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TDOP

TRUCK DESIGN OPTIMIZATION PROJECT PHASE II

ANALYSIS PLAN

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NOTICE

NOTICE

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The Analysis Plan will be supplemented by two documents: The Analytical Tool Assessment Report (FRA Report No. FRA/ORD-79/36) and The Analytical Tool Validation Report

16. Abstract

The purpose of this document is to set forth the Analysis Plan which will be followed in the Truck Design Optimization Project (TDOP) Phase II. The Analysis Plan defines the means by which Type I (standard) and Type II (premium) freight car trucks will be characterized and compared. The specific objectives of the analysis task are: 1) to define the requirements for test data and simulation results which will establish performance specifications; 2) to determine the extent to which field test data can be extrapolated; and 3) to develop criteria for validating analytical models.

The evaluation of Type I and Type II trucks addresses four important aspects of performance relating to railroad safety or effectiveness. The four are: lateral stability, trackability, curve negotiation, and ride quality. The Analysis Plan is divided into sections relating to each of the four performance regimes. Each section contains a discussion of performance indices, analysis requirements, model utilization, test data utilization, special considerations for Type II trucks, and summary.

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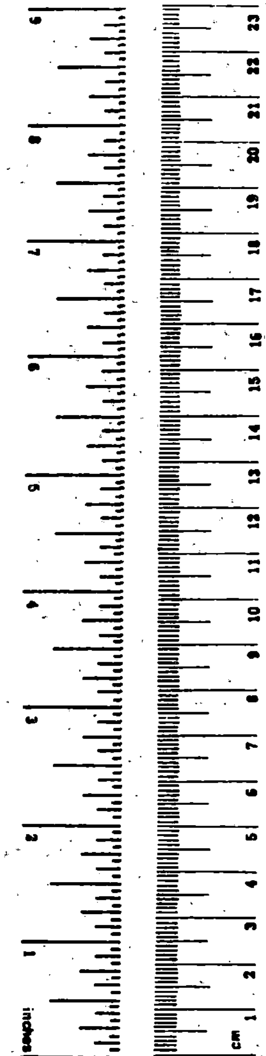
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

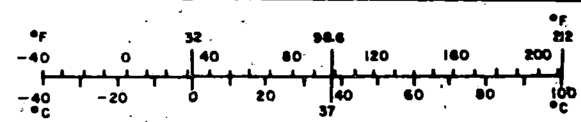
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	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
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



EXECUTIVE SUMMARY

The purpose of this report is to document the Analysis Plan that will be followed to provide the analytical support for the Truck Design Optimization Project (TDOP), Phase II. The overall objective of TDOP is the characterization and specification of Type I (standard) and Type II (premium) freight car trucks. The characterization will permit relationships to be drawn between truck parameter variations and truck performance.

The analytical support for TDOP Phase II is divided into these principal areas:

- Reduction and interpretation of test data.
- Use of analytical tools.
- Assessment of analytical tools (engineering formulae, computer simulations, etc.).
- Definition of analytical tool validation criteria.
- Validation of analytical tools.

The Analysis Plan addresses the first and second items above while the specifics of the latter three areas are addressed in two companion documents, "The Analytical Tool Assessment Report" and "The Analytical Tool Validation Report."

In evaluating Type I and Type II trucks, four performance regimes have been identified. These four regimes are: lateral stability, trackability, curve negotiation, and ride quality. Each of these regimes relates to aspects of truck performance which directly affect the safety or effectiveness of rail freight systems. Taken collectively, the four performance regimes permit a balanced evaluation of the performance of a given truck.

Analysis in the lateral stability regime will define the hunting characteristics of a truck. Field test data will be used to investigate the influence of operational and environmental factors on truck hunting. In addition, linear and nonlinear modeling techniques will be employed to predict the sensitivity of a truck to parameter variations as well as the wear rate of its key components.

Analysis in the regime of trackability (the ability of the truck to maintain adequate loads for guidance on all wheels for all extremes of in-service operation) will include a track twist load equalization evaluation and a harmonic roll and bounce analysis on standard and premium trucks. Track twist will rely mostly on test and survey data obtained during Phase II field testing of Type I and II trucks and data acquired from the Rail Dynamics Laboratory. In contrast, harmonic roll and bounce analysis will depend heavily on models for the investigation of these extreme dynamics with comparisons being made between simple and complex representations.

Curve negotiation regime studies will evaluate Type I and II truck designs with respect to a curving performance index which characterizes the severity of lateral wheel/rail loading during curving. Steady state models will be used to compare trucks on the basis of wheel wear, fuel consumption, and rail wear in curves. Time domain models will be used to analyze derailment potential and curve entry wear. Finally, the ride quality regime will deal with the normal dynamics of the freight car. Its primary objective will be to develop methodologies for predicting a lading vibration environment.

The report consists of three sections and several appendices. Sections 1 and 2 provide an introduction and overview of the objectives of the TDOP Analysis Plan as well as a more detailed discussion of the performance regimes chosen. Section 3 constitutes the main body of the report. It is organized according to the four performance regimes discussed above. The discussions pertaining to each performance regime deal with the performance indices, analysis requirements, test data utilization, model utilization, and special considerations for Type II trucks. Supplementary material including instrumentation plans are provided in the appendices.

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SECTION 1 - INTRODUCTION

As an objective of the TDOP Phase II effort, analytical studies will be conducted to evaluate and characterize the performance of Type I (standard) and Type II (premium) freight car truck configurations. These studies will also investigate the relationship between truck parameter variations and performance capabilities.

Both field test data and simulation models will be used to define performance boundaries for Type I and Type II freight car truck configurations. The analysis effort and the field testing task have been designed to complement each other in meeting the project objective. The engineering interpretation of the data will serve as a significant input into the economic cost/benefit trade-off studies. This Analysis Plan describes the work which will be performed in the TDOP Phase II analytical tasks.

The Analysis Plan provides an overview for two complementary documents which further define the analysis tasks. The first of the two complementary documents is FRA Report No. FRA/ORD-79/36, the "Analytical Tool Assessment Report," which reviews the capabilities of existing car/truck simulation programs with respect to requirements set forth in the Analysis Plan. The second document is FRA Report No. FRA/ORD-79/40, the "Analytical Tool Validation Report," which defines quantitative validation requirements for the car/truck simulations, and documents the results of the validation exercises performed.

The purpose of the Analysis Plan is to define the procedures by which the following three main objectives will be met.

- To establish the relationships between truck parameter variations and truck performance indices for both Type I and Type II trucks. Model and test data utilization will be defined for each performance regime.
- To determine the analysis requirements needed to extend and extrapolate the field test data for both Type I and Type II trucks.
- To establish model application requirements which will provide a framework for the model validation criteria.

SECTION 2 - TRUCK PERFORMANCE REGIMES

The discussion and analysis of truck performance has been divided into the following performance regimes (reference 1).

- Lateral stability
- Trackability, with the following subset regimes:
  - Track twist
  - Curve entry and exit
  - Harmonic roll and bounce
- Curve negotiation
- Ride quality

Each of these regimes identifies an aspect of truck performance which significantly influences the operational safety or effectiveness of freight systems. Evaluating trucks with respect to these four regimes then leads to a balanced assessment of overall truck performance.

2.1 LATERAL STABILITY

Lateral stability is the term used to describe the capability of a truck to inhibit self-induced lateral and yaw oscillations known as hunting. Hunting may be exhibited either by the truck alone, or by more complex interactions involving both trucks and carbody. Hunting is a safety as well as operational problem. High speed derailments, gage widening, and accelerated component and track wear are known to result from the hunting phenomenon. Finally, some lading damage may result from hunting in lightly loaded or half loaded cars.

2.2 TRACKABILITY

Trackability is the ability of the truck to maintain sufficient loads on all wheels to allow the development of guidance forces which prevent derailment for all extremes of in-service track geometry. The trackability regime includes as subsets the ability of the truck to accommodate: (1) track twist, (2) curve entry and exit dynamics, and (3) harmonic roll and bounce dynamics.

The ability of the truck to accommodate track twist refers to the maintenance of adequate wheel loads in the presence of cross level variations occurring within the wheelbase of the truck. This ability is important for successful negotiation of low sidings and extremely poor track in switch yards.

During curve entry and exit, the car and truck experience forces and motions which can impart extreme dynamic response to the carbody and truck. Excessive harmonic roll and flange contact can occur, resulting in wheel unloading and wheel climb in extreme situations. The truck must be able to maintain a smooth transition from the tangent to the curved track for a wide range of operating conditions and environmental factors.

Harmonic roll is a forced response resonance problem with the carbody responding with large amplitude, low center roll motions. The resonance problem is normally encountered on jointed rail track while traveling at speeds in the 10 to 20 mph range. Excessive carbody roll can result in side bearing contact, bottoming of the main suspensions springs and, ultimately, wheel liftoff and complete car rollover. The primary concern of the harmonic roll problem is one of safety although component deterioration, such as snubber wear and truck bolster failure, also occurs. Some lading damage may also be attributed to harmonic roll.

Bounce dynamics include carbody vertical and pitch motions. Bounce resonances are particularly a problem with shorter cars (around 20 ft in length). Car lengths that do not correspond to the rail length receive vertical and pitch as well as roll excitation. Bounce occurs at a higher speed than harmonic roll since the carbody pitch is a higher frequency mode than the roll mode (typically 3 to 5 Hz versus 0.7 to 1.8 Hz). At speeds around 40 to 65 mph, the vertical excitation from the half staggered rail corresponds to the pitch mode natural frequency of 3 to 4 Hz. Both safety and component deterioration are concerns of this regime.

2.3 CURVE NEGOTIATION

This performance regime considers the steady state curve negotiation capabilities of the truck. The ability of the truck to provide steering in curves while minimizing flange contact and sliding under a variety of operational conditions will be evaluated within this regime. The primary economic impact of poor steady state curving performance is wheel and rail wear and increased fuel consumption.

2.4 RIDE QUALITY

The ride quality regime includes all non-extreme car/truck dynamics. The truck dynamics during normal operation with the truck and carbody responding to track inputs with no abnormal or extreme motions (e.g., hunting, harmonic roll, or bounce) will be studied within the ride quality regime. The ride quality regime is essentially the study of freight car and truck forced response to random inputs. Motions resulting from system resonances, such as harmonic roll and hunting, are addressed in the other regimes. The primary economic impact of the ride quality regime is in the area of lading damage.

SECTION 3 - REGIME ANALYSIS PLANS

In this section, a plan is presented for each of the four regimes. Each regime analysis plan contains the following:

- Performance indices
- Analysis requirements
- Test data utilization
- Model utilization
- Summary

The performance indices, which are measureable quantities typical of the various regimes, are as defined in the TDOP Phase II Introductory Report (FRA Report No. FRA/ORD-78/53). Under the analysis requirements section, the problem is defined and the scope of analysis effort established. Test data from both Phase I and II will be used to investigate the actual car/truck dynamics and to validate models. The test data will also be used to quantify response levels within each regime. Next, the analytical tool requirements will be defined in the model utilization area. The formulation of validation criteria and the validation exercises themselves (to be documented in a separate report) appropriate to the depth and detail required of the analytical tools will also be carried out as part of the model utilization study.

Section 3 is divided into four major parts. Section 3.1 covers lateral stability. Section 3.2 discusses the static and quasi-static aspects of trackability which include load equalization in the presence of track twist and harmonic roll and bounce. The analysis of curve entry and exit dynamics, although a subset of the trackability regime, is included under curve negotiation, Section 3.3. The final Section, 3.4, covers ride quality.

3.1 LATERAL STABILITY

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 higher speeds on tangent track. The flow diagram shown in Figure 3-1 illustrates the lateral stability analysis tasks that will be performed (discussed in detail in the following paragraph). This flow diagram also includes the model validation effort which will be undertaken prior to using the simulation programs.

3.1.1 Performance Indices

The primary hunting performance indices are critical speed and lateral acceleration. Critical speed for a car/truck/lading configuration is the speed at which large amplitude hunting motion starts. Use of critical speed is somewhat complicated by the fact that no simple stability boundary exists for in-service car/truck configurations (reference 2). The tendency of a truck to hunt is strongly influenced by environmental factors, such as rail contamination and track geometry. A given truck may hunt, or not hunt, at different speeds on different days. However, the single critical speed predicted by linear Eigenvalue models can be used as a measure in parameter sensitivity studies and qualitative comparisons of car/truck configurations with respect to hunting susceptibility.

Each freight car truck has a basic kinematic wavelength at which it tends to oscillate laterally while traveling over tangent track. As speed is increased, the frequency of the lateral motion increases. Lateral accelerations characterize and allow comparison of the magnitude of these kinematic motions for various truck configurations and parameter variations. The lateral accelerations will also aid in determining the relative motion of the truck components for use in estimating wear.

Table 3-1 summarizes the performance criteria, performance indices, and economic impacts for the lateral stability regime. Detailed discussion of these items may be found in the TDOP Phase II Introductory Report (reference 1).

3.1.2 Analysis Requirements

Sustained limit cycle hunting motions can result in component deterioration through wear and fatigue as well as unsafe operation due to potential derailment. Test data will be used for both model validation and for quantifying in-service response levels. The influence of operating conditions and truck parameter variations on truck performance will be investigated.

Critical speed is of primary importance in characterizing truck performance. Ideally the critical speed is higher than the operating speed range expected for the truck. The effects of truck parameter variations and operating conditions on critical speed are important considerations in designing trucks and in matching trucks to carbodies for particular service conditions.

In analyzing the lateral stability regime, the following operational conditions will be varied in the model:

- Speed
- Carbody loading (empty cars are normally the worst hunters)

- Rail geometry

Any acceptable truck design should be able to maintain stable operation at a given maximum speed over the normal operating range of these variables. These operating variables will form the standard set of operating conditions against which the car/truck stability will be evaluated.

In addition, the following truck parameters will be studied:

- Yaw stiffness
- Tramming stiffness
- Primary suspension parameters between the side frame and wheelset
- Secondary suspension (consisting of the main suspension springs and damping mechanism between the side frame and truck bolster)
- Wheel profiles
- Truck kinematics (including the effects of wear and tolerance build-up)

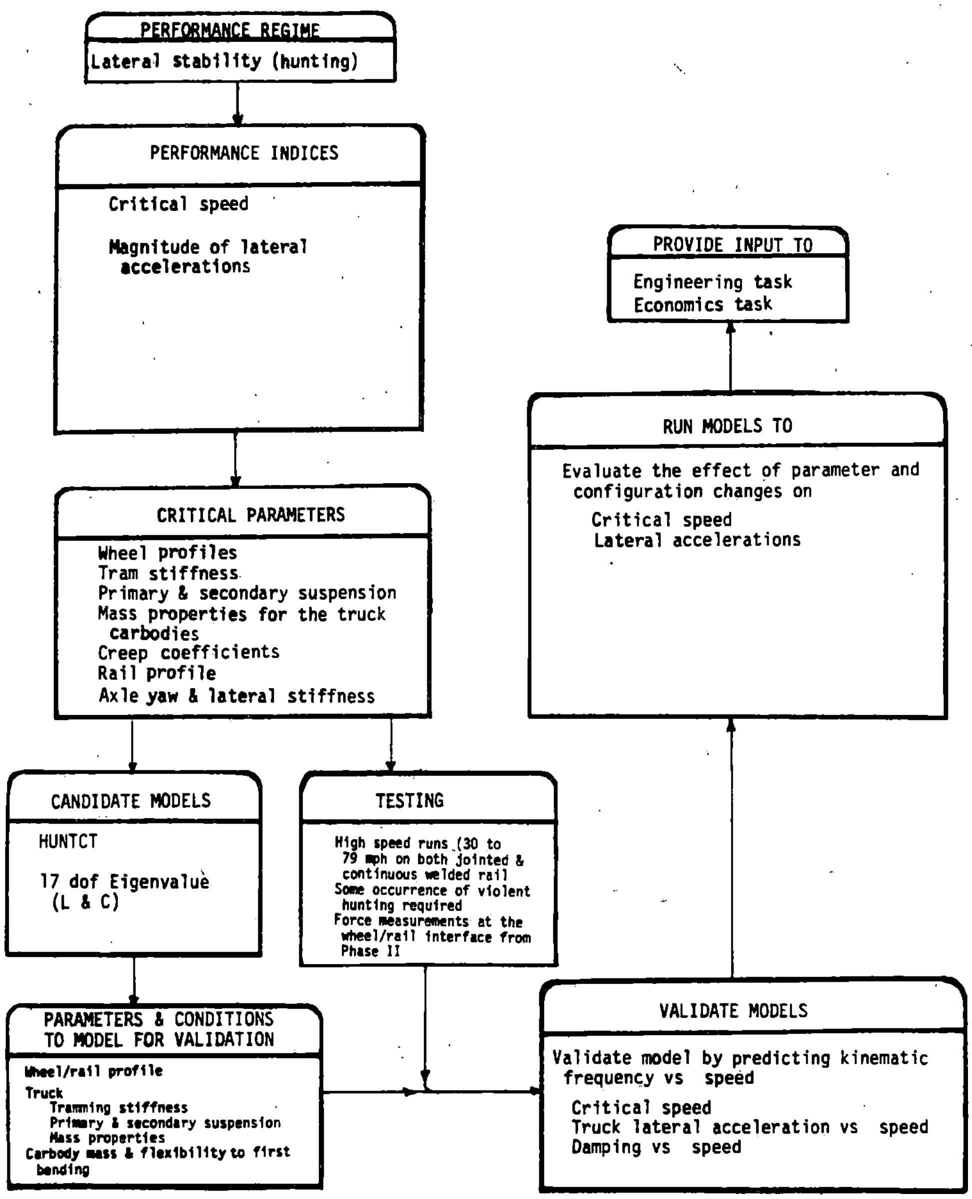

Figure 3-1. Flow Diagram for Lateral Stability Analysis

Table 3-1. Lateral Stability Regime

Performance Criteria	Performance Indices	Truck Parameters	Operational Parameters	Economic Impact Areas
Safe operation	Critical speed	Wheel profile	Track geometry	Component wear
High lateral stability	Lateral acceleration	Tramming stiffness	Speed	Safety considerations
High critical speed		Yaw stiffness	Lading	Lower maximum operating speed
Low sensitivity to worn wheels		Inter-axle lateral stiffness		Widening of gage
Low lateral acceleration near critical speed		Inter-axle yaw		

3.1.3 Test Data Utilization

During TDOP Phase I, a large number of high speed tests were run for various freight car configurations. These high speed test runs consisted of dwells of approximately 60 seconds duration at six different speeds: 30, 40, 50, 60, 70, and 79 mph. The car and truck configurations tested are shown in Table 3-2. The tests were performed on 70 and 100-ton Type I trucks manufactured by American Steel Foundries (ASF), and Standard Car Truck Company (Barber). The variations in the track type, wheel profiles, and lading condition are also shown in Table 3-2.

Of the 30 variations in test configuration, 17 have been chosen for data reduction and analysis during Phase II. The Phase I test data to be reduced and analyzed are indicated by the shadowed boxes in Table 3-2. The major portion of the tests for which the data is to be reduced are those performed on jointed rail. Emphasis has been placed on the jointed rail tests because the tendency of a truck configuration to hunt will be more easily identified on the rougher track. Once a "hunter" has been found on the jointed rail, the corresponding run on continuous welded rail (CWR) track can then be analyzed. Limiting the initial data reduction to the jointed rail tests permits a systematic means of bounding the data reduction and analysis problem to a manageable size.

The instrumentation used in the Phase I tests is shown in Figures 3-2a and 3-2b. Both the truck and carbody instrumentation are of interest for lateral stability characteristics. The mode of hunting experienced most frequently by freight trucks is one which is coupled closely with the carbody dynamics and as a result, is often referred to as "body hunting." Distinct carbody modes will be associated with each phase of the hunting motion as the test vehicle speed is increased. Preliminary data reduction of Phase I high speed tests has shown cases where the front and rear trucks hunt at different speeds and exhibit different modes.

Table 3-2. Phase I Test Runs on High Speed Track for Engineering Studies

		Empty				Loaded			
		New Wheels		Worn Wheels		New Wheels		Worn Wheels	
Truck	Carbody	Jointed	CWR	Jointed	CWR	Jointed	CWR	Jointed	CWR
ASF 100-Ton	100-Ton Hopper Car	-9- 030402 TWA DB 534 TP 416	030402 TEM/H TP 415			-16- 030401 TWA DB 434 TP 413	030401 TEM/H TP 411		
ASF 70-Ton	70-Ton Reefer Car	-2- 010201 TWA DB 579 TP 337	-7,8- 010201 TEM/H DB 404/391 TP 337	-12- 020203 TWA DB 531 TP 349	020203 TEM/H TP 348,349	-19- 010101 TWA DB 566 TP 312,314	010101 TEM/H TP 312-314	-17- 020303 TWA DB 492 TP [illegible]	020303 TEM/H TP 353
ASF 70-Ton Low Level	70-Ton Stac Pac	-10- 030501 TWA DB 533 TP 394	030501 TEM/H TP 395	-3- 030503 TWA DB 433 TP 409	030503 TEM/H TP 408	-13- 030502 TWA DB 401 TP 407	030502 TEM/H TP 406		
Barber100-Ton	100-Ton Box Car	-11- 030202 TWA DB 532 TP [illegible]	030202 TEM/H T 380,381			-18- 030201 TWA DB 567 TP [illegible]	030201 TEM/H TP 388,389		
Barber 70-Ton	70-Ton Reefer Car	-1- 030101 TWA DB 577 TP 391	-6,5- 030101 TEM/H DB 439/529 TP [illegible],381			-15- 030102 TWA DB 435 TP 377	030102 TEM/H TP 377,378		
	70-Ton Box Car	-4- 030302 TWA DB 417 TP [illegible]	030302 TEM/H TP 392			-14- 030301 TWA DB 416 TP 403	030301 TEM/H TP 403		

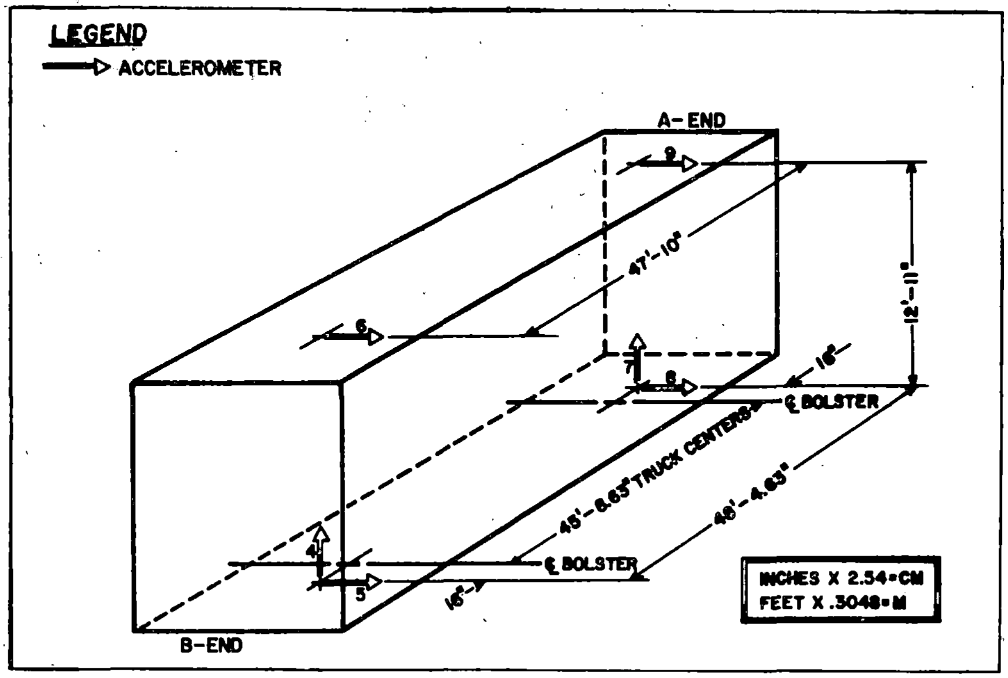

Figure 3-2a. Carbody Instrumentation

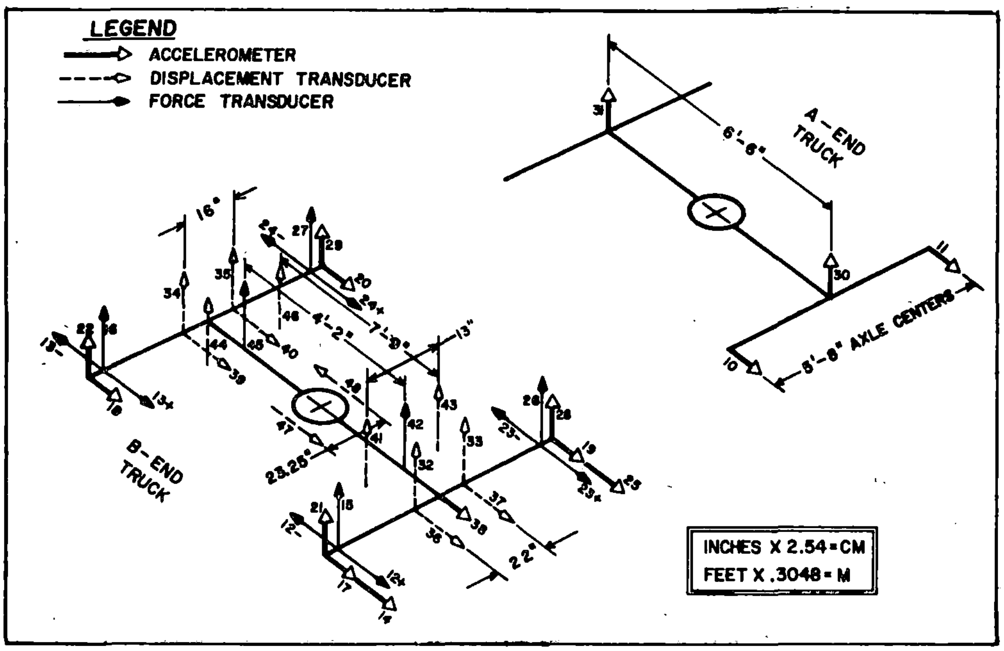

Figure 3-2b. Truck Instrumentation

The data reduction will be performed in two phases. The first phase will be a preliminary preview of the data to verify that all channels of interest were operational during the test run and that the data appear to be valid. This preview will be performed with the time domain plots of the data over the total time period of the test.

An example of a preview plot is shown in Figure 3-3. The time period of this plot covers the total 520 seconds of the high speed test run. The channels displayed are the lateral accelerations on the front and rear carbody over the center plate and the test vehicle speed. Note that this plot displays the front and rear truck difference in hunting speed discussed previously.

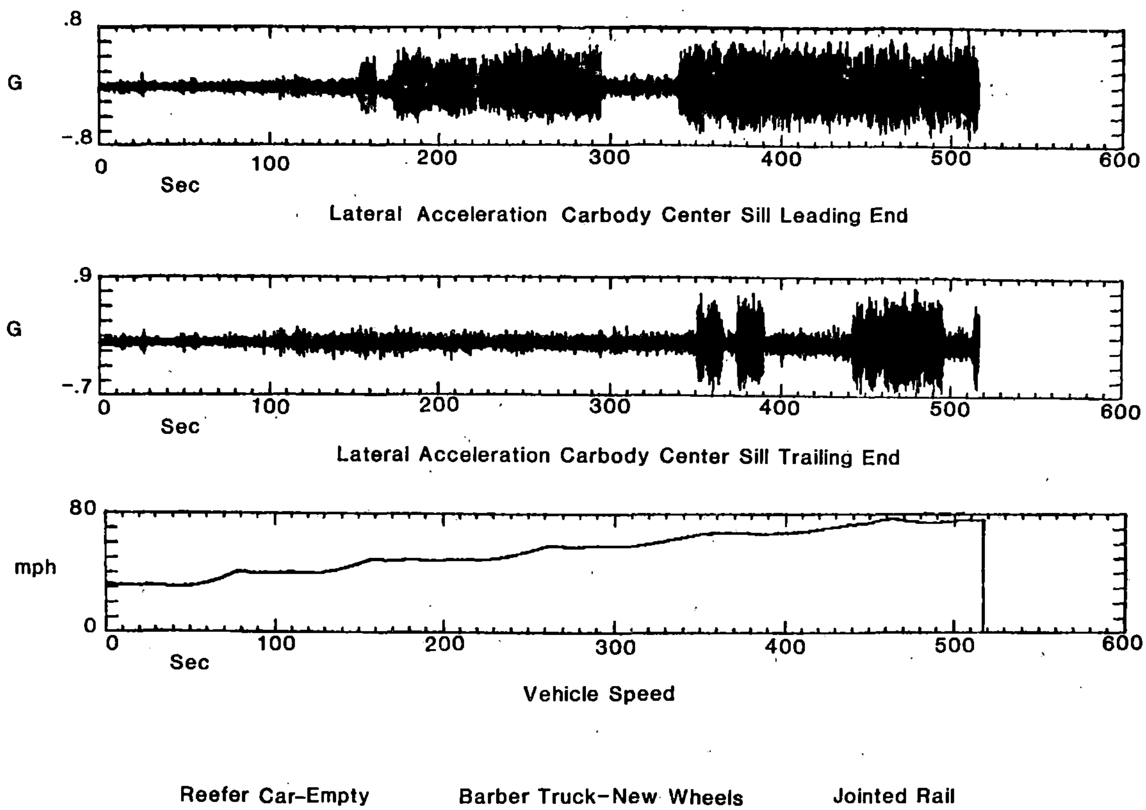

Figure 3-3. Example of a Preview Plot from a Phase I High Speed (30 to 79 mph) Test Run (This particular plot shows the front truck hunting at a lower speed than the rear truck.)

Use of carbody lateral acceleration measurements has been found to be a good indication of hunting. Measurements from the lateral accelerometers on the axles can also be used to obtain a hunting indication but these measurements contain a higher random response acceleration level and do not distinguish between the hunting and nonhunting states as clearly.

Once the data have been verified the main data reduction effort will begin. The effort will consist of the following steps for each of the test series:

Preview data

Select time periods at each dwell speed for which data are to be reduced

Establish the equations required to compute the derived carbody motions (roll, bounce, yaw, pitch, lateral, twist)

Reduce data

Truck and carbody measurements to be taken for the lateral stability regime are listed below.

Truck Measurements

- Lateral acceleration on leading axles of each truck

- Truck yaw

Carbody Measurements

- Individual carbody accelerations

- Lateral accelerations at leading and trailing ends at sill and roof levels

- Vertical acceleration at leading and trailing ends

- Combinations of the above accelerometer measurements

- Derived carbody motions

The lateral stability data reduction to be performed on each channel is as follows:

- Time history plots

- PSD (power spectral density) of the response at each speed in the frequency range 0 to 30 Hz with an analysis bandwidth of 0.1 Hz

- RMS (root mean square) vs speed for truck lateral and yaw accelerations and the derived carbody motions

- Band limited RMS calculations for truck lateral and yaw accelerations and the derived carbody motions including histograms of:

-- Probability distribution curves for the unaltered data

-- Exceedance probability distribution for absolute value for the data with the mean removed

The band limited RMS calculations will be taken over a 1-Hz bandwidth about the dominant frequency component below 5 Hz. This value is a measure of the energy at that dominant frequency.

Format examples for the histograms of the probability distribution and the distribution of the exceedance plots are shown in Figure 3-4. The histograms of the probability distribution functions indicate what percentage of the time a signal is above or below a given level. The exceedance probability distribution of the absolute value of the data with the mean removed allows one to directly read from the plot of the function the probability that *a signal is within the interval from* -x to +x.

In addition to the test data available from Phase I, testing of Type I and Type II trucks will be performed during Phase II. The instrumentation chosen for the Phase II testing was designed to provide the needed information for each regime. Table 3-3 lists the instrumentation which provides information on the lateral stability of the truck. A complete list of instrumentation is provided in Appendix B.

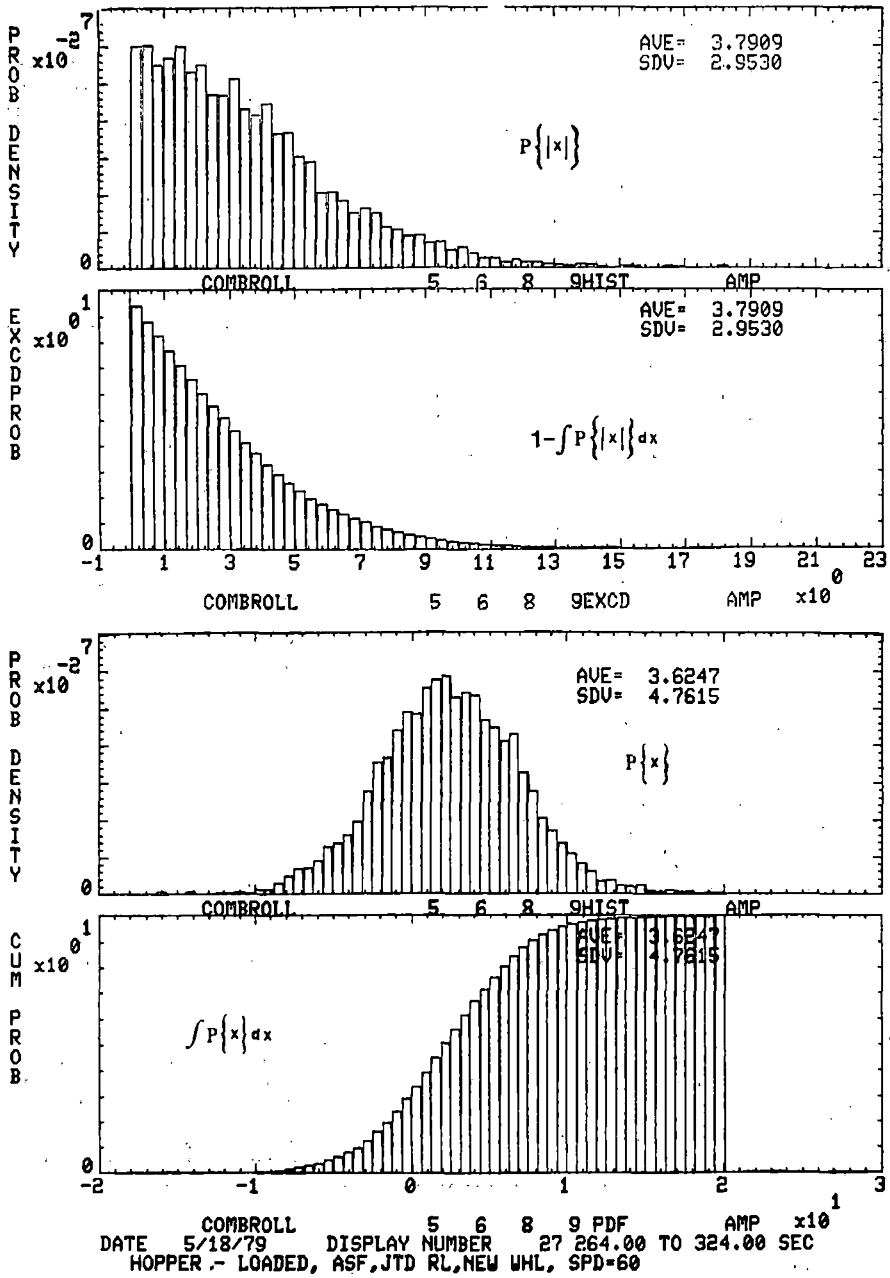

Figure 3-4. Formats for Probability Density and Distribution

Table 3-3. Instrumentation for Lateral Stability

MEASUREMENT TYPE	REQUIREMENTS	APPLICATION FOR LATERAL STABILITY ANALYSIS
ALD	Locate track test areas; mounted under Mobile Laboratory car	Required to align response data and track geometry measurements.
Speed	Speed from distance counter	Essential channel for correlation with vehicle response.
Carbody Lateral Acceleration A-End	Carbody lateral and yaw acceleration on bolster centerline	Used to determine carbody rigid and flexible body motion for correlation with truck motions. Primary interest will be coupling of body and truck motions which produce "body hunting."
Carbody Vertical Acceleration A-End	Carbody pitch, bounce, and roll acceleration on bolster centerline	
Truck Bolster Vertical Acceleration	Right truck bolster triaxial acceleration	Truck bolster triaxial will be used to correlate motions with the carbody and sideframes. Especially important in determining effectiveness of the secondary suspension in isolating truck and carbody lateral motions.
Truck Bolster Lateral Acceleration	Truck bolster triaxial acceleration	
Truck Bolster Longitudinal Acceleration	Truck bolster triaxial acceleration	
Carbody to Truck Swivel	Truck/carbody swivel and center plate slip	Provides a measure of truck yaw and lateral displacement at center plate.
Carbody to Truck Bolster Relative Displacement	Carbody/truck roll displacement	Used to measure harmonic roll and bounce responses.
Truck Bolster to Sideframe Lateral Displacement	Sideframe relative displacement truck tram	Calculation of truck modes during and prior to to the onset of hunting.
Spring Group Vertical Displacement Front	Spring group displacement sideframe pitch	Used to characterize secondary suspension dynamics.
Spring Group Vertical Displacement Rear	Spring group displacement sideframe pitch	
Fore Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	Axle bearing accelerations will be used to measure excitation resulting from the wheel/rail interaction.
Fore Axle Bearing Pocket Lateral Acceleration	Axle lateral acceleration forward	
Rear Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	
Rear Axle Bearing Pocket Lateral Acceleration	Axle lateral acceleration rear	
Vertical Acceleration Carbody Center	Triaxial acceleration at carbody center	Measures flexible body and rigid body motions when used with accelerations on carbody ends.
Lateral Acceleration	Triaxial acceleration at carbody center	
Wheel/Rail Force Measurement System	Lateral, vertical, and longitudinal forces at wheel/rail interface	These forces are required to determine actual creep coefficients in validating the detailed nonlinear time domain models.
Wheelset Angle of Attack	Angle of attack of the wheel	Provides additional information on the resultant wheelset motion and its effect on truck hunting.

3.1.4 Model Utilization

As part of the analysis task, simulation programs which had potential application to the Truck Design Optimization Project were assessed. The results of that task as related to lateral stability are shown in Table 3-4. Of the programs having potential application to the lateral stability regime, two have been chosen for primary use: the 17 Degree of Freedom (dof) Eigenvalue Model and the HUNTCT Model.

These two programs provide complementary analysis capabilities in the frequency and time domain. The 17 dof Eigenvalue program uses a linearized model of a carbody on two trucks. The program uses matrix inversion techniques to predict the critical speed, frequency, and shape of all modes including any unstable modes. The AAR Freight Car Hunting Model is another linear frequency domain model using 25 dof. It has also been identified in the model assessment exercise as an alternative to the 17 dof Eigenvalue Model. HUNTCT is a time domain simulation program capable of providing the same type of response output obtainable from an actual field test. A program of this type is intended to provide quantitative information on dynamic response.

It should be emphasized that the successful application of these models is dependent on the results of the validation exercises. None of the programs identified during the analytical tool assessments have had any significant amount of validation effort. HUNTCT, a Wyle-developed program, has been successfully compared with TDOP Phase I data in a limited number of cases. The data reduction previously discussed will provide the data against which the validation of both programs will be performed.

Validation of the 17 dof Eigenvalue Model will focus on the prediction of the critical speed and the frequency of hunting as well as the reasonableness of the shapes of the unstable modes. TDOP Phase I data will be the primary source of validation data.

Validation of HUNTCT is a more involved process. HUNTCT is a nonlinear, time domain model which has the potential for providing quantitative response data similar to that obtained from actual field tests. Complete validation of this model requires that truck and carbody responses be reproduced within a given tolerance over the speed range of interest, 30 to 79 mph.

Table 3-4. Lateral Stability Regime Tools

	TYPES OF TOOLS				
APPLICATION OF TOOLS	Qualitative Engineering Models, Simple Computations	Combinations of Simple Analytic Models	Linearized Models of Complete Vehicles	Detailed Models of Nonlinear Subsystems	Complete Nonlinear Models of Vehicles
Program or Model	Kinematic Models	DYNALIST	17 dof Eigenvalue Freight Car Hunting DYNALIST	Wheel/Rail Geometric Constraint Subroutine	HUNTCT
SAFETY Predict Dangerous Conditions			X		X
DETERIORATION Predict Damage & Wear				X	
PREDICTION OF IMPROVEMENTS - Type I - Type II Trucks	X	X	X	X	X
IMPROVEMENT OF MODEL & INSIGHT Explain discrepancies between theory and data	X		X	X	X
IDENTIFICATION OF NEEDS - Measurements - Tests			X		
EXTRAPOLATION To Conditions Not Tested (Savings in Test Effort)			X		X
CORRELATION With Models & Tests by Others			X		X

The exact validation critieria as well as the tolerances on the acceptable validation critieria are the topic of the Analytical Tool Validation Report.

The items below summarize the quantities which are of primary interest from each of the programs.

Frequency Domain Stability Analysis

Program
- 17 dof Eigenvalue

Output
- Critical speed
- Frequency at the critical speed
- Shape of the unstable mode

Time Domain Analysis

Program
- HUNTCT

Output
- Truck and carbody response as a function of speed
- Truck lateral acceleration
- Truck yaw acceleration
- Carbody accelerations: lateral, vertical, pitch, yaw, roll

The extent to which the simulation results can be used depends upon the results of the data reduction which will be performed prior to any extensive application of the models. Model use will be designed to supplement voids left by the test data and to investigate and understand the dynamics that occurred during the field testing. The analysis effort using the simulations will support both the engineering and the economics tasks. The engineering task support will be primarily in developing performance specifications for Type I and Type II trucks. The economics task support will provide data to correlate the damage identified in the maintenance records with freight car dynamics which caused the deterioration. Ideally, we would hope to correlate different types of damage with a specific dynamic regime; however, this is a goal which may not be possible in all cases.

3.1.4.1 Potential Simulation Problem Areas. Two problems have been identified which limit the usefulness of the Phase I data in recreating the track geometry excitation experienced by the test vehicle: (1) false automatic location detector (ALD) signals exist on both the test data recordings and on the track geometry recordings; and (2) no alignment data for either the right or left profile is included in the track geometry measurements on the tapes obtained from NTIS.

The first of the problems, the false ALD signals, prevents the input, the track geometry recordings, from being correlated in time with the output, the truck and carbody responses. For time domain simulations such as HUNTCT, the problem means that time histories of responses between model and test cannot be compared. With proper track alignment data, model and test results could still be compared in the frequency domain and in terms of various statistical measures; however, the Phase I track geometry recordings do not include this fundamental lateral stability input. Comparisons between the time domain model and the Phase I tests can only be done by making an assumption about the track alignment. It is proposed to substitute Phase II track geometry data which will include track alignment. The assumption is that the power spectra between the Southern Pacific test track during Phase I and the Union Pacific test track for Phase II will prove to be similar since they are of the same class and type of track. The assumption can be tested by comparing the track geometry measurements which were taken in Phase I with corresponding measurements of Phase II tests, namely the spatial PSD's of gage and right and left profile variations. If good agreement exists in these, then the use of the Phase II track geometry measurements, including the alignment data, is justified for reconstructing Phase I responses from model simulations.

3.1.5 Special Considerations Regarding Type II Trucks

A number of design features of the various Type II trucks are expected to have an influence on lateral stability. Particular attention will be paid in the analysis to the effect of wheelset interconnection, recommended wheel profiles, and tramming stiffness. Other features may also show some influence such as the lateral bolster freedom and primary versus secondary suspensions.

The lateral stability of Type II primary suspension trucks such as the Devine-Scales, NRUC Maxiride, and National Axle Motion reportedly is improved by virtue of their rigid frames (very high tramming stiffness). Secondary suspension Type II trucks such as the Dresser DR-1 retrofit, Barber-Scheffel, National Swing Motion, and ACF Fabricated designs are said to have achieved increased tramming stiffness (and thereby lateral stability) by various wheelset or side frame interconnections not found in Type I trucks.

3.1.6 Summary

The lateral stability regime treats the hunting characteristics of a truck. Field test data and simulation models will be used to investigate the influence of operational and environmental factors on truck hunting. Linear frequency domain modeling techniques will be used wherever possible to determine preliminary performance sensitivity to parameter variations. Nonlinear time domain simulations will be used to calculate motions and forces required for performance specification input not provided by the field test data.

3.2 TRACKABILITY REGIME

The trackability regime refers to the ability of the truck to maintain sufficient wheel loads for the development of forces sufficient to prevent derailment in all extremes of operation. The trackability regime is divided into the following subset regimes:

— Track twist (Section 3.2.1)

— Curve entry and exit, discussed as part of curve negotiation (Section 3.3)

— Harmonic roll and bounce (Section 3.2.2)

3.2.1 Track Twist

Track twist includes both the static and quasi-static (very low speed) load equalization capabilities of a truck in accommodating track irregularities occurring within the wheelbase length. Table 3-5 summarizes the track twist performance and the truck parameters influencing load equalization. The primary economic reason for improving load equalization is to minimize the occurrence of derailment.

Good load equalization requires that the truck maintain an adequate load on all four wheels under a range of track conditions and vehicle dynamics resulting from transient or periodic changes in these conditions. Even when a car is perfectly still, unequal wheel loads can exist depending upon the breakout force of the friction snubbers and center-of-gravity location of the car. For the quasi-static case, where the rail car is traveling at very low speeds (less than 5 mph), the unequal wheel loads plus the occurrence of a lateral force can result in derailment.

All freight car trucks have some capability for accommodating vertical rail irregularities that occur within the wheelbase distance. Flexible trucks depend upon side frame pitch and wheelset roll to accommodate these irregularities. Rigid trucks utilize the primary suspension between the wheelset and side frame to accommodate such irregularities. Adequate static and quasi-static equalization capabilities are very important for safely accommodating special track work, such as sidings and frogs.

A flow diagram for the track twist analysis task is shown in Figure 3-5. Each block of the diagram will be discussed in this section.

The track twist regime is unique in that heavy requirements will be placed on testing to evaluate static load equalization capabilities. Procedures will be developed for performing simple field and more complex laboratory tests of track twist. Some form of testing these procedures will be included in the truck evaluation methodology.

3.2.1.1 Performance Index. The performance index defined in the TDOP Phase II Introductory Report (reference 1) for the static load equalization capability with respect to track twist is the wheel unloading index (WUI).

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

Where

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

W_L = force on most lightly loaded wheel

θ = angle of twist within axle spacing of the truck (degrees, see Figure 3-6)

The WUI varies from zero for a perfectly equalized truck to $1/\theta$ for a truck with one wheel unloaded.

3.2.1.2 Analysis Requirements. The evaluation of track twist load equalization capability will be achieved primarily through the analysis of laboratory and field test data.

Vertical wheel force measurements are required at each wheel in order to evaluate the WUI. Wheel/rail force measurements will be provided by instrumented wheelsets for the field measurements. The laboratory testing will be performed at the Rail Dynamics Laboratory (RDL) located at the Transportation Test Center near Pueblo, Colorado. The Vibration Test Unit at the RDL has the built-in vertical force measurement capability required.

Test data analysis will consist of evaluating the WUI performance index for the full range of track twist measured during field tests and up to the maximum during laboratory testing.

3.2.1.3 Test Data Utilization. The proposed instrumentation for the TDOP Phase II Type I and Type II truck test program includes the necessary measurements for evaluating load equalization capability. Instrumented wheelsets will be used to measure vertical loads at the wheel/rail interfaces during the field tests. Table 3-6 lists field test instrumentation channels which have specific application to the track twist analysis. A complete list of instrumentation channels for Type I and Type II trucks, including their ranges and sensitivities, is presented in Appendix B. Field testing will be performed on Union Pacific track. Controlled dynamic testing at the Rail Dynamics Laboratory will also be performed depending upon availability of the Vibration Test Unit.

For Phase II Type I and Type II truck testing, two options are available for evaluating static load equalization capability. The simplest method is done by jacking one corner of a truck while recording the wheel loads. This procedure has been used by railroads (without the benefit of the vertical load measurement capability) to evaluate track twist accommodation. Without the wheel load measurements, the load equalization capability is determined by the jack height at which one of the other wheels on the truck lifts off.

A rigorous investigation of track twist can be performed under the controlled laboratory conditions at the RDL. The Vibration Test Unit (VTU) provides the capability for quickly and efficiently evaluating static load equalization capability. This machine allows a fully loaded car/truck configuration to be mounted on eight vertical actuators. These vertical actuators can be positioned to crosslevel differences of up to 6 inches between any of the four wheels of a truck. Overhead cranes at the RDL can be used to quickly change the load magnitude and center of gravity location.

The position and vertical load measurement capabilities required for these tests are an integral part of the VTU actuators. This capability will minimize the truck instrumentation requirements and greatly speed the static load equalization tests. These tests will be performed as part of the currently planned Phase II truck component tests to be performed at the RDL. It is anticipated that static load equalization test procedures (refined during the component phase of the test program) will form part of the overall truck performance evaluation criteria.

Table 3-5. Track Twist Regime

Performance Criteria	Performance Indices	Truck Parameters	Operational Parameters	Economic Impact Areas
Maintenance of equal wheel loads in the presence of track twist	Wheel unloading index	Torsional stiffness and free play Maximum accommodation angle Truck kinematics	Track geometry Speed Lading	Derailment costs

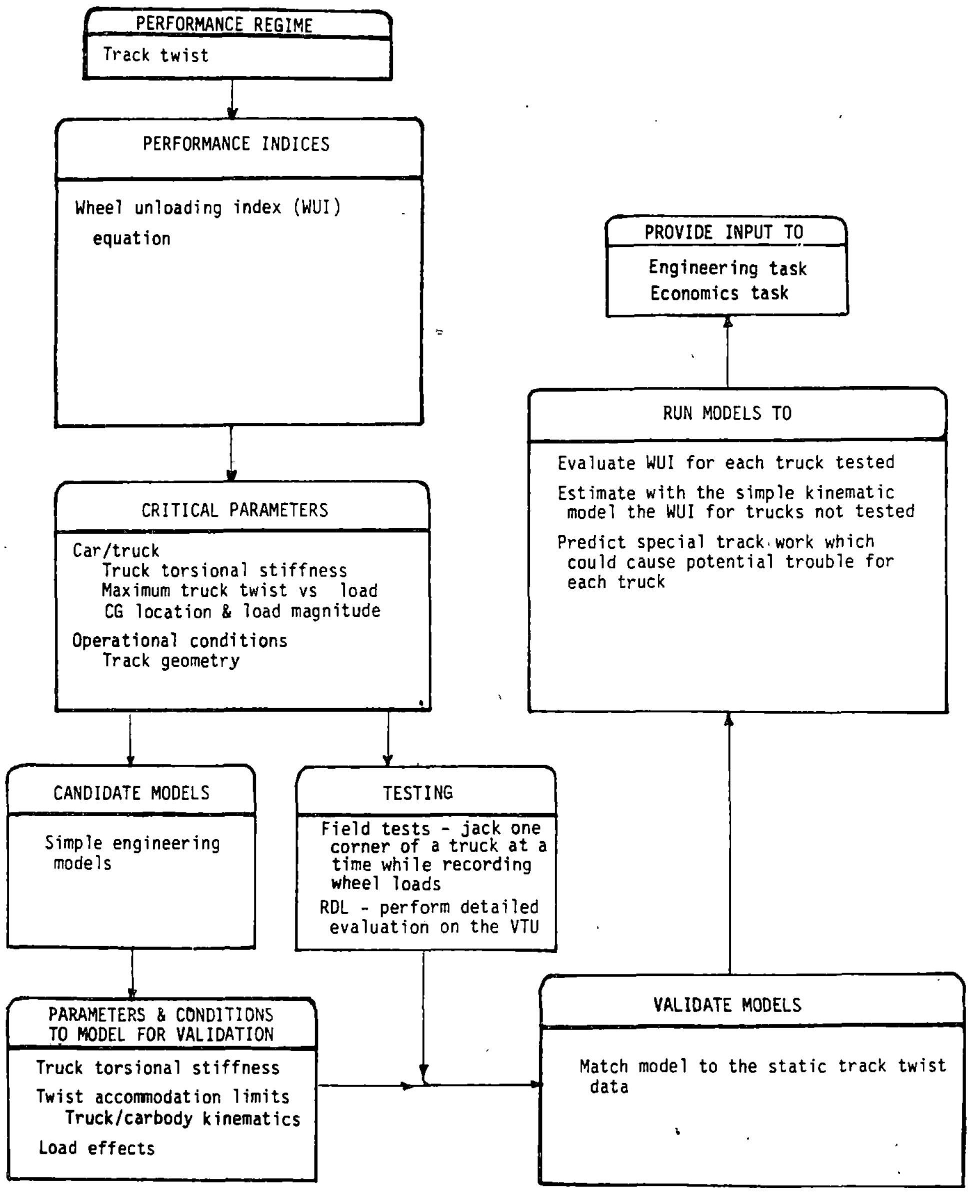

Figure 3-5. Flow Diagram for Track Twist Analysis

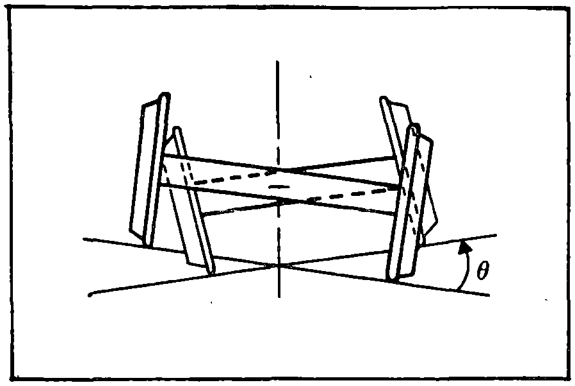

Figure 3-6. Definition of the Track Twist Angle

The VTU has the capability to duplicate a full range of actual track twist conditions by varying roll amplitude and roll center location. Investigation of all these conditions is required to fully determine the effect of carbody center of gravity variations and the effect of the different modes of lockup that can occur in the friction snubbers. Identification of the worst case conditions will allow correlation with existing track geometries encountered in yards, sidings, and special track work. The combinations of wheelset roll amplitude and roll center variations can be easily investigated on the VTU by slowly and continuously varying the actuators to test all possible configurations while simultneously recording the vertical load at each wheel.

Track twist can be set up by varying the twist amplitude and the center of rotation of wheelsets one at a time and two at a time. Separate plots of each case will be required. A typical plot of WUI versus track twist is shown in Figure 3-7. During the tests, data will be recorded for both increasing and decreasing track twist in order to detect any hysteresis. It is possible to have different wheel load distributions even for the static load cases depending upon how the friction snubbers lock up when they come to rest.

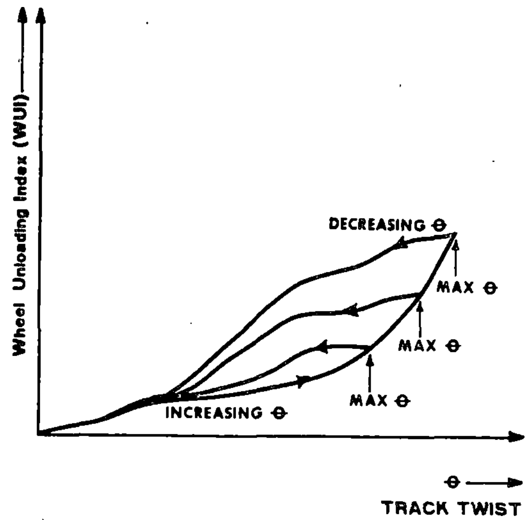

Figure 3-7. Typical WUI Plot for Several Maximum Values of Track Twist

Table 3-6. Instrumentation for Track Twist

MEASUREMENT TYPE	REQUIREMENTS	APPLICATION FOR TRACK TWIST ANALYSIS
Brake Cylinder Pressure	Measure brake line pressures to insure inadvertant braking from being mixed with data	Indicates whether brakes were on or off during the static track twist jacking tests.
Carbody to Truck Swivel	Truck/carbody swivel and center plate slip	Measures center plate lateral slip during track twist tests.
Carbody to Truck Bolster Relative Displacement	Carbody/truck roll displacement	Measures carbody roll relative to the truck bolster during track twist tests.
Spring Group Vertical Displacement Front	Spring group displacement and sideframe pitch	Measures the roll displacement of the truck bolster relative to the sideframe. Spring group displacements will be used to determine total roll of the carbody.
Spring Group Vertical Displacement Rear	Spring group displacement and sideframe pitch	
Wheel/Rail Force Measurement System	Lateral, vertical, and longitudinal forces at the wheel/rail interface	Required during the static track twist jacking tests to measure vertical load, at each wheel. These loads will be used as input to the WUI.

3.2.1.4 Model Utilization. Although track twist has not received as much attention from the rail research community as say, lateral stability, a number of applicable analytical tools are available to TDOP Phase II. The range of these analytical tools varies from extensive nonlinear time domain models, such as the AAR Flexible Carbody Model (reference 3) to relatively simple engineering calculations. In the case of the rigid trucks with primary suspension, the capability may be investigated with simple engineering models. Flexible truck twist capability can be investigated similarly by calculating the maximum twist allowed by tolerances between the bearing adapter and wheelset and from the estimated freedom allowed by the friction snubber mechanism. The friction snubber is currently thought to be the dominant limiting factor in track twist accommodation.

Car/truck models will be used to determine the effect of truck and carbody parameters on load equalization. The parameters to be varied include center of gravity height, spring travel and stiffness for both the primary and secondary suspensions, effects of the friction snubber (which will be more closely characterized as a result of a Phase II test program), and loading. The primary model outputs of interest are vertical and lateral loads at the wheel/rail interface, maximum carbody roll for the quasi-static case, duration of any wheel lift achieved, and maximum track twist angle which can be accommodated.

3.2.1.5 Special Considerations Regarding Type II Trucks. The Type II truck features which are expected to affect load equalization with respect to track twist are the frame rigidity and primary versus secondary suspension. The effect of snubbing devices will also be studied.

Vertical displacement of the primary suspension will be instrumented on the Devine-Scales and NRUC Maxiride trucks. The unique compliance characteristics of the primary/secondary suspension arrangement of the Alusuisse truck will be examined closely. Special instrumentation will be deployed to measure wheel spread and bolster bending of the Alusuisse truck.

3.2.1.6 Summary. Track twist gives the truck the ability to maintain equal wheel loads for crosslevel variations occurring within the wheelbase. It is primarily a static or quasi-static (very low speed) regime. Most data for analyzing the track twist subset regime will be accumulated from field and laboratory testing. Field data will be acquired during the Phase II Type I and Type II truck test program. Laboratory data will be obtained from the Vibration Test Unit at the RDL. Simple static and kinematic models will be developed from the test data and used to evaluate track twist capability.

3.2.2 Harmonic Roll and Bounce

Harmonic roll and bounce are forced response resonance phenomena between the track and freight vehicle. Harmonic roll is most severe when the frequency of the roll excitation resulting from rail crosslevel irregularities is close to the natural frequency of the carbody in roll. The large amplitude roll responses normally occur within the 10 to 20 mph speed range over staggered rail. Less severe forms of harmonic roll may occur at higher speeds when the upper center roll mode is excited.

For a given carbody and truck suspension, the severity of roll behavior is determined by the track characteristics. At the extremes of carbody roll, the main suspension springs may go solid and the snubber energy dissipation ceases. Continued excitation can increase the kinetic energy of the carbody to the point that separation at the center plate may occur. In the extreme cases, carbody rollover will be severe enough to cause a derailment.

Bounce dynamics include the pure vertical and pitch motions of the carbody in the plane of the vertical and longitudinal axis. Bounce resonances are particularly a problem with shorter cars having truck center distances around 20 feet in length, which corresponds to half the 39-foot staggered rail joint distance. Car lengths that do not correspond to the rail length will receive some vertical and pitch as well as roll excitation. Bounce is a higher speed problem than harmonic roll because the car body pitch is a higher frequency mode than roll, typically 3 to 5 Hz versus 0.7 to 1.8 Hz for roll resonances. At speeds around 40 to 65 mph, the vertical excitation from the half staggered rail lengths coincides with pitch mode natural frequencies of 3 to 4 Hz.

Some of the areas of economic impact for the harmonic roll and bounce regime are probability of derailment, component deterioration through impact, and fatigue loads. For example, failures of truck bolsters on 100-ton trucks have been attributed to loads resulting from harmonic roll.

Figure 3-8 illustrates the flow diagram of the analysis that will be performed within the harmonic roll and bounce regime. Each item in the flow diagram will be discussed in the following sections.

As with the other regimes, the validation of analytical models precedes their use during analysis. The harmonic roll and bounce regime is one area where considerable effort has been expended by researchers in developing and validating models. Results from previous validation efforts by the AAR and at Wyle will simplify the validation process during TDOP Phase II. The following sections discuss the analysis requirements and the use of models and test data regarding this regime.

3.2.2.1 Performance Indices. The performance indices proposed for the harmonic roll regime are maximum roll angle for a given excitation and rate of energy dissipation. The maximum roll angle for a given car/truck combination and lading allows a direct comparison of expected performance for a given track excitation. The rate of energy dissipation is a direct measure of the damping effectiveness of the snubbers.

Table 3-7 is a compilation of the performance criteria, performance indices, and truck parameter variations which affect these indices and the possible areas of economic impact for harmonic roll and bounce.

3.2.2.2 Analysis Requirements. The harmonic roll and bounce analysis requirements include the prediction of truck and carbody responses and a study of the effect on performance of certain truck parameter variations and operational conditions. The truck and carbody response predictions will be used for safety evaluations (e.g., to study derailment situations) and to predict component deterioration through wear and fatigue. Truck performance will be evaluated for both internal parameter and external operational condition variations. The parameter variations include both design variations and variations due to component wear, such as in the friction snubber.

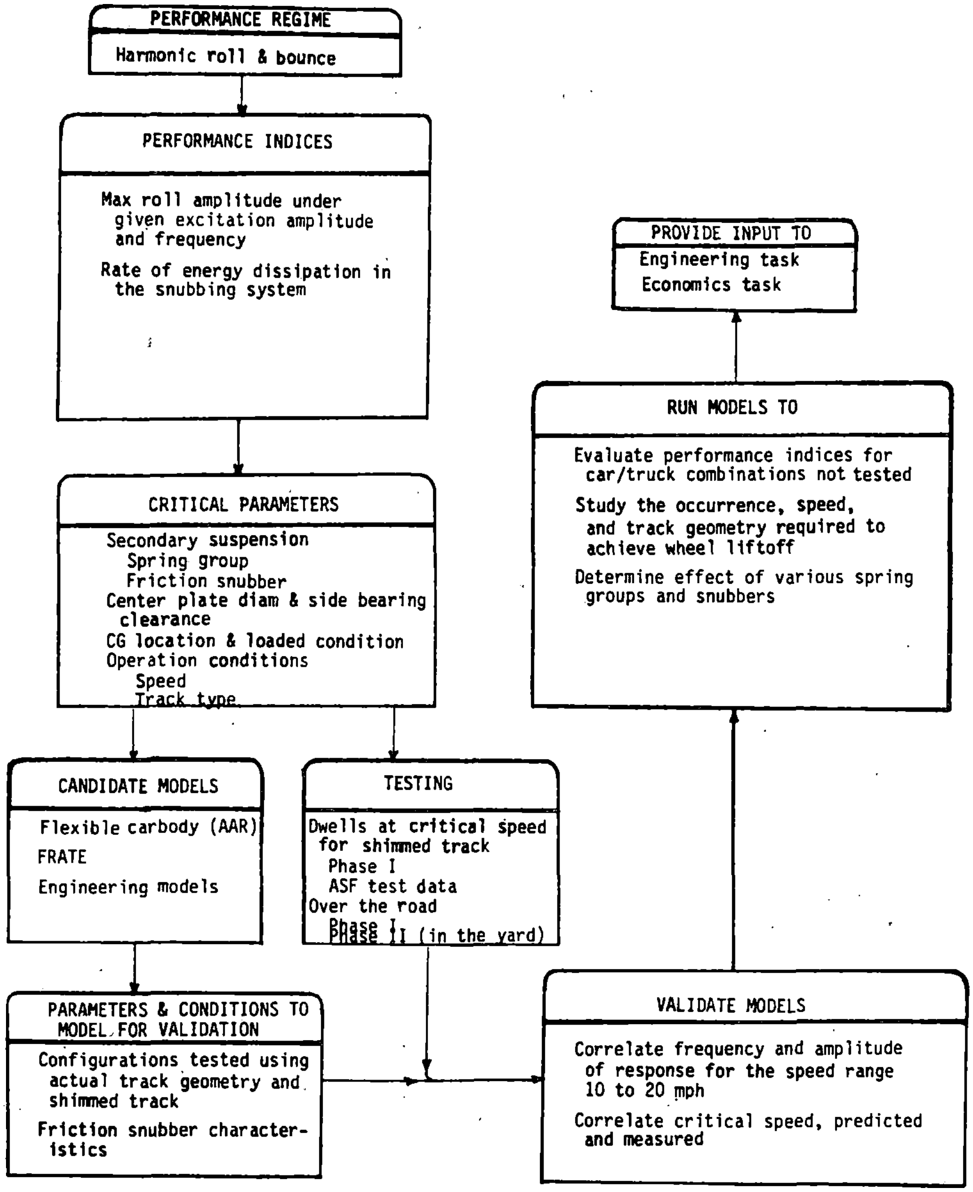

Figure 3-8. Flow Diagram for Harmonic Roll and Bounce Analysis

Table 3-7. Harmonic Roll and Bounce Regime

Performance Criteria	Performance Indices	Truck Parameters	Operational Parameters	Economic Impact Areas
Prevention of center plate and wheel liftoff	Maximum roll angle for a given excitation	Friction snubber	Track geometry	Safety considerations
Low maximum roll angle		Friction snubber augmentation	Speed	Component wear
Rapid response decay		Spring group	Lading	Design changes
		Rate of energy dissipation		

In performing the analysis of harmonic roll and bounce dynamics, a series of car/truck models will be required. Initially, simple engineering models will be required to evaluate car/truck resonances and mode shapes for both harmonic roll and bounce. Using these simple linear models, the effect of major parameter variations such as suspension spring group stiffness and loading condition will be evaluated. These linear models will also be used to estimate the excitation and response boundaries at which nonlinear simulation techniques will be required.

The parameter variations which will be studied during the harmonic roll and bounce analysis are:

<u>Carbody Parameters</u>

- Body stiffness
- Load and center of gravity variations
- Side bearing clearance
- Center plate diameter

<u>Truck Parameters</u>

- Spring characteristics (e.g., stiffness, travel)
- Friction snubber characteristics and augmentation
- Side bearing location

<u>Operational Environment</u>

- Speed
- Crosslevel variation
- Random rail excitation

The truck parameter variations for the harmonic roll and bounce analysis are primarily concerned with the secondary suspension elements between the side frames and truck bolster. Note that the harmonic roll and bounce performance is a trade-off with ride quality performance. Modifications to the suspension system which minimize the harmonic roll and bounce problem could degrade ride quality. During the parameter variation analysis, the effect of changes on ride quality will be considered.

Operational conditions which will be varied during the analysis include speed, car loading, and rail geometry. These operational conditions will be varied over ranges encountered during normal operation. Response to these operational conditions must be within acceptable bounds. The determination of "acceptable bounds" on performance is one of the requirements of both the engineering and analysis tasks.

3.2.2.3 <u>Test Data Utilization.</u> Harmonic roll test data are available from both TDOP Phase I and supplementary data to be collected during Phase II. The test data will be used for both model validation and analysis of harmonic roll and bounce dynamics.

The applicable test data from Phase I (termed medium speed data) consist of the various freight car and truck configurations operated at 12 different speeds between 12 and 45 mph. Every test configuration was run at seven different speeds between 15 and 45 mph at 5-mph increments. Then runs were made at five additional speeds at 2-mph increments. The smaller speed increment runs started at 12 mph for the fully loaded configurations and at 24 mph for the empty condition. The 2-mph increments were run for identification of the harmonic roll critical speed (the speed at which the roll response reaches its maximum amplitude).

These medium speed tests were performed over jointed rail track. Additional tests, referred to as the modified track tests, were performed over specially shimmed track. An initial study of this test series has been performed. This study showed that the only test variation which had any significant impact on the harmonic roll response was the load condition. No further data reduction is currently planned for the shimmed track tests other than for model validation.

Data reduction of the Phase I medium speed tests will be performed for the following four configurations:

- <u>100-ton hopper car</u>

  ASF 100-ton truck
  (empty and loaded)

- <u>100-ton box car</u>

  Barber 100-ton truck
  (empty and loaded)

- <u>70-ton box car</u>

  Barber 70-ton truck
  (empty and loaded)

- <u>70-ton refrigerator car</u>

  ASF 70-ton truck
  (empty and loaded)

The medium speed test data reduction will include the following output for each of the 12 different speeds.

- Time histories
- PSD's
- Narrow band RMS response versus frequency

Vehicle responses of interest include the following carbody motions calculated from the carbody accelerometers and referenced to the center of gravity.

- Lateral
- Yaw
- Roll
- Twist
- Pitch
- Bounce
- Roll center

Roll angles are calculated from these measured displacements:

- Carbody to truck bolster
- Truck bolster to side frame
- Carbody to axle

The primary application of the medium speed data reduction will be in developing performance specifications for Type I trucks. Type I truck tests to be performed during Phase II will also provide additional harmonic roll and bounce data, which will be reduced in the same manner as the Phase I data.

Table 3-8 lists the proposed instrumentation channels for the Phase II tests which have specific application areas to harmonic roll and bounce analysis. The complete list of instrumentation channels proposed for Type I and Type II testing is given in Appendix B. Included in the appendix are the measurement ranges, accuracy, and required frequency response fo each instrument.

The Phase I field test data will be reviewed to determine if any of the many test conditions which were varied influenced the harmonic roll response. During Phase I, 17 runs were performed over shimmed test track at five different speeds per run. These runs were performed to investigate the effects of the following system characteristics on roll and bounce dynamics:

- Carbody type
- Truck type
- Lading
- Spring nest configurations
- Snubbing configurations
- Auxiliary damping devices

The system response data for these runs will be tabulated in the form of carbody roll response maximum values and RMS levels. This response data will be correlated with the test parameter variations using linear regression analysis techniques. Results of this study will give direction for further investigations using the analytical models.

3.2.2.4 Model Utilization. Model requirements and model application areas of the harmonic roll and bounce regime are shown in Table 3-9. The table also lists the types of models, from simple spring mass to complex nonlinear models, which might be applicable in this area. Detailed discussion of these models can be found in the TDOP Phase II Analytical Tool Assessment Report.

Models available for harmonic roll and bounce analysis range from simple models to nonlinear time domain simulation models. The latter include all of the dominant nonlinearities in the car/truck system affecting harmonic roll and bounce response. An example of one of the more complete programs is the AAR Flexible Carbody Model (reference 3). This model can be used for performing detailed time domain response analysis of the full range of harmonic roll and bounce dynamics.

Two types of model excitation will be employed: actual track geometry field measurements, such as the class 2 test track used during the TDOP Phase II field test program and the standard deterministic track geometry. The deterministic track geometry will consist of a rectified sine wave or versed sine wave whose period corresponds to the rail length, typically 39 feet, with a 3/4-inch nominal peak displacement. Staggered rail joints are simulated by offsetting the zero values of the right and left profiles. The jointed rail track can be closely approximated by this technique. One advantage of this technique is that a large number of harmonic roll analysis runs have been made in other programs, such as the AAR Track/Train Dynamics project. Where possible results from these other sources can be compared with results from the TDOP analysis effort.

Of greatest interest to the railroads is the response of the car/truck system to actual track geometries. Actual track profiles typically consist of random sequences of high and low joints and soft spots. Track geometry measurements acquired during the Phase II field testing and the results of track geometry modeling work currently being done by ENSCO for the Department of Transportation (reference 4) can provide representative track geometry inputs to the modeling effort. The objectives of using actual track geometry inputs are to predict actual responses for service conditions and to evaluate the probability of derailment for a particular car/truck system. In addition, comparison of various Type I and Type II truck responses for the same field recorded track geometry will be made.

It has been reported by the TDOP industry consultants that service experience with high center of gravity cars shows two features:

- Derailment probability increases markedly for center of gravity height greater than 85 inches.
- Derailment probability is much lower for 70-ton cars than for 100-ton cars having similar center of gravity heights, even though the roll and wheel lift characteristics are comparable.

In order to investigate these observations, a model is required that predicts the forces at the wheel/rail interface, specifically the lateral and vertical forces at each wheel during roll oscillations. Such an output may show that critical guidance criteria are satisfied for given wheel/rail friction values, provided certain parameters such as center of gravity height and damping are restricted to certain values. Track compliance is an important parameter to vary during the studies and must be included in the model. Track compliance effects could be one of the factors that accounts for the difference in the derailment probability noted above.

Simpler techniques are needed by the industry for estimating the roll and bounce critical speed for a given car/truck combination. One of the main objectives will be to use simple models for easier and faster methods to estimate the harmonic roll critical speed, other than performing lengthy simulations using complex, nonlinear time domain models. Initial efforts on the harmonic roll and bounce analysis effort will use linear models to calculate roll and bounce resonant frequencies and amplification factors. Initial estimates of the roll and bounce critical speeds will be based on assumed track geometry, such as the rectified sine wave profile.

Table 3-8. Instrumentation for Harmonic Roll and Bounce

MEASUREMENT TYPE	REQUIREMENTS	APPLICATION FOR HARMONIC ROLL & BOUNCE ANALYSIS
ALD	Locate track test areas; mounted under Mobile Laboratory car	Used to correlate track geometry and response measurements.
Speed	Speed from distance counter	Essential channel for correlation with vehicle response.
Carbody Lateral Acceleration B-End	Carbody lateral and yaw acceleration on bolster centerline	All carbody accelerations listed will be required in various combinations to determine the rigid body and flexible body motions. Primary flexible mode of interest is the torsion mode.
Carbody Vertical Acceleration A-End Center	Carbody pitch, bounce, and roll acceleration over center plate	
Carbody Vertical Acceleration A-End	Carbody pitch, bounce, and roll acceleration on bolster centerline	
Carbody Vertical Acceleration B-End Center	Carbody pitch, bounce, and roll acceleration over center plate	
Carbody Vertical Acceleration B-End	Carbody pitch, bounce, and roll acceleration on bolster centerline	
Truck Bolster Vertical Acceleration	Truck bolster triaxial acceleration	The truck bolster triaxial accelerations will completely measure the dynamics of the bolster which will be correlated with the carbody and sideframe motions to determine carbody rocking and suspension system isolation.
Truck Bolster Lateral Acceleration	Truck bolster triaxial acceleration	
Truck Bolster Longitudinal Acceleration	Truck bolster triaxial acceleration	
Carbody to Truck Swivel	Truck/carbody swivel and center plate slip	Center plate slip measurement will be used during harmonic roll motions.
Carbody to Truck Bolster	Carbody/truck roll displacement	Essential measurements for both harmonic roll andbounce testing. Provides direct measurement of carbody to bolster roll and bounce.
Spring Group Vertical Displacement Front	Spring group displacement and sideframe pitch	Essential measurements which provide bolster roll information and displacement across the spring group.
Spring Group Vertical Displacement Rear	Spring group displacement and sideframe pitch	
Fore Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	Axle bearing pocket accelerations will be used to help identify track geometry excitation and the resulting truck excitation.
Fore Axle Bearing Pocket Lateral Acceleration	Axle lateral acceleration forward	
Rear Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	
Rear Axle Bearing Pocket Lateral Acceleration	Axle lateral acceleration rear	
Vertical Acceleration Carbody Center	Triaxial acceleration at carbody center	
Vertical Acceleration Carbody B-End	Roll center for carbody roll	Used along with carbody lateral accelerations to calculate the instantaneous roll center.
Wheel/Rail Force Measurement System	Lateral, vertical, and longitudinal forces at the wheel/rail interface	Measure wheel loads during harmonic roll and bounce.
Wheelset Angle of Attack	Angle of attack of the wheel	

Complex time domain models, such as the AAR Flexible Carbody Model, will be used for the detailed harmonic roll and bounce studies. Model outputs will be reduced and presented in the manner discussed in Appendix A. Outputs from the complex time domain simulations will be combined and reduced to provide the same information as the field test data with regard to component motion and forces. The simulations will be used to examine hazardous dynamics and very large vehicle responses which occur at the harmonic roll and bounce critical speeds.

3.2.2.5 Special Considerations Regarding Type II Trucks. 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, such as the National Swing Motion. The ACF Fabricated truck features support pads which are located over the side frames. The pads bear the weight of the carbody rather than the center plate and reportedly improve roll control. The roll behavior of the Alusuisse truck will be closely examined since lateral weight transfer causes wheelbase spread on the loaded side and hence a steering action. Type II trucks with primary suspensions which reduce the unsprung mass should reduce shocks due to roll and bounce.

3.2.2.6 Summary. The analysis of the harmonic roll and bounce regime will depend heavily upon models rather than field testing because the models will 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 1) parameter variations, such as spring characteristics and 2) operating conditions, such as speed and track condition. The emphasis will be on the use of frequency and nonlinear time domain simulations of the car/truck system, e.g., the AAR Flexible Carbody Model. Actual track geometry data acquired during Phase I and II, including field testing of the Friction Snubber Force Measurement System, will provide inputs to the modeling effort representative of service conditions. Techniques will be developed using simple models for easier and faster methods of estimating the harmonic roll critical speed.

Table 3-9. Harmonic Roll and Bounce Regime Tools

	TYPES OF TOOLS				
APPLICATION OF TOOLS	Qualitative Engineering Models, Simple Computations	Combinations of Simple Analytic Models	Linearized Models of Complete Vehicles	Detailed Models of Nonlinear Subsystems	Complete Nonlinear Models of Vehicles
Program or Model	Engineering Spring Mass Model		DYNALIST HALF, FULL	Flexible Carbody FRATE	Flexible Carbody FRATE
SAFETY Predict Dangerous Conditions					X
DETERIORATION Predict Damage & Wear			X	X	X
PREDICTION OF IMPROVEMENTS - Type I - Type II Trucks	X		X	X	X
IMPROVEMENT OF MODEL & INSIGHT Explain discrepancies between theory and data	X		X		X
IDENTIFICATION OF NEEDS - Measurements - Tests			X		
EXTRAPOLATION To Conditions Not Tested (Savings in Test Effort)			X		X
CORRELATION With Models & Tests by Others	X		X		X

3.3 CURVE NEGOTIATION

The curve negotiation regime discussed in this section considers both steady state phenomena and dynamic curving phenomena (curve entry and exit). Although curve entry and exit is generally considered a subset of the trackability regime, dynamic curving will be discussed in this section because many of the same performance indices and operational factors affect both steady state and dynamic curving. Figure 3-9 illustrates the flow of the curve negotiation analysis tasks. Table 3-10 lists the performance criteria, performance indices, and economic impacts considered.

This section presents the performance indices which will be used to quantify the curving performance of a truck. The data reduction and analysis requirement are then discussed. Specific data reduction and analysis techniques are presented in Appendix A. The primary source of data will be supplied by Type I truck tests, to be performed during Phase II. The data reduction planned for these tests is presented along with the anticipated use of curving models to supplement the test data results. The full extent of the application of curving models will depend partially on the data obtained from the Type I truck field tests.

3.3.1 Performance Indices

The primary performance index for curve negotiation is the lateral force on the leading outer wheel, per thousand pounds of axle load, per degree of curve, at balance speed. The lateral wheel/rail force is the result of the flange force, the lateral creep or friction force at the tread contact point, and the lateral component of the gravitational force. As noted in the TDOP Phase II Introductory Report, the main influences for this index are associated with differences in wheel profile and creep coefficients.

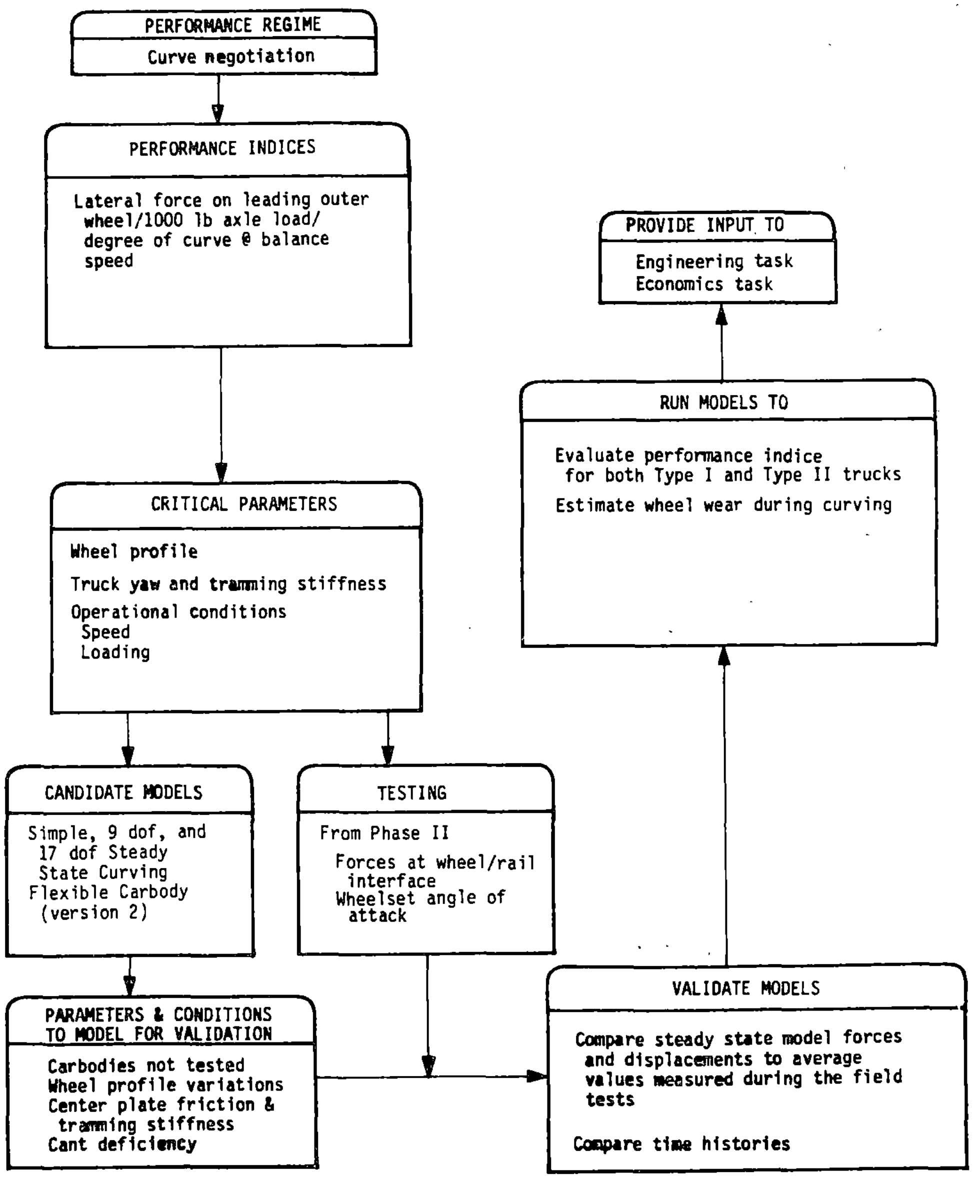

Figure 3-9. Flow Diagram for Curve Negotiation Analysis

Table 3-10. Curve Negotiation Regime

Performance Criteria	Performance Indices	Truck Parameters	Operational Parameters	Economic Impact Areas
Low wheel/rail wear	Lateral force on leading outer wheel/ 1000 lb axle load/ degree of curve at balance speed	Yaw stiffness	Track geometry	Wheel/rail wear
Low derailment probability		Tramming stiffness	Speed	Fuel consumption
Low fuel consumption		Wheel profile	Lading	

3.3.2 Analysis Requirements

The objective of the curve negotiation regime analysis is to determine the relationships between truck design and parameter variations, and the curving performance index. This objective will be achieved by using over-the-road field test data and curving simulations capable of predicting vehicle curving behavior.

The curve negotiation problem can be divided into two distinct areas. The first, steady-state curving, is concerned with the basic kinematic performance of the vehicle rounding the curve. The second, dynamic curving, is concerned with the transient behavior of the vehicle as it enters, traverses, and exits the curve. Typically the rail in curves will show the highest wear for a short stretch in the entry and exit portions of the curve, and relatively even wear through the curve. This is due to the higher forces required in the curve entry and exit to orient the trucks with respect to the rails. The wear in the entry and exit portions of a curve reflects the mix of vehicles transversing the curve. If the population of vehicles traversing a curve is dominated by a particular car/truck combination, the wear pattern in the curve will reflect the curve entry and exit dynamics of that combination.

The differences between steady state and dynamic curving suggest that two different measures of the performance index should be considered. The first is the maximum value due to transient behavior and the second is the average value while traversing the curve. In terms of derailment potential and rail wear during the entry and exit portions of the curve, the maximum value is of interest. In terms of wheel wear, fuel consumption, and wear along the entire length of the curved rail, the average value is of interest. Also, in order to compare the basic kinematic performance of different trucks, it is desirable to eliminate the transient dynamic effects due to curve entry and exit, as well as track irregularities and other extraneous factors such as over balance speed, wind, and rail surface contamination.

Table 3-11 is a list of factors affecting the performance index during curve negotiation. These factors include vehicle characteristics, track characteristics, and operating conditions. Since the purpose of the analysis is to relate the effects of truck parameter variations on the performance index, no attempt will be made to determine the effect of all the factors in the table. Rather, the effects of truck component variations will be determined for only a small set of nominal characteristics and operating conditions.

Table 3-12 lists those truck components which normally are variable on Type I trucks. According to the TDOP Phase I Final Report (reference 5), varying spring groups, gib clearance, and side bearing clearance had little effect on the curving performance of the Type I trucks tested. The report also indicated that center plate swivel (and thus center plate friction) and wheel profile had a significant effect on curving performance. The Phase I Final Report drew no conclusions with respect to the effect of varying friction snubbers on curving performance. Friction snubbers affect the tram stiffness of the truck which affects curving mechanisms.

Table 3-11. Factors Affecting Performance Indices During Curve Negotiation

Vehicle Characteristics
Wheel Profile
Truck Stiffness and Damping Characteristics
Vehicle Weight and Inertia Properties
Vehicle Physical Characteristics
Center Plate Friction
Friction Snubber Operation
Track Characteristics
Degree of Curve
Elevation of Curve
Track Curve History
Track Profile, Alignment, Gage, Cant, etc.
Track Flexibility
Grade
Length of Curve and Spiral
Type and Class of Track
Operating Conditions
Speed
Wind
Rail Contamination (Sand, Grease, Water, etc.)
Braking

Table 3-12. Variable Components of Type I Trucks

Spring Groups
Friction Snubbers
Center Plate
Gib Clearance
Side Bearing Clearance
Wheel Profile

Type II trucks generally have different flexibilities and constraints than Type I trucks. Thus, truck parameter variations which normally are not considered significant for Type I trucks must be taken into account. It is the intent of TDOP Phase II to determine the effect of variations of the most significant parameters on the performance index through testing and modeling.

3.3.3 Test Data Utilization

The TDOP Phase I data are not entirely suitable for characterizing the curving performance of Type I trucks because lateral wheel/rail forces were not measured adequately (reference 6). During TDOP Phase II, additional Type I truck testing, as well as testing of Type II trucks, will be performed. Phase II testing will be performed on trucks with instrumented wheelsets to provide wheel/rail interface forces.

The data acquired from testing Type I trucks will be used primarily for the validation of mathematical models which have been developed to predict curving performance. Once validated, the mathematical models can be used (in conjunction with the test data) to establish the baseline performance for a given car/truck system and to determine the effects of parameter variations on the performance index. The models can then be used to predict the performance of Type II trucks and can be used as an aid in planning the Type II truck tests. Testing of Type II trucks will serve to further validate the math models and to provide data for comparison of the performance index for the various trucks.

Table 3-13 describes those measurements which are of most interest to curving and indicates what use will be made of each measurement. Appendix B contains the complete list of proposed instrumentation for Type I and Type II testing.

In many cases the variables of interest will be calculated from two or more data channels. Examples are vehicle position (channels 1 and 2 of Table 3-13), B-end lateral coupler force (channels 6 and 8), A-end lateral coupler force (channels 7 and 9), carbody to truck swivel (channels 20 and 21), truck bolster lateral displacement (channels 24-29) carbody to side frame roll angle (channels 30-33). Creep forces can be estimated from the lateral wheel force measurements. If one wheel on an axle is in flange contact, the lateral force on the other wheel will be predominantly due to creep friction. Forces on the wheel in flange contact will be more difficult to decipher and will require a detailed analysis of the wheel/rail geometry.

Using the truck swivel and tram calculations and known geometrical relationships, an estimate of the wheel/rail angle of attack is possible (assuming parallel axles and no lateral freedom between axles and side frames). An estimate of the creep coefficient is possible using the estimated angle of attack and the measured creep force.

The type of data reduction required includes time history plots of acquired and derived data and calculation of average and maximum values. Maximum values are of most interest with respect to trackability, whereas average values are of interest when analyzing steady state curve negotiation. Time history plots are important in understanding both the transient and steady state behavior, and are of use in identifying anomalous behavior. Examples of the types of plots which will be generated are shown in Figures 3-10 through 3-12.

The Phase II Type I truck testing will provide data from which the effect of the following variables on the curving performance index for each carbody/truck configuration will be determined:

- Wheel profile (new or worn)
- Load (empty or fully loaded)
- Grade (uphill or downhill)
- Speed (equilibrium, over, or under)
- Braking (on or off)
- Track curvature (degree and direction)

Analytical tools will be used to interpolate and extend the effects of these variables as well as to determine the effects of the following:

- Truck stiffnesses
- Creep coefficients
- Vehicle physical properties

Consideration of the above factors allows the extension of results to other carbody/truck configurations and other operating conditions.

3.3.4 Model Utilization

The main focus in the curving studies is the forces developed at the wheel/rail interface. Analytical models to be used, then, must be capable of predicting these forces within an acceptable tolerance. Because of the need to investigate maximum and average values of lateral forces, two types of analytical models appear to be justified (see references 7 and 8 for general discussions of models). The first type includes steady state models which predict the steady state performance in curves. Both linear and nonlinear models can be used, with linear models used mainly for qualitative studies and nonlinear models used for quantitative studies. Table 3-14 shows potential application areas for these two types of models.

Linear steady state curving models have closed form solutions and may be solved easily with hand calculations or with a simple computer program. Nonlinear steady state curving models do not have closed form solutions and generally use an iterative computer process to solve the equations. Nonlinear steady state curving models are generally quite efficient in their use of computer time; the majority of the TDOP curving analysis will be performed using this type of tool. Model validation will consist of comparing the calculated steady state values of the performance index with the average of the values measured during the curve following the curve entry transients.

Table 3-13. Instrumentation for Curve Negotiation

MEASUREMENT TYPE	REQUIREMENTS	APPLICATION FOR CURVING ANALYSIS
ALD	Locate track test areas; mounted under Mobile Laboratory car	Used to correlate vehicle response data and track geometry measurements.
Speed	Speed from distance counter	Determine relation to balanced speed.
Brake Cylinder Pressure	Measure brake line pressures to insure inadvertant braking from being mixed with data.	Verify coasting condition.
A-End Coupler Force	Note: units have hysteresis of ± 200 lb but are 1 percent when either in tension or compression	Used to calculate energy expended while curving (fuel economy).
B-End Coupler Force	Note: units have hysteresis of ± 200 lb but are 1 percent when either in tension or compression	
A-End Coupler Angle	Coupler angle from longitudinal centerline of car	
B-End Coupler Angle	Coupler angle from longitudinal centerline of car	Used to determine external lateral force on the car.
Carbody Lateral Acceleration B-End	Carbody lateral and yaw acceleration on bolster centerline	Carbody accelerations will be used to determine the influence of carbody dynamics during curve entry and exit, and for the steady state portion of the curve.
Carbody Vertical Acceleration A-End Center	Carbody pitch, bounce, and roll acceleration over center plate	
Carbody Vertical Acceleration A-End	Carbody pitch, bounce, and roll acceleration on bolster centerline	
Carbody Vertical Acceleration B-End Center	Carbody pitch, bounce, and roll acceleration over center plate	
Carbody Vertical Acceleration B-End	Carbody pitch, bounce, and roll acceleration on bolster centerline	
Carbody to Truck Swivel	Truck/carbody swivel and center plate slip	Determine truck set in curves and estimate wheel/rail angle of attack.
Spring Group Vertical Displacement Rear	Spring group displacement and sideframe pitch	Determine relation of vehicle speed to equilibrium speed.
Spring Group Vertical Displacement Front	Spring group displacement and sideframe pitch	
Fore Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	Truck and carbody accelerations will be used to determine carbody dynamics.
Wheel/Rail Force Measurement System	Lateral, vertical, and longitudinal forces at wheel/rail interface	Determine vertical force on each wheel required for trackability studies during curve entry and exit; lateral force at wheel/rail interface required to evaluate the performance index; and lateral force at bearing adaptor required for determining forces transmitted from truck to axle.
Wheelset Angle of Attack	Angle of attack of the wheel	Required for estimating wheel/rail forces and for model validation.

The second type of analytical model to be used includes dynamic nonlinear time domain simulations. This type of program is useful for predicting maximum values of the performance index due to curve entry and track irregularity transients. Models of this type are generally quite expensive; minimum use will be made of them. Validation of this type of model will depend on its ability to predict the maximum values of the performance index during the entry and constant curvature portion of the curve.

Simple steady state truck models can be quite useful in analyzing the basic kinematic behavior of a truck in curves. For Type I trucks, the minimum required freedoms are yaw, lateral, and tram. The simplest Type II truck models have, in addition, the freedom for the axles to yaw relative to the truck. Basic forces which must be considered for flange free curving include creep and suspension forces. Since Type I trucks experience very little flange free curving, the inclusion of flange forces in models used for analyzing these trucks is necessary.

Additional forces which may be considered in steady state truck models are external lateral forces applied at the truck center due to draw bar forces and speeds other than equilibrium and gravitational stiffness forces. The importance of these forces depends on the situation being analyzed. The magnitude of gravitational stiffness forces depends on the wheel profile being analyzed and the amount of lateral wheel displacement relative to the rail. Full car models can be used when greater accuracy is desired. Cooperrider and Law (reference 9) have typically used models with the above freedoms for each truck with the addition of lateral axle freedoms for Type II trucks plus freedoms in lateral, yaw, and roll for the carbody. Nonlinearities of importance in steady state curving models include the creep coefficients and the wheel and rail profiles.

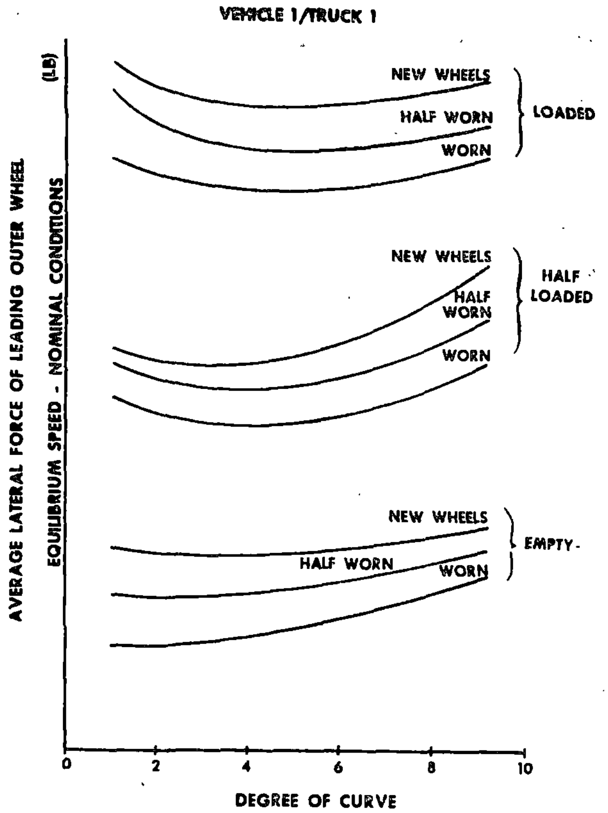

Figure 3-10. Average Lateral Force of Leading Outer Wheel-Nominal Conditions

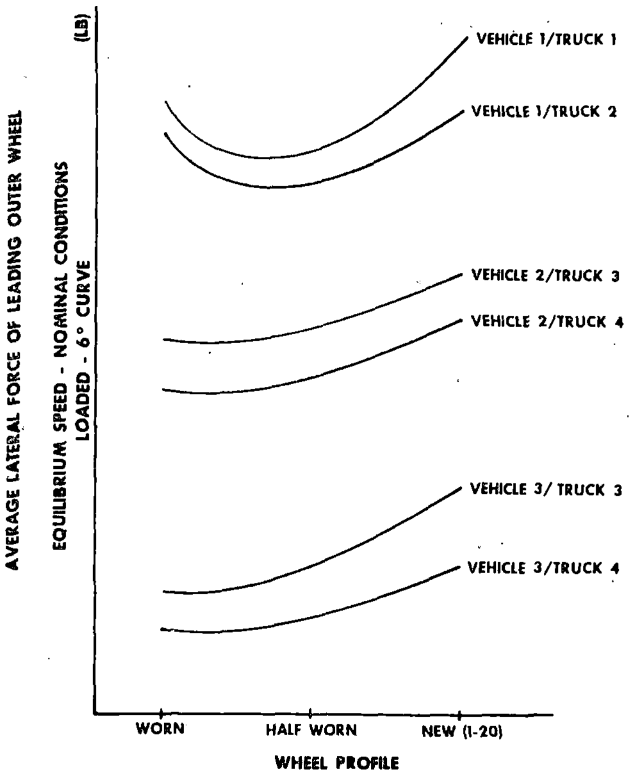

Figure 3-11. Effect of Wheel Profile on Lateral Force

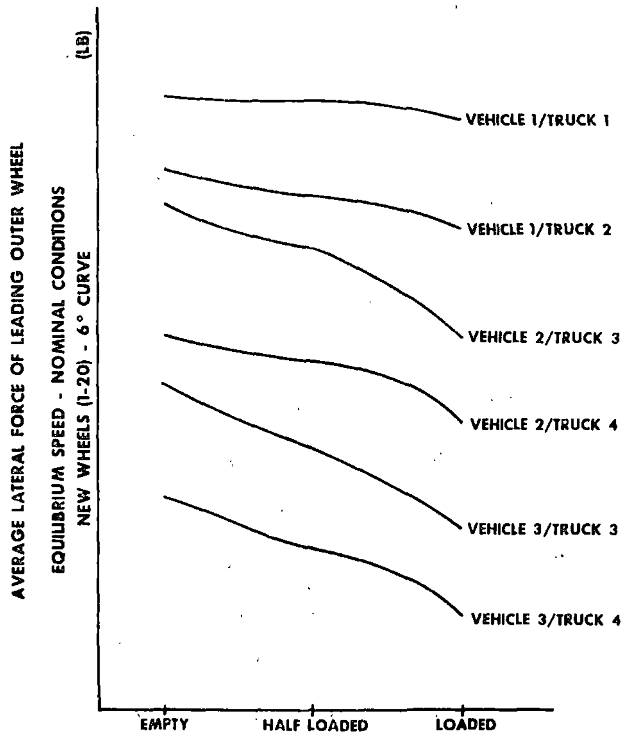

Figure 3-12. Effect of Load on Lateral Force

Table 3-14. Curve Negotiation Regime Tools

APPLICATION OF TOOLS	TYPES OF TOOLS: Qualitative Engineering Models, Simple Computations	Combinations of Simple Analytic Models	Linearized Models of Complete Vehicles	Detailed Models of Nonlinear Subsystems	Complete Nonlinear Models of Vehicles
Program or Model	Simple Steady State Models				9 & 17 dof Steady State Flexible Carbody
SAFETY Predict Dangerous Conditions					
DETERIORATION Predict Damage & Wear					X
PREDICTION OF IMPROVEMENTS - Type I - Type II Trucks	X				X
IMPROVEMENT OF MODEL & INSIGHT Explain discrepancies between theory and data	X				X
IDENTIFICATION OF NEEDS - Measurements - Tests					X
EXTRAPOLATION To Conditions Not Tested (Savings in Test Effort)					X
CORRELATION With Models & Tests by Others					

3.3.5 Special Considerations Regarding Type II Trucks

Curve negotiation, both steady state and curve entry and exit, will be affected by the unique characteristics of various Type II trucks. A number of Type II trucks feature self-steering mechanisms which are intended to reduce flanging in curves. These include the Dresser DR-1 and Barber-Scheffel. The Barber-Scheffel design recommends a high conicity wheelset profile for improved curve negotiation. Comparison of primary versus secondary suspension performance will also be examined. Provision has been made for instrumenting the longitudinal primary suspension deflections on the Dresser DR-1, Barber-Scheffel, and Devine-Scales trucks.

3.3.6 Summary

The objective of this analysis is to ascertain the relationships between variations in truck parameters and the curving performance index. The selected performance index is the lateral force on the leading outer wheel per 1000 lb axle load per degree of curve at equilibrium speed. This performance index can be related to the performance criteria using theoretical calculations. The performance criteria for the curve negotiation regime are low wheel and rail wear, prevention of derailment, low lateral wheel loads, and minimum fuel consumption.

Two types of analytical models will be used in predicting the curving performance index for variations in truck parameters. Steady state models will be used to compare the basic kinematic performance of different trucks and for calculations of wheel wear, fuel consumption, and rail wear in curves. Time domain models will be used for derailment potential analyses and for rail wear in the entry portion of the curve.

Additional curve testing of Type I trucks will be done to provide data for validating the math models and for characterizing Type I trucks. Basic to this testing is the measurement of wheel/rail forces. Models will be used to help plan and predict the results of the Type II truck field testing. The data from the Type II truck field testing will be used to further validate the models and to characterize Type II trucks.

3.4 RIDE QUALITY

The ride quality performance regime as defined includes the overall dynamic environment of a freight car (lading, carbody, and trucks), exclusive of the more extreme dynamics associated with the other performance regimes. Table 3-15 summarizes performance factors, truck parameters, and operational conditions which most affect ride quality. Figure 3-13 illustrates the flow of analysis in this regime.

To date the major economic impacts associated with ride quality in freight cars have been identified as component wear, and damage to lading. Component wear is significant on the basis of cost, since it occurs with all freight cars, including those where damage to lading is not a problem. It may be necessary to treat lading damage as a selection of special cases in order to obtain any correlation with observable economic loss. In a general sense, however, it will be desirable to consider carbody acceleration as a measure of ride quality performance, even though it may not relate directly to either component wear or damage to lading.

3.4.1 Performance Indices

The performance indices associated with this regime are transmissibility and vibration levels as measured by accelerations at specified points on the carbody and truck. Transmissibility is a well defined quantity. It may be characterized by a frequency-dependent function of amplitude ratios called a transfer function, or as a sequence of root mean square (RMS) ratios of output-to-input over selected frequency bands. The latter may be more meaningful for families of vehicles which reflect a broader range of parameter variation than any single vehicle. It may also be more meaningful from the standpoint of empirical relationships which relate to observable patterns of deterioration. RMS ratios can be derived easily from either measured or computed power spectral density (PSD) functions. In short, transmissibility of a truck directly specifies its ability to isolate the carbody from road bed vibrations, thereby producing a "good" ride quality.

3.4.2 Analysis Requirements

Included within the ride quality regime analysis are the requirements to:

- Characterize wear that occurs during the normal vibration environment of the truck
- Predict the vibration environment of the lading
- Compare the effectiveness of Type I and Type II trucks in isolating the vibration inputs due to track geometry

To date, the economic analysis has not identified any major truck components other than brake shoes and wheels whose high life cycle costs can be related to the normal vibration environment. This is especially true for the Type II trucks which have not had the amount of in-service experience which would identify high wear points. Nevertheless, one of the objectives of the ride quality task will be to identify relative motions and normal forces between components which can be used to estimate wear.

The ability to predict the lading environment for specific car/truck/lading combinations is necessary to evaluate the cause of lading damage. This is especially important for Type II trucks, whose main advantage may be that of a greatly improved ride quality.

Inasmuch as steady curving is considered for nominal track geometry only, the total dynamic environment of a freight car in curving must include both the forces which arise from steady curving on nominal track geometry, and those which are caused by track geometry irregularities. Figure 3-14 illustrates the nature of these combined effects. For the time being, it will be assumed that the forces (including wheel/rail forces) which occur within a freight car during curving may be represented by a superposition of the forces due to a steady curving and response to random irregularities on tangent track. This assumption is made with regard to overall force levels only, and does not imply that instantaneous tangent and curved track forces are additive in the time domain. Based upon the above assumption, the ride quality regime will be analyzed for tangent track only. Experimental data will be examined carefully to assess the validity of this assumption.

Table 3-15. Ride Quality Regime

Performance Criteria	Performance Indices	Truck Parameters	Operational Parameters	Economic Impact Areas
Smooth ride (minimize lading vibration environment)	Transmissibilities	Suspension system Primary Secondary	Track geometry	Lading damage
Minimization of normal truck component wear	Vibration levels as measured by accelerations at specified points on the carbody and truck	Wheel profile (as it affects the truck kinematic motion)	Speed	Component wear
			Lading	

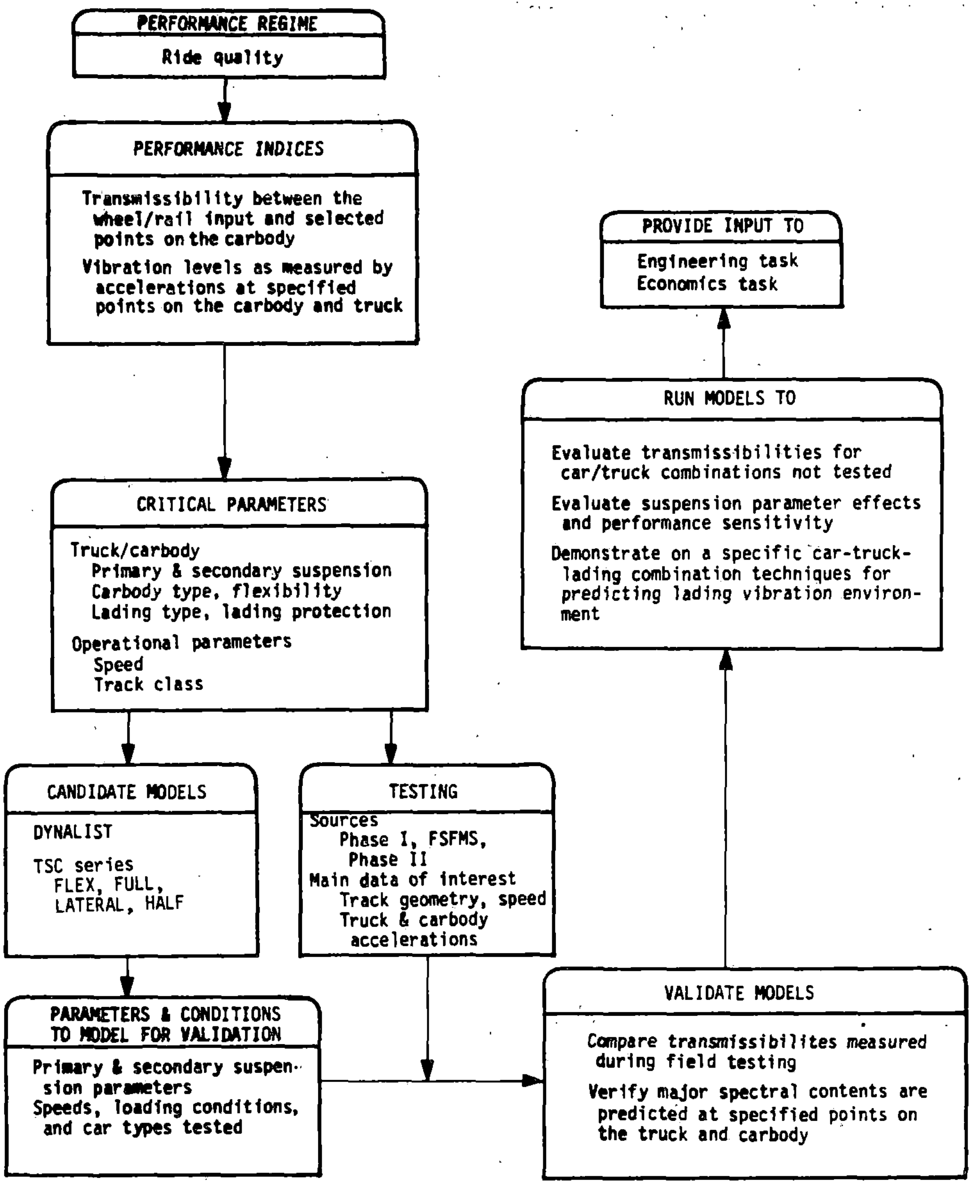

Figure 3-13. Flow Diagram for Ride Quality Analysis

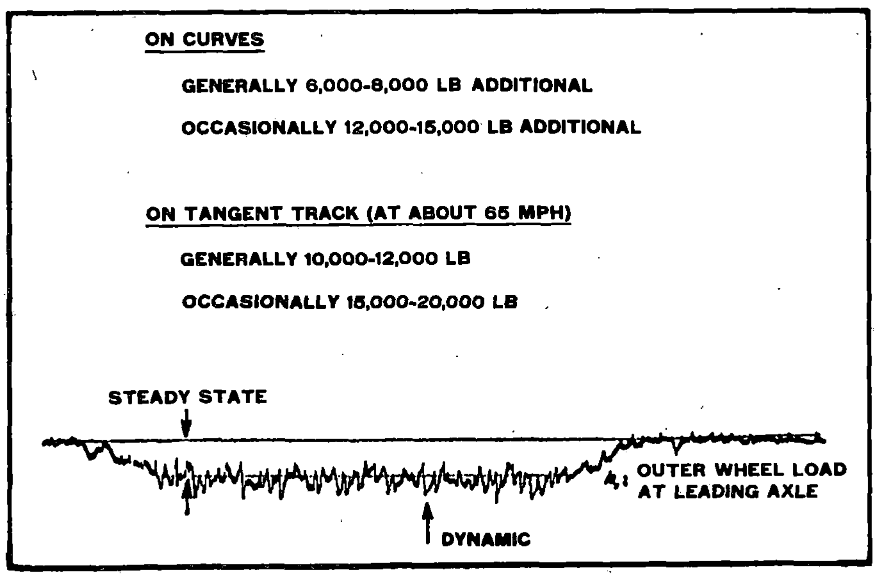

Figure 3-14. Dynamic Environment of a Freight Car in Curving

3.4.3 Test Data Utilization

Field test data applicable to the ride quality regime are available from Phase I and will be acquired in Phase II. All of the data reduction performed on the high speed runs are applicable to the ride quality regime. The additional data reduction required on the Phase I data is the calculation of transmissibility across the truck suspension elements.

For purposes of establishing the transmissibility performance index, acceleration data at selected locations will be acquired. These include:

Carbody

- Lateral measurements at front and rear ends
- Vertical measurement at mid-car
- Vertical measurements at front and rear ends

Truck Bolster

- Lateral measurement at bolster
- Vertical measurements at left and right sides

Axle Bearing Pocket

- Lateral measurement
- Vertical measurement

A detailed list of proposed Phase II instrumentation is presented in Table 3-16 along with further comments on the application of each group of measurements to the ride quality regime. Transmissibilities between rail irregularities and each of these response points can be evaluated. The transmissibility from axle bearing pockets to the truck bolster or carbody can also be evaluated. A primary difference between the two types of transmissibility is that the former includes the effect of track impedance, whereas the latter does not.

One of the primary requisites of a performance index is that it be simple enough to correlate with deterioration patterns and performance profiles while still being useful as a function of calculable and measurable response quantities. Thus, to achieve simplicity from the standpoint of correlating with deterioration, data from the four vertical axle accelerations, the two lateral axle accelerations, the two vertical truck bolster accelerations and carbody end accelerations (lateral and vertical) will be averaged separately.

The car, truck, and lading interact dynamically to form a complete system in determining accelerations on any part of the system. However, it is expected that certain parts of the system may tend to behave somewhat independently over limited frequency bands. For example, the vertical axle accelerations in the high frequency range will be influenced most by track impedance and the bearing assembly, and relatively little by the dynamic response of the truck bolster or carbody. Carbody motions in the low frequency range will be influenced most by the spring nests and snubbers (as well as the inertial properties of the carbody itself) and relatively little by track impedance and bearing assemblies. Frequency separation effects should therefore be considered in data interpretation as well as in dynamic modeling, response analysis, and the formulation of performance indices. This sort of rationale provides the basis for using band limited RMS values instead of overall RMS values for at least some purposes.

3.4.4 Model Utilization

Both vertical and lateral models will be used for ride quality analysis. Model requirements and application areas for the ride quality regime are presented in Table 3-17.

Proposed vertical models are shown in Figure 3-15. A complete freight car is labeled Model #1. It has a total of 7 degrees of freedom (dof) as indicted in the figure. However, it may be possible to replace this 7 dof model by two smaller ones: a 3 dof low frequency model (Model #2) and a 2 dof high frequency model (Model #3).

Table 3-16. Instrumentation for Ride Quality

MEASUREMENT TYPE	REQUIREMENTS	APPLICATION FOR RIDE QUALITY ANALYSIS
ALD	Locate track test areas; mounted under Mobile Laboratory car	Used to correlate vehicle response to track geometry measurements.
Speed	Speed from distance counter	Essential channel for correlation with vehicle responses.
A-End Coupler Force	Note: units have hysteresis of ±200 lb but are 1 percent when either in tension or compression	Coupler force components, both lateral and longitudinal will be required to determine the effect of the consist on ride quality measurements.
B-End Coupler Force	Note: units have hysteresis of ± 200 lb but are 1 percent when either in tension or compression	
A-End Coupler Angle	Coupler angle from longitudinal centerline of car	
B-End Coupler Angle	Coupler angle from longitudinal centerline of car	
Carbody Lateral Acceleration A-End	Carbody lateral and yaw acceleration on bolster centerline	Carbody acceleration measurements will be used as the main means of ride quality characterization, since they are necessary to establish the transmissibility performance index.
Carbody Lateral Acceleration B-End	Carbody lateral and yaw acceleration on bolster centerline	
Carbody Vertical Acceleration A-End Center	Carbody pitch, bounce, and roll acceleration over center plate	

Table 3-16. Instrumentation for Ride Quality (cont.)

Carbody Vertical Acceleration B-End Center	Carbody pitch, bounce, and roll acceleration over center plate	
Carbody Vertical Acceleration B-End	Carbody pitch, bounce, and roll acceleration on bolster centerline	
Truck Bolster Vertical Acceleration	Truck bolster triaxial acceleration	Bolster acceleration measurements will be important for calculating transmissibilities within the carbody/truck system.
Truck Bolster Lateral Acceleration	Truck bolster triaxial acceleration	
Carbody to Truck Swivel	Truck/carbody swivel and center plate slip	Truck/carbody swivel will be used to identify occurrence of and effects from kinematic truck motion on ride quality.
Spring Group Vertical Displacement Rear	Spring group displacement and sideframe pitch	
Fore Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	The lateral and vertical axle bearing acceleration will be used to separately calculate truck excitation modes: pitch, roll, vertical, and lateral.
Fore Axle Bearing Pocket Lateral Acceleration	Axle lateral acceleration forward	
Rear Axle Bearing Pocket Vertical Acceleration	Axle vertical acceleration and pitch	
Rear Axle Bearing Pocket Lateral Acceleration	Axle lateral acceleration rear	
Vertical Acceleration Carbody Center	Triaxial acceleration at carbody center	One of the primary carbody acceleration measurements for calculating transmissibilities and quantifying the vibration environment.
Lateral Acceleration Carbody Center	Triaxial acceleraton at carbody center	
Longitudinal Acceleration Carbody Center	Triaxial acceleration at carbody center	
Vertical Acceleration Carbody B-End	Roll center for carbody roll	Needed for calculation of the instantaneous roll center.

The low frequency model would represent the carbody modes, including the first vertical bending mode for very flexible cars such as the flatcar. No track compliance is included in this model, and inputs would be derived from motion of the wheels (e.g., measured at the bearing pockets).

The high frequency model would represent the wheel modes, with the track modeled as a compliant structure. In this model, the wheelsets, side frame, and a portion of the track are assumed to move together, except for a known rail irregularity which is input as a differential displacement between the two. Although not shown in the figure, it will be assumed that there is a viscous damper everywhere a spring is shown.

A schematic representation of the proposed lateral model is shown in Figure 3-16. This model is essentially the same as the 9 dof model developed by Cooperrider and Law (reference 10) except that it includes the fundamental flexible carbody torsion or twisting mode which is expected to be particularly important in characterizing the flatcar. Track inputs to this model will consist of alignment and crosslevel irregularities, whereas surface irregularities are input to the vertical models.

It is not clear in the case of lateral motion (as it is for vertical motion) that simpler models (e.g., low frequency and high frequency) may be used in lieu of the full car model. A high frequency model consisting of one 3 dof truck may be valid whereas the full car model will probably be required to model low frequency behavior. It may be that two simpler models could be used for crosslevel excitation, but not alignment. In any case, the subsystems modeling capability of DYNALIST II (reference 11) will enable the full car model to be built up of two truck components (3 dof each) and one carbody component (3 or 4 dof) depending on whether carbody torsion is included. The individual components will be analyzed separately to evaluate their dynamic characteristics.

It is anticipated that the performance indices correlating with lading damage and component deterioration will involve accelerations on the carbody and axle bearing pockets, respectively. Experimentally measured truck bolster accelerations will be compared with accelerations on the axle bearing pockets to validate the assumption that the truck bolster moves with the car bolster, and that appreciable center plate rocking does not occur in the frequency range of interest.

A model representing categories of in-service vehicles may be much easier to validate than a model representing one particular test vehicle. The broader range of parameters required for a category of vehicles than for a specific vehicle should lead to less stringent validation criteria, thereby allowing the test results to be more easily extrapolated to the national population of in-service vehicles.

Table 3-17. Ride Quality Regime Tools

APPLICATION OF TOOLS	TYPES OF TOOLS: Qualitative Engineering Models, Simple Computations	Combinations of Simple Analytic Models	Linearized Models of Complete Vehicles	Detailed Models of Nonlinear Subsystems	Complete Nonlinear Models of Vehicles
Program or Model	Simple Spring Mass Model		DYNALIST HALF, FULL, LATERAL, FLEX		FRATE Flexible Carbody
SAFETY Predict Dangerous Conditions					
DETERIORATION Predict Damage & Wear					X
PREDICTION OF IMPROVEMENTS - Type I - Type II Trucks	X		X		
IMPROVEMENT OF MODEL & INSIGHT Explain discrepancies between theory and data	X		X		X
IDENTIFICATION OF NEEDS - Measurements - Tests	X		X		X
EXTRAPOLATION To Conditions Not Tested (Savings in Test Effort)	X		X		X
CORRELATION With Models & Tests by Others	X		X		

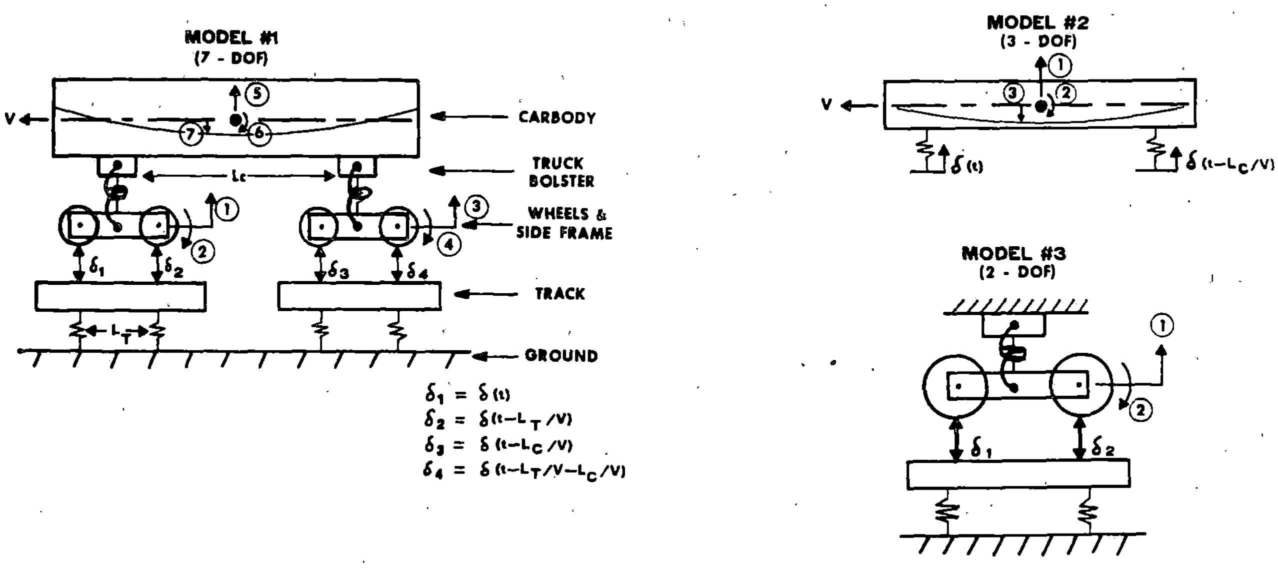

Figure 3-15. Vertical Models

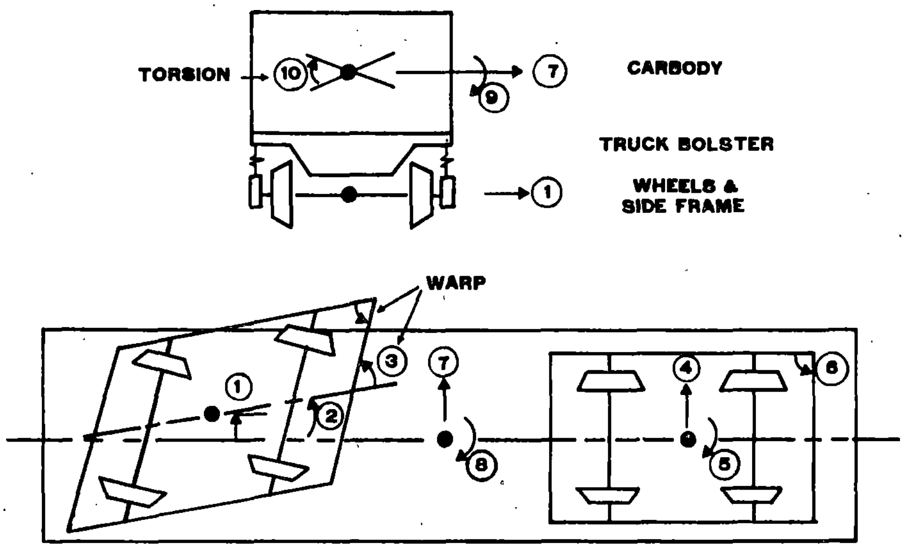

Figure 3-16. Lateral Model (10 Degrees of Freedom)

3.4.5 Special Considerations Regarding Type II Trucks

The ride quality performance index of transmissibility will largely be determined by the suspension characteristics. The analysis will therefore focus on the differences exhibited between primary and secondary suspension trucks. Vertical primary suspension displacements will be measured on the Devine-Scales and NRUC Maxiride trucks. Secondary suspension vertical displacements will be measured on the Dresser DR-1, Barber Scheffel, National Swing Motion, and ACF Fabricated trucks.

3.4.6 Summary

The ride quality regime will treat the normal dynamics of the freight car. The performance indices for ride quality are the truck transmissibility and the levels of vibration at specified points on the car. Transmissibility is a measure of the ability of the truck to isolate the track vibrations from the carbody and lading. The primary objectives of the ride quality analysis will be to develop methodologies for predicting the lading vibration environment. Simple linear models will be used to perform these studies.

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APPENDIX A - DATA ANALYSIS REQUIREMENTS

This appendix describes the data reduction requirements for both track geometry and vehicle field testing. The track geometry analysis will be performed to determine the excitation levels existing during the field test and to generate input for the car/truck simulation programs, which will be used both in the validation efforts and in the extrapolation of field test results. The data reduction and analysis capabilities will be provided by an in-house Wyle data reduction system.

TRACK GEOMETRY

Track geometry data have been acquired from the TDOP Phase I field testing and from the Union Pacific test track during an on-going Phase II field test program. Several problems have been encountered during the data reduction and interpretation of the Phase I track geometry measurements, and steps are being taken under Phase II to avoid similar problems. The two major problems have been lack of alignment data and false location detector signals.

The magnetic automatic location detectors (ALD) used during Phase I detected many false targets (spikes, crossings, cans) along with the true target. These false indications make it difficult to align the separately recorded test and track geometry from Phase I. This problem complicates data reduction where time or phase correlations are required between the track geometry and the vehicle response data. A possible solution to this problem has been identified.

The runs which have been analyzed to date indicate that the false ALD indications show some repeatability between the track geometry and the test data run, i.e., the stray pieces of metal, cans, etc. were present on both the field test and track geometry measurements. This repeatability allows a correlation routine to be used to align the track geometry and test vehicle response measurements.

The ALD signals, both true and false ones, can be aligned via two distance channels. One distance channel was measured directly during the track geometry and the other one can be derived from the speed channel of the test run. Test run speed is integrated to produce a distance channel against which the test run ALD indications are plotted.

The directly measured track geometry and calculated ALD versus distance channels are then correlated to determine any errors that have occurred.

The calculations described above showed very close correlation between the track geometry and calculated distance channels for the low speeds and errors on the order of 5 percent for the higher speeds. The error distribution indicates that the probable error source is in the speed measurement. Steps have been taken to avoid this problem during the Phase II testing.

Phase II track geometry measurements on the Union Pacific railroad will use a magnetic field ALD sensor on the test vehicle which will sense the field from permanent magnets placed in the ties. Stray metal will not produce false indications.

The test track sections for Phase II are listed in Table A-1. The track geometry will be sampled every 6 inches over these sections. Track geometry will be measured three times during Phase II.

The track geometry measurements consist of the following:

- Left and right vertical profile
- Left and right alignment
- Gage
- Superelevation
- Center line

The above spatially sampled data will be processed to produce cumulative distribution functions, probability densities, and power spectral densities.

For each test speed within a run, the spatially sampled track geometry data will have to be converted to the equivalent time domain sample of the track geometry. This processed "time domain" track geometry will form the input function for that test run. A typical data reduction sequence for Phase II consists of building an Advanced Data Analysis and Reduction System (ADARS)

Table A-1. Test Zone Locations

Zone	Site Designation	Description
1	Location	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
2	Location	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
3	Location	Las Vegas
	Mileposts	Yard Limits (0.22 miles)
	Track Type	12 and 16 Degree Curves
	Rail Type	Jointed
	Speed Limit	10 mph
4	Location	Blue Diamond Spur
	Mileposts	5 to 8 (3 miles)
	Track Type	Class 2 - Curved & Tangent
	Rail Type	131-lb Jointed
	Speed Limit	20 mph
5	Location	Balch to Crucero
	Mileposts	210.5 to 204.5 (6 miles)
	Track Type	Class 4 - Tangent
	Rail Type	133-lb CWR
	Speed Limit	79 mph

test run data file on disc and then processing the spatially sampled track geometry data to create a "time domain" track excitation file for that run. Figure A-1 illustrates the technique which is used for converting the spatially sampled track geometry to the equivalent time domain sampled track geometry. The technique consists of interpolating points between the spatially sampled track geometry data to create an equivalent time domain sampled track geometry at the 200 samples/second sample rate used during the field test data acquisition.

Once accurate time domain track geometry records have been appended to the test run data base, data reduction and analysis in either the time or frequency domain can be performed.

Track geometry will also be combined to enable the identification of specific truck and carbody excitations, such as vertical, pitch, and roll movements of the carbody. An example procedure for combining the vertical profile measurements to produce the pure vertical and pitch excitation is discussed next. Similar calculations will be performed to determine lateral excitation for a particular track geometry.

Track geometry measurements will be combined to calculate the carbody pitch and pure vertical excitation according to the scheme outlined below. The pitch and vertical excitation is a function of not only the track geometry but of the truck wheelbase and separation. The change in height of a truck center, Z_{TC}, due to track cross level variations is given by:

$$Z_{TC} = \frac{1}{2}\left(H_i + H_{(i+1)}\right)$$

Where

Z_{TC} = vertical displacement of the truck center

H_i = wheelset center height due to cross-level

$= \frac{1}{2}\left(Z_{iL} + Z_{iR}\right)$

$Z_{iL,R}$ = displacement of right, left profile for axle i

i = 1, 2, 3, 4

Pure vertical and pitch carbody excitation at a given position on the track is given by:

$$\text{Vertical excitation} = \frac{1}{2}\left(Z_{TC_A} + Z_{TC_B}\right)$$

$$\text{Pitch excitation} = \frac{1}{2}\left(Z_{TC_A} - Z_{TC_B}\right)$$

Where

Z_{TC_A} = vertical displacement of A end truck center

Z_{TC_B} = vertical displacement of B end truck center

For the above pitch excitation, the A end is assumed to be the leading end of the car.

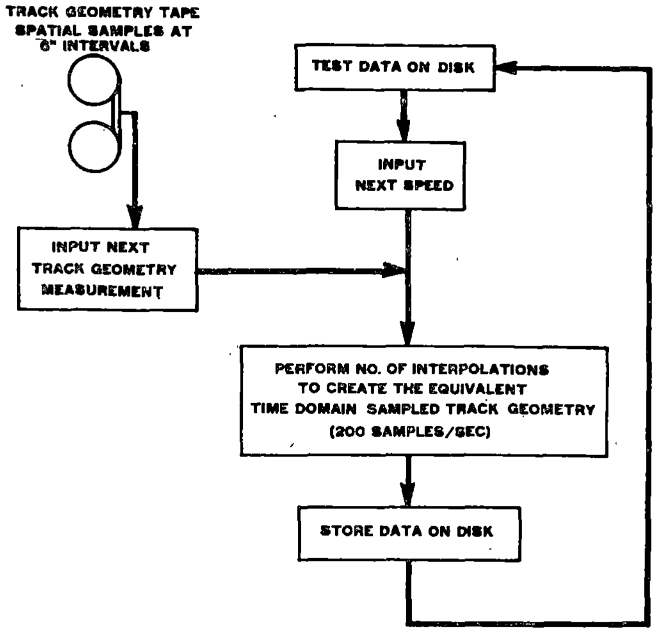

Figure A-1. Converting Spatial Track Geometry to Equivalent Time Domain Track Geometry

The frequency content of the pitch and vertical excitation will then be calculated in the same manner as track geometry data. Power spectral densities (PSD) in either the spatial or temporal domains (given the velocity) can be calculated. Spatial domain PSDs will be investigated for any of the longer wavelengths which would result in excitation at the carbody vertical and pitch mode frequencies over the normal operating speed range.

VEHICLE FIELD TESTING

Digital data tapes from both the Phase I and II test programs will be processed by first demultiplexing the data to produce a data tape compatible with ADARS.

ADARS provides the capability to perform data reduction in both the frequency and time domain and to provide plots and printouts of individual or linear combinations of channels. The general data reduction and analysis capability provided by ADARS is illustrated in the block diagram shown in Figure A-2.

Spectrum analysis (SPEC) and display capability are included within ADARS. The function of SPEC is to provide 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 provides the capability to calculate the following:

- PSD
- Cross power spectral density (CSD)
- Transfer function gain and phase
- Hanning window on time arrays
- Average of PSD and CSD values
- Mean and degree of freedom (or confidence level) with PSD plots

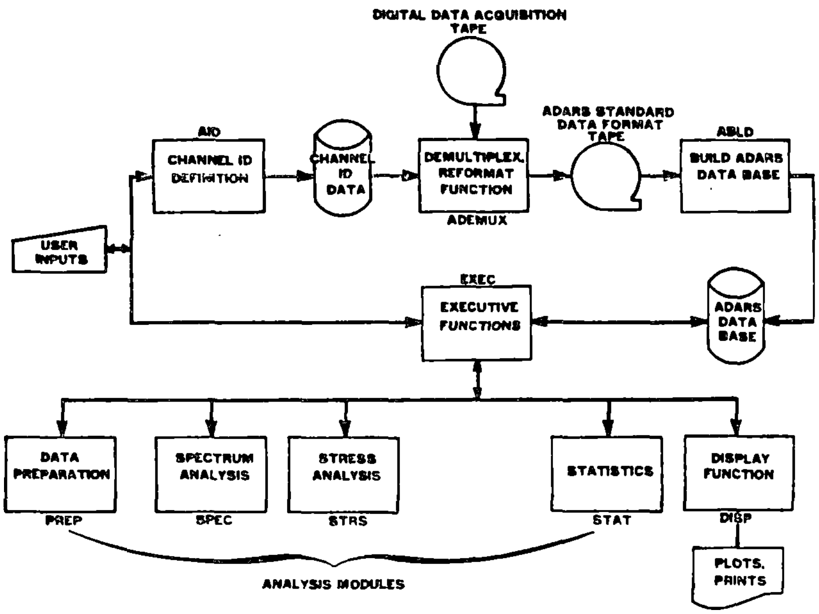

Figure A-2. ADARS Overview Flow Diagram

The statistics capability (STAT) performs the following statistical analysis functions for a given input file:

- Computes and writes to line printer the averages, standard deviation, skewness, and kurtosis
- Computes the root mean square values

The display function (DISP) provides both plots and printouts for the ADARS user. DISP can be initiated by the user directly, or any analysis module can automatically initiate DISP upon completion of its execution. The following types of plots are produced:

Time History Plot. Linear X, linear Y axis for raw or analyzed data, single channel per grid, single grid per page or multiple channels per grid with multiple grids per page

Strip Chart Plot. Raw or analyzed data with common linear X axis time frame and grid, eight linear Y axis grids per page with one channel plotted per grid for each of the eight grids

Any array of analyzed data that is suitable for plotting can be printed. Both the X and Y axis arrays will be listed in engineering units with the X axis data on the left portion of the listing.

Maximum/minimum analysis will give the largest and the smallest data value that occurs within the interval. The ADARS data analysis capability will then be used to produce the reduced data required by the model validation task.

APPENDIX B - INSTRUMENTATION CHANNELS

Appendix B contains a list of the 41 instrumentation channels proposed for Type I truck testing. Following the table are several illustrations showing the location of the instrumentation.

Further details on the TDOP Phase II Type I and Type II truck testing can be found in Wyle Document No. C-901-0004-A, "Type I Truck Test Plan," dated April, 1979, and Wyle Document No. C-901-0007-A, "Type II Truck Test Plan," dated October, 1979.

Table B-1. Instrumentation Channels for Type I and Type II Truck Testing

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY DESIRED	PURPOSE	TYPE II TRUCKS ONLY
S1	Speed	1 Hz	0-120 mph	±.5 mph	Speed from distance counter to time base unit	
S2	ALD	256 samples/in.	6-12"	±6"	Location of track test areas. Mounted under Car 210	
S3	Brake Cylinder Pressure	10 Hz	0-100 psid	1%	Brake line pressures (to insure inadvertent braking from being mixed with data)	
S4	Throttle Setting		1-8		Correlation with draw bar forces in fuel consumption study	
C1	B-End Coupler Force	20 Hz	25000 lb	5%*	Longitudinal draw bar force and correlation to fuel consumption	
C2	B-End Coupler Angle	20 Hz	+10°	±.1°	Coupler angle from longitudinal centerline of car	
C3	A-End Coupler Force	20 Hz	25000 lb	5%*	Longitudinal draw bar force and correlation to fuel consumption	
C4	A-End Coupler Angle	20 Hz	±10°	±.1°	Measure coupler angle from longitudinal centerline of car	
D1	Rt. Spring Group Vert. Disp. Front	20 Hz	+2" -4"	1%	Right spring group disp. side frame pitch	
D2	Rt. Spring Group Vert. Disp. Rear	20 Hz	+2" -4"	1%	Right spring group disp. side frame pitch	
D3	Left Spring Group Vert. Disp. Front	20 Hz	+2" -4"	1%	Left spring group disp. side frame pitch	
D4	Left Spring Group Vert. Disp. Rear	20 Hz	+2" -4"	1%	Left spring group disp. side frame pitch	
D5	Truck Bolster to Side Frame Lat. Disp. Rt. Fr. side	50 Hz	±0.8"	1%	Side frame relative disp. truck tram	
D6	Truck Bolster to Side Frame Lat. Disp. Rt. Rear Side	50 Hz	±0.8"	1%	Side frame relative disp. truck tram	
D7	Truck Bolster to Side Frame Right Bottom	50 Hz	±0.8"	1%	Used with D5 & D6 to measure side frame roll	
D11	Carbody to Truck Bolster Rel. Disp. Rt. Side	20 Hz	±1.5"	1%	Carbody/truck roll angle	
D12	Carbody to Truck Bolster Rel. Disp. Left Side	20 Hz	±1.5"	1%	Carbody/truck roll angle	
D13	Carbody to Truck Lat. Disp. Forward	100 Hz	±10°	0.1%	Truck/carbody swivel, center plate lateral slip	
D14	Carbody to Truck Lat. Disp. Rear	100 Hz	±10°	0.1%	Truck/carbody swivel, center plate lateral slip	
D15	Primary Spring Disp. Rt. Fr. Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion	√

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

Table B-1. Instrumentation Channels for Type I and Type II Truck Testing (Cont.)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY DESIRED	PURPOSE	TYPE II TRUCKS ONLY
D16	Primary Spring Disp. Rt. Rear Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion	√
D17	Primary Spring Disp. Lf. Fr. Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion	√
D18	Primary Spring Disp. Lf. Rear Axle	20 Hz	+2" -4"	1%	Side frame/axle relative motion	√
D19	Axle to Side Frame Long. Disp. Rt. Fr.	20 Hz	±0.8"	1%	Side frame/axle relative motion	√
D20	Axle to Side Frame Long. Disp. Rt. Rear	20 Hz	±0.8"	1%	Side frame/axle relative motion	√
D21	Axle to Side Frame Long. Disp. Lf. Front	20 Hz	±0.8"	1%	Side frame/axle relative motion	√
D22	Axle to Side Frame Long. Disp. Lf. Rear	20 Hz	±0.8"	1%	Side frame/axle relative motion	√
D23	Steering Arm Disp. Rt. Side	20 Hz	±2"	1%		√
D24	Steering Arm Disp. Lf. Side	20 Hz	±2"	1%		√
D25	Spread of Side Frame Legs Rt. Side	20 Hz	±2"	1%		√
D26	Spread of Side Frame Legs Lf. Side	20 Hz	±2"	1%		√
R1	Bolster/Side Frame Twist Angle Front	20 Hz	±2"	1%		√
R2	Bolster/Side Frame Twist Angle Rear	20 Hz	±2"	1%		√
A1	Carbody Vertical Accel. B-End Center	20 Hz	±5 G	1%	Carbody pitch, bounce, & roll accel. over center plate	
A2	Carbody Vertical Accel. A-End Right	20 Hz	±5 G	1%	Carbody pitch, bounce, & roll accel. over center plate	
A3	Carbody Vertical Accel. B-End Right	20 Hz	±5 G	1%	Carbody pitch, bounce, & roll accel. on bolster centerline	
A4	Carbody Vertical Accel. A-End Right	20 Hz	±5 G	1%	Carbody pitch, bounce, & roll accel. on bolster centerline	
A5	Carbody Lateral Accel. B-End	20 Hz	±5 G	1%	Carbody lateral & yaw accel. on bolster centerline	
A6	Carbody Lateral Accel. A-End	20 Hz	±5 G	1%	Carbody lateral & yaw accel. on bolster centerline	

Table B-1. Instrumentation Channels for Type I and Type II Truck Testing (Cont.)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY DESIRED	PURPOSE	TYPE II TRUCKS ONLY
A7	Carbody Vertical Accel. B-End Left	20 Hz	±5 G	1%	Carbody center of rotation on bolster centerline	
A8	Carbody Long. Accel. B-End Center	20 Hz	±5 G	1%	Carbody longitudinal accel. to correlate with braking	
A9	Fore Axle Brg. Pocket Vert. Accel. B-End Right	20 Hz	±5 G	1%	Axle vertical & pitch accel. to correlate with profile	
A10	Fore Axle Brg. Pocket Lat. Accel. B-End Right	20 Hz	±5 G	1%	Axle lateral accel. to define hunting and to correlate with alignment	
A11	Rear Axle Brg. Pocket Lat. Accel. B-End Right	20 Hz	±5 G	1%	Axle lateral accel. to define hunting and to correlate with alignment	
A12	Fore Axle Brg. Pocket Vert. Accel. B-End Left	20 Hz	±5 G	1%	Axle vertical & pitch accel. and to correlate with profile	
A13	Fore Axle Brg. Pocket Lat. Accel. A-End Right	20 Hz	±5 G	1%		
A14	Rear Axle Brg. Pocket Lat. Accel. A-End Right	20 Hz	±5 G	1%		
F1	BL-1 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation	
F2	BR-1 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation	
F3	BL-2 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation	
F4	BR-2 Bearing Adapter Vert. Force	50 Hz	20,000 lb	1%	Net vertical force. Used in L/V calculation	
F1-1	BL-1 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
F1-2	BR-1 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
F2-1	BR-1 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
F2-2	VR-1 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
F3-1	BR-2 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√

Table B-1. Instrumentation Channels for Type I and Type II Truck Testing (Cont.)

MEAS. ID	MEAS. TYPE	FREQUENCY RESPONSE	MEAS. RANGE	ACCURACY DESIRED	PURPOSE	TYPE II TRUCKS ONLY
F3-2	BL-2 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
F4-1	BR-2 Bearing Adapter Outer Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
F4-2	BR-2 Bearing Adapter Inner Strain Gage	50 Hz	5 mV	1%	Line of action of vertical load	√
G1	BL-1 Rotary Pulse Generator #1	40 kHz	0-5 V	1/2000 rev.	Rotation of axle #1, used to specify strain gage position	
G2	BL-2 Rotary Pulse Generator #2	40 kHz	0-5 V	1/2000 rev.	Rotation of axle #2, used to specify strain gage position	
B1	BL-1 Axle Bending Vertical	50 Hz	4,000 in-lb	1%	Right side vertical bending moment of lead axle. Used in L/V calculation	
B2	BL-1 Axle Bending Longitudinal	50 Hz	4,000 in-lb	1%	Right side longitudinal bending moment of lead axle. Used in L/V calculation	
B3	BR-1 Axle Bending Vertical	50 Hz	4,000 in-lb	1%	Left side vertical bending moment of lead axle. Used in L/V calculation	
B4	BR-1 Axle Bending Longitudinal	50 Hz	4,000 in-lb	1%	Left side longitudinal bending moment of lead axle. Used in L/V calculation	
B5	BL-2 Axle Bending Vertical	50 Hz	4,000 in-lb	1%	Right side vertical bending moment of rear axle. Used in L/V calculation	
B6	BL-2 Axle Bending Longitudinal	50 Hz	4,000 in-lb	1%	Right side longitudinal bending moment of rear axle. Used in L/V calculation	
B7	BR-2 Axle Bending Vertical	50 Hz	4,000 in-lb	1%	Left side vertical bending moment of rear axle. Used in L/V calculation	
B8	BR-2 Axle Bending Longitudinal	50 Hz	4,000 in-lb	1%	Left side longitudinal bending moment of rear axle. Used in L/V calculation	
P1	BL-1A Wheel/ Side Frame Position Transducer "A"	50 Hz	2"	1%	Wheel/rail angle of attack	
P2	BL-1B Rail/ Side Frame Position Transducer "B"	50 Hz	2"	1%	Wheel/rail angle of attack	
P3	BL-1C Wheel/ Side Frame Position Transducer "C"	50 Hz	2"	1%	Wheel/rail angle of attack	
P4	BL-1D Rail/ Side Frame Position Transducer "D"	50 Hz	2"	1%	Wheel/rail angle of attack	

Table B-1. Instrumentation Channels for Type I and Type II Truck Testing (Cont.)

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

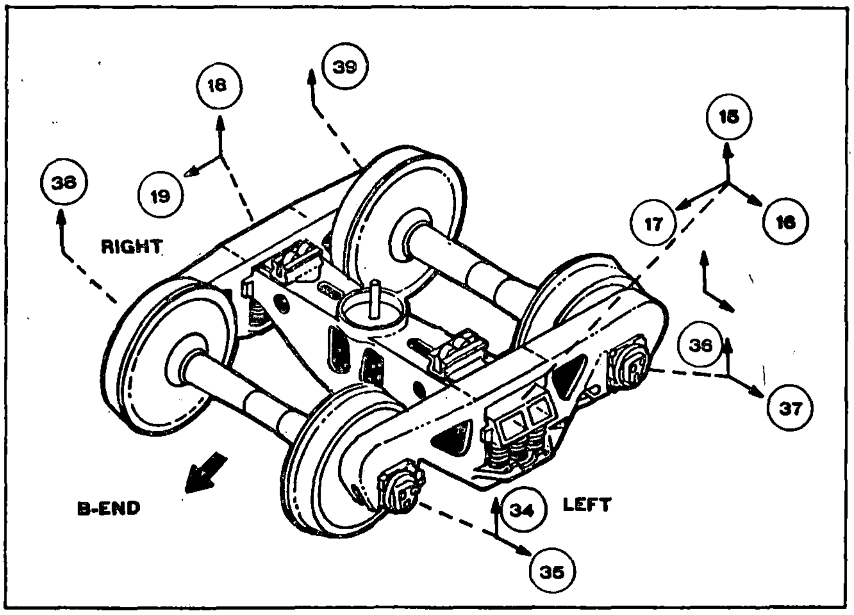

Figure B-1. Truck Instrumentation Locations (B-End)

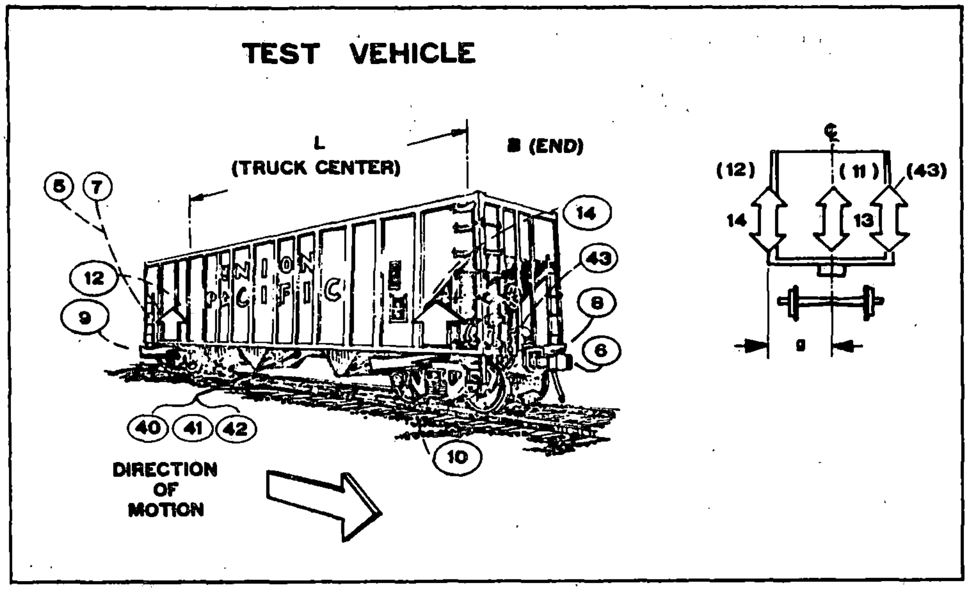

Figure B-2. Carbody Instrumentation

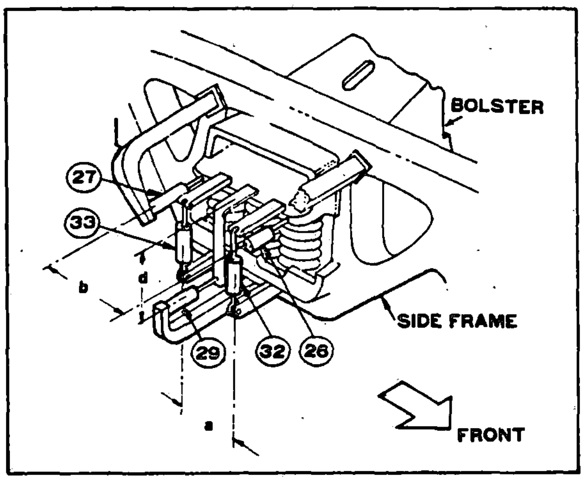

Figure B-3. Bolster Instrumentation (Right Side)

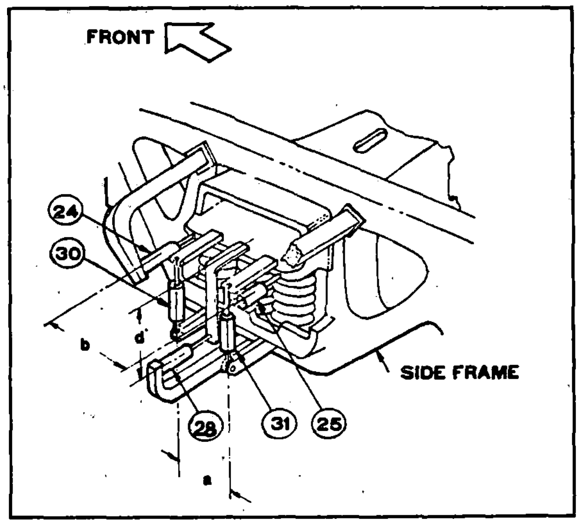

Figure B-4. Bolster Instrumentation (Left Side)

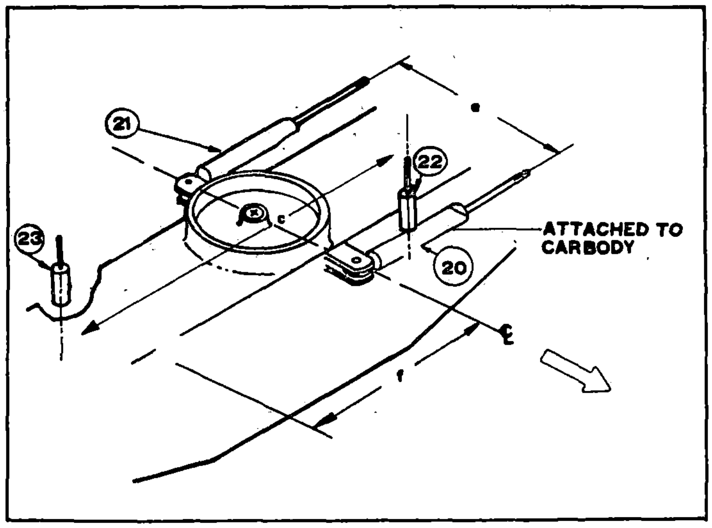

Figure B-5. Center Plate Instrumentation

APPENDIX C - GLOSSARY

ADHESION

The absence of gross slippage of the wheel on the rail in the presence of tangential forces at the interface.

ANALYTICAL TOOL

Refers collectively to a series of techniques used to study or predict the dynamics of a physical system, such as a freight car. Analytical tools are made up of mathematical models and computer programs. Models consist of a set of equations which can be used to mathematically study and predict the response of physical systems. The computer programs implement the sets of equations or models on a digital computer.

ANGLE OF ATTACK

Horizontal angle between the vertical plane of the wheel and the tangent to the rail at the point of contact.

ANOVA

Analysis of variance, a body of tests of hypotheses, methods of estimation, etc. using statistics which are linear combinations of sums of squares of linear functions of the observed values.

AXLE

The steel shaft on which the car wheels are mounted. The axle not only holds the wheels to gage, but also transmits the load from the journal boxes to the wheels.

"B" END OF CAR

The end on which the hand brake is located.

BALANCE SPEED

The speed with which a vehicle traverses a superelevated curve of constant radius when the centrifugal force exactly balances the horizontal component of the weight due to inclination.

BOLSTER (BODY)

The transverse members on the underframe of a car which transmit the load carried by the longitudinal sills through the center plate to the trucks.

BOLSTER SPRING

The main suspension spring of a car directly supporting the truck bolster and the weight of the carbody.

BOLSTER (TRUCK)

A beam placed across the frame of a truck to receive, through the center plate, the weight of the carbody and transfer it to the truck frame and wheels through the spring sets in the side frames.

BOUNCE

Vertical oscillation of the center of gravity of the sprung mass (carbody, truck bolster, etc.).

BRAKE SHOE

Friction unit contacting the wheels.

CENTER PLATE

One of a pair of plates which fit one into the other and which support the carbody on the trucks, allowing them to turn freely under the car.

CLIMB

The process of the wheel flange contacting and climbing the rail frequently onto the railhead.

COULOMB DAMPING

A damping mechanism that depends on constant amplitude friction forces which always resist the direction of motion.

CURVE NEGOTIATION

The ability of a truck to enter, provide guidance through, and exit a curve.

CURVE

In the United States it is customary to express track curvature in degrees noted by the deflection from the tangent measured at stations 100 feet apart. The number of degrees of central angle subtended by a chord of 100 feet is the "degree curve." One degree of curvature is equal to a radius of 5,750 feet.

CREEP

The capability of two bodies to displace in their plane of contact without slipping. It is made possible by shear deformation of both bodies in the region of the interface, which can support tractive forces.

CRITICAL DAMPING

Amount of damping at which no oscillatory vibration occurs after a spring-mass has been released from a non-equilibrium position.

CRITICAL HUNTING SPEED

Minimum speed at which violent truck shimmy occurs.

CRITICAL SPEED

Excitation forces applied to the vehicle are related to forward speed. A critical speed is one at which a car-truck dynamic resonance occurs.

DAMPING COEFFICIENT

Number describing the energy-absorbing property of a physical system.

DATA BASE

A collection of interrelated data stored together to serve one or more applications; data are stored so that they are independent of programs using the data, a common and controlled approach to maintenance and retrieval of data.

DEGREE OF FREEDOM

A single parameter or coordinate of choice with physical objects, it implies the ability to displace along or about one coordinate axis.

EIGENVALUE ANALYSIS

A method used to predict the natural frequencies of a physical system, such as a truck, from the natural loads of the system. Normally Eigenvalues are calculated from a linear set of equations used to model the system.

ELEVATION

The higher position of one of the two rails.

FLAT SPOT

Loss of roundness of the tread of a railroad wheel, caused by wheel sliding.

FLEXIBLE TRUCK

A truck in which the axles are allowed to displace relative to the frame in the lateral and yaw degrees of freedom, against elastic constraints, usually in the form of elastomers.

FORCE-DISPLACEMENT CHARACTERISTICS

Graph describing the force necessary to reach/maintain a specific amount of displacement.

FORCED FREQUENCY

Frequency imposed on the system by superimposed forces (rail joints, etc).

FORCED RESPONSE

A term used to describe the response of a system due to some external forcing function.

FRICTION PLATE

A removable plate to prevent wear on the main body of a component.

FRICTION SNUBBER

A device built into the secondary suspension of a truck to absorb energy. The standard designs rely on sliding friction to dissipate energy.

FRICTION SNUBBER FORCE MEASUREMENT SYSTEM (FSFMS)

Friction Snubber Force Measurement System is a device for measuring the forces within a standard friction snubber which was used during a TDOP Phase II field test late in 1978.

GAGE OF THE TRACK

The distance between the rails measured from the inside head of each rail at a right angle 5/8 inches below the top of the rail. The standard for this dimension on North American Railways is 4 feet, 8½ inches.

GRADE

Part of roadbed with changing elevation.

GROSS WEIGHT

The total weight of a car, including the lading.

HARMONIC ROLL

Periodic angular displacement of the vehicle body about its longitudinal axis, due to vertical track inputs close to the natural frequency of the carbody on its suspension, referred to as rock and roll.

HUNTING

Dynamic instability of sets of wheels or entire trucks consisting of a lateral translation along the axle and rotational vibration about a vertical axis.

HYSTERESIS

The dependence of the state of a system on its previous history, generally in the form of a lagging of a physical effect behind its cause.

INTERPOLATION

Derivation of function values at arguments not listed in the look-up table.

ITERATION

Method of computation through refinements. The iteration is called convergent if the sequence of computed values is convergent. The first element of that sequence is called initialization.

KINEMATICS

The branch of mechanics that deals with motion without consideration of inertial forces.

KINEMATIC WAVELENGTH

The wavelength of the sinusoidal motion of a wheelset or truck along the track when inertial forces are negligible.

KINETIC ENERGY

A moving body possesses an amount of energy equal to the energy needed to bring it to rest. The energy of the moving body is called kinetic energy. It is defined as half the product of the mass (mass movement of inertia) and the square of the velocity (angular velocity).

KINETIC FRICTION

Friction of motion, such as that between the brake shoe and wheel (when the wheel is turning) or as between a wheel and rail (during sliding or slipping). Kinetic friction is always less than static friction.

LATERAL STABILITY

Refers to the stability of a truck in the lateral direction. Trucks with a low degree of lateral stability tend to oscillate or hunt as speed is increased below some critical value.

LATERAL VIBRATION

Pure side to side movement in the horizontal plane.

LIFE CYCLE

The expected life of truck components over the life of a freight car.

LOAD EQUALIZATION

Ability of truck to maintain equal load distributions on all four wheels while accomodating the full range of in-service track geometry.

LOOK-UP TABLE

Values of a function at several arguments arranged in a way that a computer can access it.

LOWER CENTER ROLL

Rotation of the carbody about a virtual longitudinal axis below its center of gravity.

L/V RATIO

Defined as the ratio of the lateral force to the vertical force of a car or locomotive wheel on a rail. It is an important indicator of wheel climb, shifting of the track structure, rail turnover, and/or derailments.

MODEL

A set of equations which may or may not include nonlinearities used to simulate a physical system such as a wheelset, a truck, or a complete car/truck system.

NATURAL FREQUENCY

The frequency at which the system tends to vibrate when released after being displaced from neutral position.

OPERATIONAL CONDITIONS

Refers to the conditions or physical environment in which a truck must operate. Typical operational conditions include speed, track condition, and loading.

PARALLELOGRAMMING

Relative longitudinal displacement of truck side frames which causes the truck to go in and out of tram.

PARAMETER VARIATIONS

Refers to variations made on the parameters of a mathematical model to study the effect of component changes, configuration changes, or operational environment changes.

PERFORMANCE CRITERIA

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

PERFORMANCE INDEX

A measurable physical quantity characteristic of performance in a particular regime. An example of a performance index for hunting would be the critical speed, and for curve negotiation, the lateral load on the outer leading wheel of the truck. Each performance index must be qualified by a statement of conditions for which it applies, and which may affect its magnitude to varying degrees.

PERFORMANCE REGIME

The characteristic way in which a railcar or truck responds to a combination of track and operating conditions (such as speed). Inherent in this definition is a comparison with stable vehicle behavior on "ideal" tangent track. Performance regimes selected for truck characterization should be sufficiently distinct to permit ranking of truck performance on non-overlapping scales. The four primary regimes chosen are lateral stability, trackability, curve negotiation, and ride quality.

PITCH

Angular motion in the vertical plane about the axle perpendicular to the direction of the track.

POWER SPECTRAL DENSITY (PSD)

This represents the distribution of energy in a vibrating system under defined test conditions over the frequency spectrum specified.

PRIMARY SUSPENSION

Suspension elements between the side frame and wheelsets. For freight car trucks, these usually take the form of elastomeric pads.

RADIAL TRUCK

A truck in which the axles of the wheelsets are made to assume an approximately radial orientation in a curve. A radial truck is a special form of flexible truck in which the wheelsets or side frames are connected by special devices that determine the curving kinematics.

RECTIFIED SINE WAVE TRACK GEOMETRY

The shape of the vertical track deviations used to analytically describe jointed rail. The joints correspond to the low points of the rectified sine wave and the mid points represent the peaks of the sine wave.

REGRESSION

A statistical procedure for fitting the coefficients of an equation, usually based on minimizing the sum of the squares of the errors between the observed and predicted values of the equation.

RESONANCE

The condition at which forcing frequency is equal to natural frequency; this usually results in violent motion.

ROCK AND ROLL

An informal term for the excessive lateral rocking of cars and locomotives, usually at low speeds and associated with jointed rail. The speed range at which this cyclic phenomenon occurs is between 10 and 25 mph, with the exact speed determined by such factors as the wheel base, height of the center of gravity of each individual car or engine, the spring dampening associated with the suspension system of each vehicle, and the relative difference in elevation between successive joints in jointed rail territory. In extreme cases, actual wheel lift can occur which can result in derailments (see also Harmonic Roll).

ROLL

Rotation of the carbody about a longitudinal axis through the center of gravity.

ROLLABILITY

The relative resistance of the truck to longitudinal motion.

SECONDARY SUSPENSION

Suspension elements between the truck bolster and side frame. This is the principal means of isolating vibration in a freight car truck.

SHIMMY

A synonym for hunting.

SIDE BEARING

Bearings attached to the bolsters of a carbody, or truck, on each side of the center plate to prevent excessive rocking. The upper, or body side bearing, and the lower, or truck side bearing, are sometimes merely large flat surfaces. Other types of side bearings employ rollers, springs, and friction elements to maintain constant contact and control relative movement between body and truck.

SIDE FRAME

The frame which forms the side of a carbody or a truck. It includes the column braces, plate, etc., for the carbody; and the side member of a truck frame.

SILL (CAR)

The main longitudinal members of a car underframe which are connected transversely by the end sills, body bolsters, and cross ties. Sills are divided into side sills, intermediate sills, and center sills.

SIMULATION PROGRAM

A computer code which implements a mathematical model.

SNUBBERS

Damping devices which are used to attenuate oscillations of a car or truck. They may be similar to hydraulic shock absorbers. Friction devices are commonly used in rail vehicles.

SPRING GROUP

A helical car spring assembly formed of a number of separate springs (single or nested) and united by a common pair of spring plates.

SUPERELEVATION

The vertical distance between the heights of inner and outer edges of railroad rails.

SWING BOLSTER

A truck bolster which is suspended by hangers or links so that it can swing laterally in relation to the truck. The object of providing this swinging motion to the bolster is to prevent, as much as possible, lateral blows and shocks from being communicated to the carbody, and, vice versa, to prevent the momentum of the carbody from acting with its full force on the truck frame and wheel flanges.

SWING HANGER

Bars or links, attached at their upper ends to the frame of a swing motion truck, and carrying the spring plank at their lower ends. Also called bolster hanger.

SWING MOTION

A term applied to an arrangement of hangers or transom for the springs and truck bolster which enables a car body to swing laterally on the truck.

SWIVELING

Angular oscillation about an axis; a symmetry, usually applied to truck action when the bolster oscillates around the center pin.

THROAT (CAR WHEEL)

The curved transition between the wheel tread and flange.

TRACKABILITY

Refers to the ability of a truck to maintain equal wheel loads under all extremes of operating conditions.

TRACK-TRAIN DYNAMICS

A term used to describe the dynamic motion and the resulting dynamic forces that result from the interaction of the vehicles coupled into a train interacting with the track, under given climatic conditions, train handling, train makeup, grades, curvature, and operating policies.

TRACK-TRAIN ENVIRONMENT

All the conditions which affect the track and/or the train, such as grades, curvature, locomotive and car characteristics, train handling, etc.

TRACK TWIST

Refers to cross-level variations which occur within the wheelbase of the truck.

TRAM

This term applies to the diagonal measurement of axle bearing locations. When, in a four-wheel truck, the two diagonal measurements are equal, the truck is said to be in tram.

TRANSMISSIBILITY RATIOS

A frequency-dependent function of amplitude ratios called a transfer function, or a sequence of root mean square ratios of output to input over selected frequency bands.

TRUCK CENTER

The center point of a truck. The distance between truck centers is that distance as measured from one truck center to the other truck center on a single car.

TRUCK CLASSIFICATIONS

TYPE I: GENERAL PURPOSE DESIGN (STANDARD THREE-PIECE)

This design is interchangeable with existing trucks so as to preserve the present truck coupler height, support the carbody on center plates, utilize air brakes which are compatible with existing systems, accept standard wheelsets and journal bearings, and whose components meet applicable Association of American Railroads (AAR) requirements.

TYPE II: SPECIAL PURPOSE DESIGN (PREMIUM)

This design utilizes current wheelset and journal bearing assemblies, is compatible with existing air brake systems, and preserves car coupler height. The Type II truck may employ mechanisms other than center plate and side bearings for support and stabilization of the carbody.

TRUCK WHEEL BASE

The horizontal distance between the centers of the first and last axles of a truck.

UNDULATING GRADE

A track profile with grade changes so often that an average train passing over the track has some cars on three or more alternating ascending and descending grades. The train slack is always tending to adjust as cars on descending grades tend to roll faster than those on ascending grades.

UPPER CENTER ROLL

Rotation of the carbody about a longitudinal axis above its center of gravity.

VERTICAL VIBRATION

Pure up and down motion often described as bounce.

VIBRATION TEST UNIT (VTU)

One of the test machines located in the Rail Dynamics Laboratory, Pueblo, Colorado. This machine can provide both vertical and lateral excitation at both ends of a fully loaded railcar. The VTU will be used for component and system testing during TDOP Phase II.

WEAR PLATE

Renewable, wear-resistant, hardened steel plate which may be applied to center plates, side bearing pads, draft gear housings, etc.

WHEEL

The flanged rolling element which carries the weight and provides guidance for rail vehicles. It also serves as the brake drum for tread-braked equipment. Major classifications are "forged" (wrought) and "cast" steel wheels.

WHEEL CLIMB

This term applies to the condition where the lateral (axial) force between the wheel flange and rail head is great enough so that the resulting friction force causes the wheel flange to climb up on the rail.

WHEEL CONICITY

The slope of the wheel tread at the point of wheel/rail contact. Wheel conicity is used in linearized models to determine the change of rolling radius and the restoring force resulting from the particular wheel/rail contact.

WHEEL FLANGE

The projecting edge or rim on the periphery of a car wheel for keeping it on the rail.

WHEEL LIFT

This term applies to the lifting of a lightly loaded wheel due to the moment resulting from the high vertical force on the opposite bearing. Such forces are encountered when rail vehicles are operated at speeds too great for the existing superelevation on a curve, from very slow speed operation on a high superelevation curve, from high draft (or buff) forces on a curve, or from harmonious rocking of a car on rough track.

WHEEL PLATE

The part of a disc car wheel which connects the rim and the hub. It occupies the place and fulfills the same purpose as the spokes do in an open or spoke wheel.

WHEEL SLIDING

The situation where the wheel is rotating slower than longitudinal movement would dictate, and adhesion is lost.

WHEEL SLIPPING

The situation where the wheel rotates faster than longitudinal movement would dictate, and adhesion is lost.

WHEEL TREAD

The exterior cylindrical surface of a wheel which bears on the rails.

WHEEL UNLOADING

Reduction of vertical wheel reaction on the rail.

YAW

Angular motion in the horizontal plane about a vertical axis.

Truck Design Optimization Project: Phase II - Analysis Plan, 1980
US DOT, FRA, Larry L. Johnson, Arnold J Gilchrist