP Phase II.

In April 1979, the TDOP Phase II wear data collection program began measuring wear on six trucks: the ASF Ride Control, Barber S-2, Barber C-PEP, Dresser DR-1, National Swing Motion, and the Barber-Scheffel. A seventh, the Devine-Scales truck, was added in January, 1980. The trucks are in a unit coal train that makes a 1600-mile round trip between Colorado and California. When this program concludes in October 1980, several of the trucks will have accumulated over 100,000 miles.

Two key measurements missing from the TDOP Phase I test program were measurement of the angle of attack of the wheel relative to the rail and a measurement of the lateral over vertical (L/V) forces at the wheel/rail interface. Techniques have been developed and measurement systems acquired in TDOP Phase II to measure both of these quantities during Type I and Type II truck testing.

TDOP Phase II has completed Type I truck testing on a 100-ton ASF Ride Control truck to supplement the data acquired during Phase I. The truck was tested in both an unloaded and loaded (with coal) configuration over a variety of track. Primary measurements acquired were the angle of attack and L/V forces. In addition, truck motion, rigid carbody modes, truck/carbody relative motions, and accelerations were measured and recorded. The data from this test program, which concluded in March 1980, are being analyzed.

Type II truck testing began in April 1980 with the Dresser DR-1 truck under loaded and unloaded carbodies. Type II truck testing will conclude in September 1980 with the testing of the seventh truck, the Alusuisse.

Economics

In the economics area, considerable progress has been made in determining maintenance costs. Total repair costs by component have been tabulated for cars with known annual mileage. Thus, changes in wear measured in a service test program (e.g., the TDOP Phase II wear data collection program) can be related to dollar savings. Work is progressing in the fuel consumption study. Specifically, rolling resistance of a single car operating normally in curves has been successfully measured. The conventional rolling resistance equation used by the railroad industry has been extended to take into account off-balance speed and spiral negotiation behavior in curves. More work is underway to relate rolling resistance directly to dollar savings.

To determine the economic impact of Type I and Type II trucks on rail wear, TDOP Phase II has enlisted the aid of the Canadian Institute of Guided Ground Transport (CIGGT) under the Joint Research Project Agreement between the United States and Canada. TDOP Phase II will supply measured L/V and angle of attack for CIGGT's rail wear model. The CIGGT model has been under development for approximately three years and successfully simulates actual data on rail deterioration.

Analysis

TDOP Phase II's work in the analysis area has resulted in the development of an analysis plan, an analytical tool assessment report, and a validation report. The analysis plan defined the means by which Type I and Type II freight car trucks will be characterized and compared. The specific objectives were: 1) to define the requirements for field test data and simulated data which will establish performance specifications; and 2) to determine the extent to which field test data can be extrapolated.

The second part of the analysis task was to perform an assessment of the analytical tools (i.e., computer simulation models) that could be used in TDOP Phase II. After establishing an assessment criteria and conducting a preliminary survey of nearly 60 models, a detailed assessment was made of 17 of those models. The results of this assessment are available in a document published by the National Technical Information Service (NTIS). See Section 1 for the complete title.

Validating the candidate models selected was the third part of the analysis task. Unfortunately, the results of the validation exercises carried out thus far have been rather disappointing. 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. During the remainder of TDOP Phase II, greater emphasis will be placed on ad hoc modeling. The aim of such modeling will be the interpretation of test results. Simple models will be used to determine why a vehicle exhibits nosing, for instance, rather than attempt to construct a comprehensive hunting model.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

SECTION 1 - INTRODUCTION

The purpose of this Interim Report is to document tasks 1 through 7, and 10 of the Truck Design Optimization Project, Phase II (FRA contract DOT-FR-742-4277). A cross-reference between the Statement of Work (SOW) tasks and the corresponding sections of this report is shown in Table 1-1.

Table 1-1. Cross Reference of SOW Tasks with Report Sections

Task	Title	Report Section
1, 2.1	Definition of performance indices, and economic relationships; preparation of Introductory Report	2
2, 10	Establish requirements for additional economic data; prepare detailed plan for economic data collection and analysis	3, Appendix A
3, 4, 5	Assessment of analysis tools; development of an analysis plan; validation of analytical tools	4
6	Conduct tests and analyses	5
7	Establish formal methodology for truck evaluation	6

The material in this report has been drawn largely from documents published during the course of the TDOP Phase II project. The documents listed below with an FRA prefix are available through the National Technical Information Service (NTIS). In addition, we have listed several internal documents that are available through Wyle Laboratories, Colorado Springs Division.

NTIS Documents

- Report No. FRA/ORD-78/53, TDOP Phase II Introductory Report, November 1978.
- Report No. FRA/ORD-78/34, Phase I Data Evaluation and Analysis Plan, September 1978.
- Report No. FRA/ORD-78/52, Phase I Data Evaluation and Analysis Report, August 1979.
- Report No. FRA/ORD-79/24, Friction Snubber Force Measurement System Field Test Report, October 1979.
- Report No. FRA/ORD-78/69, Measurement of Friction Snubber Forces in Freight Car Trucks, December 1978 (prepared under FRA contract DOT-FR-T-4263).
- Report No. FRA/ORD-79/36, Analytical Tool Assessment Report, August 1979.
- Report No. FRA/ORD-80/31, Analysis Plan, March 1980.

Wyle Laboratories Documents

- Document No. C-901-0004-A, Type I Truck Test Plan, April 13, 1979 with revisions A and B.
- Document No. C-901-0008-A, Type I Truck Test Procedure, July 25, 1979 with revision A.
- Document No. C-901-0002-A, Wear Data Collection Plan, October 6, 1978 with revision A.
- Document No. C-901-0006-A, Wear Data Collection Procedure, November 1979.
- TDOP Technical Report TR-09, Type II Truck Selection, May 22, 1979.
- Document No. C-901-0007, Type II Truck Test Plan, October 1979 with revision A.
- TDOP Technical Report TR-10, Performance Characterization - Type I Trucks, September 17, 1979 (preliminary draft).

Acknowledgements

A number of individuals have contributed to the TDOP Phase II project. Their support is hereby gratefully acknowledged. A particular debt is owed Mr. Frank D. Bruner, the Assistant Chief Mechanical Officer, Union Pacific Railroad. Mr. Bruner and the Union Pacific crews under his direction have contributed many long hours supporting the various TDOP Phase II field test programs.

The TDOP Phase II project has been ably assisted by 14 consultants drawn from a cross section of the railroad industry. These individuals, who have provided Wyle Laboratories with the benefit of their expert knowledge, are:

Mr. Robert Bullock
Manager, Research & Development
Standard Car Truck Company

Mr. Bernard Eck
Director of Product Engineering
Griffin Wheel Company

Mr. Ross Gill
Manager of Research & Tests
Southern Pacific Transportation Co.

Mr. Larry Greenfield
Ass't to Vice President-Equipment
Trailer Train Company

Mr. Leo Lombardo
Manager - Research
Consolidated Rail Corporation

Mr. Al Martin
Manager, Product Engineering
Dresser Transp. Equipment Div.

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. Keith Hawthorne
Director, Track Train Dynamics
AAR Technical Center

Mr. Robert Radford
Chief Mech. & Elec. Engineer
Canadian National Railways

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

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

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

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SECTION 2 - PROJECT METHODOLOGY

The purpose of this methodology section is first to define the key terms that will be used in developing performance characterizations and performance specifications for the Type I and Type II trucks, respectively. Secondly, the section will show how testing, data acquisition/reduction, computer modeling, and engineering analysis will be used to develop the performance characterization/specifications. Finally, this section will identify the Type II trucks selected for testing and the criteria used in making this selection.

2.1 TRUCK PERFORMANCE DEFINED

A railroad needs a means for evaluating the cost effectiveness of a freight car truck with respect to the truck's operating conditions. These conditions may differ considerably from one railroad to another. For example, a railroad operating primarily in mountainous territory at relatively low speeds may be concerned with reducing wear of wheels and rails in curves. On the other hand, a railroad operating in flat terrain at high speeds will require that its trucks have high lateral stability. Railroads handling fragile cargo may be concerned with the ride quality aspect of performance. Safety and stability of the vehicle system, e.g., harmonic roll, is the concern of all operators.

A useful characterization of truck performance thus requires the identification of specific performance regimes which may be defined as sets of conditions associated with predominant features that distinguish one regime from another. Besides being distinct and non-overlapping, the set of performance regimes should be inclusive, i.e., identify all aspects of truck behavior.

In order to make possible the quantitative evaluation of truck performance, both absolute and comparative, each performance regime must be associated with performance indices, by which is meant measurable quantities typical of that regime. Examples are critical speed of hunting, lateral wheel load in curves, and minimum dynamic vertical wheel load.

A truck performance specification defines a range of performance indices for each performance regime that a truck must meet under specified operating conditions, such as speed, track quality, and degree of curves, with due regard to state of wear or other deterioration associated with age or ton mileage. Specification of performance in quantitative, operational terms will give latitude to design innovation and facilitate correlation with economic factors.

2.2 PERFORMANCE REGIMES SELECTED

Four regimes have been identified which individually can be associated with distinctly different operating conditions, and which collectively will permit an overall evaluation of truck performance. The regimes are:

- Lateral Stability (Hunting)
- Curve Negotiation
- Trackability
- Ride Quality

Lateral Stability (Hunting)

Hunting is a self-excited lateral and yaw oscillation of the truck and carbody that occurs above a certain speed (the "critical speed"). The range of the critical speed is determined by a number of factors which include the contours of the wheel tread and rail, the surface condition of the rail, the design features of the truck, the characteristics of the suspension system, and the mass and mass distribution of the carbody. With all of these parameters being equal, the contour of the wheel tread has been found to have an overriding influence on lateral stability.

Curve Negotiation

In curve negotiation, horizontal forces between the wheels and rails act to rotate the truck about the center of the curve, even though there is no relative rotation between the truck and carbody. For standard freight car trucks, the lateral force that turns the truck in the curve is usually the flange force at the outer leading wheel, and is likely to contribute to the resistance of the truck to forward motion. It is believed that this flange force is responsible for much of the wear that leads to condemnation of wheels for "thin flange."

Trackability

Trackability refers to the ability of the truck to maintain an adequate load on all four wheels under a range of track conditions, and the dynamics of the vehicle resulting from transient or periodic changes on these conditions. A combination of low vertical wheel load with a simultaneous lateral load can lead to derailment. This regime can be further subdivided into the subregimes of load equalization, curve entry/exit, and harmonic roll and bounce.

Ride Quality

Ride quality denotes a standard of performance rather than a performance regime. It is generally taken to refer to the acceleration environment in the carbody and thus reflects the capability of the truck to isolate the carbody from track irregularities. This characteristic of the truck to act as a mechanical filter is also termed transmissibility.

2.3 PERFORMANCE CRITERIA

Performance criteria express qualitatively the aspects of truck behavior considered desirable in the various performance regimes. Critieria may range from the most general, such as safety from derailment or low wear rates, to the specific, such as lateral stability or curve negotiability.

A number of more or less general performance criteria may be identified for each regime, but not all are equally suitable to characterize truck behavior in a quantitative way. Such characterization requires the selection of a measurable physical quantity that can be unambiguously associated with performance in a specific regime. The process of selecting suitable performance criteria for establishing performance indices is discussed below for each of the four major performance regimes.

Lateral Stability

The importance of selecting suitable criteria in establishing a performance index is illustrated in Table 2-1 with respect to lateral stability.

Table 2-1. Lateral Stability Criteria and Indices

Criteria	Indices
Safe Operation in Desired Speed Range	—
High Lateral Stability	—
High Critical Speed	Critical speed
Low Sensitivity to Unfavorably Worn Wheel Profiles	—
Low Lateral Accelerations Near Critical Speed	Maximum lateral accelerations

The first two criteria listed on the left are obviously important from the operational point of view but do not lend themselves easily to quantitative expression. The third criterion, high critical speed, can obviously be directly translated into a number which has long served as a performance index for lateral stability. The next criterion, low sensitivity to unfavorably worn wheel profiles, represents a desirable feature in a rail vehicle, but again would be difficult to express quantitatively. The final criterion, low lateral accelerations near the critical speed, may be the basis for a useful second performance index. It allows for differences in the severity of hunting which has been observed in different truck configurations.

Curve Negotiation

The second performance regime, curve negotiation, shown in Table 2-2 again illustrates the care required in passing from an operationally defined performance criterion to a performance index based on engineering factors.

Table 2-2. Curve Negotiation Criteria and Indices

Criteria	Indices
Low Wheel & Rail Wear	--
Safety From Derailment	--
Low Lateral Wheel Load	Lateral Force on Leading Outer Wheel, per Degree of Curve, at Balance Speed

Operational considerations would lead to such criteria as safety from derailment and low wheel and rail wear. These also are difficult to quantify. In order to isolate basic differences in curve negotiation performance between different truck designs, we must eliminate a number of extraneous factors such as the effect of unbalanced centrifugal force not compensated by superelevation, and transient effects that occur during curve entry, and focus on the basic kinematic characteristics of a given truck that determine its orientation in a curve of constant radius, under the influence of creep and gravitational forces alone.

This suggests the lateral force on the outer leading wheel of the truck during steady state curving as a likely candidate for a performance index. As is well known, and has been confirmed by both road tests and mathematical modeling, this force is strongly determined by the ability of the axles to align themselves with the radius of the curve. By contrast, any property of the truck tending to increase the angle of attack, such as the parallelogramming of the standard three-piece truck, increases the lateral wheel load directed toward the inside of the curve. Thus, a suggested performance index for curve negotiation could be the lateral force on the leading outer wheel, per degree of curve, at balance speed.

The imperfections of this index are associated with differences in wheel profile which would thus have to be specified in detail in the comparison between the curve negotiation capability of different trucks, as well as in the creep coefficients that determine the angle of attack of the leading outer wheel in flange-free curving. Nevertheless, the most important characteristics of the truck in this regime are determined by its kinematics, as embodied in the relationship of the rigid components and the properties of the elastic connections between them.

Trackability

As mentioned earlier, this regime can be subdivided into load equalization, curve entry/exit, and harmonic roll and bounce.

Load Equalization. The performance index for trackability with respect to track irregularities of short wave length, such as rail joints and track twist, is the easiest to define, at least in a static sense:

Let W_H	=	sum of forces on the three most heavily loaded wheels
Let W_L	=	force on most lightly loaded wheel
θ	=	angle of twist of track within axle spacing of truck, degrees
WUI	=	wheel unloading index

Then

$$WUI=\frac{W_H/3-W_L}{W_H/3}\div\theta=1-\frac{W_L}{W_H/3}\div\theta,\ \text{degree}^{-1}$$

It may be seen that this index may vary from zero for a perfectly equalized truck (since $W_H/3 = W_L$) to $1/\theta$ for a truck with one wheel completely unloaded, or unit for unit twist in degrees.

It may be noted that the performance index for wheel unloading due to track twist is not sufficient to measure derailment potential since it does not refer to such indices as L/V ratio, or the duration of lateral wheel impact on the rail. Derailment associated with wheel unloading is a complex dynamic process which it may be impossible to define in terms of a truck performance specification. On the other hand, the proposed wheel unloading index describes a characteristic of the truck itself that could easily be specified, and which it may be possible eventually to relate to the derailment potential on the basis of test data and analyses.

Curve Entry/Exit. This is the transient response of the vehicle as it traverses the spirals between tangent and curved tracks while entering into and exiting from constant curvature tracks. The dynamics during the negotiation of these spirals tend to set up high lateral forces at the wheel/rail interface as well as an uneven distribution of vertical loads among the wheel/rail contact points. Thus, it is quite likely that larger L/V ratios may be encountered during entry/exit as compared with steady state curve negotiation. Consequently, derailment potential, rather than wheel and rail wear, is of primary concern within this subregime. The performance indices identified in this subregime are:

- LV ratio and the duration associated with it
- Peak lateral force on the wheel
- Wheel Unloading Index, as defined above in the load equalization subregime.

Harmonic Roll and Bounce. The performance index for harmonic roll and bounce (see Table 2-3), associated with vertical roll irregularities of long wave length, illustrates the difficulty of quantifying a complex dynamic regime. From an operational point of view, one obviously wants to prevent hazardous conditions, and damage to truck components, such as would occur during center plate lift-off and wheel lift. In the first case the suspension springs are compressed solidly and are thus subject to fretting damage, and during wheel lift there is always the danger of derailment when the track has even a slight curvature.

Table 2-3. Harmonic Roll Criteria and Indices

CRITERIA	INDICES
Prevention of Center Plate Lift-Off	— —
Prevention of Wheel Lift	— —
Low Maximum Roll Amplitude Under Given Excitation	Maximum Roll Angle
Rapid Decay From Maximum Amplitude	Rate of Energy Dissipation in Subbing System

When the springs are solidly compressed during harmonic roll, the motion of the carbody is determined almost entirely by the kinetic energy of roll rotation existing in the carbody. With a given excitation due to resonance with rail joint spacing, that kinetic energy is largely determined by the energy that has been dissipated in the snubbing system prior to that point. This circumstance suggests that a performance index for harmonic roll should in some way be associated with the capability of the suspension system to dissipate energy, for example, by snubbing friction. Parameters available for variation are level of friction force, distance through which this force acts (i.e., spring travel) and rate of increase of friction level with spring compression. The effectiveness of this energy dissipation system could be measured in a number of ways, such as maximum amplitude after initial excursion. The mass and mass distribution, that is the car weight and the height of the center of gravity, will have to be specified in the establishment of this performance index.

Ride Quality

The aspect of performance shown in Table 2-4, Ride Quality, does not fit neatly into the proposed scheme. Ride quality should be "good," according to some scale. As usually expressed, ride quality is already a performance index, and the choice here is not between different criteria to be expressed as performance indices, but rather between various ways of quantifying that performance index, some of which are indicated below.

Table 2-4. Ride Quality Criteria

Function of Speed, Track Quality, and Vehicle Suspension
Referred to a Specific Location on Carbody
Identified as Acceleration Response
Expressed Statistically - Mean and Standard Deviation - Exceedances of Probability Distribution
Expressed as a Function of Frequency - Transmissibility - Power Spectral Density

2.4 DEVELOPMENT OF PERFORMANCE SPECIFICATION

After the four performance regimes and their associated criteria and indices were selected, the next step was to develop a methodology to integrate the output from the project's five major tasks into a performance specification. The methodology that has been established is shown in Figure 2-1; the project's major tasks which form the framework for the methodology are:

a. Road testing of several Type I and Type II trucks; then the reduction and analysis of the test data leading to a quantitative definition of performance.

b. Mathematical modeling of freight car trucks to augment and complement test data.

c. Determination of wear of Type I and Type II trucks in unit train service over an extended period of time.

d. Collection of economic data on truck maintenance and operation, and correlation of such data with information on truck performance (see Figure 2-2).

e. Engineering interpretation including effect on performance of eventual wear and deterioration of truck components.

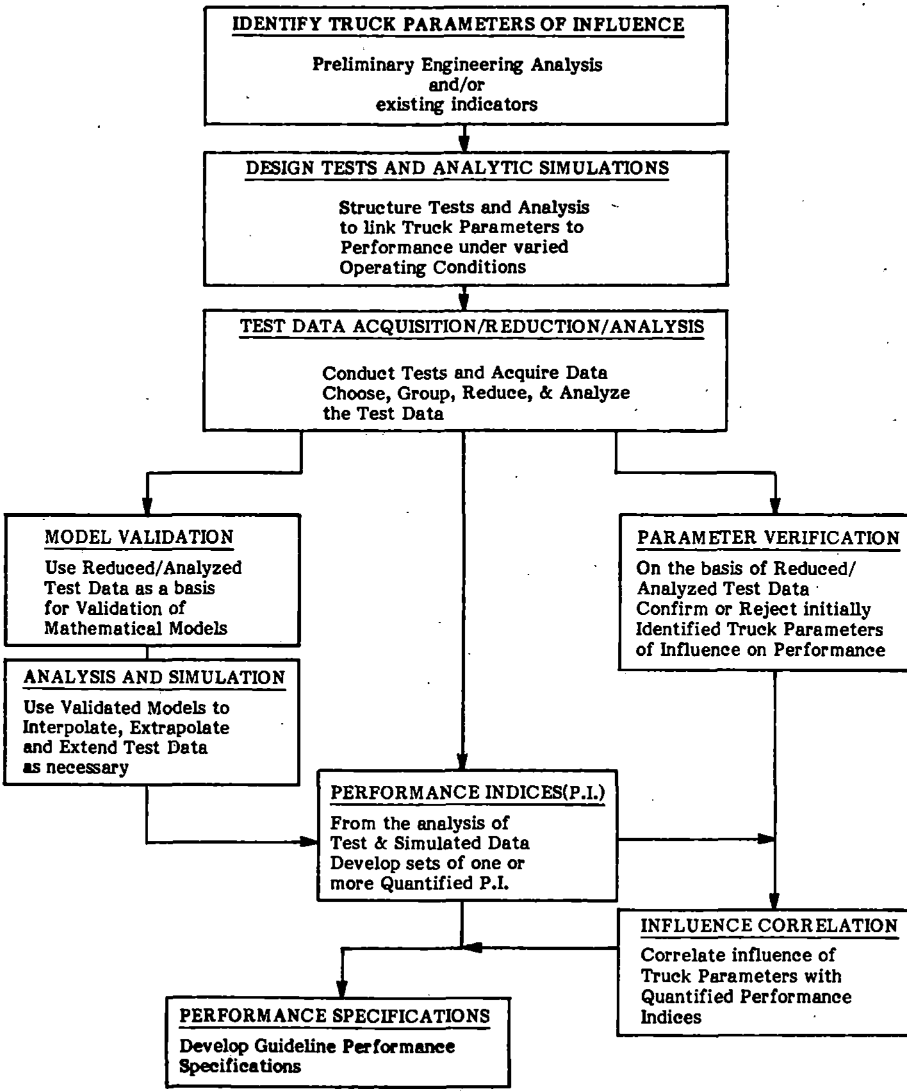


Figure 2-1. Methodology for Truck Evaluation

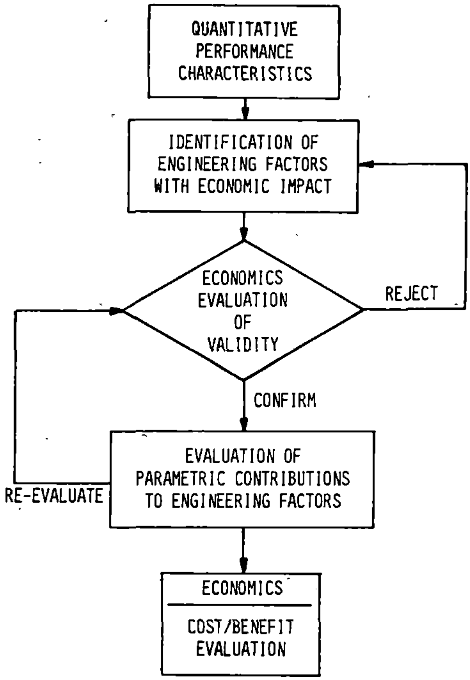


Figure 2-2. Engineering/Economics Interface

2.5 TYPE II TRUCK SELECTION

To develop Type II truck performance specifications, actual performance data will be collected through field tests of several Type II trucks. A systematic selection process based on clearly defined criteria was undertaken to ensure that the trucks selected would be representative of the state-of-the-art in truck design.

In coordination with the TDOP consultants group, who represent railroads, suppliers, and private car lines, the following criteria were established:

- structural adequacy
- suitability for U.S. conditions
- unique design features
- use of standard parts
- design information availability
- initial cost
- service experience
- truck availability/producibility
- availability of performance test data
- unit weight

The candidate trucks were categorized according to their characteristics of their suspension, i.e., primary, secondary, and others. They were further classified according to whether the connection between their side frames and bolsters is of rigid or radial construction. The trucks were also grouped according to whether they have unique load supporting devices.

On the basis of the selection criteria presented above, seven Type II trucks were selected for testing. Three of the trucks (the Devine-Scales, the NRUC Maxiride and the National Swing Motion) have primary suspension; three others (the Dresser DR-1, the Barber-Scheffel Radial, and the ACF Fabricated) have secondary suspension. The seventh truck selected is the Alusuisse truck. A brief descrption of each follows.

Devine-Scales Truck (Figure 2-3)

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. 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.

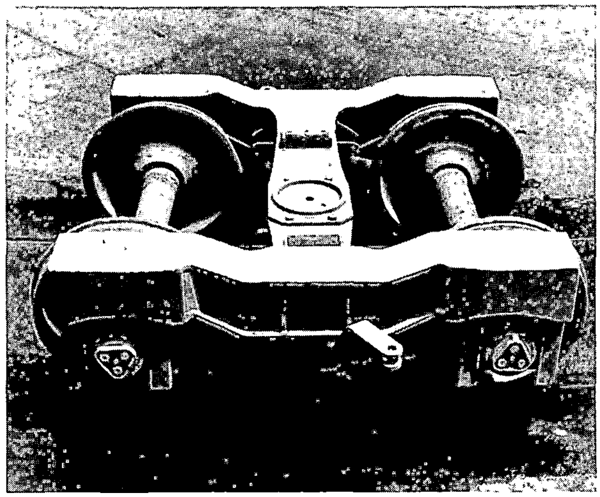


Figure 2-3. Devine-Scales Truck

National Railway Utilization Corporation (NRUC) Maxiride Truck (Figure 2-4)

The National Railway Utilization Corporation Maxiride truck is a one-piece, 100-ton capacity fabricated truck derived from a series of European trucks. 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. NRUC claims the truck improves high speed performance and ride quality, and reduces truck hunting and wheel wear.

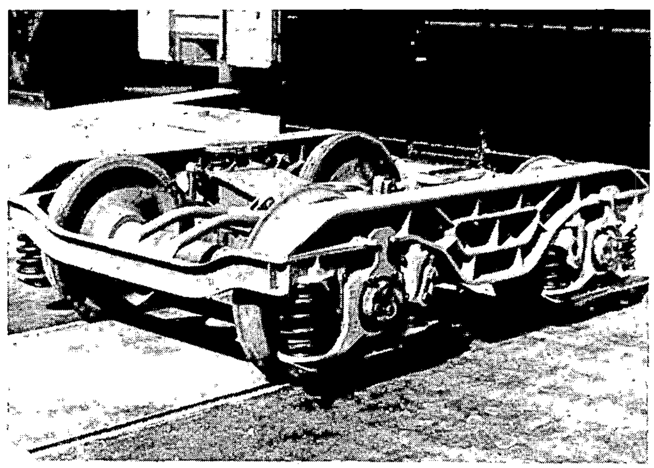


Figure 2-4. NRUC Maxiride Truck

National Swing Motion Truck (Figure 2-5)

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

Figure 2-5. National Swing Motion

Dresser DR-1 Truck (Figure 2-6)

The Dresser DR-1 Steering Asembly is a retrofit package designed to add self steering and curve negotiation control features to conventional trucks. 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 roof of the side frame pedestal with adequate clearance longitudinally to allow the wheelsets to move in seeking a radial position. Dresser maintains that a truck retrofitted with this device will result in improving curving performance, and reduced wheel wear and fuel consumption.

Figure 2-6. Dresser DR-1 Steering Assembly

Barber-Scheffel Radial Truck (Figure 2-7)

The Barber-Scheffel Radial truck consists of cast steel side frame and bolster arranged in a conventional manner. According to the manufacturer, diagonally placed steel cross arms constrain the wheel sets 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. The carbody is supported on a convential AAR center plate.

Figure 2-7. Barber-Scheffel Truck

ACF Fabricated Truck (Figure 2-8)

The ACF Fabricated truck is made up of two side frames and a bolster with a secondary spring group in a somewhat conventional arrangement. However, it has a tie between the side frames and is equiped 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. ACF claims that holding the truck rigidly in tram is designed to materially reduce hunting.

Figure 2-8. ACF Fabricated Truck

Alusuisse Truck (Figure 2-9)

A rather radical departure from conventional European or American practice is the "supple" bogie developed by Swiss Aluminum, Ltd. (Alusuisse). The 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.

Figure 2-9. Alusuisse Truck

SECTION 3 - ECONOMICS

3.0 INTRODUCTION

The TDOP Phase II economics task involves estimating the dollar savings expected with each truck design feature considered. Savings and costs in many departments of the railroad car be estimated. In particular, the major economic impact areas are:

- Car and truck maintenance costs
- Fuel consumption, particularly in curves
- Rail wear, particularly in curves
- Lading damage charges
- Derailment costs

Although other impact areas exist, the effect of a new truck design on those other areas would be relatively small compared to the areas listed above.

The accomplishments of the economics task thus far have been in the areas of maintenance costs and fuel consumption. Total repair costs by component have been tabulated for cars with known annual mileage. The resulting tables allow changes in wear measured in a service test program (e.g., the TDOP wear data collection program) to be related to dollar savings.

The other area where results have been obtained is fuel consumption. Curving resistance for a single car operating normally has successfully been measured as part of the TDOP Phase II test program. The rolling resistance equation used by the industry has been extended to take into account off balance speed and spiral entry behavior in curves. However, more work remains to be done to relate curving resistance to dollar savings.

In the area of rail wear, TDOP has enlisted the aid of the Canadian Institute of Guided Ground Transport (CIGGT) under the Joint Research Project Agreement between the U.S. and Canada. TDOP Phase II will supply measurements of lateral/vertical (L/V) forces and angle of attack for CIGGT's rail wear model. The CIGGT model has been under development for approximately three years and successfully reproduces real world data on rail deterioration.

Two operational considerations have arisen with respect to Type II trucks. First, most of the benefits accrue to the handling line as opposed to the owner. Thus, it is likely that Type II trucks will only be used in special service in the immediate future. The owner pays virtually all the added costs of the truck and must retain some degree of control over the car to benefit from it. Second, the trucks that seek to reduce the car's curving resistance are likely to encounter lading damage problems in hump yards and from yard impacts in general. The braking equations used in control systems of automated hump yards may be unable to handle a significantly improved truck. It is likely that increased lading damage will result from lower rolling resistance.

Progress has been slower in the other economic impact areas; however, detailed plans have been established for each of these areas, as outlined in subsequent paragraphs and in Appendix A, the TDOP Economic Data Collection and Analysis Plan.

3.1 CAR AND TRUCK MAINTENANCE COSTS

There are a number of direct impacts an improved truck (Type II) will have in the area of car maintenance (e.g., a radial truck is expected to reduce flange wear on wheels). In addition, there are indirect costs or savings associated with an improved truck:

- Reduction in lost car days due to reduced frequency of repairs
- Increases in inventory costs due to non-standard parts
- Added repair costs associated with non-standard parts
- Increased standard repair costs due to increased truck complexity

Using service test data and repair data from the AAR's Car Repair Billing System, estimates for the impact of individual truck design features on each of these areas will be produced. The analyses already performed will be discussed first, followed by a description of the way in which the results will be used to estimate each of the impact areas.

3.1.1 Present Value of Car Maintenance

The present value of car maintenance is based on the extra amount a railroad should be willing to pay to eliminate a particular type of repair. For example, suppose as a result of TDOP's wear data collection program, it is found that an improved truck will give 30 percent longer wheel life and 10 percent reduction in other truck repairs for a car averaging 25,000 miles per year. Then the value of the improved performance of the truck can be estimated using the tables developed for these analyses (see Appendix B).

Some of these data are summarized in Table 3-1. In the 25,000 mile per year category of Table 3-1, eliminating wheelset repairs is calculated to be worth $2,006.35 and eliminating other truck repairs is worth $1,281.78. Multiplying the percentage improvements by these values (.30 x $2006.35 + .10 x $1281.78) gives $730.09. This is how much the improvements are worth per car. Since there are two trucks per car, the improvements are worth $365.05 per truck. Adding a 10 percent investment tax credit to this for purchasing a new truck brings the value to $401.56. This is the maximum extra cost a railroad would be economically justified in paying for this improvement.

3.1.1.1 Discount Rate. The present value of repairs at the time of purchase is based on a discount rate, which adjusts for the length of time before any money is saved from improved performance. Inflation is ignored in the calculation; it is assumed that a freight car holds its value, i.e., that as general prices increase so does the price of used freight cars. Instead, the discount rate is being used to represent the "opportunity cost" of the investment in a freight car. For example, capital commands a rate of return; if $10 is put in a bank, it will return approximately 5 percent per year. The incentive to invest the $10 in freight cars instead of putting it in a bank is that the freight cars should yield a higher rate of return.

Table 3-1. Repair Costs by Annual Mileage per Car

	ANNUAL MILEAGE				
	12,500	25,000	37,500	50,000	62,500
Brakes	481.49	668.00	845.33	643.77	513.94
Couplers, Yokes, & Draft Gear	290.52	766.05	1060.09	565.47	646.25
Misc. Labor & Mfg. Material	452.67	1212.69	1913.32	766.12	1248.93
Other Car Repairs	350.53	560.99	898.68	911.13	483.28
Truck Braking System	725.29	1380.53	1913.47	1867.05	1940.11
Wheelsets	953.60	2006.35	2859.78	2517.78	2770.63
Other Truck Repairs	421.87	1281.78	1829.18	268.47	228.80
TOTAL	3675.98	7876.38	11,319.86	7539.78	7831.95
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20
NUMBER OF CARS	21,620	42,972	28,153	5901	1867

To further illustrate why discounting is used in these calculations, suppose complete maintenance records were available for a large number of cars that had just been scrapped. Adding the costs to determine the total maintenance cost would not be the same number as was calculated here. The difference is the discounting of the costs to the time of purchase. One would not pay $10 today for a savings of $10 ten years from now. In these analyses, a 10 percent discount rate is used, which means it would cost $\$10/1.1^{10} = \$3.86$ today to save $10 in ten years.

In the case of complete maintenance records for a large number of cars that had just been scrapped, it is obvious that some adjustment would need to be made for the changes in repair costs for the same repair over the life of the car. A repair that cost $200 in 1950 when the cars were reasonably new probably costs $800 today. To interpret the maintenance records for these scrapped cars, the repairs should be re-priced to today's dollars. If the results were to be used in 1981, they could simply be scaled up the appropriate amount. The results would be interpreted as the total cost of car maintenance at today's prices, discounted at 10 percent over the life of the car. The inflation rate (which is currently impossible to predict over long periods of time) does not really enter the discussion.

3.1.1.2 Classification of Cars by Repair. The tables on which these calculations are based are derived from data made available by the Union Pacific Railroad: repair records for on and off-line repairs to UP's own cars, repair records for heavy and program repairs, annual car mileage, and car age. The process used to produce the tables is shown schematically in Figure 3-1.

First, the annual mileage was entered into the UMLER (Universal Machine Language Equipment Register) for each car. This was done by averaging the annual mileage for UP cars by AAR car type categories (all four digits). This averaging greatly reduces data requirements. Also, if a per car basis had been used, cars with high maintenance would have tended to have artificially low mileage.

Next, a table is constructed showing how many times each particular repair has been made, counting all the cars over the approximately 2½ years for which data have been collected. The number of records with job codes assigned to each repair type is counted, except where more than one part can be reported in one record; in these cases quantity is counted as long as it is in units indicating how many parts are repaired.

As the car repairs are categorized, the total dollars spent and total number of repairs in each category are recorded; however the cost is added into the calculation only if the owner is responsible for the repair. This allows the handling line to be charged for the repairs it is responsible for, and also allows a shorter period of time to be used for calculating average costs. In this analysis, average prices from the first half of 1979 were used, which resulted in all the repairs being priced on that basis (i.e., the first half of 1979). An example of the results of these calculations is shown in Table 3-2. Some combinations of repairs were necessary to keep the data base a manageable size; however, this introduces some errors. For example, both the angle cocks and the air brakes were averaged in the car brake category. Any slight variation in the percentage of angle cocks versus air brakes shows up as a relatively large error because of the differences in price.

Finally, a similar table was constructed of heavy repair data by car. Because UP does not keypunch their heavy repair data or ship it to a central location, Wyle obtained the 1977 UP data from the Omaha shop, keypunched it, and scaled it up to represent the entire railroad. Prices for each category of heavy repairs were not available, so light repair prices were used for heavy repairs. The resulting numbers look reasonable with the exception of 100-ton truck bolsters, which appear too low. Another railroad has provided more complete heavy repair records, which will be compared to the UP data to see if more work needs to be done in this area.

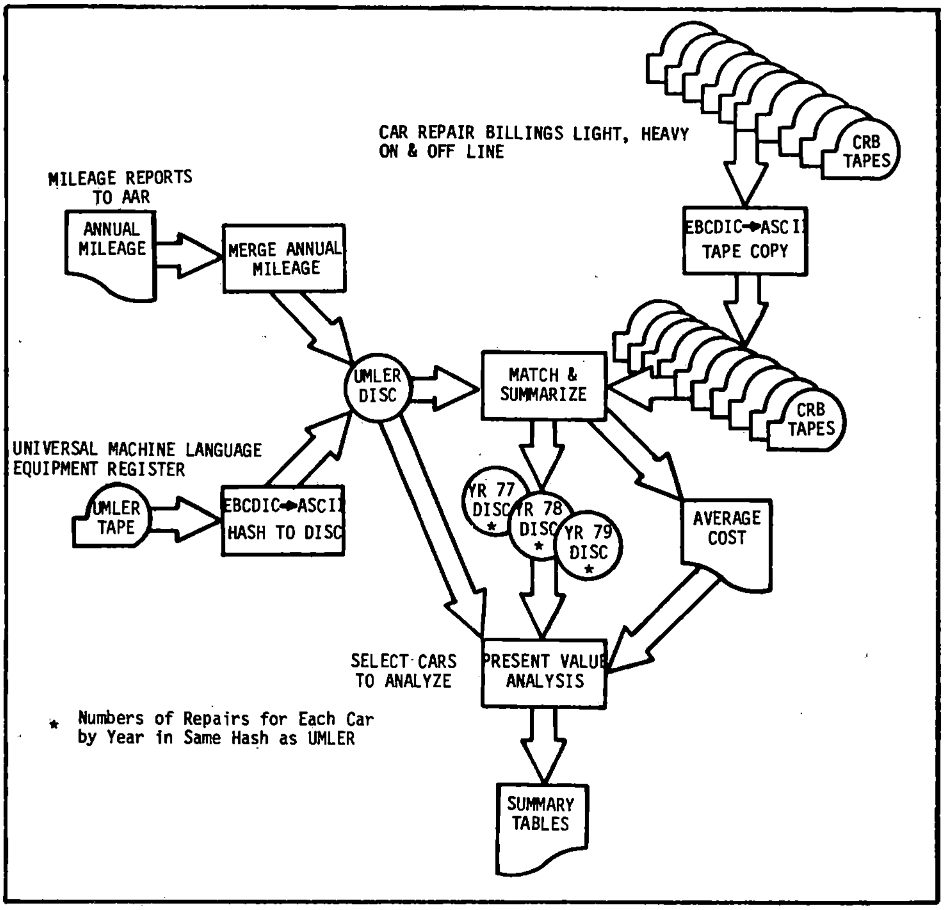


Figure 3-1. Methodology for Calculation of Present Cost of Repairs at Time of Purchase

Table 3-2. Example of Cost Calculations for Repairs

ROW NUMBER	DESCRIPTION	COUNT	JOB CODES	NUMBER OF REPAIRS*	AVERAGE COST
61	Truck Bolsters (Replaced)	R	3500-3554	1846	$ 978.53
62	Truck Bolsters (Repaired)	R	3556	145	285.15
63	Center Pins	R	3560	691	32.00
64	Center Plates	R	3564	32	148.81
65	Center Plate Liners	R	3568	3942	24.41
66	Truck Side Bearings	Q	3572-3580	4722	21.36
67	Friction Casting (Ride Control)	Q	3582	1522	68.14
68	Friction Casting (Stabilized)	Q	3584	6667	35.56
69	Side Bearing Shim	R	3588	303	34.33
70	Sideframes (Replaced)	R	3700-3768	3546	683.00
71	Sideframes (Repaired)	R	3772	168	223.23

R = Records, Q = Quantity

*On-Line Repairs First Half of 1979

3.1.1.3 Processing from Summary Data. Cars can be selected from the UMLER file by annual mileage ranges, by the leading digit of their AAR car type, or by nominal capacity. In Table 3-3, cars have been selected by annual mileage, their ages calculated, and then placed in the appropriate category under the cumulative car mileage heading. Next, each car's mileage for which repair records exist is added to the total of all miles for each cumulative mileage category. For example, if a car were two years old and went 30,000 miles per year, it would be placed in the 50,001 to 75,000 cumulative category; if there were only six months of data for the year used in this particular calculation, 15,000 miles would be added to the total miles for all cars in the category. Finally, the maintenance records for each car are added into the table.

After all the cars have been included, the number in each category is multiplied by the average cost of each repair and then divided by the miles traveled to obtain cents per mile for each repair. Table 3-3 was then extrapolated to 1.2 million car miles. This figure was based on the oldest cars in UP's fleet (which have journal bearings and have not been included). The extrapolation was based on the average of the last ten categories of cumulative car mileage.

The extrapolation was done just once, for "all roller bearing cars." The other tables were based on this first extrapolation by using a ratio of the averages in each category for which data existed.

By assuming an annual mileage and a discount rate of 10 percent, the present value of the repairs at the time of purchase can be calculated. For example, if the car averaged 25,000 miles per year, after one year the cost of wheel repairs would be \$.0091 (from Table 3-3) x 25,000 miles = \$227.50. However, this would be discounted by dividing by 1.1. In the second year the expense would be \$.0045 x 25,000 miles = \$112.50, which would then be discounted by 1.1^2, and so on for each year. These numbers are added to get the one number (in this case, \$1,823.41) in the final tables contained in Appendix B.

3.1.1.4 Limitations and Modifications. Because maintenance records were used for large numbers of active cars of all different ages, it was necessary to assume that cars with similar characteristics experience similar repairs. Then, if there were enough "similar" cars to obtain a reasonable estimate of the cost at each age, the calculation could be done. However, this introduces some complications. For example, we may be using 100-ton hopper cars to estimate the repair costs at 1 to 3 years of age and 50-ton box cars to estimate the repair costs at 20 to 30 years of age.

The hypothetical situation alluded to earlier about complete maintenance records for scrapped cars is not ideal either. If we had maintenance data for cars that are currently being scrapped, they would contain a large number of journal bearing cars and would contain very few modern 100-ton cars. The ideal situation lies between the extremes of obsolete scrapped cars with long histories and brand new car types for which no history exists. The availability of data (three-year histories) in these analyses is closer to the latter case. A great deal more parameterization (e.g., breakdowns by AAR car type) would be helpful. In the next three or four years, it will be possible to obtain complete maintenance histories extending for six to seven years over much of the fleet. Considerably better parameterization will be practical at that time, and much of the uncertainty involved in the analysis presented here will be eliminated. The methodology used is textbook material and will produce better results with more complete data.

Table 3-3. Repair Costs by Cents per Mile

CUMULATIVE CAR MILEAGE	TOTAL MILES OF ALL CARS	CENTS/MILE	
		WHEELSETS	TOTAL OF ALL REPAIRS
1 to 25,000	7,045,877	0.91	2.26
25,001 to 50,000	74,108,288	0.45	0.88
50,001 to 75,000	88,157,632	0.51	1.59
75,001 to 100,000	169,114,320	0.54	1.39
100,001 to 125,000	194,589,984	0.52	1.65
125,001 to 150,000	234,004,656	0.57	2.16
.	.	.	.
.	.	.	.
750,001 to 775,000	19,631,568	0.59	2.14
775,001 to 800,000	10,105,596	0.69	3.16
800,001 to 825,000	3,974,560	0.66	3.08
825,001 to 850,000	2,409,353	0.36	2.62
850,001 to 875,000	0	0.63	2.22
.	.	.	.
.	.	.	.
1,150,001 to 1,175,000	0	0.63	2.22
1,175,001 to 1,200,000	0	0.63	2.22

In labeling this a textbook analysis, there is one modification that has been made to the conventional analysis. Cumulative mileage was felt to be more important in terms of wear on freight cars than age. Thus, instead of constructing tables of repairs versus car age, tables of repairs versus cumulative mileage were constructed. Then, if an annual mileage is assumed, the present cost of car maintenance for any mileage can be read from a single table. This greatly reduces the data requirements. Not as long a time history is needed as would be required if the annual mileage were held constant for all the cars considered in a single table.

To check the procedure, the experiment suggested above (i.e., limiting the cars considered to those between fixed annual mileage bands) was done. The results are illustrated in Figure 3-2. As can be seen from the figure, the average of all the cars was an excellent predictor of the results for cars in fixed annual mileage bands for the middle region of the plot. For extremely high and extremely low mileage cars, the results were dramatically different. Several interpretations of this are possible (it may be an error in methodology arising from any of a number of sources) however, it seems likely that cars in these extreme categories see a different type of service (e.g., unit train service for high mileage cars) and consequently experience dramatically different costs of operation.

3.1.2 Lost Car Days

When a car is sent to a one spot or heavy repair facility for repairs, it is not earning revenue. Thus, there is another potential savings area besides direct maintenance costs associated with an improved truck - the increased utilization of the car. Typically, in other economic analyses of freight car repairs, the cost of lost car days is quite large, even approaching the direct maintenance costs. Thus, it is necessary to consider lost car days in this analysis also.

On the other hand, there is a problem in assigning major savings in lost car days to an improved truck. Usually many repairs are done on a car at one time. When repairs are done on the carbody as well as the trucks, it is very difficult to say what percentage of these repairs would have been deferred if the truck had not had a problem. Further, reducing truck repairs eliminates only part of the lost car days associated with that repair, i.e., a 30% longer wheel life will only eliminate approximately 30% of the lost car days associated with wheelset repairs.

The procedure we plan to take in evaluating lost car days is as follows:

a. Three categories of repairs currently are being recorded to calculate the present value of car repairs: repairs done on the rip track (e.g., brake shoes) with no lost revenue, repairs done at one spots, and repairs done in heavy repair shops.

b. Each of these categories will be assigned an average number of lost car days based on UP data. These lost car days will be costed based on the annual mileage for each car and the per diem and mileage revenues from the UMLER file. The incentive per diem will be ignored because of the complexity it introduces (the date of each car repair would have to be recorded).

c. These numbers will be discounted to present value, as with the direct car maintenance costs. This will yield a new category in the tables, the cost of the lost car days.

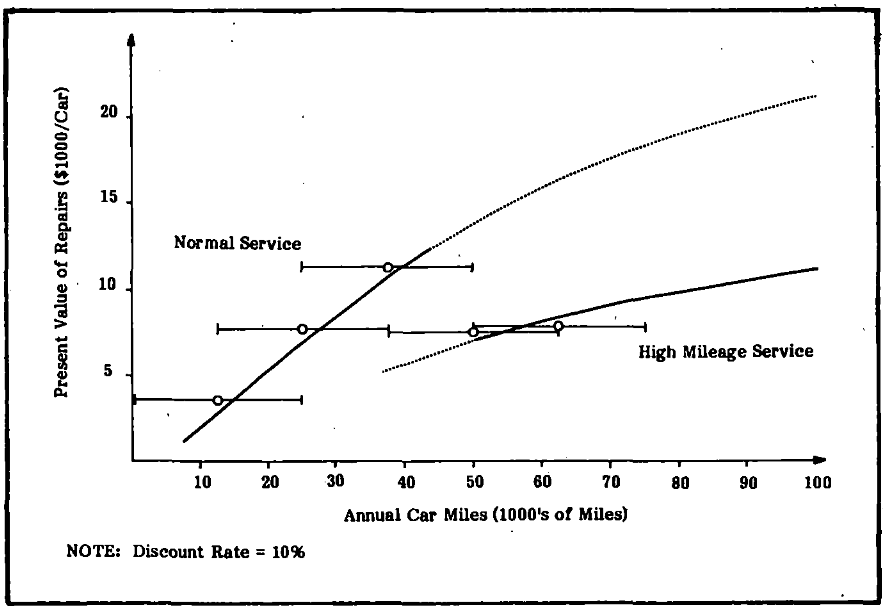


Figure 3-2. Present Cost of All Car Repairs at Time of Purchase

d. Car repairs will be classified into the following categories:

   1. Repairs to carbodies, air brakes, etc. not involving trucks
   2. Repairs to carbodies and trucks
   3. Replacement of brake shoes only
   4. Replacement of wheelsets only (with or without brake shoes)
   5. Mixed replacement of wheelsets and other truck parts (with or without brake shoes)
   6. Replacement of other truck parts only (with or without brake shoes)

   For the purposes of this classification, a single car repair will be defined as all repairs to a single car, taking place at a single standard place location code (SPLC), on a given railroad, on a given day.

e. Finally, based on the TDOP Phase II wear data collection program results, savings in lost car days will be computed. For example, suppose the total discounted value of lost car days is $5,000 for a car of 25,000 miles per year, and suppose categories (d) and (e) above are 20% of all car repairs (excluding category (c) since it was also excluded from the lost car days calculation), and suppose there is a 30% increase in wheel life; then the potential savings in lost car days would be $5,000 x .2 . x .3 = $300.

3.1.3 Inventory Costs

The Type II trucks being studied by TDOP all involve nonstandard parts unique to that truck. The extent to which this is true ranges from trucks accepting standard brake shoes, wheelsets, and springs with all other parts being nonstandard to retrofit kits that add a steering arm to a standard truck. Clearly, it will require extra capital costs to stockpile these nonstandard parts. Even more important, when a nonstandard part is being introduced, one can not rely on its being available on other railroads; thus there are added lost car days associated with a car waiting for parts to arrive from the owner railroad. To some extent this and other problems can be expected to limit the Type II trucks to captive service cars during the introductory of the trucks. (Another problem with free interchange service of cars equipped with Type II trucks is that once the car is interchanged, it will probably never be returned to the owner line. Hence, many of the benefits from a Type II truck would accrue to the handling line instead of the owner.)

Several problems with inventory cost analysis have become apparent. This is the only place in the cost/benefit analysis of Type II trucks that it becomes necessary to address the rate of introduction of Type II trucks. The inventory strategy and costs are highly dependent on whether 3 or 3000 cars are being introduced per year. Further, the parts inventory of other railroads must be considered. There are two issues here: 1) if the benefits of an improved truck are widespread enough, other railroads might reasonably be expected to stock the nonstandard parts, and 2) given the diversity of Type II trucks being offered, the railroad community will likely have difficulty selecting just one new truck.

These sorts of problems have become somewhat muddled in the last several months with regards to wheels. The Canadian railroads have adopted a standard for wheel profiles different from the traditional AAR profile, and will only purchase wheels with this new profile. The implications of this are far from clear at this time. There seems to be agreement that the new Canadian profile will prove beneficial in increasing wheel life through longer flange life. Since thin flange is a leading cause of wheelset replacement (at least a third), it seems likely that major economic benefits will accrue to railroads using the new profile. Under the rules adopted, it appears that the added inventory costs of stockpiling the new profile will also be carried by the railroads that do not adopt the new profile. When an AAR wheel profile is taken off a car, the wheelset will likely be turned to the new profile (the argument is that this will remove the same amount of metal as turning to the AAR profile, or perhaps less, given that the wheelset is worn).

The wheel profile situation has simplified economic analysis of inventory costs, while complicating the rest of the project. In the testing area, we have reexamined which wheel profile should be tested under the Type II trucks. The new CN profile will be used for all Type II testing. In the inventory analysis area, it has the effect of removing wheel profile from consideration as a variable for Type II trucks. The Type II manufacturers who were insisting on yet another new profile no longer have the added inventory cost of stockpiling a nonstandard profile. The issues of Type II trucks and nonstandard wheel profiles have effectively become unrelated.

3.1.4 Changes in Repair Costs

The last economic impact identified to date in the area of car maintenance is changed repair costs. Typically, the charge for each job code (i.e., type of repair) is determined through polling of AAR member railroads. If a Type II truck requires significantly more work than a Type I truck to perform the same repair, this would tend to either raise the overall charge for the repair or force creation of a new job code for repairs of that type of truck. Neither of these adjustments would happen very rapidly. The vast majority of all trucks are Type I trucks, which will likely be the case for some time to come.

An estimate of the relative difficulty required to repair the various Type II trucks is being conducted by the Wyle engineers working in the wear data collection program. Based on their impression, adjustments to some of the truck repair job codes may be suggested and included in the economic analysis. For obvious reasons, any penalty along these lines will be small.

It is possible that a Type II truck will reduce the cost of doing repairs. Indeed, the wear data collection staff feel the Swing Motion truck is the easiest to work on of all the Type I or Type II trucks in the wear data collection program. However, the Swing Motion truck has been used for a long time and is costed at the same levels as the standard truck. It seems doubtful that the costing structure can be made to reflect a reduction in repair changes.

3.2 FUEL CONSUMPTION

The major economic impact of improved truck performance on fuel consumption is expected to come from reduced rolling resistance in curves. A number of trucks are designed to steer through curves by aligning their axles rather than being dragged through on the wheel flange. Any truck which reduces flange wear through modified truck dynamics might reasonably be expected to also reduce rail wear (the other grinding surface) and fuel consumption (the energy consumed in the grinding process). While TDOP Phase II has succeeded in measuring the curving resistance of a single test car operating normally, it is difficult to isolate the effects of curves and requires sophisticated data reduction procedures. The change in fuel consumption that can be achieved through reduction of curving forces is quite small compared to fuel consumed by grade or acceleration. On the other hand, with fuel prices going up the way they have been, even a small change can represent a large number of dollars.

TDOP's investigation of fuel consumption is divided into two parts: the empirical identification of the curving resistance associated with a given truck, and the estimation of the cost or saving associated with the measured change. Measurement of curving resistance is being accomplished through the use of a pair of instrumented couplers, a string potentiometer system to measure the coupler angles, a specially processed DC coupled longitudinal accelerometer, a rotary pulse generator to measure speed, FRA's T-6 track geometry car's measurement of curvature and cross level, and an automatic location detection system used to correlate track position between the track geometry car and subsequent truck dynamic test runs.

The estimation of costs/savings associated with a measured change in the curving resistance will be accomplished through the use of an FRA-developed fuel consumption simulator. A statistical representation of track structures will be developed to represent U.S. railroads. Then statistically representative consists will be run through the simulator using both Type I (standard) and Type II (improved) trucks to estimate the change in fuel consumption caused by the improved truck. Consists made up entirely of cars equipped with Type II trucks, and consists with only one car in the train with Type II trucks will be investigated to see if the savings are the same. In the complete consist case, the feasibility of reducing the number of locomotives will be investigated.

3.2.1 Curving Resistance

Curving resistance is only one of a large number of determinants of coupler forces. Other forces that act on the coupler are the inertial effects of the car, the grade, aerodynamic drag, roller bearing drag, wheel/rail interface forces, and so on. An exhaustive list of the factors that affect rolling resistance is probably not possible (e.g., state of wear of the car, rail surface conditions, track modulus, etc.). It is necessary to restrict one's attention to the largest forces involved. Further, only the component of the coupler force that is in the direction of motion of the car actually contributes to the fuel consumption, since work is force moved through a distance.

The approach taken in TDOP Phase II has been to empirically measure the coupler forces and angles and all the other identified contributions to the coupler force. Then the various effects are sorted to eliminate them from the coupler force until what remains is the coupler force due to curves, the accumulated statistical error in removing the other forces, and any neglected effects. Other procedures are possible, however, this one was felt to fit most easily into the existing test program, to be the most convincing to the industry, and to offer the most hope of providing new information through correlation to TDOP's lateral/vertical (L/V) and angle of attack measurement system. On the other hand, this approach involves customized instrumentation that would be difficult for a railroad to reproduce; also the accuracy required to obtain meaningful requires sophisticated techniques to eliminate errors.

Curving resistance is measured with the following instrumentation.

a. Force is measured using instrumented couplers on both ends of the test car (actually located on the buffer cars). The couplers have a nominal full scale of $\pm$ 25,000 lb and a nominal resolution to $\pm$ 25 lb. Load cells are used to measure the force. The basic coupler design was first used at the Transportation Test Center for FRA-sponsored full scale aerodynamic drag testing and was modified for use on TDOP Phase II.

b. Coupler angles are measured with a set of bending beams set up in a parallelogram structure and attached to the coupler body on the test car and to the test carbody.

c. Longitudinal acceleration and grade (summed) is provided through the use of a DC coupled longitudinal accelerometer. The signal is filtered using a two pole analog, low pass filter, and gain is set up to $\pm$ 0.1 full scale through an auxiliary stage of amplification before digitizing. This accelerometer reads grade as well as acceleration because the DC accelerometer picks up the sine of the vertical acceleration of gravity. The longitudinal accelerometer has proved to be very important in finding curving forces. It should be standard procedure on all future rolling resistance testing involving instrumented couplers. It is inexpensive, easy, and provides the needed boost in accuracy. It also eliminates the need to separate grade from acceleration forces since they are summed. Unfortunately the technique experiences some thermal drift because of the extreme amplification involved.

d. Train speed is provided through a rotary pulse generator (RPG) on the instrumentation car. Two RPGs are also available on the test car. To date the instrumentation car speed has been used for convenience; however, the RPGs on the test car will be used as a future refinement. Wind-up in a flexible shaft attaching the RPGs to the axles has caused problems with the test car RPGs. Optical sensing of the position of a target on the wheel has been added to correct the wind-up problem, however, it complicates the analysis enough that reduction of the data has not been attempted yet.

e. Brake line pressure deviation and notch setting of the throttle are provided as part of the standard instrumentation package on the instrumentation car (UP car 210).

f. FRA's T-6 track geometry car has been run through the test zones at two separate times during testing. In particular, their measurement of curvature and cross level are used to provide curvature and balance speed for the curves. Position on the track was established using the automatic location detection system outlined below, and data from the track geometry runs were merged into the test runs using a linear interpolation routine. Considering the low frequency of the data desired (less than 1 Hz), the merge procedure is sufficiently accurate.

g. TDOP used a Wyle-designed automatic location detection system (ALD) to establish position along the track from run to run. Permanent magnets were installed on the road bed in holes drilled in the ties. A sensor located on the instrumentation car detects the magnets and a channel is recorded indicating when the magnet is passed. This system has performed extremely well. It provides excellent discrimination between ALD targets (the magnets) and the normal background.

More detail on the testing, consist, and test zones is provided in Section 5, Field Test Programs.

The equation under investigation states that the sum of the forces in the longitudinal direction is equal to the mass times the longitudinal acceleration. The following formula is used:

$$CF_f \cos(0.01745\ C_f) - CF_r \cos(0.01745\ C_r) = (2000W + 8W_w)a + 20\,W\theta + 88b + 1.3b^2 + 29 + 1.3W + 0.045\ Wv + 0.0006\ v^2 + .66\ W\phi + .07W\phi^2 + .0003\ W\phi \left| v^2 - V_b^2 \right| + .00003\ Wsv$$

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, $.5\ I_{rr}/r_w^2$ (pounds)
a =	longitudinal car acceleration
θ =	percent grade ratio (percentage)
b =	brake line pressure deficit (psi)
v =	car speed (mph)
ϕ =	absolute value of 100 ft chord curvature (degrees)
V_b =	balance speed (mph)
s =	absolute value of $d\phi/dx$ where x is distance along the spiral (degrees/mile)

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, $(2000\ W + 8W_w)a$, is the mass times acceleration term. The coefficient 2000 converts tons to pounds. The term $8W_w$ is actually a simple way of estimating the rotary inertia of the wheel-set. It represents the resistance of the wheels to having their rate of rotation changed. Theoretically the term should be $4I_{rr}/r_w^2$ 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 $8W_w$ is a close approximation to the real rotary inertia term.

The next term on the second line, $20\ W\theta$, reflects the effect of grade. The coefficient 20 converts tons to pounds and percent grade to grade ratio. The last two terms, $88b + 1.3b^2$, 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 the train out.

The two terms were included to see if this is what was really done. Of course the positive sign on the second coefficient suggests it is. Small applications of air brakes were not attempted during our test program, thus this result is not highly reliable.

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, and resistance at the wheel/rail interface. These terms appear virtually as a constant in most of the TDOP test runs. As a result, the terms are not fitted accurately enough to provide useful information for choosing between the various formulations of the equation.

The last line of the equation is the one in which TDOP is really interested. Several Type II trucks are designed to reduce the rolling resistance in curves. These terms, $.66\ W\phi + .07\ W\phi^2 + .0003\ W\phi\left|v^2 - V_b^2\right| + .00003\ Wsv$, are intended to represent the behavior of the coupler in curves and spirals. The first term, $.66\ W\phi$, is widely known in the industry. The coefficient .66 represents .66 pounds per ton per degree of curvature. It is the flanging force of the wheel on the rail in a curve at equilibrium speed. The next term, $.07\ W\phi^2$ is a reasonable extension of this idea indicating that the flanging force is more than proportional for sharper curves (the sharpest curve considered is about 6.2 degrees of curvature).

The next term is intended to represent the flanging force off balance speed for the curve. At balance speed, the superelevation cancels the centripetal acceleration going around the curve. If the expression for centripetal acceleration is written mv^2/r, this is proportional to $W(V_b^2 + 2V_bV_e + V_e^2)\phi$ where V_e is the excess velocity; but the term $WV_b^2\phi$ is cancelled by the superelevation. The third term on the fourth line, $.0003\ W\phi\left|1v^2 - V_b^2\right|$, represents the off balance speed portion of the centripetal acceleration. At balance speed, $V_e = 0$, so the term is zero. There are absolute value signs around it because the rolling resistance increases irrespective of whether it pushes on either the high or low rail.

The last term, .00003Wsv, represents the effects of spiral curve entry and exit. A term of this type (i.e., proportional to $d\phi/dt$) was first suggested by Leonard McLean of the Seaboard Coast Line Railroad (one of the TDOP consultants). Its interpretation is complicated by the fact that spirals tend to be set up so that $d\phi/dx$ is a constant. If the term represents the rotational energy of the trucks and carbody going through the spiral, it should be scaled by some proportion of the rotary inertia of the trucks and carbody. However, if the term represents the energy dissipated by rotation of the centerbowl, it should only be effective when the car is in a spiral and there is rotation of the centerbowl. When it is effective, it should only be proportional to the car weight. It is very difficult empirically to tell these two effects apart, so they have been lumped together in one term. With spirals, it is necessary to hypothesize such a term to get the correct sign on the $W\phi^2$ term. Otherwise the relatively large forces in the spiral at very low degrees of curvature tend to give a negative coefficient for $W\phi^2$.

3.2.1.1 Statistical Procedure. This is a rather difficult equation to fit. In the test zone used for TDOP, the second line of the equation (i.e., inertial effects, grade, and braking) completely dominates all the other terms. Any error in the bias or gain of the accelerometer used obscures the other terms. As a result, it was necessary to use a statistical procedure known as a staged regression, which is designed to accommodate this situation. First, the acceleration and grade terms are fitted and removed from the measured rolling resistance. Then the remaining terms are fitted from the residual (the part left over) from the first stage of the regression. Details of the procedure are summarized below.

a. Construct a standardized milepost channel for each test run based on the ALD. Merge track geometry curvature and cross level into the data for each run using the standardized milepost.

b. Estimate grade for each run in the test zone:

   1. differentiate car speed and filter it
   2. subtract the filtered car speed from the longitudinal accelerometer as an estimate of grade
   3. integrate the result times the car speed to estimate elevation change
   4. estimate a constant bias to correct the elevation change to agree with track charts
   5. add this bias back into the estimated grade

c. Average the available estimates of grade in each test zone. Merge the average back into the data for each run using the standardized milepost.

d. Avoiding the parts of the data where air brakes were applied, form the following for each test run (low pass filter down to .25 Hz taking a sample every two seconds):

   1. form the difference in the coupler forces
   2. further filter the longitudinal accelerometer (includes grade and acceleration terms)
   3. form the best available estimate of all other terms
   4. subtract step three from step one
   5. estimate a constant and gain relating steps four and two using least squares
   6. subtract this estimate from step one to form the residual

e. Using least squares, estimate the rest of the equation based on the residuals of all the test runs. This gives a new best available estimate of all other terms. Iterate at step d until the results converge.

f. Go back and find the runs where air brakes were applied. Subtract everything else off except the brake line pressure terms. Using least squares, estimate the coefficients of the brake line pressure terms.

Since any empirical study involves errors, the data are adjusted in the way described above to take advantage of those parts of the equation which are known more accurately than can be measured. In this case, the F = ma terms associated with grade and car acceleration are exactly as stated by the equation. However, this cannot be verified because of small errors in the gain and bias of the accelerometer, and the two coupler forces, and errors in measuring the weight of the test car. Rather than allowing these errors to obscure the curving forces, a procedure is devised to correct the errors. First, remove all the other terms as well as they are known. Estimate the remaining F = ma terms and attribute all the gain error to the accelerometer (typically a few percent). Integrate the accelerometer (with the adjusted gain) to find how much of the bias can be attributed to the accelerometer (typically 0.02g). Finally, the remaining bias is attributed to the difference in coupler force (typically 500 pounds). Removing the adjusted data produces the curving forces.

The results have been very encouraging. Figure 3-3 illustrates the measured difference in coupler force (the dotted line) compared to the predicted difference in coupler force from the equation described earlier (the solid line) for the near equilibrium speed run through the curving test zone. Figure 3-4 illustrates the estimated grade for this test zone (the dotted line) compared to the track chart grade (the solid line), including the grade compensation needed for each curve. Although not indicated on the track chart, the curves are grade-compensated, according to UP.

Figure 3-5 illustrates the data remaining after the acceleration and grade terms have been adjusted and removed from the test data. Again the dotted line is the test data and the solid line is the prediction from the equation. While it is obvious that every curve is not predicted precisely, the data rather closely resemble the theoretical numbers. The differences that remain are being investigated. A possible explanation is road grading in the area, which may be contaminating the rail surface on one of the test curves. Also, there is some suggestion in the literature that curve memory may exist.

The analysis to date agree rather well with the results widely known in the railroad industry for the Type I truck; however, the real issue is how this changes with a Type II truck. Tests thus far have shown that the methodology works for a case in which the answers are known. Additionally, the expressions for curving forces to express off balance speed and spiral entry behavior have been refined.

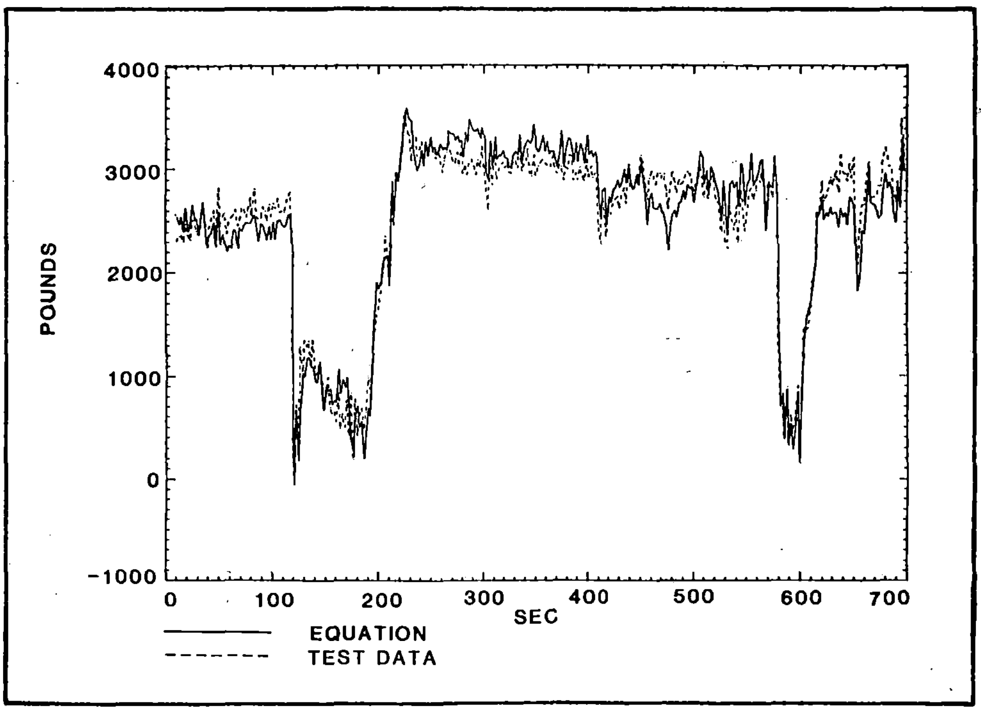


Figure 3-3. Filtered Rolling Resistance

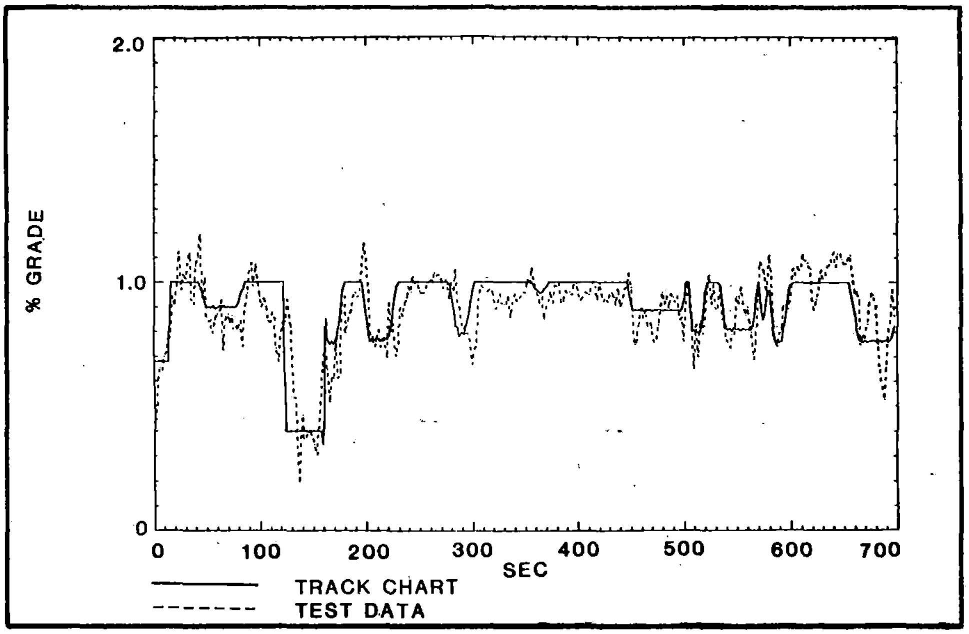


Figure 3-4. Estimated Grade

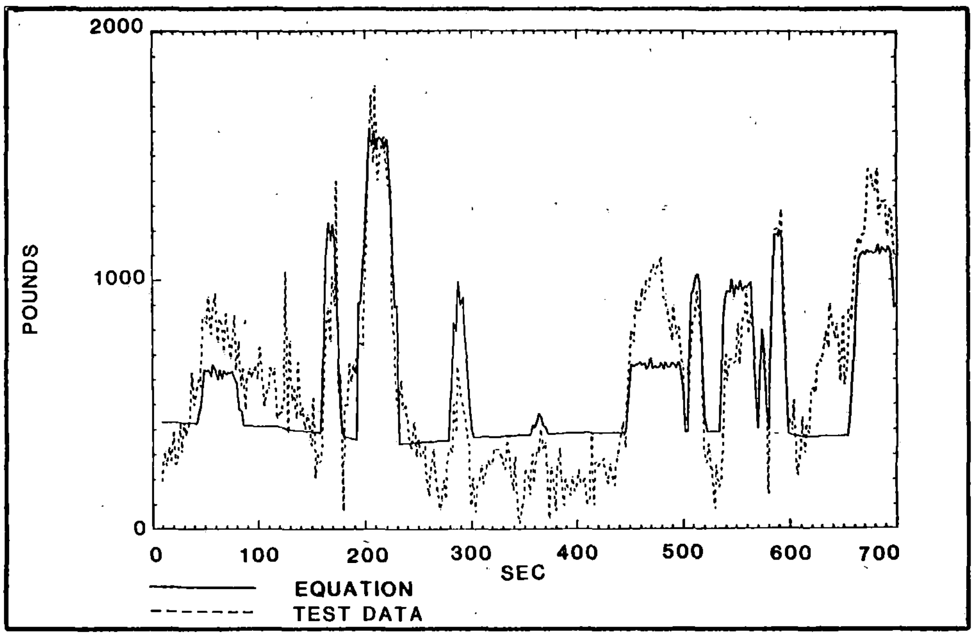


Figure 3-5. Davis Equation Plus Curves

3.2.2 Statistical Track

In order to estimate the cost/savings associated with a change in curving resistance, it is necessary to know how much curved track there is on the railroad being investigated. This leads to the concept of a statistical model for track structures. The variables that must be considered for use with a fuel consumption simulator are posted speed, grade, and curvature. Other variables influence fuel consumption (e.g., track modulus); however, the fuel consumption simulator is not sophisticated enough to account for them.

The basic plan is to categorize UP's track into categories (data groupings are coded: speed range/curvature range/grade range).

1. 5-25 mph/0-2^{o}/0-0.2%
2. 30-40 mph/0-2^{o}/0-0.2%
3. 45-50 mph/0-2^{o}/0-0.2%

⋮

28. 30-79 mph/2-4^{o}/>1.0%
29. 5-25 mph/0-2^{o}/>1.2%
30. 30-79 mph/0-2^{o}/>1.2%

The actual number of categories and the choice of categories will be based on the data, attempting to minimize the number of categories without leaving too large a range of variables and still keeping the categories all about the same size. Using an existing computerized track chart for UP, the gross ton miles/year (i.e., annual gross tons times miles) will be counted in each category. Adding all the categories and dividing each category by the sum gives the probability of observing data in each category.

Next, statistics will be run on each category. The mean and standard deviation of each of the three variables (speed, curvature, and grade) in each category will be determined, as well as the mean and standard deviation of the miles of track for each entry into each category (in other words, the length of time the track stays in each category without changing parameters will be determined). The probability of a sign reversal on either grade or curvature (e.g., going from uphill to downhill) will be calculated. Finally, the conditional probability of switching from one category to another will be estimated.

This information will be used to generate track data for the fuel consumption simulator. Statistically representative data will be generated instead of using actual routes for comparison of trucks, since an actual route may not be representative. By adjusting some of the parameters for which data are available on a national basis, the results can be made representative of national averages. By using UP data from mountainous regions, a model of mountainous terrain can be built, and so on.

3.2.3 Fuel Consumption Simulator

Having estimated the change in rolling resistance of a car and established the track to be considered, it is still necessary to estimate the effect on fuel consumption. This will be done in TDOP through the use of a fuel consumption simulator developed by the MITRE Corporation under contract to the FRA (references 1 and 2). The program basically reproduces measured fuel consumption to approximately five percent from test data. The program has been installed at Wyle and results from the Wyle version have successfully been compared with those from the MITRE version. Some modifications to the program will be made at Wyle; however, the basic operation of the program will not be changed.

The program will be used to estimate fuel consumption for statistically representative consists over statistically representative track. A baseline will be established through the use of the car rolling resistance equation introduced earlier. Next a single car will be changed to have the rolling resistance measured for one of the Type II trucks. The baseline data will be rerun precisely as it was before except for this change. Any change in the fuel consumption will be attributed to the change in rolling resistance. Next the entire consist (except for locomotives and caboose) will be modified to reflect the Type II truck. Again the baseline data will be rerun and the change attributed to the Type II truck. Finally, the number of locomotives in the consist will be reduced to see if performance can be maintained with less motive power. One would expect the answer to depend on whether the curves are assumed to be grade-compensated. Both cases will need to be investigated.

Rather than simply keeping track of the overall fuel consumption, the fuel consumed and the miles traveled in each track category will be tabulated. In this way the savings can be associated directly to the track category, allowing estimation of savings for variations in the weighting of the track categories without having to do the entire simulation again. The savings in the form of gallons saved per car mile by track category will be tabulated as the final result of the fuel consumption study.

3.3 RAIL WEAR IN CURVES

The third area of savings from a steering truck is expected to come from reduced rail wear in curves. The high rail on sharp curves wears out much faster than similar rail on tangent track. As a result, there are schemes like rail lubrication, exchanging the high and low rail, special metallurgy, and so on. Improving the truck may offer an attractive alternative to these procedures. If the truck is improved, it reduces wear on every curve that truck encounters. This is in sharp contrast to the rail-oriented schemes which only improve one curve at a time. By improving heavy, high mileage cars, it may be possible to realize significant savings in rail wear in curves.

Studying rail deterioration is difficult. The mechanisms of rail failure are complicated, and rail technology is highly specialized. Therefore, TDOP has been fortunate to enlist the aid of the Canadian Institute of Guided Ground Transport (CIGGT) through the support of Transport Canada. CIGGT (associated with Queen's University in Kingston, Ontario) has been a pioneer in the area of rail deterioration simulation.

In CIGGT's approach to rail deterioration modeling, failure mechanisms are postulated and the engineering aspects of each mechanism are modeled. The models are substantiated and calibrated through correlation to observed, real world track deterioration. Finally, the economic consequences of what is being modeled are predicted. This type of analysis is needed for TDOP to relate engineering parameters to economic savings. Given vertical and lateral forces and angle of attack, CIGGT will predict wear in curves among a variety of other mechanisms of track failure.

TDOP will supply CIGGT with the results of the engineering analysis of each truck in the form of L/V and angle of attack measurements for various degrees of curvature. CIGGT will use their model along with the same economic assumptions TDOP is using to produce savings per car mile for each truck studied. The results of both studies will be integrated into one cost/benefit analysis of Type II trucks which will be issued jointly.

3.3.1 Statistical Track

The track model for the rail wear study will be very similar to the one already outlined for the fuel consumption study. Several additional variables need to be considered, however. The most significant variable for the rail study is annual gross tons associated with each category (e.g., 40-50 mph/0-0.2°/0-0.4%) considered. Tonnage plays the same role in a track study that annual mileage plays in a car study - it determines the time span over which benefits are realized. As the tonnage increases, repairs are needed sooner, thus there is more savings to be realized from an improvement. Tonnage is available for the data base TDOP plans to use to generate the statistical track, thus this presents no problem.

Several other variables are more of a problem; CWR versus jointed rail, rail weight, lubrication, and surface treatment are all modeled in the CIGGT simulation. It is difficult to choose a particular set of these parameters to consider. Statistics will be developed for these variables (to the extent they are available) to assist in the choice of assumptions. For example, one would expect the very low speed, high curvature categories to be dominated by spurs and branch lines with lower rail weight and jointed rail.

Two of the variables from the fuel consumption study (speed and grade) are probably less important for rail wear. Particularly, posted speed plays only a secondary importance in the analysis of rail wear in curves. Thus, these variables will probably be treated more coarsely in the rail study. For example, if the fuel consumption study uses two categories that are the same except that in one the posted speed is 40-50 mph, and in the other it is 50-60 mph, these would be combined into one 40-60 mph category for the rail study.

3.4 LADING DAMAGE

Reduced lading damage is another area where major 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 loaded, or a spring nest over each roller bearing, or in one case, leaf springs as the primary means of suspension). These modifications are intended to provide less vertical excitation and thus, a better ride.

With two exceptions, the trucks that attempt to reduce rolling resistance are not the ones that attempt to provide a better ride. Reduced rolling resistance and reduced lading damage tend to be mutually exclusive because of hump yards. A car with dramatically reduced rolling resistance is going to hump very hard. One would expect an increase in lading damage from a car with reduced rolling resistance. While this problem may be resolvable (e.g., one could paint DO NOT HUMP on the side of the car), it does not seem reasonable that a very large percentage of the fleet could have this requirement without dramatically affecting railroad operations, or more likely, the request would simply be ignored. In any event, it is unlikely that reduced curving resistance and reduced lading damage could be realized by the same truck under present operating conditions.

Estimating savings from reduced lading damage due to vertical excitation is virtually impossible in any general way. The problem has to do with separating the effects of longitudinal excitation (especially in hump yards) from the effects of vertical excitation. Both effects tend to show up in the lading damage statistics compiled each year by the AAR. While this category accounts for approximately half of all lading damage, it is impossible to say how much of the damage is from vertical excitation and is thus possible to eliminate by an improved truck. Research in this area has been done to justify improved couplers. The results suggest that at least 50 percent of the lading damage is longitudinally related.

On the other hand, specific cases as outlined by the TDOP consultants' group often involve large potential savings. If an improved truck could carry the right commodity, the savings might be substantial. Of almost equal importance, an improved truck with better ride characteristics might be able to draw new commodities to the railroads. Both of these arguments make it important that TDOP not dismiss savings in this area as inestimable.

The planned approach is to demonstrate that large savings may exist in the lading damage area. Rather than attempt to estimate the percentage of lading damage due to vertical excitation, all lading damage costs for individual commodities will be calculated. Deciding whether these costs can be meaningfully reduced through an improved truck will be left to the individual railroad.

The cost of lading damage to specific commodities can be estimated in a format that is consistent with the rest of the analyses being done for TDOP. The key to making this estimate is to reduce the cost of lading damage to a cents per mile figure. This can be done by correlating the AAR lading damage reports to the 1% waybill sample by commodity code. After making an assumption about car miles (i.e., 50% loaded, 50% empty) a ratio of the numbers in the two reports can be made to estimate the cost per mile of lading damage. Results of these calculations are illustrated in Table 3-4. These results will be improved for the final cost/benefit analysis in that the same year's data will be used for both the 1% waybill sample and the lading damage reports.

Table 3-4. Lading Damage per Mile by Commodity Type

		PERCENT ALL CLAIMS	1% WAYBILL 1000 MI	CAT 3 1000 $	CAT 4 1000 $	CAT 9 1000 $	CAT 3 CENTS/MI	CAT 4 CENTS/MI	CAT 9 CENTS/MI
01	FARM PRODUCTS	18.7	6709.	5453.	3202.	10374.	0.31	0.16	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	NOMETALLIC 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.	4669.	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

3.5 DERAILMENTS

To the extent that an improved truck reduces the L/V ratio, it would tend to derail less often. Thus, it might seem reasonable that an improved truck would experience lower derailment costs. Unfortunately, this probably is 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. Most cars are victims, not perpetrators. 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, an improved truck will tend to reduce the overall incidence of derailments. Even this effect will be fairly distant. One might typify derailments 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.

REFERENCES

1. Muhlenberg, J.D., "Resistance of a Freight Train to Forward Motion," MITRE Technical Report 7664, prepared under DOT/FRA contract DOT-FR-54090, November, 1977.

2. Muhlenberg, J.D., "Freight Train Fuel Consumption," MITRE Technical Report, MTR-79W00010, prepared under DOT/FRA contract DOT-FR-54090, January, 1979.

SECTION 4 - ANALYSIS

The specific objectives of the analysis task are: (1) to define the requirements for test data and simulation computer models; (2) to determine the extent to which field test data can be extrapolated; (3) to assess the most promising of the computer models; (4) to develop criteria for validating the models, and (5) to validate the models.

The Analysis Plan (FRA Report No. FRA/ORD-80/31) addressed the first two objectives, while the Analytical Tool Assessment Report (FRA Report No. FRA/ORD-79/36) dealt in detail with the third; these two reports are summarized below in paragraphs 4.1 and 4.2. The last two analysis task objectives (developing model criteria and validating the models) are covered in paragraphs 4.3 and 4.4.

4.1 ANALYSIS PLAN

An analysis plan was developed for each of the four performance regimes of lateral stability, trackability, curve negotiation, and ride quality. Each plan contained a brief review of the performance indices associated with the regimes, the analysis requirements, and the model and test data to be used in performing the analysis.

4.1.1 Lateral Stability Regime

The lateral stability regime treats the tendency of the truck to exhibit self-excited lateral and yaw oscillations, commonly called hunting. Hunting is generally observed when operating at high speeds on tangent track. The tendency of a truck to hunt is strongly influenced by environmental factors such as rail contamination and track geometry.

The analysis plan recommended that field test data and simulation models be used to investigate the influence of these environmental factors as well as operational considerations on truck hunting. It suggested that linear frequency domain modeling techniques be used wherever 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.

A number of design features of the various Type II trucks are expected to have an influence on lateral stability. The plan suggested that particular attention be paid in the analysis to the effect of wheelset interconnection, recommended wheel profiles, and tramming stiffness. Other features such as lateral bolster freedom and primary versus secondary suspensions could also affect lateral stability.

4.1.2 Trackability Regime

The trackability regime is divided into the subregimes of: load equalization, harmonic roll and bounce, and curve entry/exit.

Load Equalization. Load equalization gives the truck the ability to maintain equal wheel loads for cross level variations occurring within the wheelbase. It is primarily a static or quasi-static (very low speed) subregime. The analysis plan concluded that most data for analyzing load equalization shall 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 at the 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 Type II truck features which are expected to affect trackability with respect to track twist are the frame rigidity, primary suspension, and the effect of snubbing devices.

Harmonic Roll and Bounce. 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. Requirements for the harmonic roll and bounce analysis include truck and carbody response prediction and the effect on performance of parameter variations (e.g., spring characteristics), and operating conditions such as speed and track condition. Actual track geometry data acquired during TDOP Phase I and II, including field testing of the Friction Snubber Force Measurement System, are expected to provide inputs to the modeling effort representative of service conditions.

A number of Type II trucks have incorporated features which are intended to improve performance in the area of harmonic roll and bounce. In particular, some Type II trucks are designed with soft lateral bolster movement to reduce harmonic roll. Type II trucks with primary suspensions which reduce the unsprung mass, should reduce shocks due to bounce-type track input.

Curve Entry/Exit. This subregime is concerned with the transient response of the vehicle as it traverses the spirals between tangent and curved tracks while entering into and exiting from constant curvature tracks. The dynamics during the negotiation of these spirals tend to set up high lateral forces at the wheel/rail interface as well as an uneven distribution of vertical loads among the wheel/rail contact points. Thus, it is quite likely that larger L/V ratios may be encountered during curve entry/exit as compared with steady state curve negotiation (see below). Consequently, derailment potential, rather than wheel and rail wear, is of primary concern within this subregime. The performance indices identified in this subregime are: (1) L/V ratio and the duration associated with it; (2) peak lateral force on the wheel; and (3) wheel unloading index.

4.1.3 Curve Negotiation

This regime considers the steady state curve negotiation phenomenon. The analysis plan recommended that two types of analytical models be used in predicting the curving performance index 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.

Curve negotiation will be affected by the unconventional characteristics of various Type II trucks. A number of Type II trucks feature self-steering mechanisms which are intended to reduce flanging in curves. Comparison of primary versus secondary suspension performance will also be examined.

4.1.4 Ride Quality

The ride quality performance index of transmissibility will largely be determined by a truck's suspension characteristics. Therefore, the plan concluded that the analysis will focus on the differences exhibited between primary and secondary suspension trucks. The plan recommended that simple linear models will be used to perform these studies.

4.2 ASSESSMENT OF ANALYTICAL TOOLS

In order to accomplish a performance evaluation of Type II trucks versus the standard, three-piece Type I truck, TDOP Phase II will conduct an extensive series of field tests on Type I and Type II trucks. TDOP Phase II will also make use of test data acquired during TDOP Phase I. Analytical tools will then be applied to extend and interpret the results of these field test programs.

The term "analytical tool" refers to any analytical method employed to predict and understand the car/truck dynamic behavior. The set of analytical tools includes models which are considered here to be the set of equations describing the car/truck dynamics and the computer program implementing these equations. The analytical tools of most interest to TDOP Phase II are those 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 4-1.

Table 4-1. Summary of Assessment Criteria

1.	Is the analytical tool applicable to one or more of the TDOP II performance regimes?
2.	Is the tool useful in studying truck performance in terms of the performance indices?
3.	Is the tool capable of performing or supporting analyses that meet TDOP II objectives?
4.	Is the tool compatible with the digital computers available to the Contractor?
5.	Is the tool capable of analyzing required truck/carbody configurations with minor modifications?
6.	Is the tool available in terms of the TDOP II schedule?
7.	What is the validation status of the tool?
8.	What is the accuracy of the tool?
9.	What is the precision of the tool?
10.	Can the tool be verified?
11.	Is the utility of the tool acceptable?
12.	Does the tool complement the other tools properly?

After completing a preliminary survey of 59 tools, 14 of the most promising analytical tools (as listed in Table 4-2) were selected for detailed assessment and validation. This assessment is summarized below.

4.2.1 DYNALIST II (Transportation Systems Center)

DYNALIST is a general-purpose, dynamic analysis simulation program developed by the Transportation Systems Center (TSC). Models with up to 50 degrees of freedom can be analyzed in both the time and frequency domain. The DYNALIST package was specifically developed for rail dynamics, hence the basic elements which can be used to define a dynamic system consist of wheelsets, truck components, carbody lading, springs, dampers, etc.

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. It is expected that DYNALIST will be applied to the lateral stability, trackability, and ride quality regimes of TDOP Phase II. Due to its versatility, it is expected to be useful and in predicting the effects of minor changes in model configuration or comparisons of Type I and Type II trucks. The model's limitation is that it is restricted to linear analysis.

4.2.2 HUNTCT (Wyle Laboratories)

HUNTCT is a nonlinear, time domain model. The model has 21 degrees of freedom (dof) in the basic rigid body representation. The effects of carbody flexibility may be accounted for by including an additional degree of freedom for each principal mode of carbody flexure (e.g., bending).

HUNTCT was selected for validation on the basis of its capability to perform detailed analysis of lateral stability phenomena, including the representation of significant nonlinearities in the truck model and the wheel/rail interface. Also, HUNTCT has the advantage of having already been given some analytical and experimental scrutiny. It can be easily modified to incorporate specific aspects of a Type II truck. HUNTCT will be used in the lateral stability régime; however it could also be used if required in curve entry and exit analysis to supplement curve negotiation models.

4.2.3 FRATE (MITRE, Wyle Laboratories)

The FRATE models (FRATE/MITRE, FRATE 11, and FRATE 17) are nonlinear, time domain models. The most basic of the three is FRATE 11, developed by Wyle Laboratories. It is an 11 dof representation of the rigid body dynamics of wheelsets, trucks, and carbody with provision for additional degrees of freedom to represent carbody flexibility in terms of its normal modes. FRATE/MITRE is an extension of FRATE 11 in which additional lumped elements are added for lading such as a trailer on a flatcar. FRATE 17 includes extra degrees of freedom in the truck. The FRATE models have been partially validated against test data both by Wyle Laboratories and MITRE.

Table 4-2. Candidates for Validation

Model	Degrees of Freedom	TDOP Areas of Application	Linear/Non-Linear	Frequency/Time Domain Steady State Equilibrium	Carbody Model
DYNALIST II	up to 50	Any (depending on particular model definition)	Linear	Frequency and/or Time	Rigid or Flexible
HUNCT	21	Lateral Stability, Curve Negotiation	Nonlinear	Time	Rigid or Flexible
FRATE	27	Harmonic Roll and Bounce, Ride Quality	Nonlinear	Time	Rigid or Flexible allows for lumped masses for lading
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
Freight Car Hunting	25	Lateral Stability (critical speed, stability margins)	Linear	Frequency	Rigid
17 dof Eigenvalve	17	Lateral Stability	Linear	Frequency	Rigid
9 dof Steady State Curving	9	Curve Negotiation	Nonlinear	Stady State Equilibrium	Rigid
17 dof Steady State Curving	17	Curve Negotiation	Nonlinear	Steady State Equilibrium	Rigid
Flexible Carbody Vehicle	20	Harmonic Roll	Nonlinear	Time	Two Lumped
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

FRATE 11 has been selected as the primary tool of the three to be validated. The selection is based on past results showing that FRATE 11 and FRATE 17 produce very similar results regarding carbody motion. In addition, FRATE 11 can be run on Wyle's Interdata computer as opposed to FRATE/MITRE which, due to its large core requirements, must be run on an outside computer. FRATE 11 will be applied to the trackability (harmonic roll and bounce subset) and ride quality regimes, providing a detailed analysis capability including nonlinear effects.

4.2.4 Freight Car Hunting (Association of American Railroads)

The Freight Car Hunting Model is a linearized representation developed by the Association of American Railroads (AAR) for studying lateral stability. The model uses 25 degrees of freedom. Matrix methods are used to obtain natural frequencies and mode shapes.

The Freight Car Hunting Model was selected for validation on the basis of the insight it can provide in investigating lateral stability. The program is sufficiently well documented so that it can be used readily. Another advantage of the Freight Car Hunting Model is the efficiency of the frequency domain analysis which it uses. No previous validation work is cited by AAR.

It is expected that the Freight Car Hunting Model will be used to obtain qualitative rather than quantitative results, such as identifying trends and establishing relationships in its application to the lateral stability regime.

4.2.5 17 dof Eigenvalue (Law and Cooperrider)

The Law and Cooperrider 17 dof Eigenvalue Model is a linear representation developed for analyzing hunting behavior. 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 17 dof Eigenvalue Model was selected for validation as a complementary program to the AAR Freight Car Hunting Model, which is also a linear frequency domain model. In particular, the Law and Cooperrider program is well suited for addressing investigations of Type II truck behavior and the effects of vehicle front/rear asymmetry. It is expected that the results obtained with the 17 dof Eigenvalue Model will, like the AAR hunting model, identify trends and establish relationships which will then be examined in greater detail by nonlinear models.

4.2.6 9 & 17 dof Steady State Curving (Law and Cooperrider)

These two models are nonlinear representations of a freight car in steady curving. The program iterates until convergence to an equilibrium solution is achieved. An extra eight degrees of freedom in the 17 dof Steady State Curving Model are used to account for lateral and yaw motions of individual wheelsets. These two models were selected for their ability to relate truck and wheelset parameters to curving behavior. Both models make use of the Law and Cooperrider Wheel/Rail Constraint Routines which have been validated experimentally.

The 9 and 17 dof curving models are expected to be the primary analysis tools applied to steady state curve negotiation. One of the results of the validation exercise will be to establish the range of application of the 9 and 17 dof versions. It is expected that these models will be capable of directly relating parameters to the performance indices.

4.2.7 Flexible Carbody Vehicle Model (AAR)

The Flexible Carbody Vehicle Model is a nonlinear, time domain program which represents freight car vertical and roll dynamics. The model uses 20 degrees of freedom. The model was selected to complement the FRATE models and other linear models in the analysis of the trackability (harmonic roll and bounce) and ride quality areas. The model has been analytically verified previously and is well documented. The Flexible Carbody Vehicle Model is expected to produce detailed results relating carbody roll, bounce, and twist motions with trackability and ride quality performance indices.

4.2.8 HALF, FULL, FLEX, AND LATERAL (TSC)

These four models are intended to be complementary frequency domain models. In relation to other models selected for TDOP validation, these would be classified as simple, linear analytical tools. HALF, FULL, and FLEX deal strictly with vertical motions. LATERAL computes transmissibilities for the carbody suspension with respect to sinusoidal track alignment variations. These models were selected to provide an efficient means of obtaining qualitative results in the ride quality area.

Results from these simple, linear models will be scrutinized with respect to results from more sophisticated nonlinear models such as FRATE and the AAR Flexible Carbody Vehicle Model. Validation of HALF will receive a low priority because the representation of the track is overly elaborate considering the simplification of the vehicle portion of the model. HALF was not altogether eliminated, however, because of its close relationship to the other three TSC models which are expected to be validated.

4.2.9 WHRAIL, and WHRAILA, Symmetric and Asymmetric Wheel/Rail Geometric Constraints (Law and Cooperrider)

In addition to the models described in paragraphs 4.2.1 through 4.2.8, two routines (WHRAIL and WHRAILA) were selected for use in TDOP Phase II as the most sophisticated wheel/rail representation available. These two routines will be used as auxiliary programs for such models as HUNTCT and the 9 and 17 dof Steady State Curving models which require detailed simulation of wheel/rail geometrical relationships. It may also be possible to adapt other models with less sophisticated wheel/rail geometrical representations for one or both of these routines.

The Symmetric Wheel/Rail Geometric Constraint program will be used predominantly, except where significant left/right asymmetry in either wheelsets or rails is identified by test data. In such cases the asymmetric version will be used. Both routines have been validated with laboratory "mock-ups." In TDOP Phase II, they will be implicitly validated when the models which use them are validated.

4.2.10 Models Not Selected for Validation

After the detailed assessment was completed, three models were not selected for further validation because either the same capability was provided by another tool, or they were overly complex and costly to run. A description of the three models is given below.

Lateral/Vertical Model. The Lateral/Vertical (L/V) Model was developed at AAR by the Track/Train Dynamics group as a tool to investigate lateral stability of freight cars. In particular, the model can be used to make a determination of the approximate ratio of lateral to vertical forces at the wheel/rail interface, thereby giving an indication of the potential for wheel climb and derailment. The model involves 14 degrees of freedom. Only a single truck and half a carbody are represented. The representation of the truck allows for nonlinearities such as center plate Coulomb damping. The solution technique is by time integration. The strength of the model is in the detail with which the wheel/rail profiles are defined; however, other tools selected for Phase II assessment treat the profiles in a similar, but easier to use, manner. Therefore, the Lateral/Vertical Model has not been selected for validation.

Freight Car Curving Model. The Freight Car Curving model is a nonlinear analysis program which uses time integration techniques to simulate the dynamic curving behavior of railroad freight cars. This model allows 43 degrees of freedom and features particularly detailed representation of the trucks. Nonlinearities include spring bottoming, clearances, and Coulomb damping. Time integration is used to solve the equations of motion. The level of detail in the simulation offers the possibility of good validation with test results, however, no validation has been carried out to date. Because of the complexity of the model, lack of documentation, and relatively high cost, this model was not selected for validation.

TDOP Phase I Model. This model is a linearized frequency domain model which attempts to represent the lateral and vertical dynamics of a freight car. A total of 13 degrees of freedom are used to model trucks and carbodies. The model has been used to make comparisons with TDOP Phase I test data but with only poor results. The model has received a great deal of scrutiny which has identified fundamental technical flaws contributing to discrepancies with test data. Because of its serious limitations, the model is not recommended for use as a TDOP Phase II analysis tool.

4.3 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 could be used with confidence without successfully being validated by comparing results from the model against actual tests of a freight car truck. 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 (reference 1). 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 critiera 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 assessing the range and sensitivity of the performance indices. A detailed discussion of the validation criteria for the four performance regimes follows.

4.3.1 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 of other tests.

A detailed review of TDOP Phase I test data 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 in both the development and assessment of models. At speeds below critical, 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 an often abrupt increase in the frequency of oscillation which, for the cars tested by both TDOP and the AAR (reference 2) is close to the natural frequency in yaw of the carbody on its suspension. Since the mass moment of inertia of the carbody about the center plate 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 center plate at the higher frequency. It has been observed in hunting tests that violent body hunting can coexist with very small lateral truck displacements, which is an indication of the small amount of energy required to maintain a limit cycle (reference 3).

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 supporting evidence of 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 $\pm$ 5 mph has been chosen based on $\pm$10 percent of the 50 mph critical speed range (see Figures 4-1 and 4-2). 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 4-3 shows the envelope within which Phase I data fell. A tolerance of $\pm$0.3 Hz based on $\pm$10 percent of the maximum observed frequency of approximately 3.0 Hz has been set for comparison of hunting frequencies.

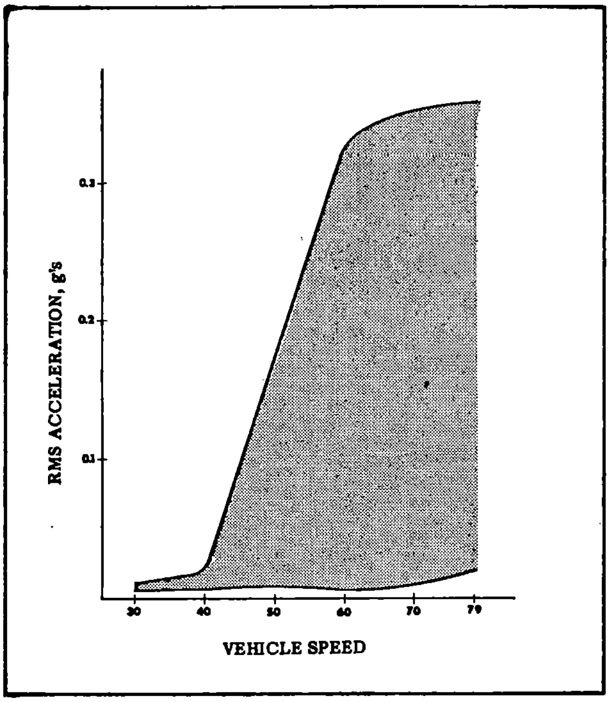


Figure 4-1. Lateral Stability Performance Bounds - RMS Lateral Carbody Acceleration Vs. Vehicle Speed (Box Cars)

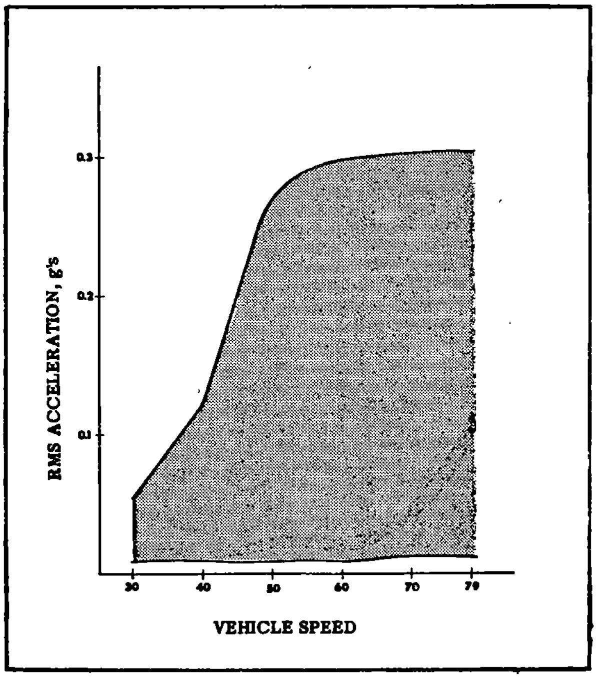


Figure 4-2. Lateral Stability Performance Bounds - RMS Lateral Carbody Acceleration Vs. Vehicle Speed (Flatcars)

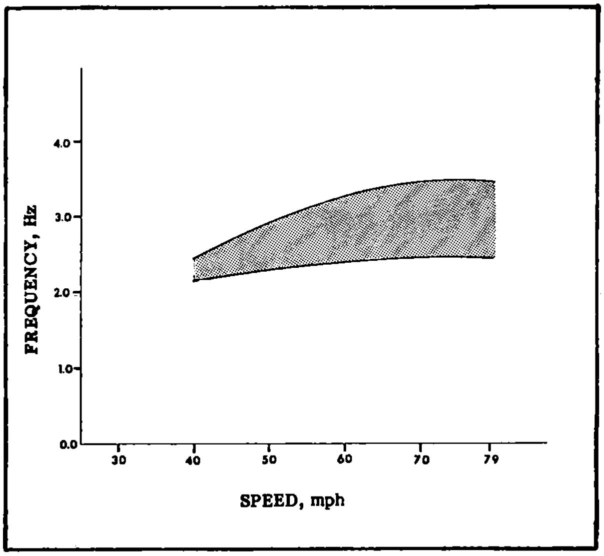


Figure 4-3. Range of Hunting Frequencies Vs. Critical Speed for Type I Truck

4.3.2 Validation Criteria for Trackability Models

The trackability regime encompasses 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 load equalization with respect to track twist, curve entry and exit, and harmonic roll and bounce.

The track twist/load equalization problem is largely a quasi-static phenomenon (speeds 5 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

$$WUI = \frac{W_H/3 - W_L}{W_H/3} \div \theta = 1 - \frac{W_L}{W_H/3} \div \theta, \text{ degree}^{-1}$$

Where:

W_L = force on the most lightly loaded wheel

W_H = sum of 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 terms of the curve negotiation performance regime (see Section 4.3.3). 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 of the vehicle occurs at higher speeds between 40 and 65 mph on 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 is more data available for comparison for that subregime. Figure 4-4 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 4-4. Figure 4-5 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 5 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 $\pm$ 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).

4.3.3 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 lateral force is inevitable due to track irregularities which cause the Coulomb friction elements such as the center plate 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 is expected to improve the measurement accuracy. With improved experimental techniques, a ± 20 percent tolerance in the prediction of the performance index is considered reasonable.

Curve entry and exit can be characterized by peak carbody roll angle and wheel load distributions. Following the case of harmonic roll excitation, the validation criterion for roll angle is ± 1 degree and for wheel loads ± 5 percent of the nominal static value.

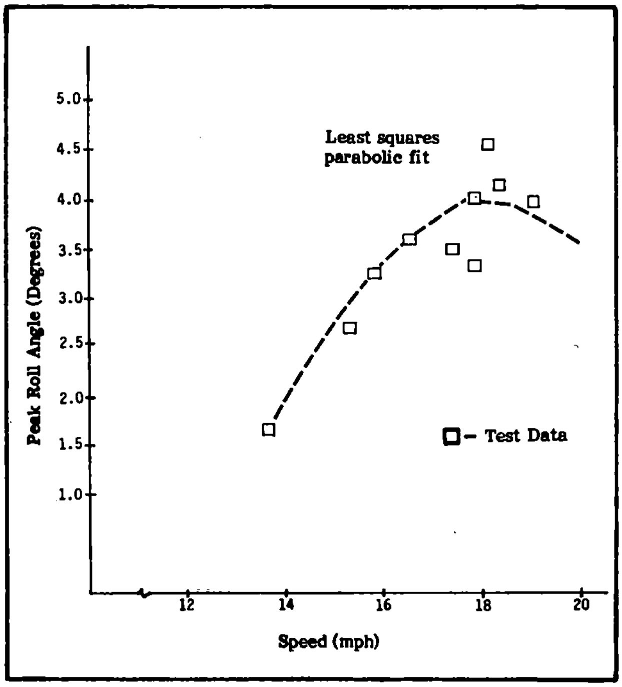


Figure 4-4. Variation in Harmonic Roll Test Data for Peak Roll Angle

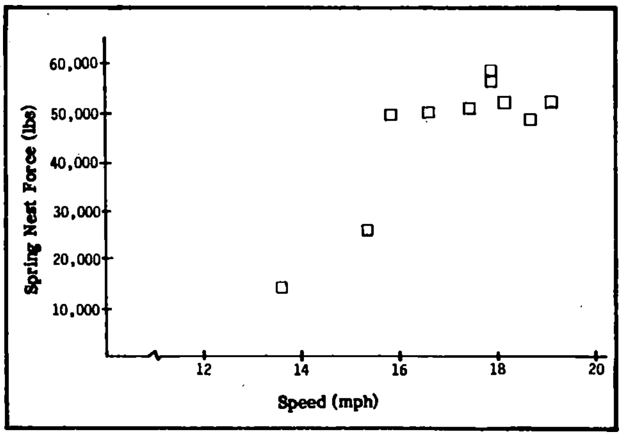


Figure 4-5. Variation in Harmonic Roll Test Data for Peak Spring Nest Force (Single Amplitude)

4.3.4 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 transmissibiity 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 ± 2.0 Hz is considered acceptable which is 10 percent of the highest frequency of interest. The tolerance for reproduction of output/input ratios, either transfer functions or RMS, is set at ± 20 percent.

4.4 VALIDATION RESULTS

This section documents the results of the analytical tool validation efforts which have been carried out in each of the performance regimes. See Table 4-2 for the list of models originally chosen as candidates for validation.

4.4.1 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 originally selected as a candidate for validation. It was felt that this would be more productive on the basis of the similarity between the Freight Car Hunting Model, HUNTCT and the 17 dof Eigenvalue Model.

4.4.1.1 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 1). The model was selected for validation as one of the most sophisticated linear models of freight car lateral stability.

At low speeds the natural frequencies for the carbody motions were also checked against simple theoretical predictions. The successful comparison of these identifiable frequencies provided confidence in the model's formulation and implementation.

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 phenomenom 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 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 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 parameter which showed a significant sensitivity was the conicity. By artificially 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 4-6. 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.

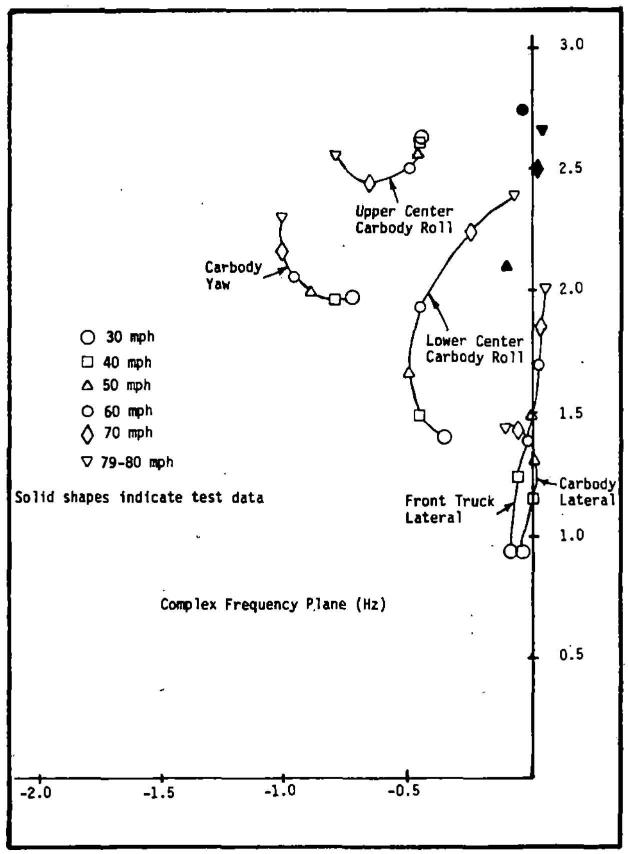


Figure 4-6. Root Loci for Principal Motions of Empty 70-ton Reefer from 17 dof Eigenvalue Model Vs. Test

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 then 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 the instability appear to be beyond the model's capability.

4.4.1.2 HUNTCT. HUNTCT is a nonlinear time domain model developed for lateral stability analysis. The model was developed at Wyle Laboratories and was chosen as a validation candidate on the basis of the detail of its formulation and Wyle's familiarity with its capabilities. The formulation uses 21 basic degrees of freedom. Although additional degrees of freedom to represent carbody flexibility can be included, in the validation exercises conducted with data from virtually rigid box-type carbodies, this feature was not used. To simulate actual tests the model requires track geometry data including left and right rail profile and alignment data. The model makes use of WHRAIL, the Symmetric Wheel/Rail Constraint Subroutine (reference 4) to relate the track input to wheelset motions. 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.

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 assumption made about the input data.

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.

4.4.1.3 Wyle Ad Hoc Modeling. Concurrent with the assessment and validation work that has been performed with the more comprehensive lateral stability models, a modeling effort has focused on the determination of the causes of the more enigmatic aspects of behavior observed in TDOP Phase I testing. Explanations were sought for the occurrence of "nosing" (leading truck oscillating, trailing truck quiescent), fishtailing (opposite of nosing), intermittent hunting, large variations in critical speed between similar configurations, abrupt frequency increases with increasing speed etc. It is believed that these quirks of behavior to a large extent can be traced to particulars of the wheel and rail contours and truck tramming equilibrium position.

Early simulations of the lateral stability regime were based on the assumption of conical wheel treads without throat or flange contact. It was established that, in rigid and three-piece trucks, the critical speed decreases with increasing wheel taper or "effective conicity," i.e., the angle of the contact plane between wheel and rail. Later models were able to simulate mild states of wheel wear by linearization techniques that included the increase in wheel taper and shift of the contact point with lateral displacement of the wheelset. However, none of these simple models was able to account for the sometimes high, sometimes low critical speeds of freight cars with severely or lightly worn wheels which sometimes differed by over 20 mph. An analysis of the results of two test programs (references 5 and 6) showed that the observed dynamic behavior could be explained by the degree of asymmetrical wear of the wheel contours.

In the cars unstable at low speeds, the kinematic frequency of the wheelsets with low levels of asymmetry was close to the frequency of the carbody's lateral modes at a fairly low speed, so that a low critical speed could be expected. By contrast, the leading wheelsets of the stable trucks, because of their asymmetry, were in contact with the rail at the throat of one wheel, where the effective conicity was very high. The kinematic frequency of the stable trucks was high enough above the lateral natural frequency of the carbody to minimize amplification, while the reverse applied to the unstable trucks.

It was clearly concluded in reference 5 that the worn wheel profile was considered the most important determinant of lateral stability, far outweighing such other factors as snubber wear, flange geometry or carbody load distribution. This conclusion, and its verification by simulation, has important implications for both model building and model validation. It is believed that, in the simulation of hunting, relatively simple models of both trucks and carbody are adequate as long as the worn wheel contours are adequately represented. This approach is considered the more important in view of the fact that a wheel stays "new" for only a very short time since the wear rate is high due to the small area and resulting high bearing stress of the contact patch on the taper. Thus, the stability of the truck on worn wheels is of greater interest to the railroad industry.

The incorporation of worn wheel profiles in a freight car model involves these three separate steps:

a. Representation of wheel/rail contact geometry, i.e., transformation of wheel and rail contours to digital form.

b. Representation of wheel/rail contact mechanics, i.e., creep coefficients, loss of adhesion, multiple contact points, etc.

c. Incorporation of the wheel/rail contact model in the vehicle model.

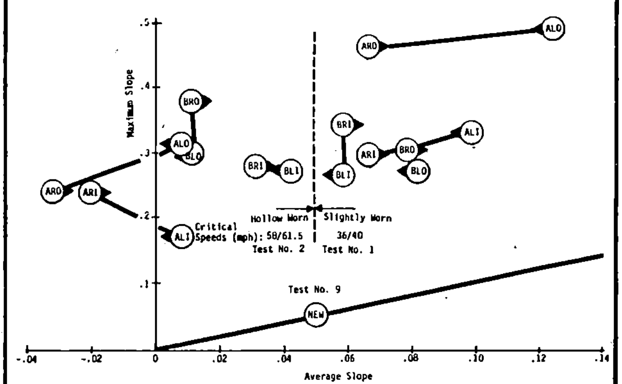
It must be emphasized that the validity of these three modeling procedures can only be demonstrated after completion of the third stage, when results of simulation by means of the model containing the presentation of the worn wheels has successfully reproduced the behavior of the real vehicle.

Representation of Wheel/Rail Geometry. In connection with the analyses of hunting of flat cars described in reference 6, Wyle Laboratories developed a program to model unsymmetrically worn wheels. Subsequently, Heller and Cooperrider (reference 7) have published results using a similar approach. The objectives of this wheel/rail geometry representation are:

- To determine the minimum amount of information required to characterize an unsymmetrically worn wheelset with respect to its dynamic performance.

- To produce a graphic means that would make it possible to identify subtle differences between worn wheel profiles, as a means of predicting their dynamic performance without necessarily carrying out the simulation with the mathematical model of the complete vehicle.

The Wyle procedure codes the contours of the wheel and rail as a pair of polynomials (one for the tread or rail head, and one for the throat and flange or gauge side contour) and then determines the radius difference ($\Delta R/R$) and rise of the wheelset center as the wheel is displaced laterally across the rail. The derivative of the latter, multiplied by the axle load, is the gravitational stiffness. Plots of the wheel tread slopes were found to correlate with critical speeds (See Figure 4-7, taken from reference 6).



NOTE: ARO is A-end, right outer wheel, etc.

Figure 4-7. Wheel Tread Characteristics - 50 Ft. Box Car

By restricting the application of the wheelset model to the small displacements associated with the region of initial instability, the characterization of the wheelset may be further simplified by expressing the $\Delta R/R$ and gravitational stiffness variations with lateral displacements by combinations of straight line segments and parabolas, obtained by simple least square computations. Plots of the wheelset characteristics, such as those shown in Figures 4-8 and 4-9 (which depict average of rolling radii and normalized differences of rolling radii), indicate such conditions as unsymmetrical wear, slight hollows worn into the profile, and whether a wheelset has been rotated to counteract unsymmetrical flange wear. The assumption of small lateral displacements (verified by results from the complete vehicle program) justifies the assumption that the lateral creep forces always act in a horizontal plane.

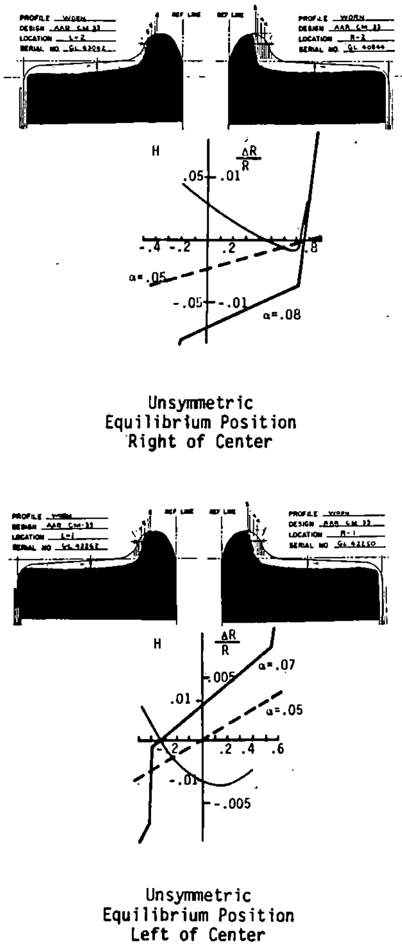


Figure 4-8. Service Worn Wheels on Southern Pacific Rail, Cross Worn

Representation of Wheel/Rail Contact Mechanics. The representation of wheel/rail contact mechanics is also simplified. Linear creepage is assumed, a single creep coefficient is used both laterally and longitudinally, and spin creep is neglected. However, an adhesion limit is specified. Secondary phenomena are neglected, such as the change in wheel load due to vehicle roll, and the destabilizing effect of gravitational stiffness on a yawed wheelset.

It should be emphasized that these features were deliberately omitted from the ad hoc model so as to determine the minimum number of essential elements required to characterize the wheelset. Features now absent in the model can be easily introduced when the need is demonstrated, as was done for some of the parameters in the suspension.

Incorporation of Wheel/Rail Representation in Truck and Carbody Models. The ad hoc model of the three-piece truck has three degrees of freedom: lateral displacements of the two wheelsets, and wheelset/bolster yaw. A linear torsional spring rate at the side frame/bolster connection is assumed.

Since each truck, assembled with the carbody, both provides excitation to itself and is in turn excited by the other truck through the carbody, not only the kinematic wavelength of the isolated truck, but also its kinematic response to sinusoidal lateral excitation is of interest. From transfer function analysis, the results of interest are, first, the attenuation of the response as the excitation frequency is increased beyond the frequency of kinematic hunting, and, second, the rise in response as the speed is increased and the damping correspondingly decreased.

The carbody is assumed to have three rigid body degrees of freedom: lateral displacement, yaw, and roll. Inclusion of torsional flexibility adds another degree of freedom. The forward and rear halves of the carbody are each assumed to have one half the mass moment of inertia of the whole body, and are connected by a torsion spring with a stiffness twice that calculated from the end-to-end torsional deflection as given in the TDOP Phase I Final Report (reference 8).

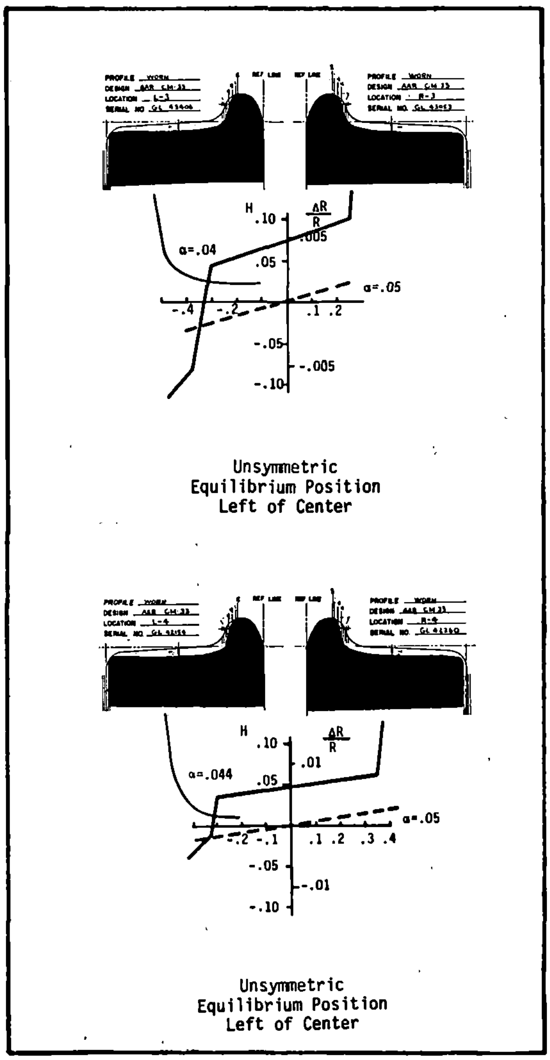


Figure 4-9. Service Worn Wheels on Southern Pacific Rail, Cross Worn Reversed

In the initial version of the model, viscous friction was assumed at the snubbers, both vertically and laterally, and at the center plate. Modifications in the damping characteristics were made during validation trials described below. No limiting due to finite gib clearance was assumed.

The phenomenon of "nosing" was simulated by means of a model of a three dof rigid carbody on an undamped suspension, excited at one end by a small sinusoidal force over a range of frequencies (see Figure 4-10). It is evident that near 2.5 Hz a very small excitation at one end produces widely differing responses at the two ends.

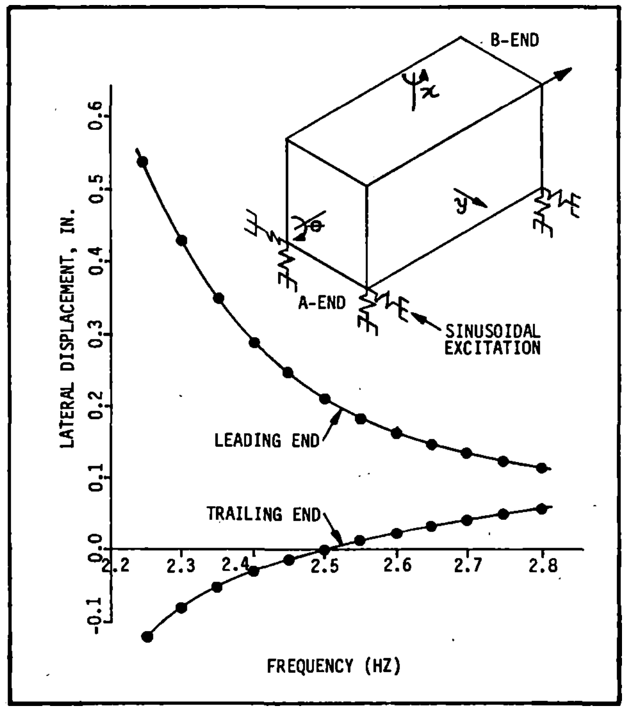


Figure 4-10. Combined Lateral, Yaw, and Upper Center Roll of 70-ton Refrigerator Car

First Model Validation Trial: Empty Refrigerator Car on New and Half-Worn (Machined) Wheels. Results of these simulations, comprising four-second computer runs, are shown in Figures 4-11 and 4-12. They show the lateral accelerations at both ends of the carbody. An initial displacement of 0.1 inch was set as an initial condition at the A-end (trailing). The profiles of one truck set of the machined wheels are shown in Figure 4-13. Except for the asymmetry, they resemble new wheels in that the effective conicity is constant over most of the contours, so that the gravitational stiffness was not taken into account. Viscous damping is assumed for both the snubber and the center plate, and the creep coefficient is assumed as 600,000 lb per wheel.

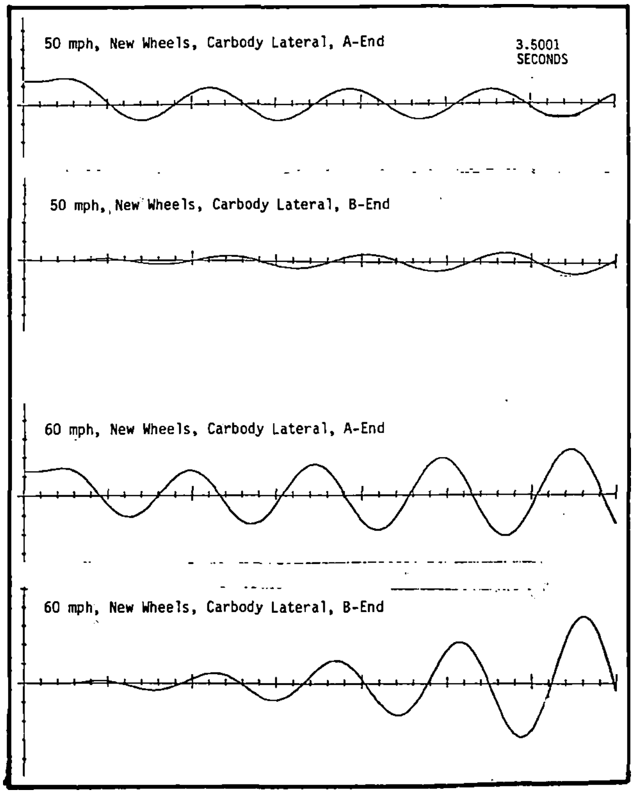


Figure 4-11. Simulated Refrigerator Car with New Wheels at 50 and 60 mph

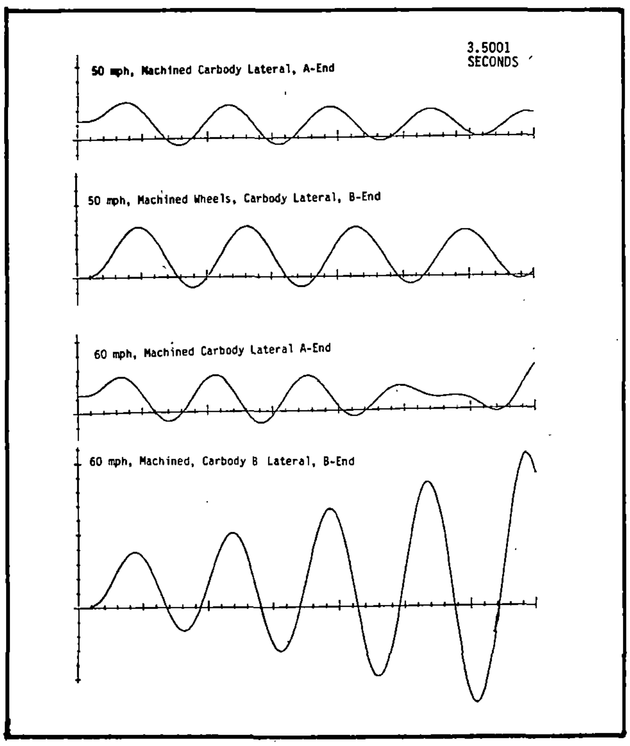


Figure 4-12. Simulated Refrigerator Car with Half Worn (Machined) Wheels at 50 and 60 mph

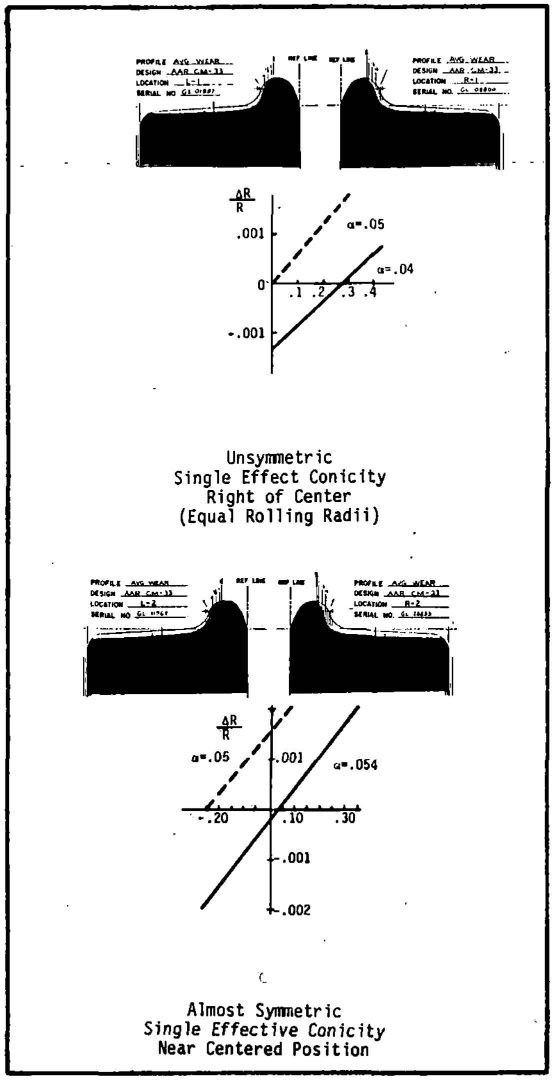


Figure 4-13. Profile of Average Worn (Machined) Wheels

The main resemblance between the simulation and the test data is in the critical speeds which are between 50 and 60 mph in both cases. In addition, the B-end moves with greater amplitude than the A-end (although the difference is less than that observed in the tests), thus, indicating some degree of "nosing". This does not apply to the case of new wheels at 50 mph, but this may be due to the short duration of the simulation which did not allow the final dynamic state to develop.

However, it is evident that the frequencies of oscillation are much closer to the kinematic frequencies of the isolated trucks than to any of the identified yaw frequencies of the carbody (between 2.0 and 3.1 Hz). Since the frequency is one of the determinants of lateral acceleration, this simulation is useless for the prediction of lateral g. However, these results raise the interesting question as to whether simulation can be trusted to predict the correct critical speed at the wrong frequency.

An obvious condition for exciting the higher frequencies of the carbody is stick-slip friction at the snubbers, which was consequently incorporated in the model. This is consistent with the overall approach in which the model is elaborated only when required.

<u>Second Model Validation Trial: Empty 89-Ft. Flat Car on Service Worn Wheels.</u> The 28-inch wheel profiles and their characteristic curves (reference 8) are shown in Figures 4-14 and 4-15. From the column load of 4110 lb per snubber and a friction coefficient of 0.37, the lateral friction force per truck is about 6000 lb. To preclude numerical problems, a dead band of 0.01 in. was assumed, in which the force at the snubber is proportional to the relative displacement between side frames and bolster. This gives a spring rate of 1,200,000 lb/in., which is sufficiently high above the lateral suspension spring rate of 10,000 lb/in. to leave the lateral dynamics unaffected. However, to avoid numerical stability problems, viscous damping was added so as to act in this region only. Equivalent viscous friction in the vertical snubbing direction was retained, as well as viscous center plate friction.

The calculated RMS lateral accelerations for a speed range of 40-60 mph are shown in Figure 4-16, superimposed on the accelerations from the Phase I test as given in reference 9, together with the calculated and measured frequencies. The introduction of Coulomb damping has evidently succeeded in exciting the correct carbody frequencies, and the rate of

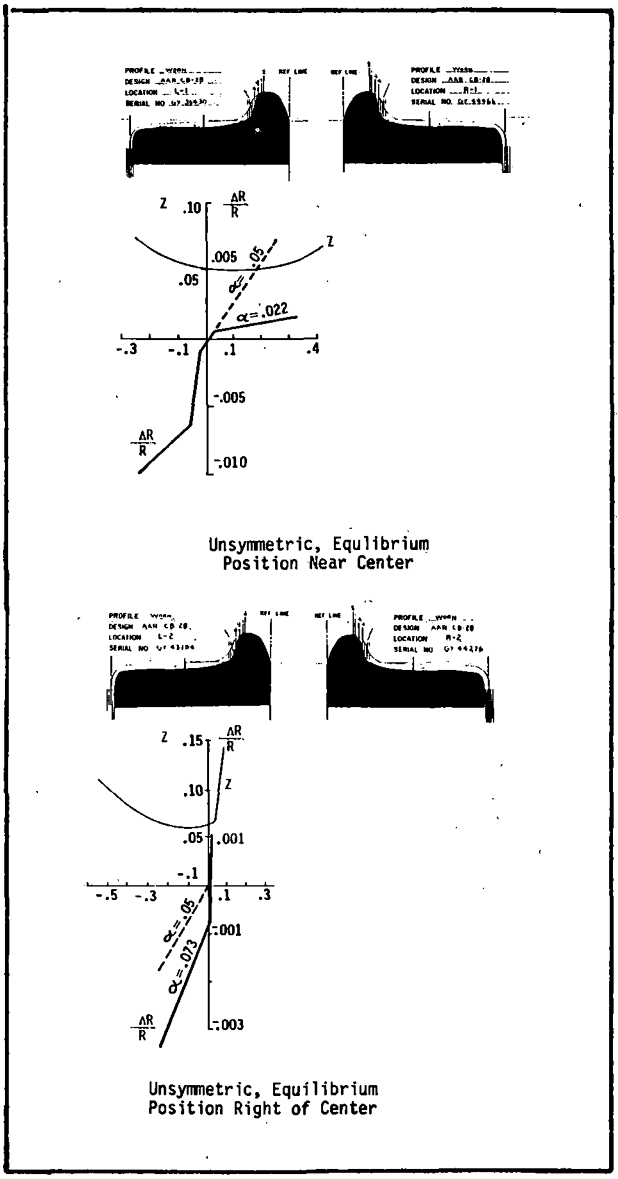


Figure 4-14. Profile of Service Worn, 20-Inch Wheels (Unsymmetric Equilibrium, Position Near Center)

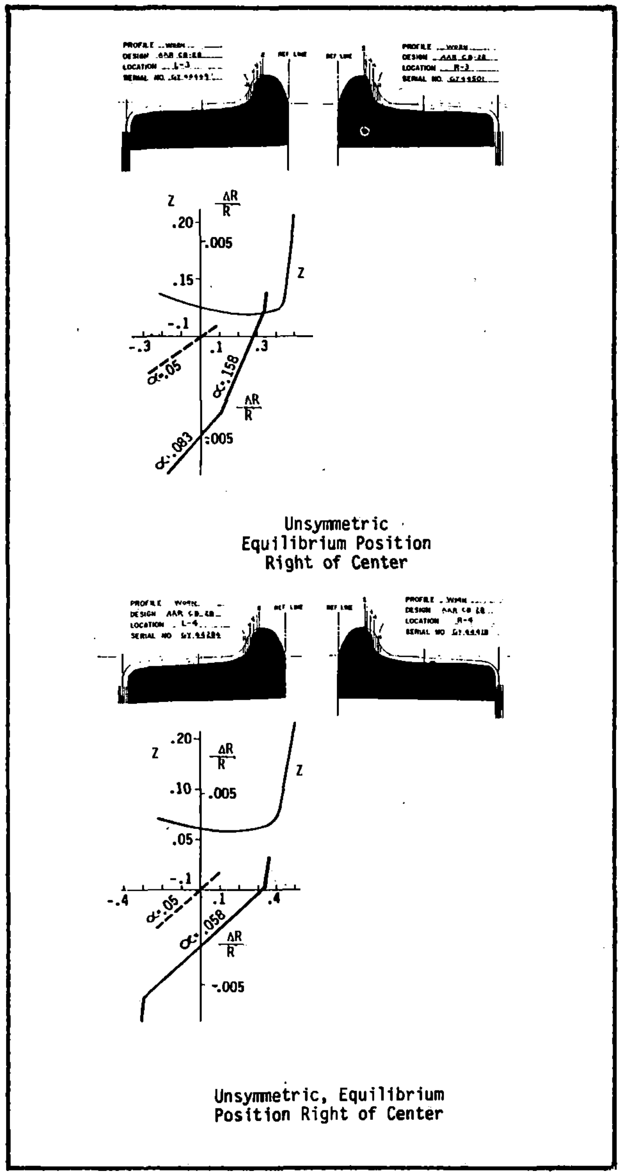


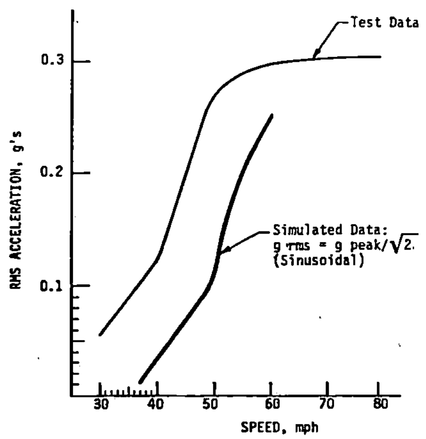
Figure 4-15. Profile of Service Worn, 28-Inch Wheels (Unsymmetric Equilibrium, Position Right of Center)

rise of acceleration with increasing speed is practically the same for the simulated and test data. However, the curve of simulated accelerations appears laterally shifted, indicating a critical speed about ten mph higher than that derived from the test data.

MODEL INCLUDES: ACTUAL WHEEL PROFILES
BOLSTER COULOMB FRICTION/STICTION
CREEP COEFFICIENT = 800,000 lb/Wheel

FREQUENCIES

SPEED MPH	TEST DATA	SIMULATION
40	2.15 Hz	1.92 Hz
50	2.54 Hz	2.63 Hz
60	2.93 Hz	2.94 Hz



TYPE 1 70 TON TRUCK WITH FLAT CARS
- Worn Wheels
- Empty Cars

RMS LATERAL ACCELERATION Vs SPEED

Figure 4-16. Comparison of Test and Simulated Data

4.4.2 Validation Results of Trackability Models

The validation work in the trackability regime has focused on harmonic roll and bounce models. Curve entry and exit phenomenon are discussed in relation to validation of curve negotiation models (see Section 4.4.3). Furthermore, since the harmonic roll and bounce models can generally be applied to the load equalization 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 4-2 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 has been treated.

4.4.2.1 Flexible Carbody Vehicle Model. The Flexible Carbody Vehicle Model was developed by Tse and Martin of AAR in conjunction with the Track/Train Dynamics Program (reference 10). Version II of the model has 20 degrees of freedom as shown in Figure 4-17.

The model includes nonlinear representations of the friction snubbers, optional auxiliary snubbing devices, center plate rocking, gib and side bearing clearances, flange clearance, and finite spring travel. The carbody is divided into two half-carbodies connected by springs to allow a first order representation of carbody flexibility.

The model was chosen for validation on the basis of the level of detail of the representation. In addition, the model has been widely disseminated among the railroad community. The model provides the option for a variety of stylized track inputs which, depending on the track stagger, excite harmonic roll or bounce behavior on tangent, curved, or ramped curved track. For the model validation exercise, the harmonic roll on tangent track option was chosen because more data is available for this combination.

The experimental data used were acquired by American Steel Foundries (ASF) in tests with a loaded, 100-ton hopper car on half-staggered shimmed track at Hartford, Illinois in 1968. This data had been used in prior validation work with Wyle's harmonic roll model, FRATE (reference 11).

The results of the comparison of roll angle (single amplitude) for the tests and the model are shown in Figure 4-17a. 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 in Figure 4-17a 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.

Wyle scrutinized the model for other sources of error and found two aspects of the program of which users should be mindful. The first was a pair of statements in the subroutine ACCEL in which the instantaneous accelerations of the various generalized inertias are calculated. The two statements were:

```
IF (ABS (DIS(I).LT(1.0E-05) DIS(I) = 0.0
IF (ABS (VEL(I).LT.(1.0E-05) VEL(I) = 0.0
```

DIS(I) and VEL(I) are the displacement and velocity along the Ith generalized coordinate. The two statements are ostensibly included to protect against numerical underflows. It was found, however, that with an otherwise stable integration step size, the threshold of 10^{-5} was too large and prevented smooth integration. That is, small but significant contributions to the acceleration calculations arising from small finite displacements, and velocities were being negated by the two statements above. By decreasing the size of the threshold to values just within the machine underflow, the integration problem could be avoided and the original intent of the statements preserved. The correction of the two statements above by itself did not significantly change the results of Figure 4-17a.

The second anomaly noted is more fundamental and is associated with the formulation of the model. As noted in Figure 4-17, the model includes separate lumped masses for the truck bolsters. The bolsters are attached to very stiff springs which represent center plate rocking stiffnesses. The combination of the relatively small bolster mass with the stiff springs gives rise to natural frequencies of 200 to 250 Hz. Such frequencies would dictate integration time steps of 1/(10*250 Hz) = 0.4 ms. This is consistent with the time step of 0.256 ms recommended in the model's documentation (reference 10). However, the bolster is also attached to nonlinear Coulomb damping elements modified with a narrow viscous range at very low relative velocities (see Figure 4-18).

The effect of the nonlinear damping representation is to restrict the integration step size to even much smaller values to obtain numerical stability with respect to the bolster accelerations. In this regard, AAR carried out a study of the effects of the numerical instability of the bolster acceleration and showed that the instability was of such a high frequency that it did not significantly affect the integration of other truck motions (see Figures 4-19 a, b, c, and d).

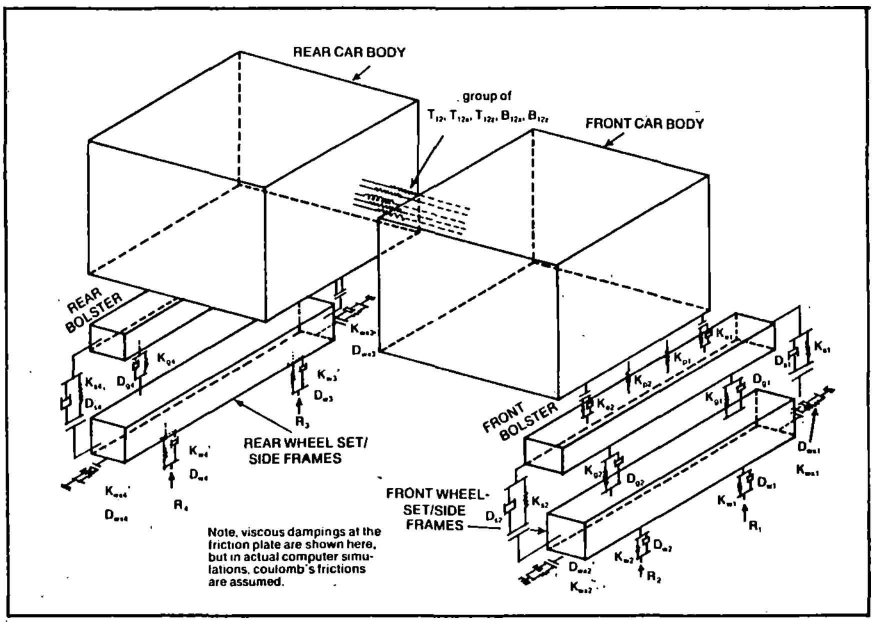


Figure 4-17. Comparison of Roll Angle Test and Model Data

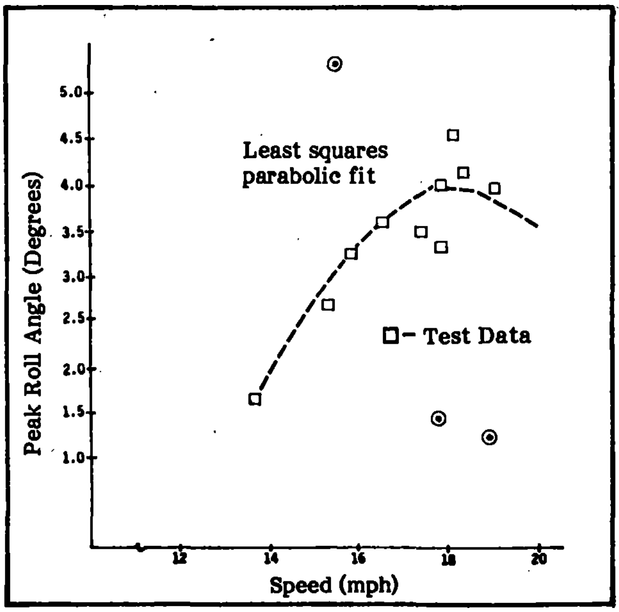


Figure 4-17a. Flexible Carbody Vehicle Simulation Results (Single Amplitude)

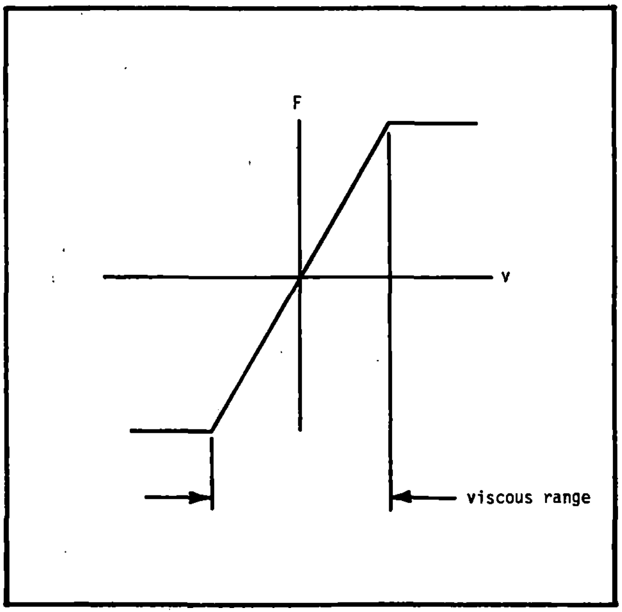


Figure 4-18. Flexible Carbody Vehicle Model Column Damping Characteristics

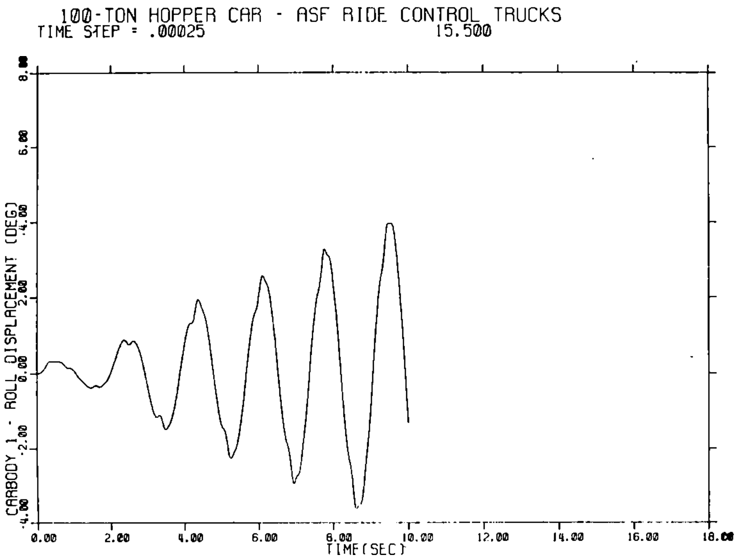


Figure 4-19a. Flexible Carbody Vehicle Model - Roll Displacement, Time History at Recommended Time Step

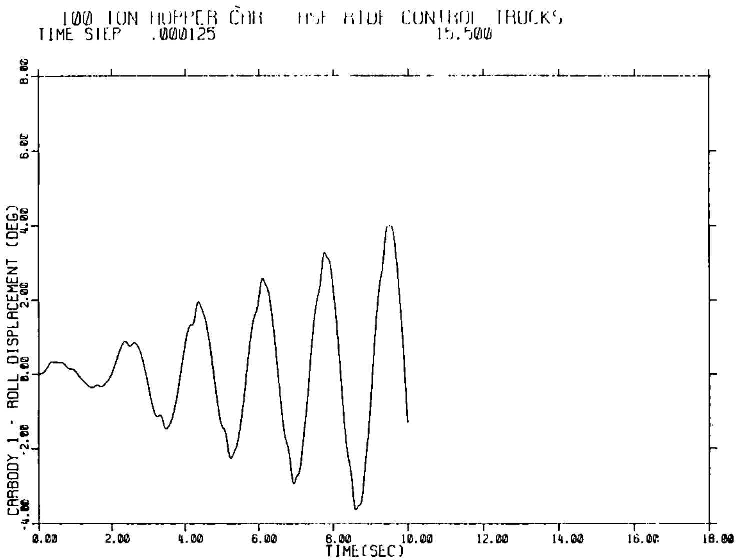


Figure 4-19c. Flexible Carbody Vehicle Model - Roll Displacement, Time History at Half-Recommended Time Step

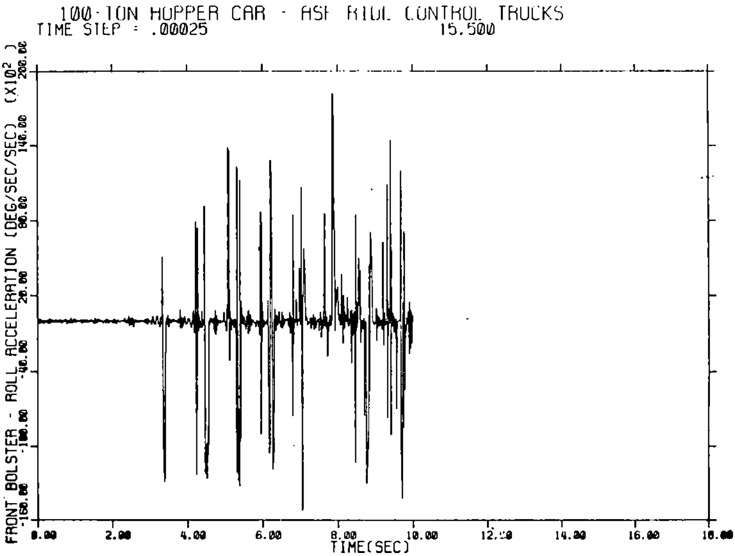


Figure 4-19b. Flexible Carbody Vehicle Model - Bolster Roll Accelerations, Time History at Recommended Time Step

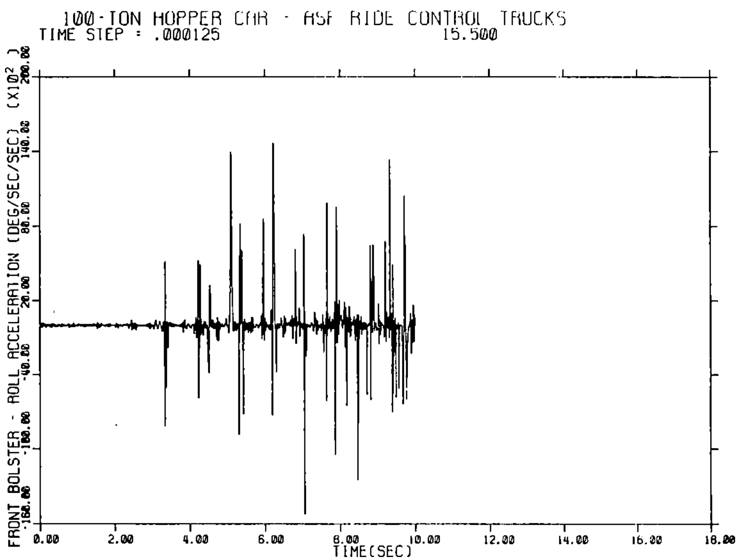


Figure 4-19d. Flexible Carbody Vehicle Model - Bolster Roll Acceleration, Time History at Half Recommended Time Step

4.4.2.2 FRATE. The second trackability model addressed was the FRATE model developed by Wyle Laboratories (reference 11). FRATE exists in 11 and 17 dof versions. The validation cases discussed here pertain to the 11 dof version as the test configurations used did not warrant the additional complexity of the 17 dof version. The appropriate use of the more complex version will be discussed later. Figure 4-20 provides a description of the 11 dof FRATE model. Note that the small bolster mass has been lumped with the rest of the truck in contrast to the Flexible Carbody Vehicle Model. At each location where spring elements are indicated in Figure 4-20 provision is made to include a viscous or Coulomb damping element. Other nonlinearities such as finite spring travel, clearances, and stops are also included.

The FRATE model was selected for validation based on the favorable comparisons with test data that the model had previously produced and by virtue of Wyle's familiarity with it. 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. FRATE is implemented on Wyle's in-house Interdata computer.

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 4-3 summarizes the results for the validation exercise with the ASF data against the validation criteria described in paragraph 4.3.2.

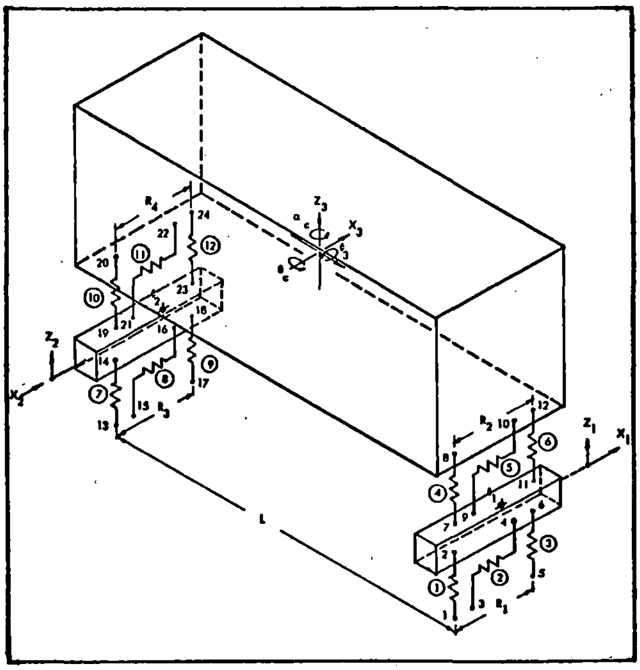


Figure 4-20. FRATE 11 Model

Table 4-3. Comparison of FRATE vs. Test Results

Comparison	Criterion	Deviation
Peak Roll Angle at Resonance	$+1^{o}$	0.6^{o}
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 results with a different test case from the TDOP Phase I series. The case chosen was test number 191 (reference 8), 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 ft. Approximately 400 ft. 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 4-21. Figure 4-22 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

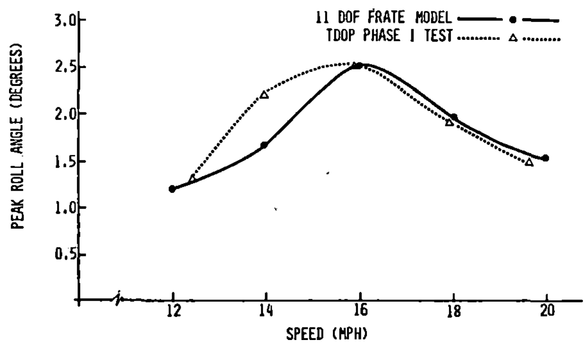


Figure 4-21. Comparison of 11 dof FRATE Model with Shimmed Track Data

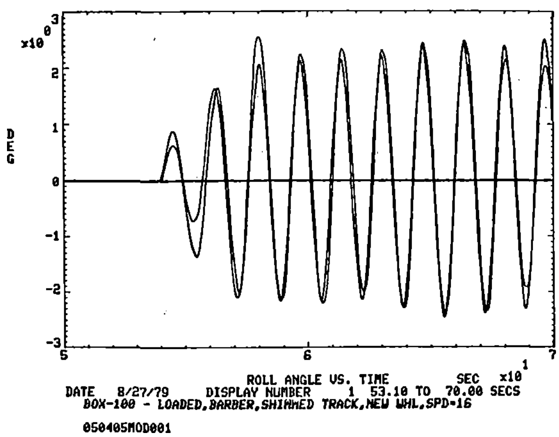


Figure 4-22. Comparison of Time Histories of 11 dof FRATE Model and Test Data at Resonant Speed

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.

Although the 11 dof FRATE model has produced favorable comparison 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 that the harmonic roll response of flexible flat cars requires a model of greater sophistication. Likewise,a more complex model such as the 17 dof version is recommended in cases where the excitation is sufficiently great to cause center plate rocking. For the more rigid box type cars at excitation levels below that causing center plate rocking, the 11 dof FRATE model produces satisfactory results.

4.4.3 Validation Results of Curve Negotiation Models

Two similar models have been identified as candidates for validation in the curve negotiation regime. These were the 9 and 17 dof Steady State Curving Models. 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 Rail Research Centre 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. A few sample runs were made initially to satisfy the suitability of using this model in TDOP Phase II. On that basis, the model was adopted.

The CN Rail Curving Model is an interactive program for the solution of steady-state behavior of two- or three-axle 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 approach to the solution is centered around the choice of a geometric state-vector which includes the lateral and longitudinal displacements of wheelset centers, yaw angles of wheelsets about an axis normal to the superelevation plane, dimensionless wheelset rotational velocities, lateral truck position, and truck yaw with respect to a rectangular cartesian coordinate system fixed to ground. An iterative procedure is used to improve the accuracy of the chosen trial state-vector.

Wheel/rail interaction is modeled as a two-point contact condition. At the tread, tangential forces are modeled using the nonlinear creep formulation of Johnson with Kalker's initial slope. Creep coefficients are assumed equal for longitudinal and lateral creepages. Flange contact is modeled as being at a constant inclination to the horizontal, and at a constant level below rail top. Both normal and tangential forces at this point are considered, with the normal force effecting a steering moment. Lateral forces arising from spin creep as well as from gravitational stiffness are neglected since these are assumed to be equal and opposite effects to a first order approximation.

Some of the other effects accounted for in the program include individual wheel radii, tread conicity, variable non-linear stiffness characteristics, wheel diameter mismatch, truck tram errors, and weight transfer and lateral force due to superelevation.

The efforts in validating this model involved a two step approach. Initially, the model attempted to simulate data generated by the Canadian railroads (Canadian National and Canadian Pacific) during their test programs. In particular, the Canadian Pacific study, "Comparative Performance of Type II Trucks" (reference 12) was used for comparison of specific sets of simulations. This study included the curving performance of a standard truck on new AAR standard wheel profiles. For lack of specific definition of parameter sets, the comparison was on a qualitative level stopping at trend correlations between the simulated data and test results. The results of these trend correlations were satisfactory in that the model did indeed reproduce the trends exhibited by the test data.

The second step in the validation efforts consisted of an attempt to reproduce test data obtained from ASF, which has been used in TDOP Phase II to characterize curving performance of Type I trucks. In this attempt, although the trends in the predicted lateral and vertical forces were closely reproduced, the goal of quantitative correlation between simulated data and test results fell short of accomplishment. Specifically, the forces predicted by the model varied from those indicated by the test data by as much as thirty percent in some cases.

This prompted a deeper examination of the model capabilities and the reason for such a large discrepancy between simulated and test data. This examination failed to uncover anything suspect with respect to the model; however, pursuing the matter of the accuracy of the test data, it was discovered that the data were indeed contributing to the discrepancy. Because of the broad uncertainty associated with the test data, it was decided that further fine tuning of the model to obtain a closer match between the two was not worthwhile.

The sources of the band of uncertainty within the data have been investigated and many of them have been identified. The contribution from these sources, at a maximum level, have been identified and it is proposed that a quantitative estimate of these uncertainties can be used to refine the results obtained from the test data and which could then be used as a basis for comparison with the simulated data. This effort is presently underway. Furthermore, the validation of the CN model is expected to advance as test data is acquired during the TDOP Phase II field testing of Type I trucks.

4.4.4 Validation Results of Ride Quality Models

The models selected as candidates for validation in the ride quality regime include the DYNALIST modeling program (reference 13) and a set of complementary models, HALF, FULL, FLEX, and LATERAL (reference 14). 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's subcontractor, the J.H. Wiggins Company. J.H. Wiggins followed the procedure used by other modelers (reference 15) 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.

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, we found that other modelers have also found it necessary to introduce unrealistically high damping in order to make the results of simulation and tests agree. The authors of reference 15 state, "The assumptions of symmetry and linearity allow the vertical motions to be independent, or uncoupled, from the lateral and rolling motions." However, they state on page 66 "that the response of the model in the lowest frequency range is consistently higher than in the actual measured response." Referring to Figures 4-23 a, b, c and d (taken from reference 15), the authors state, "Regarding the model response, note that high damping of the truck suspension yields much better agreement with experimental results than does the lower damping in the mid range of the spectrum." The authors 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 while viscous friction lowers it. 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 the figures cited from reference 16 that the major discrepancies are fairly sharply localized in the frequency range between about 2 and 5 Hz 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, equivalently, 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 Wiggins model, the introduction 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. Nevertheless, it is proposed to investigate the effect of lateral/vertical coupling by additional work with the flexible structure of DYNALIST. Based on the results with DYNALIST, the other structured frequency domain models (HALF, FULL, FLEX, and LATERAL) have not been treated since they are also uncoupled configurations.

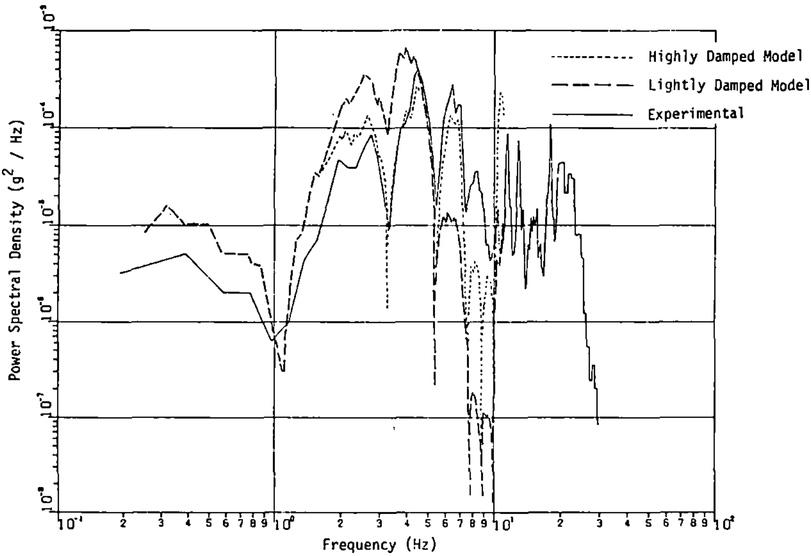


Figure 4-23a. Carbody Center of Gravity Vertical Acceleration Spectra (Empty Vehicle Traveling at 100 ft/sec over CWR)

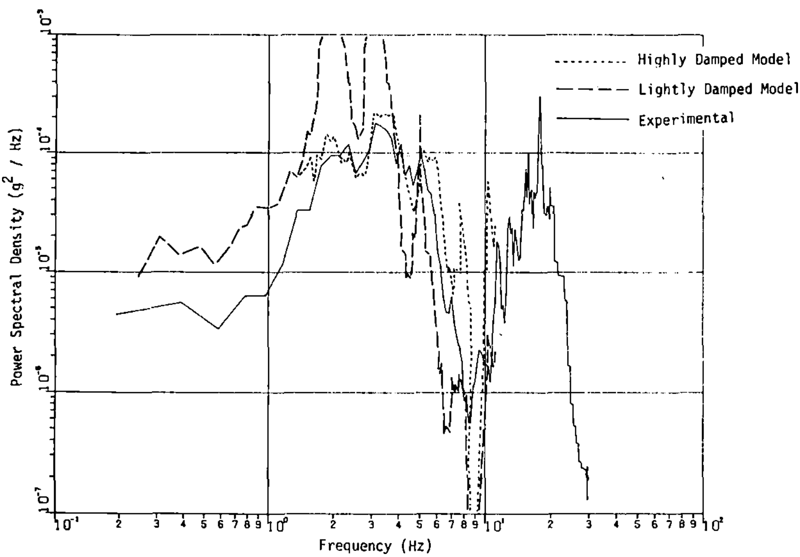


Figure 4-23c. Carbody Front End Vertical Acceleration Spectra (Loaded Vehicle Traveling at 100 ft/sec over CWR)

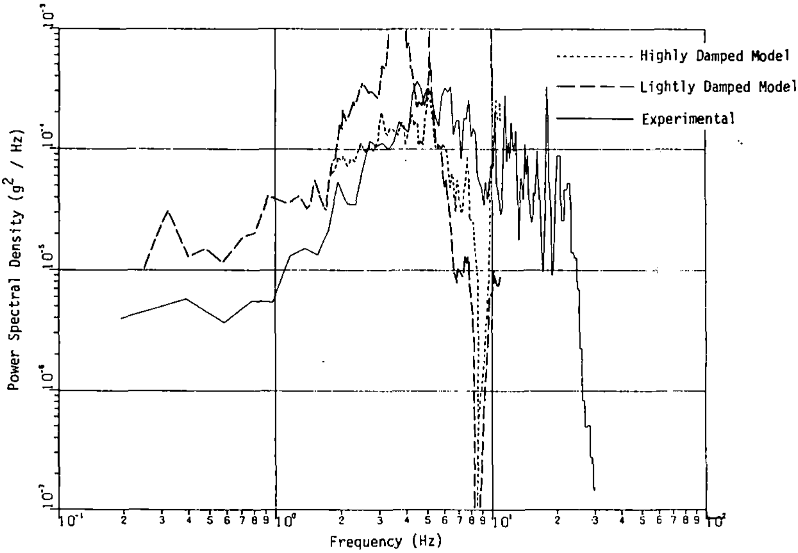


Figure 4-23b. Carbody Front End Vertical Acceleration Spectra (Empty Vehicle Traveling at 100 ft/sec over CWR)

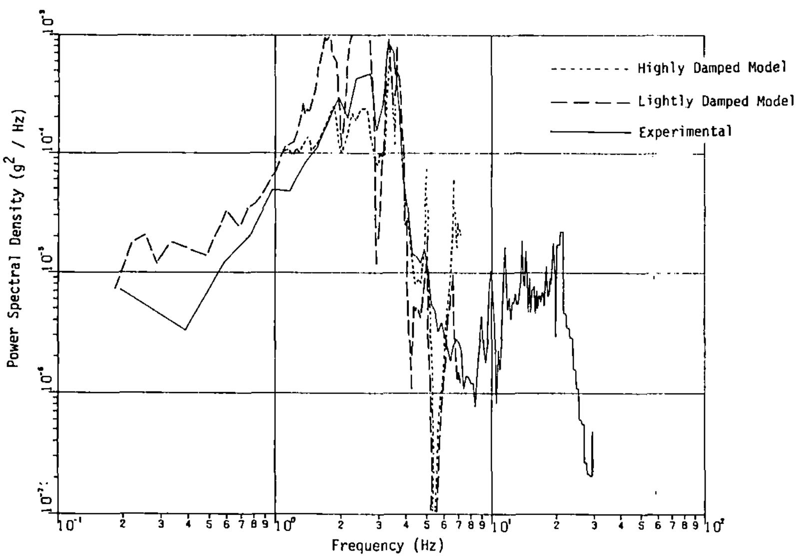


Figure 4-23d. Carbody Front End Vertical Acceleration Spectra (Loaded Vehicle Traveling at 64 ft/sec over Jointed Rail)

4.5 CONCLUSIONS AND RECOMMENDATIONS

The results of the validation exercises carried out thus far have been rather disappointing. 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., Figures 4-24 and 4-25).

During the remainder of TDOP Phase II, greater emphasis will be placed on the type of ad hoc modeling described in paragraph 4.4.1.3. The aim of such modeling will be the interpretation of test results. Simple models will be used to determine why a vehicle exhibits nosing, for instance, rather than attempt to construct a comprehensive hunting model. For some models such as the CN Curving Model and HUNTCT, additional validation work is justified pending the collection of more complete and accurate data from Phase II field testing.

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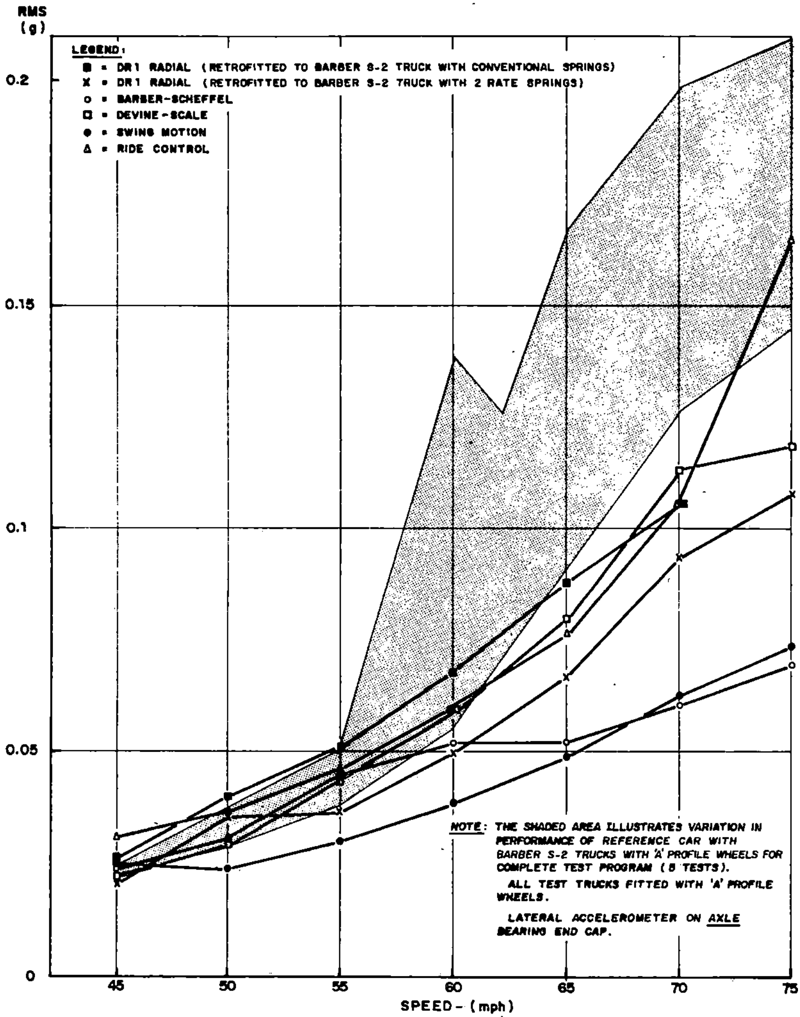


Figure 4-24. Lateral Acceleration on Axle Bearing End Car - RMS (g) Vs Speed (Reference 12)

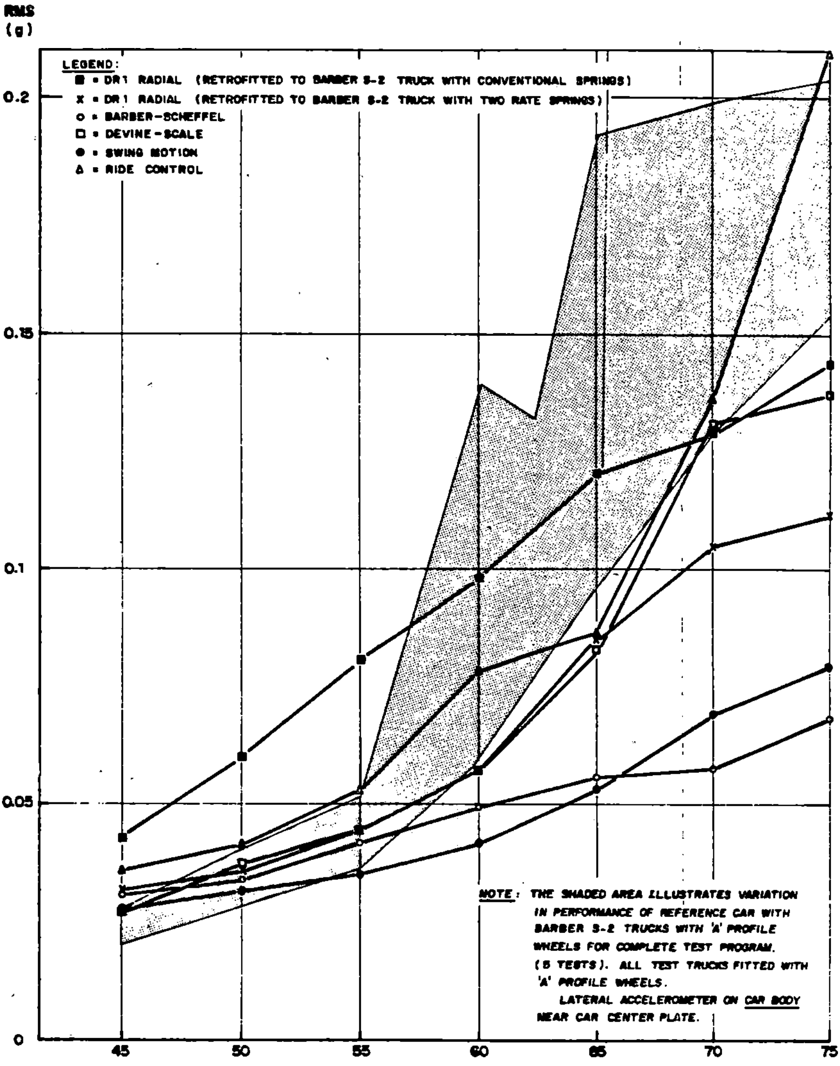


Figure 4-25. Lateral Acceleration of Carbody - RMS (g) Vs Speed (Reference 12)

SECTION 5 - FIELD TEST PROGRAMS

Field testing for the TDOP Phase II project consists of four separate programs: Type I truck testing, Type II truck testing, the wear data collection program, and the over-the-road test of the Friction Snubber Force Measurement System. This section discusses all of these test programs with the exception of Type II truck testing which will be covered in the project's Final Report.

Prior to the start of Type I testing, an evaluation was made of the test data collected on Type I trucks by the Southern Pacific Transportation Company (SPTCo.) during TDOP Phase I. The purpose was to ascertain if the data collected was sufficient to develop a performance characterization of the Type I truck without further field testing. A summary of this evaluation is included below.

5.1 PHASE I DATA EVALUATION

To determine the usefulness of the Phase I data, the quantity and scope of the data was first evaluated. 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 5-1). This emphasis made the data more difficult to use because the refrigerator car is not a typical freight car because of its uneven weight distribution and very high empty weight.

Table 5-1. 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 on 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 Phase I data evaluation next determined which measurements taken during Phase I provided useful and accurate representations of the quantity measured. For example, did the pins on which the strain gages were mounted in the adapter give an accurate representation of the lateral load at the wheel/rail interface? 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 is of critical importance to TDOP Phase II. Without it, there is little that may be done in validating curving models or assessing the curve negotiation performance indices on the Type I truck. Also, these missing data would have a secondary influence on the analysis of lateral stability because the time domain models could not be validated.

Finally, the Phase I data were evaluated for their adequacy in performing and supporting the development of Type I truck performance characterization. This evaluation is shown in Table 5-2 which lists the performance index for each of the four regimes and the test data required to specify the performance index. For the lateral stability and ride quality regimes, the data appear to be adequate; however, the lack of accurate measurements on the lateral forces at the wheel/rail interface will make it difficult to extract from the test data meaningful information for the curve negotiation and trackability regimes.

Table 5-2. Test Data Required for Engineering Analysis

Performance Regime	Performance Index	Necessary Test Data	Availability of Test Data from Phase I
Lateral Stability	• Critical Speed of Hunting	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.
	• Maximum Lateral Acceleration	Maximum 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 degree of curve at 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.
	• Angle of Attack	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.
	• L/V Ratio	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.
	• Maximum Roll Angle	Maximum roll angle 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 will spring compression, as a function of lading.	No friction snubber force measurements were made.
	• L/V Ratio	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.

5.2 TYPE I TESTING

5.2.1 Introduction

The survey of Phase I data showed that, while much useful data was acquired in Phase I, certain omissions in testing and measurement techniques required conducting a limited number of tests on the Type I truck to complete its characterization. Thus, the primary objective of the Type I truck testing was to provide truck performance data during curve negotiation. Secondary objectives were to provide data on load equalization performance, on rolling resistance for fuel consumption studies, and on wheel/rail forces during hunting.

Testing on the Type I truck was conducted on a 100-ton ASF Ride Control truck using new wheel profiles. This truck is the identical one used in the TDOP Phase I test program and in the TDOP Phase II wear data collection program. The carbody used for this program was a 100-ton open hopper car in both an unloaded and loaded configuration. Lading consisted of coal.

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 strain-gaged axle technique. The vertical forces at the bearing adapter were measured using the strain-gaged bearing adapters from TDOP Phase I. Forty-two of the 92 channels of data 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. 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. The test program was completed on March 13, 1980. Data acquired during the testing is presently undergoing reduction and analysis; test results are expected later in the summer.

5.2.2 Test Description

Test Zones. The test sites used for the Type I truck testing consisted of main line and yard tracks of the UP's South Central District, California Division. The four test zones selected for Type I truck testing are described in Table 5-3. Test zone 1 consisted of main line track with one to six degrees of curvature. Test zone 2 was a section of tangent, jointed track over which high speed (up to 79 mph) tests were conducted. Test zone 3, a section of yard track, was used for load equalization tests. Test zone 4 was a section of class 2 branch track over which roll and bounce tests were run.

Table 5-3. Type I Truck Test Zone Locations

Zone 1:	Sloan to Arden
Mileposts:	321.5 to 314 (7.5 miles)
Track Type:	Class 4 - curved
Rail Type:	133-lb jointed
Speed Limit:	40 mph
Zone 2	Arden to Boulder Junction
Mileposts:	321.5 to 326.5 (5.0 miles)
Track Type:	Class 4 - tangent
Rail Type:	133-lb jointed
Speed Limit:	79 mph
Zone 3	Las Vegas Yard
Mileposts:	Yard limits (0.22 miles)
Track Type:	12 and 16 degree curves
Rail Type:	Jointed
Speed Limit:	10 mph
Zone 4	Blue Diamond Spur
Mileposts:	4 to 8 (4 miles)
Track Type:	Class 2 - curved & tangent
Rail Type:	131-lb jointed
Speed Limit:	20 mph

ALD Placement. Wyle developed an ALD system so that test results obtained from a specific truck can 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.

Track Geometry Measurements. To be able to correlate response measurements made on test vehicles with a known track input, Ensco, under subcontract to FRA, measured the track geometry prior to starting FSFMS testing and again prior to the start of Type I truck testing. The first set of measurements was taken during the first week in November 1978 and the second set during November of 1979. Both tests utilized the T-6 Track Geometry Survey Car. The Wyle-developed ALD system was utilized during both track surveys. The final measurement will be made after completion of the Type II testing. This measurement will be used to determine if, over a period of time, there was a change in track geometry.

Test Train Consist. A standard test train consisting of a locomotive, instrumentation car, buffer car, test car, buffer car, and 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.

UP Mobile Laboratory Car 210. The UP Mobile Laboratory Car 210 was used as the instrumentation car for all testing on Type I 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 a 92-channel capability.

Carbodies. 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 and was readily available. Two carbodies from the same series were instrumented, one empty and one loaded. The trucks were then moved from the empty to the loaded carbody.

Trucks. The Type I truck selected for testing was the 100-ton ASF Ride Control truck. Prior to the start of Type I truck testing, the truck set was in revenue service as part of the TDOP Phase II wear data collection program (see paragraph 5.3). At the completion of testing, the truck set was returned to the wear data collection program. Prior to the start of instrumentation, all bearing seals of the two ASF Ride Control trucks were examined to verify that they were the same on all wheels.

Wheelsets. The new wheel profiles used for the Type I truck testing were AAR standard 1:20 taper profiles. The two axles were instrumented with strain gages for the B-end truck and were bored and prepared for the use of slip rings prior to the start of testing.

5.2.3 Instrumentation

The primary objective of the instrumentation was to obtain response measurements required to calculate the forces at the wheel/rail interface. In addition to the instrumentation required to measure the wheel/rail interface forces, transducers were also used to measure truck and carbody relative motion, rigid body car motion, coupler forces, and wheel/rail angle of attack.

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. 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 gages.

b. Instrumented bearing adapters from Phase I to measure vertical loads.

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

d. Slip rings for the rotating axle transducers.

Strain-Gaged Axle. Each axle on the B-end truck was instrumented with eight, full-bridge strain gages on each side of the axle. The strain gages were placed with one-half the bridge at the top and one-half the bridge at the bottom, as shown in Figure 5-1. Thus, there were 16 half bridge strain gages at $22\frac{1}{2}$ degree increments around the axle. The half bridges on opposite sides 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) has placed on each axle to define the strain gage position as a function of rotation angle.

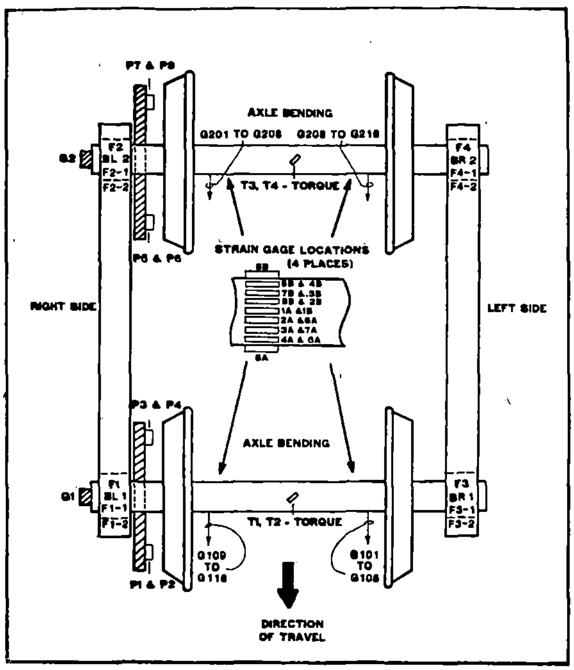


Figure 5-1. Wheel/Rail Measurement Instrumentation

Once the strain gage data was transmitted to Car 210, it was planned to multiplex the strain gages so that a vertical and longitudinal bending moment would be obtained at each side of the axle. This would result in a total of eight bending moments being recorded. After the initial set of runs using the multiplexer, it was found that the data recorded using this technique was unsatisfactory for the required data reduction. The patch panel was then reconfigured to record all 32 strain gage channels and software was developed to multiplex the data digitally during data reduction.

In addition to the 32 strain gage channels, the two rotary pulse generator channels will be recorded. The torque in each axle (T1 to T4) will be measured using two strain gages at the middle of each axle. These measurements can be used to estimate longitudinal creep forces.

Instrumented Bearing Adapter. Vertical loads were measured by means of a modified strain gaged roller bearing adapter developed by SPTCo. during Phase I and shown in Figure 5-2. The bearing surfaces on top of the adapter were partly machined off so as to leave two narrow bands to support the side frame. A deep groove was machined into the cylindrical surface so that it would mate with the roller bearing, and the top of the resulting thin section was instrumented with strain gages.

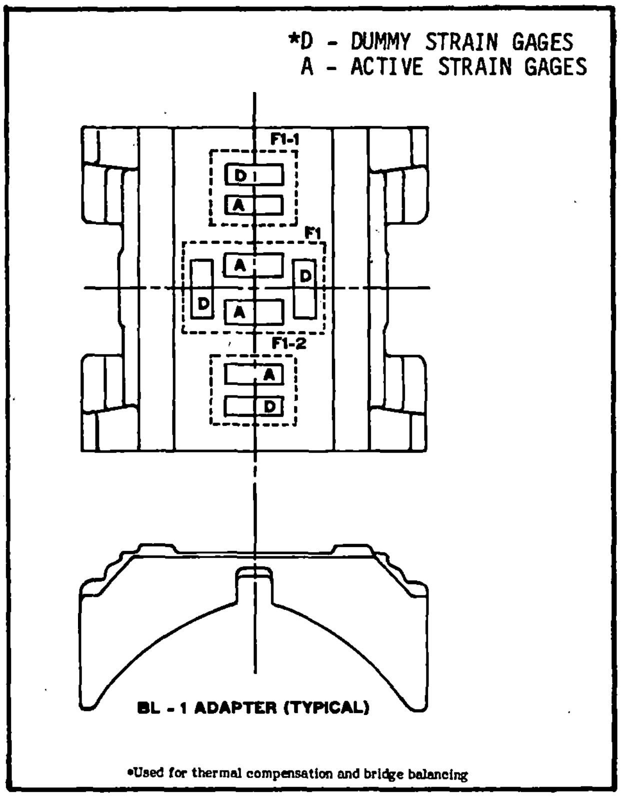


Figure 5-2. Force Transducer - Bearing Adapter

Prior to using the Phase I strain-gaged bearing adapters on Phase II, a complete calibration was performed on the adapters to determine linearity and the amount of error which is induced in the measurements by lateral shifts in the point of the vertical load. The results of the preliminary calibration tests showed the measured vertical load to be very strongly affected by the lateral shift of the load point. To correct for this, two additional gages (F1-1 and F1-2) were added to each bearing adapter as shown in Figure 5-2. The calibration series was then rerun recording all three gages. From this data it is possible to obtain a corrected estimate of the vertical load and the line of action of that vertical load.

Wheel/Rail Position Measurement. Four eddy current transducers were used at one end of each axle to measure the relative position and angle of attack of the axle relative to the rail. This concept is shown schematically in Figure 5-3. 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 5-4).

Optical Data Transmission System. Several techniques were evaluated for transmitting the strain gage data from the rotating axle. These techniques were slip rings, FM telemetry, and optics. The FM telemetry technique was eliminated early in the program because of its significantly higher cost and greater vulnerability to outside interference than the other two systems. After including the cost of on-the-axle signal conditioning, the slip ring system and optical data transmission system had very similar costs. The optical data transmission system was finally decided upon for the following reasons:

a. Ability to transmit a greater number of data channels.

b. Better quality of transmitted signal, and less vulnerability to interference.

c. Optical transmission is serialized, requiring only one transmission line per axle to Car 210.

d. No requirement to drill axles, thereby limiting its future use.

Laboratory testing of the optical data transmission system demonstrated the device to work satisfactorily. However, during field use the reliability of the optical transmission portion of the system degraded to the point that it was unacceptable for the required data. Hence, the backup plan to transmit the data from the axle using slip rings was implemented. Two sets of 36 channel slip rings were installed on each axle. The signal conditioning portion of the optical data transmission system continued to be used; auxiliary data ports built into the system were used to transmit the data to the slip ring and to the test car. The drilling of the axles to accommodate slip rings was accomplished in Las Vegas.

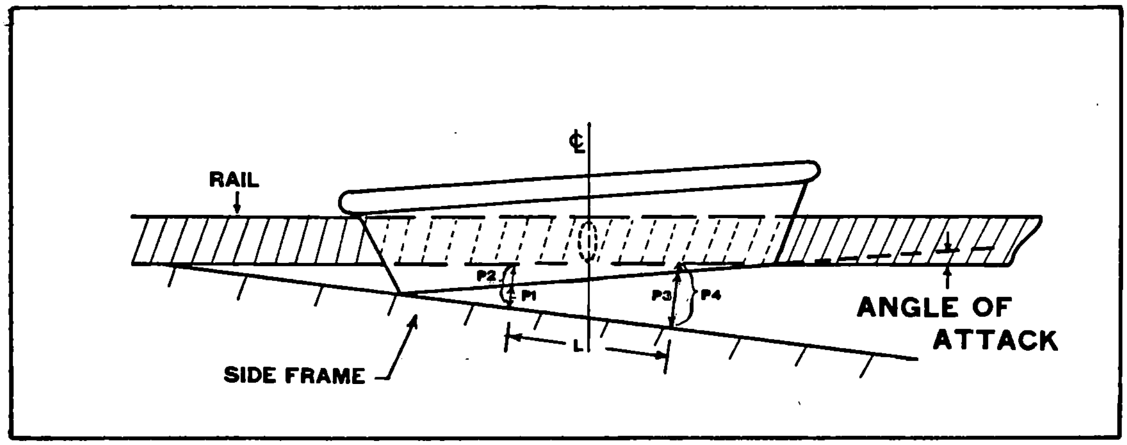


Figure 5-3. Wheel/Rail Position Measurement (BL-1 Axle)

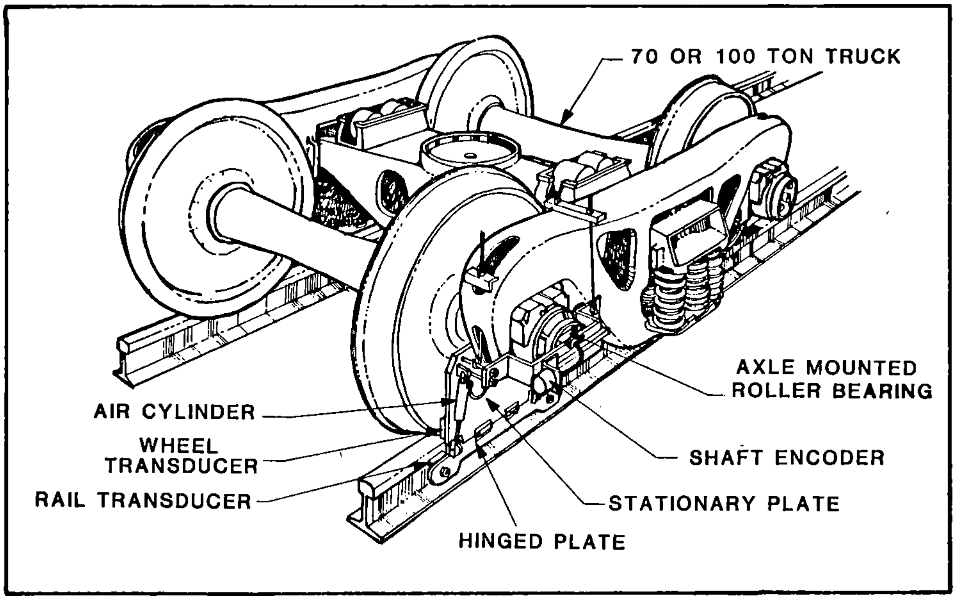


Figure 5-4. Wheel to Rail Displacement Measurement

Carbody and Truck Measurements. Displacement transducers and accelerometers were used on the truck and carbody to measure rigid body modes and relative displacements. In order to locate transducers on the carbody and trucks, the AAR standard for component location shown in Figure 5-5 was used. This enabled the exact location of a transducer to be specified and maintained continuity with the wear data collection program.

Forty-two transducers were installed on the carbody and truck. Two transducers provided train speed and position. Longitudinal coupler force and the angle of the couplers at both ends of the test car were measured by four transducers.

Fourteen transducers measured truck side frame to truck bolster, and truck bolster to carbody bolster relative motions. Eight accelerometers gave the six rigid body motions of the carbody. Three were used to determine the roll center of the carbody. Five accelerometers were on the bearing pockets. Lateral accelerometers were placed on all four axles, two on the A-end truck. These accelerometers were used as an indicator of critical speed for hunting. This was the only instrumentation used for the A-end truck. The two vertical accelerometers on each end of the forward axle of the B-end truck provided a measure of truck input to the truck.

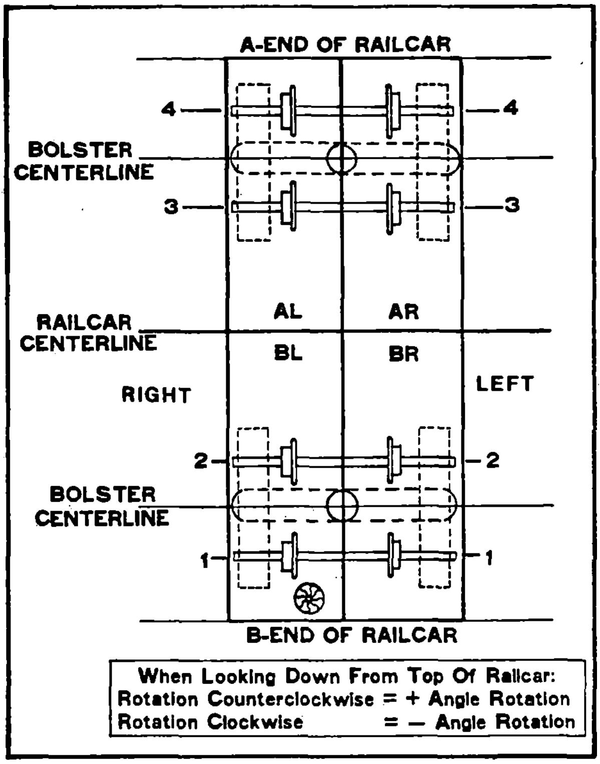


Figure 5-5. AAR Standard for Component Location

Instrumented Couplers. A pair of instrumented couplers used during the AERO-TOFC-II Program were modified to measure longitudinal draw bar forces. These couplers each consist of a pair of concentric cylinders connected by a load cell and belleville springs in series. Thus the couplers only measured forces along the axis of the coupler. Overload stops prevented the forces from exceeding the capability of the load cell. A schematic of the coupler is shown in Figure 5-6.

A coupler angle measurement was made so that the coupler forces could be broken into components along and vertical to the carbody axis. This measurement was accomplished by attaching two bending beam transducers from the carbody to the coupler. The bending beams were positioned in such a way that longitudinal translational motion can be eliminated from the summing of the two measurements, leaving only lateral motion. From the measured distance out from the carbody to the bending beam attach point, it was possible to calculate coupler rotation angle.

5.2.4 Test Operations

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 main line 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 be conducted at below and above equilibrium speed.

Secondary objectives were to acquire hunting, rolling resistance, and load equalization 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 was 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 (see Tables 5-4 and 5-5).

5.2.5 Data Acquisition, Reduction and Analysis

Data Acquisition. Data from this test program was acquired aboard Mobile Laboratory Car 210. Using the on-board computer and Wyle-developed data acquisition software, the analog signals from the transducers were preprocessed, digitized, and recorded on magnetic tape. In some cases, the preprocessing involved only signal conditioning. In other cases, analog circuitry was to be used to combine channels to obtain a measured value. The data acquisiton software provided a means of: cataloging and updating information files describing the test, organizing and writing the digital information on tape, and previewing the data written on the tape at test completion.

Quick Look Data Review. Data was recorded on a six-channel Brush Recorder to provide a real time check on the quality of the data from selected channel. The real time data consisted of time history traces of selected channels for the full run duration. Immediately at the completion of each test, a series of "quick look" data reduction runs were made to access the quality of the digitized data. They consisted of playing the magnetic tapes back and obtaining a five-second time history at selected times of each test run for all 92 channels, displayed four at a time.

Data Reduction and Analysis. After the quality of the digital tapes were reviewed using the "quick look" data techniques, the digital tapes were shipped to Wyle Colorado Springs for data reduction and analysis. The digital data tapes were then processed through a computer program which demultiplexes the test data and produces a second tape in a format compatible with Wyle's Advanced Data Analysis and Reduction System (ADARS). ADARS provides a generalized data reduction capability from which the analyzed data can easily be obtained. Figure 5-7 illustrates the various analysis functions and data interfaces within ADARS. The analysis and display capability within ADARS include Spectrum Analysis (SPEC), Statistics (STAT), Display Function (DISP), and Data Preparation (PREP).

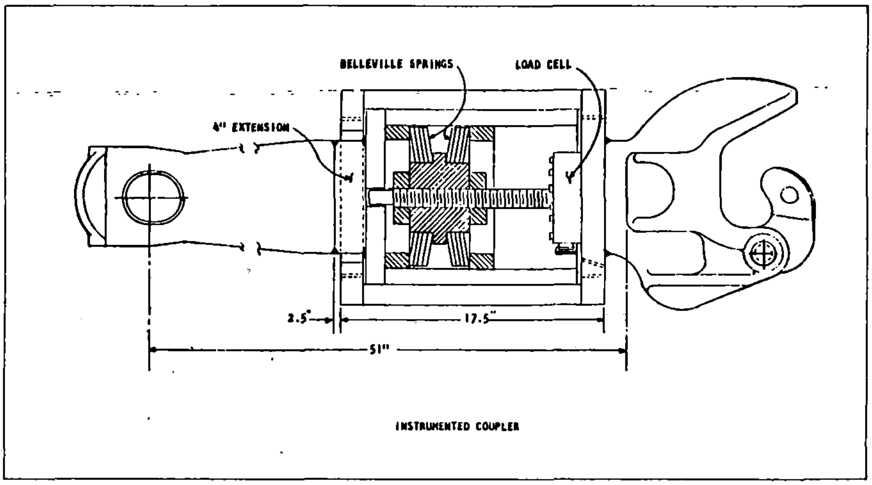


Figure 5-6. Instrumented Coupler Schematic

Table 5-4. Type I Testing - Empty Hopper Car

RUN NO.	TEST TYPE	ZONE	GRADE	SPEED
1A	Trackability	4	+1.5%	4-30-4 mph
2A	Curve Negotiation	1	+1.0%	Under Equilibrium
3A	Curve Negotiation	1	+1.0%	Equilibrium
4C	Curve Negotiation	1	+1.0%	Over Equilibrium
5B	Curve Negotiation	1	-1.0%	Equilibrium
6	Lateral Stability	2	-0.9%	40-79 mph
7	Lateral Stability	2	-0.9%	79-40 mph
8A	Trackability	3	Zero	10 mph

Table 5-5. Type I Testing - Loaded Hopper Car

RUN NO.	TEST TYPE	ZONE	GRADE	SPEED
9	Trackability	4	+1.5%	4-30 mph
10	Curve Negotiation	1	+1.0%	Under Equilibrium
11	Curve Negotiation	1	+1.0%	Equilibrium
12	Curve Negotiation	1	+1.0%	Over Equilibrium
13	Curve Negotiation	1	-1.0%	Equilibrum
14	Lateral Stability	2	-0.9%	40-79 mph
15	Lateral Stability	2	-0.9%	79-40 mph
16	Trackability	3	Zero	10 mph

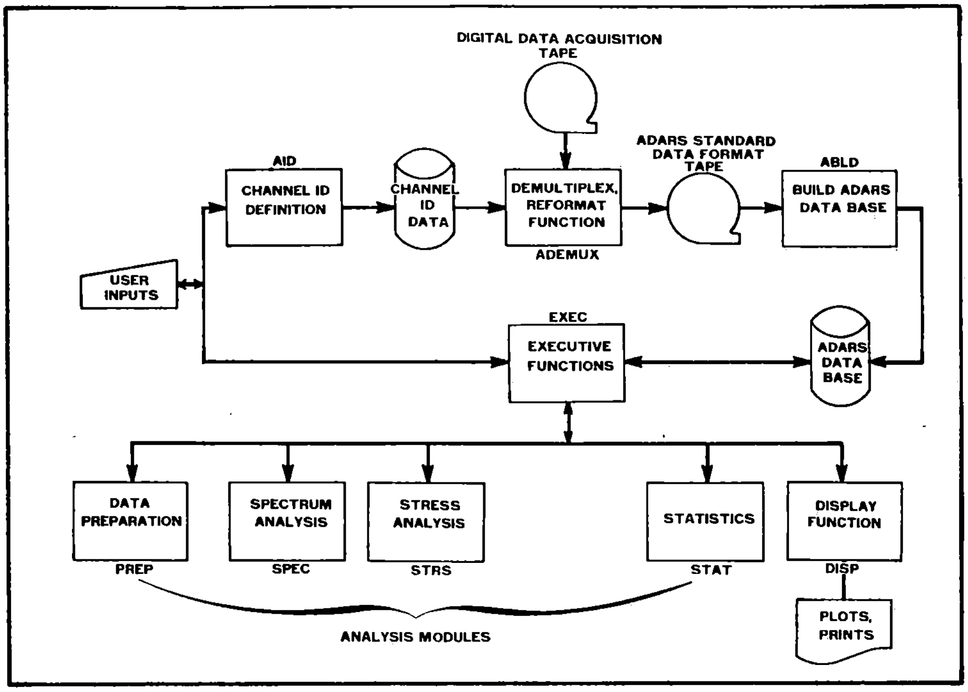


Figure 5-7. ADARS Overview Flow Diagrams

SPEC's function is to perform Power Spectral Density analysis, using a Fast Fourier Transform (FFT) routine with a sine and cosine look-up table to enhance processing speed. The module includes the following capabilities:

a. Power Spectral Density (PSD)

b. Cross Power Spectral Density (CSD)

c. Transfer function gain and phase

d. Hanning window on time arrays

e. Averaging of PSD and CSD values

f. Printout of averages, degrees of freedom (or confidence level) with PSD plots.

Statistics (STAT) performs the following statistical analysis functions for a given input file:

a. Computes and writes to line printer the average, standard deviation, skewness, and kurtosis.

b. Computes the root mean square and the mean squared data values.

Display Functon (DISP) provides both plots and printouts for the ADARS user. DISP can be initiated by the user directly, or by an analysis module which can automatically initiate DISP upon completion of its execution. Any data output by PREP or any analysis module can be processed. The following types of plots are produced:

a. Time History Plot. Linear X, linear Y axes for raw or analyzed data, single channel graph per grid, single grid per page. Also available are Power Spectral Density data plotted against frequency.

b. Strip Chart Plot. Analyzed data time history plot over linear X axis time frame and grid, eight linear Y axis grids per page with one channel graph per grid for each of the eight grids.

Data Reduction. After the test data is stored on disk, ADARS is used to calculate those parameters shown in Table 5-6. The coupler force measurements, coupler angle, vehicle speed, and vehicle longitudinal acceleration will be used to calculate average vehicle rolling resistance. The equations to be used in this calculation are listed in the table.

Table 5-6. Data Reduction Parameters

Carbody Acceleration	
Bounce	(A1 + A2)/2
Pitch	(A1-A2)/L
Yaw	(A6-A5)/L
Roll	(A3-A1+A4-A2)/2g
Sway	(A6+A5)/2
Longitudinal	A8
Center of Rotation	$g\text{-}h\ \left(\frac{A7}{A3+A4}\right)$
Sideframe/Bolster Relative Motion (Right Sideframe)	
Lateral Displacement	(D5+D6)/2
Vertical Displacement	(D1+D2)/2
Pitch Rotation	(D1-D2)/a
Yaw Rotation	(D6-D5/b
Roll Rotation	$\frac{D7\text{-}D6}{d}$

Table 5-6. Data Reduction Parameters (Cont'd)

Truck/Carbody Relative Displacements	
Truck Swivel Angle	(D14-D13)/e
Lateral Displacement	(D14+D13)/2
Roll Angle	(D11+D12)/2f
Rail/Wheel Position	
Rail to Sideframe Angle	(B-D)/L
Wheel to Sideframe Angle	(A-C)/L
Angle of Attack	(B-D-A+C)/L
Sideframe to Rail Distance	(B+D)/2
Sideframe to Wheel Distance	(A+C)/2
Wheel/Rail Force Measurements	
Vertical Force	See Note
Lateral Force	See Note
L/V	Calculated by Ratio of Lateral Force/Vertical Force
Rolling Resistance Calculation	
Average speed (ft/sec)	$\bar{V} = \frac{1.4667}{n} \sum_{1}^{n} S_1$
Energy (ft/lb)	$E = 1.4667\ T \sum_{1}^{n} S_1 (C_1 \cos C_2 - C_3 \cos C_4)$
Average rolling resistance (lb)	$\bar{R} = E/(n T \bar{V})$
Average grade (%)	$\bar{g} = \frac{.04555}{nT} (S_1 (nT) - S_1 (o)) - \frac{1}{n} \sum_{1}^{n} A_8$

Note: These relatively complex equations are discussed in Wyle TDOP Phase II Technical Note, "Measurement of Wheel/Rail Forces in TDOP Phase II Field Testing," dated February 23, 1979.

5.3 WEAR DATA COLLECTION PROGRAM

The TDOP Phase II wear data collection program will collect wear data on several Type I and Type II freight car trucks. The objectives of the program are to collect sample wear data, establish wear trends, evaluate wear measurement methods, develop a schedule for measurement occurrence, and provide data for economic models.

Six truck types (three Type I and three Type II) have been in revenue service since April 1979. A fourth Type II truck was added to the wear program in January, 1980. The trucks are run in two unit coal trains making a 1600-mile roundtrip every four days between a Colorado coal mine and the Kaiser steel plant in Southern California. These trains operate on the Union Pacific, Atchison, Topeka and Santa Fe, and Denver and Rio Grande Western railroads. The trains pass through Las Vegas, Nevada, and are easily accessible to Union Pacific's Repair In Place (RIP) track. Most of the service will be on Union Pacific's main line class 4 track, with curves of up to 10 degrees.

5.3.1 Trucks Selected

The seven trucks undergoing test are the American Steel Foundry (ASF) Ride Control truck, the Barber S-2 truck, the Barber S-2-HD Center Plate Extension Pad (C-PEP) truck, the Dresser DR-1 Steering Assembly truck, the Standard Car Radial Barber-Scheffel truck, the National Swing Motion truck, and the Devine-Scales truck.

The ASF Ride Control and the Barber S-2 will provide representative data on the Type I trucks. The Barber C-PEP represents a modified 100-ton truck, and is considered a Type II truck. This truck along with the Dresser DR-1, Barber-Scheffel, National Swing Motion and Devine-Scales trucks will provide wear information for Type II trucks.

5.3.2 Pre-Service Preparation

Before any of the trucks were placed into service, a number of essential tasks had to be performed. First, approvals had to be obtained from the three responsible railroads before acceptance of any truck type not previously approved for interchange service. Cars were selected from the Union Pacific's fleet; they were all 100-ton, self-clearing hopper cars and each had to have a remote retainer valve and separate retainer air line in order to be included in the program. The center plate on each carbody was inspected and measured. Any center plate that showed an "out-of-circular" condition was cause for rejection of the car, as was evidence of damage such as may have arisen from a broken vertical wear ring on the truck bolster.

The cars in the wear program were identified with a placard in each corner. The words "test truck" were stenciled on each side frame and bolster, and the words "test wheels" were stenciled in at least four locations on each axle. Additionally, some truck components, such as friction castings, pedestal roofs, and center plate wear liners had to have indexing features designed for them to ensure accurate wear measurements.

Next, the initial, or zero-mileage measurements were taken on the trucks. Over 6,800 individual measurements in all, or about 1000 per truck. Truck components that were measured included wheels, brakes, bearing adapters, bolsters, and side frames. Measurement techniques included the use of ultrasonic thickness gages, micrometers, calipers, and depth gages with indexing features. These inventory measurements were placed in the software data base. The trucks then were reassembled and placed into service.

5.3.3 Wear Measurement Plan

The trucks have been stopped and inspected for wear at the RIP track on the dates shown in Table 5-7. The number of miles that each car has accumulated is shown in Table 5-8. The measurements taken include wheel profiles, brake shoe wear, wear at the bearing adapters and pedestals, wear of the friction snubbers, bolster pockets, gibs, columns, center plate, and side bearings. The trucks are also inspected for any damage or changes in configuration. The wear data collection program began in January, 1979, and will conclude in October, 1980. Although mileage will vary, several of the trucks will have accumulated over 125,000 miles.

Table 5-7. Wear Measurement Occurrences

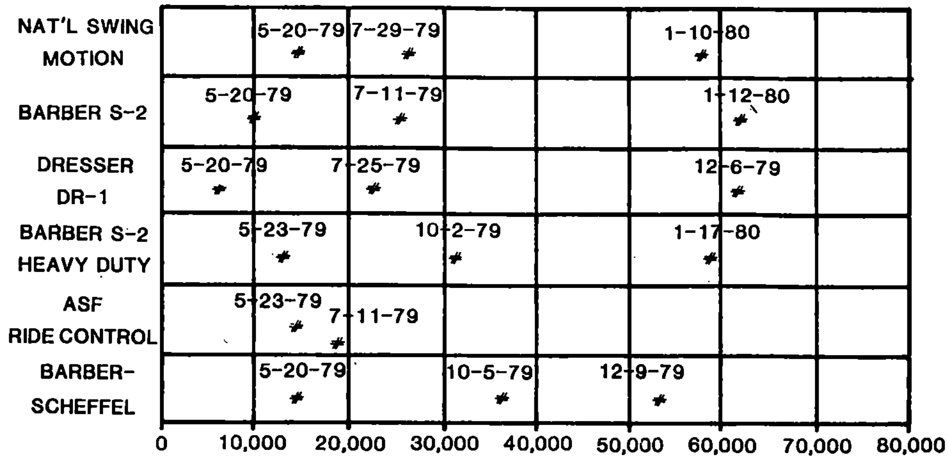


Table 5-8. Wear Data Collection Mileage Status - April 18, 1980

TRUCK	TDOP CAR NO.	UNION PAC. CAR NO.	CAR SERIES	DATE IN SERVICE	MILEAGE TO DATE
Nat'l Swing Motion	001	38192	H-100-11	4-4-79	80.057
Barber S-2	002	37708	H-100-11	4-8-79	81,586
Dresser DR-1	003	38051	H-100-11	4-8-79	51,861*
Barber S-2 Heavy Duty W/C-PEP	004	38497	H-100-12	4-4-79	83,880
ASF Ride Control	005	38-080	H-100-11	4-4-79	22,000**
Barber-Scheffel	006	38243	H-100-12	4-4-79	87,617
Devine-Scales	007	37499	H-100-11	1-16-80	18,720

*Reading taken 12-07-79; out of service for Type II testing
**Used for Type I testing; re-entered service on April 11, 1980

5.3.4 Measurement Techniques

The program's measurement techniques have been developed through evaluation of existing railroad and industry measurement techniques as well as those being used on the Facility for Accelerated Service Testing (FAST) Program at the Transportation Test Center in Pueblo, Colorado. Initial verification sampling at close mileage intervals were made to ensure that baseline data on early wear was preserved. The wear data has been entered into the TDOP data base and will be compared by engineering and economic analysts with wear data from other data sources (e.g., industry, AAR, and the FAST program).

An example of the measurements taken is the truck bolster center plate liner and wear ring, accomplished with a template and an ultrasonic gage. Eight wear ring thicknesses and twelve wear liner thicknesses are measured along the bolster longitudinal and lateral axes using the location template. The minimum thickness of the wear ring is located and this defines the major wear axis. The angle is measured using the pointer and protractor disc on the template. The template is then rotated to align with the major wear axis and the eight ring and twelve liner thickness measurements are made with the ultrasonic gage.

5.3.5 Wear Data Base

The TDOP wear data base is designed to collect data from multiple sources by truck type, manufacturer, load capacity, operational classifications, curve-to-tangent ratios, mechanical wear, and repair and maintenance costs. Data from participating railroads, the TDOP wear data collection program, and the FAST program will be utilized. The data base will be developed by means of Wyle Laboratories' Interdata 8/32 computer using TOTAL as a data base management system. This system is ideal for the wear data base in that it allows for both fixed data files (master files) and variable files. Thus, fixed information such as truck manufacturer, type, and nominal physical characteristics and specifications only need be recorded once in the data base. Variable data such as serial numbers of individual truck components, operating profiles, track characteristic measurements, maintenance labor hours, component replacements, wear measurements, etc., can be entered into the variable data files as appropriate.

5.3.6 Wear Data Collection Report

At the completion of the TDOP wear data collection program, a report will be released which will describe the program, and its collected data and data base. It will also provide selected plots of wear measurements and significant results. The contents of the data base will be recorded on computer magnetic tape and submitted to the National Technical Information Service for public access.

5.4 FRICTION SNUBBER FORCE MEASUREMENT SYSTEM

A Friction Snubber Force Measurement System (FSFMS) was developed and shop-tested during TDOP Phase I. The primary objective of this system was to measure friction coefficients and forces transmitted between the friction shoes and the wear plate of conventional freight car trucks. During Phase II of TDOP, a series of over-the-road tests was run to obtain friction snubber data in actual railroad operation.

The FSFMS was installed on both an ASF Ride Control and Barber S-2 70-ton truck. During November and December of 1978, these trucks were run through a series of tests in various load conditions over sections of Union Pacific track near Las Vegas. In addition to the instrumentation for measuring friction snubber forces, transducers were installed on the trucks and carbody to measure relative motion between carbody and truck, and carbody rigid modes.

Results from the data analysis showed the Barber truck to have a coefficient of dynamic friction between .31 and .36 while the dynamic friction coefficient of the ASF truck was between .37 and .49. The only strong correlation between relative motion in the truck and friction forces occurred in the vertical motion of the side frame relative to the truck. As the vertical motions increased, the variation in the friction forces increased.

The friction forces obtained as a result of this test program could be applied to other work, for example, as input to analytical models, to validate roll and bounce models, and as considerations in truck design. Only those data required to meet the objectives of the program were analyzed for the FSFMS test report. There remains a significant amount of analysis information which may still be extracted from the data. Some of the areas in which additional work is recommended are:

- Determining friction coefficients in curves to see if they differ from those for tangent track.
- Completing the analysis of center plate kinetic friction coefficient.
- Exploring the relationship for the half loaded to the empty and loaded car configurations.
- Determining the relationship of braking to friction forces.
- Investigating the effect of asymmetric column loading on snubber friction.

SECTION 6 – PERFORMANCE CHARACTERIZATION OF TYPE I TRUCKS

6.1 INTRODUCTION

One of the principal tasks of TDOP Phase II is to characterize the performance of the Type I truck. This performance characterization of the Type I truck will form the baseline for the ongoing evaluation and assessment of the response characteristics of the Type II truck so that the effects of design innovations on truck performance and the consequent benefits to operating railroads can be objectively assessed.

The procedure employed in developing this characterization was to establish a methodology, acquire and reduce test data on the Type I truck, perform analytical simulation through the use of mathematical models, analyze and interpret the data, and quantify the performance characteristics. The results of such a procedure yield a performance characterization for the freight car truck. The purpose of this section is to describe the work-in-progress of this performance characterization. Data used in the analysis was derived from TDOP Phase I and from American Steel Foundries tests; none of the data acquired during Phase II testing of the Type I truck has been included at the time of the publication of this Interim Report.

6.2 METHODOLOGY

As discussed in Section 2, the methodology that TDOP Phase II adopted to develop the Type I truck performance characterization included these steps:

Definition of Truck Performance. The characterization of performance requires the identification of specific performance regimes which may be defined as sets of conditions associated with predominant features that distinguish one regime from another. In order that performance be quantified, performance indices associated with each of the performance regimes have been identified. The characterization of performance is represented by a range of quantified performance indices within each regime and associated with a specified set of operating conditions such as speed, lading, and track quality. The defined performance regimes and associated performance indices are given below:

Performance Regime	Performance Index
Lateral Stability	Critical speed maximum lateral acceleration
Curve Negotiation (Steady State)	Lateral force at wheel/rail/interface L/V ratio Angle of attack
Trackability (Harmonic Roll only)	Critical speed Maximum roll angle
Ride Quality	Transmissibility

Data Acquisition. Identification of the performance indices led directly to defining the requirements for test data which in turn will permit the quantification of the indices. In general, at least a part of these data is usually available from existing sources. In the case of TDOP, extensive field tests were conducted in Phase I and performance test data acquired for Type I trucks. This body of test data was evaluated in the light of the data requirements. The evaluation of existing data resulted in the identification of additional test requirements. Thus, additional field tests of the Type I truck were conducted in Phase II to supplement the Phase I data (see Section 5).

Data Reduction and Analysis. Following established procedures, the data were reduced, verified for acceptability, and analyzed to provide quantitative measures for the performance indices. Where field test data were unavailable, simulated data were utilized. Where unusual or abnormal traits were discovered in the data, verified and/or validated mathematical models were used in explaining such traits and interpreting the results. Finally, a set of quantified performance indices, each associated with a specific set of operating conditions, have been produced.

Development of Performance Characterization. The quantified indices of performance are next interpreted to form performance characterizations within each of the performance regimes. The quantified indices were related to the specific operating conditions under which they were obtained, with due regard to state of wear or other deterioration associated with age or ton/mileage of the truck. The range of quantified performance indices for each Type I truck performance regime under specified operating conditions then comprise a characterization of performance and are represented by a set of characteristic plots of the performance indices. A flow diagram of the entire process is shown in Figure 6-1.

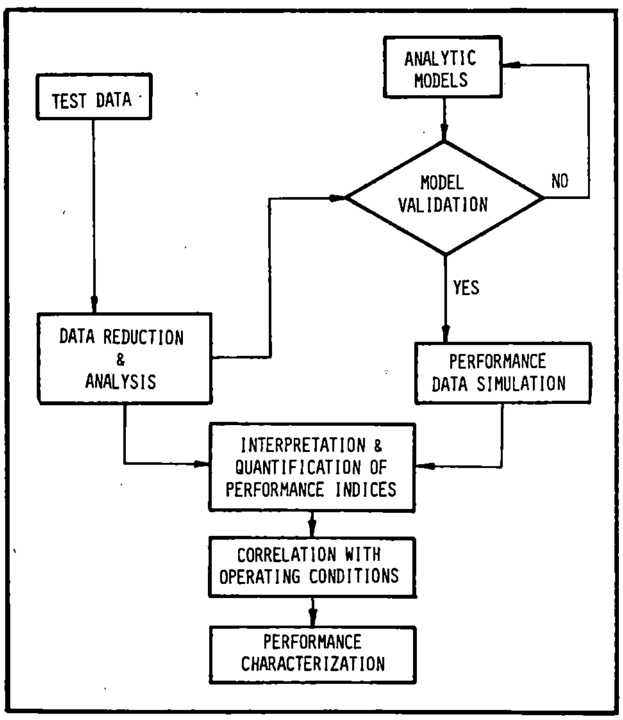


Figure 6-1. Truck Performance Characterization

6.3 FIELD TEST AND DATA ACQUISITION

The primary basis for the characterization of Type I truck performance is the test data acquired during TDOP Phase I. The Phase I instrumentation consisted of displacement transducers, accelerometers, and force transducers at different points on the truck and carbody. In addition, track geometry data were acquired to correlate vehicle response to track input. The Phase I test consist included a locomotive, the instrument car, the test car, and a caboose in that order. The test track included high speed jointed and continuous welded track, medium speed jointed track, curved track ranging in curvature from one to nine degrees, and a modified track for rock and roll tests.

To supplement the data acquired during Phase II, the recently concluded Phase II test program of Type I trucks was implemented to provide the required test data. Tests performed during Phase II were primarily intended to provide additional data on the forces at the wheel/rail interface. A vehicle-borne instrumentation system was developed to provide continuous measurement of the lateral and vertical forces at the wheel/rail interface; this instrumentation package utilized the axle-bending technique. Also implemented was a system to provide continuous reading of the angle of attack that the wheel makes with the rail as the vehicle traverses a curved track. The tests were run using an instrumented 100-ton, open hopper car. The test track consisted of medium and high speed jointed tangent and curved track ranging in curvature from two to six degrees; low speed test runs were made in yards to obtain data on trackability.

6.4 ANALYTIC SIMULATIONS

TDOP Phase II views analytic techniques as extremely valuable tools in interpreting and extending the results from field test data. A survey was conducted of existing models and computer programs which had been used in other railroad research and development projects and an effort was made to validate these models against a set of criteria (see Section 4).

The analytic efforts proved to be of great value in interpreting various phenomena encountered during data reduction, especially in such areas at the influence of wheel/rail contact geometry on hunting, and the progression of the development of hunting from the "nosing" mode to one of steady hunting. In the regime of curve negotiation, modeling was a valuable tool. The test data used in the characterization of curving performance contained information on the lateral and vertical forces at the wheel/rail interface, but not on the angle of attack. The model used in the curve negotiation regime was validated against these data as well as data from other sources and the simulated results used to quantify the angle of attack, thus complementing the wheel/rail forces obtained from test data under similar conditions. Test data have been relied upon entirely in the characterization of performance in the ride quality regime.

6.5 DATA ANALYSIS AND PERFORMANCE CHARACTERIZATION

Leading up to the quantitative characterization of Type I truck performance, field test data were reduced, analyzed, and interpreted following a general methodology outlined below:

- Review time history data on accelerations at various locations on the carbody and truck, relative motions between components, and forces on components.
- Select appropriate segments of the raw data for use in each of the performance regimes.
- Perform power spectral density analysis on selected data, where appropriate.
- Calculate rms levels for selected indices.
- Calculate transfer functions relating vehicle response to track input, where appropriate.
- Extract peak values for selected indices from time history data.
- Determine statistical significance of quantified indices, where applicable.

Data analysis within each performance regime is treated individually in the paragraphs below.

6.5.1 Lateral Stability

The lateral acceleration data on the carbody at the sill level and the roof level as well as on the truck axles were considered. In addition, the data were examined to determine distinguishing characteristics of the response at the leading and trailing ends with respect to one another. Also, having determined from the data that the response of the wheelsets followed along the response at the carbody sill level, the carbody response was chosen to characterize lateral stability performance. The carbody roof level response was studied relative to the sill level response and the characterizations arrived at represent the enveloping levels, i.e., worst case response. The frequency range of analysis was 0 to 20 Hz. Each power spectrum in the range of 0 to 5 Hz was scanned and the peak value of the spectrum selected; centered around this frequency the rms acceleration was calculated for a frequency bandwidth of 1 Hz.

i.e. the hunting encountered was body rather than truck.

Analytic means were relied on in the interpretation of some aspects of the test data, especially pertaining to the influence of wheel profiles and the predominant evidence of the "nosing" (the leading end of the carbody undergoing violent lateral motions with respect to the trailing end) phenomena preceding fully developed hunting in most of the cases. Some of the resulting characterization of performance for Type I trucks in the lateral stability regime is given in Figures 6-2 and 6-3.

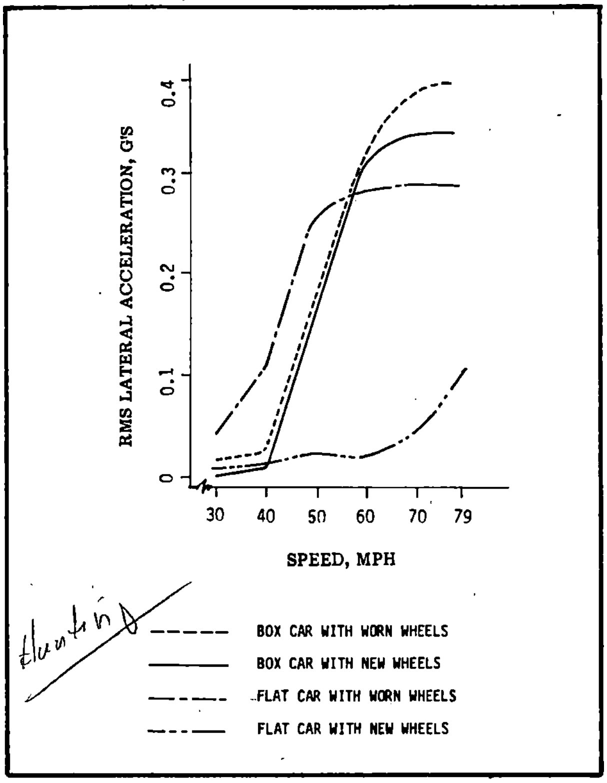


Figure 6-2. RMS Lateral Acceleration vs Speed for 70-ton, Type I trucks, Empty Cars

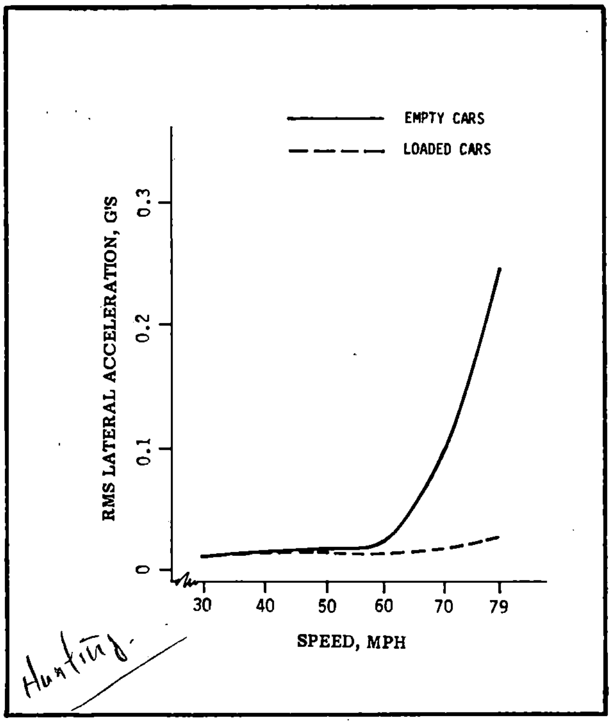


Figure 6-3. RMS Lateral Acceleration vs Speed for 100-ton, Type I Trucks, Box-Type Cars, New Wheels

6.5.2 Trackability

Although this regime has been defined to include harmonic roll, load equalization, and curve entry/exit, in terms of available field test data only the performance of harmonic roll has been characterized. Peformance relating to the subsets of load equalization and curve entry/exit will be reported on later. With respect to harmonic roll, test data available cover 70-ton Type I trucks with a mechanical refrigerator car and 100-ton Type I trucks with box cars, both using cylindrical wheels, tested on track modified according to an AAR specification for testing special devices to control stability of freight cars (see AAR Manual of Standard and Recommended Practices, 1974). Roll angle of the carbody relative to the side frame, and carbody roll accelerations have been quantified from the data. The results are presented in Figures 6-4 and 6-5.

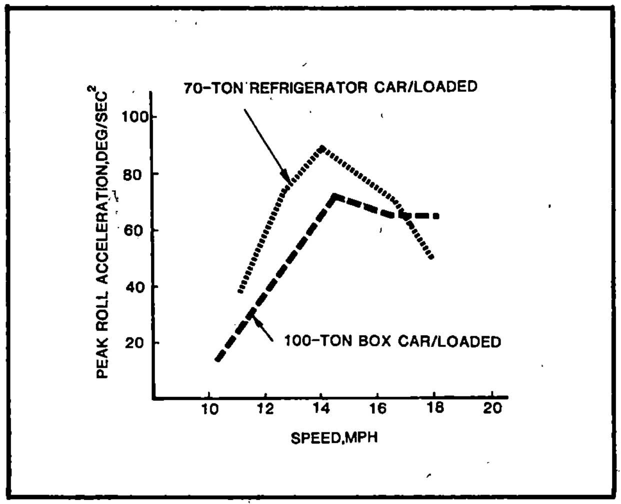


Figure 6-4. Peak Roll Acceleration vs Speed, Type I Trucks, Box-type Cars

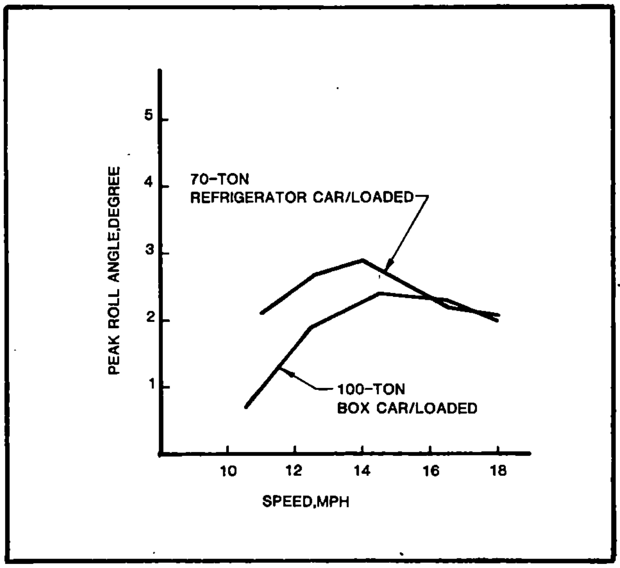


Figure 6-5. Peak Roll Angle (Between Carbody and Side Frame) vs Speed, Type I Trucks Box-type Cars

6.5.3 Curve Negotiation

100-ton, Type I trucks with loaded open hopper cars tested over curved track ranging in curvature from 1.5 to 7.5 degrees at varying speeds provided the performance test data in this regime. The lateral forces at the wheel/rail interface were obtained through the axle-bending technique of measurements which consists of strain gage instrumentation on the truck axles. Vertical forces were measured by strain gaging the side frames. In quantifying the lateral forces at the wheel/rail interface, the algebraic average, the root mean square value, and the peak value have been calculated and a description of their functional relationships to operational variables such as speed and degree of track curvature has been provided.

Using the calculated lateral and vertical forces obtained from the test data, the lateral-to-vertical (L/V) force ratio has been quantified for all four wheels in a truck. The test instrumentation did not consist of angle of attack measurements. However, analytic simulations utilizing a steady state non-linear curving model have been used in quantifying the angle of attack performance index. The model was verified against the test data for validity by using the lateral and vertical force measurements as the criteria for comparison. The results in this performance regime are shown in Figures 6-6 through 6-8.

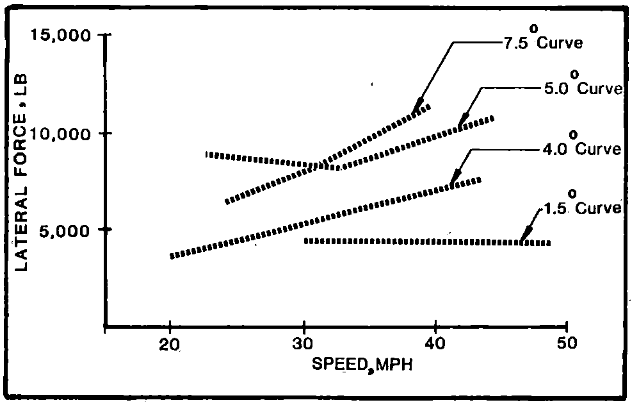


Figure 6-6. Peak Roll Angle (Between Carbody and Side Frame) vs Speed for Type I Trucks, Box-type Cars

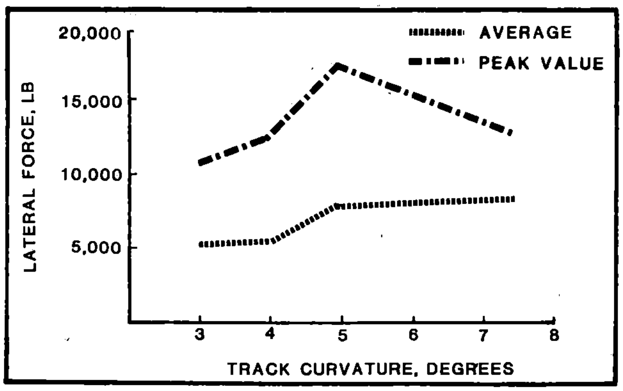


Figure 6-7. Lateral Force on Leading Outer Wheel vs Degree of Curvature 100-ton, Type I Trucks at Balance Speed

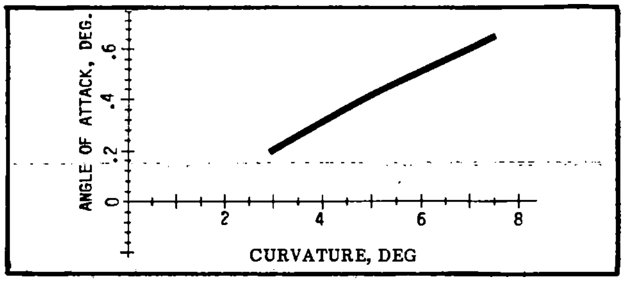


Figure 6-8. Angle of Attack of Leading Outer Wheel vs Track Curvature for 100-ton, Type I Trucks, Loaded at Balance Speed

6.5.4 Ride Quality

Ride quality, as a performance regime, refers to the acceleration environment in the carbody reflecting the capability of the truck to attenuate the excitation arising from track irregularities. This attenuating characteristic of the truck has been termed transmissibility. Extreme performance phenomena, such as resonance and other unstable conditions, have been excluded from consideration in this regime since they have been considered part of the lateral stability and trackability regimes. Test data is available covering both the 70- and 100-ton Type I trucks in the empty and loaded conditions over a wide variety of track and speed conditions. Track geometry data have been acquired from the Department of Transportation and include such track parameters as profile, alignment, gage, and cross level, as well as automatic location detection along the track.

Characterization of ride quality performance has been accomplished through calculation of the transmissibility in both the vertical and the roll modes, relating carbody response at the sill level to the track input. This process included power spectral density analysis, calculation of rms levels of vertical and roll accelerations evaluated for selected frequency bandwidths, and quantification of transmissibility by means of the ratio of rms response to the corresponding rms track input. The results from the data analysis are given in Figures 6-9 through 6-11.

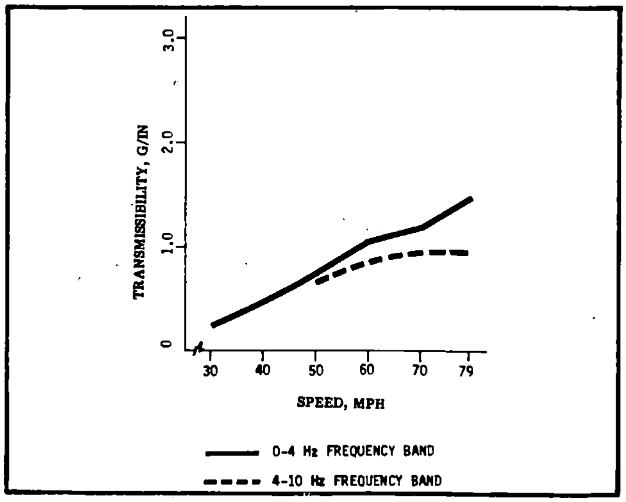


Figure 6-9. Transmissibility - Vertical Acceleration for 70-ton, Type I Trucks, Loaded Flat Cars

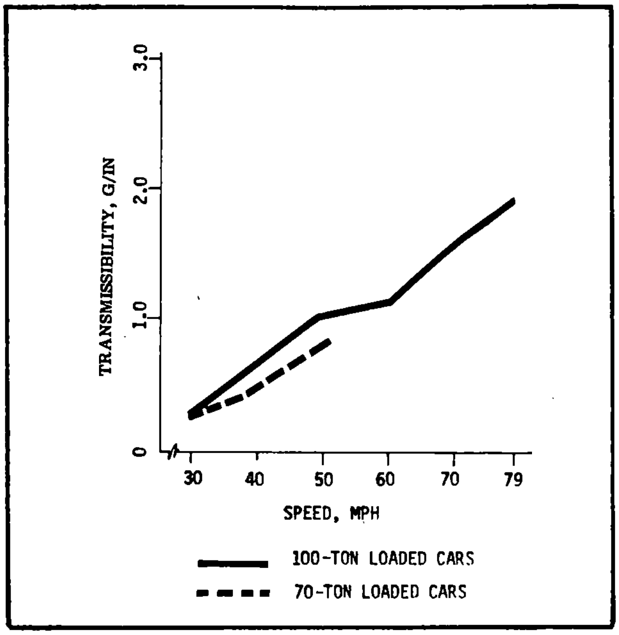


Figure 6-10. Transmissibility - Vertical Acceleration, Type I Trucks, Box-type Cars 0-4 Hz Frequency Band

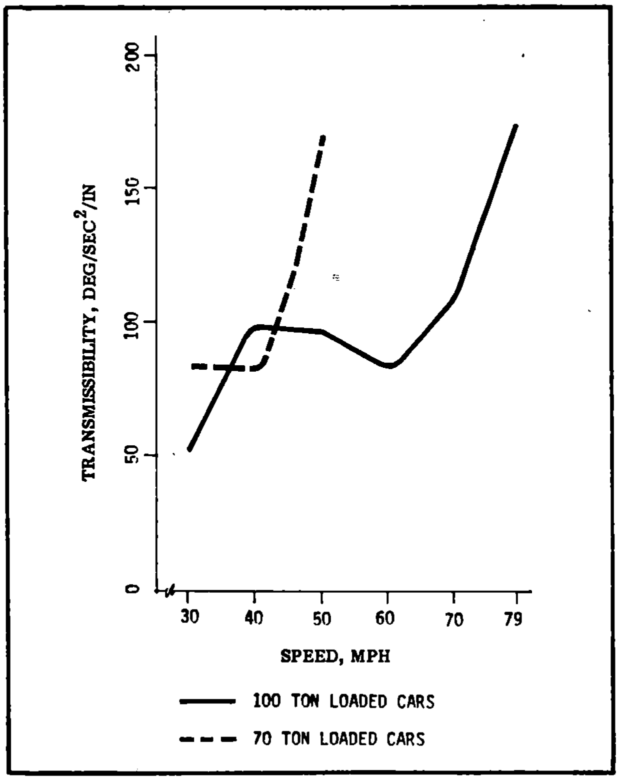


Figure 6-11. Transmissibility - Roll Acceleration for Type I Trucks, Box-Type Cars, 0-4 Hz Frequency Band

6.6 WORK IN PROGRESS

Once the data acquired during the Phase II testing of the Type I truck has been reduced and analyzed, a complete performance characterization of the standard, three-piece truck will be released.

In a related area, a Type I truck acceptance test specification guideline has been formulated and presented to the FRA and the TDOP Phase II consultants for their comments. The following procedural steps will be followed in developing the acceptance test specification:

- Determine salient performance characteristics
- Outline data requirements to quantify performance characteristics
- Define representative equipment and operational conditions
- Select necessary instrumentation
- Construct matrix of tests required to produce data
- Choose test sequence
- Outline test procedures
- Determine data acquisition, reduction, and analysis methods
- Set forth data interpretation and presentation methods and formats

APPENDIX A - ECONOMIC DATA COLLECTION AND ANALYSIS PLAN

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

INTRODUCTION

As the TDOP Phase II contractor, Wyle Laboratories, will develop:

- An illustrative benefit/cost analysis that each railroad may modify to suit its own operating conditions, and
- Economically based performance specifications for trucks that will lead to optimal truck performance.

This document briefly describes the major sources of economic data to be used. It discusses economic impact areas and the analytic methodology to connect economic and engineering performance of freight car trucks. Finally, it briefly outlines the way in which these results may be used in an incremental benefit/cost analysis to assess the cost-effectiveness of Type II trucks.

SUMMARY

Basically, three things are required to perform a benefit/cost analysis: knowledge of the benefits to be expected, knowledge of the costs, and a set of conditions under which trade-offs are to be discussed. Economic benefits and costs will be developed for improved (Type II) trucks through the use of life cycle analysis. Baseline life cycle costs will be developed for the standard (Type I) truck considering several economic impact areas as follows:

Cost Areas:	Benefit Areas:
Initial Purchase Price	Track Maintenance
Inventory Costs	Fuel Consumption
Car Maintenance	
Lading Damage	
Train Delay & Lost Car Days	
Derailment	

In each area, only that part of the cost that might be expected to change with a modified truck design must be developed. In an incremental benefit/cost analysis, only the changes are considered in the analysis. This approach greatly reduces the data requirements of a life cycle analysis.

Baseline life cycle costs will be developed from a number of data sources. This document concentrates on real-world data available from TDOP's railroad subcontractor, the Union Pacific railroad (UP), the FRA, the Association of American Railroads (AAR), and an eastern railroad and a private car line that have contributed maintenance data. Additionally, test data are available from Phase I and more test data will be available from further testing during Phase II. Analytic truck models will be used to interpret and extend the test results. A limited number of Type II trucks are being subjected to service testing as part of the TDOP Phase II wear program. We expect to have data made available from the Facility for Accelerated Service Testing (FAST) on both truck and rail wear. Information is available in the open literature and from other ongoing FRA programs. Finally, the TDOP Phase II industry consultant group regularly reviews our results providing guidance and information.

A multiplicity of data sources is very important to the success of TDOP because of the limited amount of real-world data available to assess Type II truck performance. While baseline life cycle costs are being assessed, causal relationships between the costs and engineering performance must be hypothesized and tested. Engineering performance is to be assessed in the following performance regimes:

- Lateral Stability
- Trackability
- Curve Negotiation
- Ride Quality

Causal connections between the engineering performance regimes and costs must be established so the results of test and analysis data may be applied to estimate the change in costs associated with each Type II truck. This information is the required input for an incremental benefit/cost analysis.

The last requirement for performing a benefit/cost analysis is a well defined set of conditions under which trade-offs are to be discussed. The benefit/cost ratio will vary widely depending on what service conditions are considered. For example, unit train operations insure that the owner line (who pays most of the costs) also is the handling line (who receives many of the benefits). The implications of an improved truck for car lines (who are never the handling line) are quite different from those for Class A railroads. For cars affected by special service orders (designed to insure the return of the car) we will see different benefit/cost ratios than those in general service interchange. Similarly, the engineering parameters of the car (e.g., annual mileage, car weight, cost of lading, truck center spacing, etc.) and the operating characteristics of the railroad (e.g., miles of curved track, posted speeds, track class, etc.) will produce different benefit/cost ratios. Rather than analyze every possibility, TDOP will attempt to identify those areas where Type II trucks might reasonably be expected to pay for themselves; and also to provide the basic data and methodology required to analyze each case.

SOURCES OF ECONOMIC DATA

Life cycle costs are to be developed from a wide variety of sources ranging from results of road tests to the open literature. Virtually all of the information developed as part of TDOP Phase II should eventually be reflected in the economic analysis. The economic task is most closely associated with real-world data available from the Union Pacific railroad (TDOP's railroad subcontractor) and from data contributed by an eastern railroad and a car leasing company. This is because of the need for statistical analysis of these types of data required to produce information useful to the project. It is this class of information (e.g., car maintenance data) that will be described here. Where other sources within the project must be relied upon, they will be described here; however, more interaction between the different parts of the project should be expected as the project develops.

In this section, the economic impact areas to be studied will be introduced, the data sources we propose to use for each economic impact area will be briefly described, and the analysis to be undertaken will be outlined. Because the relationship of the data to the analyses is often fairly complicated, all the data sources will be discussed before considering the detailed analysis techniques to be employed. Additionally, data sources in the area of car movements will be developed.

TRACK DATA

While TDOP Phase II deals with freight car trucks, one of the most important impact areas is probably rail wear in curves. Radial trucks are expected to reduce rail wear in curves, while rigid trucks may accelerate rail wear in curves. Additionally, improved trucks are expected to control hunting. Gage widening due to hunting should be reduced. Harmonic roll and bounce probably are related to weakening rail joints. Clearly, rail deterioration must be considered as part of TDOP. Still, TDOP is a truck design optimization project. Rather than undertake a rail study as part of the project, we will use rail wear models available in the open literature to estimate savings in this area.

A statistical model of a railroad is required to make these estimates, in addition to a model of the wear process under stated conditions (e.g., angle of curvature). In the case of the Union Pacific railroad, the necessary statistics can be developed from a computerized track file known as the Mainline Consist and History. This file is an inventory of UP rail used primarily to predict where damage should be expected. Data are available (finer than milepost-by-milepost) describing the date the track was laid, the cumulative gross tonnage passing over the track, the type of rail, curved track by angle, posted speed, super elevation, grade, and observed failures by type of failure. By taking the difference between cumulative gross tonnage for two successive years of this file, annual gross tonnage is available for the UP's entire mainline.

Similar aggregated data for mainline track in the U.S. has already been made available by the FRA as part of their investigations of fuel consumption. The annual gross tonnage associated with each category (e.g., total miles of track by degree of curvature, speed limit, and grade) is currently being developed by the FRA and will be made available to TDOP.

Our use of these data will be to form a detailed statistical model of a railroad in terms of annual gross tonnage experiencing different angles of curvature, operating speeds, grades, etc. This model will in turn be used to estimate savings in the areas of fuel consumption and rail wear. Only the Union Pacific case will be developed in sufficient detail to adequately estimate these savings; however, as part of our trade-off studies (see Benefit/Cost Analysis Section), we will assess the sensitivity of our analysis to these variables. The national data will be used to evaluate the peculiarities associated with using the Union Pacific. Each railroad may use its own track parameters with the same basic analysis to evaluate truck investments under their particular circumstances.

FUEL CONSUMPTION

Reduction in fuel consumption is another area where improved trucks might reasonably be expected to affect performance. Radial trucks, through their ability to steer through a curve, ought to consume less fuel than conventional trucks. TDOP plans to estimate fuel consumption empirically. The drawbar force on each end of the test car will be measured as part of the road test program. Since the distance integral of the drawbar force is work, this integral multiplied by the locomotive's efficiency factor gives fuel consumption. Comparing the integrals obtained for different trucks, should permit estimating the change in rate of fuel consumption with the different trucks over fixed test zones at constant speeds. The work done in each curve of the test zone will be calculated and compared between trucks. Then, the track model described in the previous section will be used to extrapolate these results to the entire railroad.

Several difficulties exist with this procedure. It will be impossible to control the aerodynamic drag caused by winds; a significant factor in fuel consumption. Wind speed will be measured, and estimated correction factors applied to eliminate its effect; however, data on different car body profiles are not available, and any correction will only be an approximation. It will be impossible to exactly replicate the same speed profile with each test run. Since the data are integrated, this will tend to alleviate some of the sensitivity to small changes in speed. Finally, there is the problem of obtaining sufficient sensitivity to measure the changes expected. The savings to be expected from improved curving performance have been estimated to be of the order of only 2% or 3% of the total fuel consumption. It should be recognized, however, that the small size of this estimate has more to do with the relatively small numbers of sharp curves in the rail network than with the savings to be realized from any given curve. Also, it should be recognized that 2% or 3% of the total railroad fuel consumption is an enormous sum (about 24 to 36 million dollars annually, and increasing).

CAR MAINTENANCE

Car maintenance is another major area where savings are expected from an improved truck. Improved curving performance should lead to reduced wheel wear and longer wheel life. Control over hunting should reduce gib and center bowl wear. If harmonic roll and bounce can be controlled, bolster fatigue, snubber wear, and side bearing damage should also be reduced.

Historical data for off- and on-line light repairs for all UP cars (and foreign cars on UP's lines) by car component have been collected for approximately two years. Light repairs are defined as those requiring less than 20 man-hours of labor. The AAR car repair billing (CRB) system is used to collect these data, which include: car initial and number, the component repaired, the reason for the repair, the component position on the car, the repair date, the geographic location of the repair, and standard labor hours and material quantities at AAR prices. In addition to the usual AAR billable repairs, information has been gathered on nonbillable repairs (e.g., inspections) as part of the UP on-line data. These nonbillable repairs have been priced by UP and can be included for on-line repairs (off-line nonbillable repairs must be estimated from these data).

The pricing system used for car repair billing complicates these data to some extent. Charges are not included for repairs for which the handling line assumes responsibility. Additionally, penalty charges and capital improvement charges are included. Thus, the records will be repriced on a standard basis and will be tabulated in three categories as follows:

a. Charges to the owner

b. Charges to the handling line on-line

c. Charges to the handling line off-line

The first category is the only one paid by a car-leasing company. The first two are paid by a Class A railroad on its own cars, and all three are of interest from an industry-wide perspective. After the numbers of repairs are tabulated, they will be multiplied by average costs from the first category to calculate standardized costs for all three categories. The average costs will be compared to the AAR master price lists and where large discrepancies exist, they will be investigated to determine if adjustments are called for to eliminate penalty charges, etc.

While light repairs represent the majority of all repairs done to UP's cars, heavy repairs are also of interest. In particular, bolster work appears to be done mostly in heavy repair shops. On-line heavy repair data have been collected by the UP, but the files are not automated at this time. Since it would be impractical to access these manual files on any large scale, we plan to request only a small sample of this information (approximately 5% of the fleet). This same approach is being used with car movements. The date and place a car enters the heavy repair shops for this sample of the fleet is known, making manual access to the data less difficult.

In addition to UP's maintenance records, TDOP has been given the repair records for 1977 from a car-leasing company. Annual mileage for each car was also supplied by the leasing company, allowing estimation of repair costs in terms of miles traveled. Finally, we are currently discussing obtaining repair data from an eastern railroad. In this case, heavy repairs are available as part of the basic data file. Geographic differences in the data will be assessed from analysis of records from a variety of railroads.

Car repair data is enhanced considerably when it is merged with data from the Universal Machine Language Equipment Register (UMLER). This is being done in the case of UP's data and will be done for the eastern railroad. Of particular interest in the UMLER data are the year and month the car was built, the nominal capacity of the car, the geometry of the car, the type of bearings (journal or roller bearings), the number of axles, the truck center distance, and any information available about the mechanical class of the car (e.g., the AAR car code and local mechanical designation).

Our analysis of these data is proceeding at two levels: first, the cross tabulation technique is being used to relate the performance regimes to car repairs, and second, wheelset life is being modeled through the use of simulation techniques. The cross tabulation study centers around finding categories of repairs that are related to classes of cars known to have problems in a certain performance regime (e.g., cars with 39-foot track centers have problems with trackability). In the wheelset life study, the age at replacement of wheelsets for different reasons is being calculated (see the discussion of the Wheel Life Model in the next section). In each case, the intent is to relate the repair to a performance regime so that the change in repair frequency with a change in performance may be estimated.

The FAST program and the TDOP wear program offer opportunities to check our results in the area of car maintenance. Data on the deterioration of Type II trucks will be gathered in these wear measurement programs. These data will be compared to our projections. Even more important, secondary effects will be assessed based upon wear program results. Perhaps this is most easily seen with an example. Suppose an improved truck were to reduce rolling resistance of the car. This suggests the standard setup for the brakes might no longer be appropriate. Operating experience is essential to properly assess this type of interaction, where changes cascade causing hard-to-assess (or even unanticipated) consequences.

CAR MOVEMENTS

Car movements relate directly to lost car days in that the utilization rate for each car may be calculated from the data. More important, however, is the potential for calculating mileage traveled for each car. This is important in the analysis of car maintenance as mileage, not time in service, wears out trucks. Additionally, it should be possible to calculate the traffic mix (e.g., 100-ton vs. 70-ton cars) at locations on the UP's line from car movement data. Because of the importance of this information, we believe it to be worthwhile to collect these data for possible use later in the project or by other projects.

Because of time constraints, the use of these data by TDOP will be very limited. We propose to use estimates of average car mileage by mechanical class or other breakouts to estimate the miles a given car has traveled. Where finer data are available, as in the case of the private car line data, we propose to use them. However, we believe it to be too large a task to attempt developing a capability to analyze on- and off-line mileage data, as well as the rest of the TDOP economic analysis data.

The sources for car-movement data at the UP are:

a. Complete Operating Information System (COINS) is a continuously updated interactive system reporting all on-line movements (i.e., UP and foreign cars) and for UP cars off-line on several other railroad's lines with whom they have data-sharing agreements. These data include: geographic departure and arrival locations (freight station accounting code - FSAC) and times for each car number; commodity carried; whether empty or loaded; gross weight; and bad order, storage and hold status.

b. The UP Jumbo File contains the same information as COINS beginning with the inception of the program but it is only available on microfiche. These data are too voluminous to access for a large number of cars.

c. Telerail Automated Information Network (TRAIN II) provides the major source of off-line car movements on a national basis but in less detail than the on-line system. The data include car placements indicating that a car has been turned over to a shipper for loading, loading reports which show when a shipper releases a car to a railroad, the origin of a loaded car, its commodity and destination, interchanging receipts and deliveries (for car tracing), regional crossing boundaries, unloading time, empty car destination, and bad order, storage and hold status.

d. The AAR Per Diem Reporting System further supplements the car movement data. Since the per diem records include empty and loaded miles traveled off the reporting railroad's line, these data may be used to estimate the distance the car has traveled.

e. The Freight Station Accounting Code/Standard Point Location Code (FSAC/SPLC) Master File provides a cross-index between the maintenance records which contain the SPLC for the geographic repair locations and the car movement records which contain the FSAC for the geographic departure and arrival locations. This file, therefore, provides the means to connect repairs to regions and car movements.

f. The Mileage Master File functions to connect miles to the car movements from station-to-station in the car movement file.

Because of the volume of data involved when dealing with car movements, TDOP has requested only a 5% sample of the UP's data. This is to be based on car number according to the following formula: if the car number ends in one and the preceding number is even (including zero), the car is part of the sample; additionally, we have asked for all the cars in the wear program unit train and for 100 cars with National Swing motion trucks that the UP already has in service. Preliminary tests on cars sampled with this formula indicate it gives a very representative sample of the UP's fleet.

LADING DAMAGE

Reduction in lading damage is another major economic impact area where an improved truck might prove beneficial. Improved performance in the harmonic roll, bounce, and hunting regimes should translate into savings in the form of reduced claims for lading damage. Additionally, if the overall ride quality of a improved truck were significantly improved, it might result in expanded markets or at least a strong selling point for using a particular railroad's cars.

Identifying the cause of lading damage in any but specific cases is virtually impossible. For this reason, our approach to date has been to identify specific areas where major problems exist and to attempt to study these areas. Where a relatively small percentage of the fleet is involved in very large claims for lading damage, it seems possible an improved truck might materially improve the situation. Unfortunately, the area identified to date, i.e., damage to automobiles in auto rack cars is so controversial it is difficult to obtain agreement that this is a reasonable area for TDOP to study. The railroads widely believe that the damage occurs not in transit but at the loading and unloading points. As a result, our current approach is to await empirical results on ride quality from the road testing, and to only pursue the matter if it appears reasonable that one of the trucks in the program actually offers a potential solution to the problem.

Data on lading damage are available from the Union Pacific including information on the car number, its AAR car code, the way bill number and date, the standard commodity code, the cause of the loss (to the extent it can be determined), the miles on UP's line involved, and the tons involved. This is difficult to relate to specific routes taken and any other railroads involved. Also, there doesn't appear to be any cost information in the record. Some headway might be made by relating this file to the UMLER and testing for disproportionate numbers of claims of unassigned cause by the car's engineering characteristics or car body type. This will be done using the cross-tabulation technique.

Estimating changes in lading damage charges is probably too difficult to be attempted without actual service experience. We plan to go as far as possible to put bounds on the savings. Where we can't show hard savings, minimum savings (probably zero) will be used to insure a conservative estimate.

LOST CAR DAYS AND TRAIN DELAYS

Reduction in the number of lost car days due to maintenance and in the incidence of train delays due to truck-related failures is yet another area where savings might be realized from an improved truck design.

Train delay and lost car days are currently estimated at the Union Pacific by using data from the dispatcher's daily time sheets. The reasons for train delay (e.g., power shortage, accidents, etc.) and the location are manually recorded on the sheets which contain the train's departing station, date and time of arrival. The data are keypunched and visual charts are manually prepared to display the data periodically.

The major effect of an improved truck design on lost car days probably is in the maintenance area. Car days are lost when the car is being maintained. Charges for lost car days will be included as part of the maintenance analysis.

The major effect of an improved truck design on train delay probably is in the track area. Rail replacement necessitates significant interruption of service and charges will be estimated for this as part of the rail study.

DERAILMENT

Reduction in the number of derailments is a likely outcome of adopting an improved truck design. However, as with lading damage, it is extremely difficult to attribute some failure of the trucks as the cause of any large number of derailments. A computerized UP accident reporting system is available which is used both internally at UP and to meet the requirements set forth in the "FRA Guide for Preparing Accident/Incident Reports". These data are in considerable detail covering the environmental and operating conditions at the time of injury or damage to persons, property and equipment. Cause codes related to certain truck components are included: bolsters, side frames and bearings, snubbers and springs (as well as truck stiff, improper lateral, or swivelling).

TDOP's use of these data can be to relate the principal car involved in the derailment to its maintenance history for comparison with other similar cars to establish any trends related to the truck, possibly providing a link to derailment. Just as in the case of lading damage estimates, conservative numbers will be used where hard savings cannot be demonstrated.

PURCHASE PRICE AND INVENTORY COSTS

In most cases, the initial purchase price of an improved truck is significantly higher than that of a standard three-piece truck. The purchase price of each truck being considered is available from the manufacturer and has been requested. The requirement that a purchase price be established is part of TDOP's selection criteria for trucks.

Probably even more important than the added purchase price is the added inventory cost necessary to maintain the nonstandard parts of an improved truck. This information will be developed by obtaining a price list for the nonstandard parts in each truck and predicting the quantities of each part required based on estimated replacement rates as predicted from the analysis of car maintenance discussed in the Statistical Analysis Section of this appendix. Analysis of inventory costs is discussed in the same section.

DATA AVAILABILITY SUMMARY

A great deal of real-world data exists that is relevant to the goals of the TDOP project. Virtually no data are available on Type II trucks. This is the challenge we are faced with, extracting information from the data that do exist to predict what would be found if premium truck data were available.

The data we are currently gathering for TDOP economic analysis are summarized in Table A-1.

In addition, we will request heavy repair data on the basis of an individual car number, repair site, and date using the COINS data. If train delay data are in a higher state of automation later in the project, we may request a representative sample for further analysis. Also, an eastern railroad and a car leasing company either have contributed data, or are currently considering contributing data.

Table A-1. Summary of TDOP Phase II Economic Data

On-Line Maintenance - All UP Cars	1/77/ through 3/80
Off-Line Light Maintenance - All UP Cars	1/77 through 3/80
In-House UMLER	Latest Update On-Hand at Wyle
Car Movements (COINS, TRAIN II, Per Diem Reporting System) for Selected Cars a. All UP car numbers with the next to last digit even (approximately 5% of the fleet) b. UP National Swing Motion Cars: UP Car Numbers 215550 through 215649 c. Wear Program Cars: UP Car Numbers 31900-32099	Monthly through 3/80 and all available
Freight Station Accounting Code/Standard Point Location Code Master File	Latest update
Mileage Master File	Latest update
Track Files: Mainline Consist and History	Last three annual
Accident File	Latest annual
Freight Loss and Damage File	Latest annual

STATISTICAL ANALYSIS

The objective in analyzing these data is to quantify the changes expected with an improved (Type II) truck. Also the timing of the changes must be estimated. In order to meet this objective, it is essential that causal relationships be established between engineering performance and costs. Requirements for establishing this relationship are outlined in Section 3 of the TDOP Phase II Introductory Report, FRA/ORD-78/53. Briefly, in each engineering performance regime, we must relate the operating environment of the truck (e.g., speed, angle of curve) and the truck's engineering characteristics (e.g., truck center distance, wheel profiles) to a mechanism of failure (e.g., unlubricated sliding friction) and a mode of failure (e.g., loss of metal). Only when this relationship is established can we make predictions with a high level of confidence. Statistical analysis alone cannot do this. As outlined in Section 5 of the TDOP Phase II Introductory Report, engineering interpretation, testing, and analysis are also required.

Statistical analysis has a major role in the success of TDOP. In many areas of truck performance, the relationship of performance to cost is simply not understood at this time. It is in these areas that statistical analysis can make its most significant contribution. While it is not possible to establish causal relationships statistically, these techniques are designed to find relationships (not necessarily causal) and to say whether a significant relationship actually exists. Additionally, once a causal relationship is established, statistical analysis can help quantify the relationship.

This should not be confused with economic analysis (with the exception of inventory costs described later in this section). The benefit/cost analysis briefly outlined in the next section is an economic analysis. What is being described in this section is simply statistical analysis. Since economists are trained as statisticians, the responsibility for doing this analysis falls in the economics area.

CROSS TABULATION

Cross tabulation studies are almost always done as the first step of a statistical analysis. Cross tabulation is a simple and remarkably powerful tool for discovering relationships between data. The procedure starts with identification of the variables available to be analyzed. Then relationships between these variables are hypothesized. A table showing the joint frequency distribution between the variables is constructed. The expected value of each cell in the table is constructed based upon the assumption there is no relationship between the variables (i.e., that they are distributed independently). Finally a chi square (χ^2) statistic is calculated to test the assumption that the variables are independently distributed. If this statistic shows that it is extremely unlikely that the variables are independent, the assumption of independence is rejected and the opposite assumption - that there is a relationship - is accepted.

An example will make this more clear. Consider the analysis of car maintenance data. The first step is to identify the variables available to be studied. Figure A-1 illustrates one of the data records for CRB maintenance data. There is a large number of variables available in this record. In particular, notice the following variables:

a. Car initial and number starting at column 20. This uniquely refers each record to a freight car.

b. The removed job code starting at column 58. The job code tells what part of the car is being repaired (e.g., job codes 3005 to 3125 refer to wheels). Each part repaired requires another record.

c. The "why made" code starting at column 56. This code explains the reason for making the repair (e.g., thin flange, high flange, etc., for wheels).

By looking up each car number in the UMLER file, more variables may be made available. Figure A-2 illustrates the UMLER data record used for most car types. Notice the following variables:

d. The nominal capacity starting at column 64. This indicates the nominal weight the car can carry in thousands of pounds.

e. The truck center spacing starting at column 67 of the second row. This indicates the distance between truck centers in feet and inches.

Having identified a few of the variables to be studied, we need to hypothesize a relationship between the variables. An engineer explains that cars with 39-foot truck centers are much more susceptible to harmonic roll because this corresponds to the spacing of rail joints for 39-foot staggered jointed rail. We expect some of the wheel repairs to be caused by harmonic roll, so we hypothesize a relationship between cars with truck center spacing around 39 feet and the "why made" codes for wheel repairs.

Next, we are ready to construct our table (known as a contingency table). First, we must separate all the car repair records with job codes 3005 to 3125 (i.e., wheel repairs) from our car repair records and match each record with its UMLER record to gain access to the additional data included in the UMLER.

At the same time, we recode the truck center spacing as follows: arbitrarily we say plus or minus four feet (10%) is close enough, if the truck center spacing is 35 feet to 43 feet we change the value to be 1; otherwise, we change the value to be zero. We then count the number of 0's and the number of 1's found for each "why made" code. This produces a table shown as Table A-2 where the number of records counted is listed as the "observed" value in each cell.

Next, we need to calculate the expected value of each cell in the table based upon the assumption that there is no relationship between the two variables. To do this, we calculate the "marginals" for the table. By summing the observed values across all the columns and all the rows, we calculate the percentages shown to the right and below the table. These numbers are the distribution of the variables ("why made" codes and truck center spacing) taken independently. If we multiply them, we calculate the distribution as if they were independent. If we also multiply by the total number of records in the entire table, we obtain the "expected" value for each cell.

Finally, we are ready to calculate the χ^2 statistic. This is done by taking the difference between the observed and expected values in each cell, squaring this difference, and dividing by the expected value. When these numbers are added, they form the χ^2 value for the table. We may go to any statistics book and look up the number in a table of the chi squared distribution where we use the number of degrees of freedom equal to the number of cells in the table minus one. The number we obtain from the table tells how likely this result is based on random chance.

For example, if we have 23 degrees of freedom, we find that $\chi^2 = 64.0$ for .99999 probability. Since we have $\chi^2 = 337$, we are well beyond the value indicating that 0.001% of the time this result will happen by chance. In the social sciences, statisticians usually accept anything less than a 5% chance as evidence that a relationship exists. Based upon this criterion, it appears there is a relationship.

Inspection of the table reveals what the nature of the relationship is. Notice that "why made" codes 64 (high flange), 75 (tread shelled), and 76 (tread built-up) occur far more often than we would expect on cars with truck centers between 35 and 43 feet. This suggests (but does not prove) that harmonic roll is part of the cause of these problems. If engineering analysis can demonstrate that there is a relationship, we might be justified in assuming a reduction of incidence of these problems down to the level of the other truck center category if an improved truck were to completely solve the harmonic roll problem.

This example illustrates the basics of the cross tabulation procedure. Numerous extensions of this basic idea exist. For example, it is not always appropriate to use the marginals as the assumed distribution. One might use the distribution of total miles traveled by cars in each of the two truck center categories. This would control for the distance traveled. Another extension to the basic technique would be to count dollars (another variable in the car maintenance records) instead of records.

RAIL WEAR IN CURVES

One of the major areas that needs to be developed under TDOP Phase II is rail wear in curves. Rather than develop expertise in rail wear, Wyle will collect and analyze rail wear data from other sources. The essential activity to be performed is to estimate savings to be realized in rail life based upon the measured performance of improved truck designs. The essentials of this analysis are indicated in the following discussion. The open literature contains examples from research institutions that have developed this analysis beyond the state discussed here. We plan to draw on this knowledge to evaluate the savings to be gained in the area of rail wear.

The proposed procedure for making this estimate of rail wear in curves is illustrated in Figure A-3. As part of the TDOP test program, lateral force on the leading outer wheel in a curve is to be measured. This force, coupled with slipping between the wheel and rail, causes wear of both the wheel and the rail. These test data are to be used to validate analytic models of truck behavior in a curve. A curving model can be used to generate continuous plots showing the lateral force as a function of angle of curve (e.g., the bottom curve in Figure A-3). Given the plot illustrating rail life for Type I trucks shown at the top of Figure A-3, an easy geometric construction produces the estimated relationship for rail life for Type II trucks.

The curve illustrating rail life for Type I trucks may be estimated directly from the Mainline Consist and History data file available at UP. This file gives information on the age of all the track in the UP's mainline by angle of curve and superelevation. This is a file listing "live" rail. We want to know the age at replacement (i.e., the age of "dead" rail). In order

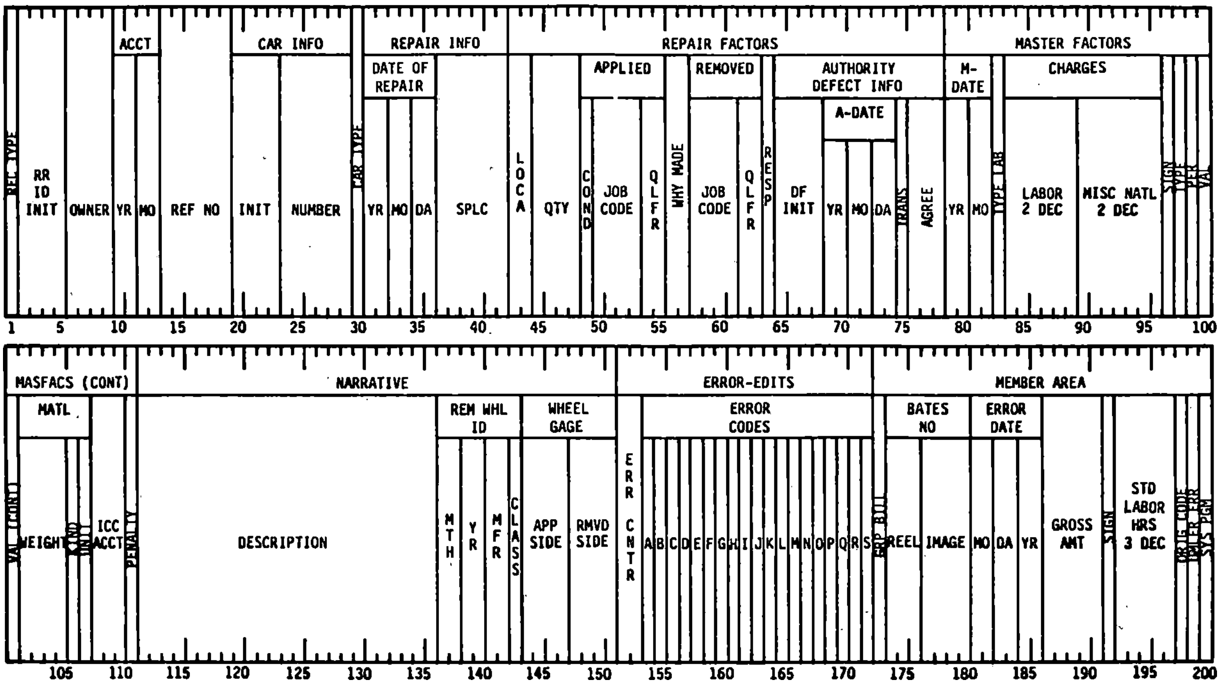


Figure A-1. Union Pacific On-Line Repair Record

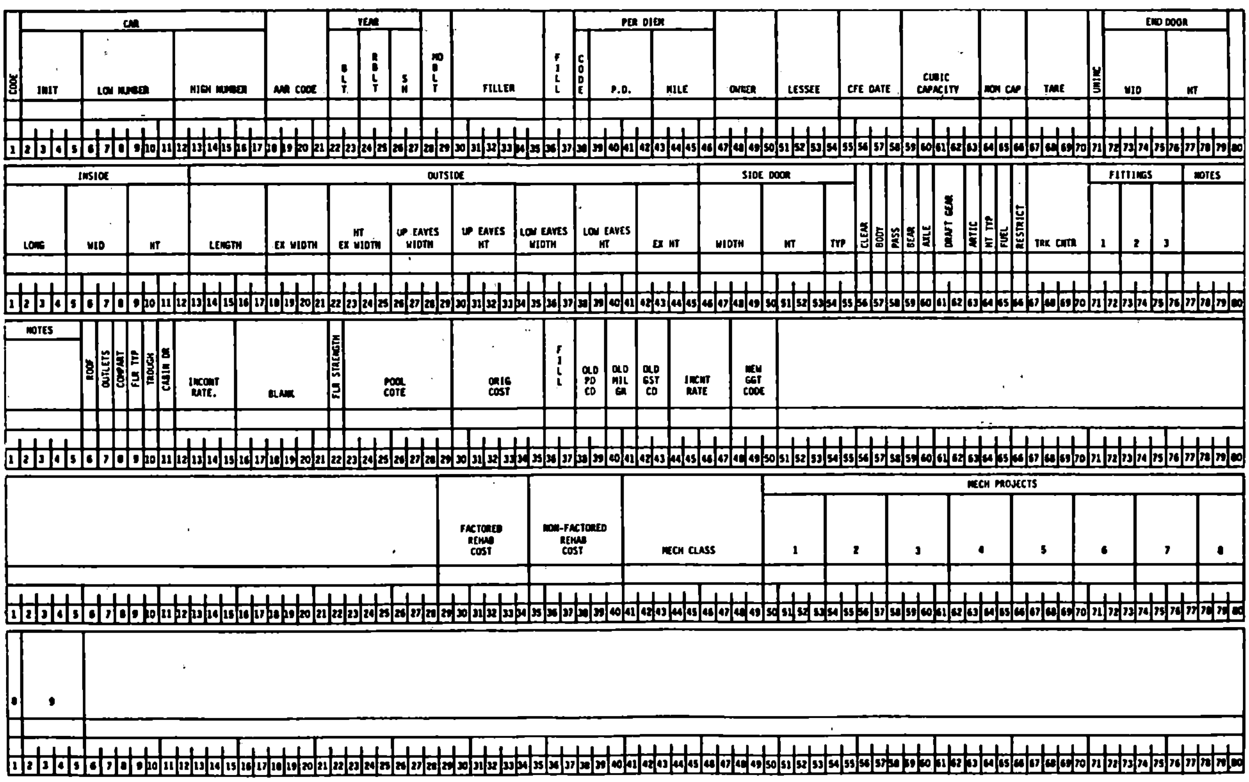


Figure A-2. UMLER Data Records - Codes 1 and 4

Table A-2. Cross Tabulation

WHY MADE CODE		TRUCK CENTER 0	TRUCK CENTER 1	MARGINALS	
11	Observed	560	547	1107	40.01%
	Expected	614	493		
60	Observed	268	77	345	12.47%
	Expected	191	154		
64	Observed	49	130	179	6.47%
	Expected	99	80		
73	Observed	41	4	45	1.63%
	Expected	25	20		
74	Observed	6	3	9	0.33%
	Expected	5	4		
75	Observed	28	99	127	4.59%
	Expected	70	57		
76	Observed	128	155	283	10.23%
	Expected	157	126		
77	Observed	6	22	28	1.01%
	Expected	15	12		
78	Observed	157	120	277	10.01%
	Expected	153	123		
79	Observed	11	8	19	0.69%
	Expected	11	8		
89	Observed	81	30	111	4.01%
	Expected	62	49		
90	Observed	201	36	237	8.56%
	Expected	132	105		
MARGINALS		1536 55.5%	1231 44.5%	Total = 2767	dof = 23 χ^2 = 337

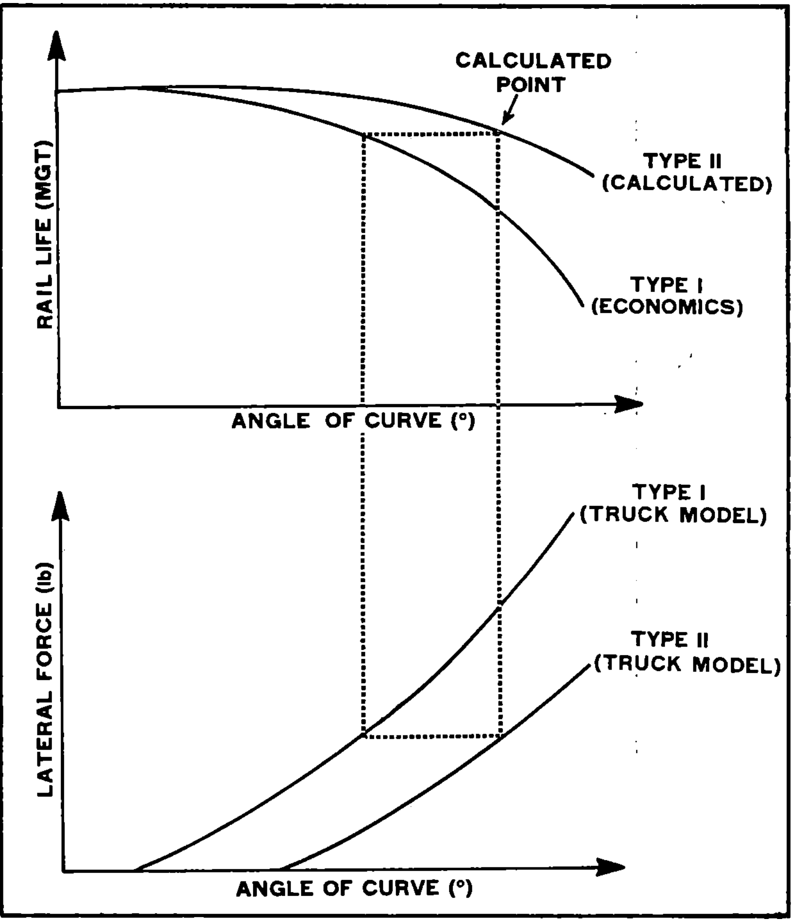


Figure A-3. Estimating Rail Life

to estimate this, we must assume that no improvements are being made in the rail on the UP. We may estimate rail life based upon the track file (e.g., by averaging the data at each angle of curve), however, we will tend to produce curves that indicate the rate at which continuous welded rail (CWR) is being introduced instead of curves that indicate rail life. Another method for estimating the top curve is to measure rail profiles and estimate the remaining rail life. This apparently has already been done by other research institutions.

The analytical models of curving may be used to estimate the effect of traffic mix. The lateral force for different car weights may be estimated from the model. Then, if the usual assumption is made that the rate of wear is proportional to the normal force, the significance of different car weights may be assessed. This assumption is more important than it might seem at first. Notice that a variety of curves (for the one lower curve in Figure A-3) will be available even for Type I trucks. Each car weight will give a different curve. The curve that should be used is a weighted average of the curves for individual car weights. The weighting should be in proportion to the traffic mix and the normal force. Again, this assumption is discussed in the open literature, where it is widely held that heavy cars greatly accelerate rail wear. This could either be because the lateral force is a great deal higher for 100-ton cars, or because the rate of wear is more than proportional to the normal force. This last possibility may in fact be the case. The wheel/rail interface is commonly beyond the yield limit for the materials involved; a situation not often considered in wear models.

Yet another problem exists in that the curve shown for Type II truck rail wear is calculated as if only Type II trucks were running on the rail. In most situations, this would not be valid. Type I trucks would be more common than Type II trucks. Intermediate points may be interpolated using the same sort of weighting scheme just discussed; however, it seems doubtful that any real savings would be realized from a small number of cars. Rail is often replaced on the basis of the million gross tons (MGT) that it has experienced, rather than strictly on the basis of the inspection results. Some major increase in life will have to be predicted before it is reasonable to attribute any savings in this area.

Finally, there is the matter of estimating the savings associated with a shift in the bottom curve. This may only be done based upon some known distribution of rail by angle of curve, posted speed, superelevation, and traffic mix. The only railroad for which sufficiently detailed information will be available will be the Union Pacific. Further, the cost of replacement of rail will tend to vary somewhat. Again, pricing information will be obtained from the Union Pacific and will only apply to them. Notice, however, that any railroad may use the source data developed and their own rail distribution and pricing information to estimate savings on their line.

WHEEL LIFE MODEL

One of the major areas where car maintenance dollars are expended is on wheel replacements. Typically, when a wheel is replaced, both wheels, an axle, both roller bearings, and labor are charged for one wheelset replacement. Wheel set replacements account for approximately 30% of all light maintenance dollars charged to the owner in the UP data. Wheelset replacement costs are expected to change with introduction of an improved truck. We will need to quantify these changes and estimate their timing as part of TDOP.

Large quantities of data are available on the replacement of wheelsets in the car repair data. It is classified as to cause in terms of "why made" codes introduced earlier in the Cross Tabulation section. Unfortunately, only limited information is available to estimate the time or mileage between replacement. Car age is available, and since April 1978, year and month of manufacture of each wheel is available. Ideally we would have a long enough duration of car maintenance data to observe the same wheels being replaced over and over. Unfortunately, it is too long between wheelset replacement (approximately 9.4 years in the UP's case) for replacement to be observed very often.

The problem with not knowing the age at replacement is surmountable, though it does complicate the analysis. A simulation procedure can be used to calculate the wheel-set age from the known car age.

If we start by assuming a normal distribution (i.e., with a mean and a standard deviation) for the age of the wheelset at replacement, then we may calculate the distribution we would observe using car age instead of wheelset age (see Figure A-4). If we also tabulate car maintenance data by car age and define an error, e.g., error = (calculated - observed)^2 , we may use a search algorithm to minimize the error through the choice of mean and standard deviation. In other words, we use a systematic procedure to guess the mean and standard deviation, taking the guess with the lowest error as the correct one. We plan to use one of a number of search algorithms (systematic procedures for guessing) already available on the computer (probably one based on the gradient method). We will not write our own algorithm.

Calculation of the distribution using car age proceeds as follows: suppose the probability of a wheelset being replaced is normally distributed. Then, the probability of a wheelset being replaced between wheelset ages of $t - \Delta t$ and t is as follows:

$$P(t) = (1/\sqrt{2\mu\sigma}) \int_{t-\Delta t}^{t} e^{-(x-\mu)^2/2\sigma} dx \qquad \text{(Eq. 1)}$$

Where μ is the mean and σ is the standard deviation. Subroutines are readily available for calculating P(t) given values for μ and σ.

For the moment, consider a fleet of 1000 cars. We can calculate how many wheelsets are replaced in the first period of time as follows:

$$X(1) = 1000\ P(1) \qquad \text{(Eq. 2)}$$

The next period, the number of wheelsets replaced from the wheelsets that were not replaced yet is simply 1000 P(2). However, some of the wheelsets that were replaced must be replaced again. This occurs with the same distribution as the new wheels

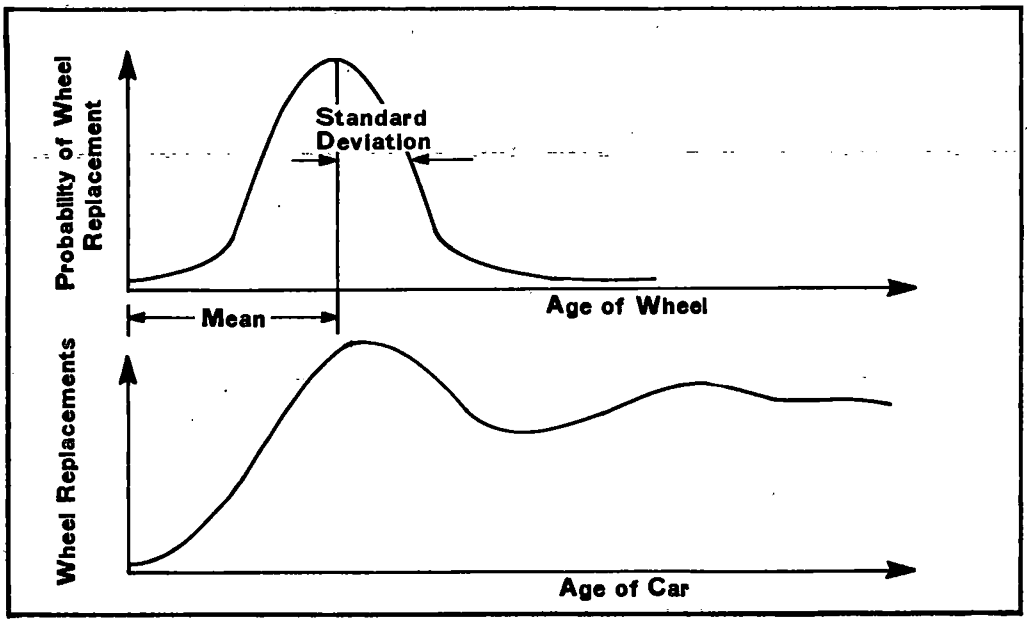


Figure A-4. Estimating Wheel Life

so it is estimated by X(1) P(1). Thus in the second period the following numbers of wheel-sets are replaced:

$$X(2) = X(1)\,P(1) + 1000\,P(2) \quad \text{(Eq. 3)}$$

$$X(3) = X(2)\,P(1) + X(1)\,P(2) + 1000\,P(3) \quad \text{(Eq. 4)}$$

$$X(4) = X(3)\,P(1) + X(2)\,P(2) + X(1)\,P(3) + 1000\,P(4) \quad \text{(Eq. 5)}$$

and so on. The shape of the X(t) is illustrated in the lower figure in Figure A-4.

A handy check on these equations is to see that 1000 cars are always present each period. This can be done by calculating the ages of all the wheelsets for the 1000 cars. The probability that a wheelset will survive until period t is given by:

$$F(t) = (1/\sqrt{2\pi\sigma}) \int_t^{\infty} e^{-(x-\mu)^2/2\sigma} dx \quad \text{(Eq. 6)}$$

Then, in each period, the numbers of cars is the sum of the wheelsets replaced and the survivors as follows:

$$1000 = X(1) + 1000\,F(1) \quad \text{(Eq. 7)}$$

$$1000 = X(2) + X(1)\,F(1) + 1000\,F(2) \quad \text{(Eq. 8)}$$

$$1000 = X(3) + X(2)\,F(1) + X(1)\,F(2) + 1000\,F(3) \quad \text{(Eq. 9)}$$

$$1000 = X(4) + X(3)\,F(1) + X(2)\,F(2) + X(1)\,F(3) + 1000\,F(4) \quad \text{(Eq. 10)}$$

and so on.

A minor problem exists in that there are no cars with ages in the negative numbers. This means that F(0) is not equal to one. To correct this, equation 1 and equation 6 may be multiplied by 1/F(0). In other words, because the car age starts at zero, it isn't really appropriate to assume a normal distribution. However, the error is very small as long as the mean is significantly larger than the standard deviation (at least by a factor of two). If this is not so, a "normal" distribution doesn't make sense as an assumed distribution.

What is available in the car maintenance data is not 1000 cars being followed through time but rather all the cars (of many ages) looked at in approximately a one-month period. This does not present any problem. If we calculate based on one car in the population, we get the distribution of wheelsets replaced by car age as X(t). Then, if we multiply by the total number of cars in the fleet of each age, we get the expected number of wheelsets replaced in a period.

The fact that there are four wheelsets on each car does not make a great deal of difference either. If we go back and replace all the references to "wheelsets" with the statement "wheelset on the number one axle," it is clear we can do this four times (one for each axle position) and look for differences in wheelset life between axle positions (there appear to be such differences).

Multiple causes for wheelset replacements do complicate the result somewhat. In order to deal with differences in wheel life between different causes for wheelset replacement, it is necessary to introduce a new variable, the probability of a wheelset being replaced due to each cause. Then we redefine P(t) to be as follows:

$$P(t) = p_1\, P(t \mid \mu_1, \sigma_1) + p_2\, P(t \mid \mu_2, \sigma_2) + \ldots \quad \text{(Eq. 11)}$$

where the notation $P(t \mid \mu_1, \sigma_1)$ means P(t) evaluated for values of the mean and standard deviation of μ_1 and σ_1. And where p_1 is the new variable, the probability of a wheelset being replaced due to cause 1.

Some of the causes of wheelset replacement are clearly associated with failures instead of wear and can happen at any time with equal likelihood. This is easily modeled by dropping the P(t) term and the mean and standard deviation for these causes just leaving p_n to be estimated.

Finally, we can switch from time to mileage by estimating annual mileage and multiplying it times the car age. Doing this on a per car basis would significantly decrease the estimates of standard deviation because variance in the mileage and variance in the wheel life are being estimated together when wheel life is calculated in terms of time. Unfortunately, mileage on an individual car basis will not be available due to time limitations.

The wheel life model described above will allow us to estimate changes in wheel life associated with Type II trucks. For example, suppose the engineering analysis suggested that "thin flange" would be eliminated by adopting a radial truck design. Then the probabilities in the wheel life model could be adjusted and a new average life of wheelsets calculated. This would tell how much longer to expect the wheelsets to last and what "why made" codes to expect them to ultimately fail in. This information is what is required as input to the benefit/cost analysis.

ANALYSIS OF INVENTORY COSTS

Perhaps the largest obstacle to the acceptance of Type II trucks is concern on the part of the railroad management about how many additional nonstandard parts will have to be stocked to maintain them. Since this concern is probably well founded, it is important that a very good job be done of estimating the added inventory costs associated with each premium truck. This information will be developed by obtaining a price and weight list for the nonstandard parts in each truck from the truck manufacturer. Competing maintenance strategies will be costed between stockpiling parts at a single site or distributing them over the potential repair sites. If there is a point of crossover, where the most cost-effective strategy changes, the time will be calculated at which the crossover occurs. Finally, our estimate of inventory costs will be verified with the Union Pacific railroad to help insure that the results are reasonably accurate.

The costs to be considered are the initial purchase price of the parts, the cost of shipping them to the railroad, the cost of shipping them to the repair site, and lost car days waiting for parts to arrive. Depending upon the strategy used, the numbers of parts required will be based upon the estimated frequency of repair of the part (in the case of dispatched parts) or the estimated frequency of repair adjusted for minimum inventory requirements (in the case of distributed parts).

A point of changeover where the least-cost procedure changes is to be expected. Early in the introduction of a Type II truck, dispatched parts should be the optimal strategy. There will be very little requirement for parts, offsetting losses due to lost car days waiting for parts to arrive. Later, the distributed parts strategy may become the least-cost strategy when the demand for parts increases and the lost car days become excessive. If a small number of cars is involved, the dispatched parts strategy may be optimal for the entire life of the cars because of the large inventory required with the distributed parts strategy.

The estimated frequency of replacement will be based upon the car maintenance analysis results discussed in the next section. Rates of replacement will be estimated for standard parts and their nonstandard equivalents will be estimated at the same rates. Where no standard equivalent part exists, engineering judgment will be used to classify the part as "similar" to some existing part. The wear data program may provide some input along these lines.

Individual components may have different optimal inventory strategies. This may be evaluated by performing the calculations described above on a component-by-component basis. In other words, a nonstandard bolster probably would be replaced so seldom that a dispatching strategy would be optimal for a much longer time than a non-standard wheel. Also, the optimal strategy will depend upon the rate of introduction of additional car sets. This will also need to be estimated.

COMPONENT LIFE CYCLE ANALYSIS

The most convincing life cycle analysis is to track trucks in real service over long enough periods of time to observe the same components being changed over again. This is not really necessary to obtain a reasonable estimate of how long that time is. Using the number of box cars in the UP's fleet taken from the National Equipment Register (issue effective 1/1/78), a ratio of that number to the number of box car components repaired in two months was developed. This method was used to estimate the mean time between replacement of several components (the small sample of data shows that wheels come out to an average replacement of 9.4 years). These kinds of numbers will be generated from the approximately 40 months of repairs anticipated by the end of Phase II.

Where cost data indicate that a full-blown life model is not justified, the method of developing ratios to obtain average life will be used. It has several drawbacks. For example, if the age distribution of the fleet of cars changes, this method incorrectly reports this as a change in the life of the components. When this technique is used, the main issue to be addressed will be to determine if there is any difference in life between components in different car classes (after taking into account mileage differences). For example, do cars with 39-foot truck centers break springs more often? Accuracy of this technique will be improved as more railroads' data are considered. If they all have changes in age distribution, it seems reasonable that the industry as a whole is undergoing such changes.

ANALYSIS OF FUEL CONSUMPTION

The changes to be expected with different Type II trucks are to be estimated based upon experimental results from the test program. Drawbar force is to be measured at each end of the test car. The energy dissipated in each curve of the test zone, and at each speed along track, etc., through the following equation:

$$\text{work/mile} = (1/\text{mi}) \int_0^t V \bullet F \, dt \qquad \text{(Eq. 12)}$$

where the data should be normalized by either the distance (mi) or time (t) spent in the curve (or whatever else is being considered). Data points of this type will be collected from along the test zone from each test.

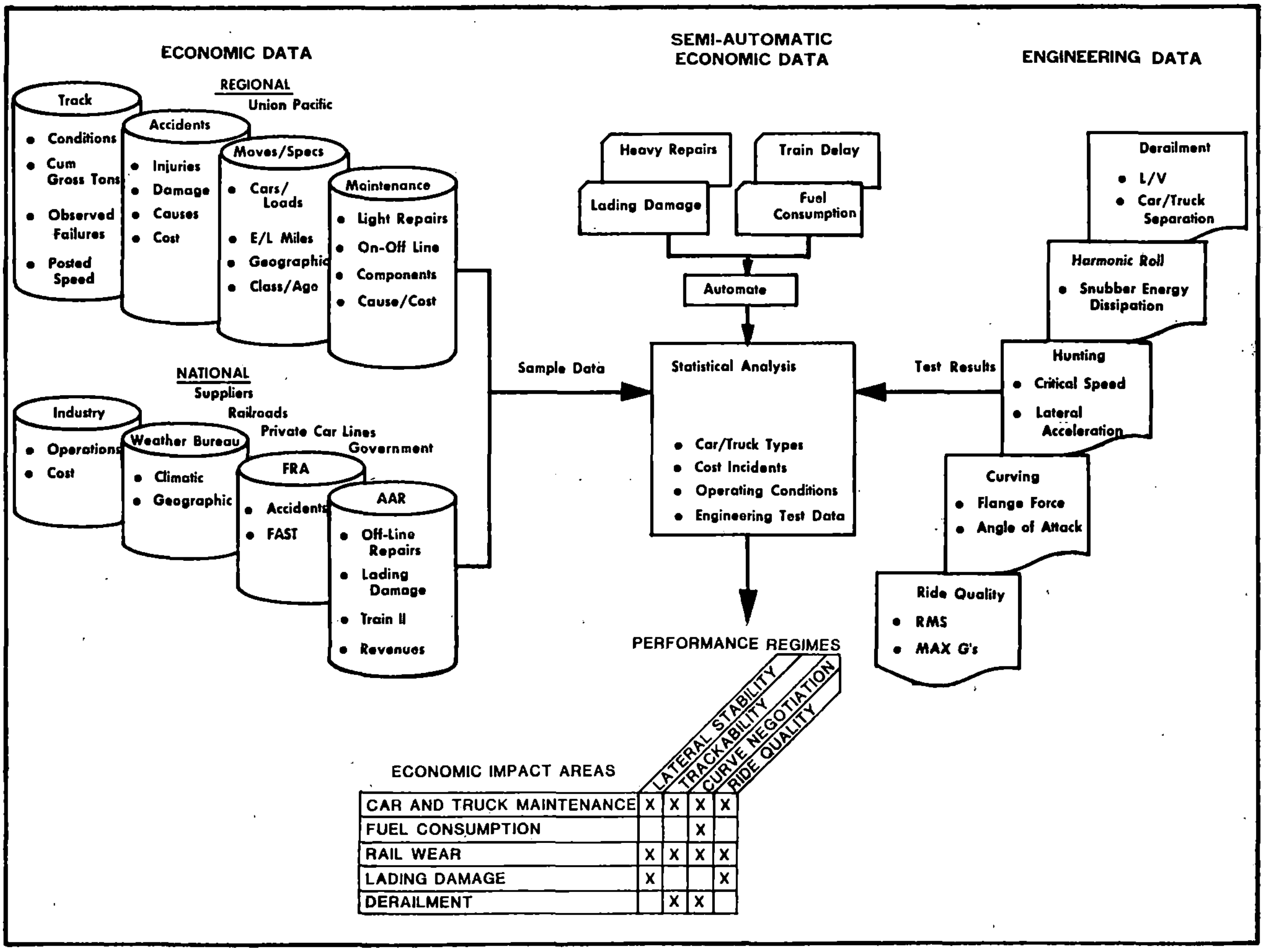
At this point, it is necessary to relate the energy dissipated to actual fuel consumption. A current FRA project can help in this regard. As part of the Locomotive Performance Analysis Program (AR 74348), both drawbar force and the rate of fuel consumption will be measured. This system can provide the calibration of energy dissipated to fuel consumption.

Finally, we briefly get to a job for the economics task. Taking the measured data from the TDOP testing and the calibration data, coefficients for the Davis equation will be fitted using regression techniques. From here, we will attempt to use existing fuel consumption analysis programs to evaluate the potential and savings of a Type II truck.

ANALYSIS SUMMARY

Further definition is required to complete this economic analysis plan. The major costs of hunting have not been identified yet, much less analyzed. This is only one example reflecting our existing state of knowledge. Often the costs of things are suspected, but there has been little systematic work to verify these suspicions. As a result, this plan cannot be "complete". Rather than telling how all the analysis is to be done, this plan has attempted to show how we expect to get there. Relationships must be found, using the cross tabulation technique and engineering analysis. Then we can attempt to analyze them.

Enough examples have been developed to make the methodology clear. Statistical analysis is being used on real-world data to obtain insight into the cost of problem areas in which Type II trucks might improve performance. A large amount of data is being collected from diverse data files belonging to many divisions of the railroad. By combining these data files into a data base (see Figure A-5), the information available is greatly enhanced. As the problem areas become identified, further analysis techniques suggest themselves. The wheel life analysis outlined earlier resulted from this methodology. As more data are analyzed using the cross tabulation technique, more detailed analyses will be identified.



ECONOMIC IMPACT AREAS	LATERAL STABILITY	TRACKABILITY	CURVE NEGOTIATION	RIDE QUALITY
CAR AND TRUCK MAINTENANCE	X	X	X	X
FUEL CONSUMPTION			X	
RAIL WEAR	X	X	X	X
LADING DAMAGE	X			X
DERAILMENT		X	X	

Figure A-5. TDOP Phase II Economic Data Base

In the case of identifying the major costs of hunting, there appears to be general agreement that increased car maintenance, operational constraints on speed, and an unacceptably high risk of derailment are the major costs. Also hunting probably damages the rail, and possibly causes gage widening. Unfortunately, there does not seem to be much agreement on which parts incur the increased car maintenance. Everyone seems to agree that center bowl wear is accelerated by hunting; however, this is a relatively small factor in light maintenance records. Probably these costs are to be found in the heavy maintenance records (which TDOP Phase II has not analyzed yet). In any event, headway can be made through the type of analysis described here. We plan to compare journal bearing cars (where hunting is not so serious) to roller bearing cars. We expect to compare data from railroads that limit their speed to the UP's data. This type of comparison coupled with engineering evaluation, test data, and analytical modeling can reasonably be expected to produce results.

BENEFIT/COST ANALYSIS

The preceding discussion centered around quantifying the changes to be expected with an improved truck design. This section discusses the analysis of these changes to determine if an improved (Type II) truck is a sound investment. The methodology recommended in this plan is to conduct an incremental benefit/cost analysis to evaluate proposed investments in improved trucks. Much of the discussion centers around arriving at a set of conditions under which the analysis is to be made. Normally this is not a major problem - the conditions under which an investment are to be made are usually well known. Because of the desire that TDOP be universally applicable, the analysis proposed here is not as simple as if a given railroad were to attempt to evaluate Type II trucks under more fixed conditions.

JUSTIFICATION OF INCREMENTAL ANALYSIS

The railroad industry purchases trucks to continue their operations. Given a level of capital investment in freight cars, there will be some rate of replacement necessary to maintain that level of investment, and this will necessitate some continuing rate of purchase of freight car trucks. Investment decisions require information we are not planning to gather such as the profitability of individual routes and car body types. Decisions about how many trucks should be purchased are clearly outside the realm of a Truck Design Optimization Project. Thus, TDOP should not be involved in decisions like whether to retrofit part of an existing fleet with improved trucks - we haven't gathered the data necessary to make such a decision. Rather, we propose to investigate the conditions under which an improved truck would be a profitable investment as part of the ongoing replacement of the existing fleet as the existing fleet wears out. In this way we can isolate our attention on the truck, ignoring the larger issues.

Given that a decision has already been made to purchase a truck, we ask the question whether it would be worthwhile to invest extra money in an improved truck to gain improved performance from the truck. The only issues involved are incremental ones: are the incremental costs of purchasing an improved truck justified by the incremental benefits expected from this purchase. If truck performance is degraded in some area with a given Type II truck, the expense of this loss of performance is part of the incremental cost. If the purchase price is higher, this is part of the incremental cost, etc.

It is within the realm of possibility that the performance improvements obtained from a Type II truck might be so large that they would materially affect purchase decisions affecting the size of the fleet. This is possible; however, TDOP is not in a position to analyze this possibility.

CONDITIONS TO BE ANALYZED

The next important issue confronting us is whether to consider that Type II trucks have not yet been approved for interchange service. Most important, do we consider that other railroads will not stock parts for these trucks? Clearly, we must start from the existing situation and work from there. If a Type II truck will pay for itself recognizing this problem, it will be introduced and ultimately arrive at the situation where parts are stocked. There is very little utility to answering a question like, assume all railroads were currently required to stockpile the nonstandard parts for this truck and further that this truck was so common that it could be introduced into unrestricted use in interchange service, then would an investment in the truck be profitable? The answer to this question is meaningless, the assumptions did not happen.

If we do not assume that a Type II truck is to be used in general interchange service, what do we have left? Type II trucks might be used on unit trains for service entirely on a single railroad's line, or Type II trucks might be interchanged between two or more railroads with mutual agreement between the railroads. These are the interchange conditions under which it currently makes sense to talk about the cost-effectiveness of a Type II truck.

Another major hurdle needs to be introduced here. The optimal truck will probably be different depending upon the operating conditions of the railroad being considered. A truck that fixes hunting problems at the expense of curving performance (call it truck A) is not likely to be acceptable to the eastern railroads. This does not mean that there is no role for such a truck. It is possible that applications exist where truck A is the best answer under the interchange conditions discussed in the previous paragraph. If the analysis were performed under the assumption that each truck was approved for interchange service, we would not be evaluating truck A fairly.

Rather than attempt to analyze a situation that doesn't exist, it makes more sense to develop the conditions under which a Type II truck will pay for itself if given limited interchange conditions. How much annual mileage does this type of truck need to be profitable? What percentage of the traffic over a given line does a Type II truck need to contribute before it is reasonable to talk about savings in rail wear? Which car bodies would an improved truck be most profitable under? What mix of curved to tangent track is needed before a given truck pays for itself? It is this sort of issue that TDOP should concern itself with.

INTERPRETATION OF BENEFIT/COST ANALYSIS

The methodology recommended in this plan is to conduct an incremental benefit/cost analysis to evaluate proposed investments in Type II trucks. Railroads follow such procedures in arriving at their profit-making rolling stock investment decisions. However, the data required to establish the operating cost of existing conventional (Type I) trucks for comparing their cost effectiveness with improved (Type II) trucks have never been available in usable form. It has now been established that the basic data may be obtained from the Union Pacific railroad as a result of their ongoing data collection efforts.

There are two broad categories of economic-related data required: 1) the cost data required are the capital investments or purchase prices of the conventional and improved trucks, adjusted for any credits and debits such as investment tax credits or additional working capital requirements. The difference plus any added inventory cost discounted to present value is the net incremental cost; 2) the benefits data required are derived from the actual operating cost of the conventional truck and the estimated operating cost of the improved truck. Their difference is the estimated gross incremental benefit which is adjusted for the income tax shield arising from depreciation allowances and discounted to present value using a railroad's cost rate to acquire the investment funds (i.e., their cost of capital). That calculation produces the estimated net incremental benefit. To measure the cost effectiveness of the improved truck, the estimated net incremental benefit is divided by the actual net incremental cost. Values greater than one provide benefits greater than cost, a profitable investment; less than one, an unprofitable investment; equal to one, an "indifferent" investment.

Values of the benefit/cost ratio greater than one do not necessarily mean a given railroad should make the investment. In principle, each railroad should estimate its own benefit/cost ratios for all the potential investments it has before it. Then, only the most profitable investments should be undertaken. For the purposes of doing trade-off studies, TDOP Phase II will treat a value of one as the point where an investment might reasonably be considered. A given railroad should regard this as the point where it is reasonable to start considering the investment rather than as a recommendation to the industry that the investment be made.

APPENDIX B - PRESENT VALUE OF REPAIRS AT TIME OF PURCHASE

Table of Contents

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR ALL ROLLER BEARING CARS

SUMMARY TABLE	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
	ANNUAL MILEAGE							
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	290.15	625.82	945.44	1200.31	1399.69	1562.28	1696.63	1808.98
COUPLERS, YOKES, & DRAFT GEAR	219.31	628.13	1001.57	1297.39	1519.62	1702.29	1854.00	1983.48
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	335.50	1001.16	1586.24	2039.94	2393.58	2676.25	2906.56	3085.85
OTHER CAR REPAIRS	206.60	543.34	943.65	1278.36	1548.71	1770.56	1958.57	2116.56
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	564.80	1268.56	1974.54	2532.37	2968.22	3328.28	3620.63	3871.12
WHEELSETS	831.31	1823.41	2738.21	3463.36	4028.00	4484.40	4857.31	5173.25
OTHER TRUCK REPAIRS	286.21	1021.20	1625.32	2078.35	2408.49	2682.21	2899.62	3076.94
TOTAL	2733.87	6911.62	10814.96	13890.07	16266.31	18206.27	19793.33	21116.18
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
	ANNUAL MILEAGE							
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	290.15	625.82	945.44	1200.31	1399.69	1562.28	1696.63	1808.98
COT&S	63.26	163.39	251.58	319.79	371.82	414.53	449.04	477.99
IOT&S	122.60	244.45	357.45	447.13	517.22	573.95	620.45	659.97
PRESSURE SYSTEM	86.61	187.01	292.87	379.61	448.87	505.45	553.34	592.60
HAND BRAKES	17.68	30.97	43.54	53.78	61.78	68.35	73.80	78.43
COUPLERS, YOKES, & DRAFT GEAR	219.31	628.13	1001.57	1297.39	1519.62	1702.29	1854.00	1983.48
COUPLER BODIES	141.67	367.77	552.53	694.61	796.31	881.57	951.72	1010.53
COUPLER KNUCKLES	23.02	71.72	118.92	156.95	186.57	210.39	230.35	247.19
OTHER COUPLER PARTS	32.62	91.51	147.97	193.14	228.74	257.13	280.66	300.61
YOKES	8.56	36.83	64.59	87.01	104.05	117.89	129.44	139.87
DRAFT GEARS, CARRIERS, AND FOLLOWERS	13.45	60.30	117.56	165.67	203.95	235.30	261.84	285.28
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	335.50	1001.16	1586.24	2039.94	2393.58	2676.25	2906.56	3085.85
OTHER CAR REPAIRS	206.60	543.34	943.65	1278.36	1548.71	1770.56	1958.57	2116.56
OTHER CAR REPAIRS	137.33	346.00	611.22	834.97	1016.96	1167.60	1295.67	1403.41
WELDING	40.70	128.32	222.97	301.34	364.01	414.47	457.24	492.82
NON BILLABLE INSPECTIONS	28.57	69.03	109.45	142.04	167.74	188.49	205.66	220.33
CAR TOTAL	1051.56	2798.45	4476.90	5816.00	6861.60	7711.37	8415.77	8994.87

TRUCK REPAIRS	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
	ANNUAL MILEAGE							
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	564.80	1268.56	1974.54	2532.37	2968.22	3328.28	3620.63	3871.12
BRAKE BEAMS	108.39	291.22	492.33	650.49	773.27	878.13	961.07	1033.85
BRAKE HEAD WEAR PLATES	0.64	1.46	1.96	2.29	2.48	2.67	2.80	2.90
BRAKE BEAM WEAR PLATES	0.11	0.29	0.41	0.49	0.55	0.60	0.62	0.66
BRAKE BEAM HANGERS	0.01	0.03	0.04	0.04	0.05	0.05	0.05	0.05
BRAKE HANGER BRACKET WEAR PLATE	0.00	0.01	0.01	0.02	0.02	0.02	0.02	0.02
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.00	0.01	0.01	0.02	0.02	0.02	0.02	0.02
BRAKE HANGER OR CONNECTION PIN	2.17	4.80	7.51	9.70	11.42	12.83	14.01	15.00
BOTTOM ROD SAFETY SUPPORT	0.84	2.22	3.83	5.30	6.50	7.48	8.34	9.04
BRAKE BEAM SAFETY SUPPORT	0.26	0.62	0.85	0.99	1.10	1.18	1.21	1.28
BRAKE CONNECTION, BOTTOM	2.06	4.27	6.07	7.50	8.54	9.37	10.12	10.71
BRAKE CONNECTION, TOP	6.35	16.89	24.55	30.29	34.18	37.49	40.18	42.42
BRAKE LEVER	1.42	2.65	3.73	4.58	5.22	5.74	6.18	6.56
BRAKE LEVER GUIDE OR CARRIER	1.77	1.99	2.10	2.18	2.23	2.27	2.32	2.35
DEAD LEVER GUIDE	0.23	0.77	1.23	1.59	1.84	2.05	2.24	2.40
DEAD LEVER GUIDE BRACKET	0.09	0.43	0.72	0.94	1.11	1.23	1.34	1.44
BRAKE SHOES	437.56	935.02	1420.52	1805.10	2107.16	2353.24	2555.10	2726.44
BRAKE SHOE KEYS	2.89	5.89	8.67	10.84	12.53	13.89	15.01	15.95
WHEELSETS	831.31	1823.41	2738.21	3463.36	4028.00	4484.40	4857.31	5173.25
LUBRICATE ROLLER BEARINGS	14.78	34.75	49.70	60.72	69.02	75.42	80.56	84.92
ROLLER BEARINGS	138.42	298.51	446.38	563.74	655.10	729.16	789.61	840.86
ROLLER BEARING CAP SCREWS	0.09	0.25	0.38	0.47	0.54	0.60	0.65	0.68
ROLLER BEARING LOCKING PLATES	0.02	0.08	0.12	0.15	0.16	0.18	0.19	0.19
ROLLER BEARING LUBRICATION FITTING	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
PEDESTAL ADAPTERS	17.54	58.22	97.79	128.95	153.60	173.23	189.36	202.61
WHEELS	308.06	673.76	1010.12	1277.08	1484.45	1652.13	1789.23	1905.80
WHEEL LABOR	341.98	735.43	1100.17	1389.88	1615.84	1798.83	1948.32	2074.92
AXLES, ROLLER BEARINGS	10.41	22.40	33.51	42.35	49.24	54.82	59.37	63.23
OTHER TRUCK REPAIRS	286.21	1021.20	1625.32	2078.35	2408.49	2682.21	2899.62	3076.94
TRUCK BOLSTERS	37.53	172.00	302.41	402.82	481.61	545.77	596.80	637.73
TRUCK BOLSTERS (REPAIRED)	0.59	3.82	7.34	10.41	12.92	14.95	16.48	18.00
CENTER PINS	1.90	5.49	9.36	12.47	14.94	16.97	18.63	20.06
CENTER PLATES	1.25	3.53	7.06	9.97	12.35	14.32	16.01	17.64
CENTER PLATE LINERS	4.60	19.73	35.54	48.60	58.88	67.23	74.05	80.14
TRUCK SIDE BEARINGS	5.10	14.82	23.42	30.06	35.19	39.31	42.61	45.35
FRICTION CASTINGS	9.97	38.77	65.95	87.37	104.07	117.42	128.21	137.47
SIDE BEARING SHIM	0.33	2.25	3.91	5.28	6.22	6.92	7.59	8.18
SIDE FRAMES	208.61	692.81	1057.60	1324.18	1508.81	1665.31	1788.45	1887.55
SIDE FRAMES (REPAIRED)	0.74	5.81	9.86	13.04	15.24	17.06	18.36	19.69
SPRING PLANKS	0.29	0.82	1.17	1.42	1.58	1.70	1.80	1.88
SPRING PLANKS (REPAIRED)	0.11	0.51	0.79	1.00	1.14	1.24	1.34	1.42
OUTER SPRINGS	6.64	27.57	45.16	58.49	68.68	76.52	82.98	88.28
INNER SPRINGS	3.36	12.92	21.14	27.34	32.19	35.88	38.92	41.38
STABILIZER SPRINGS	2.28	9.30	15.71	20.64	24.43	27.44	29.88	31.90
TRUCK SPRING FRICTION SNUBBER	0.04	0.08	0.11	0.13	0.14	0.15	0.16	0.17
TRUCK SPRING PLATES	0.02	0.06	0.09	0.11	0.13	0.13	0.15	0.15
TRUCK SPRING SHIM, WOOD	0.04	0.20	0.31	0.39	0.45	0.49	0.53	0.55
STEEL	0.32	1.38	2.14	2.70	3.08	3.38	3.67	3.87
MANUFACTURED MATERIAL (TRUCK)	2.49	9.34	16.25	21.92	26.44	30.01	33.00	35.54
TRUCK TOTAL	1682.31	4113.17	6338.06	8074.07	9404.71	10494.89	11377.56	12121.31

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR ALL ROLLER BEARING WITH ANNUAL MILEAGE BETWEEN 37500 AND 100000 MILES

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	125.89	277.17	455.95	604.17	723.09	821.69	904.72	973.88
COUPLERS, YOKES, & DRAFT GEAR	99.90	264.44	445.40	596.84	710.39	805.04	890.29	960.01
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	87.97	242.72	443.05	611.90	748.99	857.99	955.22	1034.69
OTHER CAR REPAIRS	90.04	279.19	578.17	833.58	1045.22	1221.41	1373.36	1499.95
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	356.29	756.22	1250.34	1643.66	1962.16	2228.98	2446.76	2631.58
WHEELSETS	615.88	1293.65	1949.40	2471.51	2874.51	3204.43	3479.57	3710.58
OTHER TRUCK REPAIRS	32.55	101.73	192.85	260.23	314.45	358.94	395.95	425.20
TOTAL	1408.51	3215.12	5315.16	7021.88	8378.81	9498.46	10445.86	11235.87
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	125.89	277.17	455.95	604.17	723.09	821.69	904.72	973.88
COT&S	9.18	25.36	48.14	67.37	82.59	95.83	106.55	116.37
IDT&S	67.14	144.11	222.52	285.42	335.00	375.55	409.14	437.19
PRESSURE SYSTEM	45.06	98.16	169.23	229.74	279.34	320.33	355.79	384.45
HAND BRAKES	4.51	9.55	16.07	21.64	26.17	29.98	33.24	35.87
COUPLERS, YOKES, & DRAFT GEAR	99.90	264.44	445.40	596.84	710.39	805.04	890.29	960.01
COUPLER BODIES	60.74	144.79	231.52	303.14	355.45	399.61	439.42	472.75
COUPLER KNUCKLES	14.39	36.14	60.37	80.53	96.66	109.64	120.97	130.07
OTHER COUPLER PARTS	20.03	45.69	74.15	97.86	116.64	131.99	145.46	155.96
YOKES	1.22	8.71	17.27	24.41	29.55	33.87	37.84	41.05
DRAFT GEARS, CARRIERS, AND FOLLOWERS	3.52	29.11	62.11	90.89	112.09	129.93	146.60	160.17
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	87.97	242.72	443.05	611.90	748.99	857.99	955.22	1034.69
OTHER CAR REPAIRS	90.04	279.19	578.17	833.58	1045.22	1221.41	1373.36	1499.95
OTHER CAR REPAIRS	55.13	176.71	386.18	566.17	716.16	841.58	950.40	1041.46
WELDING	19.55	64.66	127.31	180.84	224.61	260.76	291.99	317.02
NON BILLABLE INSPECTIONS	15.36	37.82	64.69	86.57	104.45	119.07	130.98	141.46
CAR TOTAL	403.80	1063.52	1922.57	2646.49	3227.69	3706.12	4123.59	4468.52

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	356.29	756.22	1250.34	1643.66	1962.16	2228.98	2446.76	2631.58
BRAKE BEAMS	27.11	66.88	161.70	236.19	299.52	354.46	397.92	436.57
BRAKE HEAD WEAR PLATES	0.00	0.12	0.19	0.25	0.28	0.31	0.34	0.36
BRAKE BEAM WEAR PLATES	0.01	0.03	0.04	0.05	0.06	0.07	0.07	0.07
BRAKE HANGER OR CONNECTION PIN	1.71	3.28	5.13	6.65	7.84	8.84	9.71	10.40
BOTTOM ROD SAFETY SUPPORT	0.69	1.92	3.46	4.88	6.04	6.98	7.85	8.50
BRAKE BEAM SAFETY SUPPORT	0.01	0.04	0.06	0.08	0.09	0.10	0.10	0.11
BRAKE CONNECTION, BOTTOM	1.35	2.72	3.86	4.76	5.44	5.98	6.45	6.78
BRAKE CONNECTION, TOP	1.15	4.59	7.71	10.10	12.00	13.46	14.65	15.66
BRAKE LEVER	0.93	1.83	2.64	3.28	3.76	4.14	4.49	4.74
BRAKE LEVER GUIDE OR CARRIER	0.00	0.02	0.04	0.05	0.06	0.07	0.08	0.09
DEAD LEVER GUIDE	0.02	0.04	0.13	0.20	0.26	0.30	0.35	0.39
DEAD LEVER GUIDE BRACKET	0.04	0.11	0.19	0.25	0.31	0.35	0.37	0.41
BRAKE SHOES	322.18	672.23	1061.23	1371.72	1620.33	1826.93	1996.71	2139.27
BRAKE SHOE KEYS	1.09	2.42	3.96	5.19	6.17	6.98	7.66	8.22
WHEELSETS	615.88	1293.65	1949.40	2471.51	2874.51	3204.43	3479.57	3710.58
LUBRICATE ROLLER BEARINGS	7.30	20.58	30.84	38.72	44.32	48.76	52.43	55.88
ROLLER BEARINGS	104.20	216.34	324.72	411.14	477.87	532.56	578.18	616.39
ROLLER BEARING CAP SCREWS	0.09	0.21	0.29	0.34	0.38	0.41	0.43	0.45
ROLLER BEARING LOCKING PLATES	0.00	0.01	0.02	0.02	0.02	0.02	0.03	0.03
ROLLER BEARING LUBRICATION FITTING	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	7.50	26.14	48.01	65.29	79.13	90.28	99.60	107.47
WHEELS	229.60	476.32	713.77	902.66	1048.36	1167.70	1267.17	1350.50
WHEEL LABOR	259.28	537.66	807.13	1022.15	1188.17	1324.30	1437.85	1533.09
AXLES, ROLLER BEARINGS	7.90	16.39	24.61	31.17	36.23	40.39	43.85	46.76
OTHER TRUCK REPAIRS	32.55	101.73	192.85	260.23	314.45	358.94	395.95	425.20
TRUCK BOLSTERS	7.78	21.90	45.91	63.10	77.78	89.96	99.07	107.16
TRUCK BOLSTERS (REPAIRED)	0.00	0.44	1.20	1.87	2.54	3.11	3.49	3.88
CENTER PINS	0.92	2.24	3.84	5.11	6.12	6.97	7.66	8.29
CENTER PLATES	0.00	0.07	0.20	0.31	0.40	0.48	0.56	0.60
CENTER PLATE LINERS	2.63	10.42	18.71	25.26	30.65	35.06	38.25	41.43
TRUCK SIDE BEARINGS	1.38	3.95	6.85	9.02	10.88	12.30	13.36	14.42
FRICTION CASTINGS	3.37	7.19	11.92	15.44	18.25	20.55	22.40	24.02
SIDE BEARING SHIM	0.01	0.12	0.31	0.47	0.59	0.70	0.79	0.86
SIDE FRAMES	5.58	32.11	69.42	96.94	118.29	135.81	152.22	163.25
SIDE FRAMES (REPAIRED)	0.00	0.10	0.24	0.35	0.47	0.57	0.63	0.69
SPRING PLANKS	0.00	0.00	0.01	0.01	0.02	0.03	0.03	0.04
SPRING PLANKS (REPAIRED)	0.00	0.00	0.01	0.01	0.02	0.02	0.03	0.03
OUTER SPRINGS	4.73	8.64	12.13	14.49	16.33	17.89	19.03	19.92
INNER SPRINGS	2.67	5.11	7.17	8.58	9.66	10.54	11.23	11.72
STABILIZER SPRINGS	0.83	1.94	3.01	3.79	4.34	4.81	5.17	5.51
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
TRUCK SPRING SHIM, WOOD	0.00	0.00	0.00	0.00	0.01	0.01	0.01	0.01
STEEL	0.48	1.14	1.57	1.91	2.15	2.21	2.44	2.49
MANUFACTURED MATERIAL (TRUCK)	1.95	6.33	10.34	13.54	15.93	17.68	19.55	20.84
TRUCK TOTAL	1004.72	2151.60	3392.58	4375.39	5151.12	5792.34	6322.27	6767.35

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ROLLER BEARING CARS WITH ANNUAL MILEAGE LESS THAN 25000 MILES

SUMMARY TABLE	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	481.49	986.81	1454.27	1822.79	2112.29	2346.31	2539.90	2701.19
COUPLERS, YOKES, & DRAFT GEAR	290.52	806.87	1284.61	1658.53	1946.04	2181.13	2371.67	2537.77
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	452.67	1065.40	1576.20	1968.33	2270.04	2506.69	2703.42	2858.21
OTHER CAR REPAIRS	350.53	839.02	1386.31	1838.32	2200.58	2496.24	2746.32	2955.70
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	725.29	1591.48	2455.44	3134.21	3664.78	4106.76	4460.67	4766.74
WHEELSETS	953.60	2206.37	3338.92	4226.28	4917.20	5478.44	5931.17	6313.21
OTHER TRUCK REPAIRS	421.87	1311.88	2071.83	2639.00	3077.14	3433.17	3708.28	3943.01
TOTAL	3675.98	8807.84	13567.57	17287.45	20188.06	22548.73	24461.43	26075.82
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	481.49	986.81	1454.27	1822.79	2112.29	2346.31	2539.90	2701.19
COT&S	124.89	296.05	442.48	553.35	640.36	709.57	765.87	812.45
IDT&S	187.82	359.96	517.52	642.19	739.23	818.37	882.70	937.42
PRESSURE SYSTEM	132.25	265.24	402.12	513.52	602.47	674.60	736.06	786.60
HAND BRAKES	36.53	65.56	92.16	113.73	130.24	143.78	155.26	164.72
COUPLERS, YOKES, & DRAFT GEAR	290.52	806.87	1284.61	1658.53	1946.04	2181.13	2371.67	2537.77
COUPLER BODIES	165.69	418.64	623.98	777.53	890.69	984.05	1057.91	1122.61
COUPLER KNUCKLES	35.35	101.05	163.57	213.37	252.36	283.77	309.74	331.76
OTHER COUPLER PARTS	52.00	139.71	222.62	288.61	340.53	382.05	416.43	445.34
YOKES	14.59	51.69	89.20	119.03	142.47	161.81	176.89	191.00
DRAFT GEARS, CARRIERS, AND FOLLOWERS	22.89	95.78	185.24	259.98	319.98	369.44	410.70	447.07
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	452.67	1065.40	1576.20	1968.33	2270.04	2506.69	2703.42	2858.21
OTHER CAR REPAIRS	350.53	839.02	1386.31	1838.32	2200.58	2496.24	2746.32	2955.70
OTHER CAR REPAIRS	233.25	527.71	873.96	1161.43	1392.68	1583.18	1744.43	1880.40
WELDING	69.45	202.14	342.68	458.44	551.11	625.13	688.09	740.08
NON BILLABLE INSPECTIONS	47.83	109.16	169.68	218.45	256.79	287.92	313.80	335.23
CAR TOTAL	1575.21	3698.11	5701.39	7287.96	8528.95	9530.36	10361.30	11052.87

TRUCK REPAIRS	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	725.29	1591.48	2455.44	3134.21	3664.78	4106.76	4460.67	4766.74
BRAKE BEAMS	156.06	396.11	661.05	867.54	1029.54	1168.88	1276.84	1372.50
BRAKE HEAD WEAR PLATES	0.10	0.26	0.35	0.41	0.45	0.48	0.50	0.53
BRAKE BEAM WEAR PLATES	0.22	0.59	0.83	1.01	1.11	1.20	1.23	1.33
BRAKE BEAM HANGERS	0.03	0.07	0.09	0.10	0.11	0.11	0.11	0.12
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.01	0.01	0.02	0.02	0.02	0.03	0.03	0.03
BRAKE HANGER OR CONNECTION PIN	2.65	5.98	9.38	12.11	14.27	16.05	17.55	18.77
BOTTOM ROD SAFETY SUPPORT	0.75	1.89	3.19	4.34	5.31	6.09	6.76	7.33
BRAKE BEAM SAFETY SUPPORT	0.55	1.22	1.62	1.85	2.04	2.18	2.22	2.35
BRAKE CONNECTION, BOTTOM	2.21	5.01	7.36	9.23	10.62	11.71	12.68	13.53
BRAKE CONNECTION, TOP	6.66	16.25	23.33	28.55	32.32	35.35	37.72	40.11
BRAKE LEVER	1.75	3.07	4.25	5.16	5.87	6.44	6.91	7.32
BRAKE LEVER GUIDE OR CARRIER	0.09	0.22	0.31	0.36	0.41	0.45	0.47	0.49
DEAD LEVER GUIDE	0.33	0.96	1.51	1.96	2.24	2.48	2.72	2.91
DEAD LEVER GUIDE BRACKET	0.14	0.81	1.39	1.85	2.15	2.41	2.67	2.84
BRAKE SHOES	549.06	1150.15	1728.07	2184.05	2540.36	2833.10	3070.92	3273.97
BRAKE SHOE KEYS	4.69	8.89	12.69	15.65	17.95	19.81	21.34	22.61
WHEELSETS	953.60	2206.37	3338.92	4226.28	4917.20	5478.44	5931.17	6313.21
LUBRICATE ROLLER BEARINGS	25.53	53.15	73.32	87.91	99.17	107.65	114.15	120.13
ROLLER BEARINGS	154.89	354.35	534.85	676.42	786.51	876.28	948.55	1009.58
ROLLER BEARING CAP SCREWS	0.09	0.29	0.47	0.59	0.69	0.78	0.85	0.88
ROLLER BEARING LOCKING PLATES	0.04	0.14	0.21	0.25	0.28	0.31	0.33	0.34
ROLLER BEARING LUBRICATION FITTING	0.02	0.03	0.03	0.04	0.04	0.04	0.05	0.05
PEDESTAL ADAPTERS	35.08	99.35	160.91	208.86	247.33	278.04	303.02	323.26
WHEELS	343.39	796.03	1204.44	1524.87	1773.62	1975.83	2139.05	2277.36
WHEEL LABOR	382.93	876.38	1324.39	1676.31	1950.18	2173.33	2353.52	2505.33
AXLES, ROLLER BEARINGS	11.63	26.65	40.30	51.02	59.37	66.17	71.66	76.29
OTHER TRUCK REPAIRS	421.87	1311.88	2071.83	2639.00	3077.14	3433.17	3708.28	3943.01
TRUCK BOLSTERS	83.46	287.85	486.70	638.72	760.89	861.33	940.20	1000.74
TRUCK BOLSTERS (REPAIRED)	1.57	6.27	11.21	15.57	19.03	21.77	24.07	26.09
CENTER PINS	3.30	8.77	14.61	19.26	23.00	26.08	28.62	30.74
CENTER PLATES	2.50	6.40	12.23	17.00	20.94	24.18	26.95	29.59
CENTER PLATE LINERS	9.02	33.49	58.67	79.43	95.77	109.08	120.00	129.42
TRUCK SIDE BEARINGS	10.58	27.11	41.69	52.93	61.72	68.82	74.56	79.07
FRICTION CASTINGS	20.65	68.49	112.82	147.63	174.90	196.89	214.78	229.60
SIDE BEARING SHIM	0.36	3.12	5.28	7.03	8.26	9.17	10.00	10.83
SIDE FRAMES	257.38	748.46	1129.51	1403.21	1608.30	1775.55	1899.81	2014.19
SIDE FRAMES (REPAIRED)	1.26	11.16	19.30	25.66	30.22	34.18	36.77	39.12
SPRING PLANKS	0.62	1.15	1.43	1.67	1.76	1.81	1.91	1.99
SPRING PLANKS (REPAIRED)	0.17	0.87	1.37	1.75	2.02	2.18	2.37	2.55
OUTER SPRINGS	13.86	52.32	84.75	109.18	128.24	142.89	154.87	164.29
INNER SPRINGS	7.14	24.98	40.35	51.88	61.13	68.09	73.81	78.31
STABILIZER SPRINGS	5.15	16.21	26.16	33.75	39.73	44.44	48.30	51.39
TRUCK SPRING FRICTION SNUBBER	0.12	0.24	0.32	0.36	0.40	0.44	0.46	0.47
TRUCK SPRING PLATES	0.06	0.10	0.13	0.15	0.16	0.17	0.18	0.18
TRUCK SPRING SHIM, WOOD	0.04	0.27	0.44	0.56	0.65	0.71	0.76	0.81
STEEL	0.17	0.46	0.67	0.82	0.91	0.99	1.04	1.12
MANUFACTURED MATERIAL (TRUCK)	4.25	14.15	24.21	32.43	39.11	44.42	48.81	52.49
TRUCK TOTAL	2100.77	5109.73	7866.18	9999.49	11659.12	13018.37	14100.12	15022.96

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ROLLER BEARING CARS WITH ANNUAL MILEAGE BETWEEN 12500 AND 37500 MILES

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	286.79	668.00	1030.73	1322.19	1550.43	1735.77	1890.72	2017.61
COUPLERS, YOKES, & DRAFT GEAR	268.70	766.05	1241.50	1625.34	1921.67	2161.41	2364.13	2532.43
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	391.00	1212.69	1922.53	2472.33	2901.55	3252.42	3530.85	3736.16
OTHER CAR REPAIRS	205.62	560.99	937.19	1253.17	1502.84	1705.72	1880.96	2024.88
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	593.86	1380.53	2101.73	2679.98	3123.54	3485.31	3785.42	4035.99
WHEELSETS	958.51	2006.35	2946.07	3695.74	4277.79	4751.27	5132.08	5453.86
OTHER TRUCK REPAIRS	321.94	1281.78	2090.25	2709.22	3158.60	3537.33	3841.09	4078.14
TOTAL	3026.42	7876.38	12269.99	15757.97	18436.43	20629.23	22425.24	23879.07
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	286.79	668.00	1030.73	1322.19	1550.43	1735.77	1890.72	2017.61
COT&S	72.66	202.03	324.04	420.41	495.72	557.68	608.20	649.43
IDT&S	114.23	250.18	376.53	478.54	558.29	622.13	675.95	720.49
PRESSURE SYSTEM	88.19	190.51	291.85	374.37	439.41	491.95	537.19	573.43
HAND BRAKES	11.71	25.27	38.31	48.87	57.02	64.00	69.39	74.26
COUPLERS, YOKES, & DRAFT GEAR	268.70	766.05	1241.50	1625.34	1921.67	2161.41	2364.13	2532.43
COUPLER BODIES	188.30	442.25	641.29	793.63	901.38	992.91	1066.44	1128.80
COUPLER KNUCKLES	24.58	81.71	138.93	185.53	222.31	251.69	276.33	297.08
OTHER COUPLER PARTS	51.95	103.26	175.54	234.79	282.16	319.52	351.51	377.80
YOKES	9.45	51.36	98.50	138.05	170.23	194.94	217.15	235.12
DRAFT GEARS, CARRIERS, AND FOLLOWERS	14.43	87.47	187.24	273.33	345.59	402.34	452.70	493.64
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	391.00	1212.69	1922.53	2472.33	2901.55	3252.42	3530.85	3736.16
OTHER CAR REPAIRS	205.62	560.99	937.19	1253.17	1502.84	1705.72	1880.96	2024.88
OTHER CAR REPAIRS	133.95	344.31	570.89	760.61	910.71	1034.57	1139.59	1226.86
WELDING	45.24	145.03	251.19	342.21	414.24	470.88	522.81	563.94
NON BILLABLE INSPECTIONS	26.44	71.66	115.11	150.35	177.89	200.27	218.56	234.08
CAR TOTAL	1152.11	3207.73	5131.95	6673.03	7876.50	8855.31	9666.66	10311.08

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	593.86	1380.53	2101.73	2679.98	3123.54	3485.31	3785.42	4035.99
BRAKE BEAMS	137.57	356.60	548.75	703.83	816.95	912.65	992.41	1058.93
BRAKE HEAD WEAR PLATES	0.90	1.88	2.47	2.86	3.07	3.30	3.45	3.56
BRAKE BEAM WEAR PLATES	0.17	0.36	0.49	0.58	0.64	0.69	0.71	0.76
BRAKE BEAM HANGERS	0.02	0.04	0.05	0.05	0.06	0.06	0.06	0.06
BRAKE HANGER BRACKET WEAR PLATE	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.00	0.01	0.02	0.02	0.02	0.02	0.02	0.02
BRAKE HANGER OR CONNECTION PIN	1.91	4.64	7.22	9.31	10.90	12.21	13.30	14.19
BOTTOM ROD SAFETY SUPPORT	0.93	1.80	2.59	3.25	3.76	4.19	4.54	4.84
BRAKE BEAM SAFETY SUPPORT	0.28	0.68	0.93	1.09	1.20	1.28	1.32	1.40
BRAKE CONNECTION, BOTTOM	2.51	4.81	6.60	8.02	9.01	9.82	10.54	11.14
BRAKE CONNECTION, TOP	8.63	20.54	28.71	34.75	38.69	42.14	44.88	47.22
BRAKE LEVER	1.41	2.69	3.82	4.69	5.34	5.87	6.32	6.71
BRAKE LEVER GUIDE OR CARRIER	4.12	4.48	4.59	4.69	4.75	4.80	4.84	4.88
DEAD LEVER GUIDE	0.30	0.95	1.51	1.96	2.26	2.50	2.74	2.94
DEAD LEVER GUIDE BRACKET	0.09	0.48	0.82	1.08	1.26	1.41	1.54	1.66
BRAKE SHOES	432.52	974.50	1483.86	1891.94	2211.75	2468.87	2681.95	2859.76
BRAKE SHOE KEYS	2.50	6.06	9.30	11.88	13.88	15.47	16.79	17.89
WHEELSETS	958.51	2006.35	2946.07	3695.74	4277.79	4751.27	5132.08	5453.86
LUBRICATE ROLLER BEARINGS	16.02	36.62	51.99	63.24	71.75	78.35	83.55	87.92
ROLLER BEARINGS	160.40	323.40	467.52	581.88	670.11	742.45	800.11	849.10
ROLLER BEARING CAP SCREWS	0.08	0.26	0.42	0.54	0.63	0.71	0.77	0.82
ROLLER BEARING LOCKING PLATES	0.02	0.08	0.13	0.16	0.17	0.19	0.20	0.21
ROLLER BEARING LUBRICATION FITTING	0.01	0.01	0.02	0.02	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	18.06	64.10	106.40	140.03	165.94	186.93	203.95	217.62
WHEELS	356.38	762.05	1133.88	1433.59	1668.21	1857.63	2011.75	2141.93
WHEEL LABOR	395.51	795.61	1150.66	1432.62	1650.64	1829.24	1971.63	2092.46
AXLES, ROLLER BEARINGS	12.03	24.22	35.05	43.65	50.30	55.75	60.09	63.78
OTHER TRUCK REPAIRS	321.94	1281.78	2090.25	2709.22	3158.60	3537.33	3841.09	4078.14
TRUCK BOLSTERS	33.74	220.26	404.47	549.95	662.37	756.04	831.70	888.54
TRUCK BOLSTERS (REPAIRED)	0.49	5.72	11.75	17.11	21.50	25.08	27.84	30.43
CENTER PINS	1.88	7.02	13.29	18.53	22.84	26.24	29.23	31.57
CENTER PLATES	1.41	4.21	10.06	14.95	19.00	22.38	25.31	28.16
CENTER PLATE LINERS	4.44	23.52	44.78	62.65	76.90	88.48	98.01	106.40
TRUCK SIDE BEARINGS	5.33	17.71	28.62	37.17	43.66	49.01	53.28	56.60
FRICTION CASTINGS	9.00	47.90	86.60	117.91	142.27	162.16	178.25	191.50
SIDE BEARING SHIM	0.34	3.02	5.41	7.43	8.83	9.81	10.79	11.69
SIDE FRAMES	249.56	866.02	1334.53	1682.13	1921.34	2127.28	2290.77	2416.97
SIDE FRAMES (REPAIRED)	0.72	8.17	14.36	19.38	22.84	25.70	27.84	29.84
SPRING PLANKS	0.40	1.02	1.43	1.73	1.91	2.04	2.16	2.26
SPRING PLANKS (REPAIRED)	0.12	0.59	0.92	1.17	1.34	1.46	1.58	1.68
OUTER SPRINGS	6.41	36.13	62.31	82.68	98.17	110.55	120.59	128.61
INNER SPRINGS	3.32	16.58	28.53	37.74	44.91	50.55	55.16	58.74
STABILIZER SPRINGS	2.06	11.86	21.51	29.10	34.91	39.64	43.52	46.55
TRUCK SPRING FRICTION SNUBBER	0.02	0.07	0.10	0.12	0.13	0.14	0.15	0.16
TRUCK SPRING PLATES	0.01	0.07	0.10	0.13	0.14	0.16	0.17	0.18
TRUCK SPRING SHIM, WOOD	0.05	0.24	0.38	0.47	0.54	0.59	0.63	0.67
STEEL	0.32	1.53	2.39	3.03	3.46	3.81	4.14	4.37
MANUFACTURED MATERIAL (TRUCK)	2.31	10.13	18.71	25.83	31.55	36.19	39.97	43.23
TRUCK TOTAL	1874.31	4668.66	7138.04	9084.94	10559.93	11773.91	12758.58	13567.99

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ALL ROLLER BEARING CARS WITH ANNUAL MILEAGE BETWEEN 25000 AND 50000 MILES

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	211.33	522.86	845.33	1109.76	1316.25	1487.98	1631.99	1749.25
COUPLERS, YOKES, & DRAFT GEAR	225.74	652.69	1060.09	1390.51	1636.45	1841.89	2016.93	2162.23
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	315.46	1158.96	1913.32	2501.51	2960.18	3338.14	3638.56	3862.47
OTHER CAR REPAIRS	145.54	476.27	898.68	1260.95	1553.40	1795.74	2005.65	2176.76
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	562.41	1262.80	1913.47	2434.63	2834.18	3161.65	3433.06	3660.54
WHEELSETS	952.37	1935.64	2859.78	3608.72	4194.50	4668.66	5061.58	5389.04
OTHER TRUCK REPAIRS	248.83	1113.99	1829.18	2378.83	2766.98	3099.36	3370.00	3574.33
TOTAL	2661.69	7123.20	11319.86	14684.92	17261.94	19393.42	21157.76	22574.60
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	211.33	522.86	845.33	1109.76	1316.25	1487.98	1631.99	1749.25
COT&S	40.26	125.22	211.95	281.61	334.18	379.58	416.84	446.81
IDT&S	93.78	207.25	316.41	404.90	474.26	530.43	577.36	616.32
PRESSURE SYSTEM	70.06	174.22	290.04	387.15	464.27	528.17	583.16	626.84
HAND BRAKES	7.23	16.16	26.93	36.10	43.54	49.80	54.93	59.29
COUPLERS, YOKES, & DRAFT GEAR	225.74	652.69	1060.09	1390.51	1636.45	1841.89	2016.93	2162.23
COUPLER BODIES	168.60	413.25	619.32	782.17	896.42	993.94	1077.55	1145.69
COUPLER KNUCKLES	18.59	67.95	118.19	159.54	191.75	217.95	240.15	258.44
OTHER COUPLER PARTS	23.99	79.48	137.65	185.72	223.77	254.42	280.38	301.71
YOKES	5.86	38.54	71.58	99.05	119.58	136.56	151.17	163.92
DRAFT GEARS, CARRIERS, AND FOLLOWERS	8.69	53.46	113.14	164.03	204.93	239.02	267.67	292.46
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	315.46	1158.96	1913.32	2501.51	2960.18	3338.14	3638.56	3862.47
OTHER CAR REPAIRS	145.54	476.27	898.68	1260.95	1553.40	1795.74	2005.65	2176.76
OTHER CAR REPAIRS	96.40	302.00	584.22	828.11	1026.32	1191.91	1336.01	1453.37
WELDING	29.79	118.11	219.93	306.68	375.99	432.62	481.48	520.84
NON BILLABLE INSPECTIONS	19.35	56.15	94.52	126.16	151.09	171.20	188.16	202.55
CAR TOTAL	898.07	2810.77	4717.42	6262.73	7466.28	8463.74	9293.12	9950.70

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	562.41	1262.80	1913.47	2434.63	2834.18	3161.65	3433.06	3660.54
BRAKE BEAMS	116.91	308.43	474.22	608.77	706.61	789.17	858.54	915.98
BRAKE HEAD WEAR PLATES	1.00	2.28	3.05	3.58	3.87	4.18	4.38	4.56
BRAKE BEAM WEAR PLATES	0.14	0.24	0.31	0.36	0.41	0.44	0.45	0.48
BRAKE BEAM HANGERS	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
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	0.60	0.01	0.01	0.01	0.02	0.02	0.02	0.02
BRAKE HANGER OR CONNECTION PIN	2.19	5.06	8.03	10.50	12.41	14.00	15.35	16.44
BOTTOM ROD SAFETY SUPPORT	1.24	3.06	5.26	7.29	8.93	10.29	11.51	12.47
BRAKE BEAM SAFETY SUPPORT	0.08	0.25	0.36	0.44	0.49	0.52	0.55	0.58
BRAKE CONNECTION, BOTTOM	2.34	4.60	6.51	8.06	9.17	10.08	10.93	11.54
BRAKE CONNECTION, TOP	7.23	19.99	29.16	36.22	40.77	44.78	48.22	50.90
BRAKE LEVER	1.22	2.61	3.90	4.94	5.70	6.32	6.88	7.34
BRAKE LEVER GUIDE OR CARRIER	5.82	6.29	6.44	6.58	6.66	6.72	6.79	6.86
DEAD LEVER GUIDE	0.23	0.81	1.34	1.75	2.06	2.30	2.52	2.70
DEAD LEVER GUIDE BRACKET	0.08	0.32	0.55	0.71	0.85	0.95	1.02	1.11
BRAKE SHOES	422.01	904.06	1366.89	1735.87	2025.07	2259.38	2452.32	2615.07
BRAKE SHOE KEYS	1.91	4.76	7.41	9.51	11.13	12.44	13.52	14.43
WHEELSETS	952.37	1935.64	2859.78	3608.72	4194.50	4668.66	5061.58	5389.04
LUBRICATE ROLLER BEARINGS	12.32	30.88	45.72	56.94	65.36	72.07	77.61	82.01
ROLLER BEARINGS	160.54	319.12	468.10	588.93	683.46	760.13	823.64	876.64
ROLLER BEARING CAP SCREWS	0.09	0.26	0.41	0.52	0.60	0.66	0.71	0.76
ROLLER BEARING LOCKING PLATES	0.02	0.06	0.09	0.11	0.13	0.14	0.15	0.15
ROLLER BEARING LUBRICATION FITTING	0.00	0.01	0.01	0.02	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	12.05	51.98	92.33	125.12	150.87	171.63	188.99	202.74
WHEELS	357.72	723.92	1066.19	1343.86	1560.37	1735.51	1880.78	2002.60
WHEEL LABOR	397.53	785.48	1151.84	1449.07	1682.44	1871.48	2027.69	2158.35
AXLES, ROLLER BEARINGS	12.11	23.93	35.09	44.15	51.26	57.02	61.79	65.76
OTHER TRUCK REPAIRS	248.83	1113.99	1829.18	2378.83	2766.98	3099.36	3370.00	3574.33
TRUCK BOLSTERS	22.49	170.45	317.09	432.66	522.11	596.62	656.16	700.99
TRUCK BOLSTERS (REPAIRED)	0.36	4.44	9.22	13.40	16.89	19.74	21.82	23.88
CENTER PINS	1.34	5.02	9.44	13.06	15.96	18.39	20.35	22.02
CENTER PLATES	0.51	2.25	6.19	9.55	12.35	14.70	16.73	18.67
CENTER PLATE LINERS	2.88	17.79	35.52	50.57	62.56	72.37	80.49	87.69
TRUCK SIDE BEARINGS	2.87	12.14	20.65	27.39	32.58	36.84	40.24	42.92
FRICTION CASTINGS	6.24	34.64	64.67	88.96	108.28	124.03	136.52	146.78
SIDE BEARING SHIM	0.25	2.54	4.65	6.45	7.71	8.61	9.50	10.27
SIDE FRAMES	202.37	804.32	1253.77	1591.23	1814.36	2011.00	2172.34	2290.14
SIDE FRAMES (REPAIRED)	0.50	5.27	9.23	12.53	14.74	16.46	17.90	19.33
SPRING PLANKS	0.17	0.79	1.24	1.54	1.77	1.95	2.06	2.17
SPRING PLANKS (REPAIRED)	0.06	0.35	0.56	0.70	0.60	0.88	0.94	0.98
OUTER SPRINGS	3.46	23.32	41.13	54.98	65.50	73.83	80.68	86.01
INNER SPRINGS	1.82	10.72	19.00	25.38	30.30	34.26	37.47	39.89
STABILIZER SPRINGS	1.34	6.78	16.24	22.07	26.58	30.28	33.22	35.49
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.03	0.03	0.04	0.04	0.04	0.04
TRUCK SPRING PLATES	0.00	0.07	0.11	0.15	0.17	0.18	0.20	0.22
TRUCK SPRING SHIM, WOOD	0.05	0.19	0.29	0.36	0.40	0.44	0.48	0.50
STEEL	0.31	1.97	3.17	4.07	4.67	5.16	5.63	5.97
MANUFACTURED MATERIAL (TRUCK)	1.80	8.92	16.97	23.74	29.20	33.59	37.24	40.36
TRUCK TOTAL	1763.62	4312.43	6602.44	8422.19	9795.66	10929.66	11864.64	12623.90

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ALL ROLLER BEARING CARS WITH ANNUAL MILEAGE BETWEEN 37500 AND 62500 MILES

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	142.62	300.38	487.46	643.77	768.97	872.93	960.16	1033.50
COUPLERS, YOKES, & DRAFT GEAR	54.08	221.90	408.67	565.47	682.80	780.44	868.13	940.40
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	236.32	393.79	595.15	766.12	905.86	1018.16	1116.90	1199.12
OTHER CAR REPAIRS	116.66	320.56	637.83	911.13	1136.67	1323.74	1485.59	1621.10
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	516.06	940.15	1454.57	1867.05	2200.85	2479.70	2708.75	2902.99
WHEELSETS	717.26	1374.49	2011.03	2517.78	2910.06	3231.09	3498.02	3723.24
OTHER TRUCK REPAIRS	32.66	105.14	198.85	268.47	324.20	369.55	407.81	437.67
TOTAL	1815.66	3656.42	5793.55	7539.78	8929.40	10075.62	11045.35	11858.01
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	142.62	300.38	487.46	643.77	768.97	872.93	960.16	1033.50
COT&S	4.54	21.26	45.17	65.44	81.45	95.56	106.77	117.17
IDT&S	99.26	179.62	261.09	327.06	379.10	421.68	457.00	486.63
PRESSURE SYSTEM	37.41	93.00	167.89	232.13	284.55	327.82	365.13	395.67
HAND BRAKES	1.42	6.50	13.31	19.14	23.87	27.87	31.25	34.02
COUPLERS, YOKES, & DRAFT GEAR	54.08	221.90	408.67	565.47	682.80	780.44	868.13	940.40
COUPLER BODIES	14.54	98.99	187.78	261.12	314.60	359.48	400.05	433.97
COUPLER KNUCKLES	14.99	36.46	61.01	81.59	98.06	111.57	123.02	132.53
OTHER COUPLER PARTS	19.81	47.23	77.42	102.65	122.59	138.80	153.10	164.26
YOKES	1.27	9.14	18.10	25.64	31.06	35.56	39.72	43.11
DRAFT GEARS, CARRIERS, AND FOLLOWERS	3.47	30.09	64.37	94.46	116.49	135.04	152.24	166.52
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	236.32	393.79	595.15	766.12	905.86	1018.16	1116.90	1199.12
OTHER CAR REPAIRS	116.66	320.56	637.83	911.13	1136.67	1323.74	1485.59	1621.10
OTHER CAR REPAIRS	69.83	200.12	421.04	612.37	771.24	903.68	1018.85	1115.68
WELDING	21.44	70.87	138.74	197.32	244.90	283.92	317.83	345.23
NON BILLABLE INSPECTIONS	25.39	49.58	78.05	101.45	120.52	136.15	148.91	160.19
CAR TOTAL	549.69	1236.64	2129.10	2886.48	3494.30	3995.28	4430.77	4794.11

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	516.06	940.15	1454.57	1867.05	2200.85	2479.70	2708.75	2902.99
BRAKE BEAMS	47.32	88.53	183.61	258.29	321.67	376.74	420.43	459.20
BRAKE HEAD WEAR PLATES	0.00	0.12	0.19	0.25	0.28	0.31	0.34	0.36
BRAKE BEAM WEAR PLATES	0.00	0.02	0.03	0.04	0.05	0.06	0.06	0.06
BRAKE HANGER OR CONNECTION PIN	2.28	4.01	5.99	7.64	8.92	9.98	10.92	11.67
BOTTOM ROD SAFETY SUPPORT	0.82	2.20	3.91	5.50	6.79	7.85	8.81	9.54
BRAKE BEAM SAFETY SUPPORT	0.01	0.04	0.06	0.08	0.09	0.10	0.10	0.11
BRAKE CONNECTION, BOTTOM	1.54	3.02	4.26	5.23	5.96	6.55	7.06	7.42
BRAKE CONNECTION, TOP	1.25	4.84	8.10	10.62	12.61	14.15	15.39	16.46
BRAKE LEVER	1.09	2.09	2.97	3.66	4.18	4.59	4.97	5.24
BRAKE LEVER GUIDE OR CARRIER	0.00	0.02	0.04	0.06	0.07	0.08	0.09	0.10
DEAD LEVER GUIDE	0.03	0.05	0.14	0.22	0.27	0.32	0.37	0.41
DEAD LEVER GUIDE BRACKET	0.04	0.12	0.20	0.26	0.32	0.36	0.39	0.42
BRAKE SHOES	460.35	832.36	1240.72	1569.57	1832.98	2051.13	2231.60	2383.21
BRAKE SHOE KEYS	1.33	2.73	4.33	5.62	6.64	7.49	8.19	8.78
WHEELSETS	717.26	1374.49	2011.03	2517.78	2910.06	3231.09	3498.02	3723.24
LUBRICATE ROLLER BEARINGS	7.94	21.85	32.49	40.64	46.44	50.99	54.85	58.37
ROLLER BEARINGS	121.26	229.46	334.19	417.68	482.34	535.36	579.40	616.51
ROLLER BEARING CAP SCREWS	0.09	0.22	0.30	0.35	0.40	0.43	0.46	0.47
ROLLER BEARING LOCKING PLATES	0.00	0.01	0.01	0.01	0.01	0.01	0.01	0.01
ROLLER BEARING LUBRICATION FITTING	0.00	0.01	0.01	0.01	0.02	0.02	0.02	0.02
PEDESTAL ADAPTERS	9.90	30.08	53.30	71.79	86.53	98.23	108.26	116.56
WHEELS	267.19	505.21	734.62	917.02	1058.16	1173.81	1269.78	1350.68
WHEEL LABOR	301.66	570.28	830.80	1038.62	1199.58	1331.65	1441.31	1533.88
AXLES, ROLLER BEARINGS	9.19	17.38	25.32	31.66	36.57	40.59	43.93	46.76
OTHER TRUCK REPAIRS	32.66	105.14	198.85	268.47	324.20	369.55	407.81	437.67
TRUCK BOLSTERS	8.39	23.51	48.22	65.97	81.05	93.42	102.88	111.08
TRUCK BOLSTERS (REPAIRED)	0.00	0.49	1.33	2.08	2.83	3.47	3.89	4.32
CENTER PINS	1.03	2.42	4.08	5.42	6.47	7.37	8.09	8.75
CENTER PLATES	0.00	0.07	0.21	0.32	0.42	0.50	0.57	0.63
CENTER PLATE LINERS	3.06	11.13	19.87	26.83	32.51	37.12	40.54	43.88
TRUCK SIDE BEARINGS	1.43	4.14	7.16	9.43	11.36	12.82	13.94	15.03
FRICTION CASTINGS	3.79	8.00	13.01	16.77	19.74	22.14	24.12	25.82
SIDE BEARING SHIM	0.01	0.13	0.34	0.51	0.64	0.76	0.86	0.94
SIDE FRAMES	6.43	34.28	72.42	100.67	122.45	140.08	156.92	168.04
SIDE FRAMES (REPAIRED)	0.00	0.11	0.26	0.39	0.52	0.63	0.70	0.77
SPRING PLANKS	0.00	0.00	0.01	0.01	0.02	0.03	0.03	0.04
SPRING PLANKS (REPAIRED)	0.00	0.00	0.01	0.01	0.02	0.02	0.03	0.03
OUTER SPRINGS	3.46	7.54	11.13	13.57	15.45	17.04	18.22	19.12
INNER SPRINGS	2.17	4.71	6.84	8.30	9.40	10.30	11.01	11.52
STABILIZER SPRINGS	0.93	2.14	3.27	4.10	4.68	5.17	5.55	5.90
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
TRUCK SPRING SHIM, WOOD	0.00	0.00	0.00	0.00	0.01	0.01	0.01	0.01
STEEL	0.00	0.01	0.06	0.11	0.14	0.17	0.20	0.22
MANUFACTURED MATERIAL (TRUCK)	1.93	6.44	10.60	13.94	16.45	18.45	20.21	21.55
TRUCK TOTAL	1265.97	2419.78	3664.45	4653.30	5435.10	6080.34	6614.57	7063.91

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ALL ROLLER BEARING CARS WITH ANNUAL MILEAGE BETWEEN 50000 AND 75000 MILES

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	125.63	237.57	351.40	441.90	513.94	571.55	621.73	661.57
COUPLERS, YOKES, & DRAFT GEAR	125.48	288.81	436.43	558.44	646.25	718.65	784.15	839.05
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	191.76	510.43	816.15	1057.91	1248.93	1403.75	1519.57	1634.44
OTHER CAR REPAIRS	84.29	170.57	296.89	398.89	483.28	552.95	612.07	660.70
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	338.89	708.80	1212.85	1608.48	1940.11	2213.22	2437.83	2627.72
WHEELSETS	606.12	1252.98	1877.55	2374.10	2770.63	3087.95	3361.37	3571.63
OTHER TRUCK REPAIRS	17.65	74.56	141.73	190.74	228.80	261.12	286.66	308.96
TOTAL	1489.80	3243.71	5133.00	6630.45	7831.95	8809.19	9623.37	10304.08
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	125.63	237.57	351.40	441.90	513.94	571.55	621.73	661.57
COT&S	9.41	21.44	30.98	38.77	44.29	48.73	53.02	55.98
IDT&S	58.62	119.61	180.24	227.79	265.69	296.13	321.85	342.78
PRESSURE SYSTEM	51.83	87.75	128.89	162.03	189.19	210.81	229.91	244.99
HAND BRAKES	5.77	8.76	11.28	13.31	14.77	15.88	16.95	17.82
COUPLERS, YOKES, & DRAFT GEAR	125.48	288.81	436.43	558.44	646.25	718.65	784.15	839.05
COUPLER BODIES	78.80	180.56	258.35	322.28	363.73	400.81	433.47	461.90
COUPLER KNUCKLES	22.38	46.75	69.30	87.44	101.20	111.49	121.36	129.21
OTHER COUPLER PARTS	15.63	29.21	43.07	54.35	63.49	70.94	76.96	81.92
YOKES	2.31	6.38	11.87	16.36	20.06	22.85	25.35	27.47
DRAFT GEARS, CARRIERS, AND FOLLOWERS	6.35	25.91	53.84	78.00	97.77	112.56	127.01	138.55
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	191.76	510.43	816.15	1057.91	1248.93	1403.75	1519.57	1634.44
OTHER CAR REPAIRS	84.29	170.57	296.89	398.89	483.28	552.95	612.07	660.70
OTHER CAR REPAIRS	55.07	112.91	200.33	270.81	329.48	377.96	419.13	452.70
WELDING	18.56	32.69	53.94	71.15	85.27	97.05	107.05	115.49
NON BILLABLE INSPECTIONS	10.66	24.97	42.63	56.93	68.53	77.94	85.89	92.51
CAR TOTAL	527.15	1207.37	1900.87	2457.13	2892.41	3246.90	3537.52	3795.77

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	338.89	708.80	1212.85	1608.48	1940.11	2213.22	2437.83	2627.72
BRAKE BEAMS	51.81	128.95	282.15	400.11	502.27	586.06	654.54	715.15
BRAKE HEAD WEAR PLATES	0.03	0.29	0.46	0.59	0.66	0.72	0.78	0.84
BRAKE BEAM WEAR PLATES	0.02	0.03	0.03	0.03	0.03	0.03	0.04	0.04
BRAKE HANGER OR CONNECTION PIN	0.38	0.77	1.55	2.19	2.69	3.11	3.49	3.80
BOTTOM ROD SAFETY SUPPORT	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
BRAKE BEAM SAFETY SUPPORT	0.01	0.03	0.04	0.05	0.06	0.07	0.08	0.08
BRAKE CONNECTION, BOTTOM	0.19	0.43	0.61	0.75	0.85	0.91	0.99	1.02
BRAKE CONNECTION, TOP	0.35	2.30	3.90	5.13	6.04	6.76	7.41	7.85
BRAKE LEVER	0.05	0.17	0.27	0.36	0.43	0.47	0.52	0.55
BRAKE LEVER GUIDE OR CARRIER	0.00	0.02	0.03	0.04	0.05	0.06	0.06	0.07
DEAD LEVER GUIDE	0.00	0.01	0.01	0.02	0.02	0.02	0.02	0.03
DEAD LEVER GUIDE BRACKET	0.00	0.00	0.01	0.02	0.02	0.02	0.03	0.03
BRAKE SHOES	285.20	573.85	920.39	1194.65	1421.50	1608.69	1762.92	1890.78
BRAKE SHOE KEYS	0.84	1.93	3.37	4.51	5.47	6.26	6.92	7.46
WHEELSETS	606.12	1252.98	1877.55	2374.10	2770.63	3087.95	3361.37	3571.63
LUBRICATE ROLLER BEARINGS	1.12	7.58	12.78	16.87	20.00	22.41	24.59	26.10
ROLLER BEARINGS	103.83	214.12	320.34	404.95	472.43	526.55	573.16	608.98
ROLLER BEARING CAP SCREWS	0.01	0.02	0.04	0.05	0.06	0.06	0.07	0.07
ROLLER BEARING LOCKING PLATES	0.03	0.08	0.11	0.13	0.15	0.16	0.17	0.17
ROLLER BEARING LUBRICATION FITTING	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
PEDESTAL ADAPTERS	6.74	14.26	24.99	32.88	40.04	45.49	49.73	53.62
WHEELS	228.34	470.14	702.15	886.65	1033.59	1151.34	1252.87	1330.75
WHEEL LABOR	258.19	530.57	792.90	1001.91	1168.60	1302.07	1417.37	1505.81
AXLES, ROLLER BEARINGS	7.87	16.20	24.24	30.65	35.76	39.86	43.39	46.11
OTHER TRUCK REPAIRS	17.65	74.56	141.73	190.74	228.80	261.12	286.66	308.96
TRUCK BOLSTERS	1.10	3.69	16.45	25.38	33.55	40.81	45.54	50.72
TRUCK BOLSTERS (REPAIRED)	0.03	0.09	0.19	0.29	0.37	0.44	0.50	0.55
CENTER PINS	0.27	0.71	1.20	1.61	1.90	2.15	2.37	2.53
CENTER PLATES	0.01	0.08	0.15	0.20	0.26	0.29	0.33	0.34
CENTER PLATE LINERS	0.18	1.47	3.06	4.36	5.45	6.38	7.07	7.68
TRUCK SIDE BEARINGS	0.63	1.30	2.71	3.76	4.64	5.40	6.02	6.53
FRICTION CASTINGS	0.17	0.63	1.85	2.79	3.64	4.40	4.92	5.43
SIDE BEARING SHIM	0.00	0.01	0.02	0.03	0.03	0.04	0.04	0.04
SIDE FRAMES	5.63	47.52	89.77	120.56	143.31	162.94	178.94	192.11
SIDE FRAMES (REPAIRED)	0.04	0.10	0.21	0.32	0.41	0.50	0.56	0.62
SPRING PLANKS	0.00	0.01	0.01	0.02	0.03	0.03	0.04	0.04
SPRING PLANKS (REPAIRED)	0.00	0.01	0.01	0.02	0.02	0.03	0.03	0.03
OUTER SPRINGS	2.85	3.85	5.03	5.88	6.58	7.12	7.56	7.97
INNER SPRINGS	1.14	1.54	2.01	2.35	2.63	2.85	3.03	3.19
STABILIZER SPRINGS	0.01	0.04	0.22	0.34	0.44	0.54	0.60	0.67
TRUCK SPRING SHIM, WOOD	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01
STEEL	4.45	10.57	14.33	17.09	18.65	19.76	21.01	21.88
MANUFACTURED MATERIAL (TRUCK)	1.13	2.94	4.50	5.73	6.67	7.43	8.08	8.58
TRUCK TOTAL	962.66	2036.34	3232.13	4173.32	4939.54	5562.29	6085.86	6508.31

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR ROLLER BEARING, 70 TON CARS

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	623.62	1114.95	1547.35	1892.03	2154.22	2368.41	2543.01	2689.37
COUPLERS, YOKES, & DRAFT GEAR	399.55	974.35	1472.93	1860.37	2150.33	2395.50	2586.66	2755.55
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	1054.19	2164.13	3084.19	3796.72	4345.58	4787.64	5142.04	5419.03
OTHER CAR REPAIRS	558.70	1058.32	1604.69	2068.24	2431.26	2733.23	2984.15	3197.70
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	1152.68	2018.67	2838.69	3487.53	3987.98	4407.51	4742.43	5031.48
WHEELSETS	1177.97	2342.67	3365.77	4166.77	4782.37	5282.47	5683.18	6022.21
OTHER TRUCK REPAIRS	699.57	1963.06	2949.78	3680.78	4214.46	4661.24	5004.34	5284.98
TOTAL	5666.48	11636.15	16863.41	20952.44	24066.19	26636.00	28685.80	30400.33
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	623.62	1114.95	1547.35	1892.03	2154.22	2368.41	2543.01	2689.37
COT&S	191.78	362.04	494.74	595.11	669.47	729.05	777.05	816.32
IOT&S	262.83	426.76	569.51	684.13	770.55	841.25	898.41	947.56
PRESSURE SYSTEM	151.05	286.57	424.19	538.36	627.97	702.03	763.63	814.86
HAND BRAKES	17.97	39.57	58.92	74.43	86.23	96.08	103.92	110.68
COUPLERS, YOKES, & DRAFT GEAR	399.55	974.35	1472.93	1860.37	2150.33	2395.50	2586.66	2755.55
COUPLER BODIES	230.02	553.16	807.45	997.16	1133.78	1254.09	1343.41	1422.83
COUPLER KNUCKLES	62.73	130.35	191.91	241.50	279.36	310.07	335.29	356.82
OTHER COUPLER PARTS	72.16	156.07	232.31	293.20	340.35	378.44	409.22	435.67
YOKES	15.12	54.26	90.44	119.19	140.97	158.81	173.24	186.48
DRAFT GEARS, CARRIERS, AND FOLLOWERS	19.51	80.51	150.82	209.31	255.86	294.09	325.51	353.76
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	1054.19	2164.13	3084.19	3796.72	4345.58	4787.64	5142.04	5419.03
OTHER CAR REPAIRS	558.70	1058.32	1604.69	2068.24	2431.26	2733.23	2984.15	3197.70
OTHER CAR REPAIRS	370.89	691.27	1058.78	1373.35	1621.22	1828.82	2001.93	2149.79
WELDING	111.93	238.78	368.91	478.15	563.06	632.67	690.20	738.74
NON BILLABLE INSPECTIONS	75.88	128.27	177.00	216.74	246.98	271.74	292.01	309.17
CAR TOTAL	2636.05	5311.75	7709.16	9617.36	11081.38	12284.79	13255.85	14061.65

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	1152.68	2018.67	2838.69	3487.53	3987.98	4407.51	4742.43	5031.48
BRAKE BEAMS	184.23	450.54	713.18	915.45	1071.41	1207.18	1311.10	1403.01
BRAKE HEAD WEAR PLATES	1.34	2.72	3.54	4.07	4.36	4.70	4.90	5.05
BRAKE BEAM WEAR PLATES	0.11	0.36	0.54	0.66	0.74	0.82	0.84	0.91
BRAKE BEAM HANGERS	0.03	0.06	0.08	0.09	0.09	0.10	0.10	0.11
BRAKE HANGER BRACKET WEAR PLATE	0.01	0.02	0.03	0.03	0.04	0.04	0.04	0.04
BRAKE HANGER BRACKET WEAR PLATE SECUREMENT	0.01	0.02	0.03	0.04	0.04	0.04	0.04	0.05
BRAKE HANGER OR CONNECTION PIN	3.30	6.78	10.30	13.15	15.39	17.28	18.81	20.12
BOTTOM ROD SAFETY SUPPORT	0.74	2.61	4.99	7.16	8.93	10.46	11.70	12.79
BRAKE BEAM SAFETY SUPPORT	0.56	1.30	1.75	2.02	2.24	2.39	2.46	2.60
BRAKE CONNECTION, BOTTOM	1.50	4.31	6.69	8.61	10.01	11.15	12.15	12.98
BRAKE CONNECTION, TOP	11.42	26.89	37.86	46.04	51.38	56.26	59.95	63.27
BRAKE LEVER	0.95	2.30	3.53	4.51	5.25	5.87	6.39	6.84
BRAKE LEVER GUIDE OR CARRIER	0.03	0.20	0.32	0.41	0.47	0.51	0.56	0.60
DEAD LEVER GUIDE	0.75	1.45	1.97	2.41	2.67	2.89	3.13	3.31
DEAD LEVER GUIDE BRACKET	0.06	0.36	0.62	0.82	0.98	1.10	1.21	1.31
BRAKE SHOES	943.83	1510.83	2041.94	2468.06	2797.94	3069.02	3290.04	3478.33
BRAKE SHOE KEYS	4.04	7.92	11.34	13.99	16.04	17.70	19.03	20.16
WHEELSETS	1177.97	2342.67	3365.77	4166.77	4782.37	5282.47	5683.18	6022.21
LUBRICATE ROLLER BEARINGS	48.51	77.03	96.70	111.36	121.55	129.58	135.63	140.73
ROLLER BEARINGS	184.51	368.06	529.80	656.29	753.68	833.09	896.66	950.41
ROLLER BEARING CAP SCREWS	0.06	0.23	0.37	0.47	0.55	0.61	0.67	0.70
ROLLER BEARING LOCKING PLATES	0.02	0.07	0.11	0.13	0.15	0.16	0.17	0.18
ROLLER BEARING LUBRICATION FITTING	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.04
PEDESTAL ADAPTERS	63.71	128.93	186.80	232.24	267.35	295.60	318.18	336.80
WHEELS	410.41	834.69	1208.83	1502.54	1727.93	1910.86	2057.83	2183.10
WHEEL LABOR	456.87	906.11	1303.50	1614.61	1854.70	2050.16	2206.87	2339.10
AXLES, ROLLER BEARINGS	13.87	27.53	39.63	49.10	56.42	62.37	67.14	71.16
OTHER TRUCK REPAIRS	699.57	1963.06	2949.78	3680.78	4214.46	4661.24	5004.34	5284.98
TRUCK BOLSTERS	100.60	354.17	580.23	753.19	889.37	999.02	1085.50	1154.36
TRUCK BOLSTERS (REPAIRED)	16.40	21.51	25.63	29.34	31.68	33.68	35.26	36.54
CENTER PINS	3.89	9.41	14.81	19.10	22.48	25.23	27.46	29.36
CENTER PLATES	2.41	5.71	9.91	13.33	16.09	18.34	20.32	22.15
CENTER PLATE LINERS	24.45	45.54	65.36	81.99	94.25	104.27	112.51	119.67
TRUCK SIDE BEARINGS	11.54	28.74	43.00	53.97	62.38	69.08	74.43	78.87
FRICTION CASTINGS	24.74	72.47	114.33	146.76	172.07	192.28	208.36	221.86
SIDE BEARING SHIM	0.71	2.61	4.11	5.26	6.06	6.71	7.31	7.75
SIDE FRAMES	472.99	1302.30	1906.93	2343.26	2647.54	2910.60	3107.62	3269.97
SIDE FRAMES (REPAIRED)	1.49	7.74	12.46	15.93	18.30	20.50	21.93	23.10
SPRING PLANKS	0.90	1.91	2.54	3.00	3.26	3.45	3.64	3.79
SPRING PLANKS (REPAIRED)	0.19	0.86	1.33	1.69	1.92	2.09	2.26	2.39
OUTER SPRINGS	15.17	46.92	72.63	91.84	106.85	118.23	127.35	134.59
INNER SPRINGS	7.90	23.03	35.49	44.77	52.21	57.73	62.17	65.69
STABILIZER SPRINGS	6.53	18.70	28.97	36.70	42.77	47.56	51.37	54.38
TRUCK SPRING FRICTION SNUBBER	0.12	0.21	0.27	0.30	0.34	0.36	0.38	0.39
TRUCK SPRING PLATES	0.69	0.88	0.97	1.05	1.07	1.09	1.11	1.13
TRUCK SPRING SHIM, WOOD	0.09	0.32	0.48	0.60	0.68	0.74	0.79	0.83
STEEL	0.66	2.40	3.62	4.52	5.11	5.59	6.05	6.37
MANUFACTURED MATERIAL (TRUCK)	8.10	17.61	26.70	34.18	40.02	44.70	48.52	51.80
TRUCK TOTAL	3030.42	6324.40	9154.25	11335.08	12984.80	14351.22	15429.95	16338.68

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR ROLLER BEARING, 100 TON CARS

SUMMARY TABLE	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	226.01	449.97	659.51	826.58	954.63	1060.78	1147.66	1222.24
COUPLERS, YOKES, & DRAFT GEAR	145.57	411.06	646.02	834.57	971.09	1081.07	1179.35	1259.84
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	179.31	471.00	731.86	935.65	1094.97	1219.64	1322.59	1410.57
OTHER CAR REPAIRS	155.43	363.16	584.62	765.04	906.84	1023.45	1120.40	1204.03
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	499.74	1105.35	1716.03	2200.88	2581.09	2891.19	3146.21	3366.54
WHEELSETS	748.94	1648.46	2515.30	3214.99	3768.91	4208.53	4576.50	4888.07
OTHER TRUCK REPAIRS	120.44	389.42	599.91	759.56	869.59	957.76	1031.56	1096.47
TOTAL	2075.45	4838.42	7453.24	9537.27	11147.11	12442.43	13524.27	14447.76
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	226.01	449.97	659.51	826.58	954.63	1060.78	1147.66	1222.24
COT&S	29.50	80.58	129.24	168.44	197.51	222.89	242.68	260.51
IDT&S	107.47	203.63	290.34	358.76	411.47	454.08	489.32	519.26
PRESSURE SYSTEM	75.25	146.10	215.06	270.25	313.33	348.78	378.35	403.21
HAND BRAKES	13.79	19.66	24.86	29.13	32.32	35.04	37.30	39.27
COUPLERS, YOKES, & DRAFT GEAR	145.57	411.06	646.02	834.57	971.09	1081.07	1179.35	1259.84
COUPLER BODIES	93.78	241.98	358.78	450.44	512.81	563.80	610.32	646.81
COUPLER KNUCKLES	16.00	48.10	78.27	102.55	120.87	135.71	148.28	159.00
OTHER COUPLER PARTS	22.69	59.39	93.40	120.48	141.35	158.03	171.94	183.86
YOKES	4.61	21.20	37.05	50.06	59.51	67.10	73.83	79.92
DRAFT GEARS, CARRIERS, AND FOLLOWERS	8.48	40.38	78.52	111.05	136.55	156.43	174.97	190.25
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	179.31	471.00	731.86	935.65	1094.97	1219.64	1322.59	1410.57
OTHER CAR REPAIRS	155.43	363.16	584.62	765.04	906.84	1023.45	1120.40	1204.03
OTHER CAR REPAIRS	101.92	221.22	355.87	466.31	553.78	626.83	687.28	739.08
WELDING	29.05	83.53	136.67	179.55	212.60	239.13	261.54	280.91
NON BILLABLE INSPECTIONS	24.46	58.41	92.08	119.18	140.45	157.50	171.58	184.04
CAR TOTAL	706.33	1695.19	2622.00	3361.83	3927.52	4384.94	4769.99	5096.68

TRUCK REPAIRS	ANNUAL MILEAGE 12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	499.74	1105.35	1716.03	2200.88	2581.09	2891.19	3146.21	3366.54
BRAKE BEAMS	70.63	188.93	327.62	439.65	527.56	600.58	660.33	712.89
BRAKE HEAD WEAR PLATES	0.37	0.81	1.07	1.24	1.35	1.44	1.49	1.55
BRAKE BEAM WEAR PLATES	0.08	0.17	0.23	0.27	0.30	0.32	0.34	0.36
BRAKE HANGER OR CONNECTION PIN	1.69	3.76	5.57	7.01	8.13	9.02	9.79	10.42
BOTTOM ROD SAFETY SUPPORT	0.79	1.53	2.15	2.66	3.07	3.38	3.67	3.88
BRAKE BEAM SAFETY SUPPORT	0.02	0.06	0.08	0.10	0.11	0.12	0.13	0.13
BRAKE CONNECTION, BOTTOM	1.97	3.52	4.65	5.51	6.11	6.59	7.02	7.35
BRAKE CONNECTION, TOP	4.52	11.19	15.90	19.34	21.73	23.63	25.24	26.41
BRAKE LEVER	1.41	2.50	3.40	4.08	4.60	4.99	5.34	5.62
BRAKE LEVER GUIDE OR CARRIER	1.81	2.01	2.09	2.16	2.21	2.25	2.28	2.30
DEAD LEVER GUIDE	0.12	0.61	1.06	1.41	1.65	1.85	2.02	2.19
DEAD LEVER GUIDE BRACKET	0.12	0.51	0.84	1.09	1.26	1.41	1.52	1.62
BRAKE SHOES	413.47	884.82	1344.14	1707.32	1992.52	2223.98	2414.46	2578.41
BRAKE SHOE KEYS	2.54	4.92	7.22	9.04	10.48	11.62	12.58	13.39
WHEELSETS	748.94	1648.46	2515.30	3214.99	3768.91	4208.53	4576.50	4888.07
LUBRICATE ROLLER BEARINGS	12.06	29.03	41.79	51.34	58.46	63.84	68.42	72.40
ROLLER BEARINGS	126.02	274.58	418.91	535.67	628.34	702.03	763.62	815.76
ROLLER BEARING CAP SCREWS	0.10	0.25	0.37	0.45	0.52	0.56	0.60	0.63
ROLLER BEARING LOCKING PLATES	0.03	0.09	0.13	0.16	0.18	0.19	0.21	0.21
ROLLER BEARING LUBRICATION FITTING	0.01	0.02	0.02	0.02	0.02	0.02	0.03	0.03
PEDESTAL ADAPTERS	9.12	30.65	51.14	67.32	79.78	89.43	97.68	104.66
WHEELS	280.18	613.71	933.84	1192.22	1396.52	1558.72	1694.51	1809.35
WHEEL LABOR	311.93	679.42	1037.46	1327.32	1557.58	1740.63	1893.67	2023.30
AXLES, ROLLER BEARINGS	9.50	20.71	31.64	40.48	47.51	53.10	57.77	61.73
OTHER TRUCK REPAIRS	120.44	389.42	599.91	759.56	869.59	957.76	1031.56	1096.47
TRUCK BOLSTERS	7.33	30.79	53.71	71.81	85.66	95.76	104.40	112.49
TRUCK BOLSTERS (REPAIRED)	0.24	3.34	6.86	9.99	12.55	14.58	16.11	17.84
CENTER PINS	1.15	3.07	5.13	6.80	8.11	9.19	10.07	10.86
CENTER PLATES	0.36	1.76	3.95	5.76	7.24	8.43	9.45	10.48
CENTER PLATE LINERS	3.00	14.37	26.27	36.13	43.80	50.03	55.08	59.88
TRUCK SIDE BEARINGS	2.63	6.24	9.18	11.46	13.17	14.49	15.56	16.55
FRICTION CASTINGS	5.00	15.90	26.26	34.64	40.96	45.73	49.79	53.69
SIDE BEARING SHIM	0.16	1.89	3.43	4.76	5.68	6.31	6.93	7.56
SIDE FRAMES	91.73	275.86	405.29	499.76	560.28	610.93	653.05	687.53
SIDE FRAMES (REPAIRED)	0.24	3.37	6.00	8.22	9.77	10.87	11.76	12.94
SPRING PLANKS	0.03	0.19	0.31	0.39	0.44	0.49	0.52	0.56
SPRING PLANKS (REPAIRED)	0.03	0.16	0.26	0.33	0.37	0.41	0.44	0.47
OUTER SPRINGS	3.70	15.24	24.76	32.15	37.52	41.49	45.01	48.30
INNER SPRINGS	1.81	6.83	11.02	14.25	16.65	18.42	19.97	21.38
STABILIZER SPRINGS	0.89	3.25	5.48	7.27	8.58	9.55	10.33	11.17
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.03	0.03	0.03	0.03	0.03	0.04
TRUCK SPRING PLATES	0.00	0.00	0.01	0.01	0.01	0.01	0.01	0.01
TRUCK SPRING SHIM, WOOD	0.02	0.10	0.16	0.21	0.23	0.25	0.27	0.29
STEEL	0.11	0.46	0.72	0.90	1.04	1.14	1.22	1.30
MANUFACTURED MATERIAL (TRUCK)	1.80	6.57	11.07	14.70	17.51	19.67	21.55	23.14
TRUCK TOTAL	1369.12	3143.23	4831.23	6175.43	7219.59	8057.48	8754.28	9351.08

**PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ROLLER BEARING, 70 TON CARS WITH AAR CAR TYPES A AND B**

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	592.72	1069.95	1521.94	1883.97	2169.43	2399.14	2588.49	2746.40
COUPLERS, YOKES, & DRAFT GEAR	404.43	936.09	1410.30	1778.36	2058.83	2289.38	2470.85	2634.70
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	692.52	1515.37	2166.28	2652.17	3021.23	3310.71	3544.36	3736.82
OTHER CAR REPAIRS	440.20	953.81	1562.44	2075.52	2489.29	2828.77	3115.22	3356.86
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	1037.93	1907.88	2773.31	3457.20	3984.62	4427.05	4777.52	5083.11
WHEELSETS	1450.67	2628.25	3648.13	4440.35	5043.32	5531.24	5924.91	6256.30
OTHER TRUCK REPAIRS	670.66	1825.52	2764.91	3447.97	3975.80	4404.75	4726.40	5002.99
TOTAL	5289.12	10836.86	15847.30	19735.54	22742.53	25191.04	27147.76	28817.18
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	592.72	1069.95	1521.94	1883.97	2169.43	2399.14	2588.49	2746.40
COT&S	153.48	307.58	434.95	530.01	605.88	664.21	712.63	751.03
IDT&S	233.66	399.12	553.99	679.08	775.64	854.72	918.93	973.97
PRESSURE SYSTEM	182.84	328.61	487.18	619.78	725.75	812.16	884.13	944.50
HAND BRAKES	22.73	34.63	45.82	55.11	62.15	68.04	72.80	76.89
COUPLERS, YOKES, & DRAFT GEAR	404.43	936.09	1410.30	1778.36	2058.83	2289.38	2470.85	2634.70
COUPLER BODIES	213.60	501.74	735.10	908.37	1038.14	1147.02	1227.54	1303.49
COUPLER KNUCKLES	84.86	155.06	220.63	273.91	314.02	346.49	373.08	396.03
OTHER COUPLER PARTS	72.85	159.07	243.70	312.93	366.96	410.26	446.21	476.77
YOKES	14.04	49.64	82.66	108.13	127.83	143.73	156.49	168.23
DRAFT GEARS, CARRIERS, AND FOLLOWERS	19.09	70.58	128.20	175.01	211.89	241.89	267.53	290.19
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	692.52	1515.37	2166.28	2652.17	3021.23	3310.71	3544.36	3736.82
OTHER CAR REPAIRS	440.20	953.81	1562.44	2075.52	2489.29	2828.77	3115.22	3356.86
OTHER CAR REPAIRS	290.55	598.71	975.50	1292.39	1548.44	1760.47	1938.14	2089.39
WELDING	109.87	267.95	454.27	614.45	743.74	848.00	937.74	1012.75
NON BILLABLE INSPECTIONS	39.78	87.15	132.67	168.68	197.11	220.30	239.34	254.72
CAR TOTAL	2129.87	4475.21	6660.96	8390.02	9738.78	10828.01	11718.92	12474.78

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	1037.93	1907.88	2773.31	3457.20	3984.62	4427.05	4777.52	5083.11
BRAKE BEAMS	209.74	487.65	791.73	1026.85	1212.94	1374.60	1496.18	1607.16
BRAKE HEAD WEAR PLATES	0.21	0.54	0.74	0.88	0.96	1.05	1.09	1.14
BRAKE BEAM WEAR PLATES	0.03	0.07	0.09	0.11	0.12	0.13	0.13	0.14
BRAKE BEAM HANGERS	0.03	0.07	0.10	0.11	0.12	0.13	0.13	0.14
BRAKE HANGER BRACKET WEAR PLATE	0.01	0.03	0.04	0.05	0.06	0.07	0.07	0.07
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	2.94	6.14	9.27	11.73	13.69	15.31	16.64	17.74
BOTTOM ROD SAFETY SUPPORT	1.18	2.52	3.92	5.12	6.11	6.93	7.60	8.20
BRAKE BEAM SAFETY SUPPORT	0.73	1.54	2.01	2.29	2.50	2.65	2.74	2.86
BRAKE CONNECTION, BOTTOM	2.24	5.07	7.38	9.18	10.54	11.60	12.51	13.35
BRAKE CONNECTION, TOP	7.86	21.35	31.28	38.54	43.89	48.28	51.57	54.72
BRAKE LEVER	1.35	2.52	3.51	4.27	4.87	5.33	5.73	6.07
BRAKE LEVER GUIDE OR CARRIER	0.79	0.89	0.94	0.97	0.99	1.01	1.02	1.03
DEAD LEVER GUIDE	0.42	1.17	1.82	2.32	2.71	3.00	3.29	3.47
DEAD LEVER GUIDE BRACKET	0.05	0.37	0.65	0.86	1.04	1.16	1.27	1.39
BRAKE SHOES	803.87	1366.47	1903.17	2333.01	2659.84	2928.79	3148.40	3334.58
BRAKE SHOE KEYS	6.47	11.46	16.61	20.85	24.19	26.96	29.09	30.98
WHEELSETS	1450.67	2628.25	3648.13	4440.35	5043.32	5531.24	5924.91	6256.30
LUBRICATE ROLLER BEARINGS	39.10	65.31	83.79	97.36	107.44	114.79	120.51	125.70
ROLLER BEARINGS	232.17	418.19	579.32	704.45	799.65	876.95	939.25	991.59
ROLLER BEARING CAP SCREWS	0.06	0.24	0.40	0.52	0.61	0.69	0.76	0.79
ROLLER BEARING LOCKING PLATES	0.01	0.04	0.06	0.07	0.08	0.09	0.09	0.09
ROLLER BEARING LUBRICATION FITTING	0.02	0.03	0.04	0.04	0.05	0.05	0.05	0.05
PEDESTAL ADAPTERS	71.27	138.18	199.35	246.80	284.02	313.41	337.30	357.14
WHEELS	514.74	941.88	1311.70	1599.33	1817.56	1994.81	2137.46	2258.34
WHEEL LABOR	575.82	1032.98	1430.01	1738.91	1973.90	2164.61	2318.96	2448.13
AXLES, ROLLER BEARINGS	17.48	31.38	43.47	52.87	60.03	65.84	70.53	74.46
OTHER TRUCK REPAIRS	670.66	1825.52	2764.91	3447.97	3975.80	4404.75	4726.40	5002.99
TRUCK BOLSTERS	135.08	375.51	598.20	763.51	896.90	1005.50	1088.56	1157.01
TRUCK BOLSTERS (REPAIRED)	0.82	3.84	6.93	9.58	11.71	13.40	14.80	15.90
CENTER PINS	3.57	8.60	13.68	17.62	20.80	23.40	25.54	27.30
CENTER PLATES	3.25	6.88	11.27	14.74	17.53	19.67	21.80	23.67
CENTER PLATE LINERS	9.67	27.99	45.78	60.22	71.36	80.34	87.90	94.23
TRUCK SIDE BEARINGS	16.94	36.99	51.88	63.14	71.79	78.68	84.07	88.65
FRICTION CASTINGS	47.75	98.37	141.65	174.97	199.88	220.04	236.19	250.18
SIDE BEARING SHIM	0.59	2.47	4.07	5.34	6.27	6.94	7.55	8.16
SIDE FRAMES	381.47	1108.22	1662.36	2054.81	2353.19	2597.93	2775.51	2931.42
SIDE FRAMES (REPAIRED)	1.78	13.42	23.05	30.50	35.89	40.59	43.49	46.24
SPRING PLANKS	0.84	1.38	1.63	1.85	1.92	1.95	2.04	2.14
SPRING PLANKS (REPAIRED)	0.13	0.70	1.10	1.41	1.65	1.78	1.92	2.10
OUTER SPRINGS	24.30	55.74	81.23	100.18	115.07	126.09	134.97	142.52
INNER SPRINGS	12.41	28.41	41.73	51.49	59.48	65.28	69.93	73.80
STABILIZER SPRINGS	12.32	25.71	37.18	45.84	52.47	57.63	61.80	65.39
TRUCK SPRING FRICTION SNUBBER	0.17	0.30	0.38	0.43	0.47	0.50	0.53	0.54
TRUCK SPRING PLATES	0.03	0.10	0.14	0.18	0.20	0.21	0.23	0.24
TRUCK SPRING SHIM, WOOD	0.04	0.29	0.47	0.60	0.69	0.77	0.82	0.88
STEEL	0.25	1.96	3.18	4.09	4.73	5.24	5.71	6.04
MANUFACTURED MATERIAL (TRUCK)	17.26	28.66	39.00	47.48	53.78	58.79	63.03	66.56
TRUCK TOTAL	3159.25	6361.65	9186.34	11345.51	13003.75	14363.04	15428.84	16342.40

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS
FOR ROLLER BEARING, 100 TON CARS WITH AAR CAR TYPES A AND B

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	336.92	882.73	1424.72	1867.77	2203.59	2482.53	2723.37	2912.72
COUPLERS, YOKES, & DRAFT GEAR	418.54	1309.38	2109.00	2762.25	3206.78	3588.53	3939.77	4182.12
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	350.82	1001.54	1572.29	2022.52	2360.12	2636.05	2860.26	3048.17
OTHER CAR REPAIRS	179.73	652.93	1235.54	1731.14	2129.78	2458.27	2739.25	2975.87
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	620.45	1661.80	2729.58	3586.05	4218.54	4757.40	5218.10	5573.06
WHEELSETS	593.64	1475.34	2278.47	2912.52	3396.67	3788.46	4111.62	4378.89
OTHER TRUCK REPAIRS	362.72	1447.17	2292.62	2948.33	3363.55	3721.50	4047.67	4270.04
TOTAL	2862.82	8430.89	13642.23	17830.59	20879.02	23432.73	25640.04	27340.87
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	336.92	882.73	1424.72	1867.77	2203.59	2482.53	2723.37	2912.72
COT&S	54.86	193.10	321.24	426.31	497.20	559.94	618.57	655.19
IOT&S	98.72	231.82	354.33	451.61	526.15	586.93	636.26	678.89
PRESSURE SYSTEM	176.91	444.42	728.92	963.90	1149.86	1301.66	1431.43	1538.92
HAND BRAKES	6.43	13.39	20.23	25.95	30.38	33.99	37.10	39.72
COUPLERS, YOKES, & DRAFT GEAR	418.54	1309.38	2109.00	2762.25	3206.78	3588.53	3939.77	4182.12
COUPLER BODIES	336.54	1002.36	1576.28	2043.88	2350.94	2618.12	2871.18	3033.66
COUPLER KNUCKLES	32.19	111.57	189.21	253.21	299.84	339.26	373.82	399.51
OTHER COUPLER PARTS	30.42	93.71	155.24	205.34	244.28	275.50	301.62	324.02
YOKES	13.99	77.44	138.61	188.40	224.01	253.83	279.02	301.53
DRAFT GEARS, CARRIERS, AND FOLLOWERS	5.40	24.30	49.66	71.41	87.71	101.82	114.13	123.41
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	350.82	1001.54	1572.29	2022.52	2360.12	2636.05	2860.26	3048.17
OTHER CAR REPAIRS	179.73	652.93	1235.54	1731.14	2129.78	2458.27	2739.25	2975.87
OTHER CAR REPAIRS	110.98	369.41	716.52	1014.57	1256.05	1457.80	1630.41	1775.67
WELDING	46.05	217.02	407.82	569.01	697.37	801.33	890.46	965.46
NON BILLABLE INSPECTIONS	22.71	66.50	111.21	147.56	176.36	199.14	218.58	234.74
CAR TOTAL	1286.02	3846.58	6341.55	8383.69	9900.27	11165.37	12262.64	13118.88

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	620.45	1661.80	2729.58	3586.05	4218.54	4757.40	5218.10	5573.06
BRAKE BEAMS	222.77	681.38	1196.21	1616.95	1914.98	2180.07	2414.29	2581.22
BRAKE HEAD WEAR PLATES	0.66	1.49	2.01	2.36	2.59	2.77	2.91	3.02
BRAKE BEAM WEAR PLATES	0.10	0.27	0.38	0.46	0.51	0.52	0.57	0.57
BRAKE BEAM HANGERS	0.02	0.03	0.03	0.03	0.03	0.03	0.04	0.04
BRAKE HANGER OR CONNECTION PIN	1.55	5.54	9.68	13.06	15.69	17.84	19.67	21.13
BOTTOM ROD SAFETY SUPPORT	0.46	3.66	7.47	10.94	13.75	16.07	18.14	19.69
BRAKE BEAM SAFETY SUPPORT	0.08	0.22	0.31	0.38	0.41	0.45	0.46	0.49
BRAKE CONNECTION, BOTTOM	2.70	8.94	14.24	18.55	21.41	23.91	26.22	27.76
BRAKE CONNECTION, TOP	13.41	39.00	58.06	72.78	82.35	90.42	97.69	102.82
BRAKE LEVER	1.77	5.40	8.85	11.52	13.64	15.33	16.65	17.83
BRAKE LEVER GUIDE OR CARRIER	10.61	11.53	11.88	12.16	12.36	12.52	12.65	12.75
DEAD LEVER GUIDE	1.23	6.29	11.08	14.78	17.43	19.67	21.44	23.07
DEAD LEVER GUIDE BRACKET	1.83	7.52	12.56	16.35	18.98	21.24	23.05	24.29
BRAKE SHOES	360.20	884.28	1387.62	[illegible]	2091.19	2341.92	2548.56	2721.63
BRAKE SHOE KEYS	3.05	6.26	9.19	11.45	13.21	14.62	15.77	16.74
WHEELSETS	593.64	1475.34	2278.47	2912.52	3396.67	3788.46	4111.62	4378.89
LUBRICATE ROLLER BEARINGS	15.50	37.20	52.67	64.05	71.80	78.35	83.03	87.57
ROLLER BEARINGS	94.50	227.83	349.88	446.10	520.12	579.96	629.30	669.69
ROLLER BEARING CAP SCREWS	0.24	0.53	0.75	0.90	1.02	1.10	1.19	1.23
ROLLER BEARING LOCKING PLATES	0.30	1.06	1.62	2.06	2.26	2.49	2.73	2.73
ROLLER BEARING LUBRICATION FITTING	0.01	0.04	0.06	0.08	0.09	0.10	0.11	0.11
PEDESTAL ADAPTERS	17.05	52.18	85.35	111.45	131.48	147.65	161.34	171.65
WHEELS	227.00	508.38	919.67	1183.23	1382.52	1544.34	1678.86	1790.70
WHEEL LABOR	231.96	551.27	842.67	1071.82	1249.11	1391.82	1508.78	1605.96
AXLES, ROLLER BEARINGS	7.08	16.84	25.78	32.82	38.27	42.66	46.26	49.24
OTHER TRUCK REPAIRS	362.72	1447.17	2292.62	2948.33	3363.55	3721.50	4047.67	4270.04
TRUCK BOLSTERS	11.34	56.75	100.76	135.51	161.41	182.54	200.80	212.48
TRUCK BOLSTERS (REPAIRED)	0.13	0.79	1.52	2.17	2.66	3.08	3.45	3.68
CENTER PINS	0.77	3.06	5.55	7.53	9.12	10.42	11.49	12.36
CENTER PLATES	0.00	0.05	0.17	0.28	0.38	0.46	0.54	0.60
CENTER PLATE LINERS	2.50	11.30	20.51	28.06	33.98	38.81	42.83	46.03
TRUCK SIDE BEARINGS	2.50	8.75	14.20	18.51	21.58	24.12	26.36	27.82
FRICTION CASTINGS	12.25	28.43	43.03	54.62	63.24	70.21	76.17	80.49
SIDE BEARING SHIM	0.97	5.59	9.38	12.55	14.35	16.00	17.64	18.65
SIDE FRAMES	311.36	1201.55	1870.19	2385.78	2698.91	2974.66	3230.12	3399.56
SIDE FRAMES (REPAIRED)	1.09	10.17	17.31	22.92	26.84	29.91	32.19	34.92
SPRING PLANKS	0.50	2.45	3.88	4.89	5.57	6.13	6.51	6.90
SPRING PLANKS (REPAIRED)	0.54	2.41	3.78	4.77	5.38	5.92	6.35	6.63
OUTER SPRINGS	10.13	70.24	123.34	165.48	195.35	219.23	240.20	256.34
INNER SPRINGS	3.03	26.29	47.71	64.82	77.46	87.26	95.83	102.89
STABILIZER SPRINGS	1.50	4.99	8.07	10.45	12.16	13.57	14.70	15.53
TRUCK SPRING FRICTION SNUBBER	0.02	0.04	0.06	0.06	0.07	0.07	0.07	0.07
TRUCK SPRING PLATES	0.01	0.05	0.07	0.08	0.10	0.10	0.11	0.11
TRUCK SPRING SHIM, WOOD	0.26	1.25	1.93	2.45	2.74	3.01	3.24	3.40
STEEL	1.25	4.99	7.88	9.87	11.43	12.60	13.41	14.22
MANUFACTURED MATERIAL (TRUCK)	2.56	8.04	13.30	17.53	20.82	23.39	25.67	27.35
TRUCK TOTAL	1576.81	4584.32	7300.68	9446.90	10978.76	12267.36	13377.39	14221.99

PRESENT VALUE AT TIME OF PURCHASE OF ALL REPAIRS FOR AAR CAR TYPES H, L, G, F, AND V

SUMMARY TABLE	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	243.02	496.32	772.04	994.59	1170.46	1314.50	1437.39	1541.64
COUPLERS, YOKES, & DRAFT GEAR	97.53	324.63	582.64	789.32	951.19	1080.70	1196.97	1293.68
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	124.19	347.27	625.74	849.21	1029.20	1173.69	1301.73	1406.68
OTHER CAR REPAIRS	163.89	438.76	880.78	1252.45	1557.85	1812.19	2037.60	2225.07
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	482.14	1027.09	1603.72	2059.77	2421.26	2716.50	2960.28	3170.87
WHEELSETS	823.32	1688.26	2532.18	3198.62	3717.76	4137.57	4491.71	4791.02
OTHER TRUCK REPAIRS	43.11	198.19	364.42	492.83	593.82	670.95	741.11	796.94
TOTAL	1977.20	4520.51	7361.51	9636.79	11441.54	12906.09	14166.79	15225.91
ASSUMED CAR LIFE IN YEARS	30.00	30.00	30.00	24.00	19.20	16.00	13.71	12.00

CAR REPAIRS:	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
BRAKES (TEST, PRESSURE SYSTEM, & HAND BRAKES)	243.02	496.32	772.04	994.59	1170.46	1314.50	1437.39	1541.64
COT&S	36.15	93.72	154.18	201.78	238.32	268.61	293.27	316.58
IDT&S	115.24	220.81	324.25	406.81	471.58	523.95	567.98	605.11
PRESSURE SYSTEM	68.47	141.39	236.94	316.37	380.94	434.25	481.39	519.42
HAND BRAKES	23.16	40.39	56.67	69.62	79.62	87.69	94.75	100.53
COUPLERS, YOKES, & DRAFT GEAR	97.53	324.63	582.64	789.32	951.19	1080.70	1196.97	1293.68
COUPLER BODIES	44.11	138.31	250.17	337.66	406.01	460.47	511.83	552.37
COUPLER KNUCKLES	15.20	49.75	85.58	114.46	137.16	155.27	170.94	184.03
OTHER COUPLER PARTS	24.10	68.55	117.61	157.43	189.10	214.51	236.39	254.81
YOKES	3.59	17.42	31.96	43.64	52.65	59.77	65.84	71.48
DRAFT GEARS, CARRIERS, AND FOLLOWERS	10.53	50.60	97.31	136.13	166.28	190.68	211.98	231.04
MISCELLANEOUS LABOR & MANUFACTURED MATERIAL	124.19	347.27	625.74	849.21	1029.20	1173.69	1301.73	1406.68
OTHER CAR REPAIRS	163.89	438.76	880.78	1252.45	1557.85	1812.19	2037.60	2225.07
OTHER CAR REPAIRS	111.64	284.76	599.45	865.94	1086.60	1272.29	1437.41	1574.71
WELDING	25.67	90.19	177.87	250.94	310.40	358.70	401.86	436.74
NON BILLABLE INSPECTIONS	26.57	63.81	103.46	135.57	160.86	181.20	198.33	213.63
CAR TOTAL	628.63	1606.97	2861.20	3885.57	4708.70	5381.07	5973.69	6467.07

TRUCK REPAIRS	ANNUAL MILEAGE							
	12500.	25000.	37500.	50000.	62500.	75000.	87500.	100000.
TRUCK BRAKING SYSTEM (MOSTLY BRAKE SHOES)	482.14	1027.09	1603.72	2059.77	2421.26	2716.50	2960.28	3170.87
BRAKE BEAMS	42.08	113.28	204.43	276.62	335.81	384.56	423.51	458.63
BRAKE HEAD WEAR PLATES	0.03	0.10	0.14	0.18	0.20	0.21	0.23	0.24
BRAKE BEAM WEAR PLATES	0.16	0.40	0.58	0.70	0.78	0.85	0.87	0.94
BRAKE HANGER OR CONNECTION PIN	2.04	4.21	6.84	9.05	10.82	12.27	13.52	14.57
BOTTOM ROD SAFETY SUPPORT	0.86	2.16	3.92	5.59	6.98	8.12	9.14	9.96
BRAKE BEAM SAFETY SUPPORT	0.06	0.23	0.36	0.43	0.51	0.56	0.57	0.62
BRAKE CONNECTION, BOTTOM	1.72	3.08	4.34	5.34	6.12	6.74	7.29	7.71
BRAKE CONNECTION, TOP	1.76	5.04	8.07	10.45	12.32	13.74	15.08	16.00
BRAKE LEVER	1.35	2.36	3.33	4.10	4.70	5.17	5.60	5.93
BRAKE LEVER GUIDE OR CARRIER	0.04	0.13	0.20	0.26	0.30	0.33	0.36	0.39
DEAD LEVER GUIDE	0.02	0.05	0.18	0.29	0.37	0.44	0.52	0.57
DEAD LEVER GUIDE BRACKET	0.02	0.05	0.12	0.17	0.21	0.25	0.28	0.30
BRAKE SHOES	429.50	891.24	1364.07	1737.56	2031.60	2271.51	2470.55	2641.37
BRAKE SHOE KEYS	2.52	4.76	7.13	9.02	10.52	11.73	12.74	13.62
WHEELSETS	823.32	1660.26	2532.18	3198.62	3717.76	4137.57	4491.71	4791.02
LUBRICATE ROLLER BEARINGS	12.23	29.95	43.93	54.34	62.09	68.02	73.20	77.50
ROLLER BEARINGS	139.15	280.38	417.85	526.48	611.09	679.69	737.38	786.18
ROLLER BEARING CAP SCREWS	0.09	0.25	0.37	0.45	0.51	0.56	0.60	0.63
ROLLER BEARING LUBRICATION FITTING	0.01	0.02	0.02	0.02	0.02	0.03	0.03	0.03
PEDESTAL ADAPTERS	10.02	38.07	70.05	95.28	115.26	131.13	145.24	156.51
WHEELS	306.00	622.17	929.86	1173.13	1362.69	1515.91	1644.85	1754.32
WHEEL LABOR	345.29	696.20	1038.44	1309.00	1519.74	1690.65	1834.46	1956.16
AXLES, ROLLER BEARINGS	10.52	21.22	31.66	39.92	46.35	51.57	55.95	59.67
OTHER TRUCK REPAIRS	43.11	198.19	364.42	492.83	593.82	670.95	741.11	796.94
TRUCK BOLSTERS	6.34	38.59	76.82	106.65	130.50	148.89	165.52	178.44
TRUCK BOLSTERS (REPAIRED)	0.32	3.35	6.51	9.27	11.47	13.20	14.51	16.04
CENTER PINS	1.39	3.99	6.95	9.30	11.14	12.64	13.94	15.06
CENTER PLATES	0.49	2.45	4.59	6.34	7.75	8.88	9.82	10.70
CENTER PLATE LINERS	4.20	18.46	33.70	46.15	55.76	63.59	70.20	76.20
TRUCK SIDE BEARINGS	2.91	8.01	12.81	16.60	19.58	21.91	23.89	25.55
FRICTION CASTINGS	5.21	19.24	34.11	45.55	54.49	61.27	67.46	72.65
SIDE BEARING SHIM	0.15	1.80	3.42	4.81	5.83	6.54	7.20	7.88
SIDE FRAMES	14.08	68.77	128.06	172.27	207.19	233.21	258.50	276.35
SIDE FRAMES (REPAIRED)	0.20	2.47	4.41	6.06	7.24	8.04	8.69	9.56
SPRING PLANKS (REPAIRED)	0.00	0.00	0.01	0.01	0.02	0.03	0.03	0.04
OUTER SPRINGS	3.65	14.70	24.62	32.05	37.79	42.16	45.83	48.88
INNER SPRINGS	1.73	6.50	10.90	14.17	16.72	18.63	20.30	21.64
STABILIZER SPRINGS	0.73	3.35	6.02	8.10	9.71	10.91	11.96	12.89
TRUCK SPRING FRICTION SNUBBER	0.01	0.02	0.02	0.03	0.03	0.03	0.03	0.03
TRUCK SPRING SHIM, WOOD	0.00	0.00	0.00	0.01	0.01	0.01	0.01	0.01
STEEL	0.01	0.03	0.14	0.23	0.29	0.35	0.41	0.45
MANUFACTURED MATERIAL (TRUCK)	1.71	6.46	11.31	15.23	18.28	20.65	22.80	24.54
TRUCK TOTAL	1348.56	2913.54	4500.31	5751.23	6732.84	7525.02	8193.09	8758.83

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US DOT, FRA, G Sheldon, G Bakken, K Cappel, D Gibson, A Gilchrist, K Glaser, DV RamaChandran