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16. Abstract
This report describes Vehicle/Track Interaction Assessment Techniques (IAT) which are developed to provide standardized procedures and tools in order to:

> Investigate the dynamic performance of railroad vehicles, and systematically identify and cure dynamic track interaction problems associated with a vehicle.

The IAT addresses ten performance issues: hunting, twist and roll, pitch and bounce, yaw and sway, steady-state curving, spiral negotiation, dynamic curving, steady buff and draft, longitudinal train action, and longitudinal impact. The report discusses the test and data analysis procedures required for each performance issue in terms of the control variables from track inputs that are required to create the test environment, the response variables to be measured, the extent of data analysis required, the data handling requirements, the performance indices to be used in interpreting the test results, and the potential test sites. This report is in two parts. Part I is contained in Volume I and covers the overall process of determining vehicle performance issues. Part II, comprised of Volumes II and III, discusses the detailed procedures to be used in the Vehicle/Track Interaction Assessment Techniques.

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Vehicle/Track Interaction, Rail Vehicle, Vehicle Dynamics, Transportation Safety, Test Analysis Procedures, Test Planning, Data Analysis Procedures

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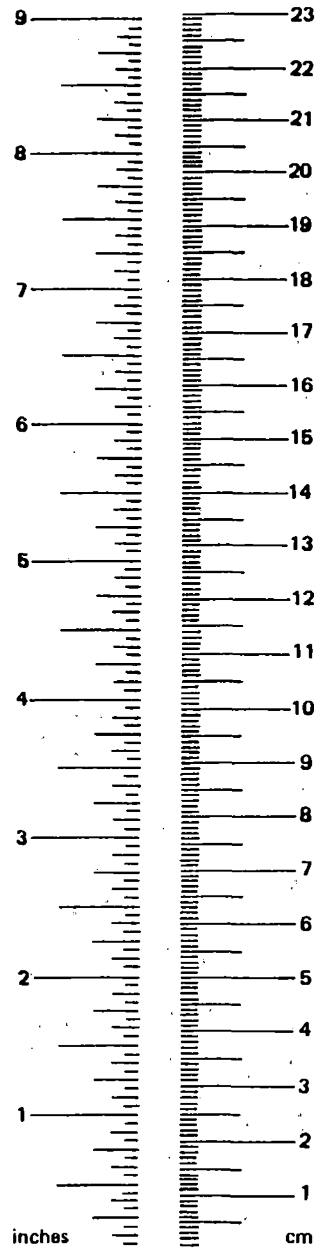
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

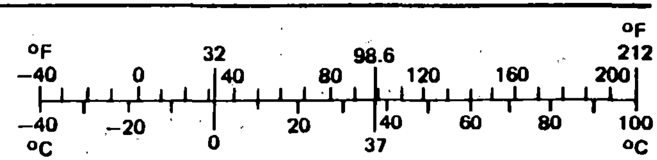
Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
	AREA			
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2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

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



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
	LENGTH			
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
	AREA			
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
	MASS (weight)			
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
	VOLUME			
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	36	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
	TEMPERATURE (exact)			
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



PREFACE

This volume is the second of three volumes dealing with the Vehicle/Track Interaction Assessment Techniques (IAT) which were developed by the Transportation Systems Center (TSC) and its contractors: Arthur D. Little, Inc. (ADL), Battelle Columbus Laboratories (BCL), ENSCO Inc., Kaman Sciences Corporation (KSC), Systems Control Technology Inc.(SCT), and The Analytic Sciences Corporation (TASC).

This information was developed from the Stability Assessment Facility for Equipment (SAFE) Program. That program had direct input from the railroad affiliated personnel of the International Government-Industry Track Train Dynamics Research Program and the Federal Railroad Administration, Track Safety Research Division.

The Vehicle/Track Interaction problems addressed by the IAT, called "Performance Issues," are listed below:

- Hunting;
- Twist and Roll;
- Pitch and Bounce;
- Yaw and Sway;
- Steady State Curving;
- Spiral Negotiation;
- Dynamic Curving;
- Steady Buff and Draft;
- Longitudinal Train Action; and
- Longitudinal Impact.

These problems have been responsible for compromising rail vehicle stability in the past and are expected to be important issues for consideration in future designs.

The IAT has evolved over the past few years through experience gained in conducting a number of tests dealing with vehicle/ track interaction. Essentially, the IAT is a systematic approach using a standardized set of procedures and tools (i.e., elements) for identifying, diagnosing and solving stability problems in a rail vehicle already in revenue service and for assessing the stability of a new or modified vehicle (freight car, passenger car, or locomotive) prior to its introduction into revenue service. The primary goal of the IAT is to provide a means of assessing the adequacy of rail vehicle stability at a minimum cost. This is accomplished by:

- Systematically developing an approach for identifying stability problems;
- Identifying the test procedures and tools necessary to assess the stability characteristics of the rail vehicles;
- Reducing, through the use of computer models, the amount of testing required;
- Summarizing the state-of-the-art in tools;
- Standardizing the nomenclature in stability assessment; and
- Providing the ability to compare data from different tests.

Although the IAT can determine the potential for derailment as a result of excessive motion between the wheel and rail or because of undesirable levels of wheel/rail interaction forces, it does not explicity deal with derailments resulting from the failure of a vehicle or track component due to wear, fatigue, or excessive stress caused by these forces. Also, the IAT has been developed to assess the dynamic performance of most types of freight cars, locomotives, and passenger cars; however, particular type of vehicle may not be sensitive to all Performance Issues. Therefore, the IAT incorporates a procedure for identifying the principal Performance Issues of concern for any vehicle design.

The IAT is organized in the form of Assessment Procedures. For each of three objectives of the IAT, a distinct procedure is identified and presented in the form of a flow chart. Thus, a procedure is defined for:

- The Modified Vehicle Assessment;
- The Vehicle Problem Diagnosis; and
- The Prototype Vehicle Assessment.

Each procedure requires a number of steps to be conducted in order to meet the Specific Assessment Objective. Often, but not always, test must be conducted to meet the Assessment Objective. These tests are distinctly different and complementary to the revenue service testing to which a new or modified vehicle is generally subjected. The IAT tests are designed to subject a vehicle or consist to a severe service environment which is simulated using test tracks or laboratory equipment. In this way, the range of dynamic characteristics of a vehicle could be brought out in a relatively short time. Achieving the same goal by means of a revenue service testing procedure may require extensive testing in many miles of track.

This document, which provides information on test and analysis procedures incorporated in the IAT, is divided into two parts. The first part introduces the IAT and provides the basic information on various Assessment Procedures and the steps to be taken in performing them. The second part consists of fifteen sections, each detailing one aspect of the Assessment Techniques. In this way, a potential user need only read Part 1 to understand the key aspects of the IAT; the details provided in the second part can be studied later while the user is gaining further knowledge of the IAT or before actually utilizing the IAT for Vehicle Performance Assessment.

This document was developed under the guidance of the TSC, with the following principal contributing individuals:

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The organizations involved in developing the document are shown on the next page.

SECTIONS	PRINCIPAL CONTRIBUTING ORGANIZATIONS*
PART I (ALL SECTIONS)	ADL
PART II	
A. Resources Available for Investigating Performance Issues	ADL
B. Accident History Investigation	ADL
C. Vehicle/Track Simulation Models	ADL/ISC
D. Rail Vehicle Model Validation	SCT
E. Test Plan Summaries	TSC
F. Test Facilities	ADL
G. Track Geometry Perturbations	TSC
H. Rail/Track Stiffness Measurements, Variations, and Simulations	TSC/BCL
I. Performance Indices	TASC
J. Analytical Techniques	ADL/ENSCO
K. Wayside and Onboard Instrumentation	TSC/ENSCO
L. Data Management	TSC
M. Field Test Planning	KSC
N. Vehicle Characterization	ADL/ENSCO
O. Reference Vehicle Usage	TASC

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This volume, the second of a three volume set, includes the following sections of Part II:

A. Resources Available for Investigating Performance Issues

B. Accident History Investigation

C. Vehicle/Track Simulation Models

D. Rail Vehicle Model Validation

E. Test Plan Summaries

F. Test Facilities

G. Track Geometry Perturbations

H. Rail/Track Stiffness Measurements, Variations, and Simulations

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SECTION A

RESOURCES AVAILABLE FOR INVESTIGATING PERFORMANCE ISSUES

The importance of performing a literature search was discussed in Subsection 2.1 and illustrated in Figure 2-1. This section contains a list of literature pertinent to each Performance Issue. Included in this literature are reports and technical papers describing each issue and various field and laboratory test programs investigating vehicle/track interaction. Through the use of past documentation, a user may be better able to understand and isolate the symptoms of a dynamic problem.

A great deal of research has been done in the area of vehicle track interaction. Theoretical studies, analytical solutions and test results have been documented for each of the following issues: hunting, twist and roll, pitch and bounce, yaw and sway, steady state curving, spiral negotiation, dynamic curving, steady buff and draft, longitudinal train action and longitudinal impact. Since the quantity of literature available to a reader is massive, this section isolates those documents that would be of particular interest to a potential user, with an intention to provide a reference list and indicate what performance issue(s) each reference addresses (see Table A.1). The papers and reports selected for this list are classical papers on the subject, and documents describing a field test or an analytical solution that had been tested. Additional literature is cited in the AAR Track/Train Dynamics Bibliographies (Volumes 1, 2, 3) or can be found by performing an additional literature search according to the procedure described in A.2. Document availability is discussed in A.3, with particular emphasis on the major organizations a potential user would need to access.

It should be noted that each section in Part 2 contains a bibliography documenting the reference literature. While these lists may overlap, they are oriented to the subject area addressed by the particular section.

A.1 Reference Documents

Table A.1 catalogues the reference documents with respect to the performance issue(s) that they address. Knowing the performance issue of interest, Table A.1 can be used to easily locate the relevant references. The reference list contains three types of material:

1. background documents that give a better understanding of the nature of the problem;
2. studies describing recent research into the solution of the problem; and
3. descriptions and analyses of field tests.

Because hunting is strongly dependent on creep and wheel/rail contact stresses, the literature on these two phenomena is identified under "Hunting".

A.2 Literature Search

Additional literature can be found through the process of a computer search, which provides a fast way of locating a large quantity of literature pertaining to a particular subject. Access to a data base is required in order to perform a literature search. There are three primary vendors of data bases: Lockheed (DIALOG), Systems Development Corporation (ORBIT) and Bibliography Retrieval Service. These vendors have a variety of data base types which they present in standardized format, update and then train users in implementation techniques. Some of the data bases that vendors have purchased the rights to are: Transportation Research Information Services (TRIS), National Technical Information Service (NTIS), Engineering Index (COMPENDEX), Information Service in Mechanical Engineering (ISMEC) and Science Citation Index (SCISEARCH). There are other less technical data bases that contain references to magazine and newspaper articles, as well as government publications.

In order to use a data base, a searcher can choose any or all three of the following methods to "talk" to the computer:

- Control vocabulary;

Legend

B: Background Work
F: Field Test
L: Laboratory Investigation
A: Analytical Study

Table A.1: Performance Issues and Associated Reference Documents

PERFORMANCE ISSUES

REFERENCE	CATEGORY	Hunting	Twist and Roll	Pitch and Bounce	Yaw and Sway	Steady State Curving	Spiral Negotiation	Dynamic Curving	Steady Buff and Draft	Longitudinal Train Action	Longitudinal Impact	
1	A										√	
2	F										√	
3	B								√	√	√	
4	L	√										
5	A					√						
6	F										√	
7	B								√	√		
8	B	√										
9	F		√	√	√			√				
10	F		√	√	√			√				
11	F		√	√	√			√				
12	F		√	√	√			√				
13	A,F		√	√							√	
14	F								√	√		
15	F	√				√		√	√			
16	B,F,L,A		√									
17	A		√									

Table A.1 (continued)

PERFORMANCE ISSUES

REFERENCE	CATEGORY	Hunting	Twist and Roll	Pitch and Bounce	Yaw and Sway	Steady State Curving	Spiral Negotiation	Dynamic Curving	Steady Buff and Draft	Longitudinal Train Action	Longitudinal Impact	
18	A		√									
19	A	√										
20	F	√				√		√				
21	A		√	√	√	√		√				
22	A	√										
23	A,L	√										
24	B								√	√	√	
25	A	√	√									
26	F,A	√	√			√				√		
27	F	√	√	√	√	√	√	√				
28	A					√						
29	A					√						
30	F		√	√	√	√		√				
31	F		√	√	√	√		√				
32	B	√	√	√	√	√	√	√		√		
33	F		√	√	√	√		√	√			
34	F		√	√	√	√		√				

Table A.1 (continued)

PERFORMANCE ISSUES

REFERENCE	CATEGORY	Hunting	Twist and Roll	Pitch and Bounce	Yaw and Sway	Steady State Curving	Spiral Negotiation	Dynamic Curving	Steady Buff and Draft	Longitudinal Train Action	Longitudinal Impact	
35	A	√				√	√	√				
36	A		√									
37	F		√	√	√			√				
38	F		√	√	√	√		√				
39	F		√	√	√	√		√				
40	B	√										
41	B	√				√	√	√				
42	B	√				√	√	√				
43	A,B	√	√	√		√		√	√			
44	F,A	√				√	√	√				
45	A								√			
46	A	√										
47	B,A	√										
48	B								√			
49	F,A	√	√	√		√	√					
50	F	√										
51	A	√			√							

Table A.1 (continued)

PERFORMANCE ISSUES

REFERENCE	CATEGORY	Hunting	Twist and Roll	Pitch and Bounce	Yaw and Sway	Steady State Curving	Spiral Negotiation	Dynamic Curving	Steady Buff and Draft	Longitudinal Train Action	Longitudinal Impact	
52	A,F		√									
53	F		√	√	√	√		√				
54	L,F,A		√			√		√				
55	L,A				√							
56	A					√						
57	A	√										
58	A	√										
59	A	√										
60	A		√									
61	L	√				√						
62	L	√				√						
63	A		√									
64	L,A	√				√						
65	B,A	√				√						
66	A	√										

- Natural language (free text);
- Identifiers.

"Control vocabulary" varies for each data base. It consists of predesignated terms used by indexers to describe each entry. "Natural language" is anything a searcher chooses to use. It can appear in the text, title, descriptor or identifier fields--or it may not appear at all. "Identifiers" are non-control words that, according to the indexer, are highly relevant to the article.

Typically, the individual requiring the literature search, and the one performing the search are not the same. Therefore, for an adequate search good communication is vital. The requester must have a clear idea of what is required. If the subject is technical, an explanation of the terminology and the purpose of the search will enable the searcher to locate the appropriate control vocabulary. Additionally, the requester should indicate how far back in time he or she wishes to look, if there are language restrictions, and whether or not titles are sufficient or abstracts are required.

Another technique for performing a literature search is through bibliographic listings. This method can be used by itself or in conjunction with a computer search. Typically, a paper or report contains a reference list used by the author. These reference lists can lead one to other pertinent documents. Additionally, there are several publications that contain lists of abstracts to existing reports and papers. Examples include the AAR Track Train Dynamics Bibliography (AAR/TTD), Railroad Research Information Service (RRIS), and National Technical Information Service (NTIS). Needless to say, this technique is lengthier than a computer search, but it is valuable in certain situations (e.g., when a computer is not easily accessible).

A.3 Document Availability

Three primary data base vendors were discussed in A.2. Their addresses may be useful to a user who does not currently participate in such a service. At this time, one can only purchase the previously mentioned data bases (TRIS, NTIS, COMPENDEX, etc...) through a vendor.

Lockheed
Dialog Information Service, Inc.
Marketing Department
3460 Hillview Avenue
Palo Alto, CA 94304

Systems Development Corporation
2500 Colorado Avenue
Santa Monica, CA 90406

Bibliography Retrieval Service
1200 Route 7
Latham, NY 12110

Information pertaining to the publications containing report abstracts can be obtained from the following sources:

Association of American Railroads
Technical Center
3140 South Federal Street
Chicago, Il. 60616

Railroad Research Information Service
Transportation Research Board
National Academy of Sciences
2101 Constitution Avenue, N.W.
Washington, D.C. 20418

National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161

An availability statement is usually included with each abstract and in the data base abstracts. To aid a potential user in obtaining desired literature, the names and addresses of several organizations from which documents can be ordered is given below:

Association of American Railroads
1920 L Street, N.W.
Washington, D.C. 20036

American Railway Engineering Association
2000 L Street, N.W.
Washington, D.C. 20036

American Society of Civil Engineers
345 East 47th Street
New York, NY 10017

Canadian Institute of Guided Ground Transport
Queen's University
Kingston, Ontario K7L 3N6
Canada

Engineering Societies Library
345 East 47th Street
New York, NY 10017

Superintendent of Documents
U.S. Government Printing Office
Washington, DC 20402

Railway Progress Institute
700 North Fairfax Street
Alexandria, VA 22314

Transportation Research Board
Publications Office
2101 Constitution Avenue, N.W.
Washington, DC 20418

Transport and Road Research Laboratory
Crowthorne, Berkshire RG11 6AU
England

International Union of Railways
Office for Research and Experiments
Oudenoord 60
Utrecht, Netherlands

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SECTION B
ACCIDENT HISTORY INVESTIGATIONS

B.1 Introduction

As discussed in Section 2, the structure of the IAT, an investigation of accident history helps not only in identifying the Performance Issues to be addressed for any particular type of vehicle or consist, but also in the overall evaluation of a vehicle's dynamic performance. To meet these objectives, one needs to be able to interpret accident investigation reports, and use them in developing hypotheses on which of the Performance Issues, if any, could be a measure of the cause of derailments. This section is designed to assist a user in interpreting accident reports.

A review of documented derailment accidents involving a particular car may be the most widely used measure of its performance in revenue service. This is appropriate since the ultimate test of a car is its ability to avoid derailing, and service demands adequate performance over a wide range of track, traffic, wear, and load conditions which cannot be easily replicated in a controlled experiment. This in-service evaluation of a car is the natural outcome of favorable interpretation of preliminary testing.

Accident reports not only help to determine if there is a dynamic problem afflicting a car, but help to define the nature of the problem as well. They can also point to various solutions or alterations which can improve performance. Further, experience gained from investigating accidents can improve the quality and efficiency of future investigations by insuring that essential evidence is not overlooked.

Simply put, accident records can be used to:

- Determine if a problem exists;
- Define conditions for derailment;
- Determine which Performance Issue is to be addressed;
- Dictate proper car, train, and track changes to compensate for that Issue; and

- Direct the course of future investigations.

In this section, we shall describe the various ways in which derailments are investigated and reported. We shall also discuss methods for using accident records to evaluate the performance of cars in service and to direct the efforts of persons investigating later accidents.

One fact should not be lost in discussing the uses of accident histories. Just as reviewing a car's accident history is useful in identifying its operational weaknesses, so the lack of derailments attributable to a particular car may be taken as confirmation of its acceptable performance. For the present, we are only concerned with using information gleaned from historical accidents and will not introduce the other consideration of assessing performance through avoidance of accidents.

B.2 Accident Investigations

The information available in accident reports and the use to which that information may be put depends somewhat upon the intensity of investigation, which, in turn, depends upon accident severity. Most accidents are minor. Minor accidents expect minor attention. They are typically investigated by railroad personnel who are also obliged to get the trains moving again. Investigators may not be afforded the time to examine all evidence and may be unable to properly determine cause for a derailment.

These accidents are generally reported in a standard format which documents certain facts, such as train number, car number, total damage, and accident location, but contain little, if any, discussion or interpretation. These are most useful for their factual content.

At the other extreme, major accidents, particularly those involving loss of life or extensive property damage, may be investigated by teams of experts from such agencies as the National Transportation Safety Board. They interview witnesses, perform laboratory analyses, reconstruct the accident, and catalog large quantities of data. The findings of such investigative teams are often reported in a written narrative including a discussion of probable cause and contributing factors, in addition to the standard report documentation. They have the added virtue of illuminating the thinking that led to a determination of cause.

B.3 Interpretation of Records

Both types of accident investigations are valuable for identifying and isolating car behavior problems. Usually, either through the detailed examination of a major accident or through the examination of a number of lesser accidents, it is determined that accidents involving one type of car or one set of circumstances have occurred with unusual frequency and should be analyzed.

Because minor accidents are much more prevalent than major ones, the bulk of statistically useful information about a car's performance is usually contained in data files which, individually, are not conclusive in identifying behavior problems. Collectively, however, they can show trends or tendencies suggesting certain performance issues. Data for those circumstances or that car type are analyzed to determine if common traits can be discerned. Some exclusions are imposed, such as collisions or derailments where an existing mechanical defect was determined to be the cause.

Examination of accident records and the identification of important performance issues affecting a rail car may be undertaken either as a unique study involving individual analytical techniques, or in a manner that has enough similarity with other studies to be conducted according to an accepted plan or format. In the first case, an individual analysis is devised to suite specific conditions and that analysis may be inappropriate for other studies. In the second case, a prepared scheme may be used, complete with specific procedures and worksheets. Both situations are briefly discussed here. The example below is included to illustrate how an individual case might be handled and to show the thinking contained in the worksheets which follow.

A rudimentary analysis would simply look at the fraction of derailments in which particular conditions were noted. That fraction might be presented as a table or graph as in the following example from hypothetical data.

Through a series of analyses of this type, it might be determined that these cars experience a disproportionate number of accidents on jointed rail and at speeds between 15 and 25 mph.

The next step is to examine all plausible explanations for this phenomenon. For example, one explanation is that "low joints" promote a rocking of the car to the point that the wheels on one side of a truck become unweighted and can no longer be guided by the wheel flanges.

If this explanation is valid, certain other observations are predicted: springs should show signs of being fully compressed; there should be a consecutive run of "low joints" in the rail; and the point of derailment may be at the beginning or end of a curve.

Finding this predicted evidence at a derailment supports the conclusion that excessive roll motion was to blame, and future investigation of derailments involving hopper cars might be predicated on the understanding that roll plays an important role.

To cite a real case, consider the problems encountered by the SDP-40F passenger locomotive in the late 1970's. After some derailments which the National Transportation Safety Board investigated, it was determined that there was no mechanical failure contribution to the accidents, suggesting a dynamic problem. At the request of the NTSB, the FRA instituted an analysis of data from accidents involving that locomotive. The FRA was able to establish that certain factors, including train speed and degree of curvature, were common to most or all of the derailments. The analysis also established that the rate of derailments for SDP-40F locomotives was slightly higher than that for the locomotives it replaced.

Drawing on those findings, which included a prediction of the locomotive's response to changes in certain components, a test program was instituted to identify the specific relationships among track, train operations, and locomotive behavior in an effort to first, determine the exact nature of the SDP-40F's dynamic behavior, and second, to indicate which alterations to the equipment would be most effective in improving its resistance to derailment.

Determining which performance issues affects a rail car may be accomplished using a less individualistic technique than that which is described here. The approach we present uses two worksheets. The first helps in identifying the recurring characteristics of derailments which have involved the cars in question. The second worksheet is used in conjunction with the first to permit a rapid and quantifiable comparison of those recurring characteristics with symptoms corresponding to specific performance issues. In the end, this process identifies the performance issue most likely to be a factor in derailments of that car.

Figure B-1 shows an example of one page of the first worksheet, which is used to extract the important information from accident records, and which may contain several pages. The task performed using this worksheet could be done by computer, but a manual approach is better for illustration. The reviewer marks the appropriate box whenever a descriptor is noted in an accident report. When all pertinent accidents have been reviewed, he calculates the percentage of accidents in which each descriptor was a factor. Using an arbitrary threshold, such as 50%, he assigns a value of 1 to descriptors occurring more often than that threshold, and 0 to those occurring less often. He then enters the worksheet shown in Figure B-2 with his results to compare them with symptoms associated with the performance issues.

In the worksheet of Figure B-2, the symptoms associated with performance issues correspond exactly with the descriptors listed on the first worksheet. Each performance issue - symptom relationship has been assigned a value representing the relative importance of that symptom in

ACCIDENT CIRCUMSTANCES		ACCIDENT NO.															%	VALUE
CATEGORY	DESCRIPTOR	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15		(0 / 1)
TRACK TYPE	Tangent - ½°																	
	Curve ½°- 2°																	
	Curve 2°- 4°																	
	Curve 4°- 8°																	
	Curve 8° +																	
	Spiral																	
	Special																	
	Unknown																	
RAIL CONSTRUCTION	CWR																	
	Stagger Jt.																	
	Parallel Jt.																	
	Unknown																	
TRACK PERTURBATION	Alignment																	
	X-Level																	
	Tight Gauge																	
	Wide Gauge																	
	Profile																	
	Soft Lat. Stiff.																	
	Soft Vert. Stiff.																	
	Unknown																	
WEATHER	Dry																	
	Wet																	
	Ice																	
	Unknown																	
SPEED (mph)	0 - 15																	
	15 - 25																	
	25 - 60																	
	Above 60																	
	Over Balance																	
	Under Balance																	
	Unknown																	
OPERATIONS & CAR COMBINATIONS	In Draft																	
	Braking																	
	Long-Short																	
	Heavy-Light																	
	Light Car																	
	High-Heavy																	
	Long/Stiff																	
CAR COMPONENTS	Suspension																	
	Draft Gear																	
	Worn Wheels																	
	Side Bearing																	
DERAILMENT MARKS & CHARACTERISTICS	Long Marks																	
	Short Marks																	
	High Side																	
	Low Side																	
	Panel Shift																	
	Rail Break																	

Figure B-1: Worksheet for Collecting Accident History Data

SYMPTOMS

PERFORMANCE ISSUES	Tangent - ½°	Curve ½° - 2°	Curve 2° - 4°	Curve 4° - 8°	Curve 8° +	Spiral	Special		CWR	Stagger Jt.	Parallel Jt.		Alignment	X-Level	Tight Gauge	Wide Gauge	Profile	Soft Lat. Stiff.	Soft Vert. Stiff.		Dry	Wet	Ice		0 - 15 mph	15 - 25 mph	25 - 60 mph	Above 60 mph	Over Balance	Under Balance		In Draft	Braking	Long-Short	Heavy-Light	Light Car	High-Heavy Car	Long/Stiff	Suspension	Draft Gear	Worn Wheels	Side Bearing	Long Marks	Short Marks	High Side	Low Side	Panel Shift	Rail Break	TOTAL / MAX = QUOTIENT	RANKING
HUNTING	2								1				1			1					1						3	2								2			1		2	1							/18	
TWIST & ROLL	2					1				3				1					1						1	3	2										3		1			1						1	/20	
PITCH & BOUNCE	2						1				3								1						2	2	1												1									1	/14	
YAW & SWAY	2												2			1		1			1					2	1											2	1			1			2			1	/15	
STEADY--STATE CURVING		1	2	2	3	2			1					1	1			1			1								1									2	1						2				/21	
SPIRAL NEGOTIATION			1	2	2	2												1											1									2	1						2				/14	
DYNAMIC CURVING		1	2	2	2	1				2					1			1	1		1						2	1	2					1					1		1		1		2		2	1	/28	
STEADY BUFF--DRAFT		1	1	2	3	2																			2	1			1	1		2	2	2		2								2					/24	
LONGITUDINAL TRAIN ACTION	1	1	1										1			1									2	1						1	2	1	1	1				1				2	2	1			/20	

FIGURE B-2: WORKSHEET FOR COMPARING ACCIDENTS WITH PERFORMANCE ISSUES defining the performance issue. For example, parallel or non-staggered joints are believed to be more important than suspension defects in leading to pitch and bounce behavior and have been assigned a higher value. The values used here are based on our understanding of the relative importances of the characteristics.

To use the second worksheet, the reviewer takes the first worksheet and holds it sideways so that the descriptors and corresponding symptoms align. He then notes when a "1" in the "VALUE" column matches with a number under a "SYMPTOM", and places a check mark in the box with value. For each row, at the end of the row, he records the sum of the "SYMPTOM" values in the marked boxes. When all matches are recorded, the values for each "PERFORMANCE ISSUE" row is divided by the maximum possible value for that row. The row with the highest quotient is determined to be the most likely performance issue afflicting the car under scrutiny. For the sake of illustration, Figures B-3 and B-4 have been included as examples of completed worksheets.

A user may prefer to supply values determined through sensitivity analysis of car behavior tempered with the judgment of experienced investigators. An alternative method for assigning values would be to derive a number from the percentage of accidents involving cars highly susceptible to a particular performance issue in which each descriptor is found. Of course, this method depends upon the confidence one has that the overriding cause of derailment for those cars is the identified performance issue.

B.4 Investigation Guidelines

As previously noted, not only is the examination of accident records useful in determining the dynamic characteristics of a particular type of car, it is an essential for guiding the course of future accident investigations. Knowing what transpired at previous derailments prepares an investigator to search for that evidence that is most useful in determining the cause of a subsequent derailment. Experience with accidents has been assembled to form accident investigation handbooks and guidelines that are available to assist investigators. [Ref. 1-5]

ACCIDENT CIRCUMSTANCES		ACCIDENT NO.																VALUE
CATEGORY	DESCRIPTOR	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	%	(0 / 1)
TRACK TYPE	Tangent - ½°	X		X		X			X	X							50	1
	Curve ½° - 2°				X			X									20	0
	Curve 2° - 4°		X														10	0
	Curve 4° - 8°																	0
	Curve 8° +																	0
	Spiral																	0
	Special										X						10	0
	Unknown						X										10	0
RAIL CONSTRUCTION	CWR			X				X									20	0
	Stagger Jt.	X	X		X					X							40	0
	Parallel Jt.					X					X						20	0
	Unknown						X										10	0
TRACK PERTURBATION	Alignment	X		X		X	X				X						50	
	X-Level	X		X				X									30	0
	Tight Gauge									X							10	0
	Wide Gauge	X	X		X				X		X						50	1
	Profile					X				X							20	0
	Soft Lat. Stiff.	X						X									20	0
	Soft Vert. Stiff.																	0
	Unknown																	0
WEATHER	Dry	X		X		X	X		X								50	1
	Wet		X								X						20	0
	Ice							X									10	0
	Unknown				X					X							20	0
SPEED (mph)	0 - 15																	0
	15 - 25			X					X		X						30	0
	25 - 60		X			X		X		X							40	0
	Above 60	X			X		X										30	0
	Over Balance		X														10	0
	Under Balance				X												10	0
	Unknown																	
OPERATIONS & CAR COMBINATIONS	In Draft																	
	Braking																	
	Long-Short	X															10	0
	Heavy-Light								X		X						20	0
	Light Car	X		X	X	X	X		X	X	X						80	1
	High-Heavy																	
	Long/Stiff																	
CAR COMPONENTS	Suspension	X		X				X		X							40	0
	Draft Gear	X			X												20	0
	Worn Wheels	X		X		X				X	X						50	1
	Side Bearing	X				X		X		X							40	0
DERAILMENT MARKS & CHARACTERISTICS	Long Marks	X				X		X		X	X						50	1
	Short Marks		X				X										20	0
	High Side		X														10	0
	Low Side							X									10	0
	Panel Shift																	0
	Rail Break				X												10	0

Figure B-3: Completed Sample Worksheet for Accident History Data

SYMPTOMS

PERFORMANCE ISSUES	HUNTING	TWIST & ROLL	PITCH & BOUNCE	YAW & SWAY	STEADY--STATE CURVING	SPIRAL NEGOTIATION	DYNAMIC CURVING	STEADY BUFF--DRAFT	LONGITUDINAL TRAIN ACTION
Tangent - ½°	2✓	2✓	2✓	2✓					1✓
Curve ½° - 2°					1		1	1	1
Curve 2° - 4°					2	1	2	1	1
Curve 4° - 8°					2	2	2	2	
Curve 8° +					3	2	2	3	
Spiral		1			2	2	1	2	
Special			1						
CWR	1				1				
Stagger Jt.		3					2		
Parallel Jt.			3						
Alignment	1✓			2✓					1✓
X-Level		1			1				
Tight Gauge					1		1		
Wide Gauge	1✓			1					1
Profile									
Soft Lat. Stiff.				1	1	1	1		
Soft Vert. Stiff.		1	1				1		
Dry	1✓			1✓	1		1		
Wet									
Ice									
0 - 15 mph		1	2					2	2
15 - 25 mph		3	2	2				1	1
25 - 60 mph	3	2	1	1			2		
Above 60 mph	2						1		
Over Balance					1	1	2	1	
Under Balance								1	
In Draft								2	1
Braking								2	2
Long-Short							1	2	1
Heavy-Light									1
Light Car	2✓							2✓	1✓
High-Heavy Car		3							
Long/Stiff				2	2	2			
Suspension	1	1	1	1	1	1	1		
Draft Gear									1
Worn Wheels	2✓						1✓		
Side Bearing	1	1		1✓					
Long Marks							1✓		
Short Marks								2	2
High Side				2	2	2	2		2
Low Side									1
Panel Shift							2		
Rail Break		1	1	1			1		
TOTAL / MAX = QUOTIENT	9/18 50%	2/20 10%	2/14 14%	6/15 40%	0/21 0	0/14 0	2/28 7%	2/24 8%	3/20 15%
RANKING	1	5	4	2					3

FIGURE B-4: COMPLETED WORKSHEET COMPARING ACCIDENT HISTORIES WITH PERFORMANCE ISSUES

In general, these guidelines direct attention to evidence which would support identifying a particular mechanical component, human factor, or performance issue as the cause of an accident. This is particularly important in the case of accidents caused by poor dynamic performance, since the supporting evidence is usually subtle and tends to be ambiguous. Without this direction, investigations would demand the acquisition and assessment of large volumes of data, which may be largely unnecessary. Thus, historical information tends to guide future investigations, while future investigations tend to confirm conclusions indicated by the earlier investigations.

Figure B-1 shows the accident characteristics found to be most indicative of certain performance issues, based on the guidelines set down in a number of investigative handbooks. As can be seen, most performance issues share characteristics with others. Most accidents do not exhibit all the characteristics of a single performance issue. The assignation of cause for a derailment often involves more than a listing of characteristics found at an accident site. Values shown in Figure B-1 are suggested weights for each characteristic. There is not, however, a consensus on the weight given to each characteristic in establishing a probable cause. The judgement of the individual investigators will be the final determinant in most cases.

To illustrate how this matrix might direct an investigator's thinking, consider a derailment occurring on tangent track. The investigator would note which performance issues involve tangent track. He would then determine, if possible, whether there were any alignment perturbations in the track before the accident, and whether the track was dry or devoid of lubricants. The next step is to determine the speed of the train prior to derailing. If the speed is relatively high, he will look for such car defects as worn wheels and suspension. In general, he progresses from evidence which he feels is most conclusive to that which is least conclusive. However, in actual investigations, much of the potential evidence is rendered useless or ambiguous by the accident. Under these circumstances, the investigator would make his judgement based on evidence he <u>can</u> see and the weight of that evidence. He would initially seek that evidence which carries the greatest weight. If that is not available or is indefinite, he would then seek those pieces of evidence which combine to offer the best support for selecting a probable cause.

Paradoxically, the interpretation of evidence surrounding an accident leads to a presumption of cause, which governs the evaluation of data: a sort of investigative tautology which can either reinforce sound judgement or sanctify fallacious reasoning. However, there is little reason to believe that evidence which has been meticulously collected and analyzed should regularly lead investigators to the wrong conclusions.

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SECTION C
VEHICLE/TRACK SIMULATION MODELS

Computer simulation models can be useful tools in planning a test program or conducting a vehicle performance assessment. Section 2-1 and Figure 2-1 of Part 1 illustrate how computer models fit into the overall IAI structure. Analytical models provide the opportunity of studying a problem under ideal and controlled conditions. With this tool, the effect of particular parameters can be individually questioned and tested.

This section is divided into two parts:

C.1 Analytical Studies of Vehicle Dynamics
C.2 Analytical Studies of Track Structure

Section C.1 summarizes the state-of-the-art models for the following performance issues: hunting, twist and roll, pitch and bounce, yaw and sway, steady state curving, quasi-steady state curving, dynamic curving, steady buff and draft, longitudinal train action, longitudinal impact and wheel/rail contact.

Section C.2 summarizes models describing the track structure and its performance. These models are of interest to a potential user because they enable one to predict stresses and deflections in track structure due to lateral and vertical wheel loads and can be used as input to several of the vehicle dynamic models. Integration of vehicle and track condition is important to both vehicle and track structure models. Track structure models would be particularly useful to assess the damage a new vehicle causes to the track. Therefore, a user would be able to estimate whether or not maintenance-of-way costs would increase as a result of placing a particular vehicle in service. A user might also wish to consider the effect to a vehicle over time due to the varying conditions of the track.

C.1 Analytical Studies of Vehicle Dynamics

Tables C.1-C.12 summarize the representative state-of-the-art analytical models. They are classified according to the following issues: hunting, twist and roll, pitch and bounce, yaw and sway, steady state curving, quasi-steady state curving, spiral negotiation, dynamic curving, steady buff and draft, longitudinal train action, longitudinal impact and wheel/rail contact. Table C.13 is intended for "general purpose" models, such as Dynalist. The tables outline the nature of the model (linear, quasi-linear, or non-linear), the degrees of freedom, the availability of a user's manual, how extensively a particular model has been used and the type of hardware that it has been run on in the past. Linear computer programs are those that contain only first order variables in the differential equations describing the system. Nonlinear models possess higher-order terms in these equations. A quasi-linear model is one in which nonlinear terms have been linearized in order to reach a solution. Degrees of freedom refers to the minimum number of independent coordinates needed to describe a system. It should be noted that the reference(s) to a particular model are indicated on each table. They are located under the column entitled "Organization" and are in parentheses. Additionally, two reference lists are included in Section C. Tables C.1 to C.12 refer to Reference List C.1. Every possible effort has been made to make these tables comprehensive and complete. They will require periodic updating as new models are created and old models are revised.

Computer simulation models are useful in studying a particular issue and seeing what parameters affect it under controlled conditions. The choice of the model depends upon the user. User constraints include: knowledge of computer programs, individual requirements (i.e., level of complexity), and facilities available for this work. A potential user should ask the following questions in order to choose the appropriate program.

1. What issue(s) am I interested in?

2. What type of vehicle do I want to study?

3. Is the annotated hardware system comparable to my own?

ISSUE: HUNTING

Table C.1: Computer Simulation Models of Hunting

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. Clemson/ASU (7, 37)	CU/ASU Freight Car Lateral Stability Models	Law and Cooperrider and Hadden	Freight	Linear	5 to 23	DOT/FRA; Authors	IBM-370/165-II and UNIVAC 1110	-	Limited
2. ASU/MIT (8, 37)	-	Cooperrider, Law & Hedrick	Freight	Quasi-Linear	9	TSC/FRA; Authors	UNIVAC 1110, DEC/VAX	-	No
3. ASU/MIT (11)	-	Hedrick, Arslan	Freight	Non-Linear	39	TSC/FRA; Authors	UNIVAC 1110, DEC/VAX	-	No
4. AAR (33, 37)	Freight Car Hunting Model	Cheung, Garg, Martin	Freight	Linear	25	AAR, Chicago	IBM-370/158	Yes	Yes
5. Southern Pacific Transportation Co. (37)	Frequency Domain Model (FDM)	Technical Research & Development Group	Freight	Linear	13	TDOP	IBM-370/168	Yes	Yes
6. Battelle	TRKVPSD (Mod IIB)	Ahlbeck, Doyle	Freight	Linear	11	Authors	CDC Cyber 70	Yes	Partial
7. Battelle	GENTRK	Hadden	Freight	Linear	11	Authors	CDC	No	Partial
8. Battelle	GENCAR	Hadden	Freight	Linear	23	Authors	CDC	No	Partial
9. Battelle	TRKHNT II	Doyle, Prause	Passenger, Loco.	Linear	7, 9	Authors	CDC 6400	No	No
10.Battelle	CARHNT II	Doyle, Prause	Passenger, Loco.	Linear	17, 21	Authors	CDC 6400	No	No
11.AAR (12, 37)	Locomotive Truck Hunting Model	Garg, Hartmann, Martin	Loco.	Linear	2,7,9,17,21	AAR, Chicago; GM, Electro-motive Division	IBM-370/158	Yes	Limited
12.MIT (13)	-	Hedrick	Loco.	Quasi-Linear	11	MIT; AAR, Chicago	DEC/VAX	Yes	No
13.MIT (13)	-	Hedrick	Loco.	Non-Linear	21	MIT; AAR, Chicago	DEC/VAX	Yes	No
14.MIT (14, 32)	-	Hedrick	Passenger	Linear	6, 15	MIT; Princeton, Pullman	DEC/VAX	Yes	Yes
15.AAR	Truck Hunting	-	Freight		-	-	Programmable Calculator		

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.2: Computer Simulation Models of Twist & Roll

ISSUE: TWIST & ROLL

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. MITRE (27,23)	FRATE	Kachadourian, Sussman, Anderes	Freight	Non-Linear	TOFC: 39 Boxcar: 33	?	CDC 7600	Yes	Limited
2. Illinois Institute of Technology (IIT: 26, 37)	Dynamics of a freight element in a RR freight car	Shum, Willis	Freight	Non-Linear	27	?	UNIVAC 1108	No	Limited
3. Battelle	PERTRK	Ahlbeck	Freight	Non-Linear	17	Author	CDC Cyber-74	No	Partial
4. Battelle	SPICOM	Doyle	Passenger	Non-Linear	6	Author	CDC	No	No
5. MIT (17)	-	Platin	Freight	Non-Linear	6	MIT, AAR, TSC	IBM-360; DEC/VAX	No	No
6. MIT (16)	-	Beaman	Freight	Quasi-Linear	6	MIT	PDP11	No	No
7. AAR (28)	-	Tse, Martin	Freight	Non-Linear	22	?	?	No	No
8. Wyle (18)	-	Healy	Freight	Non-Linear	11	?	?	No	Considerable
9. - (15)	-	Liepens	Freight	Linear	8	?	?	No	No
10. AAR	Rock & Roll Analysis	-	Freight				Programmable Calculator		

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.3: Computer Simulation Models of Pitch & Bounce

ISSUE: PITCH AND BOUNCE

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. AAR (25)	-	Garg, Chang, Goodspeed	Loco.	Non-Linear	38	?	?	No	No
2. TSC (30)	-	Perlman, DiMasi	Freight	Linear	6	TSC	DEC10	Yes	No
3. TSC (30)	-	Perlman, DiMasi	Passenger	Linear	8	TSC	DEC10		No
4. AAR	Pitch and Bounce	Track/Train Dynamics Steering Committee	-	-	-	-	Programmable Calculator	No	No

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.4: Computer Simulation Models of Yaw and Sway

ISSUE: YAW AND SWAY

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. Clemson/ASU	-	Fallon, Cooperrider, Law	Freight	Linear and Non-Linear	9	?	PDP-15 Digital/EAI	No	Partial
2. Clemson	-	?	Freight	Non-Linear	5	Doctoral Research	680 Analog/ EAI 693 Interstate	No	No
3. MIT	-	Hedrick	Freight	Non-Linear	6	MIT	DEC/VAX	No	No
4. AAR (24)	Locomotive Response Model	Chang, Garg, Hartman	Loco.	Linear	21	?	?	No	No
5. AAR (25)	-	Chang, Garg, Goodspeed	Loco.	Non-Linear	38	?	?	No	No
6. MIT (13)	-	Hedrick	Loco.	Quasi-Linear	11	MIT; AAR	DEC/VAX	Yes	No
7. MIT (13)	-	Hedrick	Loco.	Non-Linear	21	MIT; AAR	DEC/VAX	Yes	No
8. Battelle	TRKVPSD (Mod. I & III)	Doyle	Loc. and Passenger	Linear	14, 15	Battelle	CDC Cyber 70	Yes	Partial
9. MIT (32)	-	Wormely	Passenger	Linear	6	MIT; Princeton; Pullman	DEC/VAX	Yes	No
10. TSC (30)	-	Perlman, DiMasi	Passenger	Linear	15	TSC	DEC10	Yes	No
11. MIT (32)	-	Wormely	Passenger	Non-Linear	15	MIT; Princeton Pullman	DEC/VAX	Yes	Partial
12. AAR	-	Track/Train Dynamics Steering Committee	-	-	-	-	Programmable Calculator	No	No

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.5: Computer Simulation Models of Steady State Curving

ISSUE: STEADY STATE CURVING

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. Battelle	SSCUR2	Doyle, Ahlbeck	Passenger, Loco.	Non-Linear	6	TSC; Battelle	CDC Cyber 73	No	Partial
2. Clemson/ASU (20, 37)	2-Axle Vehicle Steady Curve Negotiation	Law and Cooperrider	Idealized 2-Axle	Non-Linear	7	Clemson University	?	No	No
3. Clemson/ASU (37)	Non-Linear Steady Curving	Law and Cooperrider	Freight	Non-Linear	9	Clemson University	?	No	No
4. Battelle (37)	Non-Linear Full-Car Steady Curving	?	Passenger	Non-Linear	11	Battelle	?	No	No
5. TASC	SIMCAR	?	Freight	Non-Linear	14, 16	TSC;TASC	DEC10, IBM	No	No
6. Clemson/ASU (37)	Non-Linear Full-Car Steady Curving	Law and Cooperrider	Passenger	Non-Linear	11	Clemson University	IBM-370	No	No
7. Clemson/ASU (37)	Non-Linear Steady Curving Model	Law and Cooperrider	Freight	Non-Linear	17	Clemson University	IBM-370	No	No
8. Battelle (37)	SSCUR3	Doyle	Loco.	Non-Linear	8	Battelle; TSC; TASC	CDC Cyber 73	No	No
9. AAR / EMD (34)	2-3-4 Axle Truck Curving Model	Smith	Loco.	Quasi-Static	?	GM, Electro-motive Division	IBM-370/158	Yes	Yes
10. MIT (14, 13)	-	Hedrick	Passenger	Non-Linear	17,21	MIT; AAR	DEC/VAX	Yes	No

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.6: Computer Simulation Models of Quasi-Steady State Curving

ISSUE: QUASI-STEADY STATE CURVING

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. US Department of Transportation/ Transportation Systems Center	Quasi-Static Curve Negotiating Model	Perlman, Weinstock	Idealized	Non-Linear	4	TSC	DEC-10	Yes	No
2. AAR (37)	RTCN	Track/Train Dynamics Steering Committee	Loco.	Non-Linear	10	TTD	IBM-370; 600K byte	Yes	Yes

Table C.7: Computer Simulations Models of Spiral Negotiation

ISSUE: SPIRAL NEGOTIATION

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. Battelle	CURVENT	Doyle	Passenger	Non-Linear	9	Author	CDC; IBM; UNIVAC	No	No
2. Battelle	SPICOM	Doyle	Passenger	Non-Linear	6	Author	CDC	No	Partial

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.8: Computer Simulation Models of Dynamic Curving

ISSUE: DYNAMIC CURVING

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. Battelle (37)	SPICOM	Doyle	Passenger	Non-Linear	9	Battelle	CDC	No	No
2. Clemson/ASU (37)	Non-Linear 1/2 Car Curve Entry Model	Law and Cooperrider	Passenger	Non-Linear	9	ASU; CU	IBM-370	No	No
3. AAR	Freight Car Curving Model	Garg	Freight	Non-Linear	43	TTD	?	No	No
4. AAR	Locomotive Curving Model	Garg	Loco.	Non-Linear	59	TTD	?	No	No
5. Clemson/ASU (20, 22, 37)	CURVLOCO	Law and Cooperrider	Loco (6 axle)	Non-Linear	27	TASC; TSC; Authors	IBM-370	Yes	No
6. TASC (37)	RVDCADET 2	-	Freight	Non-Linear	14	TASC	IBM-370 (3031)	No	No
7. MIT	-	Hedrick	Freight	Non-Linear	23	MIT	DEC/VAX	Yes	No
8. TASC	SIMCAR	-	Freight	Non-Linear	14, 16	TSC, TASC	DEC10; IBM	No	No
9. Battelle	CURVENT	Doyle	Passenger	Non-Linear	9	Author	CDC; IBM; UNIVAC	No	No

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.9: Computer Simulation Models of Steady Buff & Draft

ISSUE: BUFF & DRAFT

ORGANIZATION	MODEL	AUTHOR(S)	CONSIST	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. AAR (4, 37)	QLTS	Thomas, MacMillan, Martin	Freight	Non-Linear	5 Cars -- (2 each)	Limited	IBM-370/150; 292K bytes	Yes	No
2. US DOT/TSC (3)	TSA	Tanne, Brantman	Freight	Non-Linear	# of Cars (2 each)	TSC	DEC10; IBM	Yes	No
3. AAR (5)	DLTS	Track/Train Dynamics Steering Committee	Freight	Non-Linear	# of Cars (5 each)	?	?	Yes	Partial

Table C.10: Computer Simulation Models of Longitudinal Train Action

ISSUE: LONGITUDINAL TRAIN ACTION

ORGANIZATION	MODEL	AUTHOR(S)	CONSIST	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. AAR (1, 37)	TOS	Luttrell, Gupta, Low, Martin	Freight	Non-Linear	# of Nodal Blocks (1 each)	Over 50 organizations --extensive use	IBM-370/158, 216K bytes	Yes	Partial
2. AAR (2, 37)	DLTAM	Martin, Plouffe, Ahmed, Antezak, Tideman	Freight	Non-Linear	# of Cars (1 each)	Limited	IBM-370/158; 400K bytes	Yes	Partial

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.11: Computer Simulation Models of Longitudinal Impact

ISSUE: LONGITUDINAL IMPACT

ORGANIZATION	MODEL	AUTHOR(S)	CONSIST	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. Washington University (37)	Vertical Train Action	Sheng	Freight	Non-Linear	# of Cars (3 each)	?	IBM-360/65; 64K bytes	Yes	?
2. AAR (6, 37)	VTS	Raidt, Shum, Martin, Garg	Freight	Non-Linear	Up to 10 Cars (4 each)	Limited	IBM-370/158; 156 bytes	Yes	Partial

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

Table C.12: Computer Simulation Models of Wheel/Rail Contact

ISSUE: WHEEL/RAIL CONTACT

ORGANIZATION	MODEL	AUTHOR(S)	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. University of Penn. (37)	CONFORM	B. Paul; J. Hashemi	?	-	?	UNIVAC 90/70	Yes	?
2. University of Penn. (37)	COUNTACT	B. Paul; J. Hashemi	?	-	?	IBM 370/168 & UNIVAC 90/70	Yes	?
3. Clemson U (37, 42)	Kalker's Exact Theory	Goree	Non-linear	-	Author	IBM 370/165 IBM 370/3165-II	Yes	Partial
4. Clemson U (37, 43)	Kalker's Simplified Theory	Goree & Law	Non-linear	-	Author	IBM 370/165 IBM 370/3165-II	Yes	Partial
5. Delft University (The Netherlands) (37)	DUVOROL	Kalker	?	-	Author	IBM 370/158	NO	Yes
6. TSC	CREEP	?	?	-	?	?	?	?
7. ASU	WHRAIL	Cooperrider; Law	Non-linear	-	TSC; Princeton	DEC10; UNIVAC	Yes	Partial
8. ASU	WHRAILA	Cooperrider; Heller	Non-linear	-	TSC; Princeton	DEC10; UNIVAC	Yes	Partial
9. Princeton University (44)	-	Sweet, Karmel	Non-linear	3	Thesis	?	Yes	Limited

Table C.13: General Computer Simulation Models

ISSUE: GENERAL MODELS

ORGANIZATION	MODEL	AUTHOR(S)	VEHICLE	MODEL CLASS	DEGREES OF FREEDOM	USAGE	HARDWARE	USERS' MANUAL	VALIDATION*
1. J.W. Wiggins, Co.	DYNALYST II	Hasselman Bronswicki	Passenger	Linear	8, 14	DOT/TSC; Authors	CDC 6600 163K Core	Yes	No

*The term "validation" is meant to imply that the model results have been favorably compared to experimental data. It does not mean that the model has been validated over all possible operating ranges. For more details on validation techniques, see Section D.

This last question is important in terms of the ease of using or adapting a program. For example, each program has been run on a particular type of computer. Although the software can be run on any system, it could require changes in order to perform on an alternative hardware system. It is for this reason that the hardware has been described for the models wherever possible.

The author and the associated organization are also identified with each program, as well as the pertinent reference article. Potential user's are urged to locate the reference(s) they require, and the User's Manual(s) where available. Additionally, the author(s) should be contacted in order to ascertain whether or not the model has been updated and the type and status of its validation, if any. As the tables indicate, the models have not been validated over all operating ranges. Therefore, the user should determine the validation status of a model prior to its implementation. Section D offers details on validation techniques, and the respective author(s) would have particular insights. Additionally, a user may wish to have the author run his program for a negotiated cost. If such is the case, the author should be contacted and arrangements made between the individuals.

C.2 Analytical Studies of Track Structure

C.2.1 Introduction

When examining vehicle/track interaction, it is necessary to understand how wheel loads affect the response of various track structure components. Analytical techniques were originally developed as track design tools, but their concepts can also be applied to studying the effect that specific track irregularities have on overall track structure and vehicle dynamics.

C.2.2 History

The first track analysis methods were based on the infinite beam-on-elastic foundation theory (Winkler Model) which still remains the most frequently used technique today. It can adequately predict stresses and moments in the rails for simplified uniform conditions where values for the track support modulus, U, have been obtained from field measurements. It should be noted that this method gives a poor prediction of substructure stresses and displacements and is too simplified for today's general analytical requirements. Since individual substructure layer properties are not characterized in the Winkler Model, the effects of substructure component properties cannot be evaluated.

More recently, three-dimensional finite element models have been developed that can predict roadbed stresses and displacement. These analytical methods, listed in Table C-14, have been developed in the U.S. and Canada over the past ten years. Table C-14 documents the model's name, researcher and basic description, as well as the important features of each model. Caution should be used when studying this table since researchers are continually expanding their models. Therefore, authors should be contacted for current model status.

Three basic substructure analytical design models have been developed and used by foreign railroad design engineers. The German Federal Railway, Hungarian State Railway, Czechoslovakian State Railway and the Japanese National Railway use multi-layer elastic methods. This empirical method is based on varying the elastic properties throughout a multi-layer material. British Railways uses a threshold stress approach which is also an elastic theory based on limiting the resilient stress to a particular value of residual deformation. The Indian Railways combines an elastic method and subgrade stresses based on an effective stress and Mohr-Coulomb failure model. Further details can be obtained in the report, Ballast and Subgrade Requirements Study: Railroad Track Substructure Design and Performance Evaluation Practices [63].

C.2.3 Aspects of Analytical Models

In most cases, a track analytical computer model is a three-dimensional multi-layer elastic study that examines the dynamic response of conventional railroad track structure. The output consists of railseat load reactions, tie/ballast reactions, deflections, bending

TABLE C-14: TRACK STRUCTURE ANALYTICAL MODELS

MODEL NAME	RESEARCHER(s)	MODEL DESCRIPTION
Pyramid Model 1970	Meacham et al. (BCL)	Beam on elastic foundation analysis with modified track modulus, U
1970	Lundgren et al. (Illinois)	Two dimensional finite element model (FEM)
Analysis of Rail Track Structures (ARTS) 1978	Svec, Turcke, Raymond et al. (Queen's Univ.)	Three dimensional FEM. Beam elements for superstructure, hexahedronal and tetrahedronal elements for substructure.

DEVELOPED IN THE UNITED STATES AND CANADA

IMPORTANT FEATURES	REMARKS
Used theoretical approach to determine U which included effects of rail type, tie type and width, tie bearing area, ballast type, depth and stiffness, and subgrade type and and stiffness.	One of earliest attempts to rationally include the effects of substructure properties in track analysis. Poor correlation with field test results.
Analyzed longitudinal section along centerline of track. Plane strain behavior of substructure assumed.	Early forerunner of ILLITRACK. Poor correlation with measured results.
Detailed description of physical track substructure. Stress path dependent and nonlinear elastic behavior of ballast, subballast, and subgrade accounted for using "bicubic spline" functions. No-tension capabilities of substructure materials accounted for. Beam element can be employed to model rails & ties.	Emphasized geotechnial aspects of track behavior. Bicubic spline functions developed from triaxial test data. Partially successful correlation with full-scale model data.

MODEL NAME	RESEARCHER(s)	MODEL DESCRIPTION
ILLI-TRACK 1976	Tayabji, Thompson and Robnett (Illinois)	Pseudo-three dimensional FEM. Two plane strain two dimensional FEM used in combination.
(PSA, BURMISTER) Track Structure Models 1975	So, Ma and Martin (AAR)	Series of 15 computer models to predict stresses and strains in various track components. Multiple models (simple and sophisticaed) to perform same task.

IMPORTANT FEATURES	REMARKS
Element thickness increased with depth according to value ϕ in longitudianal analysis to represent transverse load spreading in plane strain analysis. Initial thickness of surface element made equal to effective tie bearing length, L to represent effective load transfer area between tie and ballast. Resilient modulus, E_r used to represent nonlinear elastic behavior of ballast, subballast and subgrade.	Emphasize geotechnial aspects of track behavior. Attempts to simplify and reduce cost of analytical models.
Multiple models (simple and sophisticated) developed to perform same task. Model used depends on degree of of analysis (preliminary or detailed). BURMISTER multi-layer elastic model developed for substructure. Prismatic Solid Analysis (PSA), a three dimensional FEM developed for superstructure Prismatic Solid Analysis (PSA), a three dimensional FEM developed for superstructure analysis.	Computational requirements minimized for type of analysis needed. Components interactions may be lost through model subdivisions. PSA and Burmister model results agreed well with field data from others. Models used to perform parametric studies and develop sample design charts.

MODEL NAME	RESEARCHER(s)	MODEL DESCRIPTION
Multi Layer Track-Analysis Model (MULTA) 1978	Prause, Kennedy et al. (BCL)	Combination of two models developed by AAR. The three dimensional FEM called LAC for super-structure analysis, and the Burmister multi-layer elastic sub-structure model.

IMPORTANT FEATURES	REMARKS
Includes essentially all important aspects of individual track componnet performance in analysis. Interactive approach used between LAC and Burmister model to solve for stresses and strains in track structure components: Wheel-rail, rail-tie, and tie-ballast reactions are obtained from LAC. Influence coefficients generated by Burmister using uniformly loaded circular areas which represent the vertical pressure from equivalent tie bearing areas. Influence coefficients used in LAC to generate rail-tie reactions, rail-tie displacements, and tie-ballast pressures. Tie-ballast pressures used in Burmister to obtain stresses and displacements in substructure layers.	Allows the effects of changes in various track components on other components to be studied. No relative displacement betwen tie and ballast. Allows unrealistic tension to develop. Used homogeneous, isotropic, linearly elastic substructure properties. Substructure materials are nonlinear and stress dependent. Analytical results compared well with dynamic data from FAST.

MODEL NAME	RESEARCHER(s)	MODEL DESCRIPTION
GEOTRACK 1978	Adegoke, Chang and Selig (UMASS)	Modification of MULTA for studying substructure behavior

IMPORTANT FEATURES	REMARKS
Interactive procedure used to vary the resilient modulus, E_r for the stress state in each layer. Stresses and E_r varied until a sufficiently converged solution is obtained. Can compute results for six depth locations within five layers. Modulus is a function of the stress state rather than a constant.	Emphasizes geotechnial aspects of track behavior. Improved characterization of roadbed materials by including stress dependent, nonlinear behavior. Analytical results compared well with dynamic data from FAST. Uses truck loadings as opposed to axle loadings. Simplicity, efficiency, and costs improved for MULTA.

moments, deviator, bulk, and principal stresses. Parameters included in the three dimensional finite element analysis are the following.

- Load conditions -- single axle (two wheels), or truck loading (pair of adjacent axle loads on the rail).
- Rail -- size and stiffness.
- Tie -- size, stiffness, bending and elimination of ties.
- Ballast and subballast -- strength, stiffness and thickness.
- Subgrade -- strength and stiffness.

Some additional aspects which are incorporated in the finite element analysis are the boundary conditions, triaxial test results, rail/tie system geometry and material properties, such as Young's modulus, Poisson's ratio, unit weight and moduli used in computing the stresses. Careful consideration should be given when selecting the input parameters for these analytical models, particularly the track modulus, U. This parameter is defined as stiffness per unit length along the rail and should not be confused with the soil mechanics modulus obtained from plate load tests.

C.2.4 Selecting an Analytical Model

Proper selection of an appropriate analytical model to solve a particular problem requires a good understanding of the different degradation modes involved. Table C-15 indicates the different types of degradation modes, the analytical requirements needed to investigate the problem, and the models that may apply. Although this table is a general guideline for selecting analytical models, a more detailed analysis is required prior to implementing a particular computer program.

C.2.5 Additional Information of Track Analytical Models

Further explanation of the capabilities of ARTS, Illi-Track, PSA-Burmister, MULTA and the GEOTRACK models are presented in Table C-16. Before using these models, it is recommended that a detailed study of the user's manual be undertaken.

TABLE C-15. SUMMARY OF TRACK DEGRADATION MODE ANALYSIS REQUIREMENTS

Degradation Modes	Performance Issues	Analysis Model Requirements	Related Models
Tie failure from bending and torsion	1) Pitch & Bounce 2) Twist & Roll	Single vertical tie finite element model with rail seat loads and moments and variable stiffness ballast support to predict tie bending moments. Estimate of maximum torsional moment based on predicted statistical tie plate loads.	PSA Illi-Track MULTA ARTS GEOTRACK
Rail fastener failure a) Pull-out of tie inserts b) Failure of rail clips	1) Yaw & Sway	Three demensional finite element track model which includes non-symetrical vertical and lateral W/R loads, fastner stiffness, rail torsion and non-linear stiffnesses for fastener and ballast.	ARTS GEOTRACK
Track surface deterioration (vertical profile and cross level) a) Ballast failure and flow b) Subgrade failure and settlement	1) Twist & Roll 2) Pitch & Bounce	Vertical track model using Burmister's multi-layer roadbed model and load distribution program to predict ballast and subgrade pressure and tie and tie deflections.	PSA Illi-Track MULTA ARTS GEOTRACK
Track alinement deterioration	1) Yaw & Sway	Vertical Track model using Burmister's multi-layer roadbed model and load distribution program to predict vertical tie loads. Two dimensional finite	PSA Illi-Track MULTA ARTS GEOTRACK

Degradation Modes	Performance Issues
Track alinement deterioration (cont.)	Yaw & Sway
Rail rollover	1) Yaw & Sway
Wide gauge	1) Yaw & Sway

Analysis Model Requirements	Related Models
element lateral track model with thermal loads, rail fastener torsion al resistance and nonlinear ballast resistance which is dependent on vertical tie loads.	
Three demensional finite model which includes non-symmetrical vertical and lateral W/R loads, fastener stiffness, rail torsion and non-linear stiffness for fastener and ballast.	ARTS GEOTRACK
Three demensional track model which includes non-symmetrical vertical and lateral W/R loads, fastener stiffness, rail torsion and non-linear stiffnesses for fastener and ballast.	ARTS GEOTRACK

TABLE C-16: DETAILS OF SELECTED TRACK ANALYTICAL MODELS

ANALYSIS OF RAIL TRACK STRUCTURES-ARTS (48)

Developed by G. P. Raymond, D. J. Turcke, D. W. Siu

Department of Civil Engineering Queen's University for Transport Canada Research & Development Centre - 1978 updated - 1980.

Finite element method is used and a choice of hexahedronal, tetrahedornal and beam elements is provided which can model non-homogeneous isotropic elastic materials properties.

Capability - This program is a static linear or non-linear analysis for moments, deflection, stresses and strains of a three deminsional railroad track structure under static loads.

- Programming Language - Fortran IV
- Hardware: B6700 240K
- Usage-The program has been used mainly in experimental tests run at Queen's University
- User Manual avaiable from the authors or sponsoring agencies but at present the program is resticted

ILLI-TRACK-A FINITE ELEMENT ANALYSIS OF CONVENTIONAL RAILWAY TRACK SUPPORT SYSTEM (58)

Developed by: M. R. Thompson, S. D. Tayabji, A. L. Robnett-Department of Civil Engineering, University of Illinois at Urbana-Champaign-1976.

The model is a two stage procedure the first being a longitudinial analysis by a rail-tie representation as a beam spring system. Loads are input as point loads acting on the rail. The second stage transverse analysis is performed. Considers a tie resting on the ballast. The maximum deflection at a tie-obtained from the longitudinal analysis is used as input. In each stage a plane-strain type analysis is performed. Usage-this model has been validated using the measured response of the Kansas Test Track.

PSA, BURMISTER - TRACK STRUCTURE MODELS MATHEMATICAL MODELS FOR TRACK STRUCTURE (54)

Developed by: W. So,G.C. Martin, B. Singh, I.C. Chang, E.H. Chang Association of American Railroads.

Model Description & Capabilities - Fifteen mathematical models were developed to predict stresses and diflections due to vertical and lateral wheel loads in track structures.

The following list indicates which models apply to what area:

1. Vertical Track Models: 1) Beam on Elastic Foundation 2) Finite Element Model
2. Lateral Track Models: 1) Beam on Elastic Foundation 2) Finite Element Model
3. Tie Models: 1) Simple Beam 2) Finite Element Model
4. Rail-Fastener Model
5. Ballast-Subgrade Models: 1) Talbot's Equation 2) Pyramid of Stress 3) Broussinesq's Equations 4) Westergaard's Equations 5) Curruti's Equations 6) Burmister's Multi-Layer Elastic System
6. Three-Dimensional Track Models: 1) Finite Element Model with Prismatic Elements 2) Finite Element Model With Member Representation

MULTI-LAYERED TRACK ANALYSIS (MULTA) (53)

Developed by: R.H. Prause, J. C. Kennedy D. AhlbeckBattelle Columbus Laboratories 1978-

Capabilities and Description - This program has a multi-layer representation of the track roadbed which predicts realistic stress distributions in the ballast and subgrade. It also takes into account the effect of the bending and changes and in ballast depth, ballast and subgrade material properties, the size, and tie spacing. features are the following:

1. Model use 2-7 layers of homogeneous isotropic elastic material each having distinct material properties & depth.
2. Flexible rail fastener and tie pad can be adjusted via vertical spring stiffness between each rail tie.

3. Loads can be supplied over tie or between ties.

4. Direct response to deformation due to bending and its effect on the tie/ballast contact area.

Usage - Concrete tie track from Revenue Service used for validating the model

GEOTRACK MODEL FOR RAILROAD TRACK PERFORMANCE GEOTRACK (64)

Developed: C.S. Chang , C.W. Adegoke and E.T. Selig, Department of Cviil Engineering, Unversity of Massachusetts 1978 up dated 1980-1981

The program is a three-dimensional, multilayer model for the elastic response of railroad track, considering stress dependent material properaties and separation of tie and ballast. Output of the model includes prediction of permanent settlement of the track, rail-tie reactions, tie-ballast reactions, tie and rail bending moments. It also provides diflections and the three dimensinal stress state at specified locations within the roadbed layer. Solutions may be obtained for single axle or for truck loading.

- Hardware-Capabilty of running on TSC DEC System 10/KL - 127K
- Usage - Extensive field test correlation with FAST, Pueblo Co, and revenue service track in conjunction with Battelle Laboratory and Office of Research & Development FRA.

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SECTION D.
RAIL VEHICLE MODEL VALIDATION

D-1 INTRODUCTION

D-1.1 MODELING AND MODEL VALIDATION

Mathematical models have been found to be extremely useful tools for predicting the performance of complicated systems in place of costly test programs. The development of modern high-speed digital computers and the continuing sharp decline in the cost of computing have made modeling an increasingly attractive alternative to testing of dynamic systems. However, even the most sophisticated models are of no use unless they can be proven to adequately represent reality (the performance of the physical system being modeled). Validation is the process of proving the adequacy of a model by use of appropriate test data.

It is neither feasible nor desirable to design a mathematical model to represent all possible modes of response of a vehicle for every anticipated purpose. Models are tools which are best designed for specific, well-defined purposes. In fact, a model ideally suited for one purpose may be totally inappropriate for another, so the model purpose must always be kept in mind. The design of a model always includes trade-offs among different features. The intended purpose should govern how these tradeoffs are managed. Increasing the complexity of a model (including more degrees of freedom or nonlinearities) tends to permit it to represent more types of behavior more accurately, but imposes the penalties of increasing the cost to develop, debug, and execute the model and making user interface more difficult (more input requirements, more difficult to understand). Similarly, the model which is designed to be as general as possible will probably be more difficult and costly to use than one which is designed for a specific purpose and specific conditions.

Mathematical models, once they are properly validated, can offer distinct advantages over full-scale testing for studying many aspects of the dynamics of rail vehicles:

(1) Models are much less expensive to run than fullscale tests and they are not subject to weatherrelated problems. Full-scale tests require the use of very costly vehicles, train crews, instrumentation, data processing equipment, and testing personnel (technicians), as well as track which either has to be specially constructed for testing or must be taken out of revenue-producing uses for a period of time.

(2) Models can be run more rapidly than full-scale tests (even when they are slower than real time on the computer), permitting more conditions to be studied in the same period of time.

(3) Models pose no safety hazards and do no damage when used to represent hazardous situations such as potential derailments.

Many mathematical models of rail vehicle dynamics have been developed, but these models have, in most cases, not been validated, and the work that has been performed to validate rail vehicle models has not been completely successful. Often, it is incomplete in that some data is looked at, occasionally parameters are modified, but the final steps of comparing the model outputs to independent data sets and defining the range of validity of the models are missing. These shortcomings are not necessarily oversights but generally are limitations of the available data and funding. In some cases, the nonlinearity of the dynamics is not well understood and causes the modeling to be invalid.

The inadequacy of test data has been a major contributor to problems in past validation attempts. Some test data sets do not include important, but difficult to measure, quantities such as wheel/rail forces and wheel/rail displacements. Oversights in test planning have been a common problem. In general (with the exception of models that require detailed wheel/rail measurements for validation), the problems have not been with the state of the art of testing or instrumentation technology, but with the omission of needed measurements or test conditions.

Given the difficulty of validating rail vehicle dynamics models, the potential benefits to be enjoyed from improved validation methods and test planning procedures are substantial. If validated models could be applied to predict performance with confidence, testing could be reduced and many questions about dynamic performance of rail vehicles could be answered more quickly, accurately, and inexpensively than they can now.

D-1.2 SCOPE AND PURPOSE OF THIS SECTION

This section is designed to provide guidance for those in the railroad and the railroad-related industries who are interested in validating vehicle dynamic models. This section has been written for engineering staff members who have a good understanding of vehicle dynamic performance and railroad equipment and testing, but it is not necessary to have a comprehensive mathematical background. The mathematical analysis and statistical aspects of model validation have therefore been de-emphasized and in some cases simplified. The emphasis here is more on intuitive insights and physical "feel" for vehicle performance. The procedures suggested in this section have been designed for practical use by industry, rather than being directed at the research community.

A principal purpose of this section is to "de-mystify" the model validation process and make it accessible to more potential users within the railroad industry. This includes not only the dynamics analyst, but also the instrumentation personnel, test engineers, and planning and operating organizations who must all interact with each other to integrate the testing and analysis activities in a model validation program. Previous model validation efforts have had limited success. The systematic, step-by-step approach presented here has been designed to avoid the mistakes of the past and to make it easier to validate models in the future. The procedures which are described, if followed in an orderly way, should lead to either a successful model validation or a determination that the model cannot be validated, without wasted effort.

The instructions for validating a model are arranged in a logical sequence which should be followed closely in practice. Because of the great diversity of the models and test data which could be used, it is impossible to reduce these instructions to a single universally applied format. It is still necessary for the analysts doing the validation work to make many important decisions using their own judgement. Wherever possible, examples have been inserted in the text to relate the instructions to some specific rail vehicle dynamics problems.

D-1.3 OVERVIEW OF REMAINING CHAPTERS

Chapter D-2 provides answers to some of the basic background questions about model validation, explaining what it is and why it is worth doing. That chapter also addresses the fundamental questions of model complexity and the relative economics of validating models or relying exclusively on testing to learn about rail vehicle performance. Chapter D-3 then covers the preliminary steps which should be followed in all model validation programs before thinking about doing the testing. These include defining the structure and purpose of the model to be validated and using those to select the validation criterion, steps which have not received enough attention in most previous validation programs.

Chapters D-4 and D-5 respectively cover the different steps which should be followed when using existing test data and when running a new test program for validating a model. The emphasis in Chapter D-4 is on determining whether the data are adequate for the desired validation, while Chapter D-5 concentrates on designing a test program which will produce the needed data. In each case, the recommended procedures include numerous runs of the model which is being validated. Each validation program will require the use of either the Chapter D-4 or the Chapter D-5 procedures, but not both.

Chapter D-6 describes the process of comparing the model predictions and test results to determine the validity of the model. This step, which applies to all validation attempts regardless of the source of the data, is what makes model validation "special", and is therefore covered in considerable detail. A critical feature of this comparison process is the adjustment, or modification, of the model to improve its realism and the re-checking of the adjusted model for all the tested conditions. The validation process is summarized in compact form in Chapter D-7.

In the interest of keeping the main text of this report from becoming too mathematical, equations have been restricted to use in tables and figures, and the statistical issues have been segregated in the Appendix D-A, where they are available for those who may be interested in them. Appendix D-B contains a glossary of some of the technical terms used here, and Appendix D-C is an example showing how the validation procedure can be applied to a simple linear frequency domain model.

D-2. ANSWERS TO BASIC QUESTIONS

D-2.1 WHAT IS A MATHEMATICAL MODEL?

An analytical model is an approximate mathematical representation of reality. A vehicle dynamic model represents the performance of a vehicle (or significant components of a vehicle) under certain operating conditions. Typically, different models are used to represent different aspects of vehicle performance. The fundamentally distinct aspects of vehicle performance are referred to as dynamic processes or performance issues (explained in Section 3.1). A model is normally designed to represent only one performance issue in order to avoid excessive complexity.

Models can be designed to represent vehicle dynamics in several different ways, referred to as analytical techniques (also explained in Section 3.1). The different analytical techniques offer trade-offs between accuracy and complexity. They are also suitable for revealing different characteristics of vehicle performance (such as frequency response using one analytical method and peak response amplitude using a different method).

Vehicle dynamic models can range from the very simple to the very complicated. The simplest models are generally those which describe performance trends based on one very specific physical process (such as axle and truck spacings or wheel rotations). These are describea more thoroughly in Section D-2.4. The most complicated models typically represent the coupling among the different modes of response of a complete vehicle using as many as a hundred degrees of freedom. These models are so diverse that the amount and type of effort required for validations will vary greatly from model to model.

Although a vehicle dynamic model may be executed on a digital computer, references to a model should not be interpreted as references to a computer program. Some models (the very simple ones) may not require the use of a computer at all. The same model could be implemented on a variety of different computers by a variety of different people, all using different computer programs. Validation of the model, as described in this section, is completely different from validation of these computer programs. Validation of a computer program involves some very specific computer programming tasks, such as debugging, transferring code from one computer to another, and solving machine specific problems. A validated analytical model could be implemented in a computer program which was not validated, or a model which has not been validated physically (the kind of validation treated in this section) could be implemented in a validated computer program. Useful results can only be obtained when both types of validation have been accomplished.

D-2.2 WHAT IS MODEL VALIDATION?

Validation of a model is the process of ensuring that the model offers a valid representation of reality. There is no absolute standard of validity which can be applied to all models all the time. Whether or not a model is considered valid depends on what it is expected to be used for. A model could, for example, be found valid for rough preliminary design but not for detailed final design. Likewise, a model could be valid for one vehicle or track design but not another; or it could be valid under one set of operating conditions (speed, track inputs, loading), but not another.

In the process of validating a model, one tries to show that the model is able to predict the results which actually occurred in tests. The model's predictions must be close enough to the test results to be useful for the model's intended purpose. The principal elements of the validation process are:

(1) Defining the validation criterion.- Based on the purpose the model is expected to serve, specify which model predictions are to be compared to the test data and how close the agreement should be for validation. For each validation, there is a single criterion to be satisfied, but if the model is to be used for a different purpose, a new validation could be needed, using a different criterion.

(2) Comparison of model and test results.- Run the model and tests for identical conditions (inputs, track, and vehicle characteristics) and compare the predicted and actual performance.

(3) Adjustment of model.- Change the model, as needed, to improve its ability to represent the test conditions.

These elements of validation are not the steps which must be followed in a validation exercise. Those steps are outlined and then discussed in detail in Chapters D-3 through D-6.

When the validation process has been concluded, one should be able to summarize the outcome by stating that the model under consideration was validated (or not validated) for:

(1) the stated purpose,

(2) the vehicle(s) or class of vehicle(s) used,

(3) the dynamic regime(s) considered; and

(4) the type or condition of the track used.

All-inclusive statements about model validation which do not specifically refer to these four qualifications are not useful. Once the model has been validated, it can be used in place of testing to predict vehicle performance under conditions satisfied by the above qualifications. It can not generally be considered validated for any other conditions.

D-2.3 WHY VALIDATE A MODEL?

The planning, testing, and data analysis required for model validation typically require substantial time, effort, and expense. This would not be worth doing unless there were a significant benefit attached to the validated model.

Mathematical models of vehicle dynamics are viable alternatives to testing for several reasons, most of which have strong economic impacts. Full-scale tests require the use of very costly vehicles, train crews, instrumentation, data processing equipment, and testing personnel (technicians), as well as track which either has to be specially constructed for testing or must be taken out of revenue-producing uses for a period. Changes to vehicle design require fabrication and assembly of the new equipment, which is much more costly and time-consuming than changing model parameter values. Models can also be run more rapidly than full-scale tests (even when they are slower than real time on the computer), permitting more conditions to be studied in the same period of time. Finally, models pose no safety hazards and do no damage when used to represent hazardous situations such as potential derailments. In each of these cases, models offer significant advantages relative to testing.

Models cannot be used to evaluate or predict vehicle performance with confidence unless they have been validated. An unvalidated model is little more than a hypothesis or a set of assumptions about vehicle performance, and should not be used to guide important decisions. Thus, the benefits of modeling for vehicle design and evaluation cannot be enjoyed until the model is validated.

The cost effectiveness of model validation depends on the trade-off between the effort required to validate the model and the additional testing which would otherwise be required. If the model is expected to be used for only a very limited set of conditions and very few runs, it may be faster and less costly to run tests only for the cases of immediate interest. However, if the model is expected to be used repeatedly, for a substantial variety of cases, the cost savings from eliminating extra test cases should be enough to justify the cost of the validation process. In general, the highest payoff should come from validating the most heavily used models.

D-2.4 AREN'T MODELS TOO COMPLICATED FOR ME TO USE?

Some vehicle dynamics models are in fact very complicated and difficult to use, but many others are not. The real "art" in modeling is to be able to choose or design the simplest possible model which will adequately do the job. The complexity of the model should thus depend on the complexity of the behavior one is trying to predict. If one is concerned about avoiding suspension and primary structural resonances in designing a railcar, for example, the models could be very simple and the validation procedure straightforward.

Although many models require the use of computer programs, not all do. The computer-oriented models are typically the time-domain simulations and quasi-linear frequency domain analyses. Many linear frequency domain and steady-state models can also be applied much more quickly and easily by relying on the computer rather than desk-top methods. However, the simplest frequency domain models, such as those used to identify basic resonance problems, can generally be worked entirely by hand. The input frequencies can be calculated directly from vehicle speed, axle and truck spacings and wheel circumference, while suspension natural frequencies can be simply determined from the mass and spring properties.

Models such as this are used widely without always being referred to as models. Indeed, it would be virtually impossible to design a new railcar without some elementary modeling. The need for validation and the applicability of the validation procedures to be described in Chapters D-3 through D-6 are as pertinent to these simple models as they are to the complicated full-vehicle and track models.

D-2.5 HOW MUCH IS IT GOING TO COST TO VALIDATE A MODEL?

The cost of validating a model can vary by as much as the cost of running a series of tests. Obviously, the cost will depend on the model being validated, the purpose it is intended to serve, the type of vehicle or consist needed, the dynamic regime(s) considered, and the difficulty of making the required measurements. Most of the cost elements associated with a model validation project should also be present in any well-conceived performance testing project. (By performance testing is meant a test program designed to investigate the performance of a specific vehicle under specific operating conditions, without regard for the applicability of the results for other vehicles or conditions.) However, performance testing is often conducted using a "brute force" approach, with very little advance planning. While that approach can sometimes yield the desired results when the testing is aimed at answering a very specific question of limited scope, it is not suitable for model validation.

D-2.5.1 Test Planning

For example, the test planning needed for model validation may require as much as 50% more effort than the planning for performance testing alone, in order to ensure that all performance regimes and combinations of regimes needed to characterize vehicle performance are included. Validation tests also require careful measurements of track geometry, which performance testing generally does not require.

D-2.5.2 Instrumentation

The on-board instrumentation requirements for model validation are typically more demanding than for other kinds of testing. Responses other than the primary outputs of interest must be measured to validate most models, requiring additional instrumentation and data channels. This could include measurements of wheel/rail forces and displacements for models which are influenced by lateral wheel/rail interactions, suspension deflection measurements for models which incorporate suspension effects, or a selection of body-mounted accelerometers for choosing the number of body-bending or torsion modes to include in the model. In each case, the additional instrumentation and data channels would increase the cost of the test program and data analysis. This cost increase is most pronounced when the wheel/ rail forces and displacements must be measured, owing to the difficulty of making those measurements accurately. In other cases, the cost increase could be insignificant.

D-2.5.3 Data Analysis

Model validation also requires some more thorough data analysis than performance testing does. The most important special data analysis consideration for model validation is the synchronizing of track geometry and vehicle dynamics measurements. Because most test programs are not set up for this type of analysis, it usually requires some special operations (to be described later). The track test section needs to be equipped with location markers which can be detected by automatic location detectors (ALDs) on the track geometry vehicle and the test vehicle. The on-board instrumentation and recording system needs an additional data channel for the ALD information, and the post-test processing must include interpolations to ensure that the ALD signals all remain properly synchronized. The costs of the ALD instrumentation are not generally significant relative to the costs of the other instrumentation required.

The data processing for alignment of ALD signals requires some skill and experience, as well as significant computer time. The first time it is done, it will be costlier than future runs, after experience has been established.

In particular, if the processing is only to be done a few times and the development of a computer program to automate the procedure cannot be justified, it will be necessary for a skilled analyst to intervene manually in the computer processing of all of the test data. This effort could add significantly to the cost of the data analysis, although the relative cost increase depends on the amount of other data processing required. For a test program which already requires substantial data reduction, the relative increase could be of the order of 20%. On the other hand, if only minimal data reduction were otherwise required, the synchronization process could double the data processing burden.

D-2.6 HOW MUCH MONEY CAN I SAVE BY VALIDATING A MODEL?

The cost of validating a model should be evaluated by comparison with the cost of obtaining comparable information from testing alone (if indeed it can be obtained only by testing). In each case, a thorough, well designed, and well planned test sequence must be assumed if data of comparable quality are to be obtained. The validation approach requires more advance planning and data analysis, and probably more instrumentation, but less direct testing expense (fewer test cases to be run). The tradeoffs among these cost elements will be different for every potential application, and should be evaluated on a case-by-case basis.

All of the cost-increasing aspects of model validation described in Section D-2.5 can be outweighed by the reduction in direct testing expense relative to pure performance testing. Each computer run of a vehicle dynamics model costs only a fraction as much as a full-scale test for the same conditions. The model validation tests should be designed to include just enough cases to reveal vehicle performance in the dynamic regimes of interest. This is not nearly as many test cases as one would need to completely demonstrate the vehicle performance trends and relationships within the same dynamic regimes. The validated model could be used to "fill in" the large number of other cases in place of this testing.

The cost advantage of the validated model is even more pronounced when it is necessary to evaluate the effects of changes in vehicle design. Testing of a variety of different designs involves not only the direct cost of the test runs (which are probably numerous), but also the substantial costs of fabricating the new design equipment and installing it on the vehicle. The ability of a model to represent the effects of vehicle design changes can be validated using a limited number of carefully chosen test cases. The validated model can then be used to predict performance under a wide variety of conditions which do not have to be tested. This can save a large number of test runs, as well as the time and expense of building and installing vehicle components of new design.

D-3. PRELIMINARY STEPS FOR ALL VALIDATION PROGRAMS

This chapter reviews the preliminary steps which should be covered before getting deeply involved in testing and data analysis for model validation. These preliminaries are essential if the validation exercise is to produce useful results.

The chapter begins with a review of the types of models which can be validated using the approach presented in this section. The critical issue of defining model purpose is then covered, followed by the equally important subject of selecting the validation criterion. The steps in the validation process which are covered in this chapter are shown schematically in Figure D-3.1. However, before discussing these steps, it is necessary to devote some attention to the types of models being validated.

D-3.1 TYPES OF MODELS

The validation procedures covered in this report are designed for use with rail vehicle dynamics models. These models are so diverse that it is often hard to generalize for all of them. Because of the wide range of purposes and requirements the models have been designed for, their complexity and level of detail vary widely. However, the dynamic processes represented by the models and the analytical (mathematical) techniques used to calculate predictions of vehicle behavior are so clearly distinct from each other that these can be useful for categorizing the models. Within each of the categories defined by dynamic process and analytical technique, the models can vary greatly in level of detail (e.g., portions of vehicles or number of vehicles described, degrees of freedom and nonlinearities included, etc.).

D-3.1.1 Analytical Techniques

The analytical techniques which are typically used to solve rail vehicle dynamics models are listed in Table D-3.1. Each analytical technique requires the use of a different mathematical solution method to calculate the responses of interest. Furthermore, the responses calculated using the different analytical techniques are fundamentally different from each other, so that

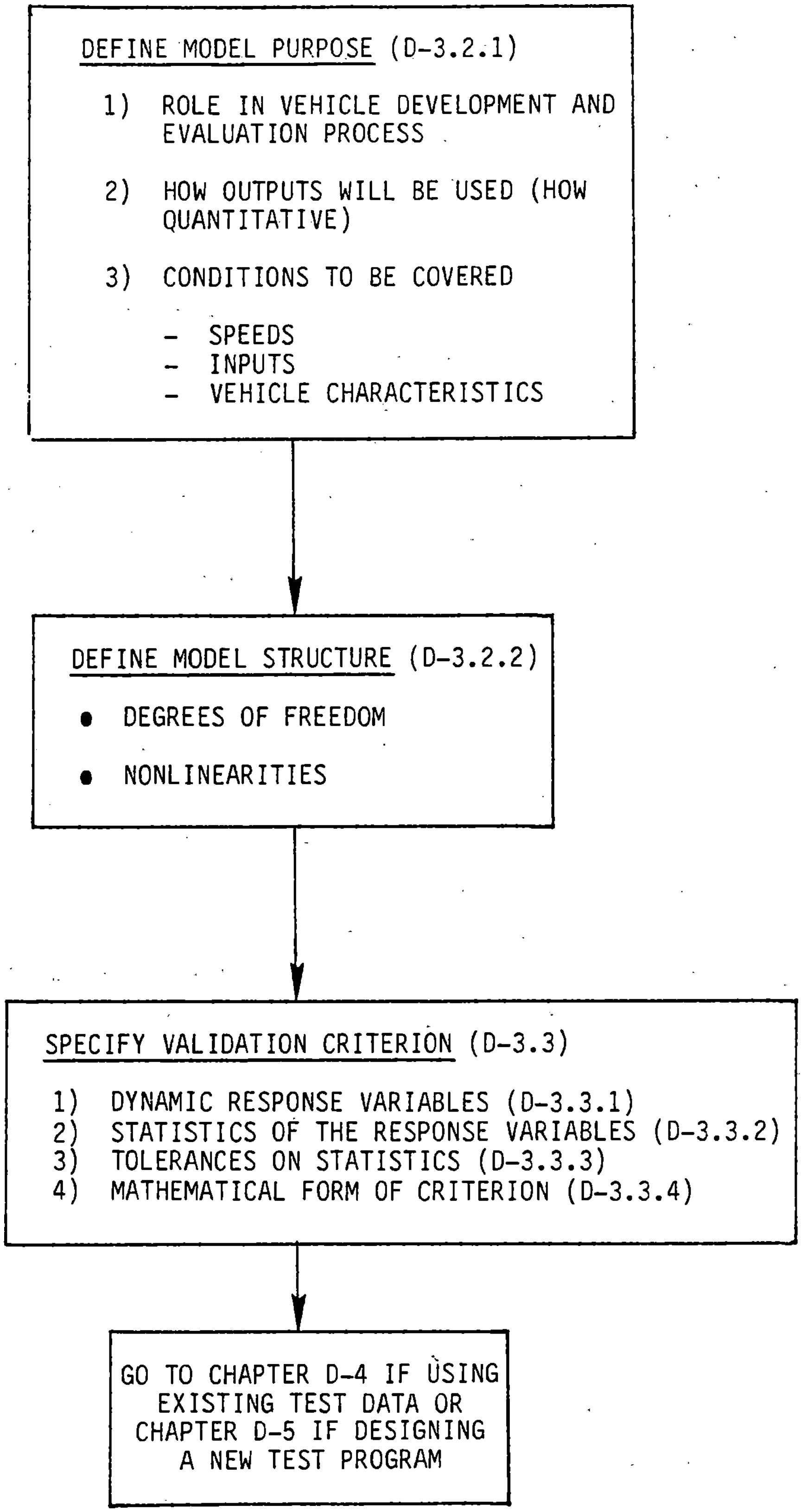


Figure D-3.1 Preliminary Steps In Model Validation Process (Chapter D-3)

Table D-3.1

Analytical Techniques Applied To Rail Vehicle Dynamics Models

• QUASI-STATIC (ALGEBRAIC)
• FREQUENCY DOMAIN - LINEAR - QUASI-LINEAR - SPECTRAL ANALYSIS - MODAL ANALYSIS
• TIME DOMAIN - LINEAR - NONLINEAR

different kinds of comparisons with test data are required to evaluate model validity.

Quasi-Static (Algebraic) Solutions

The quasi-static (or algebraic) analytical technique is applied to steady-state models which are designed to predict the dynamic equilibrium performance of vehicles. These models are the simplest to validate because each output quantity is a single number, which can be compared with a single number describing the same performance experienced in testing. An example of this type is a model which calculates equilibrium forces and displacements in steady curving (with constant speed and curve radius).

Frequency Domain Solutions

Several types of frequency domain analysis techniques can be applied to rail vecnicle dynamics models. These methods are based on linear assumptions, but can be adapted for use on nonlinear systems by using quasilinearization techniques such as describing functions [1]. Eigenanalyses are used to determine natural frequencies, damping ratios and mode shapes of vehicle response. Although eigenanalyses can be very efficiently calculated, they are difficult to compare with test data because testing cannot directly produce evidence of the natural frequency. However, eigenanalyses can be used to predict the critical speed and damping of each response mode for the onset of hunting, and that critical speed can be compared with the speeds at which hunting becomes apparent in tests. More commonly applied frequency domain analyses involve the use of transfer functions to calculate vehicle response spectra and root mean square (rms) values. These are typically used in models of ride quality or lading response, for example, in which the outputs describe the vibration environment on-board a vehicle.

Time Domain Solutions

The majority of the rail vehicle dynamics models which have been developed use time-domain solution techniques. For linear systems the solutions to the model equations can be computed using linear algebra (via the state transition matrix). For general linear or nonlinear systems, the system differential equations can be solved by a variety of numerical integration techniques. In either case, the model can produce a sampled time history of each response variable and any auxiliary variables the modeler may choose.

D-3.1.2 Dynamic Processes or Performance Issues

Table D-3.2 lists the dynamic processes which are typically represented by rail vehicle dynamic models. For each process, different variables are the significant indicators of vehicle performance and of model validation. The validation procedures and criteria must reflect this diversity. For any particular model validation effort, the choice of which variables to use in establishing validation must be made on the basis of an intimate understanding of both the dynamic process being modeled and the individual candidate model.

The dynamic processes listed in Table D-3.2 are distinct and not interchangeable. Most models are designed to represent only one of these processes, with the exception of the lateral forced response models, which sometimes also include vertical forced responses. Combining several processes in one model is generally more complicated than designing separate models for the separate processes. Separate models can be designed to focus on the most important phenomena for each dynamic process, while ignoring the less important. These separate models are likely to have very little in common with each other, even when designed to represent the same vehicle.

D-3.1.3 Combination of Analytical Techniques and Performance Issues

The cross-categorization of the five analytical techniques and 10 performance issues which have been considered here is shown in Table D-3.3. For a given performance issue, there are usually a number of analytical techniques that can reasonably be applied, depending upon the type and level of information desired. The reasonable combinations are marked in Table D-3.3, in which each "X" can be considered to represent a model category.

It is obviously impractical to specify separate validation procedures for the nearly 30 categories of models indicated in Table D-3.3. The dimensions of this problem become even worse when one considers that each model category can include many different models, all having different degrees of freedom and

Table D-3.2

Typical Rail Vehicle Dynamic Processes (Performance Issues)

SINGLE VEHICLE:

- TWIST AND ROLL
- HUNTING (LATERAL STABILITY)
- SPIRAL NEGOTIATION
- STEADY-STATE CURVING
- DYNAMIC CURVING
- PITCH AND BOUNCE (VERTICAL FORCED RESPONSE)
- YAW AND SWAY (LATERAL FORCED RESPONSE)

MULTIPLE VEHICLES:

- STEADY-STATE BUFF AND DRAFT
- LONGITUDINAL TRAIN ACTION
- LONGITUDINAL IMPACT

Table D-3.3

Cross-Categorization Of Rail Vehicle Dynamics Models

PERFORMANCE ISSUES	ANALYTICAL TECHNIQUES				
	FREQUENCY DOMAIN			TIME DOMAIN	
	QUASI-STATIC	LINEAR	QUASI-LINEAR	LINEAR	NONLINEAR
<u>SINGLE VEHICLES</u>:					
TWIST & ROLL			X		X
HUNTING (LATERAL STABILITY)		X	X		X
SPIRAL NEGOTIATION					X
STEADY-STATE CURVING	X		X		X
DYNAMIC CURVING			X		X
PITCH AND BOUNCE (VERTICAL FORCED RESPONSE)		X	X	X	X
YAW AND SWAY (LATERAL FORCED RESPONSE)		X	X	X	X
<u>MULTIPLE VEHICLES</u>:					
STEADY-STATE BUFF AND DRAFT	X				
LONGITUDINAL TRAIN ACTION			X	X	X
LONGITUDINAL IMPACT			X	X	X

nonlinear elements, and representing different types of vehicles (e.g., locomotives or hopper cars, radial or rigid frame trucks, etc.). The tremendous variety of possible models makes it impossible to design one universal algorithmic "black box" validation procedure which can be applied by an analyst who may not thoroughly understand vehicle dynamics, mathematical modeling and data analysis procedures.

Before beginning the validation process, it is necessary to review the model thoroughly enough to understand which dynamic process it represents and which analytical method it uses. The dynamic process will determine which aspects of vehicle performance are the primary outputs and which secondary. The analytical method will determine which measures of performance (such as maximum values, rms, spectral densities, natural frequencies, etc.) are appropriate to consider for the validation criterion (Section D-3.3).

D-3.1.4 Nonlinearities

If rail vehicles displayed linear performance they would be much easier to model and analyze than they are in fact. It is the nonlinear aspects of their behavior which cause most of their dynamic response problems, as well as complicating their models and the validation of those models. In general, these nonlinearities are physical processes whose performance changes qualitatively as the level of the input changes. The presence of any nonlinear element or relationship in a system makes the entire system nonlinear, requiring special care in analysis.

The details of nonlinear system analysis are much too complicated to explain thoroughly here. All that can be recommended, in general, is that one use great caution when analyzing the performance of a system which contains any nonlinearities, particularly the "stronger" nonlinearities. The "strength" of a nonlinear relationship is a rough measure of how much different it is from a linear relationship.

Figure D-3.2 shows several different kinds of nonlinearity which are prevalent in rail vehicles, in order of increasing nonlinearity. Any rail vehicle model which includes any of these elements is therefore nonlinear and cannot be treated using simplified linear analysis methods.

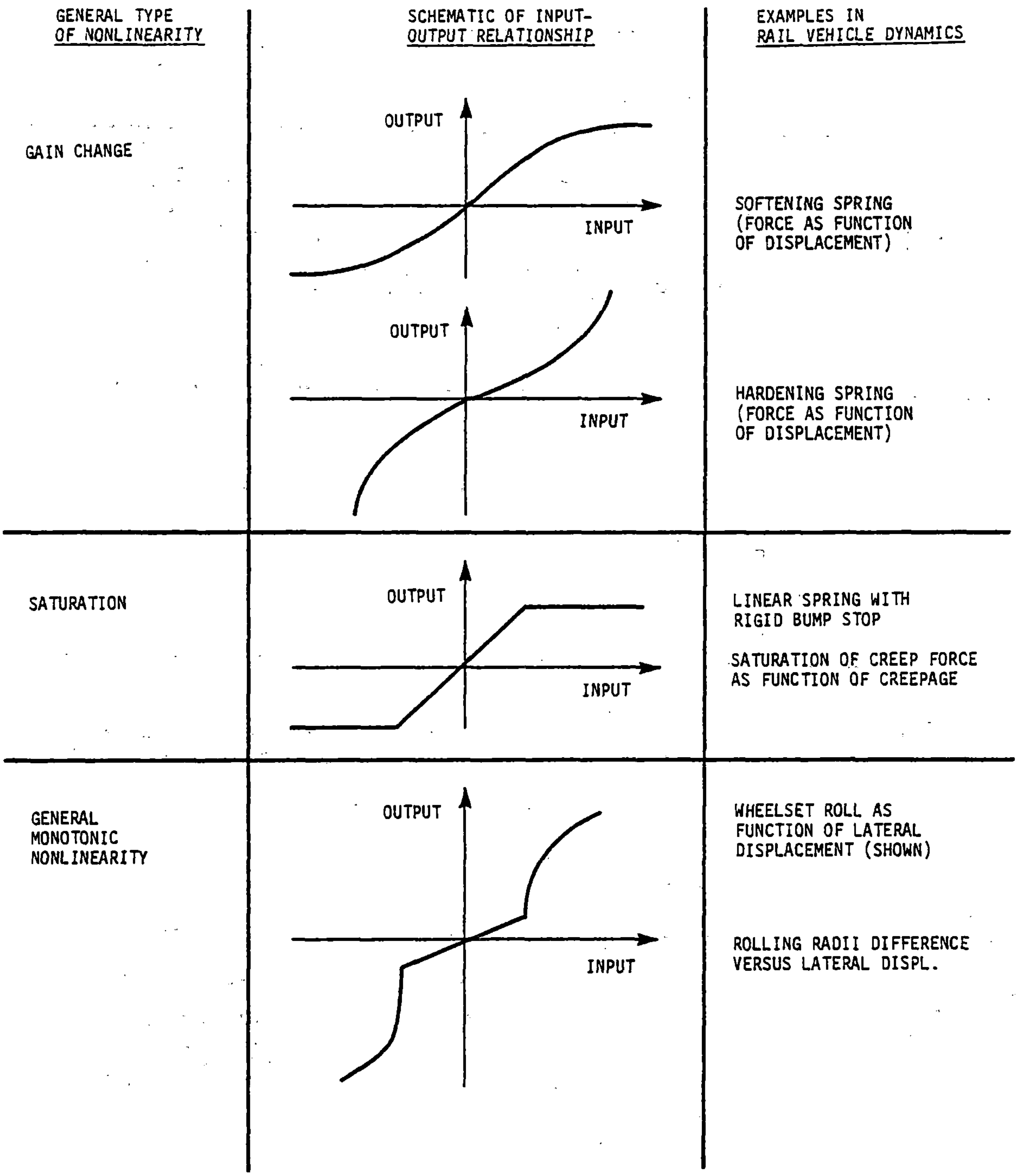


Figure D-3.2 Typical Rail Vehicle Nonlinearities

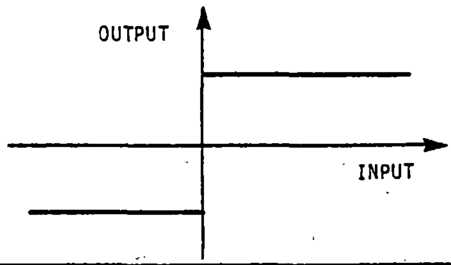
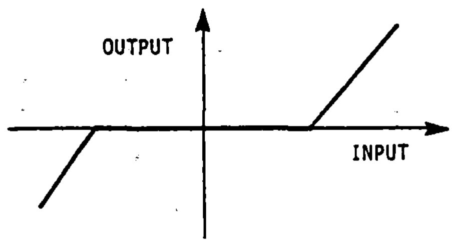
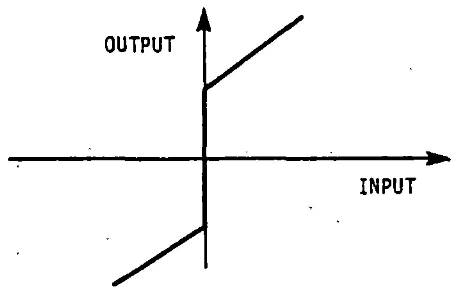
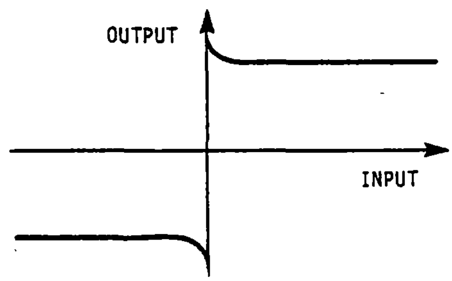
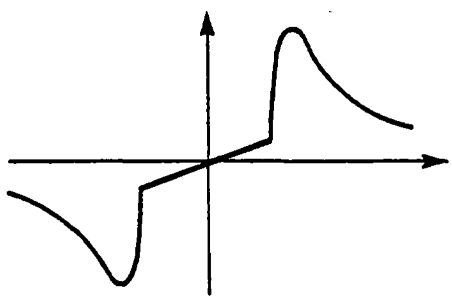
IDEAL RELAY		DRY (COULOMB) FRICTION SUSPENSION DAMPERS (RESISTANCE FORCE AS FUNCTION OF VELOCITY)
DEADBAND (THRESHOLD)		WHEELSET FLANGING (SIMPLE MODEL FOR FORCE AS FUNCTION OF LATERAL DISPLACEMENT) SIDEBEARING CONTACT
PRELOAD		SUSPENSION SPRING PRELOADS
STICK-SLIP (STICTION)		CENTERPLATE FRICTION (TORQUE AS FUNCTION OF ANGULAR VELOCITY)
GENERAL NONMONOTONIC NONLINEARITY		DIFFERENCE BETWEEN LEFT AND RIGHT WHEEL/RAIL CONTACT ANGLES AS FUNCTION OF WHEELSET LATERAL DISPLACEMENT

Figure D-3.2 (Continued)

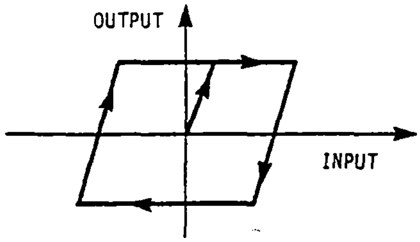
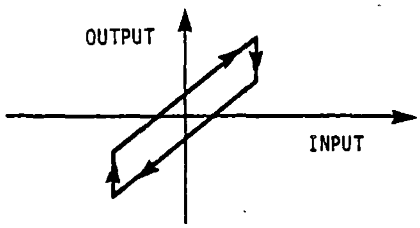
BACKLASH (HYSTERESIS) FRICTION-CONTROLLED (SERIES COMBINATION OF SPRING AND DRY FRICTION)		FREIGHT TRUCK WHEELSET YAW SUSPENSION-YAW MOMENT AS A FUNCTION OF YAW ANGLE
INERTIA-CONTROLLED (PARALLEL COMBINATION OF SPRING AND DRY FRICTION)		FREIGHT TRUCK WHEELSET LATERAL SUSPENSION — LATERAL FORCE AS FUNCTION OF LATERAL DISPLACEMENT

Figure D-3.2 (Continued)

D-3.2 MODEL PURPOSE AND STRUCTURE

The most important thing to bear in mind throughout the validation process is the purpose which the validated model is expected to serve. The model is, after all, a tool to be used in the design and/or evaluation of vehicle performance, rather than being an end product itself. The validation procedure has to be tailored to the anticipated use of the model by choosing a unique validation criterion which best indicates the model's suitability or unsuitability for the intended purpose. The validation criterion (to be discussed more thoroughly in the next section) is the set of measures (and their tolerances) by which the model predictions and test results are compared in the validation process.

The validation process does not produce the result that the model is either "validated" or "not validated" across the board. Rather, the assessment of model validity must be made in terms of model purpose. For example, the same model could be found valid for one purpose but not for another, more demanding, purpose. The distinction between the two cases is in the validation criterion, which may include either more or different types of output variables and tighter tolerances for the more demanding model purpose.

D-3.2.1 Definition of Model Purpose

The first step which should be taken in validating a model is defining the purpose(s) the model is going to serve. This definition of purpose is a three-level process which should begin by considering the model's general role in the vehicle development and evaluation process, such as:

- preliminary conceptual design
- detailed final design
- comparison of alternate designs
- predicting performance under untried operating conditions
- test planning
- evaluating compliance with performance specifications
- defining limits for safe operation.

The requirements for agreement between model predictions and test data will be different for these differing purposes.

The second level of detail in the definition of model purpose is needed to set tolerances in the validation criterion. This is the definition of how the model outputs will be used:

- establishing performance trends (qualitative)
- ranking alternate designs (relative quantitative)
- quantifying performance levels (absolute quantitative)

For the first two purposes, the model must be able to represent the proper performance trends, while for the third it must also be able to predict correctly the level of performance on an absolute scale, which is considerably more difficult.

The third level of detail in the definition of model purpose is the description of the range of conditions for which the model is expected to be used. These include:

- speed range
- inputs
    - track geometry and curvature
    - track roughness and compliance
    - external parameters (wind, temperature, creep coefficient, etc.)
- vehicle characteristics
    - vehicle type(s)(locomotive, flat car, hopper, etc.)
    - range of vehicle parameter values (masses, stiffnesses, dimensions)

These conditions are useful for designing the validation test program and for specifying the conditions for which it will be necessary to determine whether the model is valid. Of course, the dynamic process represented by the model must also be appropriate for vehicle behavior under the specified conditions.

Throughout the remainder of the validation process, it will be helpful to keep these specifications of model purpose in mind whenever questions of judgement arise. The model validation is not an end in itself, but rather the validated model is a tool designed for specific purposes.

D-3.2.2 Definition of Model Structure

Model structure is the specific selection of degrees of freedom and nonlinearities which distinguishes one model from another. If any of the degrees of freedom or the forms of the nonlinearities are modified in the course of the validation process, the model structure should be considered different. On the other hand, changes in the values of model parameters (masses, stiffnesses, dimensions, etc.) are not changes in model structure. Validation of a model for use on a general class of vehicles (e.g., six-axle locomotives) refers to the validation of the model structure and the method used to select the model's parameter values. In contrast, validation of a model for a specific vehicle (e.g., an E-8 locomotive) involves both the model structure and the specific parameter values used to describe that vehicle. The validation of the general model structure is a more difficult and costly process, since it requires, in effect, the validation of the model structure for several different vehicles.

D-3.3 SPECIFYING THE VALIDATION CRITERION

The validation criterion is a uniquely defined set of measures used to decide whether the agreement between model predictions and test results is close enough that the model can be considered validated. This criterion must be selected before any of the later steps in the validation process can be performed. The selection of the criterion must be based on the model purpose, which was already defined, and on the analyst's understanding of the dynamics of the vehicle being tested. Specific validation criteria for all models which one may wish to validate cannot be provided here, but the thought process which should be used to select the criterion for each validation will be described. It should be emphasized here that a single criterion is to be applied to the validation of a model for a specific purpose. Whether or not this criterion is completely satisfied will determine whether the model is validated. If another validation is to be conducted (for a different model purpose, for example), a different criterion will have to be selected.

The selection of the validation criterion involves four elements, which must proceed in sequence:

(1) which dynamic response variables to compare

(2) which measures (or statistics) of each variable to examine

(3) what tolerance level to apply to each

(4) what mathematical form to use to combine the individual measures

In the following four sections guidance will be offered for selecting these elements of the validation criterion, independent of any limitations posed by the need to use an existing set of test data. Sometimes the test data which are available for use in the validation program are not compatible with the preferred variables and statistics. This may limit the scope of the model-validation which can be done, as explained in Chapter D-4. For example, tests on perturbed track which do not produce statistically stationary results (i.e., results which have the same statistical properties for all time intervals) will not be suitable to use with most of the frequency domain statistics.

D-3.3.1 Dynamic Response Variables

The choice of dynamic response variables should be based on the model purpose and structure. These variables should be those which are most important for the ultimate use of the model (the primary outputs) and those intermediate outputs which are expected to be most revealing of model deficiencies. The primary outputs are normally easy to specify as those which are needed to satisfy the purpose which the model is intended to serve. These must be included to ensure that the model is at the very least predicting the essential responses correctly.

The choice of which intermediate outputs to include in the validation criterion is somewhat more subtle, depending heavily on the analyst's understanding of the model and of the physics of the vehicle being modeled and tested. This choice should be designed to test the internal workings of the model, making sure that the important response modes of the vehicle which do not directly produce the primary outputs are still modeled properly. For example, the validation criterion for a forced vertical response (ride quality) model of a vehicle should include not only the body accelerations (primary outputs) but also the suspension deflections (intermediate outputs). If the important intermediate outputs are not well represented by the model, this indicates that the model has a structural defect which will probably lead to erroneous predictions of the primary outputs under some other conditions. The intermediate output quantities which are important for validating one model may be completely different from those which should be used for validating another model, making it difficult to specify general validation criteria.

Table D-3.4 shows some examples of the primary and intermediate response variables which could be used in the validation criteria for models of different dynamic processes. Any individual model may only require the use of a subset of these variables, and some models will require the use of variables which do not appear in the table. The table is included here to stimulate thought about the types of variables which should generally be included in validation criteria, and is not an all encompassing tabulation of the variables to choose from.

D-3.3.2 Statistics of the Response Variables

The dynamic response variables of a rail vehicle can be characterized in many different ways, corresponding to different statistics. The selection of these statistics for each variable to use in the validation criterion has a very important influence on the significance of the validation (some being easier, and others harder, to match). In particular, the statistics which are chosen should be appropriate for the analytical method of the model being validated. Examples of the statistics which could be used for models based on the three major classes of analytical techniques are listed in Table D-3.5, using the definitions in Table D-3.6.

The definitions of the statistics must be handled very carefully, because many serious errors have been made in this area in the past. This is particularly true of the frequency domain measures, which are based on Fourier Transforms of the response time history measures. The definitions must be consistent to avoid common errors by factors of 2 and 2π which often result from improper definitions of frequency (Hertz or cycles per second, f, and radians per second, ω), improper mixing of double sided spectra, $S_{xx}(f)$, with single-sided spectra, $G_{xx}(f)$, and improper normalization of the Fourier

Table D-3.4

Examples Of Candidate Response Variables To Include In The Validation Criterion (Not An All-Inclusive Listing)

PERFORMANCE ISSUE	PRIMARY RESPONSE VARIABLES	INTERMEDIATE RESPONSE VARIABLES
TWIST AND ROLL	BODY ROLL ANGLE WHEEL LIFT SUSPENSION DEFLECTIONS	BODY ACCELERATIONS WHEEL/RAIL FORCES NATURAL FREQUENCIES AND DAMPING RATIOS
HUNTING (LATERAL STABILITY)	CRITICAL SPEED TRUCK LATERAL AND YAW DISPLACEMENTS DAMPING RATIOS OF RESPONSE MODES	WHEEL/RAIL CONTACT ANGLES AND FORCES SUSPENSION DEFLECTIONS BODY ACCELERATIONS
SPIRAL NEGOTIATION	LATERAL WHEEL-RAIL FORCES WHEEL-RAIL DISPLACEMENTS	SUSPENSION DEFLECTIONS BODY ACCELERATIONS
STEADY-STATE CURVING	WHEEL-RAIL DISPLACEMENTS AND FORCES SUSPENSION DEFLECTIONS	BODY ROLL ANGLE TRUCK DISPLACEMENTS
DYNAMIC CURVING	CRITICAL SPEED BODY ACCELERATIONS TRUCK LATERAL AND YAW DISPLACEMENTS WHEEL-RAIL FORCES	SUSPENSION DEFLECTIONS
PITCH AND BOUNCE (VERTICAL FORCED RESPONSE)	BODY VERTICAL AND PITCH ACCELERATIONS	SUSPENSION DEFLECTIONS TRUCK ACCELERATIONS NATURAL FREQUENCIES AND DAMPING RATIOS
YAW AND SWAY (LATERAL FORCED RESPONSE)	WHEEL-RAIL DISPLACEMENTS WHEEL-RAIL FORCES BODY YAW AND LATERAL ACCELERATIONS	SUSPENSION DEFLECTIONS TRUCK ACCELERATIONS NATURAL FREQUENCIES AND DAMPING RATIOS
STEADY-STATE BUFF AND DRAFT	COUPLER LATERAL FORCES	DRAFT GEAR DEFLECTIONS AND ANGLES BUFF AND DRAFT FORCES
LONGITUDINAL TRAIN ACTION	BUFF AND DRAFT FORCES TIMES OF BUFF AND DRAFT TRANSITIONS L/V FORCE RATIOS	BODY ACCELERATIONS COUPLER ANGLES
LONGITUDINAL IMPACT	BUFF AND DRAFT FORCES DRAFT GEAR DISPLACEMENTS	BODY ACCELERATIONS SUSPENSION DEFLECTIONS

Table D-3.5

Examples Of Statistics Of Response Variables Which Can Be Used In The Validation Criterion

ANALYTICAL METHOD	STATISTICS	DEFINITIONS (VARIABLES DEFINED IN TABLE D-3.6)
QUASI-STATIC (ALGEBRAIC)	STEADY-STATE VALUE	
TIME DOMAIN	MEAN VALUE	$x = \frac{1}{N} \sum_{i=1}^{N} x(i\Delta t)$
	VARIANCE	$\sigma_x^2 = \frac{1}{N} \sum_{i=1}^{N} (x(i\Delta t)-\bar{x})^2$
	COEFFICIENT OF VARIATION	$\epsilon = \sigma_x/\bar{x}$
	PEAK AMPLITUDE	A_{PEAK}
	EXCEEDANCES	A_{95}, A_{T40}, etc.
	FIT ERROR	$\frac{1}{N} \sum_{i=1}^{N} [x(i\Delta t) - \hat{x}(i\Delta t)]^2$
	AUTOCORRELATION	$R_{xx}(\tau) = \frac{1}{N-r} \sum_{i=1}^{N-r} [x(i\Delta t)x(i\Delta t + \tau)]$, $r=\tau/\Delta t$
	CROSS-CORRELATION	$R_{xy}(\tau) = \frac{1}{N-r} \sum_{i=1}^{N-r} [x(i\Delta t)y(i\Delta t + \tau)]$, $r = \tau/\Delta t$
	TIME LAGS BETWEEN EVENTS	$t_{PEAK} - \hat{t}_{PEAK}$
	BIAS ERRORS	$B = x_{STEADY\ STATE} - x_{STEADY\ STATE}$
	RESPONSE WAVEFORM SHAPES	"EYEBALL" COMPARISONS AND FIT ERROR

Table D-3.5 (Continued)

ANALYTICAL METHOD	STATISTICS	DEFINITIONS (VARIABLES DEFINED IN TABLE D-3.6)
FREQUENCY DOMAIN	POWER SPECTRAL DENSITY	$S_{xx}(f) = \int_{-\infty}^{\infty} R_{xx}(\tau) e^{-j2\pi f\tau} d\tau$ (FOURIER TRANSFORM OF AUTOCORRELATION)
	CROSS SPECTRAL DENSITY	$S_{xy}(f) = \int_{-\infty}^{\infty} R_{xy}(\tau) e^{-j2\pi f\tau} d\tau$ (FOURIER TRANSFORM OF CROSS-CORRELATION)
	COHERENCE FUNCTION	$\gamma^2_{xy}(f) = \frac{\lvert S_{xy}(f)\rvert^2}{S_{xx}(f)\, S_{yy}(f)}$
	FREQUENCY RESPONSE FUNCTION	$H_y(f) = \frac{S_{xy}(f)}{S_{xx}(f)}$
	NATURAL FREQUENCY OF RESPONSE MODE i	$f_{ni} = (a_i^2 + b_i^2)^{1/2} / 2\pi$ (FROM EIGENVALUE ANALYSIS AND INSPECTION OF FREQUENCY RESPONSE FUNCTION)
	DAMPING RATIO OF RESPONSE MODE i	$\zeta_i = a_i/(a_i^2 + b_i^2)^{1/2}$ (FROM EIGENVALUE ANALYSIS
	PHASE RELATIONSHIPS OF RESPONSE MODES i, j	ϕ_{ij} (FROM EIGENVALUE ANALYSIS)
	RELATIVE AMPLITUDES OF RESPONSE MODES i and j	A_i/A_j
	MEAN VALUE	$\bar{x} = S_{xx}(0)$
	MEAN SQUARE	$\psi_x^2 = \int_{-\infty}^{\infty} S_{xx}(f)dt = \sum_{i=-M/2}^{M/2} S_{xx}(i\Delta f)$
	ROOT-MEAN-SQUARE (RMS)	ψ_x
	VARIANCE	$\sigma_x^2 = \psi_x^2 - \bar{x}^2$
	FREQUENCY-WEIGHTED RMS	$\left[\int_{-\infty}^{\infty} S_{xx}(f)W(f)df\right]^{1/2} = \left[\sum_{i=-M/2}^{M/2} S_{xx}(i\Delta f)W(i\Delta f)\right]^{1/2}$

Table D-3.6

Definitions Of Response Variable Statistics Terminology

TIME DOMAIN	
$x(i\Delta t)$ $y(i\Delta t)$	RESPONSE TIME HISTORIES OF x and y, at time interval i
$\hat{x}(i\Delta t)$	ESTIMATED (MODELED) TIME HISTORY OF x
$\bar{x}$	SAMPLE MEAN VALUE OF x
B	BIAS ERROR BETWEEN x and $\hat{x}$
τ	TIME DISPLACEMENT (LAG)
A_{PEAK}	PEAK AMPLITUDE FOR A TIME HISTORY EVENT
t_{PEAK}	TIME AT WHICH PEAK OCCURS
A_{95}	AMPLITUDE EXCEEDED 5% OF THE TIME (95% ILE)
A_{T40}	AMPLITUDE EXCEEDED FOR DURATION OF 40ms DURING PEAK EXCURSION
$R_{xx}(\tau)$	AUTOCORRELATION OF x
$R_{xy}(\tau)$	CROSS-CORRELATION OF x and y
Δt	TIME INTERVALS BETWEEN SAMPLES
N	NUMBER OF SAMPLES IN TIME HISTORY RECORD

Table D-3.6 (Continued)

FREQUENCY DOMAIN	
a_i	REAL PART OF EIGENVALUE OF MODE i
b_i	IMAGINARY PART OF EIGENVALUE OF MODE i
$X(f)$	FOURIER TRANSFORM OF x(t)
f	FREQUENCY (CYCLES PER SECOND)
Δf	FREQUENCY INTERVALS FOR SPECTRAL ESTIMATES
$S_{xx}(f)$	POWER SPECTRAL DENSITY FUNCTION OF x
$S_{xy}(f)$	CROSS SPECTRAL DENSITY FUNCTION OF x and y
σ_x	STANDARD DEVIATION OF X
σ_x^2	VARIANCE OF x
ψ_x^2	MEAN-SQUARE VALUE OF x
ψ_x	RMS VALUE OF x
$W(f)$	FREQUENCY WEIGHTING FUNCTION
$\gamma_{xy}^2(f)$	COHERENCE FUNCTION OF x AND y
$H(f)$	FREQUENCY RESPONSE FUNCTION
A_i	AMPLITUDE OF MODE i
f_{ni}	NATURAL FREQUENCY OF MODE i
ζ_i	DAMPING RATIO OF MODE i
M	NUMBER OF FREQUENCY SAMPLES IN TWO-SIDED SPECTRUM (INCLUDING BOTH NEGATIVE AND POSITIVE FREQUENCIES)

Transform (2π factor). These points are highly technical, but they can seriously influence the validation process. For guidance on how to handle these data analysis issues, it would be best to consult an authoritative text on the subject, such as Ref. [2].

As Table D-3.5 shows, some of the basic statistical measures such as the mean and variance and the statistics derived from them (standard deviation, mean square and rms) can be computed in either the frequency or time domain, as appropriate. The definitions of these and other statistics in Table D-3.5 are not the most theoretically precise versions available, but have been defined in such a way as to show how they should be computed digitally from the sampled data which would be available from a validation test or simulation model run.

Beginning with the listing of time domain statistics in Table D-3.5, the mean is simply the average of all the samples, while the variance describes the "spread" of those samples around the mean value. The coefficient of variation is the standard deviation normalized by the mean, a measure of the "shape" of the distribution of the sample results. The fit error is the sum of the squares of the differences between the model and test results at each time step. It is best used for comparing the "fit" of several different models to the test data, to help choose the "best" model, once all time-scale (phase) errors between the model and test results are eliminated. The correlation measures are not single numbers, but are mathematical functions which must be calculated from the data for each value of τ (time displacement). These functions are mainly used for deriving the spectral density functions in the frequency domain, rather than being used directly for assessing model validity.

Bias errors are generally the first ones which should be eliminated, because they can produce serious mistakes in the calculations for the other statistics. Biases are generally isolated by examining vehicle performance in the steady state (unperturbed), where it is often easy to estimate from the basic physics what values many of the response variables should have. Apparent inconsistencies among the values for several variables are also good indicators of biases which need to be removed (such as mean value of total vertical forces between wheels and rails not equalling vehicle weight in steady state, or total truck lateral forces on tangent track not summing to zero).

The sources of the biases are often in the selection of zero scale points for instrumentation which do not correspond to the zero points defined in models. This could include examples such as force measurements which do not have vehicle weights or spring pre-loads deducted, or displacement measurements which start from arbitrary zero settings. In each case, manual corrections must be made by the analyst to ensure that the test data and model predictions are defined from the same zero points.

The frequency domain statistics of Table D-3.5 are mostly defined in terms of the spectral densities. The spectral densities are not directly measured statistics, but are calculated as the Fourier Transforms of the correlation functions which were defined in the time domain section of the table. The integral equations which are shown for the spectral densities are the formal definitions of the Fourier Transforms, but do not correspond to how they would be calculated in practice. These transforms are commonly performed on sampled test data using "Fast Fourier Transform" (FFT) computer programs which are available as standard software packages, or using spectrum analyzer equipment. In either case, the functions $S_{xx}(f)$ and $S_{xy}(f)$ will be defined for M different values of frequency, f. The power spectral density is a real-valued function (magnitude only), but the cross-spectral density is a complex function, including both magnitude and phase. When working with the Fourier analysis of data, there are several potential sources of error, which can only be avoided by careful application of the proper procedures, as described in sources such as Reference [1]. Fourier analysis of test data should only be attempted after the concepts of "data windows," "leakage," "aliasing," and proper selection of sample sizes and intervals are thoroughly understood.

Once the spectral density functions have been calculated properly, Table D-3.5 shows how to find the coherence and frequency response functions, as well as the mean, mean-square, variance and rms values. All of these measures are amplitude only, except for the frequency response function, which includes both amplitude (transmissibility) and phase. The natural frequencies, damping ratios, and phase relationships of the response modes can be found from eigenvalue analyses. The amplitudes, as well as the natural frequencies and damping ratios, of the modes can be estimated from the frequency response functions.

Some of the frequency domain statistics discussed here are derived using the assumption that the system (vehicle) which is being modeled and tested is linear. If the vehicle has significant nonlinear performance characteristics (as most rail vehicles do), these statistics can lose much of their meaning. In particular, the frequency response function can only be derived from the input and output power spectral densities and cross-spectral density for a linear system. The modal response statistics (natural frequencies, damping ratios, amplitudes and phases) are similarly only applicable to linear systems. For nonlinear systems, these statistics are amplitude-dependent and cannot be used to provide a general description of system dynamics. Even separate descriptions of vehicle dynamics for different amplitudes (i.e., linearizing at specific operating conditions) may fail to show some very important nonlinear effects such as limit cycles (for example, in hunting), jump resonances (in twist and roll), backlash and hysteresis. If there is any reason to suspect that a vehicle's response is nonlinear, the linear statistics should not be used until the nonlinear effects are demonstrated to be insignificant.

The choice of statistical measures to use in the validation criterion is simplest for the quasi-static models, because the outputs of these models are single numbers for each dynamic response variable, with no frequency or time domain complications. The frequency domain models should be validated using frequency domain measures, while time domain models can use either time or frequency domain measures (the latter after Fourier transforming the time domain response). Different measures can be applied to different dynamic response variables of the same model (e.g., rms acceleration and peak amplitude of displacement) and multiple measures can be applied to one dynamic response variable (e.g. both rms and peak acceleration).

The selection of which statistic to apply to each dynamic response variable should be based on the purpose of the model and the importance of each statistic for the physical processes at work. In some cases, the choice is obvious once the response variable is chosen. Such an example is the use of the damping ratio of the least damped mode for frequency domain models to predict the onset of hunting (lateral stability). On the other hand, validation of a ride quality (forced response) model could include use of any of the following statistics of any of the body accelerations: mean, rms, frequency-weighted rms, spectral density, and the natural frequencies and damping ratios of the principal body modes. Unfortunately, the ride quality example is more typical of most models, in that the choice of the most appropriate statistics is not obvious. Indeed, there are no general rules which can be applied to choose the statistics to use for the general model categories (dynamic processes and analytical methods).

The choice of statistics must be made separately for each individual model which is to be validated, keeping in mind the purpose of the model. If the model is to be used to estimate maximum forces or deflections, then certainly the maximum value statistics should be used in the validation criterion. Similarly, if the model is to be used to predict passenger ride quality, the frequency weighted rms or the spectral density should be included, and if the model is intended to represent forced responses over a wide range of input conditions the frequency response functions should appear in the validation criterion. The most important factor to keep in mind when choosing the statistics is to make sure that they are the ones which will most clearly show whether or not the model is valid for its intended purpose.

D-3.3.3 Tolerances

Once the dynamic response variables and their statistics have been selected, it is necessary to decide how closely the test results and model predictions must agree with each other for the model to be considered validated. These tolerances must be tight enough to guarantee that the model will be able to serve the desired purpose, while at the same time not being so stringent that they cannot be satisfied realistically. Definition of these tolerances can be reasonably straightforward for the statistics which are described by a single number (mean, rms, maximum, etc.), but can become more complicated for the statistics which are described by series of numbers (spectral densities, correlations, frequency response functions, etc.). These tolerances are independent of instrumentation system tolerances, which are considered in Section D-5.2.3.

The tolerances which should be chosen cannot be based on the dynamic processes or analytical methods embodied in the model, but must be based on the model's anticipated use. The tolerances should in general be tightest on the primary output variables, those which the model is to be used to predict. The tolerances should also generally be tighter for representing the trends from one case to another than for predicting absolute values of individual responses. This emphasis on the trends is recommended because the prediction of trends is generally a more important use of vehicle dynamic models than the prediction of responses to specific conditions and the trends are also less likely to be disturbed by random errors in testing.

There are also some fundamental physical reasons why the tolerances should be different on different types of response variables. For example, the tolerance on prediction of natural frequency should generally be tighter than that on damping ratio. The natural frequency of a system or of a response mode is probably the most important single description of its performance, and it should also be relatively easy to predict on the basis of simple estimates of mass and stiffness properties. Although damping is also an important description of a system's dynamics, the most common measure, damping ratio, is difficult to identify from test data, as well as not being applicable to nonlinear responses such as those produced by dry friction in suspensions. The exponential decay envelope assumed for linear damping cannot be matched directly to the triangular envelope produced by dry friction. Consequently, the tolerances on damping ratio should generally be much looser than the tolerances on natural frequency. A $\pm 20\%$ tolerance on damping ratio could be regarded as very stringent, while the same tolerance on natural frequency would be very loose for most applications.

The tolerances on the statistics which are not described by single numbers must be defined carefully in order to maintain statistical validity while not disguising potential problems. The comparisons between the measured and predicted values of these statistics can be made in several different ways, as listed in Table D-3.7. None of these measures is ideal, and indeed in some cases it may be a good idea to use more than one of them. The choice should, once again, depend on how the model is expected to be used (e.g., whether to evaluate average or extreme conditions). When the correlations and the frequency domain measures are used, it is necessary to consider the statistical significance of the measures which are derived from the test data. The following example for the comparison of modeled and observed response spectral densities will illustrate this point:

Table D-3.7

Ways of Comparing Multiple-Valued Statistics

COMPARISON	IMPLICATIONS
MAXIMUM AMPLITUDE DIFFERENCE	SIMPLE TO COMPUTE, BUT • ACCENTUATES POOR AGREEMENT IN ISOLATED PLACES • IGNORES GOOD AGREEMENT IN MOST PLACES • VERY SENSITIVE TO PHASING ERRORS
FIT ERROR	CONCISE MEASURE OF "GOODNESS OF FIT," BUT • VERY SENSITIVE TO PHASING ERRORS • BIAS ERRORS MUST BE REMOVED • SENSITIVE TO DIFFERENT NOISE LEVELS • MORE COMPLICATED TO COMPUTE
PHASE OR TIME OR FREQUENCY DIFFERENCE	SIGNIFICANT MEASURE OF VERISIMILITUDE OF RESPONSE CHARACTERISTIC, BUT • USUALLY VARIES ACROSS THE STATISTIC (NOT A SINGLE MEASURE) • CAN BE DIFFICULT TO MEASURE PRECISELY
WEIGHTED VALUES OF ABOVE COMPARISONS	CAN BE BETTER TAILORED TO THE PURPOSE OF EACH SPECIFIC MODEL, BUT • SHARES THE SAME IMPLICATIONS AS ABOVE • SOMEWHAT MORE COMPLICATED TO COMPUTE

Model-predicted spectral densities are "smooth" as long as the model does not include any noise. However, the spectral densities derived from test data are always "noisy," since noise cannot be eliminated in any real test arrangement. Even if the model were a "perfect" representation of the vehicle being tested, the two spectra would be substantially different. To make matters even worse, the amount of noise present in the test directly affects the comparisons between the measured and model-predicted spectra. This problem can be handled in several different ways. The best way of reducing the noise effect is to run numerous long tests, calculating separate spectral densities for each (or even several separate spectra per test, if they are long enough), and to then average these spectra together. In addition to this averaging process, the "jagged" look of the experimental spectra can be reduced by numerical smoothing of the data, averaging several adjacent data points together to get each new data point. These averaging and smoothing processes will reduce the effects of noise on both the maximum-difference and fit-error measures of the disagreements between the test results and model predictions.

The statistically "correct" approach to evaluating the differences between the tests result and model prediction statistics, taking account of noise influences and the amount of data available, should be left to the statisticians. Some of the requirements of that approach are described in Appendix D-A. For more general users of model validation procedures, the tolerances on the statistical validation criteria will have to be selected to correspond to an educated "best guess" about how close the model predications should be to the test results. This guess should be based on the analyst's confidence that the results throughout the course of each test, and when compared to other tests, remain consistent and directly related to the physical phenomenon being modeled, rather than being associated with random occurrences, noise, or test and instrumentation errors.

D-3.3.4 Mathematical Form

The final aspect of the definition of the validation criterion is the mathematical form which is used. This simply refers to the way of combining all the separate measures of a model's validity in order to reach the final verdict on whether the model has been validated. Each different mathematical form is based on different assumptions about the relative importance of the separate measures of validity. The types of mathematical forms which could be used are listed in Table D-3.8.

Table D-3.8

Mathematical Forms For Model Validation Criterion

MATHEMATICAL FORM	EVALUATION
(1) THRESHOLD ON WEIGHTED SUM OF ERRORS	TOTAL SUM OF ABSOLUTE VALUES (OR SQUARES) OF DIFFERENCES BETWEEN MODEL AND TEST RESULTS FOR ALL MEASURES, WEIGHTED BY UNITS AND RELATIVE IMPORTANCE, MUST BE WITHIN SPECIFIED THRESHOLD.
(2) THRESHOLD ON WEIGHTED SUM OF TOLERANCE EXCEEDANCES	TOTAL SUM OF ABSOLUTE VALUES (OR SQUARES) OF AMOUNT BY WHICH PERFORMANCE MEASURE TOLERANCES ARE EXCEEDED, WEIGHTED BY UNITS AND RELATIVE IMPORTANCE, MUST BE WITHIN SPECIFIED THRESHOLD.
(3) THRESHOLD ON WEIGHTED PRODUCT OF ABSOLUTE VALUES OF ERRORS	PRODUCT OF ABSOLUTE VALUES (OR SQUARES) OF DIFFERENCES BETWEEN MODEL AND TEST RESULTS, FOR ALL MEASURES, WEIGHTED BY UNITS AND RELATIVE IMPORTANCE, MUST BE WITHIN SPECIFIED THRESHOLD.
(4) NO TOLERANCE EXCEEDANCES	IF ANY ONE OF THE PERFORMANCE MEASURES EXCEEDS ITS TOLERANCE, THE MODEL IS NOT VALIDATED.

Typically, one would use a combination of these forms when validating a model. In particular, mathematical form #4, requiring performance within the tolerance band, could be applied to one or more of the most critical variables, while the other forms could be applied to the remaining variables. The quadratic measures of #1 through 3 apply relatively heavier penalties to large errors than the absolute value measures. The mathematical forms which incorporate the differences between test and model results (#1 and 3) permit relatively large errors in some variables to pass if the other variables are predicted very accurately. This is particularly true of the product form (#3), which must be applied very carefully (if at all) so that near-perfect prediction of one variable does not permit very poor prediction of all the rest. These two forms do not explicitly include the tolerances on the individual performance measures, although those tolerances would be used to set the threshold value. On the other hand, mathematical form #2 explicitly includes the individual tolerances, and only penalizes exceedances, while not rewarding very accurate predictions of variables within the tolerance bands.

The four mathematical forms of Table D-3.8 incorporate different built-in assumptions about the importance of agreement between model predictions and test results for model validation. The choice of which form(s) to use for each model validation excercise should be based on which assumptions are appropriate for each variable. It may even be advisable to use more than one form for some of the most important variables, incorporating them in the weighted combination forms (#1, 2, and 3) and also imposing firm limits on the allowable differences between model and test results (#4).

In the end, the choice of the mathematical form of the validation criterion must be based as heavily on analyst judgement as the previous choices of variables, statistics and tolerances. The discussion of the alternative forms in this section can be used as guidance in choosing the form of validation criterion which is most appropriate for any particular model and purpose.

D-3.4 OVERVIEW OF NEXT THREE CHAPTERS

Now that the validation criterion has been defined in terms of its dynamic response variables, statistics, tolerances and mathematical form, the preliminary steps of the validation process are completed and we can move on to the steps which involve the majority of the effort. These remaining steps are described in the next three chapters. Because of the very substantial differences between the procedures which should be followed when using existing test data and when designing a new validation test program, Chapters D-4 and D-5 deal with these two cases separately. The final stage in the validation process, which applies to both cases, is the comparison between model predictions and test data. This stage is, more than any other, what makes model validation "special," and therefore it is treated separately in Chapter D-6.

D-4. USING EXISTING TEST DATA

Very few of the rail vehicle test programs which have been run in the past were designed with model validation in mind. As a result, the data which were collected are generally not well suited for validating models. In spite of this, it is sometimes necessary to "make do" with existing data because limited time and money do not permit a new test program. This chapter describes the approach which should be followed when it is necessary to try to validate a model using existing test data, rather than being able to design a new test program from scratch. This approach does not necessarily lead to a successful model validation in all cases, because the available data may simply be inadequate to validate the desired model for the stated purpose. The steps which are covered in this chapter are shown schematically in Figure D-4.1.

D-4.1 REVIEW AND SUMMARY OF AVAILABLE DATA

Before investing a great deal of effort in trying to validate a model with inadequate data, it is useful to make a preliminary assessment of the adequacy of the data. This involves reviewing and summarizing the test data so that any important deficiencies can be spotted immediately. The features of the data set which should be covered in this review are listed in Table D-4.1.

The basic review of the data which are known and unknown should provide some initial indication of whether the desired validation is feasible. For example, if these test results are only available for a limited speed range and the model is to be used over a broad range of speeds, the validation effort may be in vain. Similarly, if the model is to be used to evaluate the effects of various vehicle design changes and the test results are only available for one vehicle configuration, the data could be inadequate. Of course, data which were collected for vehicles and dynamic processes totally different from those which the model is intended to represent will not be useful for validation. It should also be apparent that the validation of models which include wheel-rail interaction effects will require track geometry data which can be synchronized with vehicle dynamic measurements. If the basic require-

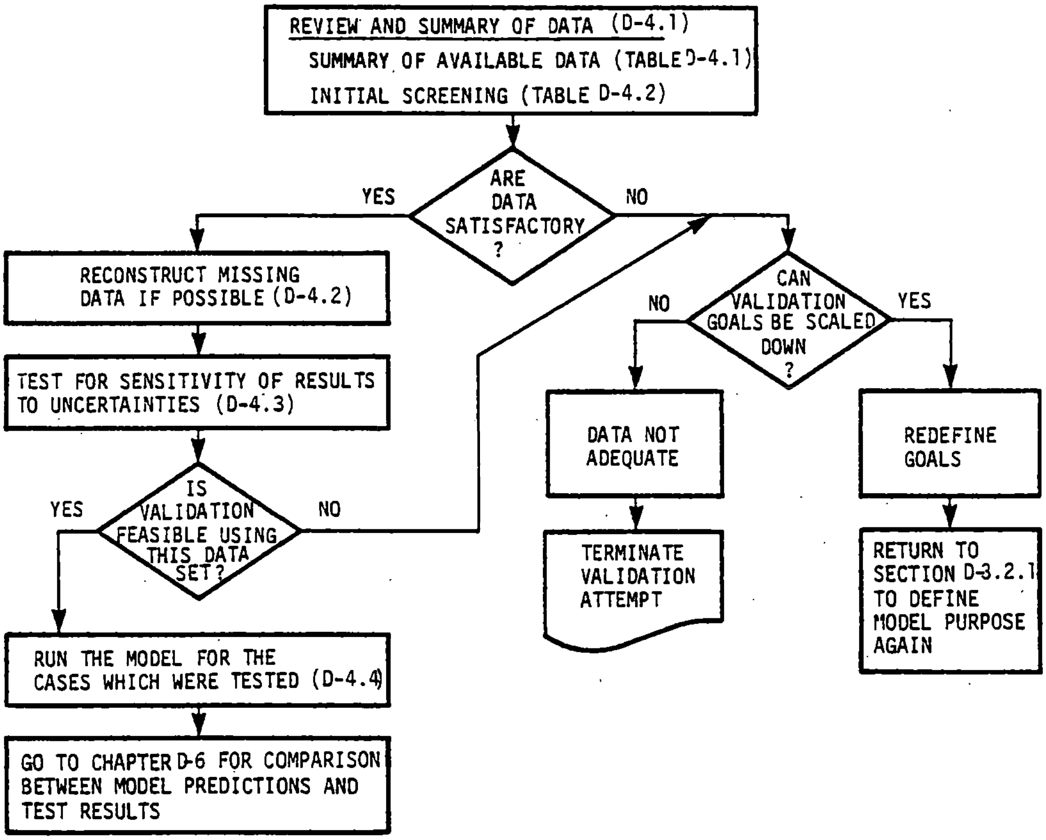


Figure D-4.1 Preparing to Use Existing Test Data (Chapter D-4)

Table D-4.1

Summary Of Existing Test Data

TEST CONDITIONS:

PURPOSE OF ORIGINAL TEST

ENVIRONMENTAL CONDITIONS:

- TEMPERATURE AND HUMIDITY
- PRECIPITATION
- WIND SPEED AND DIRECTION

OPERATING CONDITIONS:

- SPEED
- ACCELERATION OR BRAKING
- VEHICLE LOADING

TRACK CHARACTERISTICS - WHICH WERE MEASURED AND WHEN?

- TANGENT OR CURVE
- GRADE AND SUPERELEVATION (IF ANY)
- VERTICAL AND LATERAL MODULUS (STATIC AND DYNAMIC)
- GEOMETRY (PROFILE, GAUGE, ALIGNMENT, CROSSLEVEL)
- ROUGHNESS
- CROSS-SECTIONS OF RAIL HEAD
- REVENUE SERVICE AND MAINTENANCE HISTORIES

VEHICLE CHARACTERISTICS - WHICH ARE KNOWN?

- GEOMETRY (DIMENSIONS)
- MASS PROPERTIES
- SPRING AND DAMPING CHARACTERISTICS
- DETAILS OF NONLINEAR CHARACTERISTICS
- WHEEL PROFILES
- REVENUE SERVICE HISTORY (TYPE AND LADING)
- MAINTENANCE HISTORY

DYNAMIC RESPONSE MEASUREMENTS

- WHICH WERE MEASURED?
- HOW ARE THEY SYNCHRONIZED WITH TRACK DATA?
- ARE ANY CHANNELS MISSING?
- HOW "GOOD" ARE THEY?
  - BANDWIDTH (FILTERING)
  - DYNAMIC RANGE
  - NOISE LEVELS
  - ACCURACY
  - KNOWN ERRORS

ments such as these cannot be met by the available data, it is probably not worthwhile continuing with the attempt to validate the model in question.

If the basic requirements are satisfied, the review should continue with a look at some more detailed aspects of the available data. In particular, the response variables which appear in the validation criterion must be available from the test data. These are generally vehicle dynamic response measurements, but in the case of the frequency response functions it is also necessary to have the appropriate input measurements (typically, track geometry), synchronized with the vehicle responses. If some of these variables were not measured directly during the test program, it may still be feasible to reconstruct them from the data which were collected (see Section D-4.2). It is also important to review the quality of the available test data (accuracy, bandwidth, dynamic range, etc.) to ensure that the vehicle performance is described precisely enough to fit comfortably within the tolerances in the validation criterion. Of course, it is also important that the parameters of the vehicle which are included in the model (dimensions, masses, static and dynamic spring and damping characteristics) be known for the test vehicle.

This initial screening of the test data is summarized in Table D-4.2. If the answers to any of the eight major questions of Table D-4.2 are negative, the goals of the model validation exercise will need to be scaled down to match the limitations of the data. The efforts will then need to be focused on validating the model for those conditions for which the test data are adequate and those which can be covered by reasonable extrapolations from the test data. This means that the model validation can proceed for those <u>performance regimes</u> for which test data are available.

A performance regime or dynamic regime is a set of conditions for which the trend of vehicle performance remains characteristically the same. Nonlinear vehicles (including virtually all rail vehicles) typically have different performance regimes for different input levels (speeds, track geometry). As the levels of the inputs (and therefore of the responses) increase, nonlinearities will cause either gradual or sudden changes in performance. Test data which are collected before the sudden transitions cannot be used to predict performance after the transitions (in a new performance regime). If enough test cases are available at various speed and input levels to define

Table D-4.2

Initial Screening Of Existing Test Data

(1) WERE THE TEST CONDITIONS COMPARABLE TO THE CONDITIONS THE MODEL WILL BE USED TO EVALUATE?

- SIMILAR SPEED RANGE
- SIMILAR TYPE OF TRACK (TANGENT VS. CURVE)

(2) WAS THE TEST VEHICLE BASICALLY SIMILAR TO THE VEHICLE BEING MODELED?

- LOCOMOTIVE OR FREIGHT OR PASSENGER CAR
- TRUCK DESIGN AND LOCATION (TRUCK CENTER SPACING)
- MASS AND MASS DISTRIBUTION

(3) WERE ENOUGH DIFFERENT VEHICLE CONFIGURATIONS TESTED TO SHOW THE EFFECTS OF DESIGN CHANGES?

(4) WAS THE DYNAMIC PROCESS WHICH HAS BEEN MODELED OBSERVED IN THE TEST?

(5) ARE TRACK GEOMETRY DATA AVAILABLE FOR THE TEST TRACK SECTION IN A FORM WHICH CAN BE SYNCHRONIZED WITH THE VEHICLE DYNAMIC RESPONSE DATA?

(6) ARE ALL THE RESPONSE VARIABLES WHICH APPEAR IN THE VALIDATION CRITERION AVAILABLE IN THE TEST DATA?

IF NOT, CAN THEY BE RECONSTRUCTED FROM THE TEST DATA?

(7) ARE THE TEST DATA ACCURATE ENOUGH TO FIT COMFORTABLY WITHIN THE TOLERANCES IN THE VALIDATION CRITERION?

(8) WERE THE VEHICLE PARAMETERS WHICH APPEAR IN THE MODEL RECORDED AT THE TIME OF THE TEST?

IF NOT, ARE THEY AVAILABLE FROM OTHER RELIABLE SOURCES?

accurately the gradual changes in performance, the model can be validated for that gradually changing nonlinear performance regime. However, there may be no way of knowing how much the input levels can be increased before meeting a sudden transition to a new performance regime (which could include unacceptable performance such as hunting). Some of these transitions can be anticipated by carefully watching trends in the response of important nonlinear elements (such as wheel/rail displacements approaching flange contact, suspension springs bottoming out or dry friction elements approaching break-away force levels). Unfortunately, there is no guarantee that the abrupt transitions to new and unmodeled performance regimes will be predicted this way. Therefore, model predictions outside the performance regimes which have been observed in the tests should not generally be relied upon. In other words, the model cannot be validated outside these performance regimes.

D-4.2 RECONSTRUCTION OF MISSING DATA

Because the test data were probably gathered for a purpose other than model validation, the measurements which were made were not necessarily those which are needed for the validation work. However, the available data may still be usable for reconstructing the needed quantities. The complexity of the reconstruction process will vary from case to case, being very simple in some cases and quite complicated in others.

The simplest data reconstructions are those based on proportional summing of related measurements (i.e., measurements having comparable units). For example, the vertical (and lateral) accelerations measured at the front and rear of an essentially rigid vehicle body can be combined to produce estimates of the bounce and pitch (and lateral and yaw) accelerations of the vehicle body. If accurate enough wheel/rail displacement, track geometry and suspension deflection measurements were available, these could be used to estimate body displacements. However, the wheel/rail displacement measurements are typically the most difficult to make, diminishing the attractiveness of this possibility.

Data reconstructions involving mixed combinations of units are somewhat more complicated, requiring the use of simple dynamic models to reconstruct the missing data. For example, measurements of truck and body accelerations and initial displacements can be used to reconstruct the velocity and displacement of suspension elements mounted between the trucks and body. Similarly,track geometry space curve data can be numerically differentiated (if appropriately filtered to reduce noise) to produce track geometry rate-of-change data for validation of vehicle models with primary suspension damping. The data obtained in this type of reconstruction depend for their accuracy on the appropriateness of the model used for the reconstruction, and are therefore less accurate than they would be if they could have been measured directly.

It is tempting to use this type of data reconstruction to estimate some of the difficult-to-measure wheel/rail force and displacement quantities. However, this estimating should be approached very cautiously to make sure that (1) the measurements being used were made relative to the proper frames of reference, (2) rail deflections are not overlooked, (3) measurement noise is not propagated by numerical integration, and (4) sufficiently accurate initial condition information (especially for displacements) is available to start the model properly.

Even more sophisticated techniques can be applied to reconstruct missing test data. These techniques of optimal state estimation rely on relatively complicated models of the processes which were not measured, and use sophisticated mathematical algorithms to estimate the missing measurements [3]. These methods should only be applied by users who fully understand the mathematical principles involved and are willing to invest considerable effort in reconstructing the data.

D-4.3 SENSITIVITY TESTING FOR UNCERTAINTIES

Because the existing test data which were collected for purposes other than model validation generally do not include all of the information needed for validation, it may be necessary to make educated guesses about the values for some quantities. These could include unmeasured vehicle parameters or general track geometry and roughness parameters, which could not be reconstructed from the existing test data. When such guesses are necessary, they should include not only the "best guess" value, but also the upper and lower limits on what the real value might be. If you need to be 90% sure that the model validation criterion defined in Section D-3.3 is satisfied, then you should choose upper and lower bounds which you think include 90% of the possible values of the uncertain parameters. Similarly, if the level of confidence needed in the final validation is different, the limits on the parameter values should be chosen to meet that confidence level.

Before going further with the validation procedure, it is important to test the sensitivity of the outcome (whether or not the model satisfies the validation criterion) to the uncertainties caused by estimating the unknown quantities. Doing this in a statistically "correct" way is difficult and time consuming, and is only recommended when there are compelling reasons. The approach which is described in this section is approximate, and depends heavily on the judgement of the analyst about what should be considered "reasonable" combinations of uncertain values. The basic idea is to make sample model runs using some "worst case" values for the uncertain parameters to bound the uncertainties and then to compare the range of the model output values with the tolerances established in the validation criterion. If the variations in the output values exceed the allowable tolerances, we do not have good enough information about the parameter values to complete the validation, and further validation efforts will be unproductive.

When there is only one uncertain parameter to be tested for sensitivity, the process is quite simple. The maximum and minimum expected values of that parameter should be used in separate model runs for several typical operating conditions, and the model outputs then judged against the validation criterion. If they are within the specified tolerances, the model is insensitive enough to the uncertainty for the validation to proceed. However, if the tolerances are violated in this sensitivity test, the uncertainty about the parameter value is too large for the validation to succeed. Although it may be possible to find a parameter value which satisfies the validation criterion, there may be no way of knowing if that value correctly represents the condition of the test, or if errors in some other part of the model may be compensating for errors in the value of the uncertain parameter. (System identification techniques can be used to develop maximum likelihood estimates of the values of uncertain parameters and of appropriate model structures, but these typically require high-quality measurement of both inputs and outputs, and cannot guarantee the validity of the estimates when there are too many uncertainties.)*

The process becomes more complicated, and more dependent on analyst judgement, when there are several uncertain parameters. Combinations of variations in those parameters must be considered, but if there are more than two or three parameters to vary, the number of cases which would be needed to test all the combinations of maximum and minimum parameter values can become very large (2^n for n uncertain parameters). In practice, it is highly unlikely that all or even most of the uncertain parameters would have values near the extremes of their ranges, so the evaluation of sensitivity using combinations of all worst-case values will appear to exaggerate any sensitivity problems. For example, consider mass and spring constant values which are known to $\pm 20\%$. Using the maximum mass value and the minimum spring value produces an effective natural frequency which is 33% less than that for the opposite extreme case (minimum mass and maximum spring). That level of variability in such a critical characteristic is likely to be unacceptable for most model purposes.

When there are several uncertain parameter values, the sensitivity test should begin with separate sets of model runs for the minimum and maximum values of each parameter, while holding all the others at their "nominal" (or "best guess") values. This test will indicate the relative importance of the uncertainties in the different parameters, and may indicate that one or more of the uncertainties alone is already severe enough to jeopardize the validation (analogous to the test for a single uncertainty). If this test is passed, the analyst must then choose some combinations of uncertainties to test, including reasonable but not extreme deviations from the nominal parameter values. These combinations should be chosen to have a likelihood of

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* For more complete information on system identification, refer to the Supplementary References and Reference 4: Hull, R.L., T.L. Trankle, and D.L. Klinger, Application and Evaluation of System Identification Techniques to Rail Vehicle Dynamics, Systems Control Technology, Inc. Report No. TR-5307-100, November 1979, for Transportation Systems Center, Cambridge, MA.

occurrence comparable to the acceptable level of uncertainty about the validation of the model (i.e., if you need to be 90% sure that the validation criterion is satisfied, the combination of uncertainties should be within a range which you are 90% sure will not be exceeded). The model should be run again for the chosen combinations of off-nominal conditions, and the results again checked against the tolerances in the validation criterion to see whether the effects of the combined data uncertainties are small enough to permit the validation to proceed.

The sensitivity testing in this step of the validation procedure is needed to ensure that there is sufficiently good information available in the test data to validate the desired model. After completion of this step, the model is no closer to being validated than it was before, but the vital question of whether or not it is feasible to accomplish the validation is answered here. If the answer is negative, the validation goals will need to be redefined or the validation effort abandoned (saving the effort which would otherwise be wasted in later steps).

D-4.4 RUNNING MODEL FOR CASES WHICH WERE TESTED

The validity of the model must be tested by comparing its predictions with the test results, following the approach to be described in Chapter D-6 of this report. In order to do that, the model must be run for cases which represent the test conditions as closely as possible. The issues of data reconstruction and sensitivity testing which were covered in Sections D-4.2 and D-4.3 must be considered carefully here.

The model should be run using the sets of vehicle parameter values which were selected in Section D-4.3 (the most likely values and some extreme cases for uncertain parameters). The inputs for the model runs should be those which were measured, as well as the estimated values of any inputs required by the model which needed to be reconstructed. The model outputs should be processed in exactly the same way the test data were, including sampling rates, filtering, instrumentation errors, biases, and noise levels (if known). In particular, if the test instrumentation or data processing imposed any limitations of bandwidth or dynamic ınge, these limitations must be modeled as closely as possible to produce ꞌparable saturations, phase shifts, and other filtering effects on the simulated data. This helps to ensure that the comparison of model and test data is valid (i.e., comparing "apples and apples" rather than "apples and oranges").

The model outputs should be reduced to a form compatible with that available from the test data (the same statistics of the same response variables). Once this is done, the comparisons between model predictions and test data, using the chosen validation criterion, can follow directly. Guidelines for the comparison process are found in Chapter D-6 of this report. (It is not necessary to refer to Chapter D-5 if the validation is to be done using only existing test data, although Section D-5.3 can provide some additional guidance regarding execution of the model.)

D-5. DESIGNING AND RUNNING A NEW VALIDATION TEST PROGRAM

When adequate data are not available from existing tests, the only way to validate a model is to run a new test program. If these tests are properly designed and conducted, they should produce all the data needed to determine whether a model is valid for the conditions of interest. The model which has been proposed for validation, or the "candidate model," may not be a very good representation of reality. In that case, it will be shown to be "not valid," rather than "valid." The model adjustment process, which will be described in Chapter D-6, may lead to the formulation of a better model, which can then be validated. The important thing to remember is that the original model may not necessarily be validated at all using the test data, and if that is the case, it is not necessarily the "fault" of the test data. Indeed, if the model is poor, the test results should reveal that and should lead to a "not validated" verdict. The procedures to be covered in this chapter are shown schematically in the flowchart of Figure D-5.1, which follows the outline of the chapter very closely.

D-5.1 SELECTION OF OPERATING CONDITIONS

The conditions to be tested must be chosen to provide the data which will be needed to demonstrate whether or not a candidate model is valid for its intended purpose. This requires the selection of an appropriate set of vehicle characteristics and input conditions. These must be diverse enough to cover all the performance regimes and vehicle variations of interest, but must at the same time be limited by economic and technical constraints. Some potential test conditions are too costly and/or hazardous to use in the test program, and will therefore have to be omitted. The overall number of cases to test must also be limited to keep the cost of the validation program from becoming excessive. In any event, each of the potential test cases must be evaluated by pilot runs of the candidate model before deciding which to use and which to reject.

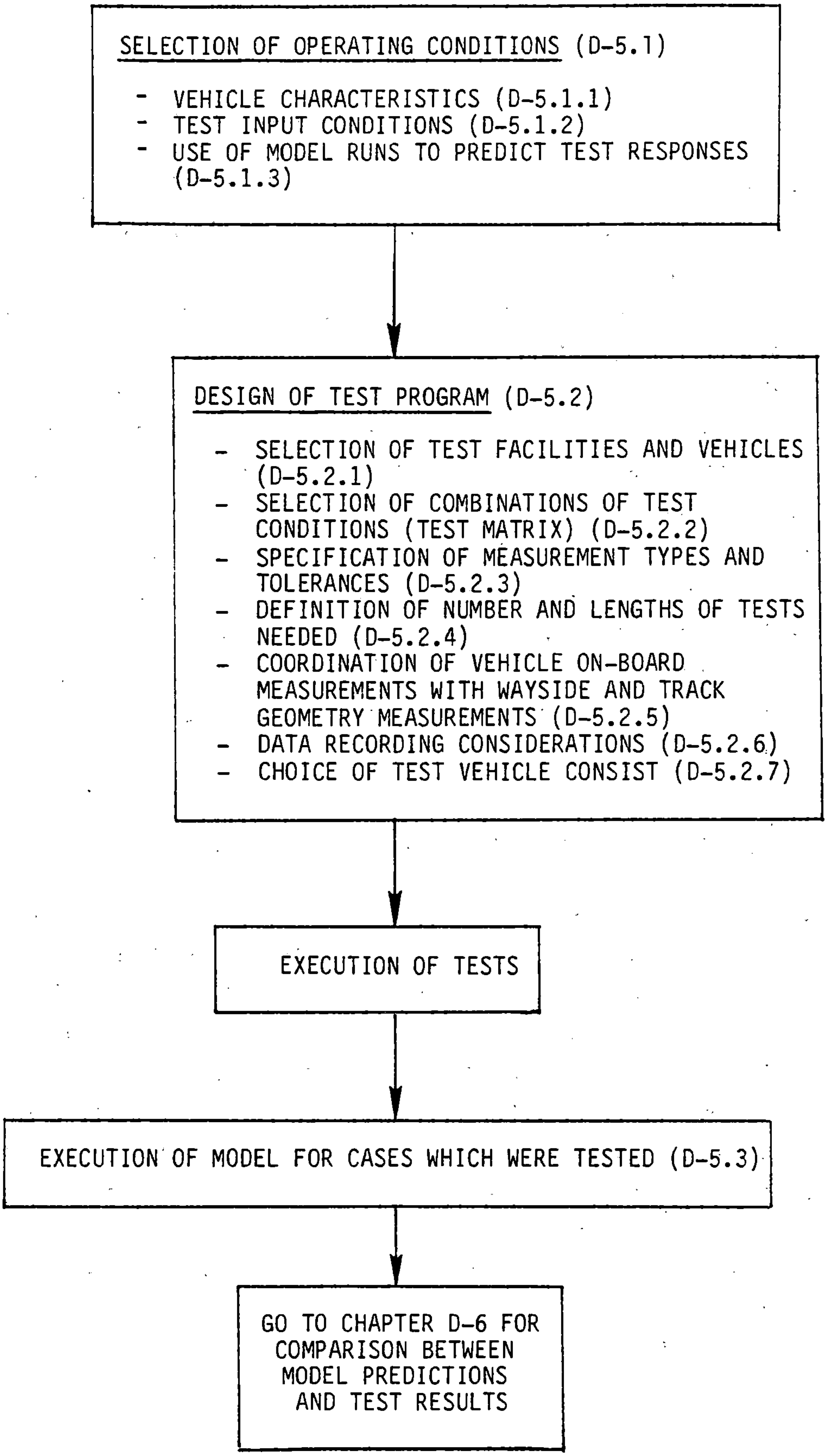

Figure D-5.1 Designing And Running A New Validation Test Program (Chapter D-5)

D-5.1.1 Selection of Vehicle Characteristics

The test vehicle(s) should be dynamically similar to the vehicle(s) which the model is expected to represent. The dynamics of locomotives, freight cars and passenger cars are so different from each other that a model developed for one should not be compared with test results for another. A model may not even be suitable for all vehicles within one of these general classes. The key differences among vehicles which limit the applicability of general models include truck designs and body stiffnesses and centers of gravity. The effects of these factors on dynamic response must be included very explicitly in a model if it is to be used to represent vehicles which differ from each other in these factors.

Models are almost always designed to be used to represent a family of vehicles, or a vehicle which may have a range of characteristics, rather than one specific, unchanging vehicle. In order to validate the model's ability to represent the effects of changes in vehicle characteristics, the validation tests should include some variations in these vehicle characteristics, such as those listed in Table D-5.1. Without this variety of test cases there is no way to tell whether the model correctly predicts the effects of vehicle design changes. The variations which are needed in the test program will only be a small fraction of the number of variations which the model can be used to represent, and they can generally be those which are relatively simple and inexpensive to change. The exceptions to this are the changes in basic truck design characteristics, which require the replacement of one truck by another during the test program.

The choice of which variations in vehicle characteristics to include in any specific validation test program must be based on the anticipated use of the validated model and the sensitivity of the model's outputs to changes in those characteristics. For example, the height of the center of gravity is much more important for twist-and-roll and curving models than for models of other dynamic processes, and it would therefore make sense to include c.g. height variations in the test designs for only those categories of models. If the model is to be used to evaluate the performance of vehicles which use only one type of truck, there is no need to include tests of other truck designs. Apart from the decision about whether to test more than one truck, the most

Table D-5.1

Typical Vehicle Characteristics To Vary In A Validation Test Program

<u>USING SAME VEHICLE:</u>
LOADING
VARIATION IN WEIGHTS
VARIATION IN DENSITY (CHANGE CENTER OF GRAVITY)
SUSPENSION -
CHANGE SIDE BEARINGS
ADD, CHANGE, OR REMOVE SHOCK ABSORBERS (DAMPERS)
CHANGE STOPS
CHANGE SPRINGS (DIFFERENT STIFFNESSES)
CHANGE TRUCKS -
DIFFERENT GEOMETRY (DIMENSIONS)
DIFFERENT LATERAL AND YAW STIFFNESSES
DIFFERENT SUSPENSION CHARACTERISTICS
DIFFERENT SPRUNG/UNSPRUNG MASSES
DIFFERENT STAGES OF WEAR -
WHEEL PROFILES
SNUBBERS
SUSPENSION CLEARANCES
<u>USING DIFFERENT VEHICLES:</u>
VARIATION IN GEOMETRY (DIMENSIONS)
VARIATION IN BODY STIFFNESSES AND ELASTIC MODES

important choice is likely to be whether to test carbodies having different stiffnesses and elastic modes. This is an important factor to consider if the same model is to be used to represent highly diverse freight cars (from flat cars to tank cars to covered hoppers), and is vital for the decision about which, if any, body structural modes need to be included in the model.

D-5.1.2 Selection of Test Input Conditions

The test input conditions should be chosen to cover all of the performance regimes which the model will need to represent. This is generally an iterative process, including the use of sample model runs, because the performance regimes are not necessarily well understood in advance. The selection of inputs includes specifying as many conditions as necessary beyond the physical characteristics of the test vehicle, which were covered in Section D-5.1.1. The test conditions which must be selected are generally chosen from among those in Table D-5.2.

Any of the items in Table D-5.2 which could influence the agreement between the candidate model and test results should be specified for the test program. In many cases, that could mean simply making sure that the condition (such as temperature or humidity or track compliance) remains fairly constant among all the tests. In other cases, especially for speed and track geometry, it would involve specifying the sets of values which should be used in separate tests to reveal all the important aspects of model performance. These choices must be based on the specific model and its anticipated use, as well as the dynamics of the test vehicle.

The validation test conditions may or may not be representative of revenue service conditions. The full range of revenue service inputs could make the validation unnecessarily complicated and could obscure the contributions of the different types of inputs to vehicle response. For validating a model, it is easier to work with special track having carefully chosen perturbations than to use "typical" revenue track. Similarly, it is much easier to validate a single-vehicle model using a special test consist (see Section D-5.2.7) than to do it in a typical revenue train consist, which introduces the additional effects of interactions with other vehicles.

Table D-5.2

Input And Operating Conditions To Specify In A Validation Test Program

OPERATING SPEED PROFILE

- CONSTANT SPEEDS
- ACCELERATION OR BRAKING RATE (AND TYPE OF BRAKING)

TRACK GEOMETRY

- TANGENT OR CURVE
- GRADE, CURVATURE AND SUPERELEVATION
- PROFILE
- GAUGE
- ALIGNMENT
- CROSSLEVEL
- RAILHEAD CROSS SECTION

RAIL SURFACE CONDITION

- ROUGHNESS
- JOINT CHARACTERISTICS (IF JOINTED)
- LUBRICATION (DRY, WET, SANDED, LUBRICATED)

TRACK COMPLIANCE

- LATERAL AND VERTICAL
- STATIC AND DYNAMIC

ENVIRONMENTAL CONDITIONS (DURING AND BEFORE TEST)

- WIND SPEED AND DIRECTION
- TEMPERATURE RANGE
- PRECIPITATION (OR HUMIDITY)

CONSIST

- LENGTH OF CONSIST AND DISTRIBUTION OF CAR TYPES
- CARS COUPLED DIRECTLY TO TEST CAR(S)

Although it is not recommended that the principal validation tests be conducted under typical revenue service conditions, it would be very helpful to "double-check" the validated model against one or two revenue service test cases after all the other tests and analyses are complete. A properly validated model should be able to represent revenue service conditions in the performance issue of interest, even though the validation tests themselves are not conducted under the same conditions.

Pilot Runs

The most important factor to consider in choosing the test input conditions is making sure that these conditions include all of the performance regimes which the model must be able to represent in order to satisfy its purpose. This should be accomplished by making pilot runs of the candidate model (or, if possible, of a model which has already been validated) for some carefully chosen sample conditions. The first runs should be for the most extreme conditions which the model is expected to have to represent in order to help define the most strongly nonlinear performance regimes expected and to help ensure that the tests will not include unsafe conditions (such as derailments). These should be followed by other runs for a large variety of intermediate conditions, which can be used to identify the intermediate performance regimes. The model runs are much less expensive than vehicle tests, so it is economical to make model runs for many more cases than are eventually tested. Based on the results of the model runs, several test cases should be chosen within each performance regime, with particular emphasis on choosing cases near the regime boundaries in order to verify where those boundaries really are.

The example shown in Figure D-5.2 should help illustrate this point for a hypothetical case in which the performance regimes are defined in two dimensions, speed and track roughness. The system described by Figure D-5.2 displays four qualitatively different types of dynamic response, represented by the four separate performance regimes. The boundaries were determined using numerous pilot runs of the model and the suggested test cases are marked by x's. These have been chosen to include only a few track roughnesses, since each requires a separate section of test track. The speeds have been allowed to take on many different values, since the speed can be chosen separately for

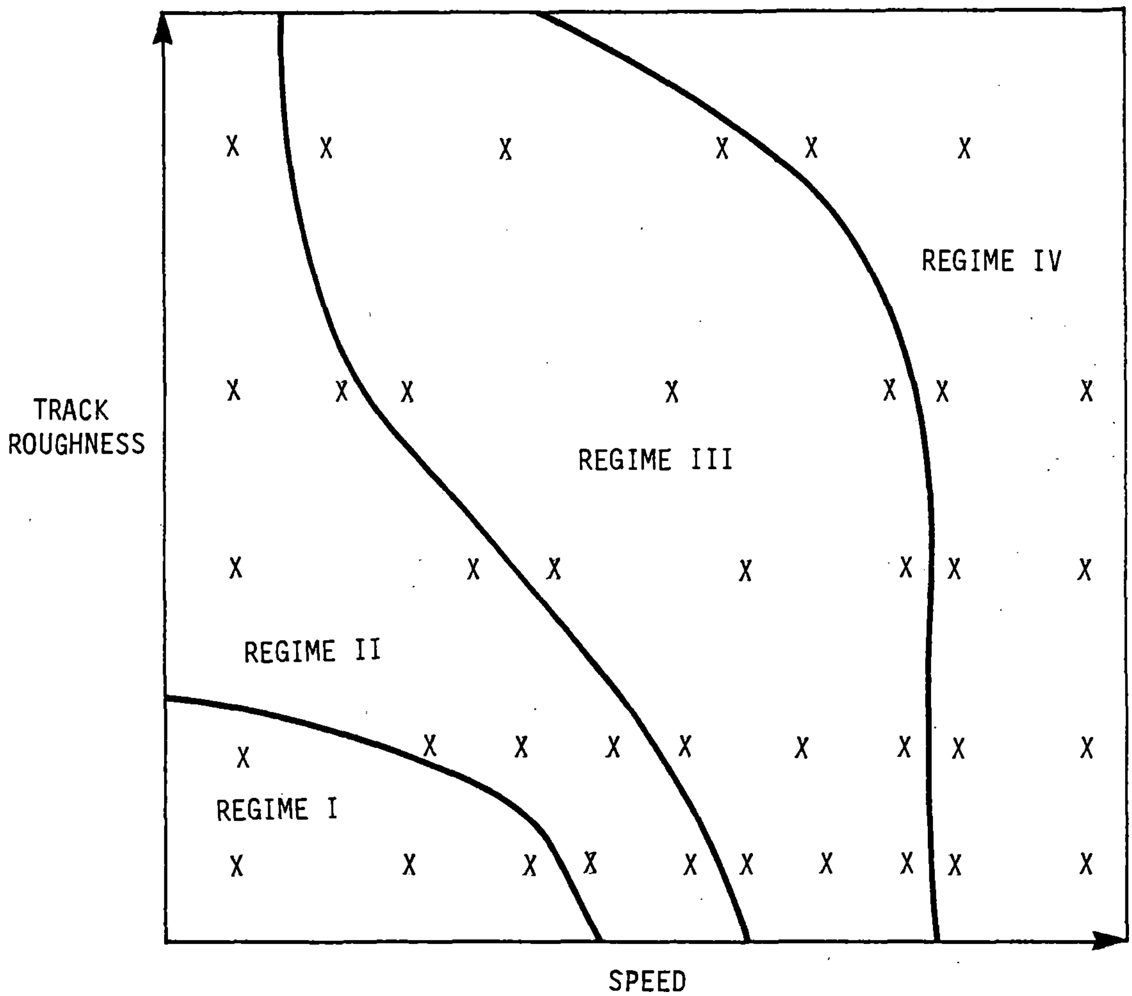

Figure D-5.2 Examples of Selection of Test Input Cases to Define Boundaries of Performance Regimes each individual test run. Where the boundaries of the performance regimes are close together, the test cases are clustered more tightly than where the boundaries are widely separated. Cases are deliberately selected close to the expected boundaries, on both sides, to help identify these boundaries as accurately as possible.

For validating most models, it is best to maintain speed as nearly constant as possible throughout each test run. Acceleration and braking conditions should only be applied for validating models which include acceleration and braking effects directly, such as the multivehicle models (longitudinal impact or train action). Similarly, track compliance should be maintained uniform across all tests except when validating a model which explicitly incorporates the effect of changing compliance.

<u>Track Geometry Perturbations</u>

The track geometry inputs to specify for model validation tests should be chosen to fit the dynamic process being modeled, particularly when testing on deliberately perturbed track. Twist-and-roll models should be validated in tests on track having periodic cross-level perturbations corresponding to staggered rail joints. Vertical forced response models could be validated with tests on track having profile (and cross-level) perturbations, while curving models of course require the use of curved track. Validation of lateral stability (hunting) and lateral forced response models, as well as dynamic curving models, could involve use of perturbations in virtually all of the track geometry measures (except perhaps grade). The amplitudes of the perturbations should be selected on the basis of pilot model runs, as already discussed for the track roughness example of Figure D-5.2.

The forced response models designed to predict ride quality under normal operating conditions and the hunting models can be validated using tests performed on "typical" track of several different classes, rather than requiring use of specially perturbed track. In these cases, long test runs of statistically stationary data can be processed quite efficiently when analyzing the results. Perturbed track tests, on the other hand, require the comparisons between model and test results to be calculated in the time domain, subject to a separate set of precautions (recognizing that multiple performance regimes may be evident within one run, that the results are not stationary and that frequency domain analyses are probably inappropriate).

Once it has been decided that one or more track geometry measures should be perturbed for running the model validation tests, it is necessary to design the exact form of the perturbation. This involves selection of the amplitude, wavelength, waveform, number of cycles and spacing between perturbations (length of unperturbed track). Once again, the model to be validated should be used as a tool for evaluating different track perturbation designs. The design of the track perturbations could be the subject of an extensive study in itself, using some very sophisticated analysis methods. Some general guidelines for input design, which can be applied without a great deal of effort, are provided in Table D-5.3. These guidelines should be applied as appropriate for each separate model validation attempt (since all of them do not apply all of the time). There is no single track perturbation design which is ideal for all model validation test programs, so compromises are necessary in the design if it is to be used for more than one series of tests.

D-5.1.3 Use of Model Runs to Predict Test Responses

Even before the model validation has been accomplished, the best way of predicting (or guessing) what will happen in the test program is still to make trial runs using the best available model of the phenomena to be tested. In many cases, this will turn out to be the candidate model for validation (which should represent the analyst's best current understanding of these phenomena), although in some cases a validated model may be available. It is very helpful to take advantage of whatever information those model runs can offer to aid in planning the test program. Some of the uses of these pilot runs of the model to help in specifying the test inputs were already discussed in Section D-5.1.2, but there are many more ways in which pilot model runs can help in the detailed design of the test program, producing the information needed for the work which will be discussed in Section D-5.2.

The model should be run using vehicle parameters and inputs which correspond to the best available information about how the tests will be conducted. Model outputs corresponding to the measurements planned for the test program should be recorded (i.e. if vertical accelerations at the front and rear of

Table D-5.3

Considerations Affecting Design Of Track Perturbations

DESIGN FEATURE	HOW TO SELECT IT
TYPE OF TRACK GEOMETRY PERTURBATION	RELEVANCE TO PERFORMANCE ISSUE(S) ADDRESSED BY THE MODEL BEING VALIDATED AVOID TYPES WHICH COULD STIMULATE MODES NOT REPRESENTED IN THE MODEL CONSIDER BOTH SINGLE AND COMBINED TYPES, AS APPROPRIATE FOR PERFORMANCE ISSUE(S)
AMPLITUDE	LARGE ENOUGH TO REPRESENT LARGEST PERTURBATION EXPECTED IN REVENUE SERVICE LARGE ENOUGH TO PRODUCE MAXIMUM VEHICLE RESPONSE (MOST EXTREME PERFORMANCE REGIME) EXPECTED IN PRACTICE NOT SO LARGE AS TO BE UNSAFE VARIETY OF AMPLITUDES FOR DIFFERENT NONLINEAR REGIMES
WAVELENGTH	REPRESENT WAVELENGTHS EXPECTED IN REVENUE SERVICE (TRACK SECTIONS) PRODUCE INPUT FREQUENCIES AT NORMAL OPERATING SPEED CORRESPONDING TO NATURAL FREQUENCIES OF IMPORTANT VEHICLE RESPONSE MODES AVOID STIMULATING MODES WHICH ARE NOT REPRESENTED IN THE MODEL, BUT STIMULATE ALL WHICH ARE REPRESENTED STIMULATE BOTH ODD AND EVEN MODES (AS APPROPRIATE TO MODEL) WITH WAVELENGTHS OF APPROPRIATE MULTIPLES OF VEHICLE CHARACTERISTIC DIMENSIONS-i.e., 1.0 AND 1.5 TIMES WHEELBASE OR TRUCK CENTER SPACING
WAVEFORM	SHOULD INCLUDE MULTIPLE FREQUENCY COMPONENTS TO STIMULATE MULTIPLE RESPONSE MODES (NOT SINUSOIDAL) PIECEWISE LINEAR SHOULD BE EASIEST TO INSTALL AND MAINTAIN ASYMMETRICAL FORM IS PREFERABLE TO SYMMETRICAL (RATHER THAN)
NUMBER OF CYCLES	SINGLE CYCLE TO SHOW EFFECT OF A STRONG TRANSIENT OR TO STIMULATE HUNTING MULTIPLE CYCLES NEEDED TO SHOW FREQUENCY RESPONSE, GROWTH OF RESONANCES OR EFFECTS OF PERIODIC INPUTS (SUCH AS BOLTED TRACK JOINTS) IF MULTIPLE CYCLES ARE NEEDED, AT LEAST FIVE SHOULD BE INCLUDED TO GET PAST STARTING TRANSIENTS (FOR SYSTEMS WHICH ARE NOT HEAVILY DAMPED)
SPACING BETWEEN PERTURBATIONS	LONG ENOUGH TO ALLOW ALL EFFECTS OF PERTURBATION TO DIE DOWN (DEPENDS ON AMOUNT OF DAMPING OF TEST VEHICLE) SHOULD BE AS SMOOTH AS POSSIBLE (CLASS 6+)
COMPLIANCE	SEVERAL VALUES, VERTICAL AND/OR LATERAL, AS APPROPRIATE FOR THE MODEL'S PERFORMANCE ISSUE(S), TO REPRESENT TYPICAL OPERATING CONDITIONS IN WARM AND COLD WEATHER

the vehicle will be measured, those are the quantities which should be recorded, rather than body vertical and pitch accelerations). If the test measurements are to be sampled rather than continuous, the model outputs should be sampled at the same rate. The simulation time step for the model run should be substantially smaller than this sampling interval (at least by a factor of 10) so that the model "appears" to be essentially continuous. Limitations of the anticipated instrumentation (filtering effects, saturations, etc.) should be modeled so that the recorded outputs of the model are as close as possible to the data which will be gathered in the validation tests.

These pilot runs for the proposed test conditions should be used to check that the responses include the desired performance regimes and to try to guarantee the adequacy of the proposed instrumentation. This includes helping to select the waveform for track perturbation inputs by observing the model's responses to several different input waveforms and developing the specifications for instrumentation (especially dynamic range and bandwidth). The results of the pilot modeling runs can be the central references for the discussions among the test planners, instrumentation, and field test engineers in developing the test program. The changes which are suggested in the test planning sessions can be reflected in a later pilot run of the model to help make certain that the desired results are achieved. The use of pretest modeling can thereby reduce the guesswork involved in choosing instrumentation, and should help avoid the large majority of the costly losses of data which often occur when the test output amplitudes or frequencies exceed the capabilities of the equipment.

D-5.2 DESIGN OF TEST PROGRAM

The detailed design of a model validation test program should follow the same general principles of test design as any other test program. However, there are some additional issues which should be considered specifically for model validation, but which may not apply to other testing. This section focuses on those issues which are specific to model validation testing, rather than attempting to cover all aspects of general test design.

D-5.2.1 Selection of Test Facility and Vehicle(s)

The selection of the test facility and vehicle(s) to be used is typically based on the technical requirements and on cost, convenience, and availability. There are no special constraints on the choice of facility to use for model validation testing, as long as it is able to provide the track inputs required. For test facilities using actual track, the track conditions should be well documented (with up-to-date information), or else they will need to be measured as part of the test program. Locations along the track must be identifiable for synchronization with vehicle data, and if track force measurements are needed for the validation it would be more economical to use track which was already instrumented.

Repeatability of test conditions is particularly important for model validation tests, which may favor the use of the wellcontrolled conditions offered by the major test units at the Rail Dynamics Laboratory in Pueblo, CO (the Roll Dynamics Unit, RDU, and Vibration Test Unit, VTU). However, the choice between fullscale laboratory tests and track tests, whether they be in the field or on special perturbed track, will still be made largely on the basis of availability of facilities and cost. As long as the basic requirements for track inputs and measurements are met, the selection of a facility should not be critical.

The selection of the test vehicle(s) requires a little more care because all vehicles are not equally well suited for validating a particular model. The test vehicle must be representative of the class of vehicles the model is planned to be used on (body stiffness, weight, dimensions, structural type, truck design and spacing, etc.), and it would help if it were also easy to modify so that the effects of changes in vehicle design could be evaluated without too much effort. If the model is intended for use on widely varying types of vehicles, it may be necessary to use several different test vehicles to validate the model's ability to represent the diverse vehicle characteristics. Any vehicle which is used must be accurately characterized, with its mass, stiffness, and damping characteristics, as well as its dimensions, measured or documented in advance. The modeling required for test planning cannot be done accurately until these characteristics are known. If they were not well documented in advance, considerable time and money may be needed to determine the vehicle characteristics before the model validation testing can be done.

D-5.2.2 Selection of Combinations of Test Conditions (Test Matrix)

The vehicle and input characteristics to include in the validation test program were discussed in Sections D-5.1.1 and D-5.1.2, respectively. Those discussions covered only the general question of what conditions to include, but did not suggest how to choose the specific combinations of conditions to use for each test run. The number of variables in both vehicle and input characteristics is typically so large that it is impractical to test all of the possible combinations of values of those variables (a "factorial" experimental design). The number of combinations can be reduced by use of a "fractional factorial" design, but even then the number of test cases may be impractically large.

The number of cases to be tested should be reduced by eliminating those cases which probably do not reveal anything new about vehicle performance. These are the cases which involve changes in variables which do not influence each other by much (variables which are largely decoupled from or independent of each other). Identification of the cases which can be eliminated depends on the analyst's understanding of the vehicle's dynamic interactions, aided by the results of some model runs. Generally, when their effects are decoupled, the tests can be performed for multiple values of one parameter, with all the others held constant. This involves many fewer tests than are required for the closely coupled systems, which require testing several values of each parameter for all combinations of values of the other parameters.

For example, it is not generally necessary to test all values of suspension stiffness for all levels of vehicle loading. If there were four values for each of these parameters, the complete experiment would involve 16 separate tests (for each combination of speed and all other parameters). By testing the four suspension stiffnesses for one level of vehicle loading, and the four loading levels for one stiffness, the experiment can be reduced to seven tests. As the number of parameters and the number of possible values for each increase, the potential saving in test cases increases rapidly. Continuing with the above example, by using five different operating speeds instead of only one, the total experiment would then include 80 tests. If the effects of speed on vehicle dynamics were completely decoupled from loading and suspension stiffness, this could be reduced to 11 tests. Assuming that there is some coupling, based on changes in suspension natural frequency, tests could be performed for two additional speeds, each for three different natural frequencies. These natural frequencies could be produced by different combinations of suspension stiffness and loading, with a few redundant cases (the same natural frequencies, obtained from different stiffnesses and loadings) to demonstrate the dependence on natural frequency, as compared to stiffness or loading. The two additional speeds and three natural frequencies, plus two redundant cases for one of the speeds, produce an additional eight test cases, for a total of 19 in the entire experiment.

A thought process similar to that just explained in the example should be followed when selecting the specific test cases. The most important trends and the most important combinations of influences on vehicle performance should be given priority in the design of the test program, since the cost of testing will usually make it difficult to include every case of possible interest.

D-5.2.3 Specification of Measurement Types and Tolerances

One of the main deficiencies of most existing rail vehicle test data for use in validating models is the inadequacy of the measurements which were made. This has involved omission of some important measurements and inadequate precision of others. Proper planning of a new validation test program can avoid these problems and ensure that the tests produce the required quantity and quality of data. The focus of this section is on the selection of which dynamic response variables to measure and the level of precision needed for each, and not on the selection of specific instrumentation such as accelerometers, LVDTs, force-measuring wheelsets, and wheel/rail displacement measurement devices.

The measurements which are needed for any specific model validation test program depend on the test vehicle, the model and its purpose, and the validation criterion. The minimum set of measurements needed to validate a model includes all of the variables which appear in the validation criterion, or variables which can be directly combined to represent those needed variables. It is also extremely useful to include additional measurements which can be used to check some of the key assumptions of the model. For example, a vertical ride quality (forced response) model would include body vertical and pitch accelerations as the primary variables in the validation criterion. These could be represented in the tests by measurements of the vertical accelerations at the ends of the vehicle body. A rigid-body modeling assumption could be checked by additional vertical acceleration measurements at other body locations. An accelerometer mounted at the center of the body could help identify the first body bending mode, and accelerometers at other locations could help evaluate the importance of higher body bending modes. Similar supplementary measurements can be used to check major assumptions or simplifications in other kinds of vehicle models.

Since it is impossible to predict with any advance certainty which response variables will be needed to resolve questions about modeling errors during the validation process, it can only be recommended that as many important response variables as practical be measured and recorded during the test program. These should not be restricted to those which are used in the validation criterion, but should include the variables associated with all the modeled degrees of freedom and the unmodeled degrees of freedom which appear to be most important (or which could be candidates for inclusion in the model).

Unless the test program was designed for model validation, it is unlikely that all of the variables which appear in the validation criterion would be measured and recorded. Indeed, the choice of those variables can differ greatly from the validation of one model to that of another, as already shown in Table D-3.4.

The measurement requirement which remains consistent for most vehicle dynamic model validation tests, and which distinguishes them from most other testing, is the requirement for comprehensive track geometry measurements, with automatic location detection for synchronizing the vehicle-based measurements. Although precise knowledge of track characteristics is not crucial for many types of vehicle testing, it is essential for model validation. The specific track geometry characteristics which must be known in any particular model validation program depend mainly on the dynamic process being modeled. Table D-5.4 shows which track measurements are typically needed for validating models of each dynamic process. In many cases it is not practical to make all of these measurements and the validation has to be conducted using only a portion of the track data indicated in Table D-5.4. This is not a serious problem unless the validation criterion has very tight tolerances and the model predictions are required to agree very closely with the test results.

Because test equipment and procedures are never perfect, it is important to include some redundant measurements of the most important test results in the test plan. This should reduce the common problems of missing or untrustworthy data channels, which are often not discovered until long after the tests have been completed. By then, it is too late to repeat the tests and there may not be enough other data available to permit reconstruction of the deficient data. By using the redundant measurements to replace or reconstruct the missing data, gaps in the validation can be avoided.

The accuracy of the measurements which are made in the test program directly influences the tolerances which can be satisfied in the validation criterion. The general category of "accuracy" includes all of the measurement defect categories shown in Table D-5.5. Each of these defects has different effects on the ability to validate a model. Before beginning a validation test program, the allowable level of each defect should be specified for each measurement, so that the measurements are "good" enough to be useful without becoming prohibitively expensive. In each case, the random and unknown defects cause more serious problems than those which are known and can therefore be compensated for. Sometimes it is possible to identify bias errors from the available data (especially when the data include static or statistically stationary conditions), and then eliminate them systematically.

If the frequency response of the measurement system is determined accurately, it may be possible to compensate for bandwidth limitations to some extent by appropriate digital filtering of the test data (i.e., artificially "boosting" the response at frequencies where it is "weak"). This must be done very carefully to avoid introducing further errors or amplifying noise. The bandwidth of the measurement system should be matched as closely as possible to the bandwidth needed for the validation. In other words, a bandwidth which

Table D-5.4

Track Measurements Needed To Validate Models Of Each Dynamic Process

DYNAMIC PROCESSES (PERFORMANCE ISSUES)	PROFILE	CROSS-LEVEL	ALIGNMENT	GAUGE	SURFACE ROUGHNESS	CURVATURE	GRADE	COMPLIANCE*	RAIL HEAD CROSS-SECTION*
TWIST AND ROLL		(X)				X			
HUNTING (LATERAL STABILITY)	X	X	X	(X)	X	X		X	X
SPIRAL NEGOTIATION	X	(X)	(X)	(X)		(X)	X	X	X
STEADY-STATE CURVING	X	(X)	X	(X)		(X)		X	X
DYNAMIC CURVING	(X)	(X)	(X)	(X)	X	(X)	X	X	X
PITCH AND BOUNCE (VERTICAL FORCED RESPONSE)	(X)	X			(X)			X	
YAW AND SWAY (LATERAL FORCED RESPONSE)	X	(X)	(X)	(X)				X	X
STEADY-STATE BUFF AND DRAFT						X	X		
LONGITUDINAL TRAIN ACTION						(X)	(X)		
LONGITUDINAL IMPACT						X	(X)		

* THESE ARE NEEDED FOR THE BEST VALIDATION RESULTS, BUT BECAUSE OF THE DIFFICULTY OF MAKING THE MEASUREMENTS THEY ARE VERY RARELY AVAILABLE.

X = ESTIMATES NEEDED

(X) = PRECISE MEASUREMENTS NEEDED

Table D-5.5

Categories Of Measurement Defects Which Degrade "Accuracy"

CATEGORY	DESCRIPTION
SCALE ERRORS	PERCENTAGE ERRORS IN CHANGES FROM ONE CONDITION TO ANOTHER
BIAS ERRORS	OFFSETS IN ZERO-SCALE POINTS
DYNAMIC RANGE LIMITATIONS	DYNAMIC RANGE OF RESPONSE EXCEEDS DYNAMIC RANGE OF MEASUREMENT SYSTEM
INADEQUATE SIGNAL-TO-NOISE RATIO	SIGNAL LEVEL IS TOO LOW RELATIVE TO NOISE IN MEASUREMENT SYSTEM
BANDWIDTH LIMITATIONS	RESPONSE FREQUENCIES NEEDED FOR MODEL VALIDATION EXCEED FREQUENCY RESPONSE OF MEASUREMENT SYSTEM
KNOWN DETERMINISTIC ERRORS	SIGN INVERSIONS, INCORRECT CALIBRATIONS AND OTHER ERRORS DETECTED AFTER TESTS

is too large can sometimes cause as many problems as one which is too small. If the bandwidth is too large, it should be reduced by filtering before any frequency response calculations are made. The data provided by the instrumentation and processing should ideally lie in the frequency band which the model represents, and should not extend much beyond that unless there are some significant dynamic responses to be found there.

The scale errors, dynamic range limitations, and signal-to noise ratio problems are the most troublesome to eliminate of the six shown in Table D-5.5. Uniform scale errors may be correctable if calibrations are done very frequently and carefully throughout the test program, but nonuniform (nonlinear) scale errors are almost impossible to eliminate. The allowable limits on scale errors must therefore be chosen to be significantly less than the corresponding tolerances in the validation criterion (about 1/3 being a good proportion in most cases). Dynamic range and signal/noise problems are the upper and lower levels of the same problem. This problem should be avoided by using pilot runs of the model being validated to predict the maximum and minimum amplitudes of each variable to be measured. The measurement system should then be specified to have a dynamic range which enables it to capture the complete dynamic range expected in the tests, without overloading or saturating at the maximum values or dropping below the noise level at the minimum values.

The general performance specifications for measurement system "accuracy" (including bandwidth, dynamic range, and noise levels) must be compared to the specifications of available instrumentation in order to select the instrumentation to use in the test program. If these performance specifications cannot be met using available or affordable equipment, it may be necessary to redefine the validation tolerances or even the scope of the validation effort to fit within the limitations of the equipment which will be used. This may even require a reevaluation of the benefits and costs of the entire effort if it is necessary to reduce the scope of the model validation substantially. It would be better to terminate the work at this point if its value does not exceed its cost rather than proceeding to completion with data known in advance to be inadequate.

D-5.2.4 Definition of Number and Lengths of Tests Needed

Although the selection of conditions to be tested has already been covered in Section D-5.2.2, that discussion did not include the selection of the number and lengths of the tests needed at each condition. These are important considerations because an insufficient quantity of data can cause as much trouble for a model validation attempt as insufficient quality of data. A strictly "correct" choice of the quantity of data which should be gathered for each test condition would require a sophisticated statistical analysis (see, for example, Ref. [2]), which is beyond the scope of what can be covered here. Therefore, this section covers some rough guidelines for deciding the number of tests to run and their lengths.

Each additional test run increases the expense and time required for the test program, and the longer each individual test run, the more costly it will be. At the same time, increasing the length and number of test runs makes it possible to have greater confidence in the model validation. There is thus an important trade-off between the cost of the validation test program and the confidence which can be placed in the validated model. The design of the test program should either maximize the confidence which can be provided for the resources available or minimize the resources used to validate the model to the needed level of confidence.

Some of the basic issues of test length were touched upon in Section D-5.1.2, in the discussion of input design for perturbed track. It is just as important to choose the proper length of test for operations on "normal" or "smooth" track, which would be used for validating forced response, lateral stability, or steady-state or dynamic curving models. In these cases, the test section must be at the very least longer than several cycles of the longest wavelength track input (such as the length of a rail or elevated structure). The test must also last at least long enough for the vehicle to reach a steady-state response (for stable configuration) and to experience several cycles of its lowest frequency response mode.

Some additional requirements on test length should be met for lateral stability (hunting) model validation and for data which are to be used in spectral analyses (frequency response modeling). Because hunting can be an intermittent phenomenon, it is important that test conditions be maintained long enough to allow the hunting behavior to come and go. This may require tests lasting several minutes, rather than several seconds. (Recent experimental results also seem to indicate that it is very important to maintain identical track conditions when testing the effects of changes in vehicle design or speed on hunting [5].

In addition to the minimum test length requirement based on the need to observe the effects of several cycles of all inputs and outputs, data to be used for frequency response model validation must also be collected in long enough records to eliminate potential noise problems. The lengths of these records should be in multiples of 2^n samples each (such as 1024 or 2048 samples) to facilitate the use of Fast Fourier Transform (FFT) procedures. There should also be enough independent test records (typically on the order of 10) of the appropriate length so that their spectra (Fourier transforms) can be averaged together to reduce the jaggedness produced by random factors in the experiment (noise). The greater the number of spectral estimates which can be averaged together, the closer will be the agreement with the predictions of a model which does not include noise.

The independent test records do not necessarily have to come from separate test runs. One long test run could produce several separate test records, whose spectra could then be averaged together. In general, however, it is desirable to have two separate runs of the same test condition rather than a single run (even if the single run is twice as long). The separate test runs can be used to check the repeatability of results, and also provide a distinct reliability advantage. If some of the instrumentation should malfunction on one test run, all data for that condition would not be lost because of the availability of the independent data set from another run. Duplicate runs are generally difficult to justify for all test conditions because of their expense, but they should certainly be used on some of the most important individual test cases to reduce the probability of a loss of data and to provide proof of the repeatability of results.

D-5.2.5 Coordination of Vehicle On-Board Measurements with Wayside and Track Geometry Measurements

The importance of coordinating vehicle on-board measurements with wayside and track geometry measurements for model validation has been discussed several times already. This coordination can only be accomplished if the tests are planned with it in mind. Instrumentation and data channels must be provided to record the location of the test vehicle precisely throughout the test section. The on-board vehicle measurements can then be coordinated with the track measurements during post-test data processing.

The most popular method for identifying vehicle location on a track seems to be the automatic location detector (ALD) system. ALD markers are placed at known locations along the track and are detected by a sensor on the vehicle when it passes, producing an impulse on one of the data channels. During the posttest processing, the ALD pulses can be used to match the time scale of the vehicle test record with the ALD marker locations alongside the track. The time intervals between ALD pulses can be matched to track locations by using the measured speed of the vehicle and integrating it to compute distance traveled, or by using counts of axle rotations from a shaft encoder mounted to a vehicle axle. In either case, it is generally necessary to "stretch" or "squeeze" one set of measurements (either vehicle or track) so that it matches the other. This requires interpolation between samples, which can become a laborious calculation. When vehicle speed varies, it is particularly important to have the corrections for "stretching" or "squeezing" adjusted frequently to maintain accurate synchronization.

The choice of which set of measurements to adjust and which to hold constant (time on vehicle or distance on track) depends on the nature of the test and the other data processing which must be done. If there are many more channels of one set than the other, it is most economical to adjust the smaller set. If frequency domain processing is to be done on the data, the times between samples must be held equal and the spatial (i.e. track) data must be stretched and squeezed. However, if the waveforms of responses to a track perturbation at several different speeds are to be compared, it would be better to plot all results at equal spatial intervals, stretching and squeezing the time scales.

The process of coordinating the vehicle and track measurements should be computerized as much as possible. If many sets of test data are to be processed, the potential savings in analyst labor hours could justify devoting substantial effort to developing software and graphic display capabilities to streamline the analyst's job of matching the vehicle and track data. On the other hand, if there are only a few test data sets involved, it should be more economical to have the analyst visually inspect plots of the data and separately execute the adjustments between each pair of consecutive ALD impulses.

D-5.2.6 Data Recording Considerations

The selection of the method for recording validation test data should be based on the same consideratons as would apply for any other test program. The choice of the storage medium (magnetic tape, semiconductor memory, floppy disks, etc.) has a variety of implications for cost, convenience, capacity, speed, and durability. This choice is generally dictated by the circumstances of the group which is doing the testing, and has little influence on the ultimate outcome of a model validation effort, as long as the capacity and speed are adequate.

The data recording issues which can have a significant impact on the success of a model validation are those of sampling, filtering, and preprocessing. If these are not done properly, the value of the data produced by the test program can be destroyed and, what's worse, this destruction may not even be detectable. Seriously incorrect conclusions could even be drawn from the data because of the lost information. The most important problem to guard against is the loss or corruption of high frequency data.

In order to specify the data acquisition system properly, it is necessary to take into account the following set of frequencies:

f_1 = the highest frequency expected to be experienced by the sensor in the test

f_2 = the highest frequency the sensor can detect adequately

f_3 = the sampling frequency or number of samples per second which are recorded

f_4 = the highest frequency needed for the model validation (for comparison with the model's predictions)

f_5 = the anti-aliasing filter cutoff frequency

Each of these frequencies could be evaluated separately for each instrumentation channel, but that is not advisable because practical considerations require that all sampling and processing be done at the same frequency for all channels. Therefore, the highest frequency requirements should generally be applied to all the channels except where this leads to impractical instrumentation specifications on some of the channels or where the highest frequencies are only needed for a very limited fraction of the total number of channels.

Typically, f_1 will be the highest frequency of those listed, however it is neither practical nor necessary to measure the highest frequencies experienced in the tests, corresponding to vibrations of high (unmodeled) structural modes. The frequency f_4 is chosen to suit the anticipated use of the validated model. In other words, f_4 is the highest frequency for which the model is to be validated, and its selection should therefore precede the design of the test program. The sensor or transducer must be able to detect a maximum frequency, f_2, at least as high as f_4 without noticeable rolloffs (loss of amplitude) or phase shift. This typically requires that it have a bandwidth specification (based on 3 dB rolloff) somewhat higher than f_4.

The sampling frequency f_3 should generally be at least five times f_4 to produce good results. If it is less than that, the higher frequency information will be lost or disguised (aliased). If the sampling frequency is less than $5f_2$, which is often the case, it is necessary to use an "anti-aliasing" filter before sampling the transducer's output signal. This filter should be chosen to have a cutoff frequency of $f_5 = f_3/5$ (typically, $f_5 \approx f_4$) in order to eliminate signals which are at too high a frequency to be sampled correctly. Failure to do this could lead to the "folding" of the higher frequency responses into lower frequencies, making the responses appear erroneously large at those frequencies. In fact, it is advisable to low-pass filter <u>all</u> of the data channels before sampling to eliminate the effects of random noise at frequencies above those which are needed or usable (f_4).

One final data recording consideration is the selection of which data channels to retain as sensed and which to combine in preprocessing prior to recording. Combining and preprocessing data can reduce the number of channels and the overall data processing effort and expense. However, it also reduces the amount of information available for later use. Any measurement which could be used at a later stage of the validation process (to cross-check other measurements, to evaluate the importance of an unmodeled response mode, etc.) should be retained in the recorded data. This would typically include any measurement which directly represents the dynamic response of the vehicle or its major components. By this argument, acceleration and suspension displacement measurements would be retained as measured, while the outputs of individual strain gauges on an instrumented wheelset could be combined in preprocessing so that only the calculated force components would be recorded. Even this level of preprocessing would run the risk of obscuring problems which could be caused by one defective strain gauge. Therefore, it should not be attempted unless the wheelset instrumentation has been proven to be thoroughly accurate and reliable.

D-5.2.7 Choice of Test Vehicle Consist

The conditions which are used for model validation testing should reflect as closely as possible the conditions which the model is expected to represent. With rare exceptions, railcars do not operate by themselves, isolated from all other cars. At the same time, dynamic models of individual railcars do not generally include the influences of adjacent cars (except for the longitudinal dynamics and whole-train models, which focus on those influences). Therefore, the test consist should normally be arranged to avoid strong interactions among cars by maintaining fairly uniform weights and geometry among all the cars.

The test consist should be as short as practical to reduce costs and dynamic interactions. For validating models of freight or passenger cars, there should be at least one car coupled to each end of the test car, and the test car should not be coupled directly to the locomotive. Validation testing of locomotive models may include operations with several different trailing loads, as well as operations as either the front or rear member of a tandem pair, or operation in the middle of a consist.

These general guidelines for choosing the test consist do not apply in all cases. The most obvious exception is for validating models of the dynamics of an entire train, in which case the choice of consist is one of the principal variables to be tested, and an integral part of the experimental design. In tests which are used to validate models of some large-amplitude disturbances such as hunting and twist-and-roll, it may be desirable to surround the test car with some very different cars which do not respond the same way. For example, a heavily loaded, high center of gravity car used to validate a twist-and-roll model could be tested between lightly loaded low center of gravity cars which would be unlikely to interact in roll with the test car. Similarly, a lightly loaded car used to validate a hunting model could be tested between heavier cars which would not be expected to hunt until higher speeds were reached. In this way, the hunting of the test car could be isolated from the possible influences of hunting of adjacent cars.

The choice of test consist will be strongly influenced by the availability of cars of different types, and it may not be feasible to follow the guidelines suggested here from test to test. If that is the case, the most important thing to remember about the consist is that it should remain as consistent as possible from test to test. If the consist is changed a lot, it may be impossible to isolate the effects of those changes on test vehicle response from the effects of the changes in the test vehicle or its operation which are represented in the model which is being validated. The results of that would be to confuse the validation process by destroying the control of the experimental test program.

D-5.3 EXECUTION OF MODEL FOR CASES WHICH WERE TESTED

The conduct of the model validation tests themselves is not covered here. The testing should proceed according to the design principles covered in Sections D-5.1 and D-5.2, using accepted experimental test techniques and normal care and prudence. As the test program progresses, the performance which is observed should be compared qualitatively with that which was expected. If there are significant discrepancies, the test plan should be adjusted to make sure that there are enough test cases in each performance regime to complete a successful model validation.

Once the testing is complete, the model should be run for the exact conditions which were experienced in the tests (even if they did not correspond completely to the test plan). The track geometry which was actually measured should be used as input, and the vehicle speed should reproduce that which was measured. Automatic location detector (ALD) impulses in the track geometry data should be written into the output of the model runs, to help indicate precisely where the modeled vehicle is at each time interval.

The best available set of vehicle characteristics data should be used for the model runs. Since not all vehicle parameters may be known with high confidence, the sensitivity of the model's predictions to a range of values for the uncertain parameters should be demonstrated using several model runs to try to represent the test conditions. If the results are consistent for these runs, one can be confident that the uncertainties are not significant. However, if there are significant differences among the results of these runs, it will be necessary to evaluate carefully the alternative choices of parameter values during the comparison process (Chapter D-6).

The model outputs should be processed in exactly the same way the test data were, including sampling rates, filtering, instrumentation errors, biases, and noise levels (if known). In particular, if the test instrumentation or data processing imposed any limitations of bandwidth or dynamic range, these limitations must be modeled as closely as possible to produce comparable saturations, phase shifts, and other filtering effects on the simulated data. This helps to ensure that the comparison of the model and test data is valid (i.e., comparing "apples and apples" rather than "apples and oranges").

The model outputs should be reduced to a form compatible with that available from the test data (the same statistics of the same response variables). Once this is done, the comparisons between model predictions and test data, using the chosen validation criterion, can follow directly. Instructions for the comparison process follow in Chapter D-6.

D-6. COMPARISON OF MODEL PREDICTIONS AND TEST RESULTS

The final phase of the model validation process is the comparison of model predictions with test results. It is this activity, more than any other, which makes model validation special. This is also the most complicated part of the process to understand because of the possibility of multiple iterations through some of the steps.

D-6.1 HIERARCHICAL ORGANIZATION OF COMPARISON PROCESS

Because the comparison of model predictions and test results can involve considerable variety in test conditions and vehicle characteristics (as specified in the experimental design), this comparison process must be carefully structured to avoid confusion. The recommended hierarchical structure is shown schematically in Figure D-6.1, as a simplified flowchart. In computer terminology, this structure would be referred to as "nested loops," with an "inner loop" for changes in test conditions and an "outer loop" for the variations in the test vehicle.

The structured analysis of Figure D-6.1 is designed to separate clearly the influences of operating conditions on vehicle response from the influences of design changes. In order to clarify the separation, the entire process of comparison and model adjustment (to be described in Section D-6.2 and Figure D-6.2) must be completed for each individual test condition of each vehicle before proceeding to any other condition. Similarly, all of the comparisons for each vehicle design (the "inner loop") must be completed before working with the next vehicle design. In this way, it is possible to demonstrate whether the same model structure can be used to represent each of the vehicle designs or whether it is necessary to change the model structure when the vehicle is modified.

Figure D-6.1 has been designed to highlight the general hierarchy of the comparison process, while ignoring some of the important details. Many of these details, which are embedded in the block called "Compare test results and model predictions," do not affect the general flow of the comparison process and are described thoroughly in Section D-6.2.1. The other important omissions are the repeated runs of the final adjusted model for all of the

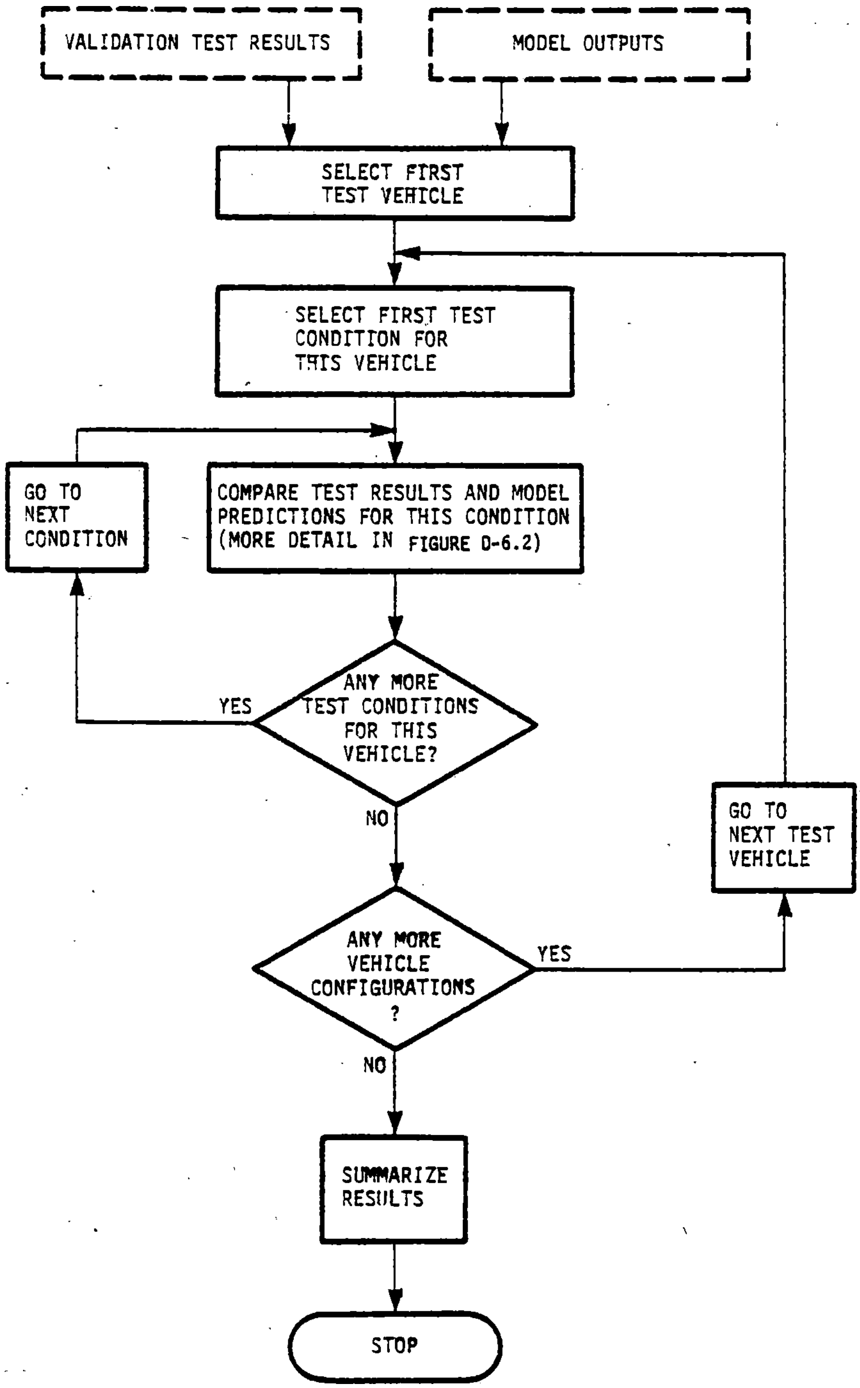

Figure D-6.1 Simplified Schematic of Hierarchical Organization of Comparison Process

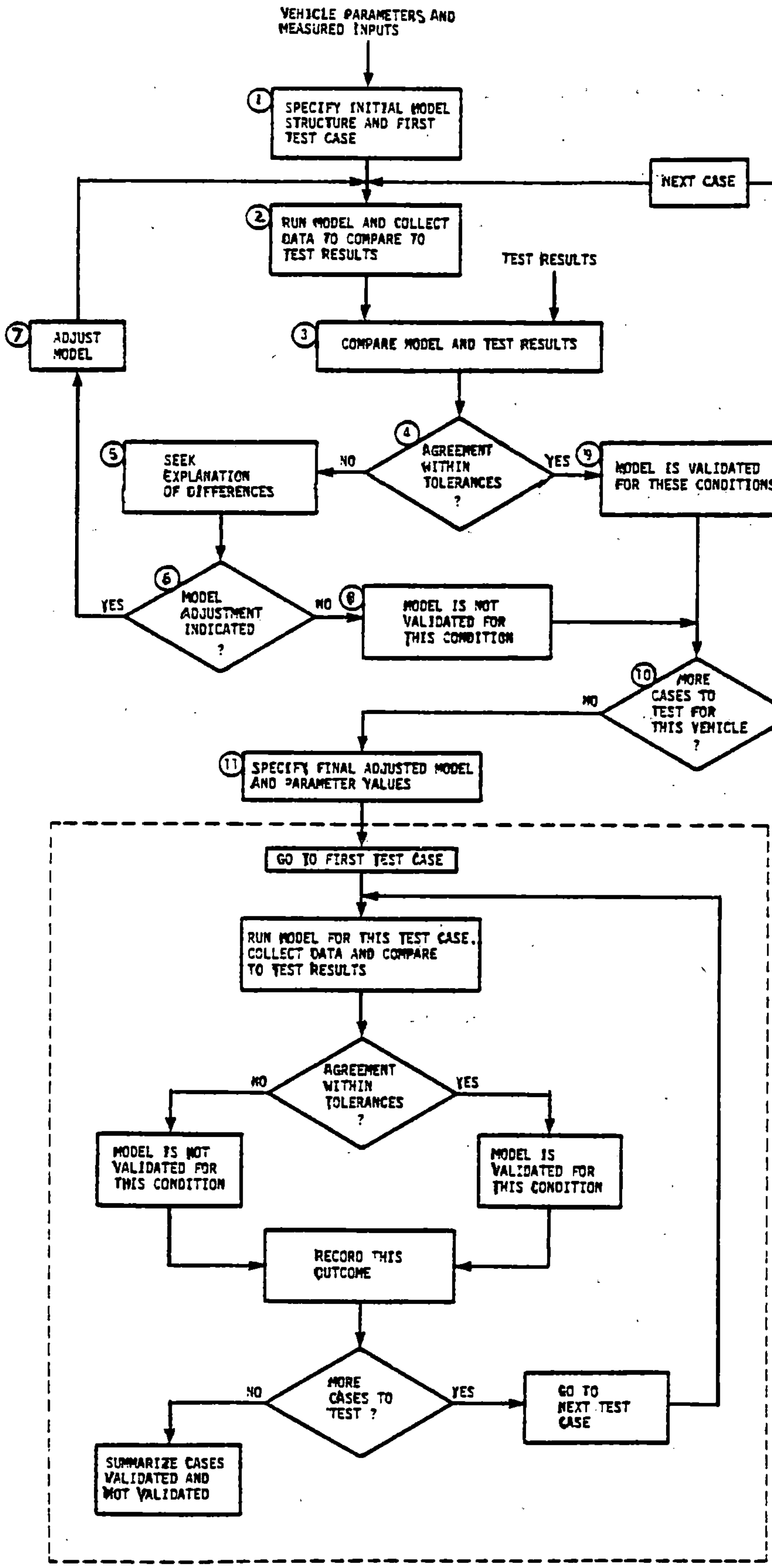

Figure D-6.2 Flowchart of Comparison Process for One Vehicle Design conditions which were tested, to be described in Section D-6.2.2. This extra set of model runs is needed to verify that the adjustments which had to be made to the model to enable it to reproduce the later sets of test conditions did not lead to a loss of the ability to reproduce any of the earlier test conditions.

The sequence of test conditions used in the comparison and model adjustment process can have a noticeable effect on the success of the validation effort. It is best to start with the simplest conditions to model, those with the smallest response amplitudes and most nearly linear characteristics. Once the comparison between the model predictions and test results is completed (and usually found satisfactory) for these conditions, the larger amplitude cases with nonlinear responses can be treated, starting with simple isolated nonlinearities and later proceeding to more complicated coupled nonlinearities. By following this sequence, the complexity of the model is not increased any faster than absolutely necessary and the experience gained with the simpler cases can be used to help decide what form the model adjustments should take in order to represent the more difficult conditions while not losing agreement for the original simple conditions.

D-6.2 COMPARISON FOR FIRST TEST VEHICLE

Following the hierarchical organization of the comparison process as just described, the entire process should be applied for the first test vehicle before looking at the results for any of the other vehicles. This process is described by the flowchart of Figure D-6.2, each step of which will be explained here. The previous steps in the model validation procedure produce the test results, measured inputs (track geometry) and vehicle parameter values which are needed before the comparison process can proceed. The initial model structure is specified in advance (at the time one decides to try to validate a model) and the first test case to validate is chosen according to the reasoning already described in Section D-6.1. This completes the preliminaries included in Block 1 of Figure D-6.2.

The model should be run using the measured inputs and test conditions, and outputs of the same form (see Section D-3.3.2) as the results obtained from the test should be recorded. The running of the model was already discussed in Sections D-4.4 and D-5.3. The important thing to add here is that the model runs appear in Block 2 of Figure D-6.2, inside the loop for model adjustments. This means that if it is necessary to adjust the model to improve its prediction of the test results, the new adjusted model must then be run and its predictions compared with the test results. Thus, it may be necessary to go through several runs of adjusted models for some of the test cases, and not just the single run per case discussed before in Sections D-4.4 and D-5.3.

The model predictions and test results are compared in Blocks 3 and 4 of Figure D-6.2, using the validation criterion chosen earlier (Section D-3.3). Each pass through the comparison process applies to one test run, and for that specific condition and the model being tested the validation criterion is either satisfied (Block 9) or not satisfied (Block 5). Recall that this evaluation is based on the unique validation criterion which was defined in Section D-3.3, based on the anticipated model purpose. This criterion incorporates the selection of response variables, statistics, tolerances and mathematical form needed to demonstrate model validity. The most important stage of the comparison process occurs when the criterion is not satisfied and the reason for this must be sought (Block 5), leading to a decision about whether to adjust the model to improve its agreement with the test results (Block 6).

Finding the sources of the differences between the model predictions and test results, which is probably the most difficult step in the entire model validation process, is described in Figure D-6.3. This requires a thorough understanding of the test conditions, instrumentation, data processing, model assumptions, and the likely effects of making different assumptions. The exact nature of the differences must be defined, and it is then necessary to apply engineering judgement to decide whether these differences were caused by (1) experimental errors, (2) measurement system or data processing problems, or (3) deficiencies in the model. This decision may be quite difficult to make on the basis of the available evidence, but when there is a reasonable doubt about whether the problems could have been caused by experimental or data processing errors, they should be assumed to be model deficiencies in order to be conservative.

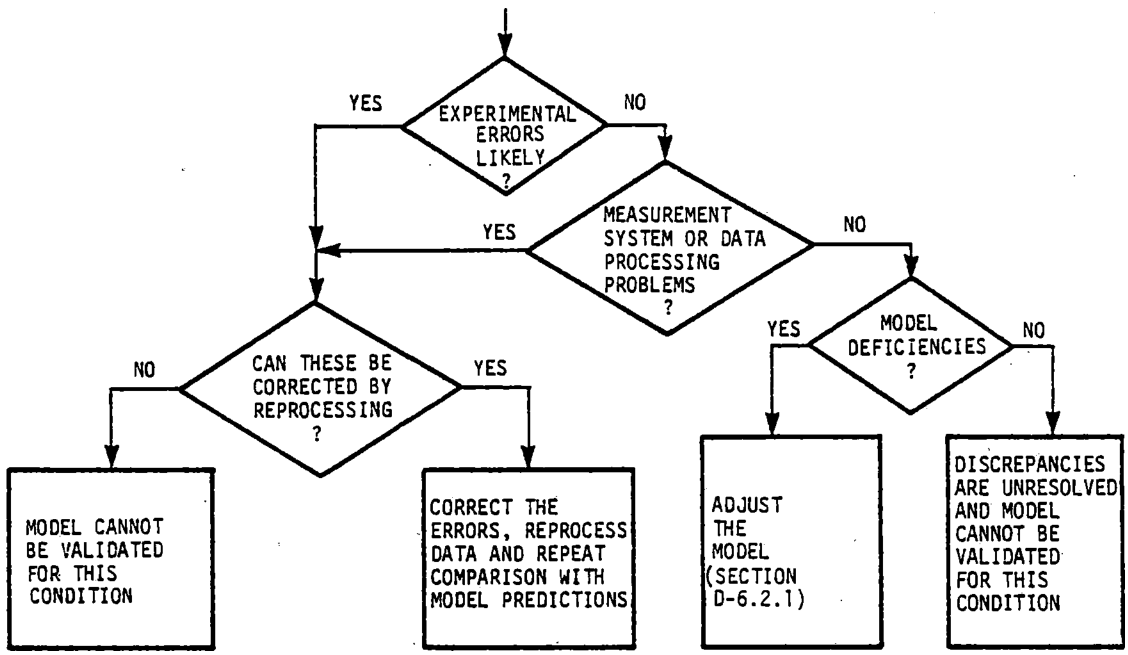

Figure D-6.3 Finding the Sources of Differences Between Model Predictions and Test Results

If the problems fall under category (1) or (2) above, model adjustments should not be considered and the next step to take depends on how well the specific problem can be identified. If it can be identified clearly enough to be corrected by the analyst working on the model validation, the correction should be made in the test data (reprocessing) and the comparison with the model predictions (Block 3 of Figure D-6.2) repeated. In most cases this is not possible and the model must be found "not validated" for the condition in question (Block 8).

D-6.2.1 Model Adjustments to Improve Agreement

The more challenging cases arise when the differences between test results and model predictions appear to be caused by deficiencies in the model. Some specific deficiencies must be identified before any model adjustments can be attempted. The types of deficiencies which must be considered include:

- incorrect parameter values
- omission of important elastic response modes
- omission of significant degrees of freedom
- linearization of important nonlinearities
- incorrect form for nonlinearities
- inclusion of unnecessary nonlinearities or degrees of freedom
- incorrect representation of vehicle elements (damping, stiffness, inertia, deadband, friction, hysteresis, etc.).

Deciding which of these deficiencies produced the observed problems requires a good understanding of nonlinear vehicle dynamics, which cannot be conveyed quickly in a report such as this. The decision can be made easily if the right vehicle response variables were measured in the test program.

The modeling deficiencies will need to be corrected by adjustments to the model structure and/or parameter values. These adjustments must be based on <u>causal</u> reasoning, directly related to the physical laws which govern vehicle dynamics. This means that vehicle parameter values which are known with good accuracy from the test program (dimensions, some masses and stiffnesses) should not be changed simply to improve the model's agreement with test data. The disagreements which remain should be attributable to other model deficiencies, and it is those which should be modified. If the original assumed parameter values were uncertain, they can be adjusted cautiously, relying on engineering judgement about the physical reasonableness of the new values of masses, stiffnesses and geometry which are implied by the adjustments. Adjustments to the model structure should be justified by the same kind of causal arguments needed for parameter value changes. In other words, the new model structure should make physical sense, relying on the physical laws which appear to be acting on the vehicle, rather than trying to gain a mathematical "best fit" to the test results. The "best fit" technique may work for one case, but unless there is a physical basis for the model adjustment it will not work for any of the other cases.

Some general guidance for identifying and correcting the different types of model deficiencies is offered in Table D-6.1, although these suggestions are not meant to replace the need for direct experience working with the test data and models. The model adjustments which are simplest to make and those which lead to simplification of the model should be attempted before the more difficult and complicated adjustments.

When several of these modeling deficiencies are present simultaneously, it may be very difficult to find and correct them. In particular, correcting any one of them would still leave significant discrepancies between the model and test results, and it would not be apparent that one of the problems had indeed been eliminated. There is no simple way of avoiding this difficulty for any individual test condition. However, comparisons of the results for a variety of different test conditions can reveal the separate effects of the different model shortcomings. This means that it may not be possible to complete the model adjustment process for each test condition totally independently of all the other conditions. Some of the model adjustments can only be put in practice after a substantial set of results has completed the comparison process. For example, an aspect of vehicle performance which is velocity dependent cannot be validated or invalidated using a single test case at one speed. Rather, the trend of performance must be tracked through several separate test cases at different speeds. It may not be possible to validate the

Table D-6.1
General Guidelines for
Identifying And Correcting Model Deficiencies

Incorrect Parameter Values

- simplest to adjust (no model structure change)
- evidence includes differences in natural frequency or damping values, but similar qualitative response

Omission of Important Elastic Response Modes or Degrees of Freedom

- test results show some responses not evident from model (more complicated waveforms or frequency response functions, particularly at higher frequencies).
- measurements made at well-chosen locations can easily identify the importance of these modes or degrees of freedom (to help decide whether to include them).

Linearization of Important Nonlinearities

- agreement between model and test results depends on amplitude of response
- test results show distinctly nonlinear response characteristics not predicted by model (limit cycles, jump resonances, saturations, etc,.)

Incorrect Form for Nonlinearities

- difficult to identify by general inspection of results
- may be found from amplitude dependence of comparison between model and test results
- best identified by use of detailed measurements very close to the nonlinearity

Table D-6.1 (Continued)

Inclusion of Unnecessary Nonlinearities or Degrees of Freedom

- model predictions show modes of response or nonlinear characteristics not evident in the test data

- correcting this problem also helps simplify model (linearize and/or reduce order of model)

Incorrect Representation of Vehicle Elements

- if nonlinear elements are involved, it will be equivalent to incorrect form for nonlinearities (as above)

- best revealed by significant loss of agreement between model and test when the incorrectly modeled element is modified.

form of velocity dependence assumed in the model until the cases for all the different speeds have been studied and their results analyzed.

Returning to the flow chart of Figure D-6.2, one could expect to go through Blocks 5 and 6 several times for each test case. Simple model adjustments could be made on the early passes, but still might not produce successful validations. After all of the test cases have gone through these simple adjustments, the larger patterns among the different cases should become clear enough to permit a second round of more subtle model adjustments to take place. These adjustments could then produce a close enough agreement between the test results and model predictions to validate the model for some of the conditions.

It is also important to consider the need to maintain validation for the earlier cases while making model adjustments to match the later cases. If this is not done, each case would have its own separate "validated" model, which would be useless for all the other cases. Therefore, the selection of which model adjustments to make has to be a compromise between the need to continue to represent the cases already covered and the need to match the latest case being considered. These choices are very dependent on the judgement of the analyst who is performing the validation work, and cannot be incorporated into the flow chart as well-defined rules. Once again, it is necessary to rely on the analyst's understanding of vehicle dynamics to decide which adjustments to the model are likely to have the best <u>overall</u> effect on its agreement with the test results. These adjustments should be designed to converge to a single "best" model once all of the test cases have been considered, and to avoid drastic changes to the model which may improve agreement with the latest set of test data while disturbing the agreement with all the previous data sets.

D-6.2.2 <u>Checking All Cases for Final Adjusted Model</u>

After all of the test cases for the baseline vehicle have been processed (Block 10), and a single adjusted model has been specified (Block 11), that model must be re-run for all of the test cases and the comparison between its predictions and the test results must then be used to determine the final pattern of cases for which the model has been found validated or not validated. This phase of verifying the final model form and parameter values, enclosed in the dashed box of Figure D-6.2, is needed to ensure that the model adjustments which were made to fit the later data sets did not destroy successful validations for the earlier data sets.

D-6.3 OTHER VEHICLE CONFIGURATIONS

Once the "inner loop" of Figure D-6.1 (the comparisons of model predictions and test results for a single vehicle) has been completed, it is time to do the comparisons for other vehicle characteristics (modifications to that vehicle or other vehicles). This "outer loop" set of comparisons is designed to validate the model's predictions of the influences of changes in vehicle design and loading on dynamic response. For each new vehicle characteristic, the comparison process of Section D-6.2 must be performed. If the model structure is reasonably good, these comparisons should become very easy after the first vehicle. In other words, after the validation has been completed for the first vehicle, the model should agree very well with the test results for the later vehicle characteristics if the effects of these characteristics have been well represented.

The final adjusted model developed for the first vehicle in Section D-6.2 should serve as the starting point for the comparisons with the remaining test data sets. The discrepancies between the predictions of this model and the test results for all the other vehicle configuratons should be recorded and then analyzed to reveal which trends (which vehicle changes and response variables) are not being modeled correctly. This analysis requires the same kind of thorough understanding of vehicle dynamics as the model adjustment analysis described in Section D-6.2. Here, different model adjustments may be recommended to represent the effects of vehicle changes more accurately. These adjustments should generally be designed to leave the model unchanged for the initial vehicle configuration so that the validation is not endangered. However, they will generally involve modifying the way changes in vehicle characteristics (weights, dimensions, stiffnesses, damping) are represented in the model.

The model adjustments which are made at this stage in the validation process should, like the earlier adjustments, be based on causal (physical) reasoning, and not purely on making a "best fit" to the test results. The danger involved in the "best fit" approach is that it loses track of the physical principles which produce vehicle dynamic response and may therefore obscure important differences among different vehicles. In other words, vehicles which should be represented by different model structures, because of some fundamentally different characteristics, would be forced into a single common model structure by the "best fit" approach. That would, at best, produce a model which happens to be valid for the specific vehicles and operating conditions used in the test program, but will not be valid for any other vehicles or operating conditions. Such a model would be useless, since the entire reason for developing a model is to be able to predict performance for new vehicles and conditions without incurring the expense of testing.

D-6.4 SUMMARY OF REGIMES FOR WHICH MODEL IS VALIDATED

After all of the comparisons of the test results with the model predictions have been completed, the outcome of the validation project must be summarized for later use, including information comparable to that shown in Figure D-6.4. This summary must provide the potential user of the model with the information he needs to determine whether the model should be considered valid for his intended purpose. This means that the summary should include the range of vehicle characteristics and input and operating conditions for which the model has been found valid to within the specified tolerances. The results could be made more widely applicable by including descriptions of more than one level of validation. This would involve specifying several different tolerance levels for validation, and then reporting the ranges of vehicle characteristics and input and operating conditions for which each tolerance level could be met.

The technically challenging aspect of this process is in defining the performance regimes for which the model can be considered valid using the available test conditions. A great many parameters could be used to define these performance regimes (i.e. any of the parameters which have a nonlinear influence on vehicle performance). It is necessary to isolate the most important of these influences and describe the validation in terms of these (which should be as few as possible). Typically, the most important influences are

THE MODEL

NAME (AND DEVELOPERS):
DEGREES OF FREEDOM:
PERFORMANCE ISSUE(S):
LINEAR OR NONLINEAR:
NONLINEARITIES INCLUDED:
ANALYTICAL TECHNIQUE:

VALIDATION #1

PURPOSE (ANTICIPATED USE):
VALIDATION CRITERION:

VEHICLE CHARACTERISTICS
TYPE (FREIGHT, PASSENGER, LOCOMOTIVE):
CARBODY TYPES:
WHEEL PROFILES:
TRUCK TYPES:

RESTRICTIONS ON OTHER CHARACTERISTICS:
(LENGTH, HEIGHT, MASS, STIFFNESSES, CARGO, TRUCK CENTER SPACING, CENTER OF GRAVITY, ETC.)

SPEED RANGE:

TRACK CHARACTERISTICS:
TANGENT, SPIRAL OR CURVE:
GRADE OR SUPERELEVATION:
SURFACE CONDITION:
STIFFNESS (LATERAL, VERTICAL):

PERTURBATIONS IN	MAXIMUM AMPLITUDE	WAVELENGTHS
ALIGNMENT:		
PROFILE:		
GAUGE:		
CROSSLEVEL:		

VALIDATION #2

•
•
•
•

Figure D-6.4 Example of Information Needed to Summarize Results of a Model Validation operating speed, track geometry inputs, and vehicle design characteristics, as discussed in Subsections D-3.2 and D-5.1.2. In particular, Figure D-5.2 showed an example of multiple performance regimes defined in two dimensions (speed and track roughness). In practice, the performance regimes are likely to be defined in more dimensions than two, significantly increasing the complexity of the analysis.

The type of track geometry input should not be an important issue here because most models are designed to be used with only one or two track inputs. Major differences in track inputs, therefore, usually require the use of a different model (representing the different dynamic processes produced by differing track inputs). The amplitude and frequency (wavelength) characteristics of the track inputs, in combination with the operating speed, will have a strong influence on the performance regimes and must therefore be included in the description of conditions for which the model is validated or not validated. Such a description could be as simple as separate limits on the operating speed and disturbance amplitude for which the model has been found valid (e.g. for speed below 81 mph and track perturbations of less than one inch). However, more typically the description of the limits of a performance regime must be in terms of <u>combinations</u> of speed and amplitude (i.e. $|vA| \leq x$, where v is velocity, A is amplitude, and x is the limit, or $|v| + kA \leq x$, where k is a scaling constant). The description becomes more complicated when the wavelength, λ, of track disturbances is included because the frequency at which those disturbances act on the vehicle is $f = v/\lambda$. The response of a vehicle may be very sensitive to the frequency of the disturbance inputs, and changes in the input frequency could shift the vehicle's response from one performance regime to another. The fact that velocity and disturbance wavelength both influence the disturbance (input) frequency means that the categorization of performance regimes for which a model is validated must include velocity, wavelength <u>and</u> their ratio (frequency), as well as amplitude.

The inclusion of vehicle design characteristics in the summary of model validation results is also challenging. Obviously, a model which is validated for one type of vehicle will not necessarily be valid for a very different vehicle. For example, a rigid-carbody model validated for a covered hopper car would not necessarily be valid under all of the same operating conditions for a flat car. The challenge here is to "cluster" the vehicle characteristics in such a way that the distinction can be made clear between the different vehicle types for which the model can be considered valid and not valid. If the test program is designed to include a broad enough selection of vehicle characteristics, this task is made easier. In effect, the conditions for which the model is valid can first be specified separately for each vehicle type. Then, the summary of validation conditions can incorporate the trends in the pattern of validations based on systematic changes in some important vehicle characteristics.

For example, the validation results for vehicles having several different body stiffness characteristics can be combined to show, in the summary, the dependence of model validity on that stiffness. In this way, a potential future model user can rely on the summary to learn the conditions for which the model has been validated for a vehicle of the appropriate type having a known body stiffness within the range which was evaluated. Other characteristics which could be incorporated in the vehicle description include mass, center of gravity height, truck center spacing, truck type, suspension (spring) stiffness, and friction and damping factors. These vary in importance for the different types of models and typically only a few of these characteristics need to be considered in assessing the validity of any individual model.

D-7. SUMMARY OF VALIDATION PROCESS

The process of validating a model of rail vehicle dynamics has been described at length in the preceding four chapters. The explanations of how the steps in this process should be performed may have obscured some of the basic flow from step to step. In order to make sure that this flow is made clear, the process is summarized in Tables D-7.1 through D-7.4, which refer back to the appropriate sections, tables, and figures in the earlier chapters for more detailed explanations.

Table D-7.1 outlines the preliminary steps common to all validation projects, as described in Chapter D-3. Tables D-7.2 and D-7.3, respectively, show the steps to follow when using existing test data and when designing a new validation test program (from Chapters D-4 and D-5, respectively). Table D-7.4, which is in effect a written explanation of Figures D-6.1 and 6.2, describes the step-by-step process of comparing model predictions and test results for any validation (Chapter D-6 contents). Taken together, these four tables can serve as the checklists to guide an analyst through the entire validation procedure.

Table D-7.1

Preliminary Steps For All Validation Projects

<u>DEFINE THE MODEL</u> (SECTION D-3.1)

PERFORMANCE ISSUE(S) ADDRESSED (TABLE D-3.2)

ANALYTICAL TECHNIQUE USED (TABLE D-3.1)

MODEL STRUCTURE (SECTION D-3.2.2)
- DEGREES OF FREEDOM
- NONLINEARITIES (EXAMPLES IN FIGURE D-3.2)

<u>DEFINE THE MODEL PURPOSE</u> (SECTION D-3.2.1)

ROLE IN VEHICLE DEVELOPMENT AND EVALUATION PROCESS

HOW OUTPUTS WILL BE USED (HOW QUANTITATIVE)

CONDITIONS TO BE COVERED (SPEEDS, INPUTS, VEHICLE CHARACTERISTICS)

<u>SPECIFY VALIDATION CRITERION</u> (SECTION D-3.3)

DYNAMIC RESPONSE VARIABLES (SECTION D-3.D-3.1, EXAMPLES IN TABLE D-3.4)

STATISTICS OF THE RESPONSE VARIABLES (SECTION D-3.D-3.2, EXAMPLES IN TABLE D-3.5)

TOLERANCES ON STATISTICS (SECTION D-3.D-3.3, EXAMPLES IN TABLE D-3.7)

MATHEMATICAL FORM (SECTION D-3.D-3.4, EXAMPLES IN TABLE D-3.8).

Table D-7.2

Steps For Analyzing Existing Test Data

<u>REVIEW AND SUMMARIZE DATA</u> (SECTION D-4.1)
SUMMARY OF TEST DATA (TABLE D-4.1)
INITIAL SCREENING (TABLE D-4.2)
PRELIMINARY DECISION WHETHER VALIDATION EFFORT CAN PROCEED
<u>RECONSTRUCT MISSING DATA</u> (SECTION D-4.2)
<u>TEST FOR SENSITIVITY TO UNCERTAINTIES</u> (SECTION D-4.3)
PILOT RUNS OF MODEL
DECIDE WHETHER VALIDATION IS FEASIBLE USING THE AVAILABLE DATA
<u>RUN THE MODEL FOR THE CASES WHICH WERE TESTED</u> (SECTIONS D-4.4 AND 5.3)

Table D-7.3

Steps For Designing And Running A New Validation Test Program

<u>SELECTION OF OPERATING CONDITIONS</u> (SECTION D-5.1)

- VEHICLE CHARACTERISTICS (SECTION D-5.1.1, EXAMPLES IN TABLE D-5.1)
- TEST INPUT CONDITIONS (SECTION D-5.1.2, EXAMPLES IN TABLES D-5.2 AND D-5.3)
- USE OF MODEL RUNS TO PREDICT TEST RESPONSES (SECTION D-5.1.3)

<u>DESIGN OF TEST PROGRAM</u> (SECTION D-5.2)

- SELECTION OF TEST FACILITIES AND VEHICLES (SECTION D-5.2.1)
- SELECTION OF COMBINATIONS OF TEST CONDITIONS (TEST MATRIX) (SECTION D-5.2.2)
- SPECIFICATION OF MEASUREMENT TYPES AND TOLERANCES (SECTION D-5.2.3, EXAMPLES IN TABLES D-5.4 AND D-5.5)
- DEFINITION OF NUMBER AND LENGTHS OF TESTS NEEDED (SECTION D-5.2.4.)
- COORDINATION OF VEHICLE ONBOARD MEASUREMENTS WITH WAYSIDE AND TRACK GEOMETRY MEASUREMENTS (SECTION D-5.2.5)
- SELECTION OF SAMPLE RATES FOR INSTRUMENTATION AND RECORDING (SECTION D-5.2.6)
- CHOICE OF TEST VEHICLE CONSIST (SECTION D-5.2.7)

<u>EXECUTION OF TESTS</u>

<u>EXECUTION OF MODEL FOR CASES WHICH WERE TESTED</u> (SECTION D-5.3)

Table D-7.4

Steps For Comparing Model Predictions And Test Results

1. SELECT FIRST TEST VEHICLE

2. SELECT FIRST (SIMPLEST, MOST LIKELY TO BE LINEAR) TEST CONDITION FOR THIS VEHICLE

3. COMPARE TEST RESULTS AND MODEL PREDICTIONS FOR THIS CONDITION (FIGURE D-6.2)

    3.1 IF AGREEMENT IS WITHIN TOLERANCE SET BY VALIDATION CRITERION, MODEL IS VALIDATED FOR THIS CONDITION. GO TO STEP 3.3.

    3.2 IF AGREEMENT IS NOT WITHIN TOLERANCE, SEEK EXPLANATION OF DIFFERENCES (FIGURE D-6.3).

        3.2.1 IF MODEL ADJUSTMENT IS INDICATED, ADJUST MODEL (TABLE D-6.1) AND GO BACK TO START OF STEP 3.

        3.2.2 IF MODEL ADJUSTMENT IS NOT INDICATED, MODEL IS NOT VALIDATED FOR THIS CONDITION. GO TO STEP 3.3.

    3.3 IF THERE ARE MORE TEST CASES FOR THIS VEHICLE, LOOK AT THE NEXT ONE (BACK TO START OF STEP 3). IF THERE ARE NO MORE TEST CASES, GO TO STEP 4.

4. IF MODEL ADJUSTMENTS WERE MADE, DEFINE THE MODEL STRUCTURE AND PARAMETER VALUES USED IN THE FINAL PASS THROUGH STEP 3 AS THE INTERIM VERSION OF THE MODEL INTERIM TO BE VALIDATED, AND GO BACK TO THE FIRST TEST CONDITION (SAME CONDITION AS IN STEP 2).

    4.1 RUN THE MODEL FOR THIS CONDITION AND COMPARE RESULTS TO TEST RESULTS

        4.1.1 IF AGREEMENT IS WITHIN TOLERANCES OF VALIDATION CRITERION, MODEL IS VALIDATED FOR THIS CONDITION. GO TO STEP 4.2.

        4.1.2 IF AGREEMENT IS NOT WITHIN TOLERANCES, MODEL IS NOT VALIDATED FOR THIS CONDITION. GO TO STEP 4.2.

    4.2 RECORD THE OUTCOME OF 4.1.1 OR 4.1.2 ABOVE. IF THIS IS THE LAST TEST CONDITION, SUMMARIZE ALL THE RESULTS FOR THIS VEHICLE AND GO TO STEP 5. OTHERWISE, GO TO THE NEXT TEST CONDITION AND RETURN TO STEP 4.1.

5. IF THIS WAS THE FINAL VEHICLE CONFIGURATION TESTED, THE VALIDATION RESULTS CAN NOW BE REPORTED IN FINAL FORM, SUPPLYING THE INFORMATION LISTED IN FIGURE D-6.4. IF MORE VEHICLE CONFIGURATIONS REMAIN, GO ON TO THE NEXT ONE AND RETURN TO STEP 2.

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APPENDIX D-A
SOME STATISTICAL CONSIDERATIONS IN MODEL VALIDATION

D-A.1 INTRODUCTION

The validation of rail vehicle dynamic models must be conducted within a statistical framework, either explicitly or implicitly. In order to avoid mathematical complications, the validation approach presented in the main body of this report is based implicitly on the relevant statistical issues without requiring explicit statistical calculations. Some of the most important of these statistical issues are described more thoroughly in this appendix.

The statistical approach is necessary because of the random processes and modeling uncertainties which pervade rail vehicle dynamics. The external forces which act on rail vehicles, particularly those produced by track variations, have significant random components. Furthermore, the instrumentation used in the test program introduces additional noise, ensuring that the test results are not repeatable in a deterministic way. The models of rail vehicle dynamic processes contain significant uncertainties as well because of the impossibility of precisely representing all of the physical processes at work on such complicated systems.

Statistical analysis is a powerful set of techniques for making explicit statements about how much is known and unknown (or certain and uncertain) about complicated processes. The separation of the certain from the uncertain is central to the concept of model validation. If the uncertainty about a model's prediction of vehicle performance is too great, the model cannot be validated.

The three specific statistical issues to be covered in this Appendix are hypothesis testing, ensuring the statistical significance of results and evaluating spectral analyses.

D-A.2 HYPOTHESIS TESTING

Hypothesis testing is basically concerned with evaluating the probability of drawing the "wrong" conclusion under given conditions. Because the model does not represent reality perfectly, the outputs it produces will differ from those experienced in practice. The values of an output measure produced by a model under a large number of different conditions form a probability distribution, while the outputs experienced in practice form another distribution. The hypothesis to be tested is that the model represents the performance observed in practice. Typical applications of hypothesis tests are based on comparisons of Gaussian distributions of the output measures, and involve consideration of two different types of potential errors:

Type 1: Rejecting the hypothesis when it is true (finding the model invalid when it is actually valid)

Type 2: Accepting the hypothesis when it is false (considering the model to be validated when it should not be).

The confidence coefficient x, (the probability of not making either type of error) can be specified and used to derive a confidence interval, which is the range of values of the output measure for which one can assume the hypothesis to be correct 100x% of the time.

The concepts of the hypothesis test and confidence interval can be used to determine whether the available test results will be usable to validate the model to within the tolerances specified in the validation criterion. Indeed, the validation criterion can even be defined explicitly as a hypothesis test (it is an implicit hypothesis test in the procedure outlined in the main body of this report). The choice of which confidence interval to use in the validation criterion (75% or 90% or 95%, for example) depends on what portion of the time the model must be able to correctly predict vehicle responses in order to serve its intended purpose. The one note of caution which must be observed is that the hypothesis test must be based on an assumed distribution of results, but if the results actually fit a significantly different distribution, the hypothesis test could be very misleading.

D-A.3 ENSURING STATISTICAL SIGNIFICANCE

The random, non-deterministic aspects of rail vehicle dynamic performance make it impossible for one test of limited duration to fully represent that performance. Increasing the length and number of the test runs provides more information about vehicle performance, and reduces the uncertainty about how well the test results represent the vehicle's true performance. Statistical analysis quantifies this reduction of uncertainty for estimates of single-valued performance measures such as the mean or rms, as well as for the multiple-valued measures such as frequency responses and spectra.

The random processes are assumed to have a Gaussian (normal) distribution, so that they are fully described by their mean value and standard deviation (or variance). These statistics were defined in Table D-3.5. For discussions of statistical significance, it is also necessary to consider the sample size, n, which is the number of independent samples used to calculate the statistic. The effect of the sample size on the confidence intervals is shown in standard statistics texts as:

$$|\bar{x} - \mu_x| \leq \frac{\sigma_x Z_{\alpha/2}}{\sqrt{n}} \quad \text{for the mean value}$$

and

$$\frac{(n-1)s_x^2}{\chi^2_{n-1;\ \alpha/2}} \leq \sigma_x^2 < \frac{(n-1)s_x^2}{\chi^2_{n-1;\ 1-\alpha/2}} \quad \text{for the variance,}$$

where $\overline{x}$ is the sample mean value of x

μ_x is the actual mean value of x

σ_x is the actual standard deviation of x

α is the level of statistical significance chosen for the test

$Z_{\alpha/2}$ is the standardized Gaussian distribution value for the level of significance $\alpha/2$ (that is, the value of the abscissa on the standardized Gaussian distribution at which the cumulative probability reaches $\alpha/2$).

s_x is the sample standard deviation of x

$\chi^2_{n,}$ is the 100α percentage point on the χ^2 (chi-square) probability distribution with n degrees of freedom (n independent Gaussian random variables).

The values for $Z_{\alpha/2}$ and $\chi^2_{n-1;\,\alpha/2}$ are obtained from standard tables or computer programs and the value of α is chosen as appropriate for the model validation being considered.

The expression for the confidence interval on the mean value shows that as the number of samples increases, the sample mean becomes an increasingly accurate estimate of the actual mean value. In order to reduce the confidence interval in half (or double the confidence in the sample mean value), it is necessary to take four times as many samples. The corresponding expression for the variance indicates that the width of the confidence interval is almost proportional to the number of samples (for more than a very few samples).

Similar expressions can be derived for the confidence limits of other statistics, but these become more complicated than the mean and variance calculations. For the multiple-valued statistics, the confidence intervals are derived separately for each individual value, using the mean and variance expressions already presented, and suitably modified values of n.

Following the results derived in Reference 1, for <u>correlation functions</u>:

$$\text{Var}\ [\hat{R}_x(\tau)] \approx \frac{1}{T} \int_{-\infty}^{\infty} R_x^2\ (\xi)d\xi$$

where T is the time interval covered by the data

$R_x\ (\tau)$ is the actual stationary autocorrelation function of x for a lag time τ

$\hat{R}_x\ (\tau)$ is the sample autocorrelation function value

ξ is the integration variable.

For <u>spectral density functions</u>:

$$\text{Var}\ [\hat{G}_x(f)] \approx \frac{G_x^2\ (f)}{B_e T}$$

where B_e is the bandwidth of the filter, centered at f, which is used to measure the power spectrum (B_e = 1/T for spectrum computed using FFT)

$G_x(f)$ is the actual one-sided power spectral density function of x at frequency f

$\hat{G}_x(f)$ is the sample spectral density value.

For estimates computed using the FFT, the (1-α) confidence interval for the spectral density function G(f) is defined as:

$$\left[\frac{n\hat{G}(f)}{\chi^2_{n;\ \alpha/2}} \leq G(f) < \frac{n\ \hat{G}(f)}{\chi^2_{n;\ 1-\alpha/2}}\right] \quad \text{where } n = 2B_eT$$

and the other previous definitions of the variables still apply.

For <u>coherence functions</u>:

$$[\tanh^2\ \{w(f) - (n-2)^{-1} - \sigma_w\ Z_{\alpha/2}\} < \gamma_{xy}^2(f) \leq$$

$$\tanh^2\ \{w(f) - (n-2)^{-1} + \sigma_w\ Z_{\alpha/2}\}\ ]$$

is the (1-α) confidence interval where

$n = 2B_eT$

$w(f) = \tanh^{-1} \hat{\gamma}_{xy}(f)$ is a convenient transformation of the sample coherence function $\hat{\gamma}_{xy}(f)$

$\gamma_{xy}(f)$ is the actual coherence function of x and y at frequency f

for <u>frequency response functions</u>:

$$|\hat{H}(f) - H(f)|^2 \leq \frac{2}{n-2} F_{2,\, n-2;\alpha} [1 - \hat{\gamma}^2_{xy}(f)] \frac{\hat{G}_y(f)}{\hat{G}_x(f)}$$

where $n = 2B_eT$ is the number of degrees of freedom of each spectral estimate

$\hat{H}(f)$ and $H(f)$ are the sampled and actual frequency response functions respectively

$F_{2,n-2;\alpha}$ is the 100α percentage point of an F distribution with $n_1=2$ and $n_2= n-2$ degrees of freedom

y and x are the output and input variables respectively

D-A.4 EVALUATING SPECTRAL ANALYSES

Some of the considerations which affect the use of spectral response data (record length, filtering and smoothing) were already covered in Section D-3.3 of the main body of this report. Even after the noise-related problems treated in that section have been handled, it is still necessary to be very careful about comparing measured and model-predicted spectral densities. The appearance of rail vehicle response spectra are dominated by the zeros produced by the cancellation of track inputs associated with the combination of fixed axle and truck spacings and constant train speed. These zeros can make the simulation and test output spectra look surprisingly similar at first glance even though the peaks in the spectra, which contain the majority of the information about vehicle dynamics, may differ by an order of magnitude or more. The problem is made even worse by the logarithmic ordinate scale used for plotting spectra, although that can be compensated for by careful examination of the differences in the amplitudes of the peaks in the simulated and experimental spectra. It is also advisable to run the validation tests at several speeds so that the zeros are shifted to different frequencies, permitting responses which would otherwise be obscured to become observable. This issue is covered more thoroughly in Chapter D-5, on the design of the test program.

Although output spectra are most commonly used for validating frequency domain models, there is considerable merit to the use of cross spectra, whether they be input/output or output 1/output 2. The output 1/output 2 cross spectra can be particularly helpful in reducing the need to rely on very accurate and simultaneous measurement of the inputs to the tested vehicle, although each output/output cross spectrum can only be used to validate portions of the model, rather than the entire model. The input/output spectral comparisons benefit greatly from simultaneous input and output measurements. If the track input information is only available from prior (or post-test) measurements, the loss of phase information can be significant, especially on flexible track.

APPENDIX D-B
GLOSSARY OF TERMS

<u>aliasing</u> - If test data are sampled at time intervals which are too long, aliasing produces confusion between the low- and high-frequency behavior. In particular, high-frequency responses may appear to be occurring at lower frequencies unless the data are properly filtered before sampling.

<u>analytical model</u> - An analytical model is an approximate representation of reality (such as the performance of a rail vehicle) using mathematical equations.

<u>analytical techniques</u> - Analytical techniques are the mathematical processes used to calculate the answers in analytical models.

<u>bandwidth</u> - Bandwidth is the range of frequencies within which a system responds as intended. As applied to a filter, it corresponds to the frequencies which can pass through without excessive losses. When applied to transducers, the bandwidth is the range of frequencies which are detected satisfactorily.

<u>data window</u> or <u>window function</u> - In order to perform frequency domain analyses on time-domain data, it is first necessary to smooth out the start and end of the data to avoid distorting the results. The data window is the mathematical function used to perform this smoothing operation.

<u>describing functions</u> - Describing functions are quasi-linear functions which describe approximately the transfer characteristics of a nonlinearity for an input of an assumed form (Gaussian random, sinusoidal and/or bias).

<u>dynamic range</u> - Dynamic range of a system is the range of amplitudes which can be handled satisfactorily. As applied to an instrumentation system, it extends from the minimum signal level which can be readily distinguished from the noise to the maximum signal level which can be received without saturation or distortion.

<u>dynamic regime</u> - A dynamic regime is a set of input conditions for which the trend of system performance remains characteristically the same, even though the quantitative measures of performance may change in value.

eigenanalyses - An eigenanalysis of a linear system model reduces the model to its characteristic eigenvalues and eigenvectors. These describe the characteristic frequencies, damping ratios and phase relationships of all the elements of the model, providing a complete description of the model's dynamics in a standardized form.

factorial experiment - (or fractional factorial experiment) - A factorial experiment on a system in which several different variables can be changed includes test cases for all possible combinations of the values of all the variables. This requires many test cases (for three variables, each of which could take on four different values, it would be 4^3 cases), but makes it possible to identify the effects of all the interactions among the variables. A fractional factorial experiment eliminates some of these test cases, with a corresponding reduction in the ability to identify some of the interactions.

frequency domain - The frequency domain is a mathematical way of describing a model or test data, which is derived using a well-defined transformation from the time domain for linear systems. The frequency domain approach permits compact descriptions of system performance with relatively modest computational requirements. It is particularly appropriate for evaluations of the vibration environment and ride quality, as well as for hunting analyses.

hysteresis - Hysteresis is a multivalued nonlinearity which can be produced by backlash, deadbands or dry friction in mechanical systems. The multivalued nature of hysteresis makes it difficult to analyze, because for any value of input, the output could have many different values. A hysteresis nonlinearity is said to have "memory" because the output value which is actually produced depends on the history of what has already occurred.

jump resonance - Jump resonance is a nonlinear type of system response which appears as a multiple-value frequency response characteristic. This means that the amplitude ratio and phase angle characteristics can experience abrupt changes with respect to frequency, and these changes will occur at frequencies which depend on the history of the test and on whether the frequency is being increased or decreased.

leakage - Leakage is the distortion of data which occurs when a proper data window is not used in signal processing. Leakage is manifested as a spreading of the main lobe of the frequency spectrum and the addition of spurious side lobes.

limit cycle - A limit cycle is a periodic oscillation of fixed frequency and amplitude which can only exist in a nonlinear system. The frequency and amplitude of the limit cycle do not depend on initial conditions or external inputs, although these factors may determine whether the limit cycle is excited. The limit cycle oscillation is self-sustained and distinct from a linear system's resonant response.

natural frequency - The natural frequency is the frequency of input disturbance which causes a system to respond with the maximum amplitude. If the system has very little damping, this maximum amplitude can become very large.

performance issue - A performance issue (or dynamic process) is an aspect of rail vehicle dynamic response which can be modeled independently of other aspects of vehicle responses. The different performance issues are described by different sets of equations, and typically it is not efficient to try to combine more than two performance issues in one model.

performance regime - see "dynamic regime"

piecewise linear - A line or curve is piecewise linear if it is comprised of a series of straight lines connected together. The application in rail vehicle model validation is to perturbed track design, for which the consecutive track segments may be straight, but they may be connected to each other at an angle (a piecewise linear track profile or alignment).

quasi-linear analysis - Linear analysis methods are much simpler and less costly than nonlinear analysis methods. An efficient way of analyzing the performance of nonlinear systems is quasi-linear analysis, which permits the use of some judicious linearization. The most popular form of quasi-linear analysis for rail vehicles is the application of describing functions to frequency-domain models.

resonance - A lightly damped system experiences resonance when it is subjected to input disturbances which contain some energy at the system's natural frequency. The resonance appears as a sinusoidal response at the natural frequency, with an amplitude that tends toward infinity as the system's damping approaches zero.

rms - rms stands for root mean square, which is also the square root of the mean-square. This concept is used in frequency domain analysis of random data as a measure of the overall level of a system's input or response. The mean square can be defined as the integral of the spectral density (or spectrum), represented as the area under the spectral density curve, including responses over the entire range of frequency. It is also the average of the squared values of the time history samples.

spectrum (or spectral density) - The spectrum is a description of the general frequency composition of a signal or signals (measured inputs and/or outputs). The autospectrum or power spectrum represents the mean square value of the signal as a function of frequency, providing a graphic description of the relative strength of the signal over a range of frequencies. The cross-spectrum of a pair of signals (one input and one output or two different outputs) shows the magnitude of their product and their phase angle as a function of frequency.

stationary statistics - A random process is stationary (or has stationary statistics) when its statistical description does not change with time (mean, mean square, autocorrelation, and possibly higher moments as well).

steady-state model - A steady-state model determines an equilibrium value of a system's response rather than the dynamics of its response. Its output is typically a constant value or values, rather than time-varying values.

system identification - System identification is a technology or set of techniques for determining a mathematical model of a dynamic system from measurements of its reponse to inputs. These techniques are applied to test design, model structure determination, estimation of parameter values, and model validation.

time-domain simulation - Time-domain simulation is one of the principal analytical techniques used in rail vehicle dynamics models. It involves the numerical integration, in successive time steps, of the differential equations which describe the dynamics of a system (such as a rail vehicle). The time-domain simulation, therefore, produces as output a sampled time history of the model's response.

transfer function - A transfer function is a mathematical expression used to describe the input/output relationships of linear time-invariant systems. The transfer function is defined to be the ratio of the Laplace transform of the output (response function) to the Laplace transform of the input (driving function), assuming initial conditions are zero. An individual transfer function describes the frequency domain response of one output with respect to one input. Extensions to additional outputs and inputs require additional transfer functions, so that a linear system with m inputs and n outputs would be described by a matrix of $m x n$ transfer functions.

APPENDIX D-C
MODEL VALIDATION EXAMPLE

In this appendix, the model validation guidelines and procedures are applied to an artificial example to provide a further illustration of how rail vehicle dynamics models can be validated. This example applies to the design of a special-purpose freight car for carrying vibration-sensitive cargo. The design process for such a car would require consideration of all of the performance issues, but here we consider only the validation of a model for evaluating pitch and bounce response.

D-C.1 MODEL DEFINITION

The validation to be considered is of the 6-degree-of-freedom linear model of freight car pitch and bounce shown in Figure D-C.1, for use in the preliminary evaluation of the car design concept. The analytical technique used to solve the model equations is a linear frequency domain spectral analysis. No nonlinearities are contained in this model. The model degrees of freedom are carbody pitch and bounce and the pitch and bounce of the two trucks. This information completes the model definition.

D-C.2 MODEL PURPOSE

The model is to be used for preliminary conceptual design of the railcar, in particular to determine whether a car which uses conventional freight trucks equipped with standard spring groups is likely to be able to provide the needed vibration isolation in pitch and bounce. For this purpose, the outputs will be used to establish performance trends in a general sense, but will not be used to quantify performance absolutely. The model will need to

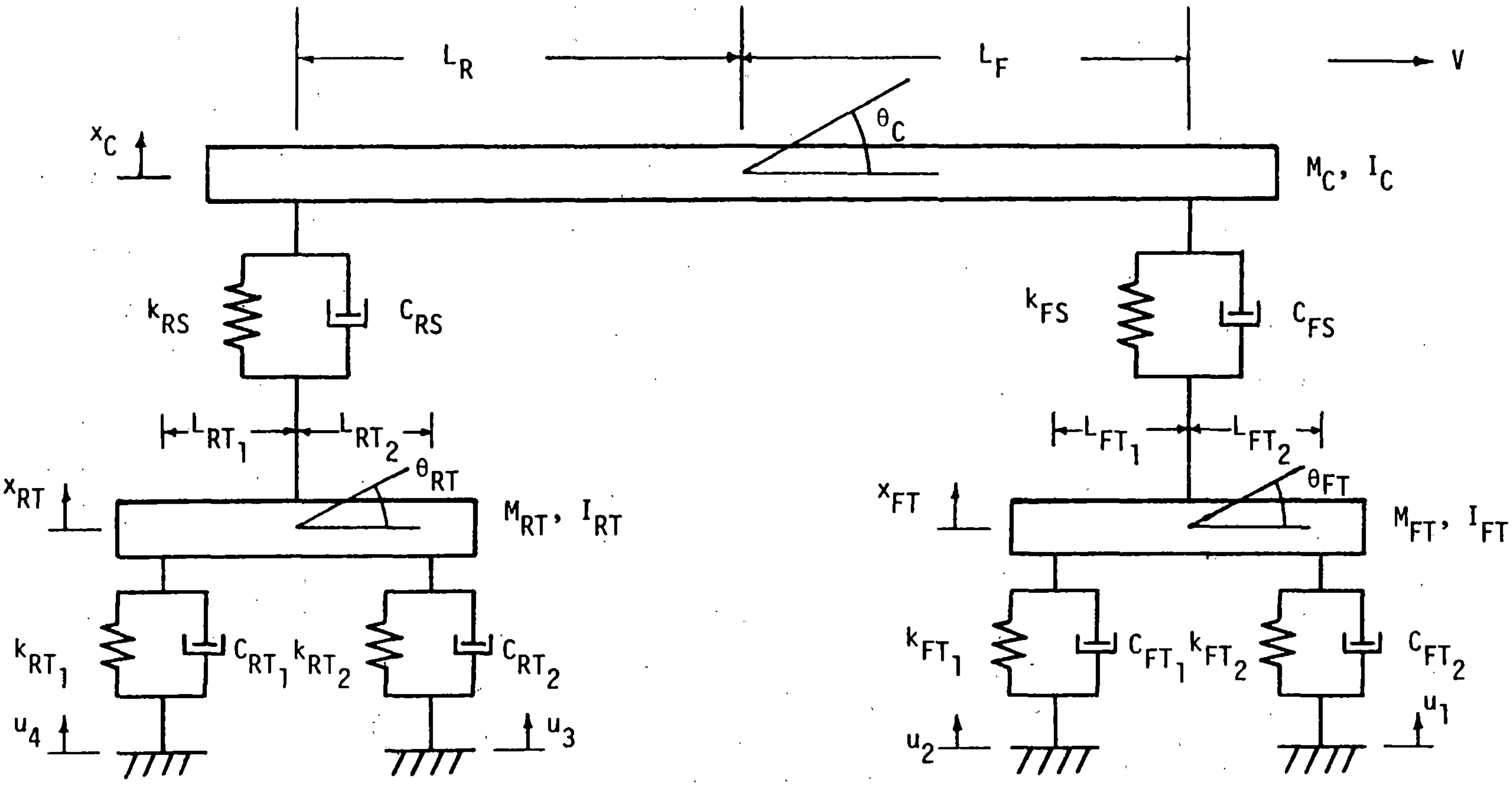


Figure D-C.1 Linear Six-Degree-of-Freedom Model of the Freight Car Vertical Dynamics (Source: Fallon, et al.)

be useful for the entire range of speeds encountered by freight cars (0-80 mph), and for track geometries encountered in all six classes of track. In particular, since this example concerns pitch and bounce response, the track profile and surface roughness inputs will be the ones of interest. At the level of detail considered here, the external parameters need not be considered. The vehicle characteristics which the model should be able to represent should include the range of masses, stiffnesses and dimensions being considered for the special-purpose boxcar design.

D-C.3 VALIDATION CRITERION

Now that the model purpose has been defined, it is necessary to specify the validation criterion which is most suitable for this model structure and purpose. This criterion incorporates all of the information needed to evaluate the validity of the model for the stated purpose - the response variables, their statistics and tolerances and the way they must be considered together (mathematical form). The dynamic response variables should be the body vertical and pitch accelerations (primary) and the secondary (carbody) suspension deflections (intermediate). These suspension deflections are important to the validation because of the model purpose of evaluating alternative suspension concepts. On the other hand, the natural frequencies and damping ratios are not included because these are not explicit outputs of the model, which is solved using spectral analysis methods.

Because the model is to be used to check on the vibration isolation ability of a railcar, it is important that it be able to represent the response spectra, and not just the total rms responses, of the chosen variables. In the validation procedures, this can be handled in either of two ways: (1) comparing the test outputs with the outputs of the model when it is driven by the inputs measured in the test, or (2) comparing the analytical frequency response functions of the model with the transfer functions derived empirically from the test data. The latter procedure is preferable if the input and output data can be synchronized well enough to produce frequency responses from the cross-spectral densities of outputs with inputs. However, the former procedure is simpler to apply and leaves fewer opportunities for errors.

The comparisons of the frequency responses or response spectra should be based on several different statistics. Restricting the discussion to the response spectra, these should be compared using the following measures: fit error and rms value error in the range from 0.1 to 10.0 Hz and errors in the frequencies and amplitudes of the principal peaks. Based on the anticipated use of the model for preliminary conceptual design, the following tolerances are suggested for these measures:

rms value error:	$\pm$20% absolute, $\pm$10% trend
fit error:	25% of mean-square value
discrepancies in frequencies of spectral peaks:	$\pm$10%
discrepancies in amplitudes of spectral peaks:	$\pm$30%

The rms values of acceleration are good indicators of the amount of vibrational energy being transmitted to the vehicle's lading, so these should be represented fairly accurately by the model ($\pm$20% being as good as one could reasonably expect from a simple model). More important than absolute rms accuracy is the accurate replication of the performance <u>trends</u> between different cases, leading to the tighter tolerance on the trend. The rms results could be matched very closely without necessarily having good agreement in the shapes of the response spectra. The fit error measures the agreement in the shapes of the spectra across all the frequencies, and is more difficult to reduce than the rms error. This is because it is a mean-square error (so positive and negative errors do not cancel each other), and it is sensitive to the "valleys" as well as "peaks" in the spectrum, even though those may not be very significant in terms of the overall model purpose. These considerations taken together lead to the choice of a fit error tolerance of 25% of the mean-square value.

In addition to these overall measures of model performance, it is important to match the individual response peaks in the model predictions and test results. The frequencies of these peaks should be relatively easy to match, and discrepancies in these would seem to indicate substantial errors in the modeled mass or spring characteristics. On the other hand, the amplitudes of the peaks in the spectrum are among the most difficult measures to match, particularly with a simple linear model. Based on these considerations, the tolerances of $\pm 10\%$ on frequency and $\pm 30\%$ on peak amplitude were chosen. In this example, the same tolerances have been applied to all of the responses, but that is not a necessary restriction.

The mathematical form of the validation criterion chosen for this example is the simplest of the four which were shown in Table 3.8, which is #4 (no tolerance exceedance). That form requires that each of the tolerances be satisfied separately, and does not permit poor performance in some to be compensated for by good performance in others. Although this is in a sense the "strictest" of the four forms, it is also most suited to a simple validation with only a few dynamic response variables being used. The validation thus requires good agreement between model and test in both the body accelerations and the secondary suspension deflections.

D-C.4 REVIEW AND SUMMARY OF EXISTING TEST DATA

Because limited resources are available and a vehicle of the proposed design has not yet been constructed, only existing test data can be used for this validation. The procedures of Section D-4 are therefore chosen over those of Section D-5. This begins with the review and summary of the test data, listing the data items described in Table D-4.1 and answering the questions which were listed in Table D-4.2. These are shown in Tables D-C.1 and

Table D-C.1

Summary of Existing Test Data

TEST CONDITIONS:	
PURPOSE OF ORIGINAL TEST	Ride quality test of refrigerated boxcar.
ENVIRONMENTAL CONDITIONS:	
- TEMPERATURE AND HUMIDITY - PRECIPITATION - WIND SPEED AND DIRECTION	Unknown.
OPERATING CONDITIONS:	
- SPEED	0-60 mph in 20-mph increments.
- ACCELERATION OR BRAKING	None.
- VEHICLE LOADING	Fully loaded and empty.
TRACK CHARACTERISTICS - WHICH WERE MEASURED AND WHEN?	
- TANGENT OR CURVE	Tangent.
- GRADE AND SUPERELEVATION (IF ANY)	None.
- VERTICAL AND LATERAL MODULUS (STATIC AND DYNAMIC)	Not measured.
- GEOMETRY (PROFILE, GAUGE, ALIGNMENT CROSSLEVEL)	Spectra measured before test.
- ROUGHNESS	
- CROSS-SECTIONS OF RAIL HEAD	Not measured.
- REVENUE SERVICE AND MAINTENANCE HISTORIES	Unknown.
VEHICLE CHARACTERISTICS - WHICH ARE KNOWN?	
- GEOMETRY (DIMENSIONS)	Known.
- MASS PROPERTIES	Known.
- SPRING AND DAMPING CHARACTERISTICS	Springs known, not damping.
- DETAILS OF NONLINEAR CHARACTERISTICS	Not specified.
- WHEEL PROFILES	
- REVENUE SERVICE HISTORY (TYPE AND LADING) - MAINTENANCE HISTORY	Unknown.
DYNAMIC RESPONSE MEASUREMENTS	
- WHICH WERE MEASURED?	Body end accelerations, bolster/side frame displacements.
- HOW ARE THEY SYNCHRONIZED WITH TRACK DATA?	Not synchronized.
- ARE ANY CHANNELS MISSING?	One accelerometer channel lost on one test.
- HOW "GOOD" ARE THEY?	
- BANDWIDTH (FILTERING)	0-100 Hz.
- DYNAMIC RANGE	30 dB.
- NOISE LEVELS	Unspecified.
- ACCURACY	Unknown.
- KNOWN ERRORS	Zero points not calibrated for displacements (bias).

D-C.2. The important items here include the speed range, which was only tested up to 60 mph and the lack of synchronization between the track and vehicle measurements, which means that the cross-spectral density method of deriving the frequency response of the vehicle cannot be used although the auto-spectral density method can still be used. Also, the vehicle damping characteristics and instrumentation noise and accuracy information were not available. Most of the other unknown information is not particularly important for this model validation.

The restricted speed range of the tests will make it difficult to validate the model for speeds up to 80 mph, since there are no test data available for speeds above 60 mph and no way of telling whether those speeds include a transition to a different performance regime (such as hunting). Only empty and fully loaded vehicles were tested, but none with different spring groups or snubbers. However, the differences in these components should be straightforward enough that the validation could still be performed for preliminary design evaluations without further separate test cases.

D-C.5 RECONSTRUCTION OF MISSING DATA

Some reconstruction of data will be needed. For example, the vertical accelerations at the ends of the carbody need to be summed and differenced to derive the body vertical and pitch accelerations. Likewise, the bolster/side frame displacements will have to be assumed equivalent to the secondary suspension deflections, not allowing for any centerplate lift-off. The accelerometer channel which was lost on one test cannot be reconstructed from the remaining data. However, the remaining channels for that test can be compared with the corresponding channels in the tests at lower and higher speeds to ensure that the trends remain as expected and that the other variables are

Table D-C.2

Initial Screening of Existing Test Data

(1)	WERE THE TEST CONDITIONS COMPARABLE TO THE CONDITIONS THE MODEL WILL BE USED TO EVALUATE?	
	- SIMILAR SPEED RANGE	Not quite as high.
	- SIMILAR TYPE OF TRACK (TANGENT VS. CURVE)	Yes.
(2)	WAS THE TEST VEHICLE BASICALLY SIMILAR TO THE VEHICLE BEING MODELED?	
	- LOCOMOTIVE OR FREIGHT OR PASSENGER CAR	Freight car.
	- TRUCK DESIGN AND LOCATION (TRUCK CENTER SPACING)	Similar.
	- MASS AND MASS DISTRIBUTION	Similar to loaded case.
(3)	WERE ENOUGH DIFFERENT VEHICLE CONFIGURATIONS TESTED TO SHOW THE EFFECTS OF DESIGN CHANGES?	Not directly.
(4)	WAS THE DYNAMIC PROCESS WHICH HAS BEEN MODELED OBSERVED IN THE TEST?	Yes.
(5)	ARE TRACK GEOMETRY DATA AVAILABLE FOR THE TEST TRACK SECTION IN A FORM WHICH CAN BE SYNCHRONIZED WITH THE VEHICLE DYNAMIC RESPONSE DATA?	Not completely synchronized.
(6)	ARE ALL THE RESPONSE VARIABLES WHICH APPEAR IN THE VALIDATION CRITERION AVAILABLE IN THE TEST DATA?	Not entirely.
	IF NOT, CAN THEY BE RECONSTRUCTED FROM THE TEST DATA?	Yes.
(7)	ARE THE TEST DATA ACCURATE ENOUGH TO FIT COMFORTABLY WITHIN THE TOLERANCES IN THE VALIDATION CRITERION?	Unknown.
(8)	WERE THE VEHICLE PARAMETERS WHICH APPEAR IN THE MODEL RECORDED AT THE TIME OF THE TEST?	No.
	IF NOT, ARE THEY AVAILABLE FROM OTHER RELIABLE SOURCES?	Yes. Manufacturers' specifications.

found to agree for the model and test results. In particular, for this case, the model prediction of the vertical acceleration of the body end for which the measurement is available can be compared with that measured result. The bias on the suspension displacement measurement (failure to calibrate zero point) can be removed easily by subtracting the steady-state value prior to the start of the test run from the values measured during the run.

D-C.6 SENSITIVITY TESTING FOR UNCERTAINTIES

The uncertain values from this test program which could affect the model validation are the primary and secondary suspension damping. It is necessary to make estimates of these values to use in the model, since they were not measured directly. Based on results which have been reported from prior test programs, the range of primary suspension damping can be assumed to be from 0.05 to 0.1 and secondary suspension damping from 0.2 to 0.5 for evaluating sensitivity. Pilot runs of the model for several well-chosen combinations of damping values can reveal the sensitivity of the results to the damping estimates. For example, four model runs can illustrate all combinations of the minimum and maximum damping values. The extremes of these four cases (both low values and both high values) are illustrated in a sample model output spectrum plot in Figure D-C.2. The differences between the two sets of results are small enough to fit within all of the validation criteria except for the difference in the maximum amplitudes of the lower frequency peaks.

The comparison for worst-case changes on both parameters is unnecessarily rigorous. A more realistic sensitivity test is to use the "best guess" value of one parameter while varying the other over its complete range, and then to switch the two parameters and repeat the test. Under those conditions, the extreme cases remain within the tolerance bands and the model is

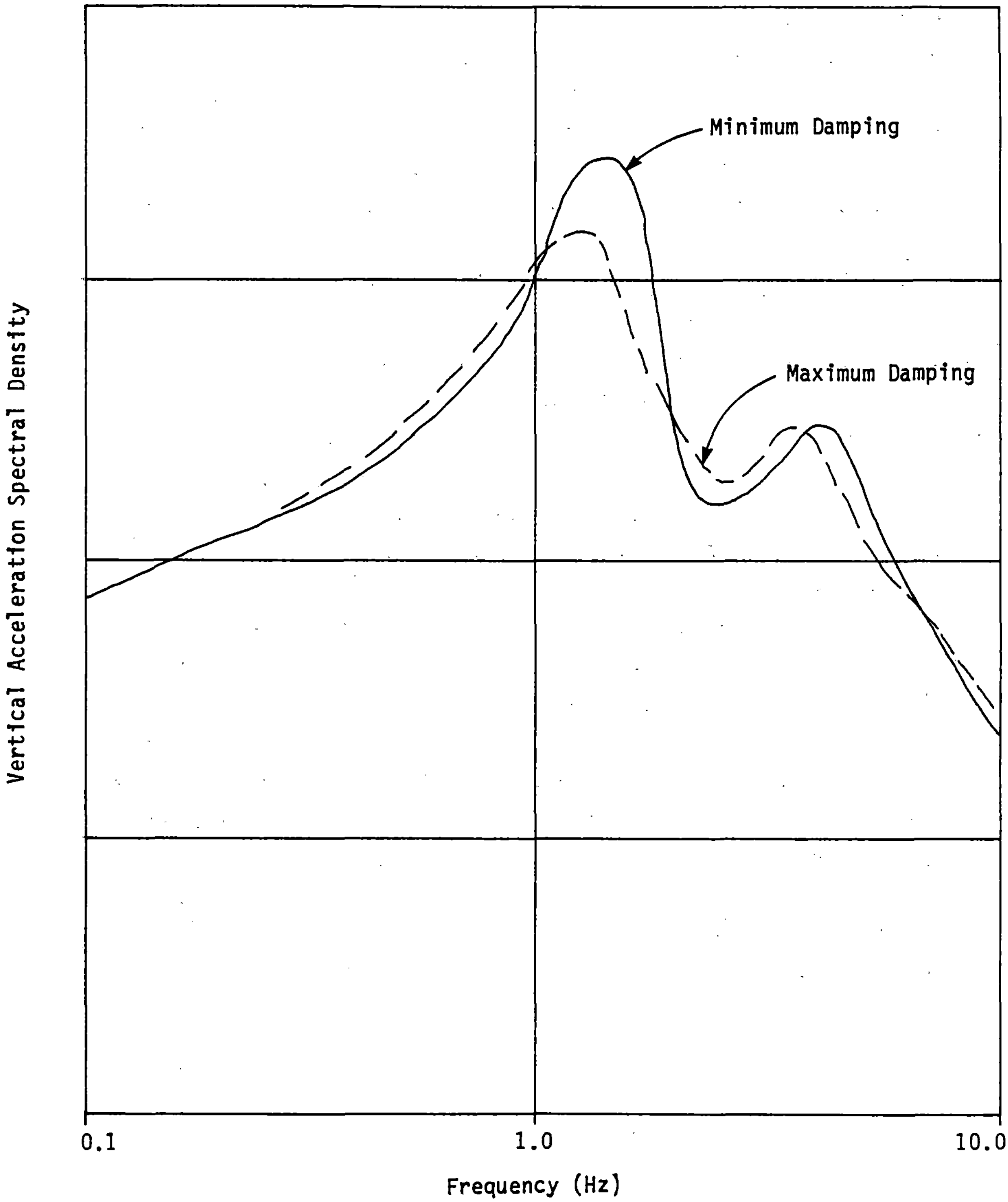


Figure D-C.2 Sensitivity Test for Suspension Damping shown to be sufficiently insensitive to the uncertainties in damping values for the validation to proceed.

D-C.7 PILOT RUNS OF MODEL

In order to generate model output data which are as directly comparable as possible to the test results data, the model should now be run for exactly the cases which were tested (as exactly as can be determined). For these runs, the measured or documented vehicle parameters are used, plus the best-guess values of the uncertain damping parameters. The inputs to these model runs are the track profile spectra which were measured before the vehicle test program and the outputs are the model's predictions of the response spectra.

These pilot runs represent six separate executions of the model for the six test conditions:

Speed	Loading
20 mph	Empty
40 mph	Empty
60 mph	Empty
20 mph	Fully Loaded
40 mph	Fully Loaded
60 mph	Fully Loaded

Before the model output predictions and vehicle test results can be compared, they must be placed in the same form (spectra of response variables). Because the model is based on a linear frequency domain analytical techniques, it calculates this directly from the input spectrum. (The input spectrum had to be calculated from the track geometry space curve data measured prior to the vehicle test.) The time history data recorded in the vehicle test (as well as the track geometry measurements) must be converted to the frequency domain using a Fast Fourier Transform computer program or a spectrum analyzer. In order for this conversion to produce spectra which can be compared directly with the model-predicted spectra, several data conditioning processes must be carried out.

The test data were recorded digitally (sampled) at 100 Hz, although the model validation need only be concerned with responses between 0.1 and 10 Hz. The test instrumentation was not documented thoroughly enough to tell whether the measurements were properly filtered before sampling. It is therefore possible that the available data are aliased (confounded with some higher frequency noise and responses). The data should first be filtered with a cutoff frequency of 10 Hz, and then sampled at 25 Hz before proceeding further. In order to be able to observe responses at frequencies as low as 0.1 Hz, it is necessary to have a data sample of at least 10s duration, while the available test results cover runs of at least 5 minutes duration (300s). The FFT processing is most efficient when it uses data records containing 2^n samples, where n is an integer (e.g., 512 or 1024 or 2048 samples). The 7500 samples available in the shortest test run are divided into seven separate records of length 1024 samples (about 41 seconds each). Each of these is converted by the FFT to a sample spectrum, and the seven sample spectra are then averaged together to produce a smoother composite spectrum which can be more readily used for the comparison between test and model results.

This processing is performed for each dynamic response variable needed in the validation criterion, for each of the test cases. When a response variable is derived by combining more than one instrumentation channel (such as the body vertical and pitch accelerations), the time history data should be combined before performing the Fourier transform, not after. At the completion of this step, there is an average response spectrum computed for each of the three response variables (body yaw and pitch acceleration and one secondary suspension deflection), for each of the six operating conditions, for both model and test outputs, making a total of 36 spectra.

D-C.8 COMPARING MODEL PREDICTIONS AND TEST RESULTS

The comparison of model predictions and test results begins with the fully loaded vehicle, which is likely to show more nearly linear responses, and proceeds from the low-speed test cases to the higher speed cases with their larger amplitude responses.

Figure D-C.3 is an example of the comparison of carbody bounce acceleration spectra for the first test condition. The most immediately visible features are the abrupt dips in the model predictions. These are produced by the mathematical filtering effect associated with the axle and truck center spacings in the model, and must be disregarded when comparing the model and test results. The model predicts peak responses which are lower and slightly more damped than the test results, although only the magnitude of the first peak shows a large enough error to risk violating the validation criterion. The rms predictions are very close, as are the frequencies of the peaks. The fit error can only be computed meaningfully if the spurious abrupt dips are "filled in" first, and when that is done the fit error is well within the allowable tolerance.

The remaining dynamic response variables show similarly good agreement between model and test for this case, so the validation can be considered successful <u>for this case</u>.

The comparison between the 40 mph results is not substantially different from that at 20 mph, and is not illustrated here. Proceeding to the 60 mph fully loaded case, two sample comparisons of spectra are shown in Figures D-C.4(a) and (b). Figure D-C.4(a) shows accelerations substantially higher than at the lower speed, and the dips in the model-predicted spectra have moved to higher frequencies, corresponding to the higher speed. Here, the model-predicted acceleration spectrum peaks are somewhat higher than the measured peaks, but still close enough to remain within the tolerances.

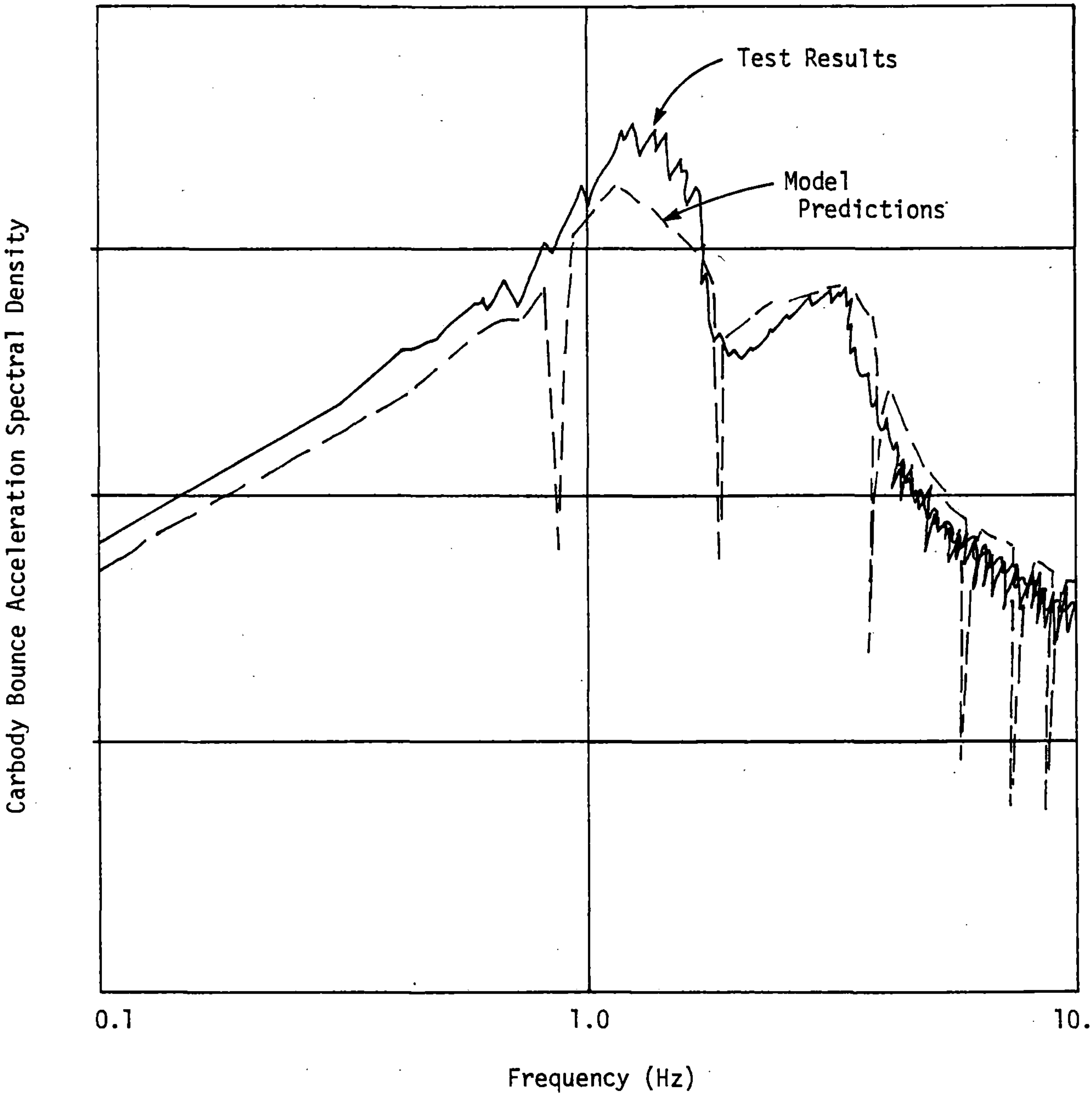


Figure D-C.3 Comparison of Model Predictions and Test Results For Fully Loaded Car at 20 mph

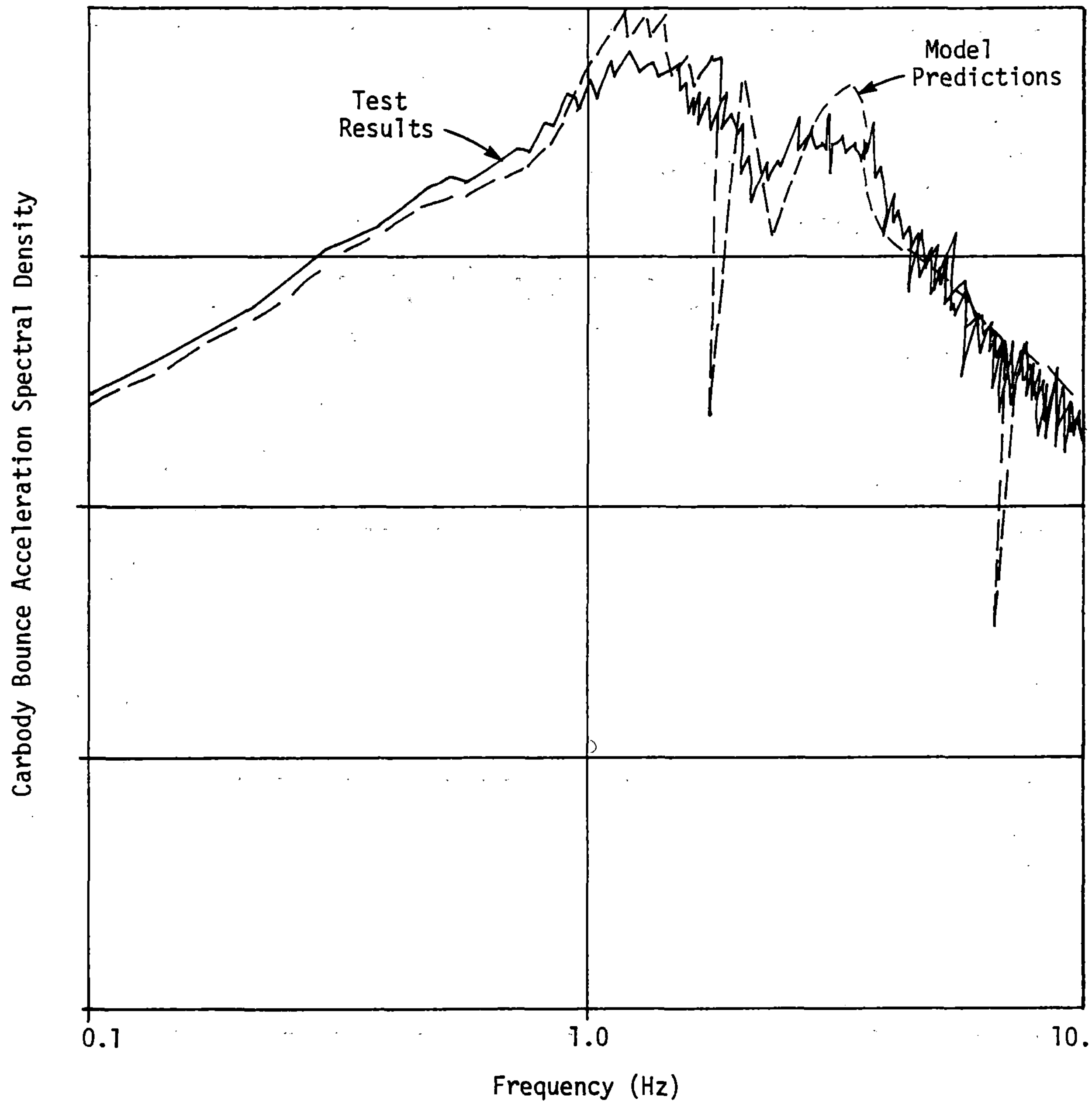


Figure D-C.4(a) Comparison of Model Predictions and Test Results For Fully Loaded Car at 60 mph

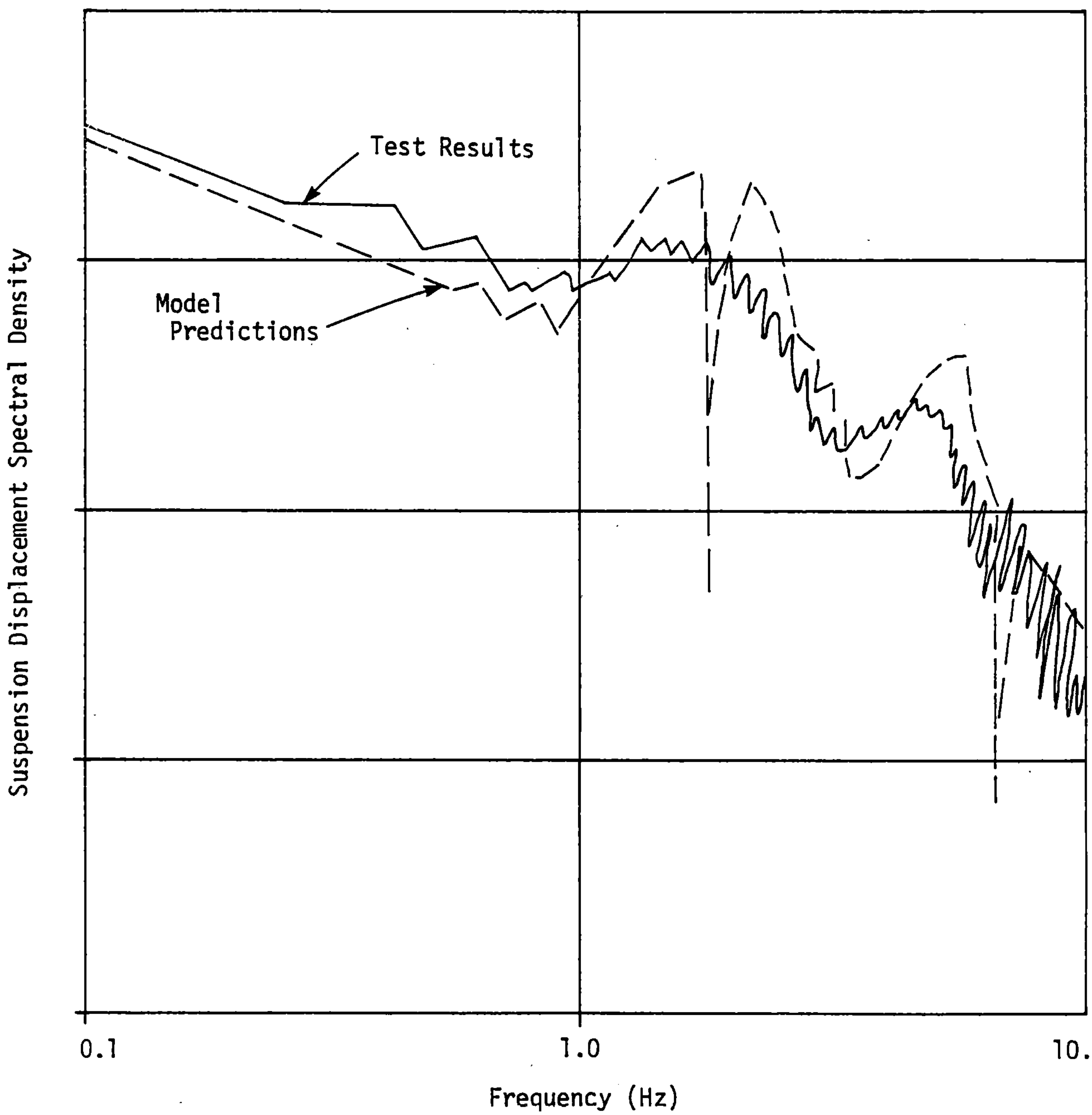


Figure D-C.4(b) Comparison of Model Predictions and Test Results For Fully Loaded Car at 60 mph

Figure D-C.4(b) shows the suspension displacement spectra for this case, in which the modeled peaks are noticeably higher than the test results indicated. Although the validation criterion has not been violated, there is a noticeable trend for the model damping to decrease relative to the observed damping as the speed and overall response amplitude increase. This could be attributed to the linearization of the dry friction in the suspension, which would imply that at even higher speeds and amplitudes the disagreement in damping estimates would increase. If the model is to be applied for speeds in the 60 to 80 mph range, it will be necessary to increase the damping parameter value to remain within the required tolerances.

A modest increase in the secondary suspension damping (10%) was inserted into the model and the model was re-run, producing better agreement with the test results at 60 mph. When the 20 mph case was re-run with this damping increase, the discrepancy between the model and test amplitudes of the first peak in the spectrum increased to the maximum allowable in the validation criterion. This means that the damping cannot be increased any further without endangering the validation for the lower speeds. These results mean that the linear model cannot be validated to the level of accuracy defined in the validation criterion for the entire speed range from 0 to 80 mph. Either the tolerances must be relaxed or a nonlinear representation of the suspension dry friction must be used if the model is to satisfy the criterion over the entire speed range for the fully loaded vehicle. The approach followed in this example is to limit the speed range to 0 to 60 mph rather than changing the model or criterion.

The model with the increased damping is now compared against the test results for the second set of conditions, the empty car. If the model is to be usable for more than one specific type of car and loading condition, it must be able to represent changes in vehicle characteristics with modifications only to the physical parameter which is different (in this case, the carbody mass and inertia). The comparison of body bounce acceleration spectra for the empty car at 20 mph is shown in Figure D-C.5.

The important differences between model predictions and test results in Figure D-C.5 are the offsets in the frequencies of the peaks and the significantly higher accelerations at the higher frequencies in the tests. The peaks are of higher frequency and amplitude than for the fully loaded vehicle, as one would expect. With the smaller vehicle mass used here, the break-away force level of stick-slip friction in the suspension appears more significant than in the earlier cases, and the performance is more strongly nonlinear. The friction acts as an effective spring until break-away, increasing the natural frequencies above those predicted by the purely linear model. This probably accounts for the discrepancies in the frequencies of the peaks, which are as large as can be permitted under the validation criterion. These discrepancies and the differences in high-frequency response also make the fit error statistic difficult to maintain within the tolerance band. The model fails to represent the higher frequency accelerations accurately because of the nonlinear stick-slip phenomenon which it does not include.

Although the model predictions for 20 mph in Figure D-C.5 are barely able to satisfy two of the tolerances in the validation criterion (peak frequencies and fit error), when the speeds are increased to 40 mph and 60 mph the validation criterion is no longer satisfied. A model structure change is needed to enable the model to represent all the cases which were tested. The addition of nonlinear friction could greatly improve the model's agreement with the test results, but would introduce major complications, including a change in analytical technique to the significantly more complicated quasi-linear frequency domain technique. A simpler way of trying to make the model represent the effects produced by the dry friction is to make the suspension spring stiffness in the model a function of the car's

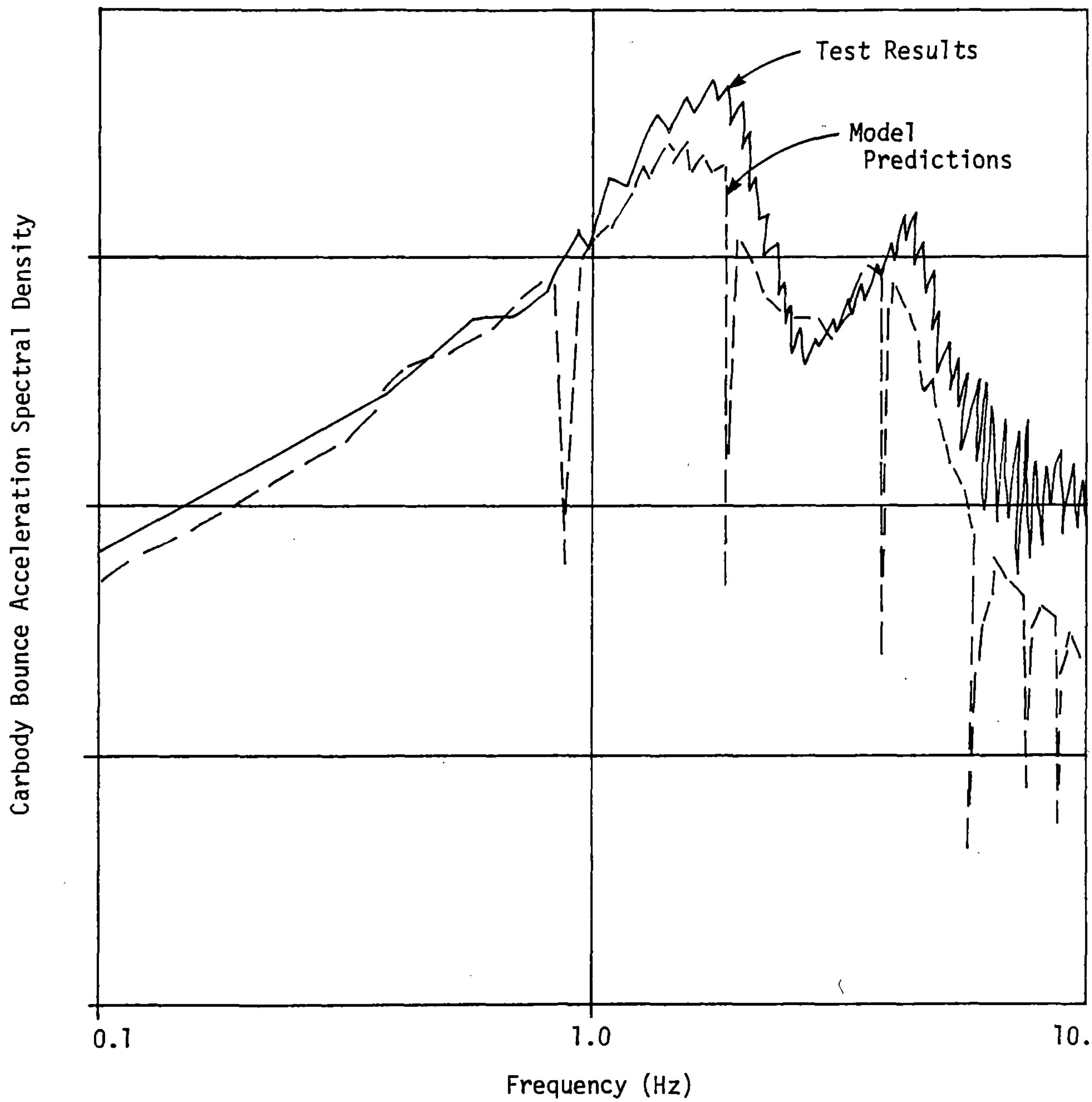


Figure D-C.5 Comparison of Model Predictions and Test Results For Empty Car at 60 mph loading as well as the actual stiffness of the springs. The size of this correction can be estimated from the offset in the frequencies of the peaks in Figure D-C.5, which are about 10% higher in the tests than the model predictions. To increase the model's predicted natural frequencies by 10%, the effective spring constants must be increased by 21% for the empty car relative to the fully loaded car. For use over the entire range of car loadings, these spring constraints should therefore be defined as

$$k_e = k\left(1 + 0.21\,\frac{m_f - m}{m_f - m_o}\right)$$

where

k_e	is equivalent spring constant
k	is actual spring constant
m_f	is mass of fully loaded carbody in validation test
m_o	is mass of empty carbody in validation test
m	is mass of carbody being modeled

It is essential to recognize the serious limitations which this adjustment to the spring constant introduces. The model can only be validated for a car which has a body mass and capacity comparable to that of the test car, as well as the same suspension friction characteristics. The model can therefore not be used to investigate the effects of different types of suspension damping, such as the replacement of friction snubbers with hydraulic dampers. This may make the model less than ideally useful for its intended purpose of evaluating the vibration isolation characteristics of different suspension design concepts.

The model with the adjusted spring constants is now re-run for the empty car at 20 mph, producing the results shown in Figure D-C.6. Figure D-C.6 illustrates the shift of the model predicted peaks to higher frequencies, providing a much closer match to the peaks in the test results and thereby significantly improving the fit error. The under-prediction of higher

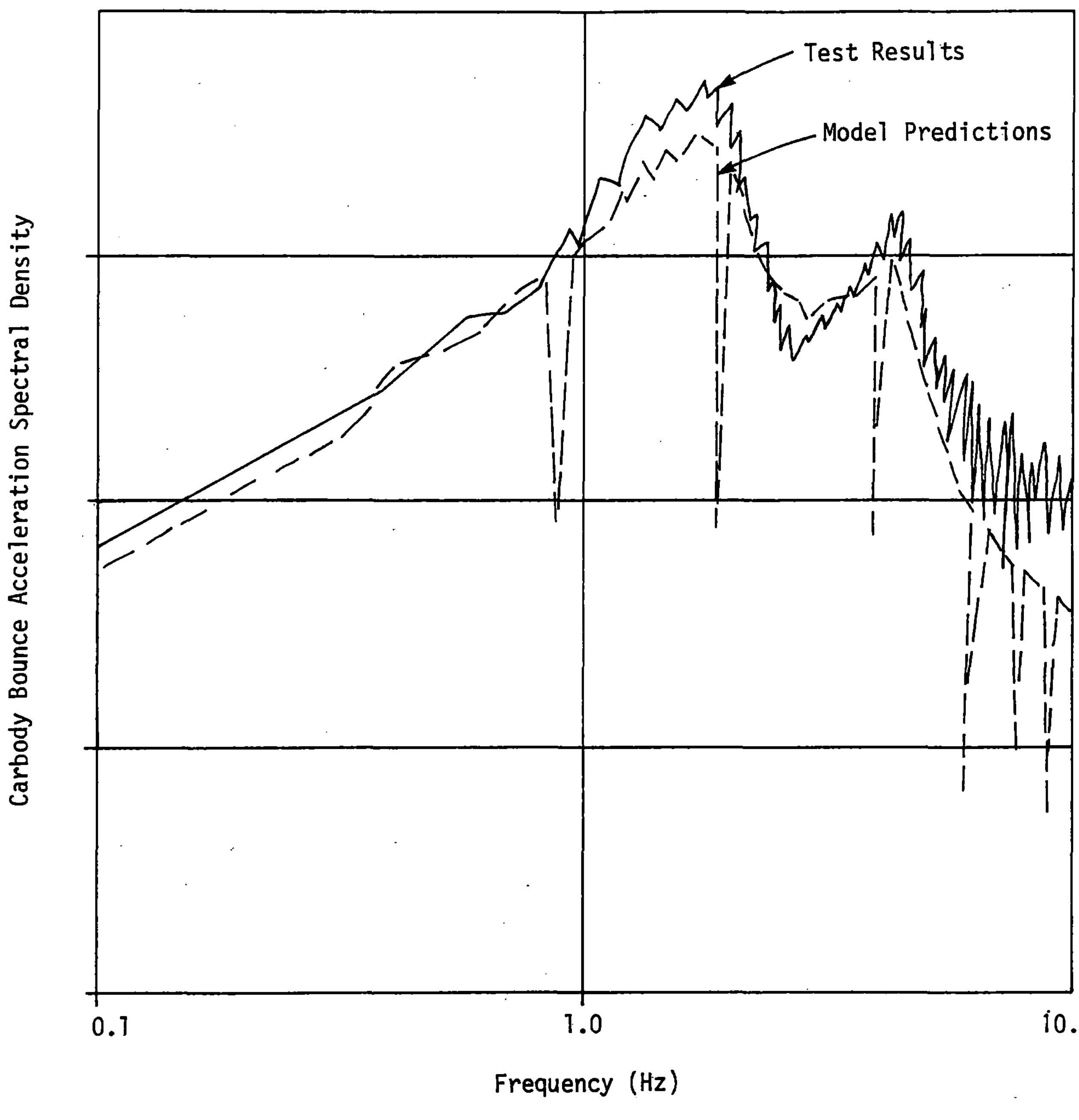


Figure D-C.6 Comparison of Adjusted Model Predictions and Test Results For Empty Car at 20 mph frequency response is only partially alleviated by this change. The results for the 60 mph test and model prediction are larger in amplitude, but illustrate a slightly lower degree of correspondence between model and test.

Although all of the test cases have gone through the process of comparison with model predictions, the model has been adjusted twice since the first comparison. It is thus necessary to go back and re-do the comparisons using the final adjusted version of the model. For this example, that process is simplified because the second adjustment (making the effective spring constant a function of body mass) was designed not to change the model for the fully loaded case. The first adjustment (10% increase in secondary suspension damping) was re-checked for the earlier case at the time it was suggested, so that it is not necessary to re-do any of the comparisons here. However, in most validations, which are more complicated than this, it is necessary to go back through all of the tests cases at the end with the final adjusted version of the model.

D-C.9 SUMMARIZE RESULTS OF MODEL VALIDATION

The results of this example validation are summarized for any potential users of the model in Figure D-C.7, using the general format which was defined in Figure D-6.4. Because the test conditions were not reported as fully as one would prefer for use in model validations, some of the entries in Figure D-C.7 are not as specific as they should be. The track compliance was not measured, and must be assumed to be in a "standard" range rather than being exceptionally high or low. This is an unfortunate loss because of its direct influence on the primary suspension stiffness. Similarly, the track class and surface conditions were not reported, although the class could, in theory, be derived from the track geometry measurements. The model is assumed to apply for the normal range of track classes

THE MODEL

NAME (AND DEVELOPERS):
DEGREES OF FREEDOM: 6
PERFORMANCE ISSUE(S): Pitch and bounce
LINEAR OR NONLINEAR: Linear
NONLINEARITIES INCLUDED: None
ANALYTICAL TECHNIQUE: Linear frequency domain

VALIDATION #1

PURPOSE (ANTICIPATED USE): Preliminary evaluation of car design concept
VALIDATION CRITERION: Variables - Body pitch and bounce acceleration suspension deflections
Statistics - rms value error ± 20% absolute
± 10% trend
fit error < 25% of mean square value
frequencies of spectral peaks ± 10%
amplitudes of spectral peaks ± 30%
Form - satisfy all of tolerances

VEHICLE CHARACTERISTICS
TYPE (FREIGHT, PASSENGER, LOCOMOTIVE): Freight
CARBODY TYPES: Boxcar (including refrigerated)
WHEEL PROFILES: Unspecified
TRUCK TYPES: Standard 3-piece freight truck

RESTRICTIONS ON OTHER CHARACTERISTICS: (LENGTH, HEIGHT, MASS, STIFFNESSES, CARGO, TRUCK CENTER SPACING, CENTER OF GRAVITY, ETC.) Standard day friction in truck
Carbody empty weight 1/3 of fully loaded weight

SPEED RANGE: 0-60 mph

TRACK CHARACTERISTICS:
TANGENT, SPIRAL OR CURVE: Tangent
GRADE OR SUPERELEVATION: None
SURFACE CONDITION: "Normal"

PERTURBATIONS IN	MAXIMUM AMPLITUDE	WAVELENGTHS
ALIGNMENT:	Not applicable	
PROFILE:	Normal classes 1-6, at permitted speeds	
GAUGE:	Not applicable	
CROSSLEVEL:	Not applicable	

COMPLIANCE (LATERAL, VERTICAL): "Standard" range - not exceptionally high or low.

Figure D-C.7 Summary of Results of Sample Validation with their allowable geometric deviations, but not for abrupt isolated perturbations such as those produced by cross-overs or turnouts.

The test results were available at 20, 40, and 60 mph, permitting validations within that speed range without significant qualitative changes in dynamic performance. Because there are no known changes in performance regime for pitch and bounce response at speeds below 20 mph, the range of validation has been extended down to zero. This does not rule out the possibility of a transition to severe twist-and-roll response at the lower speeds, since that would be modeled as a separate performance issue. However, the validation could not be extended to speeds above 60 mph because of the increasing amplitude of response and the possibility of encountering further nonlinearities such as wheel lift, centerplate separation or suspension stops.

This artificial example validation has been kept simple to illustrate some of the basic issues in the validation process. The restriction to a linear model has been shown to be quite severe when trying to represent the performance of a substantially nonlinear system such as a railroad freight car over a range of speeds and loading conditions. The need to use existing test data has also been shown to impose significant limitations on the level to which a model can be validated.

SECTION E

TEST PLAN SUMMARIES

E.1 INTRODUCTION

The Vehicle/Track Interaction Assessment Techniques (IAT) is organized in the form of assessment procedures. For each of the following three objectives a procedure is developed as described in Section 2 of Part I of the document:

- o The Modified Vehicle Evaluation
- o The Vehicle Problem Diagnostic Evaluation; and
- o The Prototype Vehicle Evaluation.

As described in Section 2, the objective can often be achieved by performing analytical studies without having to conduct a test. However, if a test is found to be necessary, this section provides information required to determine the scope of that test. This information is presented in an easy-to-use format; one table is provided for each combination of the following three Test Categories:

- o Proof Test;
- o Diagnostic Test; and
- o Service Environment Test;

and the following 10 Performance Issues:

- o Hunting;
- o Twist and Roll;
- o Pitch and Bounce;
- o Yaw and Sway;
- o Steady-state Curving;
- o Spiral Negotiation;
- o Dynamic Curving;
- o Steady Buff and Draft;
- o Longitudinal Train Action; and
- o Longitudinal Impact.

For each combination, the following test aspects are addressed:

- o The characteristics of a desirable test site;
- o The required control variables (i.e., the conditions under which the tests should be conducted);
- o The required response variables (i.e., the variables to be measured);
- o The data handling requirements;
- o The elements of the Performance Indices (needed to evaluate vehicle performance); and
- o The safety criteria.

In some cases the speeds listed in these tables exceed the Track Safety Standards for some performance issues and their associated perturbations. These inputs are required to assess the dynamic capabilities of the vehicles, and their sensitivites at conditions near the stated Track Safety Standard limits. A waiver from the FRA's Office of Safety will be required for those testing conditions where the FRA Track Safety Standards may apply and where testing may exceed them. In all testing situations, safety precautions should be taken in the planning and execution of these tests.

The above information is brief and suitable only to gain an understanding of the scope of the test program. The user should consult Section 3 in Part I and Sections F, G, H, K, L, M, and O in Part II while developing the Test Details Document.

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

HUNTING

TEST CATEGORY:

PROOF

RECOMMENDED TEST SITE:

- o Test Track or RDL.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 3000 ft of unperturbed tangent track for maximum speeds of 115 mph for freight and 130 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56.5	1	o Body Accelerations at C.G. - Sway
2	Track Class	3-6		
3	Track Alignment Amplitude, inches	1/2-2	2	o Truck Frame Accelerations Instrumented Truck - Sway
4	Rail Profile	New		
5	Test Speed, mph Freight Passenger	 30-115 30-130		
6	Rail Surface Condition	Sanded,* Dry		
	*Rail Friction Coefficient of 0.15 to 0.3			

TEST PLAN SUMMARY (CONT'D)

HUNTING PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	7	None
SAMPLING RATE (Hz)	100	None
QUICK LOOK CHANNELS	4	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Acceleration	Damping (From Time Response	>0.1
2	Carbody Sway Acceleration	Damping (From Time Response	>0.1
3	Carbody Sway Acceleration	Peak	<0.55g
4	Carbody Sway Acceleration	RMS	<0.1g

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Sway Acceleration	Maximum	0.6g

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

TWIST & ROLL

TEST CATEGORY:

PROOF

RECOMMENDED TEST SITE:

- o Test Track or RDL (See Section F for perturbation details).
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 240 ft of perturbed tangent track, for maximum speeds of 30 mph for freight and 35 mph for passenger cars;
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2		o Body Accelerations at C.G.
			1	- Roll
2	Track Crosslevel Amplitude, inches	2		o Bolster Displacement
			2	- Roll
3	Track Crosslevel Wavelength, ft	39	3	- Bounce
4	Vertical Track Stiffness, kips/in *	> 225		
5	Test Speed, mph Freight Passenger	 10-30 10-35		
	*Tangent Stiffness under 12,000# Vertical Load			

TEST PLAN SUMMARY (CONT'D)

TWIST & ROLL

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	8	None
SAMPLING RATE (Hz)	20	None
QUICK LOOK CHANNELS	3	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Roll Angle	Peak to Peak	$<7^{o}$
2	Carbody-Bolster Relative Roll	Peak to Peak	$<4^{o}$

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Roll Angle	Maximum	3.5^{o}

TEST PLAN SUMMARY

PERFORMANCE ISSUE: PITCH AND BOUNCE

TEST CATEGORY: PROOF

RECOMMENDED TEST SITE:

- o Test Track or RDL (See Section F for perturbation details).
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 640 ft of perturbed tangent track, for maximum speeds of 30 mph for freight and 35 mph for passenger cars;
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2		o Body Accelerations at C.G.
			1	- Pitch
			2	- Bounce
2	Track Profile Amplitude, inches	3		o Bolster Displacement
3	Track Profile Wavelength, ft.	19.5-39	3	- Bounce
4	Vertical Track Stiffness, kips/inch*	>225		o Truck Frame Displacement Instrumented Truck
			4	- Bounce
5	Test Speed, mph			o Coupler Displacement Both Couplers
	Freight	10-30		
	Passenger	10-35	5	- Vertical
	*Tangent Stiffness Under 12,000# Vertical Load.			

PITCH & BOUNCE

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	11	None
SAMPLING RATE (Hz)	20	None
QUICK LOOK CHANNELS	5	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Peak to Peak	$<2^{0}$
2	Carbody Bounce Accelerations at C.G.	Peak	<0.5g
3	Truck-Car Relative Bounce Displacement.	Peak	<3"
4	Carbody-Bolster Relative Bounce Motion	Peak	<2"

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Maximum	1^{0}
2	Truck Car Relative Bounce Displacement	Maximum	3"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

YAW AND SWAY

TEST CATEGORY:

PROOF

RECOMMENDED TEST SITE:

- o Test Track or RDL (See Section F for perturbation details).
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 240 ft of perturbed tangent tracks for maximum speeds of 30 mph for freight and 35 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2		o Body Accelerations at C.G.
			1	- Yaw
2	Track Alignment: Amplitude, inches	3	2	- Sway
3	Track Alignment Wavelength, ft.	39		o Coupler Displacement Both Couplers
			3	- Lateral
4	Lateral Rail Stiffness, kips*	>40		
5	Rail Profile	New		
6	Test Speed, mph			
	Freight	10-30		
	Passenger	10-35		
7	Rail Surface Condition	Sanded** Dry		
	*Secant Stiffness with Zero Vertical Load with Zero to 4000# Lateral Load.			
	**Rail Friction Coefficient of 0.15 to 0.3.			

TEST PLAN SUMMARY (CONT'D)

YAW & SWAY PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	9	None
SAMPLING RATE (Hz)	20	None
QUICK LOOK CHANNELS	5	None

*Includes five channels for speed, ALD, temperature and such.

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Yaw Angle	Peak to Peak	$<2^{o}$**
2	Carbody Sway Acceleration at C.G.	Peak	<0.5g
	**For 40' truck center distance. Proportionally lower for higher truck center distances.		

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Yaw Angle	Maximum	1^{o}

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

STEADY STATE CURVING

TEST CATEGORY:

PROOF

RECOMMENDED TEST SITE:

- o Test Track,
- o Class of track will be appropriate with the maximum test speed.
- o Test zone requires 2000 ft of 1° curve for maximum speeds of 115 mph for freight and 130 mph for passenger cars; 200 ft of 2° curve with 3" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 800 ft of 2° curve with 6" S.E. for maximun speeds of 80 mph for freight and 90 mph for passenger cars; 1200 ft of 2° curve with 1" S.E. for speeds of 65 mph for freight and 80 mph for passenger cars; 500 ft of 5° curve with 0" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 800 ft of 50' curve with 2.5" S.E. for maximum speeds of 45 mph for freightand 55 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unitentional perturbations in the unperturbed parts should be less than these for class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56-57		o Wheel/Rail Forces
2	Track Curvature, degrees	1-10		Instrumented Truck Total Truck
3	Superelevation, inches	0-6	1	- Lateral
4	Track Class	2-6	2	- Vertical
5	Rail Profile	New		Instrumented Axle
6	Test Speed, mph		3	- Lateral
	Freight	20-115	4	- Vertical
	Passenger	20-130		
7	Underbalance (ΔE), inches	0-8		
8	Rail Surface Condition	Sanded,* Dry		
	*Rail Friction Coefficient of 0.15 to 0.3.			

TEST PLAN SUMMARY (CONT'D)

STEADY STATE CURVING

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	11	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	4	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px)*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Wheel Lateral Force (Leading, High Rail)	Mean	<20 kips
2	Wheel L/V (Leading, High Rail)	Mean	<0.8

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Wheel Lateral Force	Maximum	25 kips
2	Single Wheel L/V	Maximum	0.8

TEST PLAN SUMMARY

PERFORMANCE ISSUE: SPIRAL NEGOTIATION

TEST CATEGORY: PROOF

RECOMMENDED TEST SITE:

- o Test Track will consist of two types of spiral track, one perturbed, the other unperturbed.
- o With each, the Class of track will be appropriate for the maximum test speed.
- o Test zone requires about 350 ft spiral to traverse from tangent track to 2^o curve with 3" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 200 ft spiral to traverse from tangent track to 10^o curve with 2.5" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 430 ft spiral to traverse from tangent track to 5^o curve with 2.5" S.E. for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 50 ft of reverse spiral between th 10^o and 5^o curves and 500 ft of spiral from tangent to 2^o curve with 6" S.E. for maximum speeds of 80 mph for freight & 90 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts shall be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Curvature, degrees	1-10	1	o Wheel Displacement Instrumented Truck, High Rail - Vertical*
2	Superelevation, inches	0-6		
3	Track Class	2-6		
4	Vertical Track Stiffness, kips/inch*	>225		*All four wheels.
5	Lateral Rail Stiffness, kips/inch**	>40		
6	Rail Profile	New		
7	Test Speed, mph Freight Passenger	 20-115 20-130		
8	Underbalance (ΔE), inches	Variable		
9	Rail Surface Condition	Sanded,*** Dry		

*Tangent Stiffness under 12,000# Vertical Load.
**Secant Stiffness with Zero Vertical Load with Zero to 4000# Lateral Load.
***Rail Friction Coefficient of 0.15 to 0.3.

TEST PLAN SUMMARY (CONT'D)

SPIRAL NEGOTIATION PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	9	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	3	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Wheel Vertical Displacement Relative to Rail	Peak	<0.5"

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	W/R Vertical Displacement	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:
DYNAMIC CURVING

TEST CATEGORY:
PROOF

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 240 ft of 2^o curve with 3" S.E., for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 1190 ft of 2^o curve with 1" S.E., for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 870 ft of 5^o curve with 2.5" S.E., for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 720 ft of 10^o curve with 2.5" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56-57		o Wheel/Rail Forces Instrumented Truck, Total Truck
2	Curvature, degrees	2-10	1	- Lateral
3	Superelevation, inches	0-3	2	- Vertical
4	Track Class	2-4		o Body Accelerations at C.G.
5	Track Alignment Amplitude, inches	1.5-3	3 4	- Roll - Pitch
6	Track Alignment Wavelength, ft.	19.5-78	5 6	- Bounce - Yaw
7	Crosslevel Amplitude, inches	2	7	- Sway
8	Crosslevel Wavelength, ft.	19.5-78		
9	Vertical Track Stiffness, kips/inch*	>225		
10	Lateral Rail Stiffness, kips/inch**	> 40		
11	Rail Profile	New		
12	Test Speed, mph Freight Passenger	 10-65 10-80		
13	Underbalance (ΔE), inches	0-8		
14	Rail Surface Condition	Sanded,*** Dry		
	*Tangent Stiffness Under 12,000# Vertical Load. **Secant Stiffness with Zero Vertical Load with Zero to 4000# Lateral Load. ***Rail Friction Coefficient 0.15 to 0.3.			

DYNAMIC CURVING

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	12	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	6	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force (Leading)	L_{95}***	<60 kips
2	Truck L/V (Leading)	$(L/V)_{95}$	<0.5
3	Carbody Yaw Angle	Peak to Peak	$<2^{o}$**
4	Carbody Roll Angle	Peak to Peak	$<7^{o}$

**For 40' truck center distance. Proportionally lower for longer distances.

***L_{95} indicates 95 percentile level

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5
3	Carbody Roll Angle	Maximum	3.5^{o}
4	Carbody Yaw Angle	Maximum	1^{o}

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

STEADY BUFF AND DRAFT

TEST CATEGORY:

PROOF

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 1200 ft of 2° curve with 1" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 800 ft of 5° curve with 2.5" S.E. for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 500 ft of 10° curve with S.E. 2.5", for maximum speeds of 30 mph for freight and 35 mph for passenger cars joined to a 500 ft reverse curve with 0" S.E.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56-57		o Wheel/Rail Forces
				Lead Truck,
				Total Truck
2	Curvature, degrees	2-10	1	- Vertical
3	Superelevation, inches	0-3	2	- Lateral
4	Grade, percent	0-2		Trailing Truck
5	Track Class	2-4		Total Truck
6	Test Speed, mph	Variable	3	- Lateral
7	Underbalance (ΔE), inches	Variable	4	- Vertical
8	Acceleration/Deceleration Rates, mph	-0.45 to +0.3		
9	Longitudinal Forces, kips	Up to ±250K		
10	Rail Surface Condition	Sanded,* Dry		
	*Rail Friction Coefficient of 0.15 to 0.3.			

TEST PLAN SUMMARY (CONT'D)

STEADY BUFF & DRAFT

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	9	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	4	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force (Both Trucks of a Selected Car)	Mean	<60 kips
2	Truck L/V (Both Trucks of a Selected Car)	Mean	0.5

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Forces	Maximum	60 kips
2	Truck L/V	Maximum	0.5

TEST PLAN SUMMARY

PERFORMANCE ISSUE:
LONGITUDINAL TRAIN ACTION

TEST CATEGORY:
PROOF

RECOMMENDED TEST SITE:

- o Test Track, FAST, RTT and/or well maintained revenue service track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 10,000 ft of unperturbed tangent track with various grades.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56-57		o Wheel/Rail Forces
2	Grade, percent	Various		Leading Truck, Total Truck
3	Test Speed, mph	Variable	1	- Lateral
4	Acceleration/Deceleration Rates, mph/s	-0.45 to +0.3	2	- Vertical
				Trailing Truck, Total Truck
5	Longitudinal Forces, kips	Up to ±250K	3	- Lateral
			4	- Vertical
				o Body Accelerations at C.G.
			5	- Longitudinal

LONGITUDINAL TRAIN ACTION

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	10	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	4	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force (Both Trucks of a Selected Car)	L_{95} **	<60 kips
2	Truck L/V (Both Trucks of a Selected Car)	$(L/V)_{95}$	<0.5
	**L_{95} indicates 95 percentile level		

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5

TEST PLAN SUMMARY

PERFORMANCE ISSUE:
LONGITUDINAL IMPACT

TEST CATEGORY:
PROOF

RECOMMENDED TEST SITE:

- o Test Track.
- o Test zone requires 1000 ft of unperturbed tangent track for maximum speeds of 5 to 15 mph.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.
- o Add additional lengths for acceleration/deceleration for test speed range.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Test Speed, mph	0-15		o Body Accelerations at C.G.
			1	- Pitch
			2	- Bounce
			3	- Longitudinal
				o Bolster Displacement
			4	- Bounce
				o Truck Frame Accelerations Instrumented Truck
			5	- Pitch
			6	- Bounce

TEST PLAN SUMMARY (CONT'D)

LONGITUDINAL IMPACT

PROOF

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	11	None
SAMPLING RATE (Hz)	100	None
QUICK LOOK CHANNELS	7	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Peak	<1°
2	Carbody-Bolster Relative Bounce Displacement	Peak	<2"

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle *Unless it is a destructive test.	Maximum	1°*

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

HUNTING

TEST CATEGORY:

DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track or RDL
- o Test zone requires 3000 ft of unperturbed tangent track for maximum speeds of 115 mph for freight and 130 mph for passenger cars
- o Class of track will be appropriate for the maximum test speed
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56.5		o Wheel/Rail Forces
2	Track Class	3-6		Instrumented Truck,
				Total Truck
3	Track Alignment		1	- Lateral
	Amplitude, inches	0.5-2	2	- Vertical
				Instrumental Axle
4	Rail Profile	New, Worn	3	- Lateral
5	Test Speed, mph		4	- Vertical
	Freight	30-115		o Body Accelerations at C.G.
	Passenger	30-130	5	- Yaw
6	Rail Surface Condition	Sanded,* Dry	6	- Sway
				o Truck Frame Accelerations
				Instrumented Truck
			7	- Yaw
			8	- Sway
	*Rail Friction Coefficient of 0.15-0.3			o Truck Frame Displacement
				Instrumented Truck
			9	- Yaw
			10	- Sway
				o Axle Acceleration
				Instrumented Truck
			11	- Lateral
				o Axle Displacement
				Instrumented Truck
			12	- Lateral
			13	- Yaw
				o Wheel Displacement
				Instrumented Truck, High Rail
			14	- Lateral
			15	- Angle of Attack

TEST PLAN SUMMARY (CONT'D)

HUNTING

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	19	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	8	none

*Includes five channels for speed, ALD, temperature and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Acceleration	Damping, (from Time Response)	>0.1
2	Carbody Sway Acceleration	Damping (from Time Response)	>0.1
3	Carbody Sway Acceleration	Peak	<0.55g
4	Carbody Sway Acceleration	RMS	<0.1g

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Wheel Lateral Force	Maximum	25 kips
2	Single Vertical Wheel Force Time Duration at Zero Value	Maximum	0.5 secs.
3	Single Wheel L/V	Maximum	0.8
4	Truck Lateral Force	Maximum	60 kips
5	Truck L/V	Maximum	0.5
6	Carbody Sway Acceleration	Maximum	0.6g

TEST PLAN SUMMARY

PERFORMANCE ISSUE:
TWIST & ROLL

TEST CATEGORY:
DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track or RDL (See Section F for perturbation details).
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 870 ft of perturbed tangent track for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 390 ft of perturbed tangent track for maximum speeds of 65 mph for freight and 80 mph for passenger cars.
- o All additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES

NO.	VARIABLE	RANGE
1	Track Class	2-4
2	Crosslevel Amplitude, inches	1-2
3	Crosslevel Wavelength, ft	39-78
4	Vertical Track Stiffness kips/in*	>225
5	Test Speed, mph	
	Freight	10-65
	Passenger	10-80

*Tangent Stiffness under 12,000# Vertical Load.

RESPONSE VARIABLES

NO.	VARIABLE
	o Wheel Rail Forces
	Instrumented Truck, Instrumented Axle
1	- Vertical
	o Body Accelerations at C.G.
2	- Roll
	o Bolster Displacement
3	- Roll
4	- Bounce
	o Truck Frame Displacement
	Instrumented Truck
5	- Roll
	o Wheel Displacement
	Instrumented Truck,
	- High Roll
6	- Vertical

TWIST & ROLL DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	14	none
SAMPLING RATE (Hz)	200	none
QUICK-LOOK CHANNELS	8	none

*Includes five channels for speed, ALD, temperature and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Roll Angle	Peak to Peak	<7°
2	Carbody-Bolster Relative Roll	Paak to Peak	<4°
3	Vertical Wheel Force	Maximum Zero Force Duration	<0.5 sec

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Vertical Wheel Force Time Duration at Zero Value	Maximum	0.5 sec
2	Carbody Roll Angle	Maximum	3.5°
3	W/R Vertical Displacement	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

PITCH + BOUNCE

TEST CATEGORY:

DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track or RDL (See Section F for perturbation details.).
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 640 ft of perturbed tangent track for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 240 ft of perturbed tangent track for maximum speeds of 65 mph for freight and 80 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2-4		o Body Accelerations at C.G.
			1	- Pitch
2	Track Profile Amplitude, inches	2-3	2	- Bounce
3	Track Profile Wavelength, ft	19.5-39		o Bolster Displacement
			3	- Bounce
4	Vertical Track Stiffness, kips/in*	>225		o Truck Frame Displacement Instrumented Truck
			4	- Pitch
5	Test Speed, mph		5	- Bounce
	Freight	10-65		o Coupler Displacement
	Passenger	10-80		- Vertical
	*Tangent Stiffness under 12,000# Vertical Load			

PITCH + BOUNCE

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	12	none
SAMPLING RATE (Hz)	20	none
QUICK LOOK CHANNELS	5	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Peak to Peak	<2°
2	Carbody Bounce Accelerations at C.G.	Peak	<0.5 g
3	Truck-Car Relative Bounce Displacement	Peak	< 3"
4	Carbody-Bolster Relative Bounce Motion	Peak	< 2"

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Maximum	1°
2	Truck-Car Relative Bounce Displacement	Maximum	3"

TEST PLAN SUMMARY

PERFORMANCE ISSUE: YAW + SWAY

TEST CATEGORY: DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track or RDL (See Section F for perturbation details).
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 240 ft of perturbed tangent track for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 870 ft of perturbed tangent track for maximum speeds of 65 mph for freight and 80 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2-4		o Wheel/Rail Forces Instrumented Truck, Total Truck
2	Alignment Amplitude, inches	1.5-3	1	- Lateral
			2	- Vertical
3	Alignment Wavelength, ft	19.5-78		o Body Accelerations at C.G.
4	Lateral Rail Stiffness, kips/in*	>40	3 4	- Yaw - Sway
5	Rail Profile	New		o Truck Frame Displacement Instrumented truck
6	Test Speed, mph Freight Passenger	 10-65 10-80	 5 6	 - Yaw - Sway
7	Rail Surface Condition	Sanded,** Dry	 7	o Coupler Displacement - Lateral
	*Secant Stiffness with zero Vertical Load with zero to 4000# Lateral Load. **Rail Friction Coefficient of 0.15-0.3			

YAW + SWAY

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	13	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	7	none

*Includes five channels for speed ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Car Body Yaw Angle	Peak to Peak	<2°**
2	Carbody Sway Accelerations at C.G.	Peak	<0.5 g
3	Truck Lateral Force	L_{95}***	<60 kips
4	Truck L/V	$(L/V)_{95}$	<0.5
	**For 40' truck center distance. Proportionally lower for higher truck center distance.		
	***L_{95} indicates 95 percentile level		

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
1	Truck L/V	Maximum	0.5
3	Carbody Yaw Angle	Maximum	1°

TEST PLAN SUMMARY

PERFORMANCE ISSUE:
STEADY STATE CURVING

TEST CATEGORY:
DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will appropriate with the maximum test speed.
- o Test zone requires 2000 ft of 1^o curve for maximum speeds of 115 mph for freight and 130 mph for passenger cars; 200 ft of 2^o curve with 3" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 800 ft of 2^o curve with 6" S.E. for maximum speeds of 80 mph for freight and 90 mph for passenger cars; 1200 ft of 2^o curve with 1" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 500 ft of 5^o curve with 0" S.E. for maximum pseeds of 30 mph for freight and 35 mph for passenger cars; 800 ft of 5^o curve with 2.5" S.E. for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 500 ft of 5^o curve reversing into 10^o curve with S.E. going from 0" to 2.5" for maximum speeds of 30 mph for freight and 35 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES

NO.	VARIABLE	RANGE
1	Track Gauge	56-57
2	Curvature, degrees	1-10
3	Superelevation, inches	0-6
4	Track Class	2-6
5	Rail Profile	New
6	Test Speed, mph	
	Freight	20-115
	Passenger	20-130
7	Underbalance (ΔE), inches	0-8
8	Rail Surface Condition	Sanded,* Dry
	*Rail Friction Coefficient of 0.15-0.3.	

RESPONSE VARIABLES

NO.	VARIABLE
	o Wheel/Rail Forces
	Instrumented Truck,
	Total Truck
1	- Lateral
2	- Vertical
	Instrumented Axle
3	- Lateral
4	- Vertical
	o Truck Frame Displacement
	Instrumented Truck
5	- Yaw
	Wheel Displacement
	Instrumented Truck,
	High Rail
6	- Angle of Attack

TEST PLAN SUMMARY (CONT'D)

STEADY STATE CURVING

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	13	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	6	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Angle of Attack of Leading Axle	Peak	<1°
2	Wheel Lateral Force (Leading, High Rail)	Mean	<20 kips
3	Wheel L/V (Leading, High Rail)	Mean	<0.8

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Wheel Lateral Force	Maximum	25 kips
2	Single Wheel L/V	Maximum	0.8

TEST PLAN SUMMARY

PERFORMANCE ISSUE: SPIRAL NEGOTIATION

TEST CATEGORY: DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track will consist of two types of spiral track, one perturbed, the other unperturbed.
- o With each, the class of track will be appropriate for the maximum test speed.
- o Test zone requires about 350 ft spiral to traverse from tangent track to 2° curve with 2.5" S.E. for maximum speeds of 45 mph for freight + 55 mph for passenger cars, 50 ft of reverse spiral between the 10° and 5° curves, and 500 ft spiral from tangent to 2° curve with 6" S.E. for maximum speeds of 80 mph for freight +90 mph for passenger cars.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Curvature, degrees	1-10		o Wheel/Rail Forces
2	Superelevation, inches	0-6		Lead Truck
				All Wheels
3	Track Class	2-6	1	- Lateral
			2	- Vertical
4	Vertical Track Stiffness, kips/in*	>225		Trailing Truck
				All Wheels
5	Lateral Rail Stiffness, kips/in**	>40	3	- Lateral
			4	- Vertical
6	Rail Profile	New		o Body Accelerations of C.G.
7	Test Speed, mph		5	- Rail
	Freight	20-115	6	- Yaw
	Passenger	20-130	7	- Sway
8	Underbalance (ΔE), inches	Variable		o Bolster Displacement
9	Rail Surface Condition	Sanded,** Dry	8	- Roll
				o Truck Frame Displacement
				Instrumented Truck,
				High Rail
				- Vertical
	*Tangent Stiffness under 12,000# Vertical Load.			*All four wheels.
	**Secant Stiffness with zero Vertical Load with zero to 4000# Lateral Load.			
	***Rail Friction Coefficient of 0.15-0.3			

SPIRAL NEGOTIATION

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	26	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	7	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Side L/V (All Trucks Both Sides)	$(L/V)_{95}$ **of Maximum	<0.6
2	Truck Side V (All Trucks Both Sides)	V_{95} of Minimum	>0
3	Wheel Unloading Index (See Twist + Roll Perform. Indices)	Peak	<0.7
4	Wheel Vertical Displacement Relative to Rail	Peak	<0.5"
	**$(L/V)_{95}$ indicates 95 percentile level		

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Vertical Wheel Force Time Duration at Zero Value	Maximum	0.5 sec
2	Wheel/Rail Vertical Displacement	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE: DYNAMIC CURVING

TEST CATEGORY: DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 240 ft of 2° curve with 3" S.E., for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 1190 ft of 2° curve with 1" S.E., for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 870 ft of 5° curve with 2.5" S.E., for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 720 ft of 10° curve with 2.5" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES

NO.	VARIABLE	RANGE
1	Track Gauge, inches	56-57
2	Curvature, degrees	2-10
3	Superelevation, inches	0-3
4	Track Class	2-4
5	Alignment Amplitude, inches	1.5-3
6	Alignment Wavelength, ft	19.5-78
7	Crosslevel Amplitude, inches	2
8	Crosslevel Wavlength, ft	19.5-78
9	Vertical Track Stiffness, kips/in*	>225
10	Lateral Rail Stiffness, kips/in*	>40
11	Rail Profile	New
12	Test Speed, mph Freight Passenger	 10-65 10-80
13	Underbalance (ΔE), inches	0-8
14	Rail Surface Condition	Sanded,*** Dry

*Tangent Stiffness under 12,000# Vertical Load.
**Secant Stiffness with zero Vertical Load with zero to 4000# Lateral Load.
***Rail Friction Coefficient of 0.15-0.3

RESPONSE VARIABLES

NO.	VARIABLE
	o Wheel/Rail Forces
	Instrumented Truck, Total Truck
1	- Lateral
2	- Vertical
	Instrumented Axle (High/Four Wheel)
3	- Lateral
4	- Vertical
	All Wheels
5	- Lateral
6	- Vertical
	o Body Accelerations at C.G.
7	- Roll
8	- Pitch
9	- Bounce
10	- Yaw
11	- Sway
	o Bolster Displacement
12	- Roll
	o Truck Frame Displacement
	Instrumented Truck
13	- Roll
14	- Yaw
15	- Sway

DYNAMIC CURVING

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	22	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	8	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Wheel Lateral Force (Leading, High Rail)	L_{95}***	<20 kips
2	Wheel L/V (Leading, High Rail)	$(L/V)_{95}$	<0.8
3	Truck Lateral Force (Leading)	L_{95}	<60 kips
4	Truck L/V (Leading)	$(L/V)_{95}$	<0.5
5	Carbody Yaw Angle	Peak to Peak	<2°**
6	Carbody Roll Angle	Peak to Peak	<7°

**For 40' truck center distance. Proportionally lower for longer distances.

***L_{95} indicates 95 percentile level

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5
3	Carbody Roll Angle	Maximum	3.5°
4	Carbody Yaw Angle	Maximum	1°

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

STEADY BUFF + DRAFT

TEST CATEGORY:

DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 1200 ft for 2° curve with 1" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 800 ft of 5° curve with 2.5" S.E. for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 500 ft of 10 curve with 2.5" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars joined to a 500 ft reverse curve with 0" S.E.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES

NO.	VARIABLE	RANGE
1	Track Gauge, inches	56-57
2	Curvature, degrees	2-10
3	Superelevation, inches	0-3
4	Grade, percent	0-2
5	Track Class	2-4
6	Test Speed, mph	Variable
7	Underbalance (ΔE), inches	Variable
8	Acceleration/Deceleration Rates, mph/s	-0.45 to +0.3
9	Longitudinal Forces, kips	Up to +250K
10	Rail Surface Condition	Sanded, Dry*

*Rail Friction Coefficient of 0.15 to 0.3.

RESPONSE VARIABLES

NO.	VARIABLE
	o Wheel/Rail Forces
	Lead Truck,
	Total Truck
1	- Lateral
2	- Vertical
	Lead Axle (High/Low Wheel)
3	- Lateral
4	- Vertical
	Trailing Truck,
	Total Truck
5	- Lateral
6	- Vertical
	o Coupler Forces
	Both Couplers
7	- Axial
8	- Lateral
	o Coupler Displacement
	Both Couplers
9	- Axial
10	- Lateral

TEST PLAN SUMMARY (CONT'D)

STEADY BUFF + DRAFT

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	21	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	6	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force (Both Trucks of a Selected Car)	Mean	<60 kips
2	Truck L/V (Both Trucks of a Selected Car)	Mean	<0.5
3	Coupler Longitudinal Force (Both Couplers of a Selected Car)	Mean	<200 kips
4	Coupler Lateral Angle (Both Couplers of a Selected Car)	Mean	<20°

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

LONGITUDINAL TRAIN ACTION

TEST CATEGORY:

DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track, FAST, RTT and/or well maintained revenue service track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 10,000 ft of unperturbed tangent track with various grades.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches.	56-57		o Wheel/Rail Forces
2	Grade, percent	Various		Lead Truck,
3	Test Speed, mph	Variable		Total Truck
			1	- Lateral
4	Acceleration/Deceleration Rates, mph/s	-0.45 to +0.3	2	- Vertical
				Trailing Truck,
5	Longitudinal Forces, kips	Up to ±250K		Total Truck
			3	- Lateral
			4	- Vertical
				o Body Accelerations at C.G.
			5	- Longitudinal
				o Coupler Forces
				Both Couplers
			6	- Vertical
			7	- Axial
			8	- Lateral
				o Coupler Displacement
				Both Couplers
				- Vertical
				- Axial
				- Lateral

LONGITUDINAL TRAIN ACTION | DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	22	none
SAMPLING RATE (Hz)	200	none
QUICK LOOK CHANNELS	6	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force*	L_{95}***	<60 kips
2	Truck L/V*	$(L/V)_{95}$	<0.5
3	Coupler Longitudinal Force**	Peak	<200 kips
4	Coupler Lateral Angle **	Peak	<20

*Both trucks of a selected car.
**Both couplers of a selected car.
***L_{95} indicate 95 percentile level

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

LONGITUDINAL IMPACT

TEST CATEGORY:

DIAGNOSTIC

RECOMMENDED TEST SITE:

- o Test Track.
- o Test zone requires 1000 ft of unperturbed tangent track for maximum speeds of 5 to 15 mph.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.
- o Add additional lengths for acceleration/deceleration for test speed range.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Test Speed, mph	0-15		o Body Accelerations at C.G.
			1	- Pitch
			2	- Bounce
			3	- Longitudinal
				o Bolster Displacement
			4	- Bounce
				o Truck Frame Accelerations Instrumented truck
			5	- Pitch
			6	- Bounce
			7	- Longitudinal
				o Truck Frame Displacement Instrumented Truck
			8	- Longitudinal
				o Coupler Forces Both Couplers
			9	- Vertical
			10	- Axial
			11	- Lateral
				o Coupler Displacement Both Couplers
			12	- Vertical
			13	- Axial
			14	- Lateral

LONGITUDINAL IMPACT

DIAGNOSTIC

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	25	none
SAMPLING RATE (Hz)	100	none
QUICK LOOK CHANNELS	8	none

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Peak	< 1°
2	Carbody-Bolster Relative Bounce Displacement	Peak	< 2"
3	Coupler Vertical Force	Peak	50 kips

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle *Unless it is a destructive test.	Maximum	3"*

TEST PLAN SUMMARY

PERFORMANCE ISSUE: HUNTING

TEST CATEGORY: SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- o Test Track or RDL.
- o Test zone requires 3000 ft of unperturbed tangent track for maximum speeds of 115 mph for freight and 130 mph for passenger cars.
- o Class of track will be appropriate for the maximum test speed.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unintentional perturbations in the unperturbed parts should be less than those Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56.5		o Wheel/Rail Forces Instrumented Truck Total Truck
2	Track Class	3-6	1	- Lateral
3	Track Alignment Amplitude, inches	0.5-2	2	- Vertical
4	Rail Profile	New/Worn		Instrumented Axle
			3	- Lateral
5	Test Speed, mph		4	- Vertical
	Freight	30-115		o Body Accelerations at C.G.
	Passenger	30-130	5	- Yaw
6	Rail Surface Condition	Sanded,* Dry,Wet	6	- Sway
				o Truck Frame Accelerations Instrumented Truck
			7	- Yaw
			8	- Sway
	*Rail Friction Coefficient of 0.15-0.3			o Truck Frame Displacement Instrumented Truck
			9	- Yaw
			10	- Sway
				o Axle Acceleration Instrumented Truck
			11	- Lateral
				o Axle Displacement Instrumented Truck
			12	- Lateral
			13	- Yaw
				o Wheel Displacement Instrumented Truck, High Rail
			14	- Lateral
			15	- Angle of Attack

TEST PLAN SUMMARY (CONT'D)

HUNTING

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	22	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	8	None

*Includes five channels for speed, ALD, temperature and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Acceleration	Damping (From Time Response)	>0.1
2	Carbody Sway Acceleration	Damping (From Time Response)	>0.1
3	Carbody Sway Acceleration	Peak	<0.55g
4	Carbody Sway Acceleration	RMS	<0.1g

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Wheel Lateral Force	Maximum	25 kips
2	Single Vertical Wheel Force Time Duration at Zero Value	Maximum	0.5 secs
3	Single Wheel L/V	Maximum	0.8
4	Truck Lateral Force	Maximum	60 kips
5	Truck L/V	Maximum	0.5
6	Carbody Sway Acceleration	Maximum	0.6g

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

TWIST & ROLL

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- Test Track or RDL (See Section F for perturbation details).
- Class of track will be appropriate for the maximum test speed.
- Test zone require 1110 ft of perturbed tangent track for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 390 ft of perturbed tangent track for maximum speeds of 65 mph for freight and 80 mph for passenger cars.
- Add additional lengths for acceleration/deceleration for test speed range.
- Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2-4	1	o Rail & Tie Deflections - Vertical
2	Crosslevel Amplitude, inches	1-3		o Wheel/Rail Forces Instrumented Truck Instrumented Axle
3	Crosslevel Wavelength, feet	39-78	2	- Vertical All Wheels
4	Vertical Track Stiffness, kips/inch*	90-150& >225	3	- Vertical
				o Body Accelerations at C.G.
5	Test Speed, mph Freight Passenger	 10-65 10-80	4	- Roll
				o Bolster Displacement
			5	- Roll
			6	- Bounce
	*Tangent Stiffness under 12,000 # Vertical Load			o Truck Frame Displacement Instrumented Truck
			7	- Roll
				o Wheel Displacement Instrumented Truck, High Rail
			8	- Vertical

TEST PLAN SUMMARY (CONT'D)

TWIST & ROLL

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	15	10
SAMPLING RATE (Hz)	200	200
QUICK LOOK CHANNELS	5	6

*Includes five channels for speed, ALD, temperature, and such

PERFORMANCE INDICES (Px):*

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Roll Angle	Peak to Peak	< 7°
2	Carbody - Bolster Relative Roll	Peak to Peak	< 4°
3	Vertical Wheel Force	Max. Zero Force Duration	< o.5 SEC
4	Wheel Unloading Index $I = WL/\frac{WH}{3}$ WL = Vert. Force on most lightly loaded wheel WH = Sum of Vert. Forces on three most heavily loaded wheels	Peak	< 0.7

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Vertical Wheel Force Time Duration at Zero Value	Maximum	0.5 secs
2	Carbody Roll Angle	Maximum	3.5°
3	Wheel Rail Vertical Displacement	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

PITCH & BOUNCE

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- Test Track (see Section F for perturbation details).
- Test zone requires 880 ft of Class 2 perturbed tangent track for maximum test speed of 30 mph for freight & 35 mph for passenger cars; 870 ft of Class 4 perturbed tangent track for maximum test speed of 65 mph for freight and 80 mph for passenger cars.
- Add additional lengths for acceleration/deceleration for test speed range.
- Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2-4		o Rail & Tie Deflections Either Rail
2	Track Profile Amplitude, inches	2-3	1	- Vertical
3	Track Profile Wavelength, feet	19.5-78		o Body Accelerations at C.G.
			2	- Pitch
			3	- Bounce
4	Track Vertical Stiffness, kips/inch*	90-150 >225		o Bolster Displacement
			4	- Bounce
5	Test Speed, mph Freight Passenger	 10-65 10-80		o Truck Frame Displacement Instrumented Truck
			5	- Pitch
			6	- Bounce
	*Tangent Stiffness under 12,000 # Vertical Load			o Coupler Displacement Both Couplers
			7	- Vertical

TEST PLAN SUMMARY (CONT'D)

PITCH & BOUNCE

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	12	10
SAMPLING RATE (Hz)	20	200
QUICK LOOK CHANNELS	5	6

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Peak to Peak	<2°
2	Carbody Bounce Acceleration at CG	Peak	<0.5g
3	Truck-Car Relative Bounce Displacement	Peak	<3"
4	Carbody - Bolster Relative Bounce Motion	Peak	<2"

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Maximum	1°
2	Truck Car Relative Bounce Displacement	Maximum	3"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

YAW & SWAY

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- o Test Track (see Section F for perturbation details).
- o Test zone requires 480 ft of perturbed tangent track for maximum test speeds of 30 mph for freight and 35 mph for passenger cars; 710 ft of perturbed tangent track for maximum test speeds of 65 mph for freight and 80 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections but does not include transition lengths that would be needed between sections.
- o Unintentional pertubations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Class	2-6		o Rail & Tie Deflections
2	Alignment			Either Rail
	Amplitude, inches	0.5-5.0	1	- Lateral
3	Alignment			o Wheel/Rail Forces
	Wavelength, feet	19.5-78		Instrumented Truck
				Total Truck
4	Lateral Rail Stiffness,	15-240	2	- Lateral
	kips/inch*	>40	3	- Vertical
5	Rail Profile	New,		Instrumented Axle
		Worn	4	- Lateral
			5	- Vertical
6	Test Speed, mph			o Body Accelerations at C.G.
	Freight	10-115	6	- Yaw
	Passenger	10-120	7	- Sway
7	Rail Surface Condition	Sanded**, Dry,Wet		o Truck Frame Displacement
				Instrumented Truck
			8	- Yaw
			9	- Sway
				o Axle Acceleration
				Instrumented Truck
	*Secant Stiffness with zero Vertical Load with zero to 4000 # Lateral Load		10	- Lateral
				o Axle Displacement
				Instrumented Truck
	**Rail Friction Coefficient of 0.15-0.3		11	- Yaw
				o Wheel Displacement
				Instrumented Truck
			12	- Lateral
			13	- Angle of Attack
				o Coupler Displacement
				Both Couplers
			14	- Lateral

YAW & SWAY

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS*	21	10
SAMPLING RATE (Hz)	200	200
QUICK LOOK CHANNELS	7	6

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Yaw Angle	Peak to Peak	<2°**
2	Carbody Sway Accelerations at CG	Peak	<0.5g
3	Truck Lateral Force	L_{95}***	<60 kips
4	Truck L/V	$(L/V)_{95}$	<0.5

**For 40' truck center distance. Proportionally lower for higher truck center distance.

***L_{95} indicates 95 percentile level

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5
3	Carbody Yaw Angle	Maximum	1°
4	Track Gage at Rail Head	Maximum	58.5"
5	Panel Shift	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

STEADY STATE CURVING

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- Test Track.
- Class of track will be appropriate for maximum test speed.
- Test zone requires 2000 ft of 1° curve for maximum speeds of 115 mph for freight and 130 mph for passenger cars; 200 ft of 2° curve with 3" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 800 ft of 2° curve with 6" S.E. for maximum speeds of 80 mph for freight & 90 mph for passenger cars; 1200 ft of 2° curve with 1" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 500 ft of 5° curve with 0" S.E. for maximum speeds of 30 mph for freight & 35 mph for passenger cars; 800 ft of 5° curve with 2.5" S.E. for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 1050 ft of 5° curve reversing into 10° curve with S.E. going from 0" to 2.5" for maximum speeds of 30 mph for freight and 35 mph for passenger cars.
- Add additional lengths for acceleration/deceleration for test speed range.
- Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES

NO.	VARIABLE	RANGE
1	Track Gauge, inches	56-57
2	Track Curvature, degrees	1-10
3	Superelevation, inches	0-6
4	Track Class	2-6
5	Track Profile	New, Worn
6	Test Speed, mph	
	Freight	20-115
	Passenger	20-130
7	Underbalance (ΔE), inches	0-8
8	Rail Surface Condition	Sanded,* Dry, Wet

*Rail friction coefficient of 0.15-0.3

RESPONSE VARIABLES

NO.	VARIABLE
	o Rail & Tie Deflections
	High Rail
1	- Lateral
	o Wheel/Rail Forces
	Lead Truck,
	Total Truck
2	- Lateral
3	- Vertical
	Instrumented Axle
4	- Lateral
5	- Vertical
	Trailing Truck
	Instrumental Axle
6	- Lateral
7	- Vertical
	o Truck Frame Displacement
	Instrumented Truck
8	- Yaw
	o Axle Displacement
	Instrumented Truck
9	- Lateral
10	- Yaw
	o Wheel Displacement
	Instrumented Truck
11	- Lateral
12	- Angle of Attack

TEST PLAN SUMMARY (CONT'D)

STEADY STATE CURVING

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	18	4
SAMPLING RATE (Hz)	200	200
QUICK LOOK CHANNELS	6	4

*Involves five channels for speed, ALD, temperature and such.

PERFORMANCE INDICES (Px):*

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Angle of Attack of Leading Axle	Peak	$<1^{0}$
2	Wheel Lateral Force (Leading, High Rail)	Mean	<20 kips
3	Wheel L/V (Leading, High Rail)	Mean	<0.8

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Wheel Lateral Force	Maximum	25 kips
2	Single Wheel L/V	Maximum	0.8

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

SPIRAL NEGOTIATION

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- Test Track will consist of two types of spiral track, one perturbed, the other unperturbed.
- With each, the class of track will be appropriate for the maximum test speed.
- Test zone requires about 350 ft spiral to traverse from tangent track to 2° curve with 3" S.E. for maximum speeds of 65 mph for freight & 80 mph for passenger cars; 200 ft spiral to traverse from tangent track to 10° curve with 2.5" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars; 430 ft spiral to traverse from tangent track to 5° curve with 2.5" S.E. for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 50 ft of reverse spiral between the 10° and 5° curves and 500 ft spiral from tangent to 2° curve with 6" S.E. for maximum speeds of 80 mph for freight & 90 mph for passenger cars.
- Unintentional perturbations in the unperturbed parts shall be less than those for Class 6 track.
- Add additional lengths for acceleration/deceleration for test speed range.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Curvature, degrees	1-10		o Rail & Tie Deflection High Rail
2	Superelevation, inches	0-6	1	- Lateral
3	Track Class	2-6		o Wheel/Rail Forces Lead Truck, All Wheels
4	Vertical Track Stiffness, kips/inch*	90-150& >225	2	- Lateral
5	Lateral Rail Stiffness, kips/inch***	15-25 & >40	3	- Vertical
6	Rail Profiles	New, Worn		Trailing Truck, All Wheels
			4	- Lateral
7	Test Speed, mph		5	- Vertical
	Freight	20-115		o Body Accelerations at C.G.
	Passenger	20-130	6	- Roll
			7	- Yaw
8	Underbalance (ΔE), inches	Variable	8	- Sway
9	Rail Surface Condition	Sanded*** Dry,Wet		o Bolster Displacement
			9	- Roll
				o Truck Frame Displacement Instrumented Truck
			10	- Roll
	*Tangent Stiffness under 12,000# Vertical Load		11	- Yaw
			12	- Sway
	**Secant Stiffness with Zero Vertical Lead with Zero to 4000 # Lateral Load			o Wheel Displacement Instrumented Truck
			13	- Lateral
	***Rail friction coefficient of 0.15 to 0.3		14	- Yaw
				o Wheel Displacement Instrumented Truck, High Rail
			15	- Lateral
			16	- Angle Attack
			17	- Vertical

SPIRAL NEGOTIATION

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	30	4
SAMPLING RATE (Hz)	200	200
QUICK LOOK CHANNELS	7	4

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Side L/V (All Trucks Both Sides)	$(L/V)_{95}$**of Max.	<0.6
2	Truck Side V (All Trucks Both Sides)	V_{95} of Max.	>0
3	Wheel Unloading Index (see Twist & Roll Perf. Indices)	Peak	<0.7
4	Wheel Vertical Displacement Relative to Rail	Peak	<0.5"
	**$(L/V)_{95}$ indicates 95 percentile level		

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Single Vehicle Wheel Force Time Duration at Zero Value	Maximum	0.5 sec
2	W/R Vertical Displacement	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:
DYNAMIC CURVING

TEST CATEGORY:
SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 870 ft of 2" curving with 3" S.E. for maximum speeds of 65 mph for freight & 80 mph for passenger cars; 1900 ft of 2" curve with 1" S.E., for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 870 ft of 5° curve with 2.5" S.E., for maximum speeds of 45 mph for freight and 55 mph for passenger cars; 160 ft of 5° curve with 0" S.E. for maximum speeds of 30 mph for freight and 35mph for passenger cars; 1360 ft of 10° curve with 2.5" S.E., for maximum speeds of 30 mph for freight & 35 mph for passenger cars.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Includes internal transition lengths between subsections, but does not include transition lengths that would be needed between sections.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES

NO.	VARIABLE	RANGE
1	Track Gauge, inches	56.57
2	Curvature, degrees	2-10
3	Superelevation, inches	0-3
4	Track Class	2-4
5	Alignment Amplitude, inches	1.5-5
6	Alignment Wavelength, feet	19.5-78
7	Crosslevel Amplitude, inches	1-2
8	Crosslevel Wavelength, feet	19.5-78
9	Vertical Track Stiffness, kips/inch*	90-150 & >225
10	Lateral Rail Stiffness kips/inch**	15-25 >40
11	Rail Profile	New,Worn
12	Test Speed, mph Freight Passenger	10-65 10-80
13	Underbalance (ΔE), inches	0-8
14	Rail Surface Condition	Sanded,*** Dry,Wet

*Tangent Stiffness under 12,000 # Vertical Load

**Secant Stiffness with zero Vertical Load with zero to 4000 # Lateral Load

***Rail friction coefficient of 0.15-0.3

RESPONSE VARIABLES

NO.	VARIABLE
	o Rail & Tie Deflections
	High Rail
1	- Lateral
2	- Vertical
	o Wheel/Rail Forces
	Lead Truck,
	Total Truck
3	- Lateral
	- Vertical
	Instrumented Axle
5	- Lateral
6	- Vertical
	All Wheels
7	- Lateral
8	- Vertical
	Trailing Truck,
	Instrumented Axle
9	- Lateral
10	- Vertical
	o Body Accelerations at C.G.
11	- Roll
12	- Pitch
13	- Bounch
14	- Yaw
15	- Sway
	o Bolster Displacement
16	- Roll
	o Truck Frame Accelerations
	Instrumented Truck
17	- Yaw
18	- Sway
	o Truck Frame Displacement
	Instrumented Truck
19	- Roll
20	- Yaw
21	- Sway
	o Axle Displacement
	Instrumented Truck
22	- Lateral
23	- Yaw
	o Wheel Displacement
	Instrumented Truck
24	- Lateral
25	- Angle of Attack
26	- Vertical

DYNAMIC CURVING

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	31	20
SAMPLING RATE (Hz)	200	200
QUICK LOOK CHANNELS	8	8

*Includes five channels for speed, ALD, temperature, and such

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Wheel Lateral Force (Leading, High Rail)	L_{95}***	<20 kips
2	Wheel L/V (Leading, High Rail)	$(L/V)_{95}$	<0.8
3	Truck Lateral Force (Leading)	L_{95}	<60 kips
4	Truck L/V (Leading)	$(L/V)_{95}$	<0.5
5	Carbody Yaw Angle	Peak to Peak	<2°**
6	Carbody Roll Angle	Peak to Peak	<7°

**For 40' truck center distance. Proportionally lower for longer distance.

***L_{95} indicates 95 percentile level

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5
3	Carbody Roll Angle	Maximum	3.5°
4	Carbody Yaw Angle	Maximum	1°
5	Track Gage at Rail Head	Maximum	58.5"
6	Panel Shift	Maximum	0.5"

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

STEADY BUFF & DRAFT

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- o Test Track.
- o Class of track will be appropriate for the maximum test speed.
- o Test zone requires 1200 ft of 2° curve with 1" S.E. for maximum speeds of 65 mph for freight and 80 mph for passenger cars; 800 ft of 5° curve with 2.5" S.E. maximum speeds of 45 mph for freight and 55 mph for passenger cars; 1050 ft of 5° curve reversing into 10° curve with 2.5" S.E. for maximum speeds of 30 mph for freight and 35 mph for passenger cars joined to a 500 ft reverse curve with 0" S.E.
- o Add additional lengths for acceleration/deceleration for test speed range.
- o Unitentional perturbations in the unperturbed parts should be less than those for Class 6 track.
- o

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56-57		o Rail & Tie Deflections
2	Curvature, degrees	2-10		High Rail
			1	- Lateral
3	Superelevation, inches	0-3		o Wheel Rail Forces
4	Grade, percent	0-2		Lead Truck,
5	Track Class	2-4		Total Truck
			2	- Lateral
6	Test Speed, mph	Variable	3	- Vertical
7	Underbalance (ΔE), inches	Variable		Instrumented Truck,
			4	- Lateral
			5	- Vertical
8	Acceleration/Deceleration Rates, mph/s	-0.45 to +0.3		Trailing Truck,
				Total Truck
9	Longitudinal Forces, kips	Up to $\pm$250K	6	- Lateral
			7	- Vertical
10	Rail Surface Condition	Sanded,* Dry,Wet		o Coupler Forces
				Both Couplers
			8	- Axial
			9	- Lateral
				o Coupler Displacement
				Both Couplers
			10	- Axial
	*Rail friction coefficient of 0.15-0.3		11	- Lateral

STEADY BUFF AND DRAFT

SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	21	4
SAMPLING RATE (Hz)	200	200
QUICK LOOK CHANNELS	6	4

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px):*

* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force (Both Trucks of a Selected Car)	Mean	<60 kips
2	Truck L/V (Both Trucks of a Selected Car)	Mean	<0.5
3	Coupler Longitudinal Force (Both Couplers of a Selected Car)	Mean	<200 kips
4	Coupler Lateral Angle (Both Couplers of a Selected Car)	Mean	<20°

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

LONGITUDINAL TRAIN ACTION

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- Test Track, FAST, RTT and/or well maintained revenue service track.
- Class of track will be appropriate for the maximum test speed.
- Test zone requires 10,000 ft of unperturbed tangent track with various grades.
- Add additional lengths for acceleration/deceleration for test speed range.
- Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Track Gauge, inches	56-57		o Wheel Rail Forces
				Lead Truck,
				Total Truck
2	Grade, percent	Various	1	- Lateral
3	Test Speed, mph	Variable	2	- Vertical
4	Acceleration/Deceleration Rates, mph/s	-0.45 to +0.3		Trailing Truck
				Total Truck,
5	Longitudinal Forces, kips	Up to +250K	3	- Lateral
			4	- Vertical
				o Body Accelerations at C.G.
			5	- Longitudinal
				o Coupler Forces
				Both Couplers
			6	- Vertical
			7	- Axial
			8	- Lateral
				o Coupler Displacement
				Both Couplers
			9	- Vertical
			10	- Axial
			11	- Lateral

LONGITUDINAL TRAIN ACTION — SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	22	None
SAMPLING RATE (Hz)	200	None
QUICK LOOK CHANNELS	6	None

*Includes five channels for speed, ALD, temperature, and such.

PERFORMANCE INDICES (Px): *

*(SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force**	L_{95}****	<60 kips
2	Truck L/V**	$(L/V)_{95}$	<0.5
3	Coupler Longitudinal Force***	Peak	<200 kips
4	Coupler Lateral Angle ***	Peak	<20°

**Both trucks of a selected car
***Both couplers of a selected car
****L_{95} indicates percentile level

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Truck Lateral Force	Maximum	60 kips
2	Truck L/V	Maximum	0.5

TEST PLAN SUMMARY

PERFORMANCE ISSUE:

LONGITUDINAL IMPACT

TEST CATEGORY:

SERVICE ENVIRONMENT

RECOMMENDED TEST SITE:

- o Test Track.
- o Test zone requires 1000 ft of unperturbed tangent track for maximum speeds of 5 to 15 mph.
- o Unintentional perturbations in the unperturbed parts should be less than those for Class 6 track.
- o Add additional lengths for acceleration/deceleration for test speed range.

CONTROL VARIABLES			RESPONSE VARIABLES	
NO.	VARIABLE	RANGE	NO.	VARIABLE
1	Test Speed, mph	5-15		o Body Accenerations at C.G.
			1	- Pitch
			2	- Bounce
			3	- Yaw
			4	- Sway
			5	- Longitudinal
				o Bolster Displacement
			6	- Bounce
				o Truck Frame Accelerations Instrumented Truck
			7	- Pitch
			8	- Bounce
			9	- Longitudinal
				o Truck Frame Displacement Instrumented Truck
			10	- Longitudinal
				o Coupler Forces Both Couplers
			11	- Vertical
			12	- Axial
			13	- Lateral
				o Coupler Displacement Both Couplers
			14	- Vertical
			15	- Axial
			16	- Lateral
			17	o Structural Stress
			18	o Body Deformation

TEST PLAN SUMMARY (CONT'D)

LONGITUDINAL IMPACT | SERVICE ENVIRONMENT

DATA HANDLING REQUIREMENTS:

	ONBOARD	WAYSIDE
NO. OF DATA CHANNELS *	37**	None
SAMPLING RATE (Hz)	100	None
QUICK LOOK CHANNELS	8	None

*Includes five channels for speed, ALD, temperature, and such.
**Assumes ten channels for structural stress and deformation

PERFORMANCE INDICES (Px): *
* (SEE SUBSECTION 3.2 FOR OBTAINING Px'S FROM ELEMENTS)

NO.	OUTPUT RESPONSE VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Peak	<1°
2	Carbody - Bolster Relative Bounce Displacement	Peak	<2"
3	Coupler Vertical Force	Peak	<50 kips

SAFETY CRITERIA:

NO.	SAFETY VARIABLE	STATISTIC	THRESHOLD
1	Carbody Pitch Angle	Maximum	1°*
	*Unless it is a destructive test.		

SECTION F
TEST FACILITIES

F.1 Introduction

While planning for the tests prescribed by the IAT, the user is confronted with several choices in selecting a test site. Essentially, the tests can be performed on a revenue service track, as is or modified, or they can be performed on an existing test facility. Primary among the test facilities for user consideration are:

- The Test Tracks at the Transportation Test Center (TTC), at Pueblo, Colorado;
- The Rail Dynamics Laboratory (RDL), located at TTC; and
- The Stability Assessment Facility for Equipment (SAFE).

Of these, the first two already exist; the last facility does not, but its detailed plans are available. This design will prove useful, should a user contemplate modifying revenue service track or building a dedicated test facility.

The selection of the proper test site is based on many considerations. The primary consideration is the ability of the test site to provide the input excitations described in Subsection 3.1.1 of Part 1 and summarized in Table 3-2 in Part I. This means that the test site should be able to provide the required:

- Nominal Track Geometry (Curvature, Grade);
- Track Geometry Irregularities;
- Track/Rail Stiffnesses; and
- Rail Geometry.

In addition, the site should be able to run vehicles within the required speed range.

With this background, Table 3-7 in Part I provides a guideline for selecting proper test site for each combination of Performance Issue and Test Category.

The table shows that one of the existing TTC tracks can be used to address all Performance Issues, except longitudinal train action, assuming that the perturbations required to study twist and roll, pitch and bounce, yaw and sway, and dynamic curving will be installed on one of the test tracks by the user. The details of each appropriate test track at TTC are provided in the following subsections.

F.2 The Existing Test Tracks at TTC

Located near Pueblo, Colorado, the TTC enjoys a relatively rain-free climate, which is ideal for field testing. Figure F-1 (Ref. [1]) shows an overview of the existing tracks at TTC. These tracks are of ballast and wood tie constructions with cut spikes except for a few test segments of concrete ties with elastic fasteners. Of the tracks shown, the following can be used for performing the IAT related tests:

- the Facility for Accelerated Service Testing (FAST),
- the Railroad Test Track (RTT),
- the Train Dynamics Track (TDT),
- the Precision Test Track (PTT), previously called the LIM Track
- the Turn Around Track, and
- the Impact Track.

A brief description of the capabilities of each appear in the following paragraph.

The Facility for Accelerated Service Testing (FAST)

The FAST, a continuous loop of 4.8 miles of tangent and curved tracks, is used to simultaneously test various track structures, rail, ties, ballast, fasteners, switches and switch components, track stability, safety equipment, vehicle components, lading techniques, maintenance methods, and equipment under heavy demand conditions (See Ref. [2]). Although not designed to address the Performance Issues of

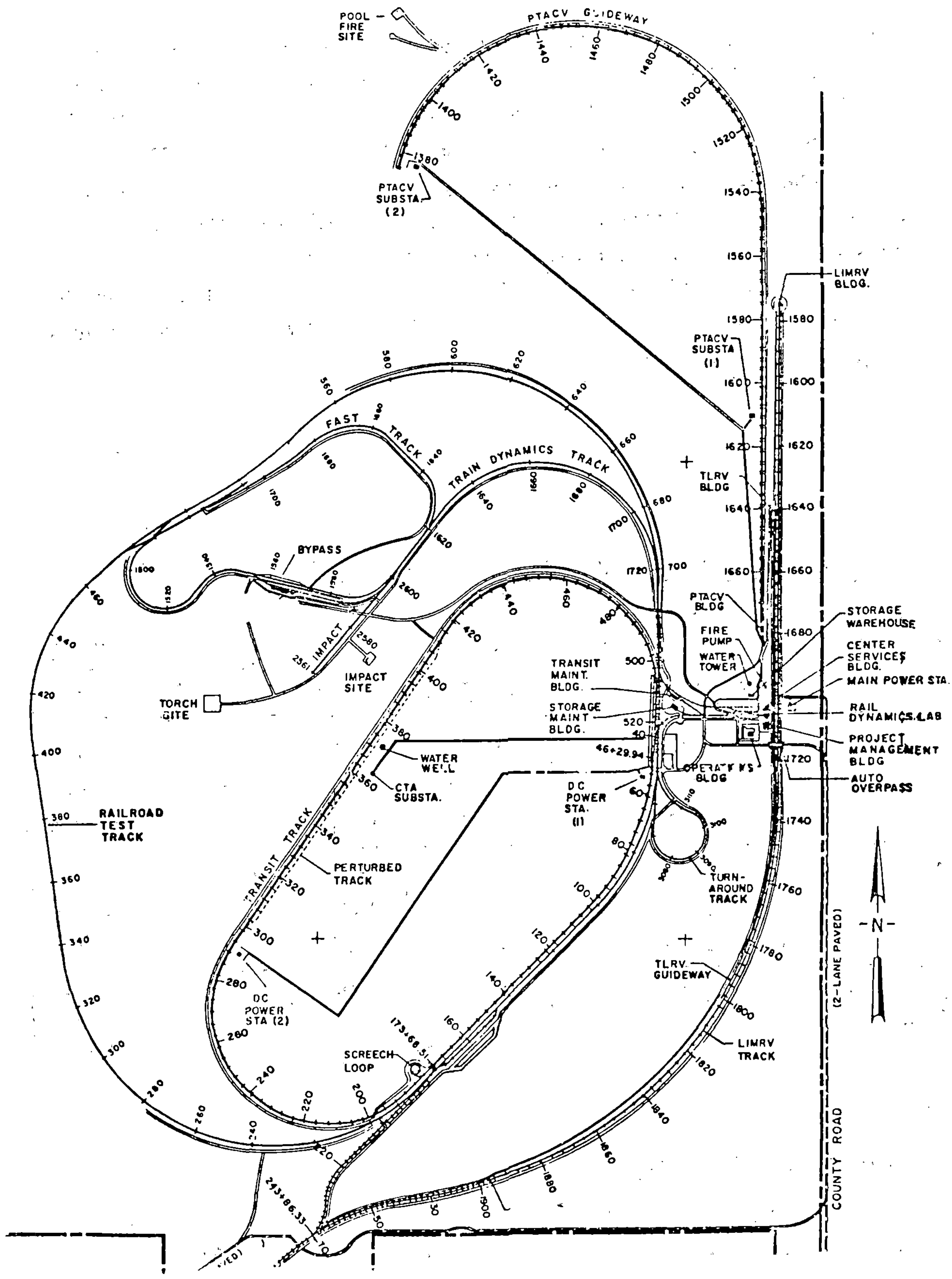


FIGURE F-1 THE EXISTING TEST TRACKS AT TTC

interest to the IAT, the FAST includes many track sections which can be used to perform tests related to steady-state curving, spiral negotiation and steady-state buff and draft. A list of principal curves on the FAST is available in Table F-1. As can be seen in the table, the FAST is useful for testing the above Performance Issues on curves up to 5 degrees. In addition, the FAST has the capability of endurance testing vehicles and track components.

Also, the FAST service facility, adjacent to the track, can be used for inspecting and performing static measurements on the various types of rail vehicles.

TABLE F-1: CHARACTERISTICS OF MAJOR CURVES IN TTC

TRACK	LOCATION (See Figure F-1)	CURVATURE	LENGTH	SUPER ELEVATION	BALANCE SPEED
RTT	260	0° 50'	13791'	5.6 "	100 mph
	430	0° 50'	7752'	5.6 "	100 mph
	640	0° 50'	15900'	5.6 "	100 mph
FAST	1500	5°	3673'	4.0 "	34 mph
	1540	5°	1000'	4.0 "	34 mph
	1600	4°	1325'	3.0 "	34 mph
TURN AROUND TRACK	3090	7° 30'	2775'	5.0 "	31 mph

Railroad Test Track (RTT)

The longest test track in the TTC is the 14 mile loop of the Railroad Test Track (RTT). It is intended to be used for high speed testing of locomotives and cars, such as the new 120 mph AMIRAK coaches. As shown in Table F-1, no curves in the RTT are tighter than 0°50'. This makes the RTT ideally suited for high speed steady-state curving. Also, the tangent parts of RTT can be used for evaluating the hunting performance of a vehicle.

Unlike FAST, the RTT does not have a regularly scheduled consist running over it all the time. Thus, it may be possible to install perturbations on the tangent, for studying twist and roll, pitch and bounce, and yaw and sway issues, and on curve for studying the dynamic curving performance. In fact, for the Perturbed Track Test conducted in 1978 (See Ref. [3]), several perturbed sections were installed on a tangent part of the RTT located between markers 340 and 370. Although these perturbations have subsequently been removed, the feasibility of incorporating them has clearly been demonstrated. If such perturbations are to be installed, it is useful to know that only the track between locations 335 and 488 (See Figure F-1) is bolted, the rest of the RTT is made up of continuously welded rail.

Train Dynamics Track (TDT)

The TDT incorporates a 1°30' curve of bolted rail which was used in the Perturbed Track Test for studying dynamic curving. These perturbations have subsequently been removed, just as in the case of the RTT, but they can be installed once more, if such need arises.

Precision Test Track -- Originally the Linear Induction Motor Test Track

The 6.2 mile-long Precision Test Track, originally constructed to study the performance of linear induction motors and the site of a world speed record for wheel-on-rail vehicles, provides an ideal location for testing the hunting, rock and roll, pitch and bounce, and yaw and sway performances of a rail vehicle. In fact, the track currently incorporates a set of profile and crosslevel perturbations (39' wavelength, 3/4" amplitude and 10 wavelengths long) for studying pitch and bounce and rock and roll performances of test vehicles. The track is of ballasted wood tie construction with elastic clip fasteners. The maximum speed on the curve is set to 100 mph.

Turn Around Track

The turn around track shown in Figure F-1 has a 7°30', 5.06" superelevation curve which is almost 2800' long. This can be used to test a vehicle consist for its performance in steady-state curving, dynamic curving, and steady buff and draft.

Impact Track

An extension of the TDT has been used for impact testing. At present, this 0.75 mile long track is being used to unload and store ballast for FAST. It may be available, with some modifications, for performing longitudinal impact assessment.

F.3 Rail Dynamics Laboratory (RDL)

The Rail Dynamics Laboratory (RDL) at TTC permits evaluation of various vehicles in a safe, controlled and reproductive laboratory environment (See Ref. [4], [5], and [6]). This is accomplished by the two testing units housed in the RDL: the Vibration Test Unit (VTU) and the Roll Dynamics Unit (RDU), along with the necessary data collection, analysis and service facilities. These are described in the following paragraphs.

The Vibration Test Unit (VTU)

The VTU, shown in Figure F-2, can subject a stationary rail vehicle to controlled vertical and lateral vibration inputs at the wheel/rail interfaces, through use of servo-controlled hydraulic actuators. The measurement, recording and analysis of dynamic responses of the test vehicle and its lading can be carried out using this unit. As currently configured, the VTU can accommodate both ends of a 320,000 pound rail vehicle with two two-axle trucks or one end of a vehicle with three- or four-axle trucks.

The input to the VTU actuators can be provided by a magnetic tape with actual profiles or by computer programs which generates synthetic

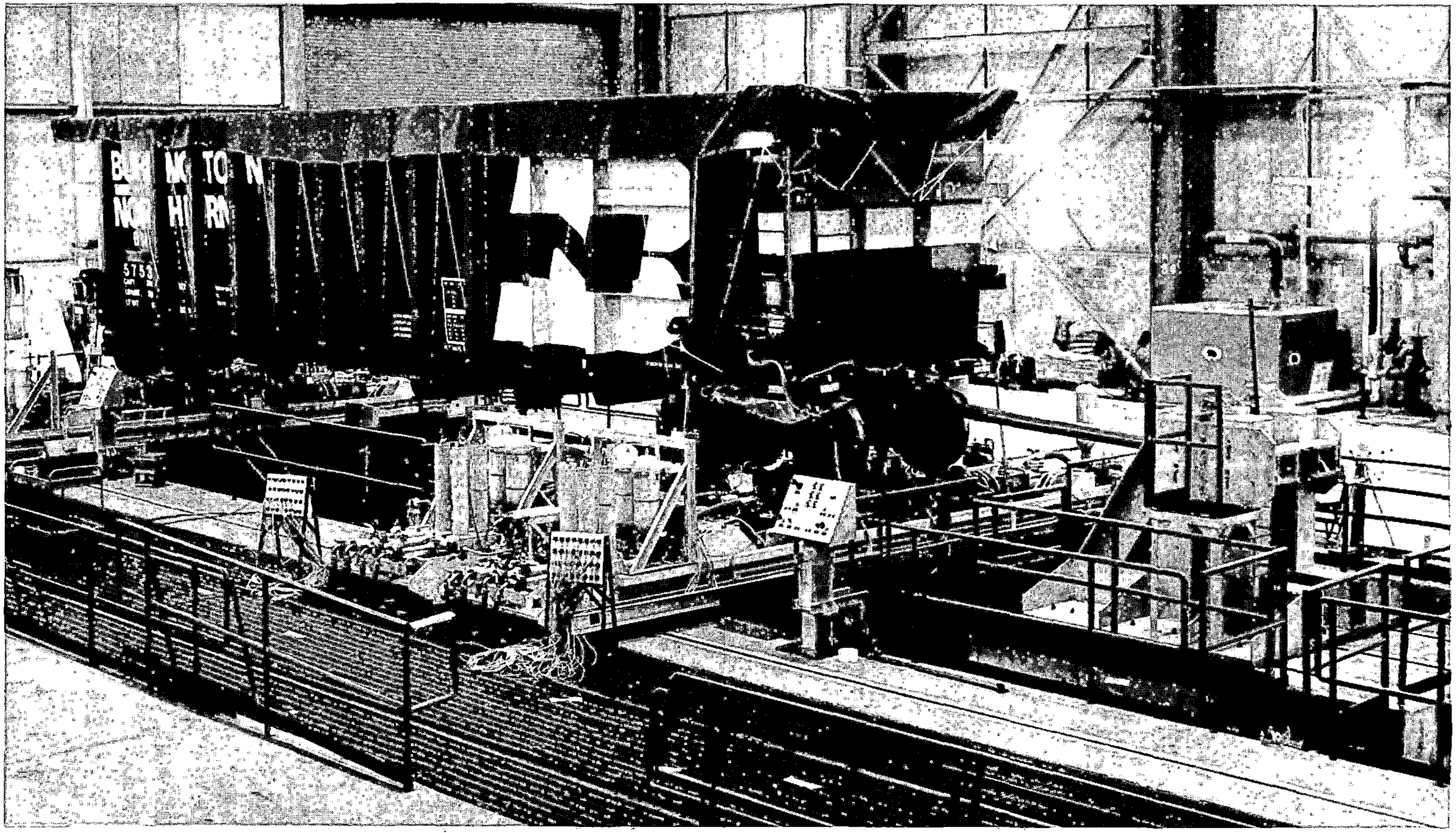


FIGURE F-2 VIBRATION TEST UNIT

sinusoidal signals. Using these inputs, the VTU can test a vehicle for twist and roll, pitch and bounce, and yaw and sway characteristics. In addition, the static twist for spiral negotiation can also be determined.

Table F-2 summarizes the capabilities of VTU. Although this summary table specifies that the vertical and lateral excitation displacement capabilities are ±2" and ±1.5" respectively, such high displacements can be achieved only at the lower end of the frequency range. This is illustrated by the amplitude versus frequency plots in Figure F-3. Now, the frequency of the simulated input depends on the speed of the vehicle and the perturbation wavelength being simulated, as shown by the equation below:

$$\text{Required frequency of input} = \frac{\text{speed of vehicle (ft/sec)}}{\text{perturbation wavelength (ft)}}$$

This assumes that the perturbations are sinusoidal. For most other shapes, the required frequency of input will be higher.

Since the speed of the vehicle and the perturbation wavelength, type and amplitude depend on the Performance Issue being addressed, as well as on the test category under consideration, the adequacy of the VTU capabilities needs to be evaluated for each realistic combination of these parameters. This is done as shown in Table F-3. In this table, the data on the speed, wavelength, and amplitude for different Performance Issues are obtained from Table 3-2 in Part I. From these data, the required amplitude values are computed for comparison with the VTU displacement capability at the required frequencies. As can be seen, the VTU seems adequate for performing twist and roll assessment, but only partly adequate for addressing the other two issues. Since the VTU cannot handle more than 3" peak to peak lateral displacement, the assessment on Track Class 1 for yaw and sway is limited to 3" amplitude (peak to peak). For the other two instances where the VTU is not adequate (i.e., Track Class 2 for pitch and bounce, and Class 4 for

TABLE F-2: SUMMARY OF VTU CAPABILITIES

Vehicle Size:

Up to 90.0' (27.4 m) long
12.0' (3.66 m) wide
Weight to 320,000 pounds (145,000 kg)

Truck Center Distance:

20' (6.10 m) Minimum
70' (21.3 m) Maximum

Truck Axle Spacing:

54.0" (1.37 m) Minimum
110.0" (2.79 m) Maximum

Center of Gravity to Railhead:

18" (.457 m) Minimum
98" (2.49 m) Maximum

Coupler Centerline to Railhead:

17.5" (.444 m) Minimum
34.5" (.876 m) Maximum

Gauge:

56.5" (1.44 m) Minimum
66.0" (1.68 m) Maximum

	Vertical Excitation	Lateral Excitation
Frequency Range	0.2 to 30 Hz	0.2 to 30 Hz
Displacement	±2" (5.08 cm) ($\leq$ 2Hz)	±1.5" (3.81 cm) ($\leq$ 2Hz)
Velocity	25 inch/sec. ($\leq$ 2Hz) (63.5 cm/sec)	15 inch/sec. ($\leq$ 2 Hz) (38.1 cm/sec)

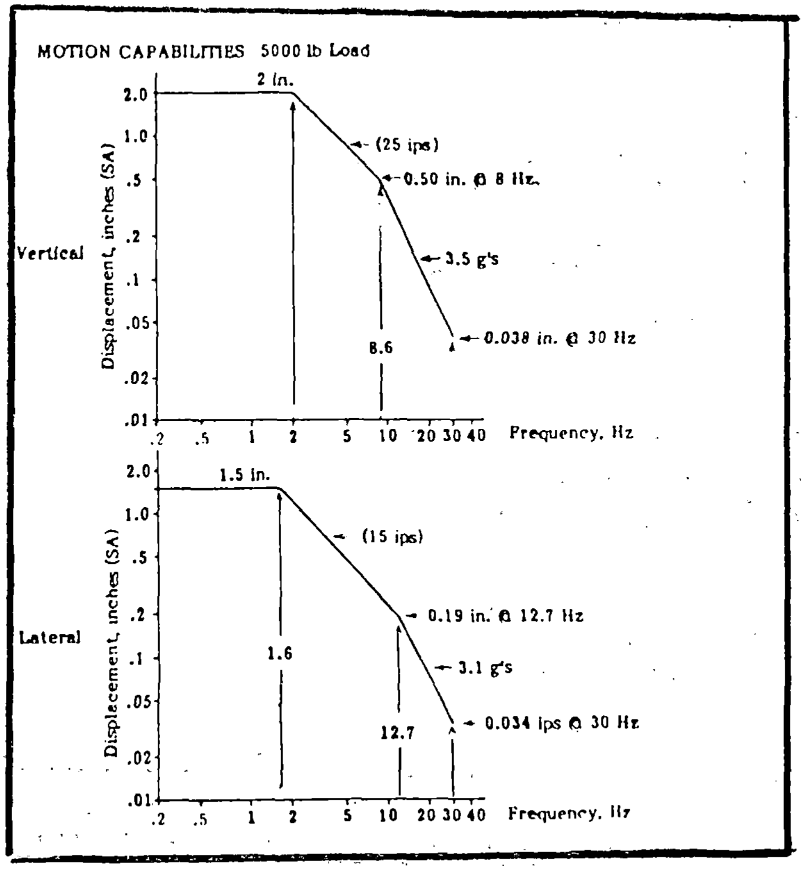


FIGURE F-3: DISPLACEMENT VERSUS FREQUENCY PLOT FOR THE VTU

TABLE F-3: ASSESSMENT OF VTU CAPABILITY TO MEET THE MOST SEVERE IAT REQUIREMENTS

TEST CATEGORY*	TRACK CLASS	MAXIMUM SPEED (MPH)	MINIMUM WAVELENGTH (FT.)	REQUIRED INPUT FREQUENCY (Hz)	REQUIRED AMPLITUDE** (PEAK INTO PEAK)	VTU CAPABILITY (PEAK INTO PEAK)	VTU ADEQUATE UP TO THE MOST SEVERE CONDITION?
Twist & Roll							
P*	2	35	39	1.3	2	4	Yes
D, S	4	80	39	3.0	1	2	Yes
Pitch & Bounce							
P, D, S	2	35	19.5	2.6	3	2.4	No
P, D, S	4	80	39	3.0	2	2.4	Yes
Yaw & Sway							
S	1	15	19.5	1.1	5	3	No
P	2	35	39	1.3	3	3	Yes
D, S	4	80	19.5	6.0	1.5	0.8	No
S	6	120	78	2.2	0.5	2	Yes

*P = Proof Test, D = Diagnostic Test, S = Service Environment Test

**In vertical direction for Twist & Roll and Pitch & Bounce, and in lateral direction for Yaw & Sway.

yaw and sway), a compromise has to be made in terms of reducing the simulated speed or amplitude; or increasing the wavelength.

In summary, the VTU may be able to test a rail vehicle for twist and roll, pitch and bounce, and yaw and sway in all except the most extreme situations.

The Roll Dynamics Unit

The RDU is capable, through a system of drive motors, flywheels and rollers, of simulating relative rail forward motion for unpowered vehicles (such as freight cars) and absorbing power produced by self propelled vehicles, including locomotives. The unit, shown in Figure F-4, consists of modular elements which can be positioned to match various truck spacings, axle spacings, and gauges of rail equipment. Each wheel of a test vehicle rests on and is driven by a supporting roller. Each pair of rollers, mounted on a common shaft, are attached to a drive train which provide sufficient inertia to assure that the interface between the vehicle wheels and the rollers adequately simulates a vehicle travelling on a stationary track.

The roller rotation simulates vehicle velocities on "perfect" tangent track. Thus, by somehow providing an initial lateral input, a vehicle's hunting behavior can be studied on the RDU. The RDU can also simulate very crudely the vehicle performance in steady-state curving by skewing the axles. However, since no capability is provided for having a differentially rotating input for opposite wheels of a wheelset, this is not considered adequate for the IAT.

The simulated speeds on RDU is from 3 to 144 mph for axle loads up to 100,000 lb. The capabilities of the RDU are outlined in Table F-4.

The Support Facilities

The VTU and RDU are supported by a data acquisition system and a computer system network for monitoring the progress of tests and for post-test data processing. The highlights of these systems are provided in Table F-5. Both of these seem adequate to meet the requirements of the IAT tests that can be performed in the RDL.

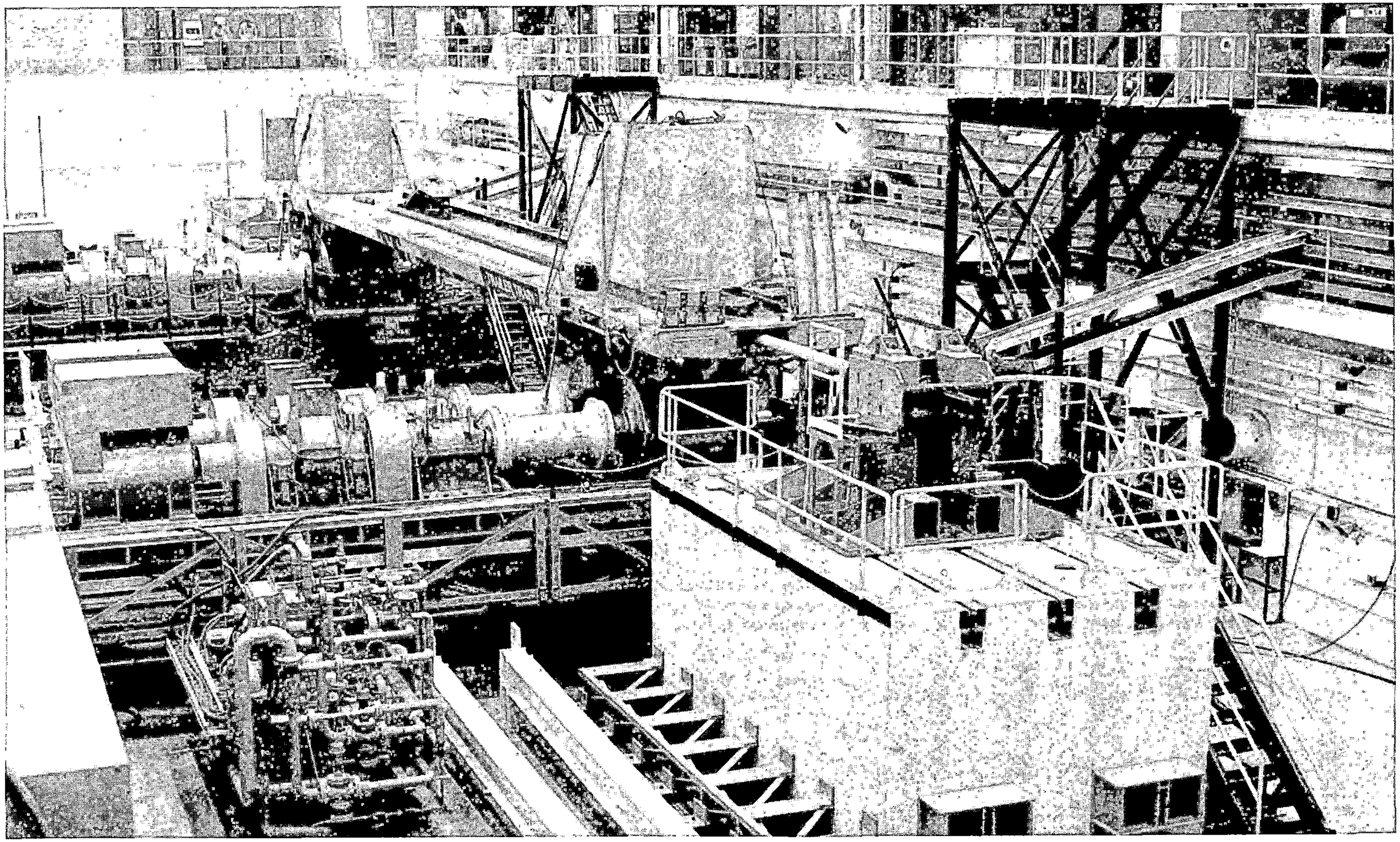


FIGURE F-4 ROLL DYNAMICS UNIT

TABLE F-4: SUMMARY OF RDU CAPABILITIES

Vehicle Size:

Up to 108' (32.9 m) long
12' wide (3.66 m)
Weight to 400,000 pounds (181,000 kg)

Truck Center Distance:

20' (6.10 m) Minimum
80' (24.4 m) Maximum

Truck Axle Spacing:

66" (1.68 m) to 110" (2.79 m) with 60" (1.52 m) Diameter Rollers

Center of Gravity to Railhead:

18" (.457 m) Minimum to 98" (2.49 m) Maximum

Coupler Centerline to Railhead:

17.5" (.444 m) Minimum to 34.5" (.876 m) Maximum

Gauge:

56.5" (1.44 m) Minimum to 66" (1.68 m) Maximum

Speed (simulated):

144 mph (232 km/h) Maximum with 60" (1.52 m) Diameter Rollers

Steady-State Curve Simulation (minimum curve radius):

100 ft. (30.5 m) for Truck Center of 50 ft. (15.2 m) or less
150 ft. (45.7 m) for Truck Center between 50 and 80 ft. (15.2 and 24.4 m)

TABLE F-5: THE SUPPORT SYSTEM AT RDL

SPECIMEN DATA ACQUISITION SYSTEM

- Multichannel System to Acquire Test Vehicle Response Data
- Consists of Sensors, Signal Conditional Electronics, Cables, Cable Junction Boxes in Laboratory, Patch Panels in Control Room, Analog Recording
  - 100 Linear Accelerations
  - 16 Angular Accelerations
  - 38 Strains
  - 10 Linear Displacements
  - 6 Angular Displacements
  - 12 Surface Temperatures
  - 4 Infra-Red Temperatures
  - 6 Sounds

  - Connect to Digital Input of ICSN
  - 14 Channel Analog Tape Recording
  - 36 Channel Light-Beam Oscillographic Recording

INTEGRATED COMPUTER SYSTEM NETWORK

- Complex of 5 Minicomputers and Peripherals
- Controls Vibration Test Unit Operations; Limited To Sinusoidal and Track Geometry Waveshapes
- Monitors Either Test Unit and Test Vehicle for Malfunctions or Exceedances
- Records 256 Channels: 128 from Test Unit (RDU or VTU)
128 from SDAS (Test Vehicle)
- Performs Post-Test Data Processing
  - Measurement Waveshapes
  - Transfer Function Plots
  - Power Spectral Density
  - Cross-Power Spectral Density
  - Amplitude vs. Frequency Plots
  - Coherence

Source: Ref. [6]

F.4 Stability Assessment Facility for Equipment (SAFE)

One of the test facilities described in this section does not exist except in terms of detailed plans. This facility is SAFE -- The Stability Assessment Facility for Equipment (Ref. [1], [7] and [8]). This facility was designed originally for construction at the TTC. If built, this facility would provide a comprehensive test ground to test almost all (except longitudinal train action and longitudinal impact) Performance Issues for all three test categories. Also, the design assures that different types of rail vehicles (freight, locomotive, and passenger) can be tested in their specific speed ranges.

Figure F-5 shows a sketch of the track layout designed for SAFE. Essentially, it consists of an outer loop made of 2° and 5° curves joined by two 2-mile long tangents. A track consisting of a 5° to 10° reverse curve and several tangent sections is placed inside the outer loop. In addition, parts of the RTT were to be modified to be included in the SAFE track.

All of the darkened lines in Figure F-5 represent test sections, some of which have specially installed crosslevel, alignment and profile perturbations (information on how this perturbation can be installed is presented in Section G-Track Geometry Perturbations). The details of each test section are presented in the Table F-6 following the figure, and a summary of their characteristics is provided in Table F-7.

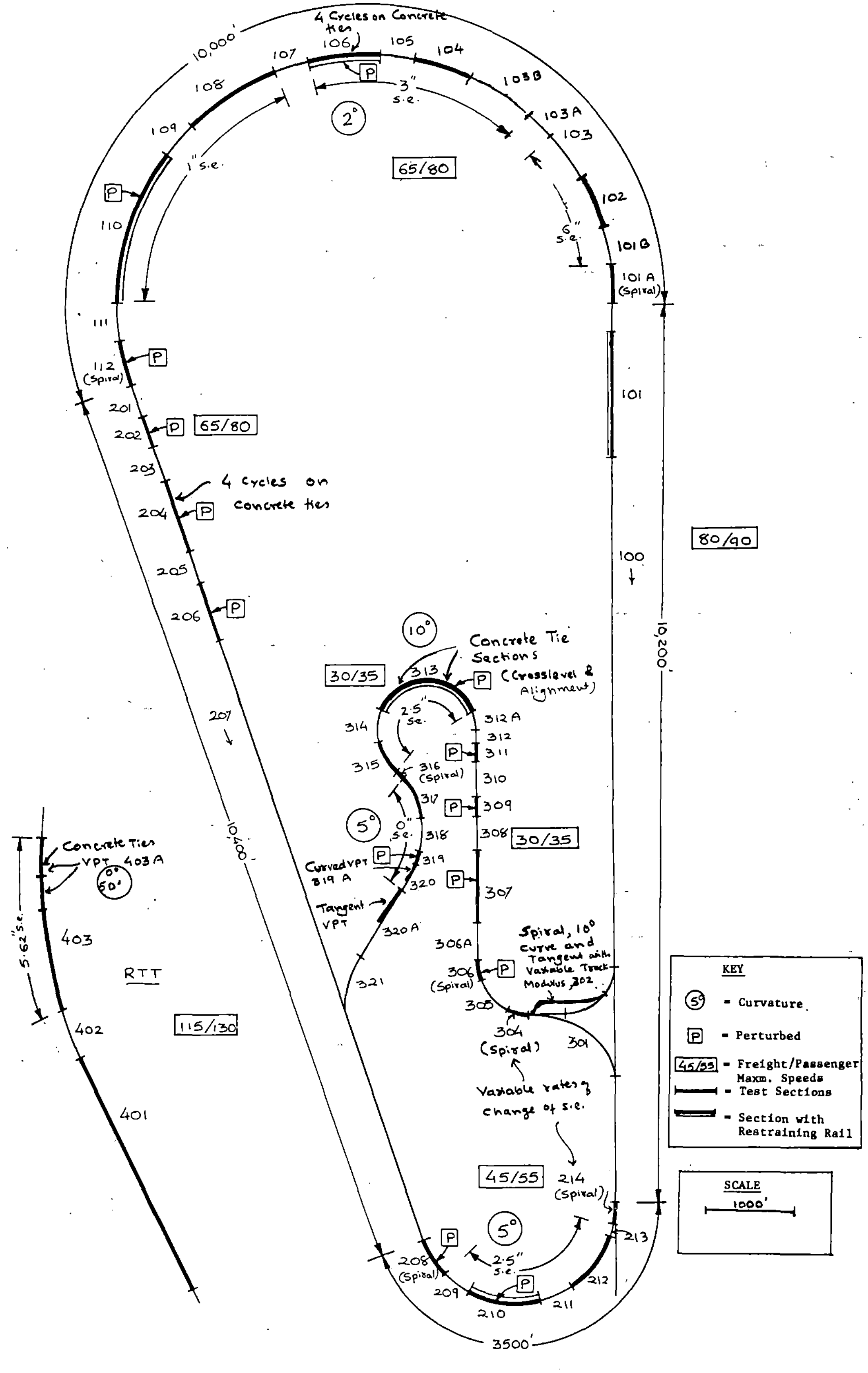
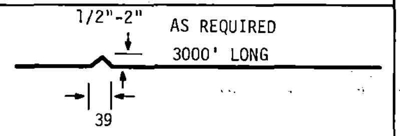


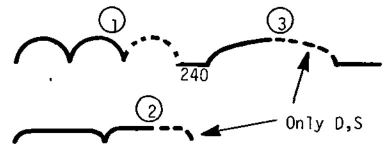
FIGURE F–5 THE PLANNED LAYOUT OF SAFE TEST TRACK

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE

TRACK CHARACTERISTICS FOR HUNTING

TEST CATEGORY	REQUIRED TRACK TYPE	CLASS	PICTORIAL REPRESENTATION	TEST SECTION NO. (Fig. F-5)
P,D,S	TRANSIENT PERTURBED TANGENT TRACK	6	 ALIGNMENT PERTURBATION	101 401

TRACK CHARACTERISTICS FOR TWIST AND ROLL

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS	PICTORIAL REPRESENTATION	
P,D	PERTURBED TRACK	① ② ③	2 4		307 202

(--- REPRESENTS CYCLES NOT SHOWN)
SERVICE TESTING REQUIRES TANGENT VPT AND VMT DESCRIBED LATER

SUBSECTION CHARACTERISTICS FOR TWIST AND ROLL

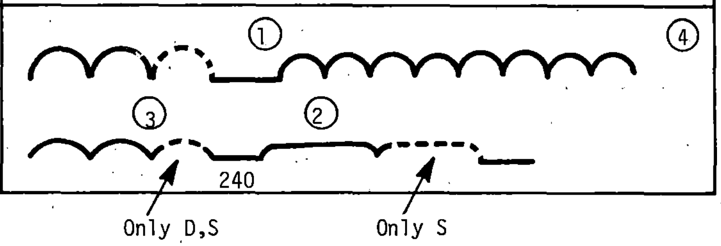
SUBSECTION NO.	TYPE	PERTURBATION CHARACTERISTICS AMPL.** IN	WAVELENGTH FT.	COMPLIANCE	NO. OF CYCLES	LENGTH FT.	CLASS	RECOMMENDED MAX. SPEED, MPH FREIGHT	PASSENGER	
①	CROSSLEVEL	2	39		6	240	2	30	35	307
②	CROSSLEVEL	1	78		5	390	4	65	80	202
③	CROSSLEVEL	2	78		5	390	2	30	35	307

**PEAK TO PEAK AMPLITUDE WITH ZERO MEAN VALUE

ΔΔ = P = PROOF TEST
D = DIAGNOSTIC TEST
S = SERVICE ENVIRONMENT TEST

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

TRACK CHARACTERISTICS FOR PITCH AND BOUNCE

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS	PICTORIAL REPRESENTATION	TEST SECTION NO. (Fig. F-5)
P,D	Perturbed Track	1 2 3	2		309
			4		204

(--- REPRESENTS CYCLES NOT SHOWN)

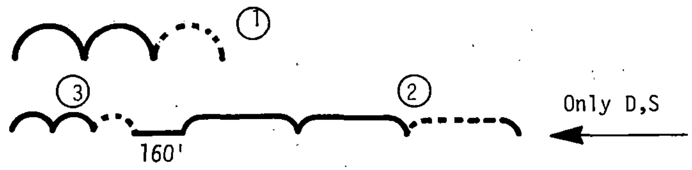
SUBSECTION CHARACTERISTICS FOR PITCH AND BOUNCE

SUBSECTION NO.	PERTURBATION CHARACTERISTICS					LENGTH FT	CLASS	RECOMMENDED MAX. SPEED, MPH		
	TYPE	MAX. AMPL. IN	WAVELENGTH FT.	COMPLIANCE	NO. OF CYCLES			FREIGHT	PASSENGER	
1	PROFILE	3	39		2-4	240	2	30	35	309
2	PROFILE	2	78		5	390	4	65	80	204
3	PROFILE	2	39		6	240	4	65	80	
4	PROFILE	3	19.5		8	160	2	30	35	309

Service Environment Test requires tangent VPT and VMT described later.

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

TRACK CHARACTERISTICS FOR YAW AND SWAY

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS	PICTORIAL REPRESENTATION	TEST SECTION NO. (Fig. F-5)
P,D	Perturbed Track	1 2 3	2 4		311 206

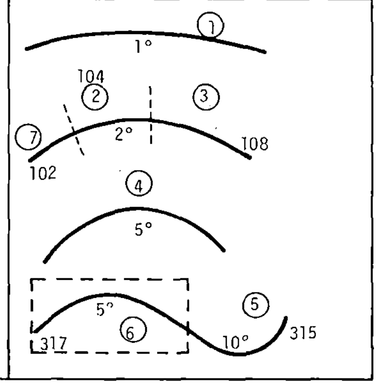
(--- REPRESENTS CYCLES NOT SHOWN)

SUBSECTION CHARACTERISTICS FOR YAW AND SWAY

SUBSECTION NO.	PERTURBATION CHARACTERISTICS					LENGTH	CLASS	RECOMMENDED MAX. SPEED, MPH		TEST SECTION NO.
	TYPE	AMPL. IN.	WAVELENGTH FT.	COMPLIANCE	NO. OF CYCLES			FREIGHT	PASSENGER	
1	ALIGNMENT	3	39		6	240	2	30	35	311
2	ALIGNMENT	1.5	78		5	390	4	65	80	206
3	ALIGNMENT	1.5	19 1/2		8	160	4	65	80	

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

TRACK CHARACTERISTICS FOR STEADY-STATE CURVING

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS**	PICTORIAL REPRESENTATION	TEST SECTION NO. (Fig. F-5)
P,D,S	UNPERTURBED CURVES	1 2 3 4 5 6* 7	6		403
			4		102 104 108
					212
			2		315 317

*6 ONLY FOR SERVICE ENVIRONMENT TESTING
**FOR SPEED LIMITS; TRACK IS MAINTAINED TO 6+

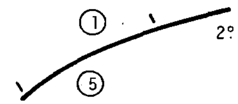
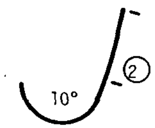
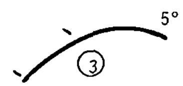
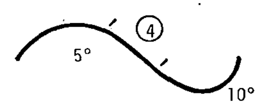
SUBSECTION CHARACTERISTICS FOR STEADY-STATE CURVING

SUBSECTION NO.	CURVATURE DEGREE	SUPERELEVATION IN.	BALANCE SPEED MPH	RECOMMENDED MAX. SPEED, MPH				LENGTH FT.	
				FREIGHT		PASSENGER			
				MPH	ΔE	MPH	ΔE		
1	1	6	92	115	3"	130	6"	2000	403
2	2	3	46	65	3"	80	6"	790	104
3	2	1	27	65	5"	80	8"	1200	108
4	5	2.5	27	45	5"	50	5"	800	212
5	10	2.5	19	30	4"	35	6" } FORMS A REVERSE CURVE	500	315
6	5	0	0	30	3"	35	4"	500	317
7	2	6	65	80	3"	90	5"	800	102

CONCRETE TIE SECTION ON 403.

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

TRACK CHARACTERISTICS FOR SPIRAL NEGOTIATION

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS*	PICTORIAL REPRESENTATION	TEST SECTION NO. (Fig. F-5)
P,D	UNPERTURBED SPIRALS	2 3 4 5	4		112(P) 101A
			2		304 306(P)
			4		208(P) 214
S	PERTURBED SPIRALS	1 2 3	2		316

SUBSECTION CHARACTERISTICS FOR SPIRAL NEGOTIATION

SUBSECTION NO	CURVATURE DEGREE		SUPERELEVATION IN		LENGTH FT.	CLASS	RECOMMENDED MAX. SPEED, MPH		
	START	END	START	END			FREIGHT	PASSENGER	
1	0	2	0	3	350	4	65	80	112(P)
2	10	0	2.5	0	200	2	30	35	304, 306(P)
3	0	5	0	2.5	430	4	45	50	208(P), 214
4	5	-10	0	2.5	50	2	30	35	316
5	0	2	0	6	500	5	80	90	101A

*FOR SPEED LIMITS; TRACK IS MAINTAINED TO 6+
(P) = PERTURBED

304 and 214 with variable rates of change of superelevation

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

TRACK CHARACTERISTICS FOR DYNAMIC CURVING

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS	PICTORIAL REPRESENTATION
P, D	Perturbed Track	1 - 8	2, 4	See Figures on Next Page
S		1 - 14	2, 4	

SUBSECTION CHARACTERISTICS

See Table following the Figures

TABLE F-6: DESCRIPTION OF THE VARIOUS TEST SEGMENTS ON SAFE (continued)

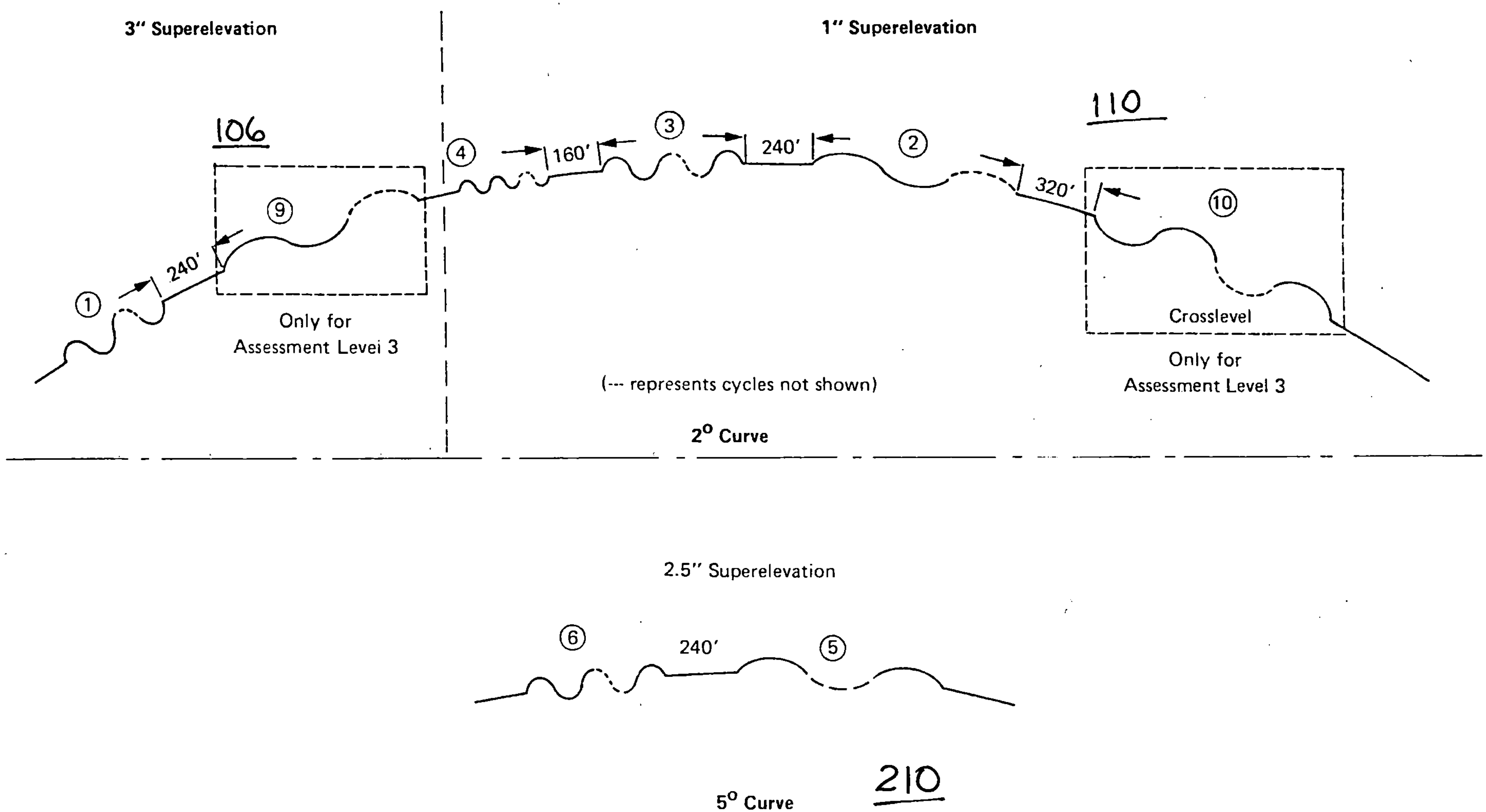
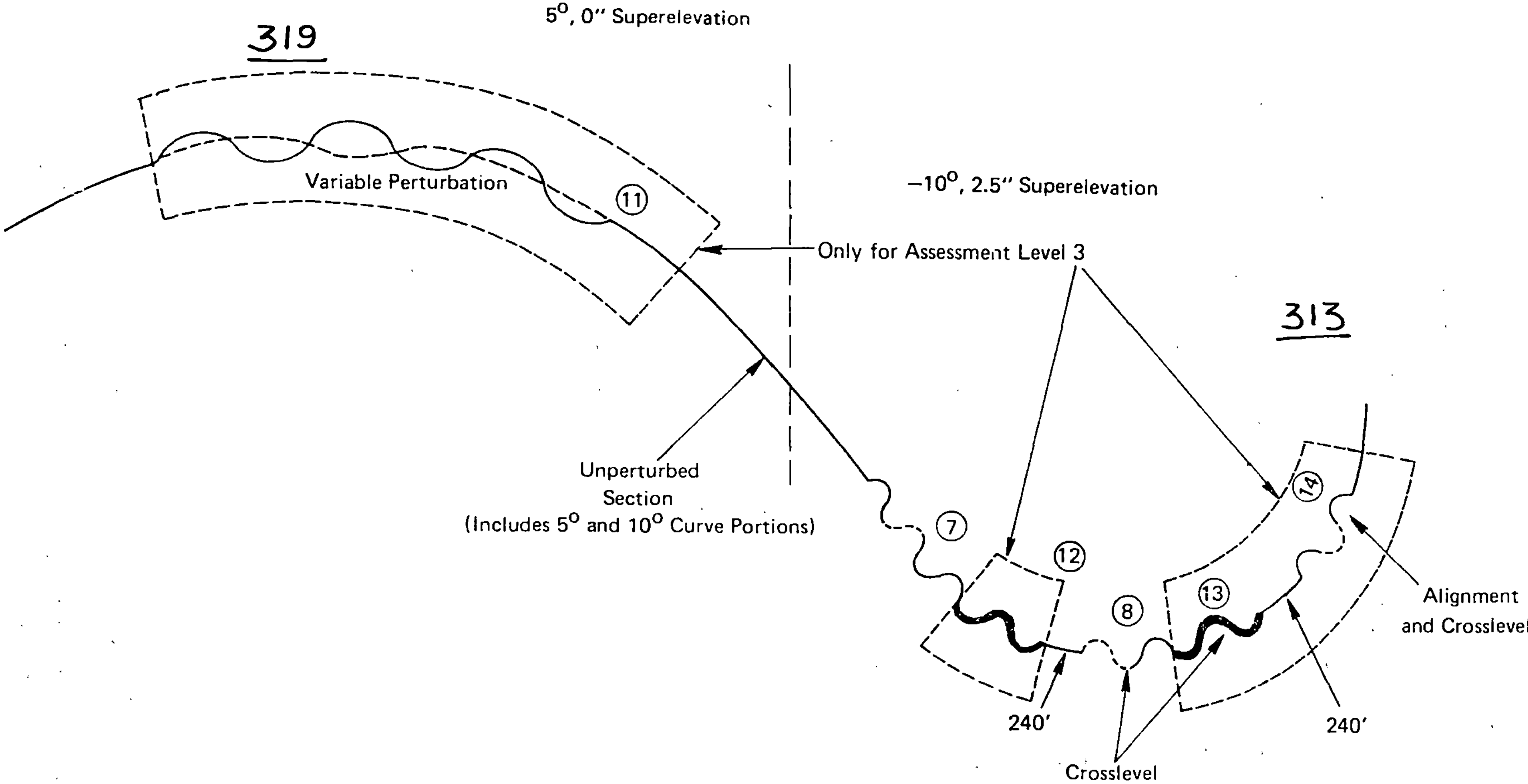


TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (continued)



5° → −10° Reverse Curve

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

SUBSECTION CHARACTERISTICS FOR DYNAMIC CURVING TRACKS

	Subsection No.	Curvature Degrees	Super Elevation in.	Balance Speed mph	RECOMMENDED MAXIMUM SPEED				Class	PERTURBATION CHARACTERISTICS						Length	Test Section Number (Fig. F-5)
					Freight		Passenger			Type	Amplitude in.	Wavelength ft.	Forced Frequency Hz		No. of Cycles		
					mph	ΔE in.	mph	ΔE in.					At Max. Speed*	At Balance Speed			
PROOF AND DIAGNOSTIC TESTING	1	2	3	46	65	3	80	6	4	Alignment†	1.5	39	3.0	1.7	6	240	106
	2	2	1	27	65	5	80	8	4	Alignment	1.5	78	1.5	0.5	5	390	110
	3	2	1	27	65	5	80	8	4	Alignment	1.5	39	3.0	1.0	6	240	110
	4	2	1	27	65	5	80	8	4	Alignment	1.5	19 1/2	6.0	2.0	8	160	110
	5	5	2.5	27	45	5	55	8	4	Alignment	1.5	78	1.0	0.5	5	390	210
	6	5	2.5	27	45	5	55	8	4	Alignment	1.5	39	2.0	1.0	6	240	210
	7	10	2.5	19	30	4	35	6	2	Alignment	3	39	1.3	0.7	6	240	313
	8	10	2.5	19	30	4	35	6	2	Crosslevel	2	39	1.3	0.7	6	240	313
ADDITIONAL TRACKS SERV. ENVRNMT. TEST	9	2	3	46	65	3	80	6	4	Alignment	1.5	78	1.5	0.8	5	390	106
	10	2	1	27	65	5	80	8	4	Crosslevel	1	78	1.5	0.5	5	390	110
	11	5	0	0	30	3	35	4	1-6	Alignment**	Up to 5	Up to 78	Variable	0	2-4	160	319
	12	10	2.5	19	30	4	35	6	2	Alignment†	3	39	1.3	0.7	2	80	313
	13	10	2.5	19	30	4	35	6	2	Crosslevel†	2	39	1.3	0.7	2	80	313
	14	10	2.5	19	30	4	35	6	2	Alignment & Crosslevel	3/2	39	1.3	0.7	6	240	313

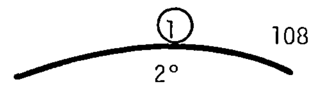
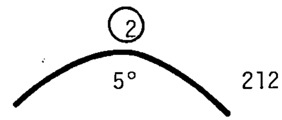
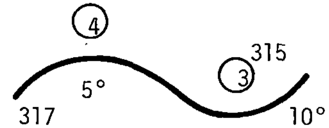
*For passenger operation.
**Variable perturbation track
†Different compliance

(Concrete tie sections)

Service Environment Testing also requires curved VPT and VMT described later.

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (CONTINUED)

TRACK CHARACTERISTICS FOR STEADY BUFF AND DRAFT

TEST CATEGORY	REQUIRED TRACK TYPE	SUBSECTION NO.	CLASS*	PICTORIAL REPRESENTATION	TEST SECTION NO. (Fig. F5)
P,D	UNPERTURBED CURVES	1 2	4		108
			4		212
S	UNPERTURBED CURVES	3 4	2		317 315

*FOR SPEED LIMITS; TRACK IS MAINTAINED TO 6+.

SUBSECTION CHARACTERISTICS FOR STEADY BUFF AND DRAFT

SUBSECTION NO.	CORRESPONDING SUBSECTION NO. IN STEADY STATE CURVING
1	3
2	4
3	5
4	6

TABLE F-6: DESCRIPTION OF VARIOUS TEST SEGMENTS ON SAFE (continued)

Variable Perturbation Tracks (VPT)

1. Tangent VPT (320A): 400' long with possibilities to vary perturbation type, wavelength, # of cycles and amplitude.

2. Curved VPT (319A): 400' long with possibilities of varying perturbation type, wavelength, # of cycles and amplitude.

3. Curved VPT (403A): similar to above, but 1000' long.

Variable Modulus Tracks (VMT)

1. Tangent, curve and spiral VMT (302): possibilities to vary the vertical and lateral track modulii on a spiral, 10° curve and tangent.

TABLE F-7: A SUMMARY OF SAFE TEST SECTION CHARACTERISTICS

TEST SECTION #	TANGENT (T) CURVE (C) SPIRAL (S)	CURVATURE	SUPERELEVATION	PERTURBATIONS (P) CONCRETE TIES (C) VARIABLE PERTURBATIONS (VP) VARIABLE MODULUS (VM)	RECOMMENDED MAX. SPEED FREIGHT/ PASSENGER	PRIMARY PERFORMANCE ISSUES ADDRESSED*
101	T	-	-	P	80/90	1
101A	S	0° → 2°	0" → 6"	-	80/90	6
102	C	2°	6"	-	80/90	5
104	C	2°	3"	-	65/80	5
106	C	2°	3"	C, P	65/80	7
108	C	2°	1"	-	65/80	5, 8
110	C	2°	1"	P	65/80	7
112	S	0° → 2°	0" → 3"	P	65/80	6
202	T	-	-	P	65/80	2
204	T	-	-	P	65/80	3
206	T	-	-	P	65/80	4
208	S	0° → 5°	0" → 2.5"	P	45/50	6
210	C	5°	2.5"	P	45/55	7
212	C	5°	2.5"	-	45/50	5, 8
214	S	0° → 5°	0" → 2.5"	-	45/50	6
302	S, C, T	0° → 10°	0", 2.5"	VM	30/35	2, 3, 4, 7
304	S	10° → 0°	2.5" → 0"	-	30/35	6
306	S	10° → 0°	2.5" → 0"	P	30/35	6
307	T	-	-	P	30/35	2
309	T	-	-	P	30/35	3
311	T	-	-	P	30/35	4
313	C	10°	2.5"	C	30/35	7
315	C	10°	2.5"	-	30/35	5, 8
316	S	5° → 10°	0" → 2.5"	-	30/35	6
317	C	5°	0"	-	30/35	5, 8
319	C	5°	0"	P	30/35	7
319A	C	5°	0"	VP	30/35	7
320A	T	-	-	VP	30/35	2, 3, 4
401	T	-	-	P	115/130	1
403	C	0° 50'	5.62"	-	115/130	5
403A	C	0° 50'	5.62"	VP, C	115/130	5, 7

*Performance Issues addressed by SAFE:

1. Hunting
2. Twist & Roll
3. Pitch & Bounce
4. Yaw & Sway
5. Steady State Curving
6. Spiral Negotiation
7. Dynamic Curving
8. Steady Buff & Draft

REFERENCES -- SECTION F

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SECTION G

TRACK GEOMETRY PERTURBATIONS

G 1.0 Introduction

One of the essential aspects of a test performed under the Vehicle/Track Interaction Assessment Techniques (IAT) is providing controlled input to the vehicle being tested through track geometry perturbations as described in Part I Section 3 of this document. Specifically, the track geometry perturbations provide inputs necessary to evaluate the dynamic response of a test vehicle in hunting, twist and roll, yaw and sway, pitch and bounce, and dynamic curving.

This section deals with track irregularities (or perturbations) that can intentionally be installed in a track to provide a test vehicle with sufficient excitation to bring out its dynamic characteristics, which is generally the intent of a test done under the IAT. Included in the description in this section are methods of creating alignment, crosslevel and profile perturbations; and measuring, and maintaining them. Reference is made to the past tests where such perturbed tracks have sucessfully been used.

G 2.0 Nomenclatures and Definitions

Gauge, is the distance between the heads of rails at right angles to the rails in a plane five-eights of an inch below the top surface of the rail.

Alignment, is the horizontal location of a railroad as described by curves and tangents.

Profile, is the surface uniformity of each rail.

Cross-level, is the amount of elevation of one rail above the other.

Superelevation, is the amount of elevation of the outside rail on curve above the inside rail.

High rail/low rail, is the designation given to the rails in curves. The outside rail on curves with superelevation is the high rail, the inside rail is the low rail.

Curvature, in degrees, is a measure of the angular change in track direction per 100-foot track chord.

Chordal offset, is the distance between the chord and the rail

M C O (Mid-chord offset), is the chordal offset at mid-chord

62 ft MCO, is chordal offset at mid-chord of a 62 feet chord. This measurement in inches is equal to the rail curvature in degrees.

Loaded measurement, is a track geometry measurement under vehicle load.

Unloaded measurement, is a track geometry measurement without vehicle load.

Rail cant, is the inward inclination of a rail, effected by the use of inclined-surface tie plates, usually expressed as a rate of inclination, such as 1 in 40, etc.

Rail size, is expressed in pounds per yard with a standardized cross-section.

Super-position of perturbation on curve, is the perturbation on curve track using the nominal curve as reference.

G.3.0 Technical Discussion

The overall test track arrangement, including amplitude, wavelength, number of cycles and perturbation used for test for any performance issue are detailed in Section F.4. This section deals with local track geometry perturbation design, fabrication, installation, measurement, and the use of adjustable fasteners.

G.3.1 TRACK GEOMETRY PERTURBATION DESIGN

The most convenient type of construction for perturbations appears to be that made up from standard 39 foot bolted joint rails. Each segment is 39 ft long which can be easily cut or constructed so that many (if not all) cusps fall at the joints where they can be easily controlled. Such piecewise-linear waveforms so constructed are easily surveyed, and can be measured with great accuracy using simple techniques (e.g., a stringline). A piecewise linear waveform is rich in harmonic content, and offers multiple cusps to excite wheel/rail impacts. A disadvantage is that such a shape has no energy content at the frequency corresponding to the segment length or its harmonics. This could make it difficult to relate the responses to those on more conventional track without recourse to a model. Nevertheless, this consideration is outweighed by the benefits of simplicity and controllability, so that piecewise linear perturbations will be treated exclusively in what follows. It should be emphasized there are other waveforms that can be used, such as rectified sine and versed sine formations constructed by bending or forming rail sections to create the cusps. Figure G-1 are examples of some typical waveforms.

G.3.1.1 SINGLE PERTURBATION WAVEFORMS

Figure G-2 shows a typical single perturbation waveform for gauge. Alignment, profile, and crosslevel waveforms can be designed by the same principle. These waveforms may be super-positioned on tangent, spiral, or curved tracks to meet the objectives and constraints of the test. The amplitudes of perturbations are indicated by symbols which can be assigned appropriate values to satisfy specific requirements. For example, Table G-1 lists the typical perturbation amplitudes for a track geometry. The symbols in the figures describing the waveforms are defined below:

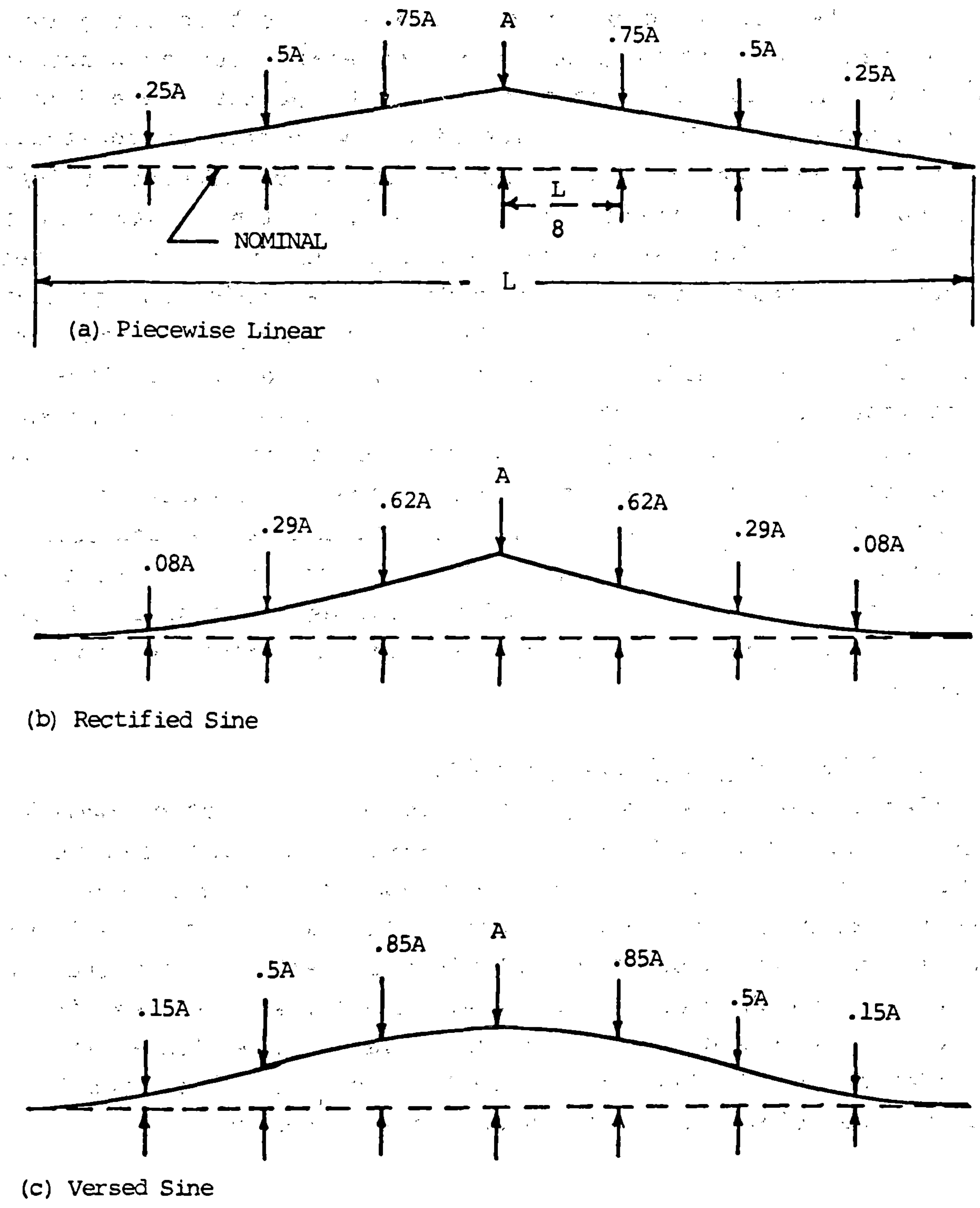


Figure G-1 Typical Waveforms

L = A rail length (typically 39 ft)

d_1, d_2 = Peak lateral perturbation amplitude over nominal track position

($d_1 + d_2$ = peak-to-peak amplitude; d for $d_1 = d_2$)

h_1, h_2 = Peak vertical perturbation over nominal track position

($h_1 + h_2$ = peak to-peak amplitude; h for $h_1 = h_2$).

x = Direction along the track centerline.

y = Direction perpendicular to track centerline and in the plane containing both rails.

z = Direction normal to x-y plane; positive upward.

λ = Fundamental wavelength of perturbation.

TABLE G-1

TYPICAL PERTURBATION AMPLITUDES FOR TRACK

Perturbation	d_1	d_2	2h	λ
Gauge	0.25 in	0.75 in	-	3L
Alignment	1.0 in	1.0 in	-	3L
Profile	-	-	2.2 in	4L
Crosslevel	-	-	2.2 in	4L

Note that the construction of gauge and alignment waveforms as shown in figure G-2 requires two bent rails per cycle. The panel-joint (joints square or opposite, rather than half-staggered) bolted rail construction is required. The waveforms should be repeated over several cycles as recommended in section F.4 to permit the vehicle to reach steady state.

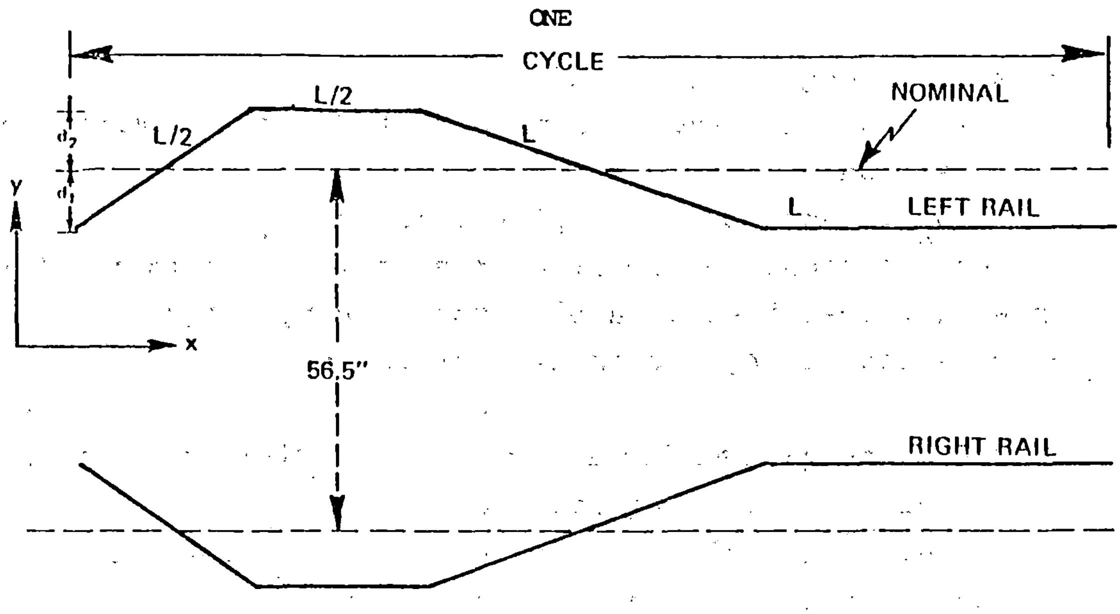
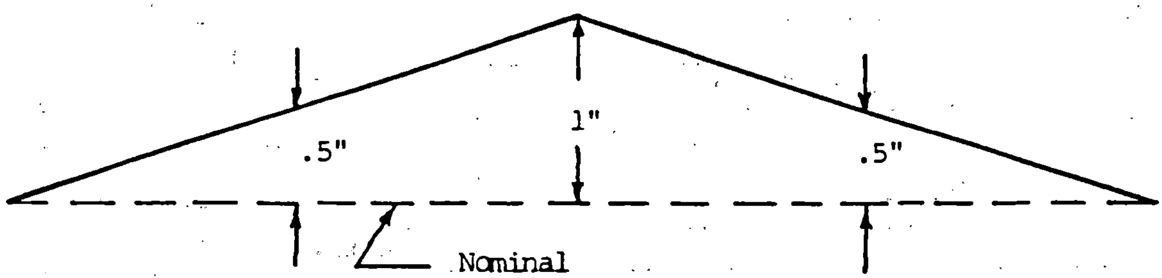


Figure G-2 **Plan View:** Typical Gauge Perturbation Waveform

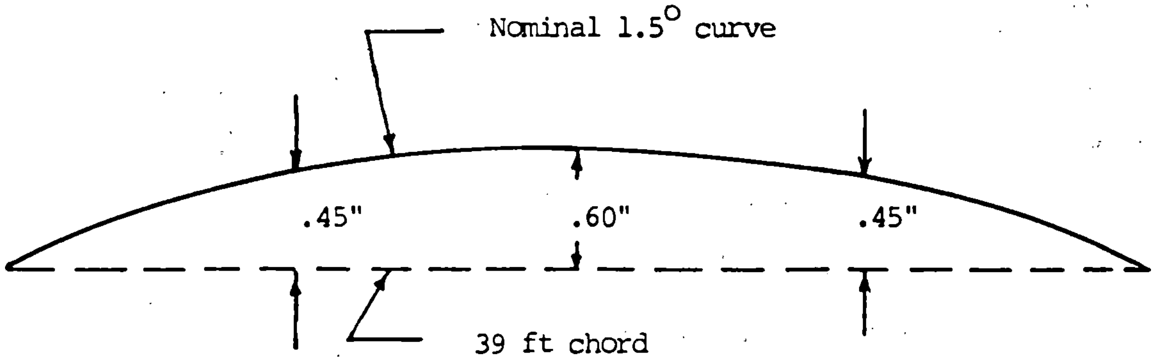
It should be emphasized that the references for the perturbation dimensions are the nominals. For example, figure G-3 a, b, c, illustrate a one inch amplitude triangular transient alignment perturbation super-position on an 1.5° curve. This example uses a 39 foot chord length to obtain the actual track geometry for this case. The super-position, when using the stringline technique may apply the perturbation either to the inside or the outside of the curve by adding or subtracting the nominal chord offset for the specific curve and location on the stringline.

G.3.1.2 COMBINATION PERTURBATION WAVEFORMS

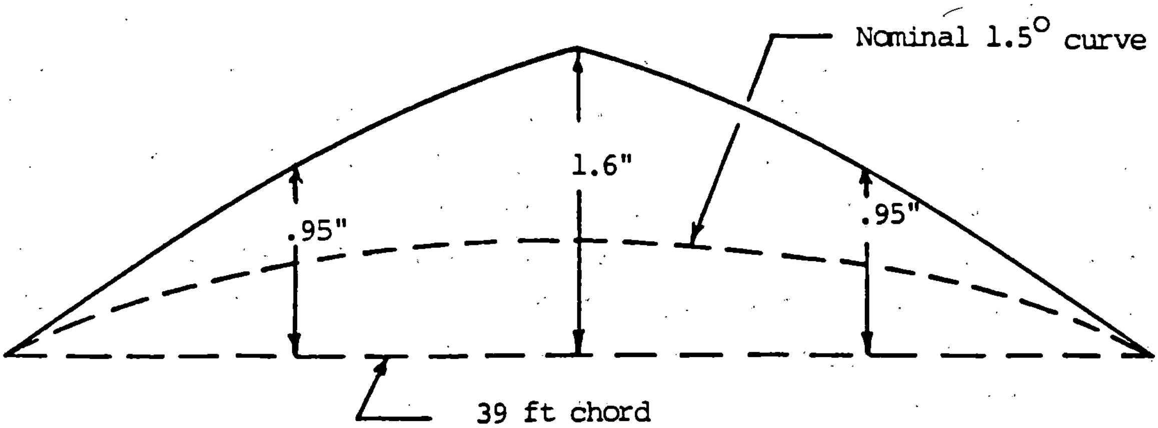
Waveforms for a combination of lateral and vertical perturbations can be created by constructing the track for the lateral perturbation and then shimming it appropriately to superimpose the vertical perturbation waveform. For example, Fig. G-4 shows a crosslevel-plus-alignment waveform for Class 3 geometry. The intermediate ties are to be appropriately shimmed to provide linearly varying vertical perturbations. Note that the combined perturbations have a wavelength of 4L.



(a) Piecewise Linear Perturbation



(b) Nominal 1.5° curve



(c) Piecewise Linear Perturbation Installed on Curve

Figure G-3 Perturbation on Curve

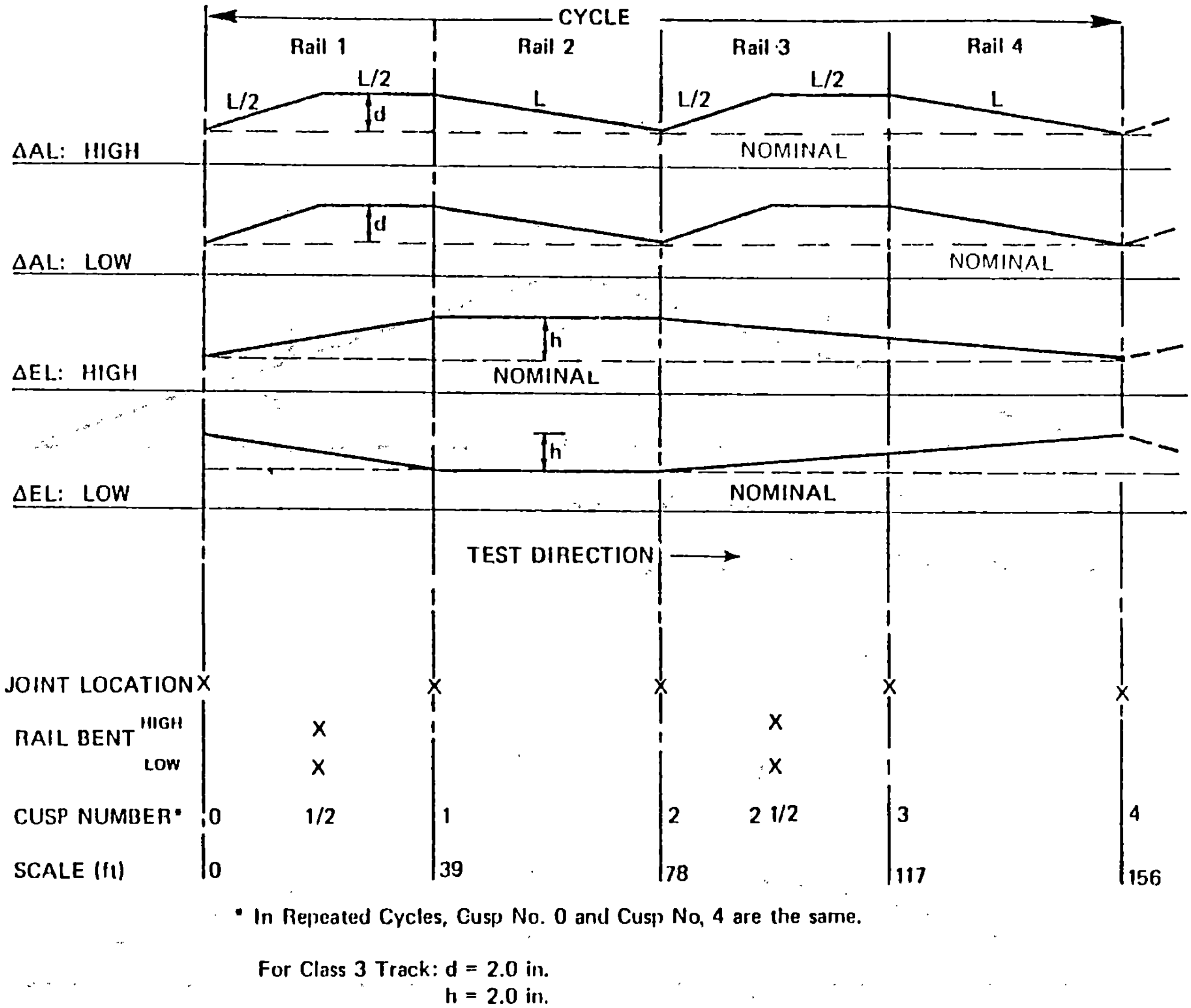


Figure G-4 Typical Alignment Plus Crosslevel Track Perturbation Waveform

G 3.1.3 SPECIAL PURPOSE PERTURBATION

The force due to superelevation imbalance and high rail misalignment on curves and spirals has been identified as a prominent possible cause of vehicle derailment, and therefore a test section can be designed combining controlled high rail misalignment with staged superelevation runoff in the curve body and exit spiral.

Figures G-5 and G-6 shows two ways of achieving high rail misalignment. The track may be of the staggered or non-staggered joint bolted rail type, since the low rail is unperturbed laterally. Each outer rail is less curved than the nominal (possibly straight, as illustrated in Fig. G-5 or even curved outward as in Fig. G-6 and is geometrically tangent to the nominal track curve at its midpoint. The rail ends may be simply pushed outward the tie replugged and the rail respiked to the required position. The objective of these perturbations is to simulate the joint cusp impacts experienced by a vehicle as it traverses a poorly maintained curve at an overbalance speed.

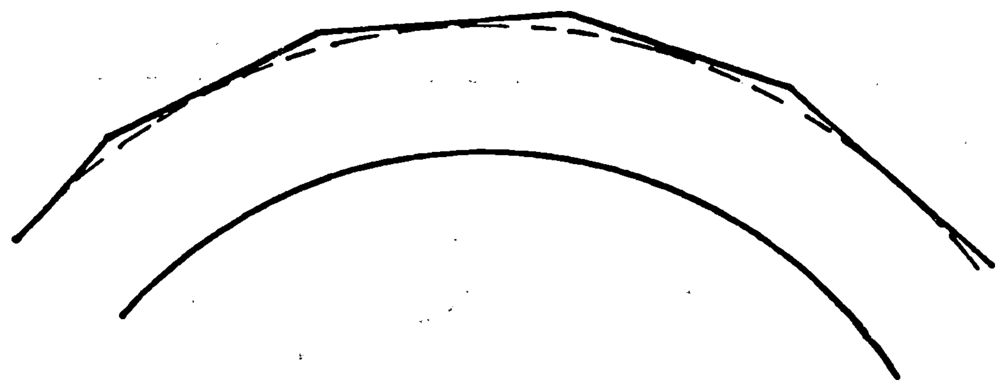


Figure G-5 High Rail Misalignment

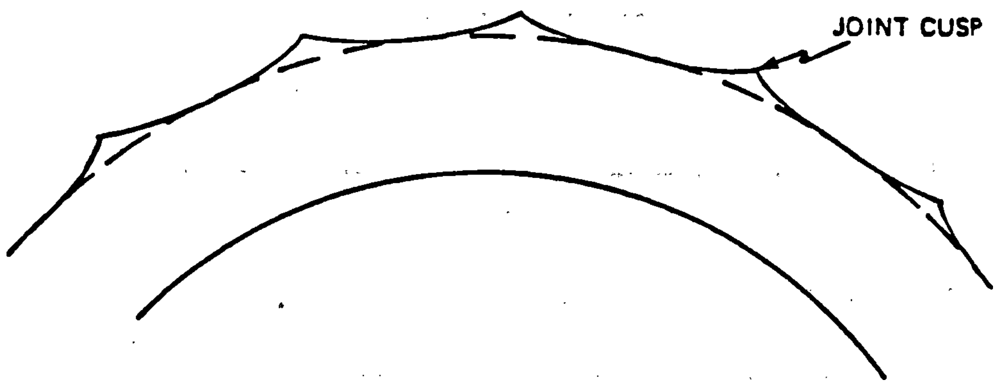


Figure G-6 Alternate High Rail Misalignment

Figure G-7 shows the superelevation runoff profile of the curve and spiral sections. The corresponding curvature and balance speeds are also qualitatively shown for clarity. This test is intended to simulate various degrees of imbalance on a single curve, which will enable one to separate the effects of speed and imbalance.

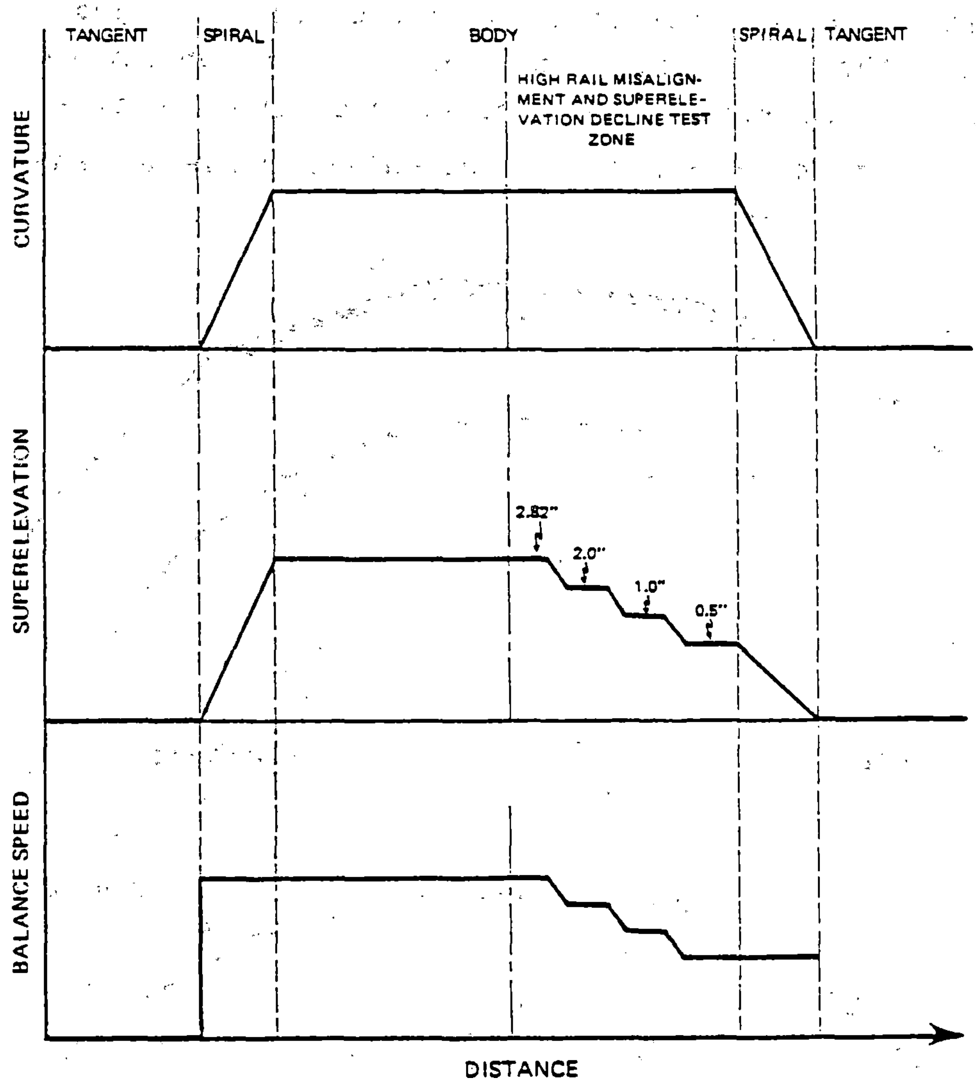


Figure G-7 Superelevation Run Off Profile

G.3.2 TRACK GEOMETRY PERTURBATION FABRICATION

The bending of rails and joint bars to create perturbations had been performed and tested on the Train Dynamics Track of the Transportation Test Center (TTC) Pueblo, Colorado. The results were not conclusive due to the limited amount of traffic over the test segments, however, indications showed that this technique may be adequate for the intended uses. Pre-bent rails appear to be a better approach than pre-bent joint bars because of the following reasons:

- there is more control in rail bending because the bending machine was designed for this purpose.
- Rail can be bent both vertically and laterally while joint bars can be bent only laterally.
- Pre-bent rails can be adjusted easily with the rail bender prior to installation; however, pre-bent joint bars can be adjusted only after installation with heavy track maintenance equipment, thus, there is not much control on pre-bent joint bar adjustments.

G.3.2.1 PROCEDURE FOR BENDING OF RAILS

(a) Rails were cut to the designed length and appropriate joint bar holes were drilled at each end at a rail fabrication plant.

(b) The rails were warmed in hot water prior to bending.

(c) Then the vertical bend was made followed by the lateral bend, including overbend to allow for bending tolerance and bend loss by the bending machine.

(d) Fine adjustment to the lateral bends was made with a rail bender before spiking.

G.3.2.2 PROCEDURE FOR BENDING OF JOINT BARS

(a) Joint bars were bent for lateral cusps only.

(b) Adjustment to perturbations when using bent joint bars was difficult, and was done after installation with track repair tools and equipment.

G.3.2.3 ADJUSTABLE FASTENERS

The adjustable fasteners are designed to facilitate the adjustment of the test rail vertically and laterally without disturbing the ties and ballast. The adjustable fasteners may be used advantageously for a permanent test facility but not for track uses for occasional testing because the adjustable fasteners cost considerably more than the standard fasteners.

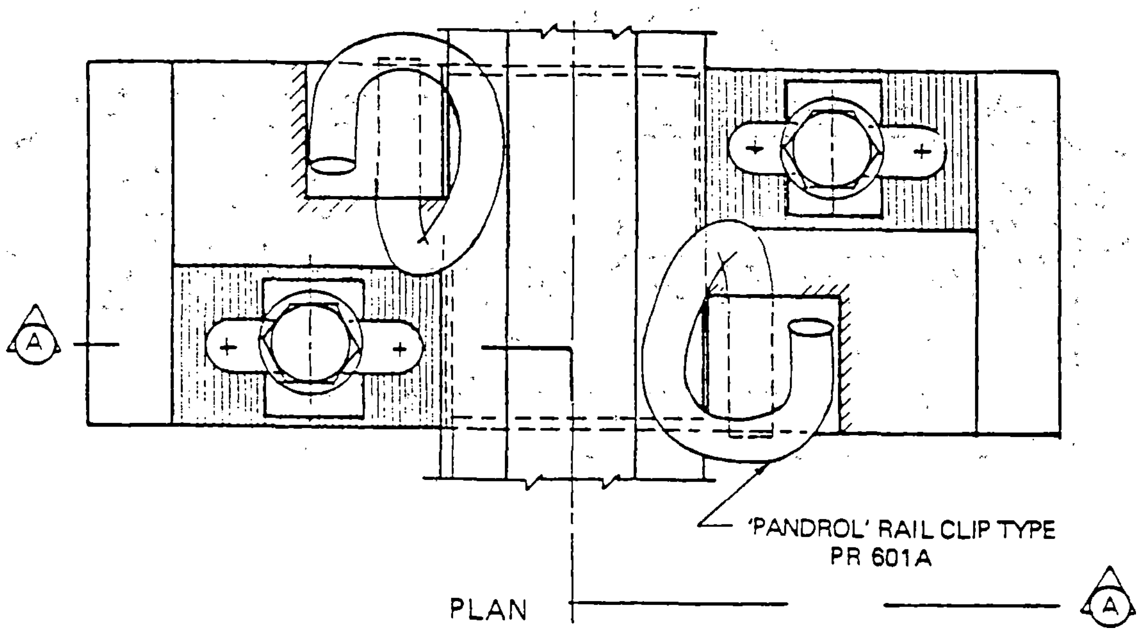
Six types of fasteners were evaluated at TTC by field testing which accumulated .70 million gross ton of traffic. The six types of fasteners evaluated are described in the following paragraphs.

Adapted Alaska Railroad Tieplate. This tieplate (Figure G-8), developed and tested on the Alaska Railroad, can adjust rail laterally and vertically to compensate for frost heave without disturbing the ties (see reference 6). In the original design, it was used with concrete ties and 115-lb rails. The rail-tieplate connection is the Pandrol clip inserted into weld-on shoulders, a Pandrol proprietary item.

The tieplate was first fastened to the concrete tie by proprietary "coil bolts" screwed into inserts in the tie. Correct bolt tension was achieved by using Belleville washers under the bolt head. The tieplate is slotted, and lateral adjustment is provided by moving a serrated block on the plate with matching serrations. This arrangement has the advantage of positive location, but is costly, and the adjustment can only be made in steps of 0.15 inch. Vertical adjustments were made by using plywood shims between the tie and tieplate. The modifications required for adapting the fastener to wooden ties are as follows:

- A 1-in-40 taper is machined on the base of the tieplate to provide the necessary rail cant for the wood tie.
- The rail seat is widened to 6" for the 136-lb/yd rail.
- Two alternative tieplate-tie fasteners were tried. One is a conventional 1" lag screw to U.S. standard dimensions. This has a cost advantage, but there is concern that with repeated adjustment, it may become loose in the wood and allow the serrated block enough freedom to lift clear of the serrations. The other fastener uses a special insert in the wood tie (Figure G-9) with a conventional heavy hex structural bolt to secure it.

The prototype wood tie inserts, made in the TTC workshop, were relatively expensive. However, if a large quantity were required, they could be made with appropriate machinery by a screw manufacturer. The procedure for installation of the wood tie inserts was as follows: Pre-bore the tie to the root diameter of the outside thread, coat the outside thread with a suitable adhesive, and screw in the insert using an adapted socket-wrench drive. The top of the insert should be 1/8" below the surface of the tie. In large-scale production, a socket for



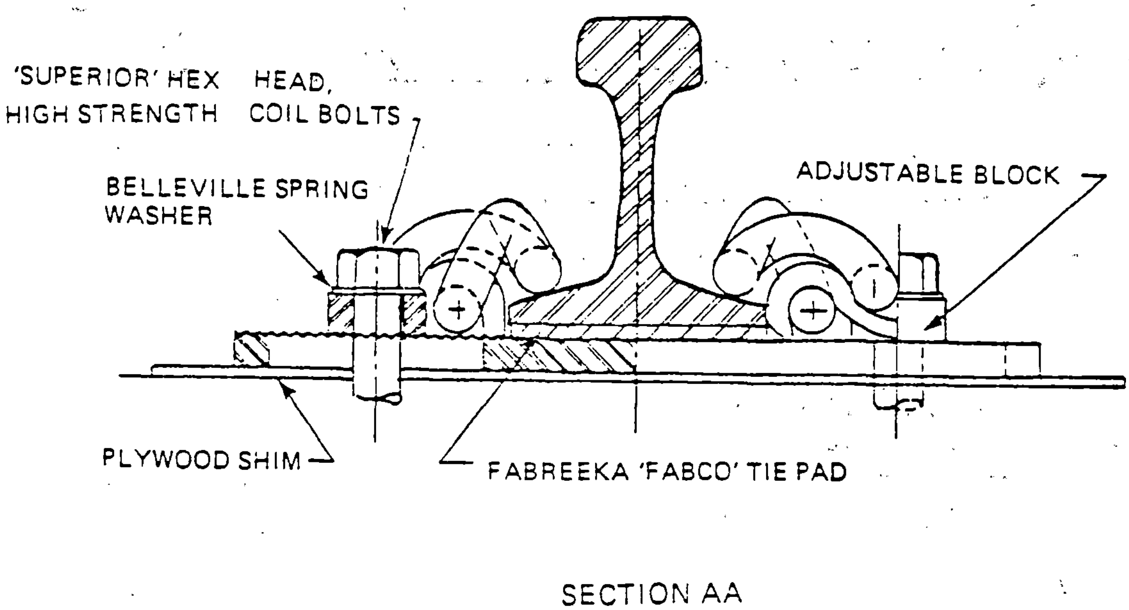
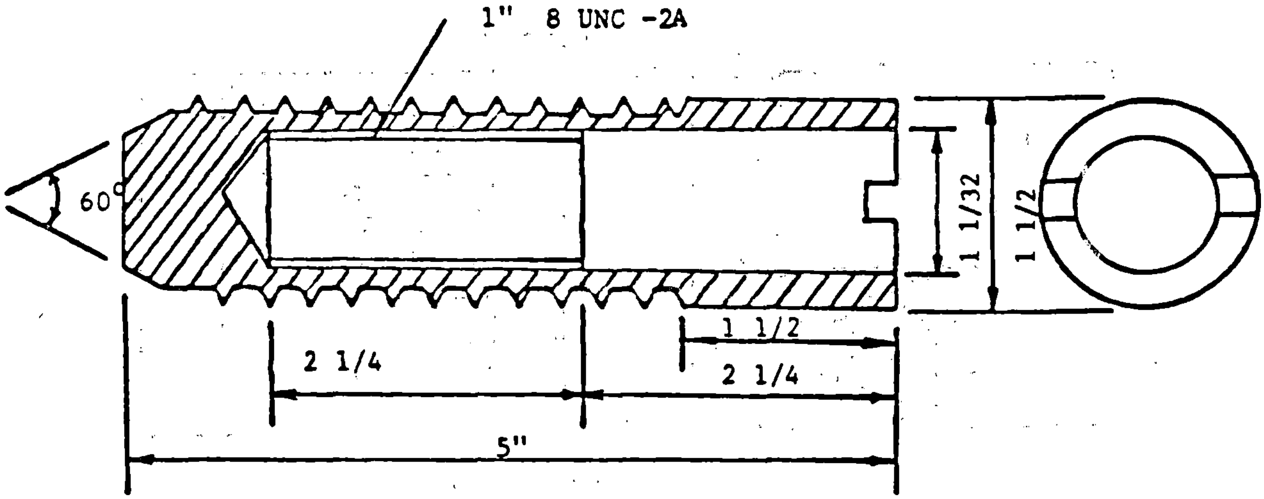
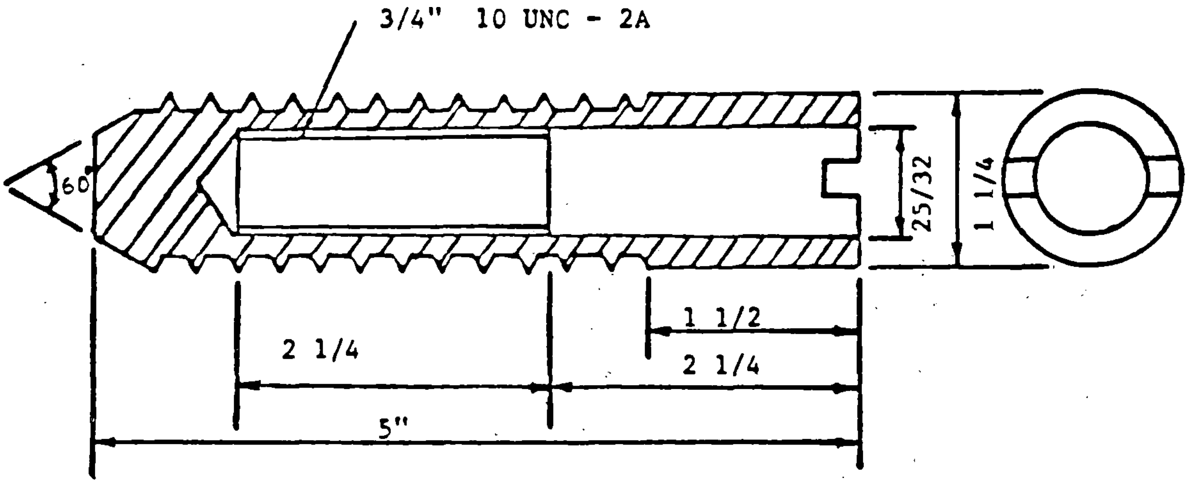


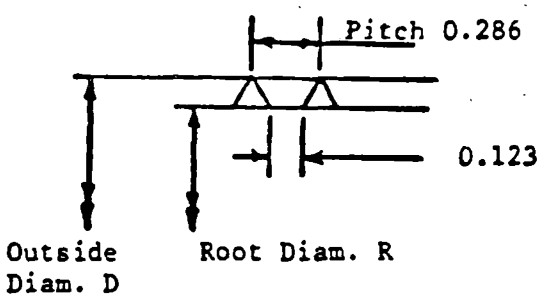
Figure G-8 ALASKA RAILROAD ADJUSTABLE FASTENING SYSTEM FOR CONCRETE TIES

1 Inch Size



3/4 Inch Size





Size	R	D
1	1.28	1.50
3/4	1.03	1.25

Figure G-9 Wood Tie Insert

an Allen wrench in the head of the insert would be a viable alternative.

Potential uses for the insert are not confined to adjustable tieplates. These inserts could be used to attach any tieplate to a tie where there is a particular need for a more secure fastening than is obtainable from the conventional wood tie/cut spike system after it has become somewhat spike-killed. Such a need often occurs on sharp curves, where rail is replaced frequently.

Pandrol Tieplate on Baseplate. The tieplate is the conventional wood tie Pandrol tieplate used at FAST (Figure G-10). The normal square-punched spike holes are replaced by 13/16" diameter round holes with a 1-1/2" diameter spotface to take washers and bolt heads. The pilot test used the conventional tieplates, because unpunched plates were not available.

The baseplate is slotted and recessed to take the 3/4" heavy hex head bolts. Figure G-10 shows that the slots for the bolt heads run across the baseplates. This is not strictly necessary; depending on the machinery available, short slots just long enough to allow the bolt to move the full length of the bolt shank slot could be used. The system relies on the friction from the clamping force between tieplate and baseplate to keep the tieplate in position. It has the advantage of infinite adjustability, and avoids the cost and complication of serrations. Nuts and heavy lock washers complete the assembly, which is fastened to the tie with four 3/4" conventional drive spikes.

Compression Clip on Baseplate. The baseplate of this system is similar to that for the Pandrol tieplate, but the slots are positioned to match the punchings on the conventional 8-hole, A-punch tieplate (Figure G-11). Four 3/4" bolts fasten the tieplate to the baseplate, two of them directly and two through the conventional spring clip. The system avoids the use of Pandrol components, but is otherwise similar to the Pandrol scheme. The bolt which holds the spring clip may remain properly torqued longer because of the controlled pretension provided by the clips. As before, heavy hex nuts and heavy lock washers complete the assembly, which is fastened to the tie with four 3/4" drive spikes.

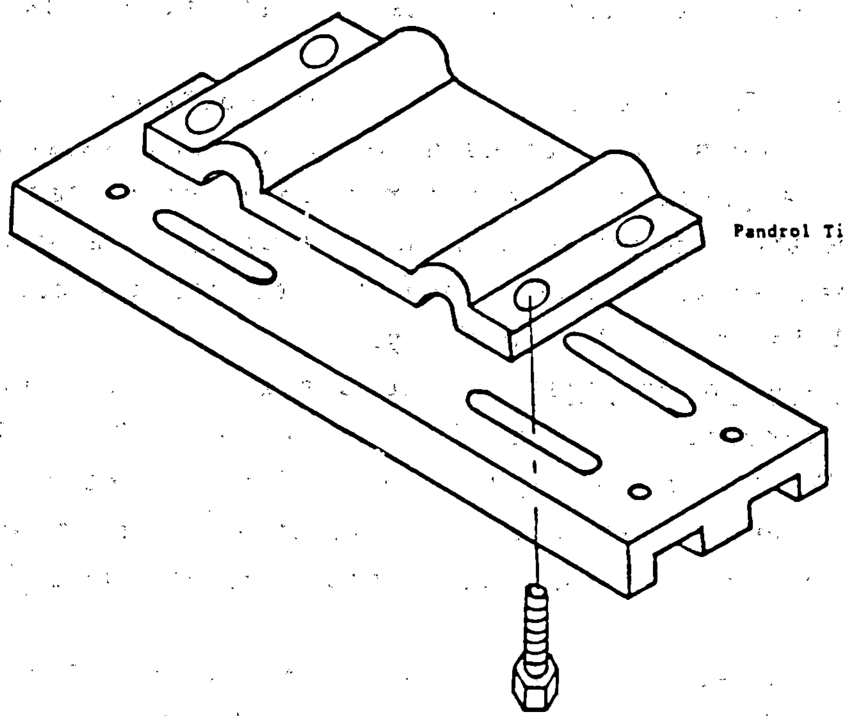


Figure G-10 Pandrol Tieplate on Baseplate

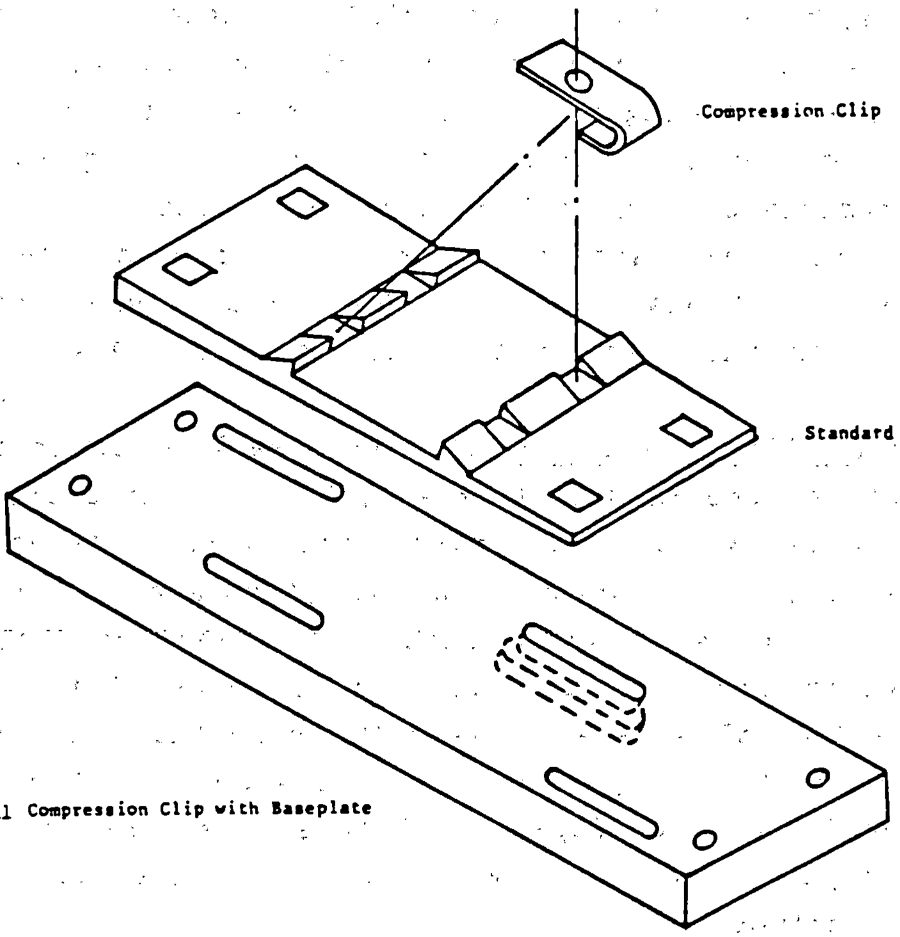


Figure G—11 Compression Clip with Baseplate

Adapted Hixon Tieplate. The Hixon tieplate used in these adaptations is the one used on FAST. It had performed satisfactorily for 30 million gross tons, so there appear to be no major flaws with the basic tieplate-rail fastening design.

The first adaptation is illustrated in Figure G-12. The conventional bolt holes are replaced by a slot, and a baseplate with tapped holes is required. The tieplate is simply bolted down onto the baseplate with a screw and heavy lock washer, and the whole assembly is bolted to the tie with four 3/4" drive spikes. Because the length of the Hixon tieplate is limited, only 2 1/2" of adjustment are possible; therefore, two tapped holes are provided in the baseplate to increase the range of adjustment.

The second adaptation uses the same tieplate, but the screwed inserts (Figure G-9) are used in place of the baseplate. The inserts are put directly into the tie in the same places as the tapped holes in the baseplate.

The third adaptation is illustrated in Figure G-13. It consists of a lengthened version of the FAST Hixon tieplate, and uses the same rail-tieplate fastening. A 3 1/2" x 1" slot is provided in place of the holes for the normal hold-down bolts. Serrations are provided as on the Alaska Railroad tieplate, but these can be omitted at some cost reduction if they are found to be unnecessary. The wood tie inserts are used for the holddown bolts. Although not immediately available, this design has a number of advantages:

- o It is a cast item, and thus suited to medium production runs (100-1,000).
- o Variations in the design can be provided for the modest cost of new patterns to suit different applications.
- o If serrations are required, they can be cast in, saving costly machining.
- o If the wood tie inserts prove successful, they can be installed on ties that are in place.
- o This design can be used with concrete ties in the same way as the original Alaska Railroad tieplate.

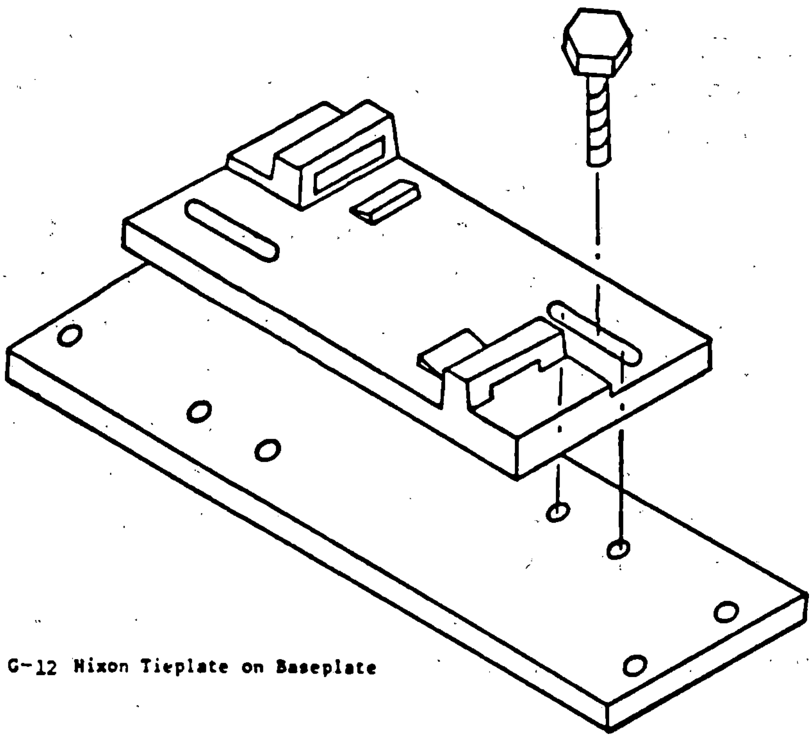


Figure G-12 Hixon Tieplate on Baseplate

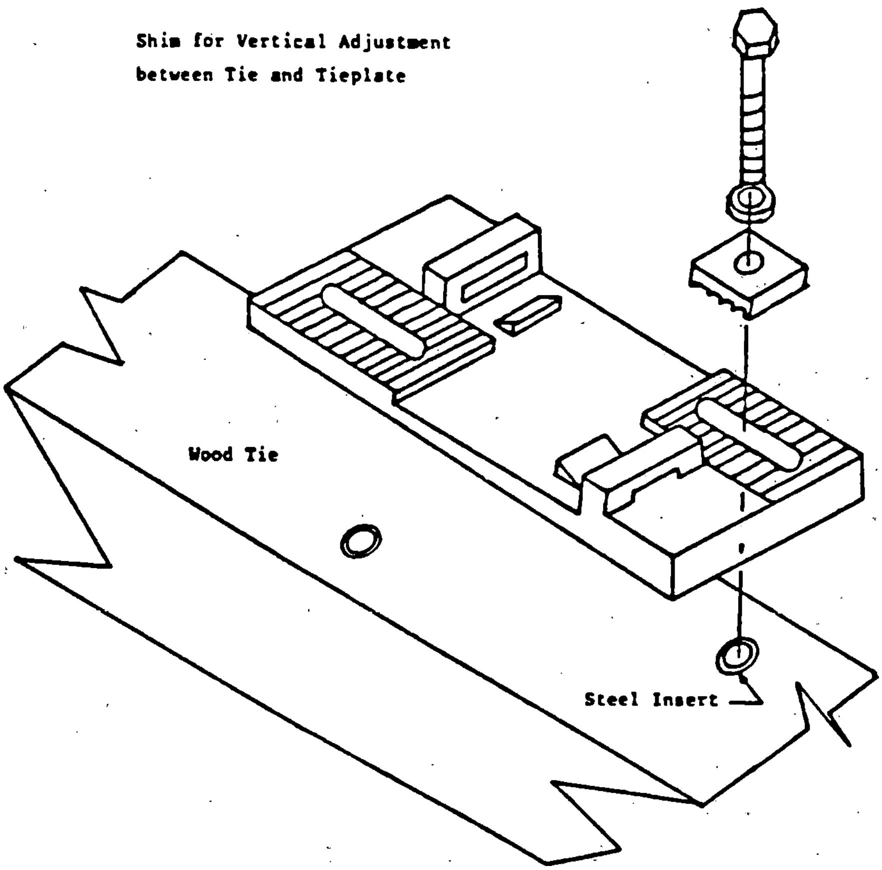


Figure G-13 Adaptation of the Hixon Tieplate

G.3.3 TRACK GEOMETRY PERTURBATION INSTALLATION

G.3.3.1 NEW TRACK

New track must be stabilized before the track perturbations can be installed. To stabilize the track, the standard rails should be installed first, then followed with ten million gross tons of traffic at speeds of 10-25 mph.

Remove the standard rails and replace with the pre-bent perturbed rails. In the process of installing the perturbations, gauge and crosslevel for any point along the track can be measured directly with a standard gauge and crosslevel bar; profile and alignment can be measured with a stringline technique.

The rail segment between the two end points is then adjusted vertically by shimming or laterally by spiking to the prescribed position as required by the distances from the stringline chord. Vertical and lateral distances from the stringline chord are to be measured every 1/8 of a rail length. After the entire cycle of perturbation is installed, the stringline chord is moved to the next cycle of perturbation and the process is repeated.

G.3.3.2 EXISTING TRACK

If an existing track is used for the conversion to a test track, the track is considered already stabilized. It should be noted that the ties should not be displaced or raised by tamping since this will disturb the stabilization of the track for testing. In this case, the existing rail segments can be removed and replaced with the pre-bent perturbations following the same installation procedure as that described in section G.3.3.1. After the completion of tests, the track can be restored to the original condition by removal of perturbations and replacing them with the original rails.

G.3.4 TRACK GEOMETRY MEASUREMENTS

Several methods of measuring the geometrical properties of the test track are necessary for various reasons:

- o To provide a reference during the installation of the track perturbations.

- To define the geometric input under load for correlation with vehicle response.
- To measure the elastic and permanent movements of the track at locations of peak/dynamic loads.

The measurement methods and their primary application are given below:

- Hand measurements (with gauge-crosslevel bar and string line) can be used during initial perturbation construction, subsequent readjustments, and periodic spot checks.
- Track Survey Device (TSD), a laser based measuring device , can be used at regular intervals throughout the test to provide a fairly continuous monitoring of the test track.
- A light weight commercially available track geometry measuring car such as the Plasser EM80 used at the Transportation Test Center - can be used before each day of testing to survey the appropriate test zone and thereby provide an independent source for data verification (see reference 7).
- A heavy track geometry measuring car such as the FRA developed T-6 and T-10 - are capable of providing fast loaded measurements of the track for defining the track excitation input into the vehicle and the actual track geometry under the load of the vehicle (see reference 8 and 9).
- Dynamic Displacement Measurements - can be installed at selected locations to measure gauge changes or lateral track movements, to provide run-to-run monitoring for safety, and for correlation with vehicle forces in the assessment of track strength.

The characteristics of the first four methods and the procedures used are given in more detail below.

There are advantages and disadvantages in each of the measurement methods. There are also practical limitations in the applications of these methods. Discussions are presented here on each of the methods, followed by a description of the actual procedure used in carrying out each of the track geometry measuring methods.

G 3.4.1 <u>Hand Measurements</u>

Perturbations used in the IAT consist of distinct wave shapes in one or a combination of track geometry parameters. These wave shapes are defined by prescribing the magnitude of the deviation of the rails from a perfect track.

Gauge and crosslevel perturbations are described as deviations from nominal gauge or crosslevel. Profile and alignment perturbations on tangent track are given as deviations from a straight line. In the case of curved track, alignment perturbations are given as deviations from a perfect circular curve.

In the process of installing the prescribed perturbations in existing trackage, gauge and crosslevel for any point along the track can be measured directly with a standard gauge and crosslevel bar. Profile and alignment, on the other hand, require the use of precision survey instruments if exact spatial positions of the rails are to be pinpointed with respect to an absolute reference. The use of survey instruments are not at all practical considering the number of rails which are perturbed. A manual stringline technique is therefore recommended to provide a relative rather than an absolute reference in the spatial coordinate system.

A stringline technique involves the use of either one or two rail lengths of line, depending on the basic wavelength of the perturbation, stretched between the intended end points of a basic cycle of perturbation, so that the offsets from the string may be measured.

The rail segment between the two end points is then moved vertically (by shimming) or laterally (by respiking) to the prescribed positions as required by distances from the stringline chord. Vertical and lateral distances from the stringline chord are given for every 1/8 of a rail length. After the entire cycle of perturbation is installed, the stringline chord is moved to the next cycle of perturbation and the process is repeated.

The use of a relative measurement technique, such as the stringline process described above, is considerably simpler than any survey procedure. No special equipment other than hand tools are needed to perform these measurements. A crew of three can perform these measurements on short notice and the data is immediately obtained. Therefore, a broad reliance can be placed on the use of hand measurements during the test.

However, there are many disadvantages in the hand measurement methods. The accuracy and repeatability is not as good as automated methods. The procedure is cumbersome so that measurement stations are usually spaced relatively far apart (1/2 rail length to 1/8 rail length). The measurements are made with no vertical or lateral loads so that slack in the track would make the measurements differ from the actual inputs to the vehicle.

The most serious disadvantage of using the cycle-by-cycle stringlining method for installing profile and alignment perturbations is in the relative nature of a chord measurement technique. Since the end points of each chord are the only reference points in the measurement, the final wave shape of the perturbed track would have all the end points of the chords remain in the unperturbed position. If the original track were a perfectly lined tangent or circular curve, then the perturbed track would conform to the design. However, if the end points of the chords happened to be out of alignment, the resulting perturbed track would retain those errors. The individual cycles within each chord length would conform to the prescribed wave shape, the transitions between adjacent cycles may contain significant errors. This effect has been actually observed in the alignment perturbations. Adjustment to the track will be necessary to remove the dissimilarity between successive cycles of perturbation.

G.3.4.2 Track Survey Device

The Track Survey Device (TSD) is a laser based precision track geometry measurement vehicle. The TSD consists of two separate portions. A laser source is mounted on a small rail car which can be pushed on the track. When a track segment is to be surveyed, the laser source is placed at one end of the track segment with the light beam pointing along the track. The second portion is the survey vehicle itself, which is driven by a gasoline engine. The survey vehicle is driven on the track towards the laser source during the survey. A target screen on the survey vehicle continuously intercepts the stationary laser beam as the vehicle moves forward. The portion of the intercept point on the screen provides the absolute reference for the track measuring mechanisms installed on the vehicle. Two contact wheels on the vehicle are hydraulically loaded against the gauge measurement points of the left and right rails. A gravitational pendulum is installed in the vehicle to measure the crosslevel of the track. The positions of the contact wheels and the crosslevel angle are used to determine the positions of the two rails relative to the stationary laser beam. The rail position information is measured at six-inch intervals and recorded on magnetic tape.

The TSD uses a stationary light beam external to the vehicle as a reference to determine track geometry. The principle employed is similar to that used by standard optical survey methods, which is better than the relative reference or inertial reference principles employed by other vehicle-borne track geometry systems.

The system is self-contained in the sense that it requires no supporting equipment such as a locomotive. It can be operated by a crew of three, and requires minimal setup time. The survey speed is relatively slow compared with a typical automated track geometry vehicle. However, it is considerably faster than manual stringlining. A typical six-hour operating period (usually from dusk to midnight) can cover as much as 1.5 miles of track.

The sampling frequency of the TSD can be adjusted to provide a fine resolution along the distance of the track. (Six-inch sample rate is recommended). A software package exists to process the data tape and provide the results in the form of pseudospace curves or midchord offsets of several popular chord lengths.

Even though the TSD measurements are based on an absolute light beam reference, there are significant limitations in measuring track geometry perturbation of long wavelength. On tangent track, the laser beam can provide a thin reference line only up to 150 feet long, beyond which the increase in beam aperture and loss in intensity would reduce the accuracy of the geometry measurements. A track section surveyed with respect to a common laser beam is called a survey sequence. The laser light source has to be moved forward to the next track section to establish a new reference light beam for the next survey sequence. Maximum sequence length for surveying curved track is further restricted because of the size of the target screen. On a 1.5 degree curve, the sequence length is limited to 80 feet.

Since each survey sequence is referenced to a different light beam, the track geometry data from one sequence cannot be tied to the adjacent sequences. An overlay series of sequences are made to overcome this limitation. Each of the overlay sequences covers from mid-point to mid-point of two adjacent sequences from the original survey series. A software package then performs the "splicing" of the data using survey data from the original and the overlay sequences. The accuracy of the long wavelength information in the spliced data is relatively poor. Since the same light beam is used within the length of each sequence, the accuracy does not begin to degrade until the wavelength is longer than the sequence length, which is 80 feet for the surveys conducted on the referenced perturbed track test.

The axle weight of the TSD is approximately 7000 pounds, which is sufficient to take up all or at least a large portion of the vertical slack in the track. There is, nevertheless, essentially no lateral load applied to the track. The speed of vehicle motion during a survey is usually less than 5 mph, which is considerably below the balance speed of the 1.5 degree test curve. The lateral track load due to gravity is on the order of a few hundred pounds applied to the low rail; there is essentially no lateral load applied to the high rail.

There are some other limitations of the TSD which should be mentioned. The TSD data output is not available during the survey, which limits the capability of the crew to verify that the system is collecting data properly. The unenclosed design and the use of hydraulic controls hampers the operation of the TSD in cold weather.

G.3.4.3 Light Weight Plasser EM80

The Plasser EM80 (other makes are also available) is a light-weight, self-propelled, commercially available track geometry measuring vehicle. The rated top measuring speed of the vehicle is 80 km/hr. (The same model is sometimes identified as a EM50, signifying a 50 mph top speed.)

The EM80 has two load-bearing axles, each carrying approximately

15 tons of vertical load. One of the two axles is used to measure crosslevel. Gauge, profile and alignment are measured by six measuring axles which do not carry significant vertical or lateral loads. The measuring axles have flanged wheels that are 22 inches in diameter. The measuring axles are placed as three pairs in the center, the front, and the aft ends of the vehicle. The lead axle in each pair is pneumatically loaded laterally against the trailing axle located two feet behind. The lateral loads force the lead axles in the three pairs to flange against the left rail forming the three contact points for a 10-meter mid-chord offset (MCO) alignment measurement. The trailing axles are forced in the opposite direction forming a 10-meter MCO alignment measurement for the right rail.

The vertical movements of the measuring wheels are measured from the carbody to provide the 10-meter MCO measurements for the left and the right rails.

The 10-meter MCO's for profile and alignment are converted to 62-foot MCO's by algorithms in the onboard computer. Low-pass filtering is used on the 62-foot MCO outputs to smooth out undesirable noise in the conversion process. It should be noted that the profile and alignment measurements use the carbody as the reference beam to provide the 10-meter chord.

One of the six measuring axles is made to have the wheel-flange gauge wider than the standard track gauge. This axle is hinged at the middle so that the two axle-wheel halves can be cambered at an angle for the wheel flanges to fit within track gauge. The axle halves are pneumatically loaded to maintain simultaneous flanging of both the left and right wheels at all times. Track gauge is measured by the distance between the wheel flanges, which is calculated from the measured camber angle between the axle halves.

Crosslevel is measured on one of the two running axles. A gyro-stabilized pendulum is installed in the vehicle for measuring roll angle of the carbody. Displacement transducers are used to measure carbody-to-axle roll angle. A fixed-base twist is measured by the roll angle between a running axle and the adjacent outboard measuring axle (see reference 7).

The measuring systems have several limitations. The gyro-stabilized pendulum is not fully compensated for curvature and drift effects. Therefore, the vertical reference is subject to error on curves. Testing at relatively low test speeds can reduce the magnitude of the error. The alignment and profile measurement wheels are essentially not loaded; slack in the track may introduce errors. The low-pass filters used for smoothing the converted 62-foot MCO profile and alignment data are time-based filters. These filters will have different spatial corner frequency at different test speeds.

The data output is in the form of distance-based paper pen charts. Permanent magnetic tape recording capability is featured in the EM80 design.

G 3.4.4 Heavy Track Geometry Vehicle T-6 and T-10 System Description

The T-6 and T-10 are heavy track measurement vehicles which have been developed and used by the FRA for track inspection. The T-6 and T-10 have similar measuring systems. They contain the most recent version of the inertial-based track geometry measuring instruments, including the inertial alignment system.

The systems are capable of measuring gauge, crosslevel, profile, alignment, warp, and curvature at track speeds up to 80 mph on the T-6 with on-line processing and up to 120 mph on the T-10 with off-line processing. A one foot sample rate has been used as the primary mode of measurement, and a few runs have been made at a 6-inch sample rate for the purpose of data verification.

The on-line chart display is limited to a 62-foot mid-chord offset format for profile and alignment. An off-line software package is capable of converting the data to pseudospace curves or chord data of other chord lengths (see references 8 and 9).

The T-6 has a total weight of approximately 80 tons, or 20 tons per axle. The typical test speeds are up to 80 mph. The vertical and lateral loads, though not as high as locomotives, are representative of a typical heavy vehicle.

The data collected on tape is continuous over the entire test section (as opposed to 80-foot segments for the TSD), and can be used easily for downstream data analysis and research.

Both the T-6 and T-10 have profile and alignment measurement which degrades below 15 mph, and not reliable below 5 mph; the curvature measurements are not accurate below 2 mph; however, the gauge and crosslevel measurements are good even at slow speed.

G.3.5 Perturbed Track Length

G.3.5.1 Length for Acceleration and Stopping

The minimum length of test track is constrained partly by the need for adequate test duration at each condition, and partly by acceleration and deceleration allowances. For a power-limited locomotive, the distance d required to accelerate from standing to speed V is given approximately by

$$d = \frac{(m_l + m_t)}{3\,P} V^3$$

where P is the power available, m_l is the locomotive mass, and m_t is the trailing mass. The distance to stop the same consist with locomotive brakes only is approximately

$$d = \frac{(m_l + m_t)}{2ug\,m_l} V^2$$

where u is the net permissible braking adhesion, and g is gravitational acceleration. Wind resistance and other losses haves been neglected. Figure G-14 shows the more exact relationships (including losses calculated from the Davis Equation, Ref. 5 , and adhesion limit) calculated numerically for two representative locomotives and a 200 ton trailing load; u is taken to be 0.1. It will be seen that the distance required to accelerate to 80 mph and then stop the consist is substantially more than two miles. This distance would have to be added to the total test section length to obtain the minimum length of a dedicated, 80 mph test track. For this reason, it is strongly recommended that a permanent test site, in its final form, be configured as a closed loop which will provide unlimited starting and stopping length, subject only to limitations on curvature and superelevation.

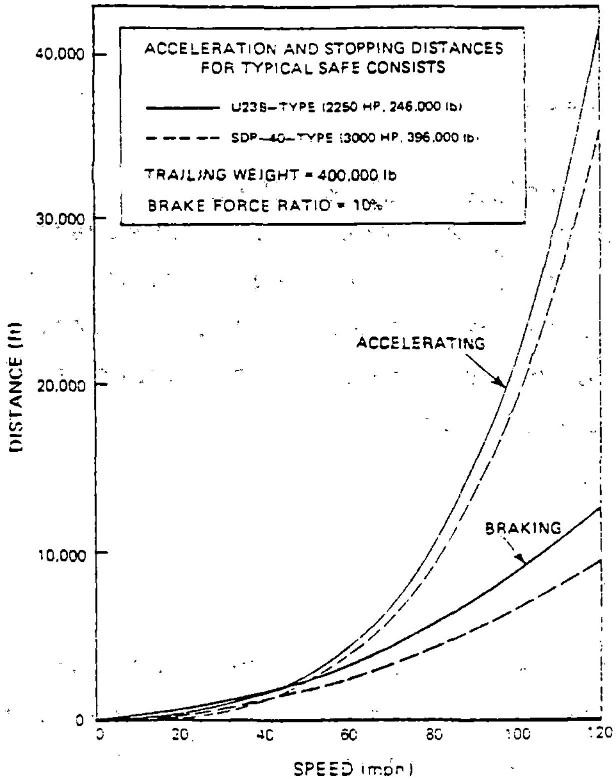


Figure G-14 Acceleration and Stopping Distances for Typical Test Consists

G.3.5.2 Length for Transition Between Test Zones

A transition unperturbed track length is required to separate the perturbed test zones, so that the vehicle dynamic responses that are generated by the preceding test zone can be damped out before the vehicle enters the next test zone. This transition track length is about equal to 3 to 4 wavelengths of the perturbations in the preceding perturbed test zone, based on the experience of the Perturbed Track Test, Pilot Test and Freight Test conducted at TTC in 1978 and 1979.

G.3.6 Curved Track and Balance Speed

In order to gain the best understanding of the effects of curvature, two or three different curvatures are desirable. Another key factor in curve negotiation is imbalance due to over - or under-speed condition. To separate the effects of curvature and imbalance, the following steps are recommended:

- The principal part of each curve within the test track should be elevated by an amount such that the balance speed is equal for all curves.
- The elevation should be sufficient so that the minimum of all the speed limits is reasonably high.
- In at least one curve, a section of track with reduced superelevation should be used, along with the part having the nominal elevation.

The first item allows the entire track to be traversed at a constant speed and unbalance situation (excepting the reduced superelevation section). The second permits a significant range of speeds to be tested without accelerating or braking (which might confuse the results). And the third provides information on the effects of imbalance as distinct from those due to curvature. The technique of declining superelevation was first proposed for the 1978 Perturbed Track Tests (Ref. 2) and used successfully there. Table G 2 contains a brief tabulation of curvature-superelevation-speed relationships.

TABLE G-2
BALANCE AND LIMIT SPEEDS

SUPERELEVATION (INCHES)	CURVATURE (DEGREES)					
	1	2	3	4	5	6
1	39(78)	28(55)	23(45)	20(39)	17(35)	16(32)
2	55(87)	39(62)	32(50)	28(44)	25(39)	22(36)
3	68(95)	48(68)	39(55)	34(48)	30(43)	28(39)
4	78(103)	55(73)	45(60)	39(52)	35(46)	32(42)
5	87(110)	62(78)	50(64)	44(55)	39(49)	36(45)
6	95(117)	68(83)	55(68)	48(58)	43(52)	39(48)

Speeds in miles per hour

3-inch overbalance speed in parentheses.

Note: Speed limits deviate slightly from limits in FRA Track Safety Standards as a result of approximations in the latter.

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SECTION H

RAIL/TRACK STIFFNESS MEASUREMENTS, VARIATIONS, AND SIMULATIONS

H.1 INTRODUCTION

The performance of a rail vehicle in revenue service is dependent not only on its own characteristics but also on the properties of the track on which it operates. Thus, parameters which affect track characteristics are important in evaluating the vehicle performance. One such set of rail and track parameters includes the vertical and lateral stiffnesses which are described in this section.

The vertical and lateral rail/track stiffness depend on the design of the track, the track construction/maintenance techniques, the environmental condition, and the degradation of the components. Some of the parameters in these categories are: the size of rail; the type of fasteners; the ties and their spacing; the type of ballast/subsoil; the traffic period since realignment and tamping; the moisture content of soil; the ambient temperature; and the loads applied.

Since the tests prescribed by the Vehicle/Track Interaction Assessment Techniques (IAT) may be performed on any of a variety of railroad tracks, it may be impossible to standardize all of the above parameters. However, several recent studies have shown that a repeatable vehicle response will be obtained as long as the overall track vertical stiffness and the rail lateral stiffness (in gauge spreading) are within certain limits.

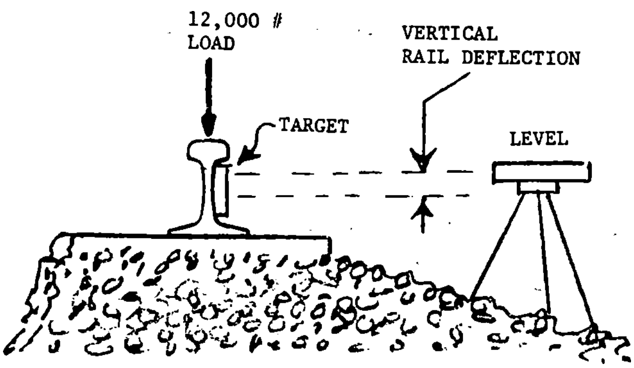
Outlined in this section are suggestions for measuring these two stiffnesses and discussions on the various factors which affect measuring and maintaining the stiffness while performing tests recommended by the IAT. Also outlined are methods to simulate both lateral and vertical track modulus or stiffness in a rail vehicle model to provide realistic response predictions.

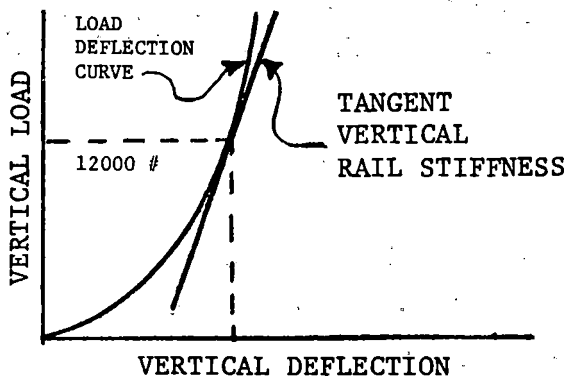
H.1.1 Stiffness Limits for Vehicle/Track Interaction Assessment Technique Testing

The objective of the Vehicle/Track Interaction Assessment Techniques is to provide guidance in determination of a vehicle's dynamic response and characteristics when exposed to varying track geometry conditions. This is done by installing known perturbations in the track at specific wave lengths and measuring the vehicle's response when traveling over them at varying speed ranges. The varying speed provides the frequency inputs to the vehicle due to the various patterns of track perturbations used for specific performance issues under investigation. With variations of the input frequencies a vehicle's response changes as does its sensitivity to the track's response or compliance. In order to compare results for varying track input conditions, certain limits must be placed upon the track vertical and lateral stiffness. The track is an extension of the vehicle's suspension system and therefore the analysis of a vehicle's performance should incorporate the track stiffness. The track input condition for (IAT) testing translates into the vehicle inputs which are caused by the vehicle's motion over the various track geometries and strength conditions.

Studies have been undertaken to assess the variations associated with a vehicle's response due to changes in a track's vertical and lateral stiffness. These studies show that in order to have certain repeatable responses (measurement results of $\pm 10\%$ or better) that the track stiffness in the vertical and lateral orientation must be above certain limits. Table H-1 recommends the limits on track vertical stiffness and rail lateral stiffness for different Test Categories and Performance Issues. As shown in the table, the Service Environment Test should be performed not only on a nominally stiff track, but also on a highly compliant track. This is so that the vehicle performance can be evaluated in a more comprehensive manner than that for a Proof or Diagnostics Test.

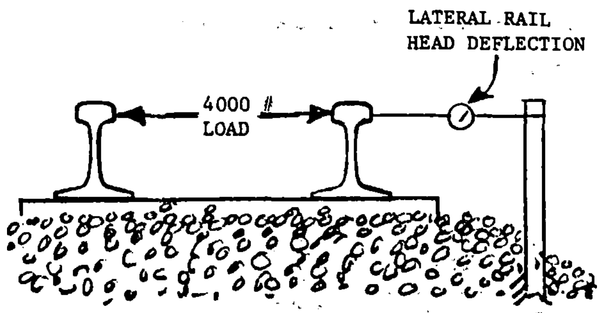
Since track stiffness or modulus is nonlinear and is very sensitive to the initial conditions under which the values are taken the following procedures for making these measurements relate to the values in Table H-1. The static vertical stiffness is a stiffening spring rate with increasing vertical load with much of the deflection taking place over the first few thousand pounds. Thus to eliminate the influence of slack under the light loads a value of 12,000 pounds is used as the standardized value, and the stiffness value from the table is the tangent value of the load deflection curve at the 12,000 pound value. This is a single point load on one rail as shown below.

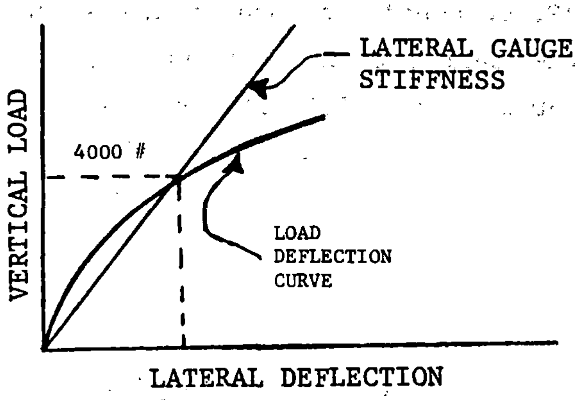




If a four-axle vehicle is used, the correction for the influence of the adjacent wheel must be taken into account and the deflection measurement should be taken under one wheel. The measurements should not be taken over a joint bar. If maintenance was performed in the area, an average of about 10 readings should be used, dispersed around the site and using both rails, to guarantee that the site is stable and the measurements are within tolerances.

The static lateral rail stiffness is also a highly nonlinear spring effect relative to both the lateral and vertical loads. The lateral stiffness values presented in Table H-1 are measured without a vertical load. The load is applied between the rails at the gauge measurement locations on a per rail basis as shown below.





These values give an indication of the strength of the fastener/rail system. The values are only valid if all of the lateral play is removed and the rails are against the tie plates and spikes. The values do not hold if the measurements are taken over joint bars. When inspecting the track for test sites the ballast condition should be observed and if any clearance is found between the ballast and the end of the ties, or if the depth of ballast in the cribs is not sufficient, the site is not acceptable for Vehicle/Track Interaction Assessment Techniques testing.

TABLE H-1

STATIC TRACK STIFFNESS REQUIREMENTS
-VS-
PERFORMANCE ISSUES & TEST CATEGORIES

PERFORMANCE ISSUES	* Vertical Track Stiffness in kips per inch		** Lateral Gauge Stiffness Per RailHead in kips per inch	
	Proof & Diagnostics Tests	Service Environment Tests	Proof & Diagnostics Tests	Service Environment Tests
TWIST & ROLL	> 225	90-150 & > 225	-----	-----
PITCH & BOUNCE	>225	90-150 & >225	-----	-----
YAW & SWAY	-----	-----	> 40	15-25 & >40
SPRIAL NEOGTIATIONS	> 225	90-150 & > 225	> 40	15-25 & > 40
DYNAMIC Curving	> 225	90-150 & > 225	> 40	15-25 & > 40

* Tangent Stiffness at a 12,000 # Vertical Load per rail...See Subsection H.3

** Secant Stiffness with zero Vertical Load and zero to 4,000 # Lateral Load....See Subsection H.3

Summary of Tables 3.2 & 3.8 in Part I of the documentation.

H.2 NOMENCLATURE

E	Modulus of elasticity of rail steel
L_W	Net lateral load of wheelset
L_G	Gauge-spreading lateral load
L_L	Lateral wheel/rail load, left rail
L_R	Lateral wheel/rail load, right rail
V_z	Vertical wheel/rail load
I	Section moment of inertia of rail about horizontal neutral axis
K_b	Tie/ballast effective lateral stiffness
K_B	Tie/ballast effective vertical stiffness
K_p	Rail/tie (pad/fastener, etc.) vertical stiffness
K_r	Rail/tie effective lateral stiffness
K_R	Rail/tie effective vertical stiffness
K_y	Track overall lateral stiffness (rail to ground)
K_z	Track overall vertical stiffness (rail to ground)
L_t	Tie (fastener) spacing along track
L/V	Ratio of lateral to vertical load
P	Point vertical load on rail (beam-on-elastic-foundation (BOEF) Formulation)
U	Track modulus per rail (force/deflection per distance along rail)
x	Longitudinal distance along rail
y	Lateral displacement
y_r	Lateral displacement of rail head
z	Vertical displacement
β	Inverse characteristic length (BOEF formulation)
e_y	Rail alignment geometry error
y_t	Lateral deflection of tie
V	Vehicle speed, ft/sec
f	Frequency, Hz
t	Time, sec.
λ	wavelength, ft.
λ_L	Lateral deflection influence coefficient
λ_v	Vertical deflection influence coefficient
V_a	Static vertical axle load, lb
δ_o	Angle between contact plane and horizontal, rad

ε Rate of change of contact plane slope with lateral displacement of wheelset

ξ Rate of change of distance between wheelset centerline and contact points with lateral displacement of wheelset

λ_c Effective conicity (rate of change of rolling radius with lateral displacement of wheelset), rad

g Effective gauge (lateral distance between contact points with wheelset centered), in.

r_o Wheel tread radius, in.

f_{11} Longitudinal creep coefficient, lb

f_{22} Lateral creep coefficient, lb

f_{23} Lateral/spin creep coefficient, lb-in.

M_r The rail effective mass, lb-sec^2/in.

K_{zr} Rail/tie stiffness, lb/in.

w_r Rail weight per unit, lb/in.

K_t Track structure stiffness, lb/in.

a Tamped length, tie end, in.

b Tamped width of tie, in.

A_T Tamped area, tie end in.^{2}

C Shape factor for soil modulus conversion

u_b Soil modulus, lb/in.^{3}

E_b Ballast modulus (use 40,000 lb/in.^{2})

M_t Tie effective mass, lb-sec^2/in.

w_t Tie weight per length along track

P_i (Track geometry) power spectrum

P_o Vehicle response power spectrum

H System transfer function

ω Frequency, rad/sec.

H.3 MEASUREMENT TECHNIQUES

The absence of standardized procedures for measuring or presenting the track/ rail stiffness or modulus data has caused confusion in interpreting test results available from various sources engaged in testing. In fact, even the proper definitions of these parameters are lacking. Thus, a vertical track modulus value given in pounds per square inch is not track modulus (which should be in pounds/inch/ inch length of rail), but is very likely soil or ballast modulus. Also, the description accompanying stiffness data usually lacks in details. Important parameters, such as the exact locations of measurements, the loads used, if one or both rails were loaded, a qualitative description of soil/ballast at the measurement sites, the number of measurements taken to arrive at the stiffness data, etc., are not specified.

This subsection deals primarily with presenting preferable measurement techniques, with a view toward eliminating some of the above confusion and thereby generating test results which can be interpreted and used in subsequent vehicle performance evaluation.

H.3.1 Methods for Measuring Static and Dynamic Displacements

Stiffness is nothing but a force/displacement characteristic. Thus, one of the first topics discussed is the various types of displacements applicable to rail/track stiffness measurements. In this discussion, "dynamic" means measurements taken with a test consist providing the necessary loads through operation over the test section, whereas "static" means measurements taken with a stationary loading fixture.

Several different absolute and/or relative measurements are needed to fully describe the upper track response to loads. The same basic measurements apply to both static and dynamic displacements. The requirements for the dynamic measurements are more stringent than the static measurement requirements because of the added frequency response needed and the need to provide for the survival of the transducer in the rugged operating environment of rail traffic. These displacements and their primary importance are: (1) Rail vertical absolute displacement, used to define the track modulus and dynamic load/deflection characteristics, (2) Rail head/tie lateral displacement, used to measure rail lateral restraint characteristics under both lateral and vertical loading; (3) Tie/ground lateral displacement, data under traffic can document the occurrence of lateral shift of the tie in the ballast, used to establish track lateral strength limits; (4) Rail rotation (roll), used to document the mode of rail deflection and loading on tie/fastener system; (5) Rail rotation (pitch), used to determine loading environment on tie/fastener system, (6) Rail/tie vertical displacement, used to determine dynamic load/deflection characteristics and loading environment on tie/fastener system, (7) Rail longitudinal displacement, used to determine tie/fastener rail restraint capabilities.

A major difficulty in measuring displacements is the establishment of references from which the measurements are to be made. Establishing "absolute" reference points adjacent to the track structure requires going deep enough, or far enough to the side of the track, to locate ground which does not move relative to the track. While pressures in the ballast/subgrade drop off quickly to something less than 3 lb/in^2 (21 kN/m^2) at a depth of 40 inches (102 cm), both track structure modeling and field experiments have shown vertical deflections to decrease with depth. At a 40-inch depth, typically half the vertical deflection will still be measured.

Past experience has shown that absolute deflection measurements related to rail joint or rail fastener performance can be referenced to "ground" by attaching the transducer to a rod driven down into the subgrade. In the concrete tie track study [29], a 1-inch diameter steel rod was driven through a concentric hollow pipe casing through the ballast into the subgrade. The casing was about 4-ft long to isolate the rod from ballast movements; while the steel rod was 8 ft long and was driven into the ballast/subgrade until only about 8 inches projected above the ballast surface. In other field experiments, shorter rods have been used driven directly through ballast into the subgrade without benefit of the casing. Vertical deflections using a 4-ft rod showed a range from 0.10 to 0.14 inch under locomotive axles, while static calibration (viewed through an off-track transit) showed 0.18 inch deflection under a 30,000 lb (133 kN) point load. "Ground stakes" such as these have been used quite successfully for establishing reference points for lateral deflection measurements. When using this type of reference, the rod must be stiff enough to minimize deflections from any loads imposed by the measurement transducer. This is particularly important when dynamic measurements are being made which might excite a vibratory response in the reference rod. For the measurement of static lateral displacements, several programs have used 3-ft ground rods driven directly into the ballast. Since measurements were to be made at a number of different sites, the shorter rod allowed the rods to be installed and removed in a minimum of time. The errors introduced by the shorter rods were negligible since the displacements needed for the measurement of lateral track strength are large when compared with normal track deflections under traffic. A schematic diagram of a setup to measure lateral track displacements is shown in Figure H-1. The response of a section of track to lateral load exerted on the rail by hydraulic cylinders is shown in Figure H-2.

Relative measurements must also be isolated from undesirable displacements. For example, if a dynamic track gauge is to be measured and the tie is used as a reference for individual rail displacements, then tie bending could readily distort the intended output. The deflection measurement fixture developed by Battelle for the Track Train Dynamics Program [30] is an example of a measurement system that provides displacement measurements of the rail without distortion from bending of the wood tie. A conceptual drawing of this fixture is shown in Figure H-3, while Figure H-4 shows the relative displacements which are measured. In addition, the fixture provides some degree of shock and vibration isolation for the transducers and signal conditioning electronics through elastomeric grommets and lag screws mounting the fixture to the tie. Acceleration levels on the tie can range typically up to 50 g under flat wheel impact loads. Typical deflection measurements from the fixture shown in Figure H-4 are illustrated in Figure H-5. Dynamic track gauge and rail rollover (of one rail only) under severe lateral impact loads due to empty freight car truck hunting are seen here, along with about 1 mm of permanent lateral shift of the tie.

In measurements on much stiffer concrete ties, a fixture which eliminates the effects of tie bending was found unnecessary. A conceptual drawing of a fixture used for recent measurements of concrete tie fastener/pad deflections [31] is shown in Figure H-6. Here the measured rail-to-tie displacements,

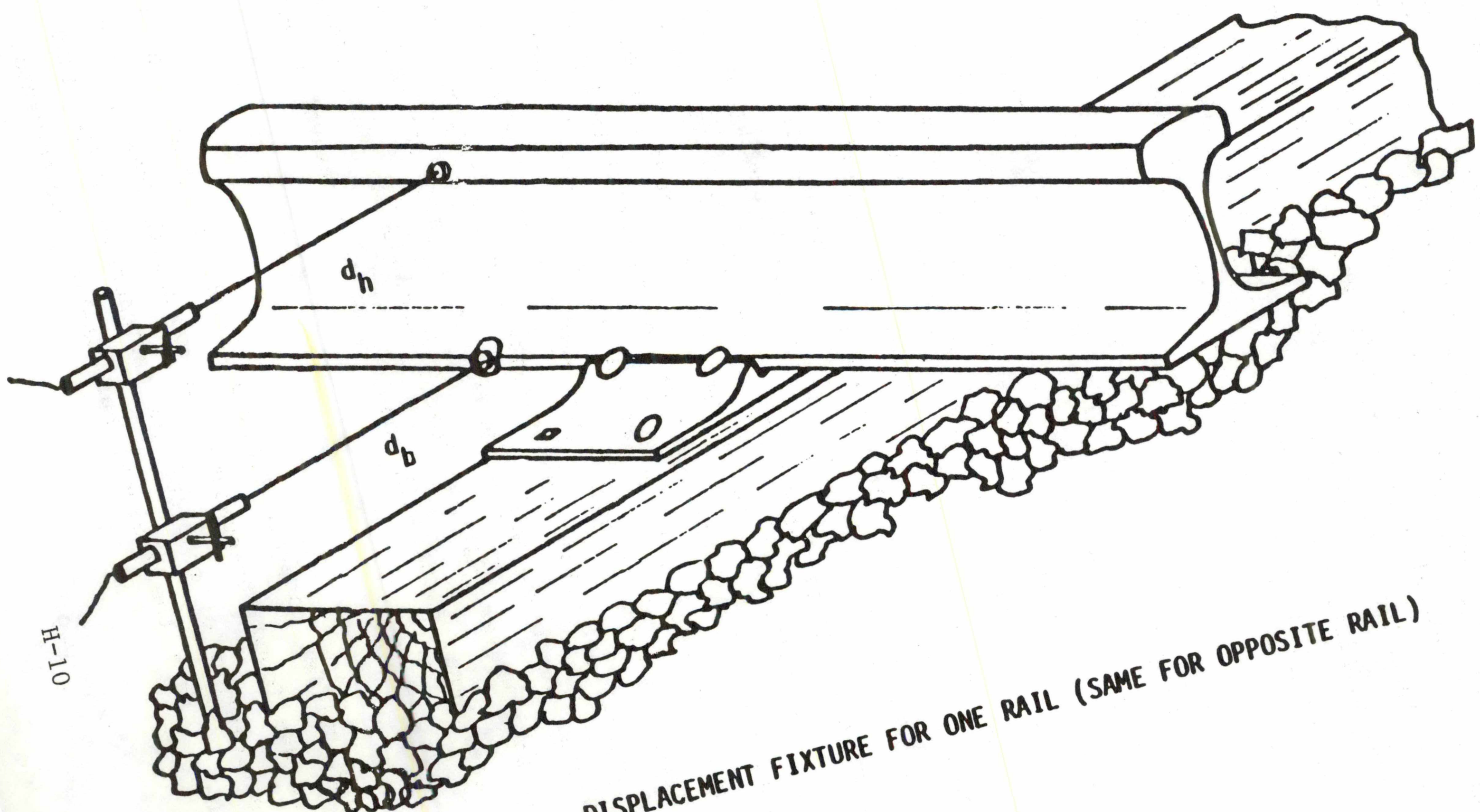


FIGURE H-1. DISPLACEMENT FIXTURE FOR ONE RAIL (SAME FOR OPPOSITE RAIL)

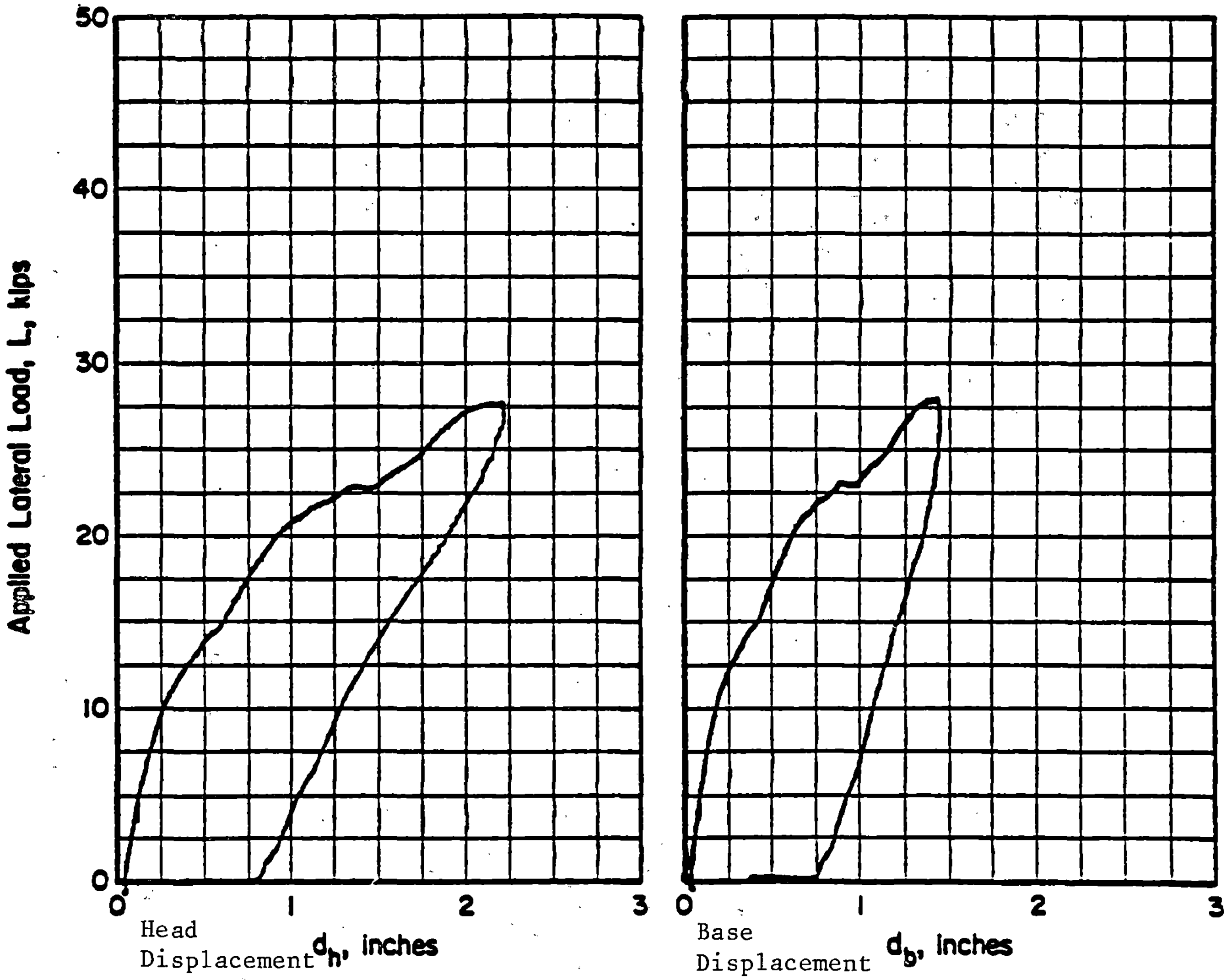


FIGURE H-2. LATERAL LOAD VS. RAIL HEAD AND BASE DISPLACEMEMTS

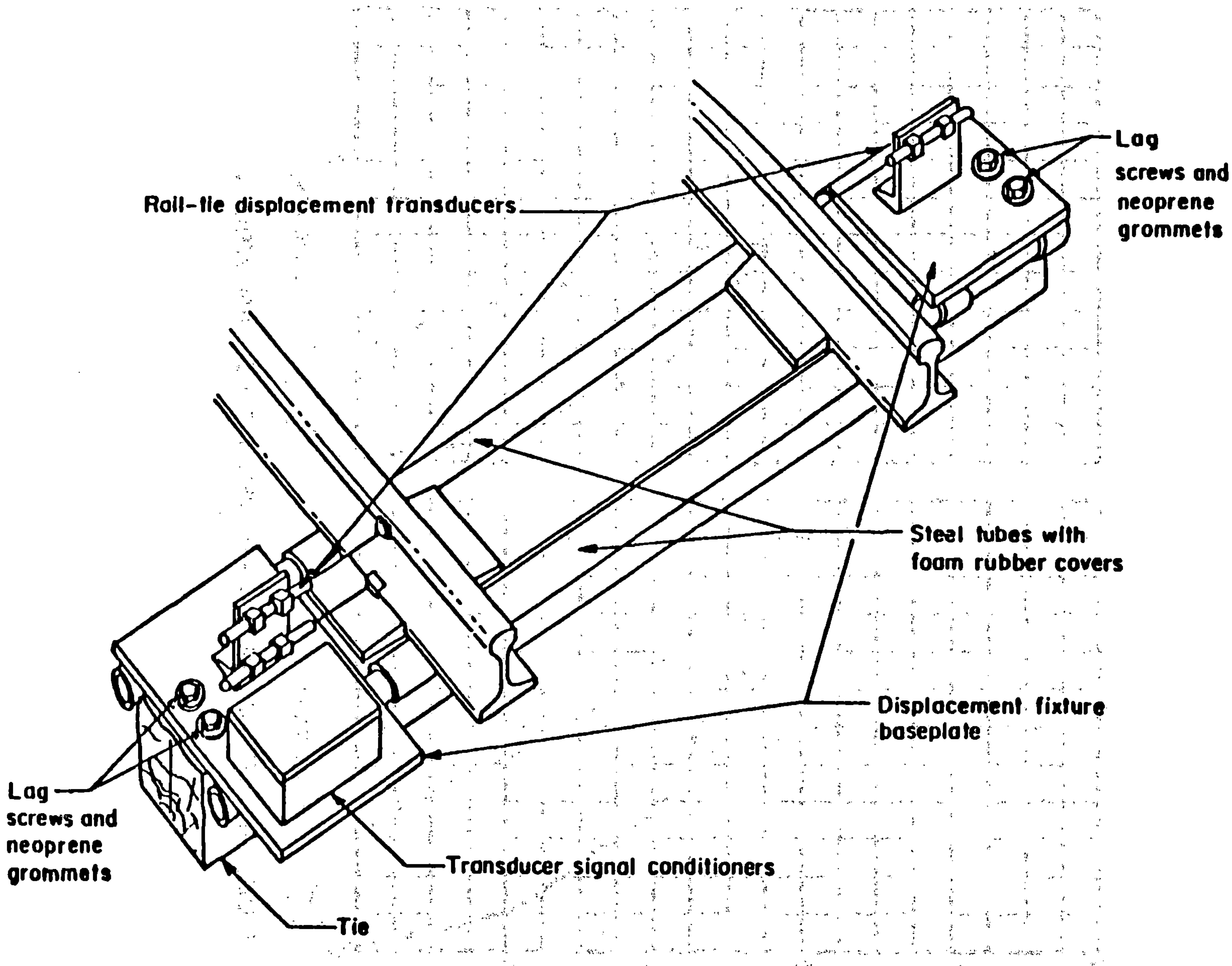


FIGURE H-3. STABLE BASE FIXTURE FOR UPPER TRACK STRUCTURE DYNAMIC RESPONSE MEASUREMENTS UNDER WHEEL/RAIL LOADS

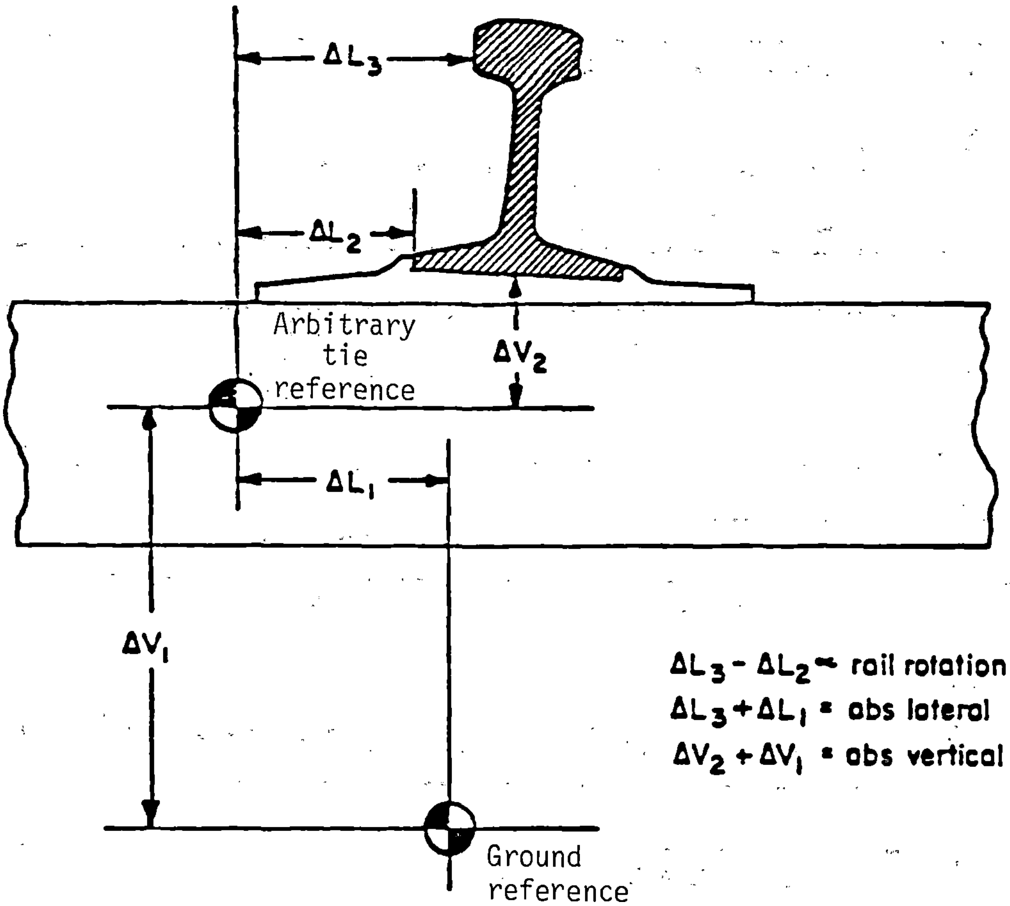


FIGURE H-4. MEASUREMENTS OF RELATIVE AND ABSOLUTE DISPLACEMENT NEEDED TO DEFINE UPPER TRACK STRUCTURE DYNAMIC RESPONSE TO WHEEL/RAIL LOADS

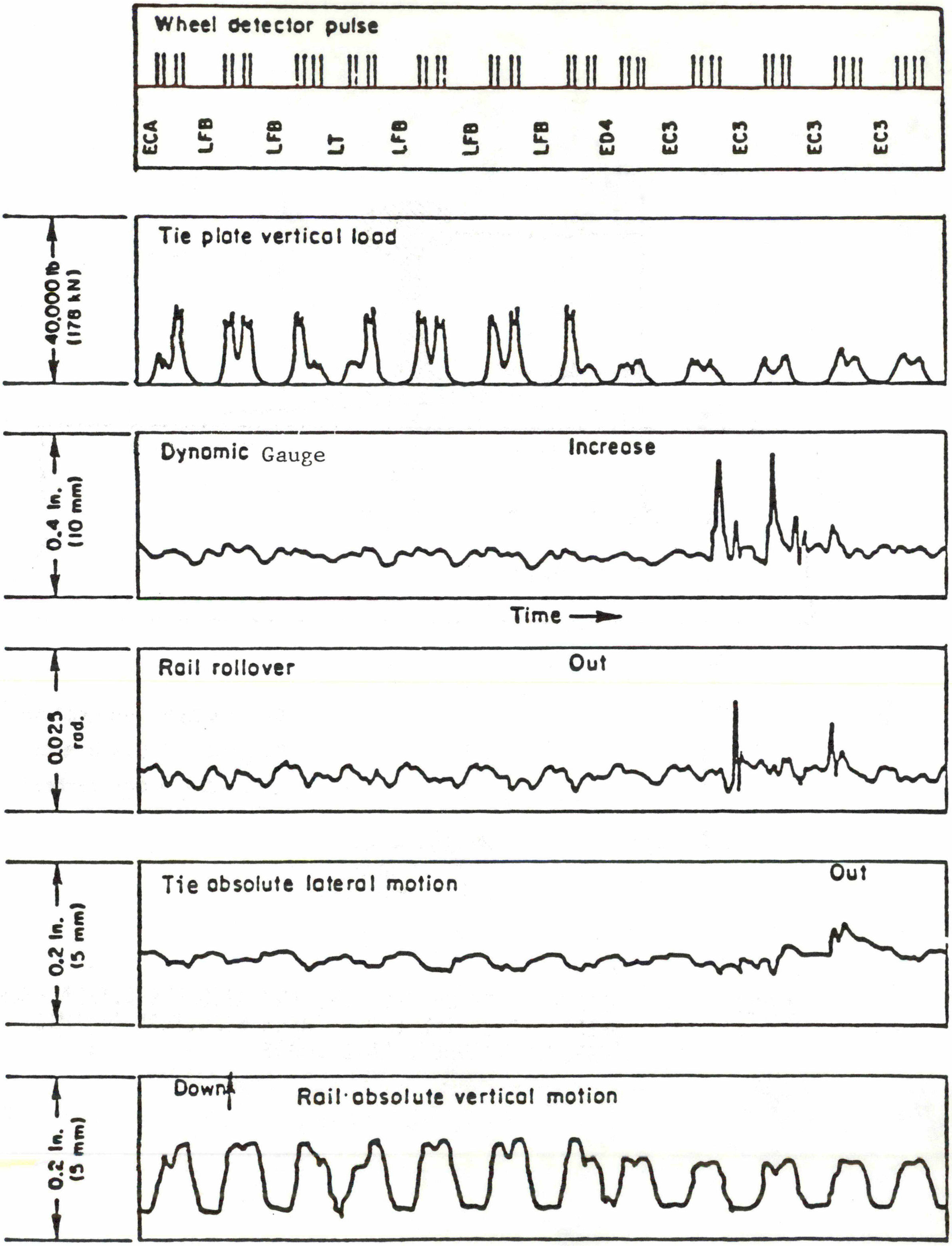


FIGURE H-5. TYPICAL TRACK DYNAMIC RESPONSE MEASURED FROM TRACK TRAIN DYNAMICS PROGRAM INVESTIGATION OF WIDE GAGE ON HIGH-SPEED TANGENT TRACK -- REVENUE FREIGHT TRAIN AT 62 MI/H

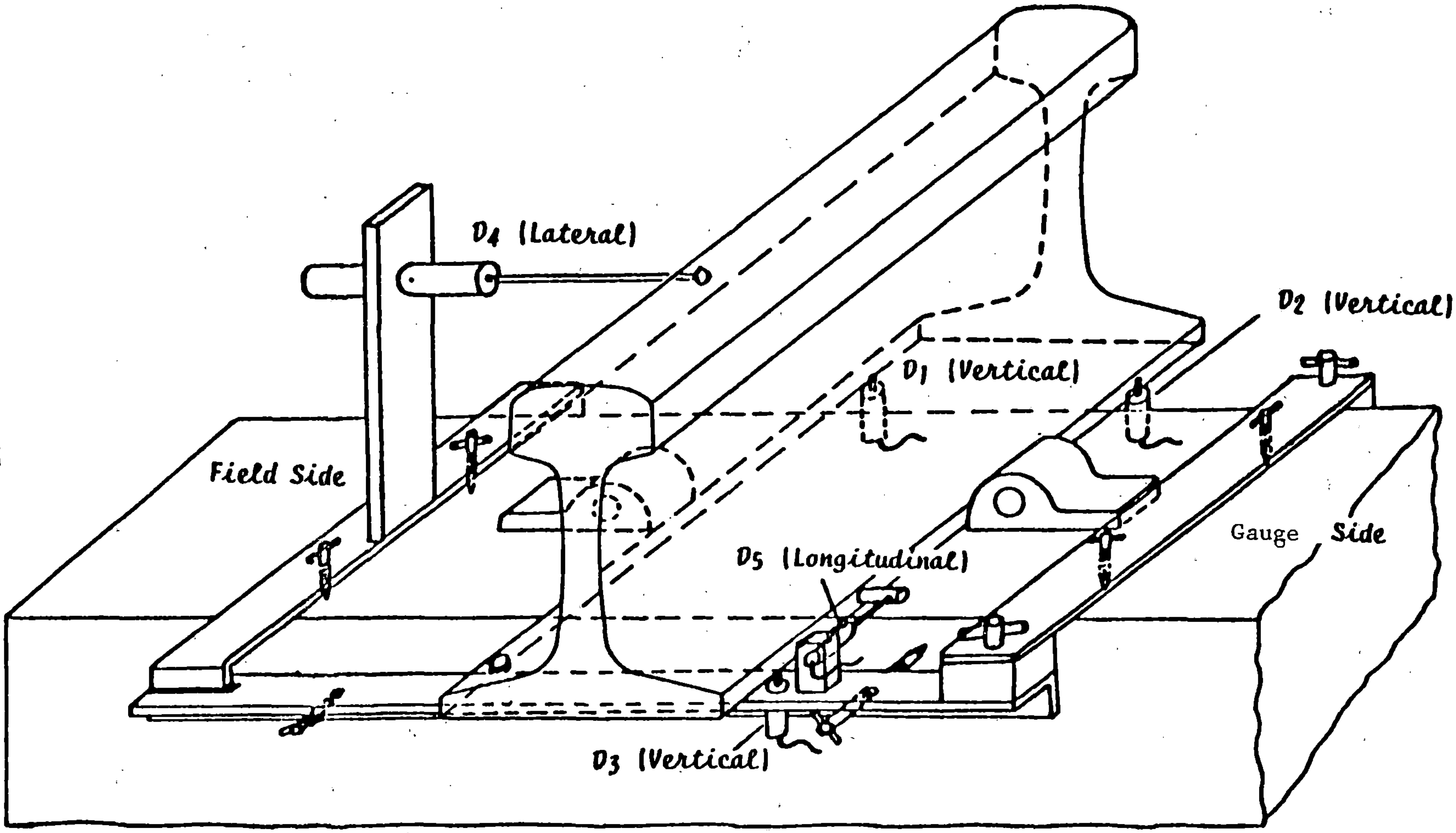


FIGURE H-6. FIXTURE FOR MOUNTING RAIL/TIE DISPLACEMENT TRANSDUCERS along with fastener clip strains, were used to define the loading environment on rail fastener systems.

In the fixture shown in Figure H-6 vertical rail deflections are measured relative to the tie. This measurement is useful in fastener evaluations but is insufficient when trying to determine overall track vertical deflections or when track stiffness calculations are of interest. In order to obtain absolute vertical deflections a reference measurement point must be established outside the load influence zone of the ballast and sub-ballast. For normal stiffness track this requires a deeply driven ground rod which is isolated from the ballast movement. Once an isolated reference point is established then the methodology used for measuring the vertical deflections is the same as that used for measuring lateral deflections. A sketch of a fixture used in a recent field test is shown schematically in Figure H-7. Although it is restricted to small motions due to errors incurred when measuring large deflections, good results were obtained from the fixture.

When analyzing dynamic track load data on highly compliant track, a problem was discovered with high rail lateral load data analysis. As shown in Figure H-8 (sensors L1H, L2H, L3H, etc.), just ahead of each leading outer wheel one can see a negative- (inward-) going pulse followed generally by a positive load pulse as the wheel passed over the gauge pattern. This "Negative Pulse" occurs just as the wheel passes over the tie and enters the instrumented crib, and is associated primarily with leading outer (flanging) wheels. It may therefore be due to the more asymmetric rail loading and to the superelevation of the outer rail thus causing a torsional strain from the approaching vertical load of the flanging wheels. The value of these "Negative Pulses" should decrease as the tie/fastener stiffness increases.

While the "Negative Pulse" can be seen visually on the oscillograph traces and eliminated from the data, there is a tendency in the data processing for peak detectors to choose the false peak, particularly on slave channels where the edge of the zone is not well defined. This effect can be minimized by a slight negative bias voltage on the lateral load data channel before taking the absolute value, at the expense of missing a few real negative load peaks.

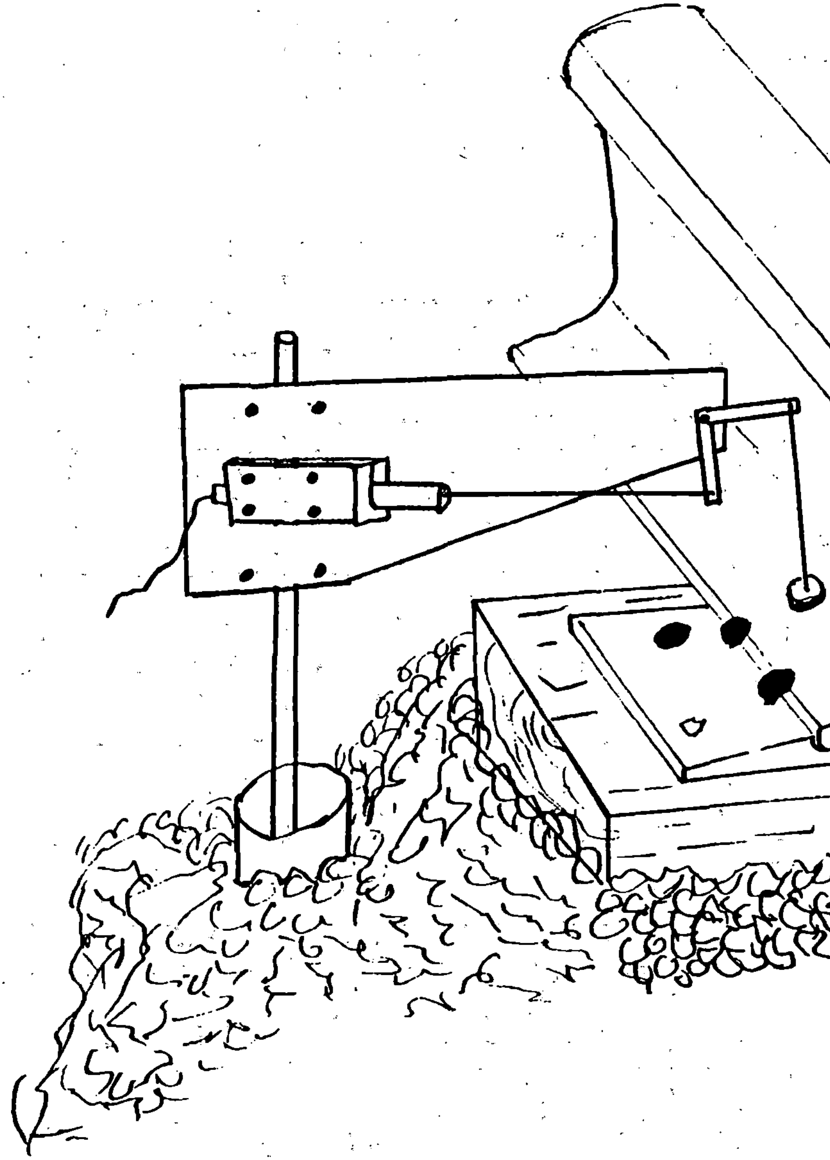
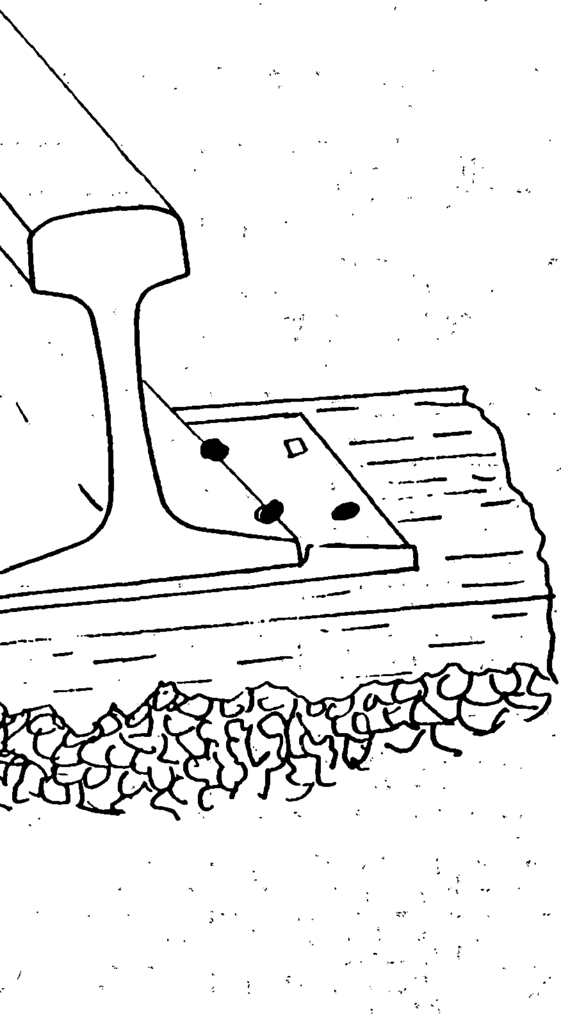
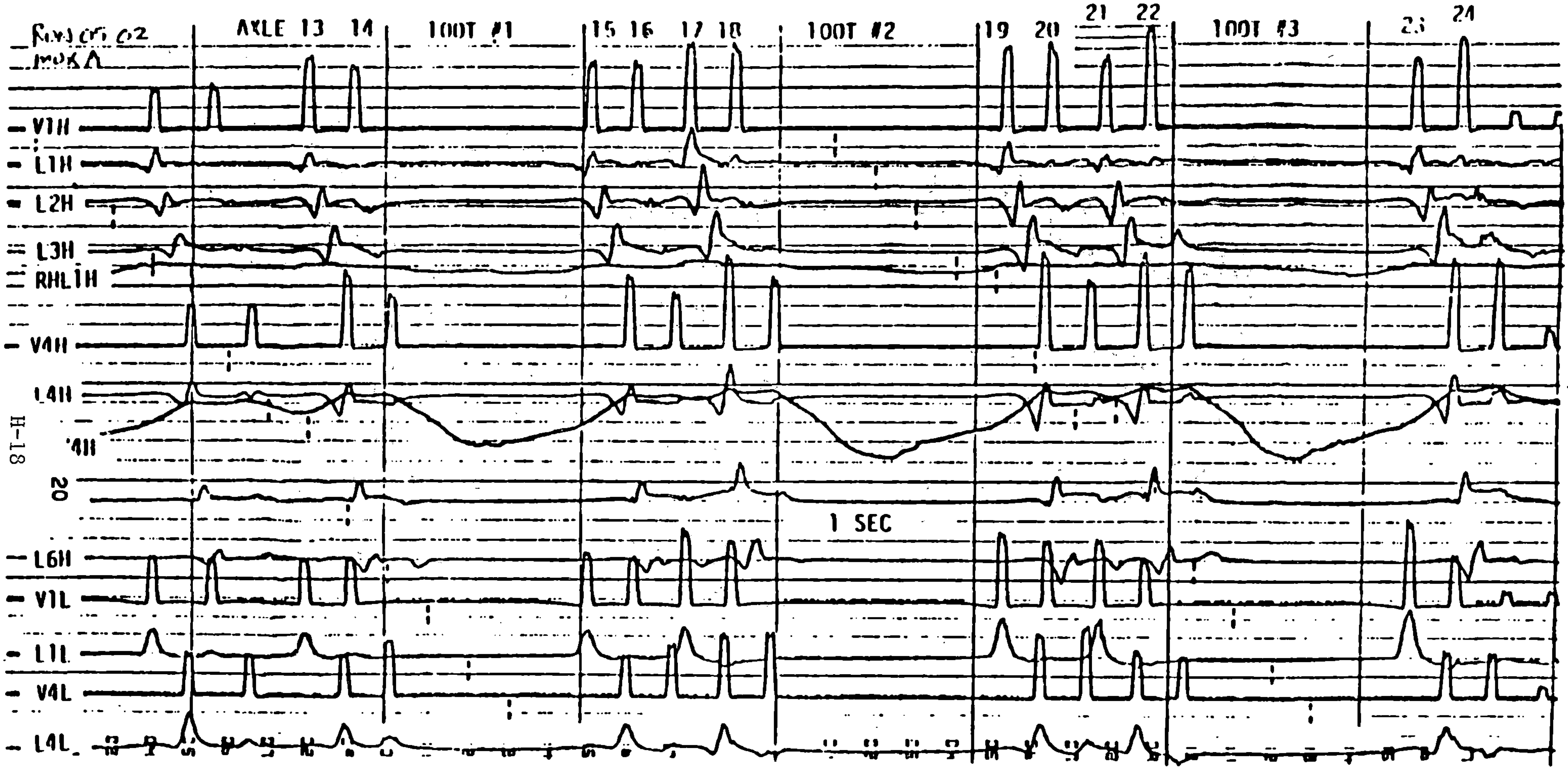


FIGURE H-7. VERTICAL DISPLACEMENT FIXTURE.





Scale: V - 10,000 lb/div; L = 5,000 lb/div; RHL & RBV = 0.2 in/div

FIGURE H-8. EXAMPLE OF OSCILLOGRAPHIC RECORDING OF WHEEL/RAIL LOADS AND RAIL DEFLECTIONS UNDER LOADED 100 TON HOPPER CARS IN SECTION 1.3 (SOFT TRACK) -- RUN 05-02, AT 17 MI/H ON A 6° CURVE FOR A VEHICLE/TRACK INTERACTION TEST.

Suspicious data points should be laboriously checked against the oscillograph traces before establishing the peak value tables for further analysis.

H.3.2 Static Force/Deflection Measurements

A Track Loading Fixture (TLF) such as that developed for rail restraint tests [32] can be used to characterize the lateral compliance and rail gauge restraint of a test track. A sketch of the TLF is shown in Figure H-9. This fixture applies independent vertical (V) and lateral (L) loads to the rail head through sections of standard AAR 1:20-taper, 36-inch freight car wheels. An hydraulic power supply with a pressure regulator powers each separately. The loads applied to the rail heads are measured through laboratory-calibrated clevis pin load cells which compensate for the misalignment of the vertical actuators under large lateral deflections. These load cells are powered by amplifiers and utilize precision shunt calibration resistors to set the proper output levels in engineering force units.

Lateral deflections of the rail head and base can be measured by direct current differential transformers (DCDTs) mounted to a ground stake driven into the ballast. The moving cores of the direct current differential transformers can be mounted to the rail through threaded "ready rod" extensions to phenolic blocks which are normally cemented to the head or base. To reduce set-up time during the TLF operation, these blocks can be c-clamped in place. Signals from the deflection transducers and lateral load cells can be recorded directly on a pair of XYY plotters so that lateral load versus rail head and base deflections for both the high and low rails are plotted simultaneously as shown in Figure H-10.

During tests, the vertical actuators of the TLF are mounted to the heavy I-beam cross bearer of a loaded 100-ton hopper car. This both supports the TLF when moving between sites and provides a reaction mass for vertical loads up to 37,000 lb per rail. A plywood platform can be built between two of the three hoppers of the car to provide space for hydraulic power supplies, instrumentation and XYY plotters, as well as adequate protection during inclement weather. The TLF has been modified so that it can be installed on freight vehicles as a single system incorporating the power supply, actuators, sensors, and chart recorders.

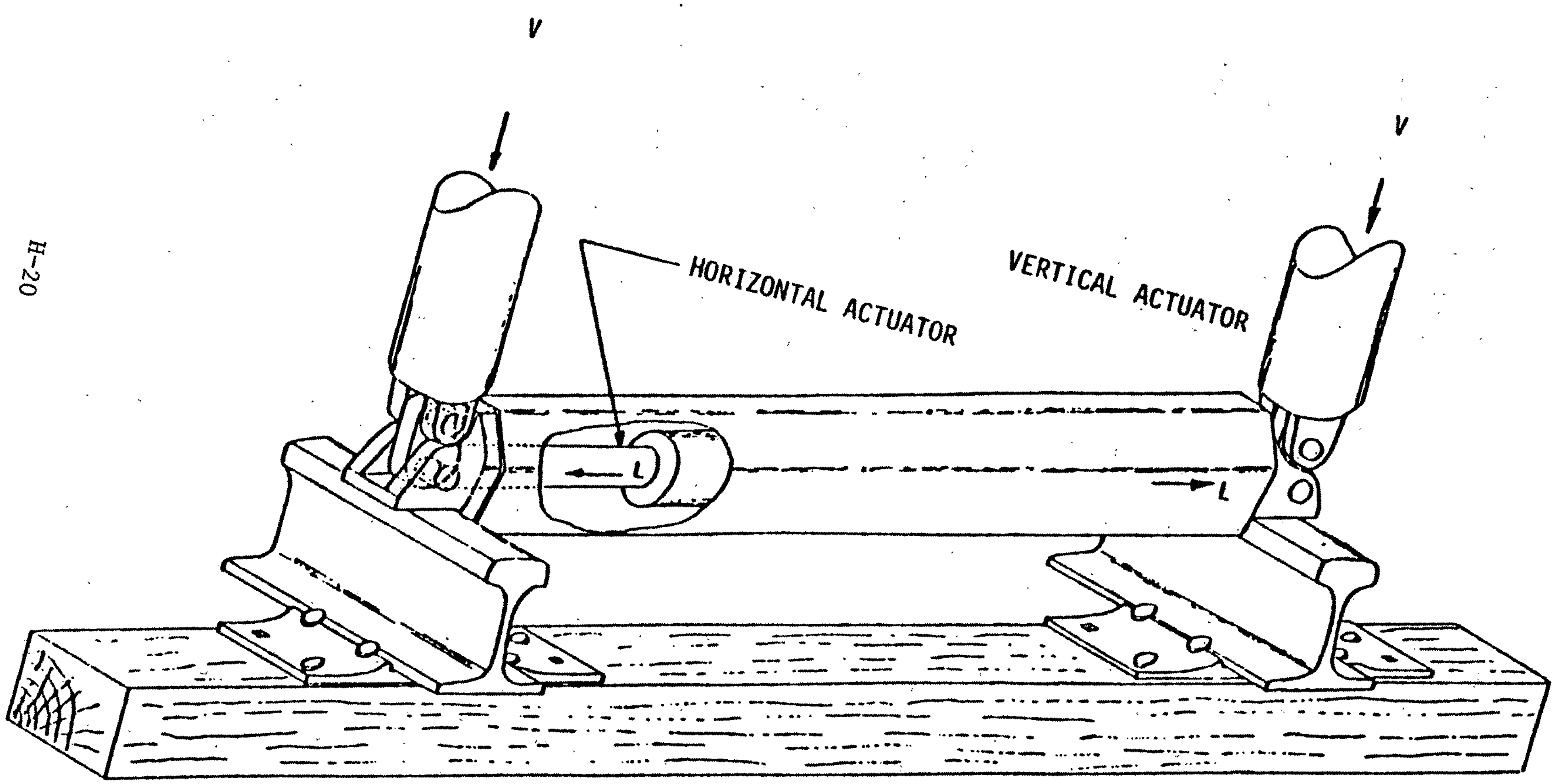


FIGURE H-9. BASIC LAYOUT OF TRACK LOADING FIXTURE

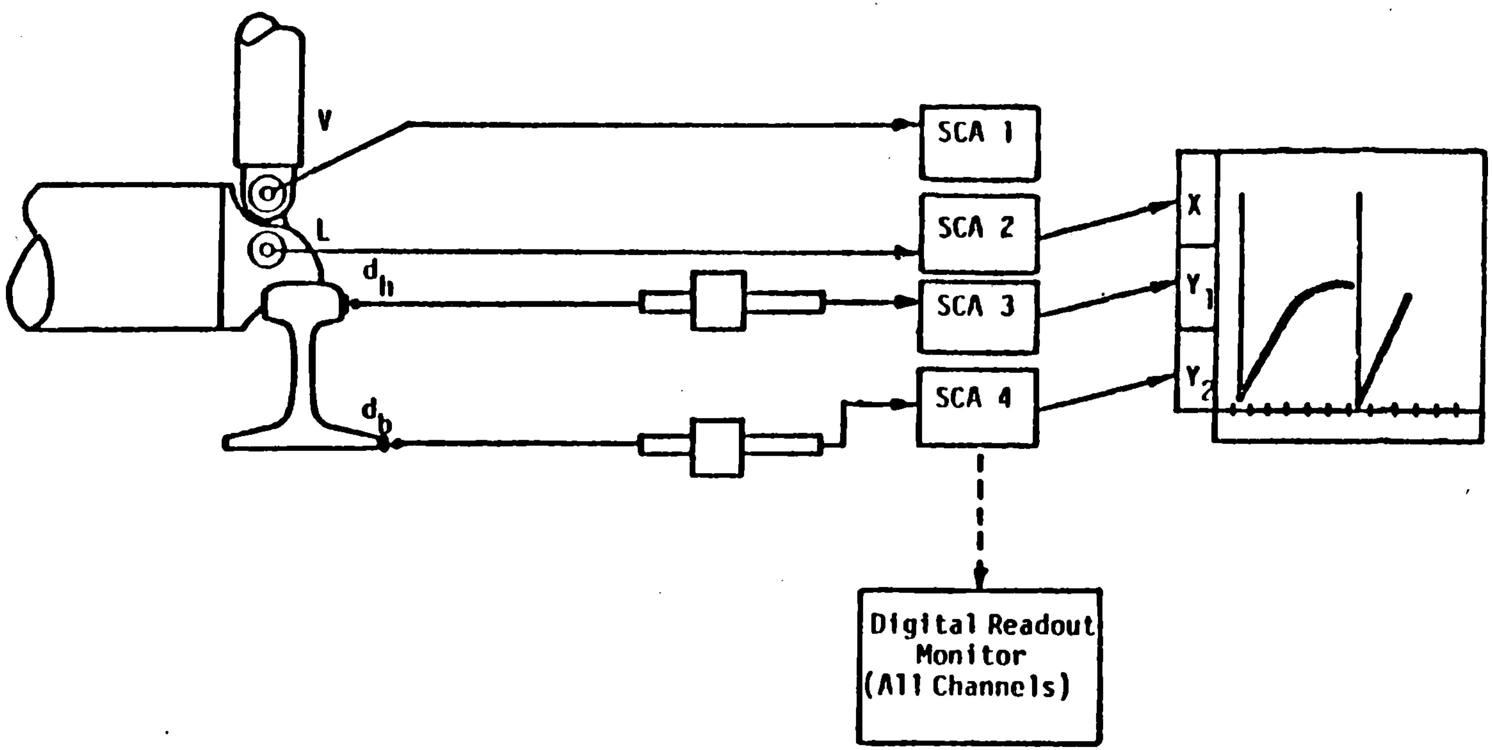


FIGURE H-10. SCHEMATIC ARRANGEMENT OF FORCE AND DEFLECTION TRANSDUCERS USED FOR RAIL LATERAL RESTRAINT INVESTIGATION [3]

H.4 FACTORS AFFECTING TRACK STIFFNESS OR MODULUS

Vertical track stiffness is shown to be a "hardening" nonlinear spring rate, increasing in value with higher vertical load, both for the rail/tie portion and for the overall track stiffness. Part of this is due to slack (clearance under the rail base, ballast voids, etc.), and part is due to pads, ties, ballast, and soil acting in a manner analogous to an elastometric element.

In the lateral direction, stiffness consists of two components, a rail-spreading stiffness (primarily the pad/fastener lateral and torsional support stiffnesses) with load path through the tie to the other rail; and a tie (or slab) to ground stiffness, responding to the net lateral load of the wheelset on both rails. Both of these are "softening" spring rates, with sharply increasing deflections per lateral load increment above a certain L/V ratio.

Both stiffnesses exhibit some deformation or "friction" load limit, above which permanent displacement of the rail or tie will occur. Both the stiffness characteristic and the deformation limit are strongly dependent on the simultaneous vertical load level.

There is a great variation of track stiffness across ties and also from tie to tie. These variations are due to traffic, tonnage, joints, track curvature, maintenance and original construction factors. The results of static vertical track modulus measurements identified that for new construction and low usage track, the scatter of modulus values was greater for segments where the earth was not disturbed to establish the track grade than where back filling and compaction of the subsoil were required. After new construction approximately 10 million gross tons of traffic were applied before the vertical track modulus would tend to stablize as shown in Figure H-11. In addition, when ties are replaced or ballast disturbed approximately 0.2 million gross tons of traffic over the disturbed location was required to re-establish a gradual change of vertical track modulus with tonnage.

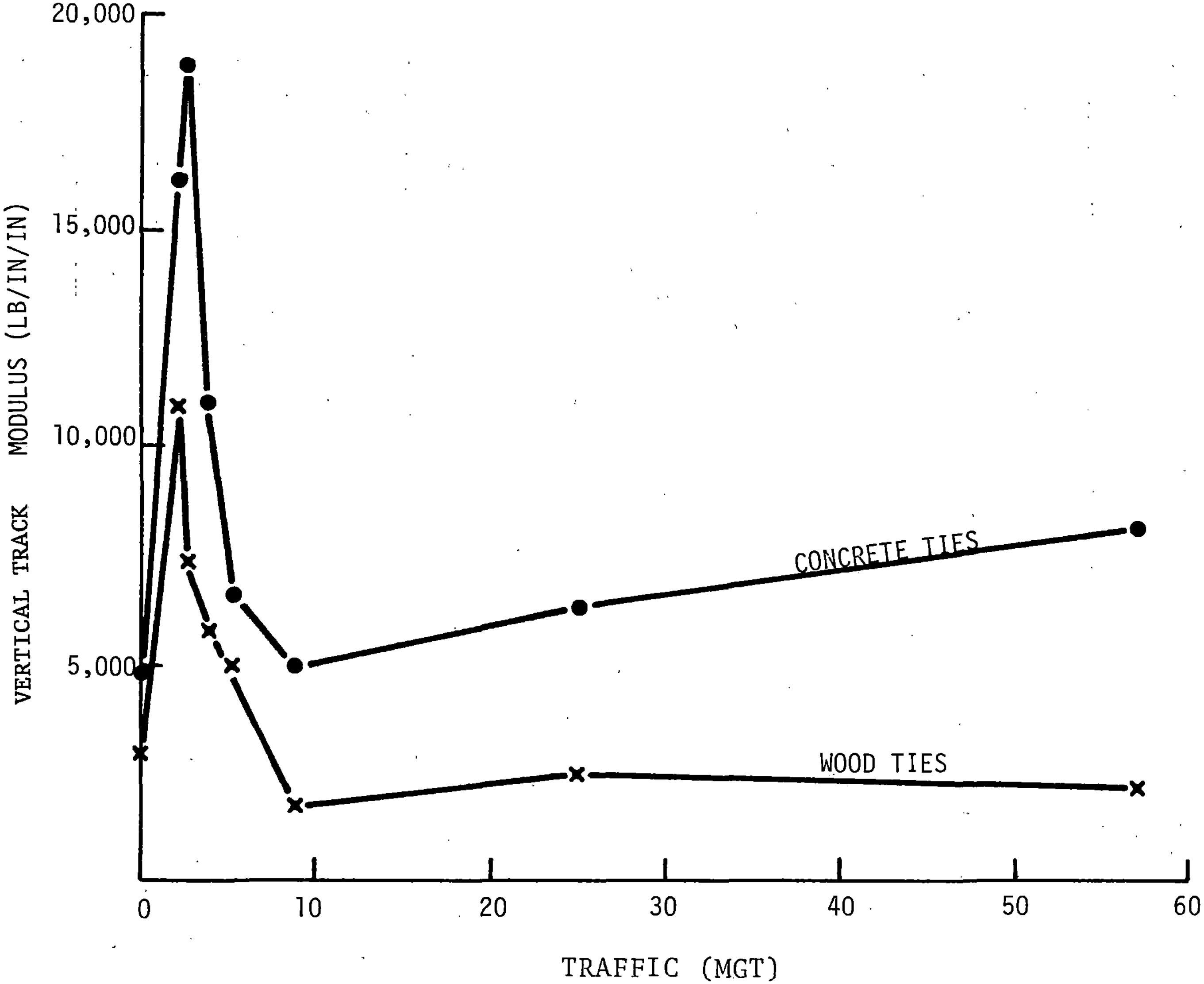


FIGURE H-11. VERTICAL TRACK MODULUS. FAST SECTION 22

H.4.1 Vertical Track Modulus or Stiffness

An accepted, approximate theory for predicting the vertical deflections of a rail under a point load is based on considering the rail as a beam continuously supported by a linear elastic foundation [4,5,6]. This method was first applied by Zimmerman (Berlin, 1888) to calculate the stresses and deflections of rails under load. Later work by Talbot [7], and Timoshenko and Langer [8] demonstrated the general usefulness and accuracy of elastic foundation theory in predicting rail deflections and bending stresses due to vertical wheel loads. Recent work by the AAR [9] has compared this theory with other methods for predicting deflections under load. The basic relationships from this theory are:

$$K_z = V_z/z_o = 2[4EIU^3]^{0.25} \tag{1}$$

$$z = (V_z/K_z)e^{-\beta x}[\text{Cos}\beta x + \text{Sin}\beta x] \tag{2}$$

$$\beta = [U/4EI]^{0.25} \tag{3}$$

$$U = K_z\beta/2 = 0.25[K_z^{\,4}/EI]^{0.333} \tag{4}$$

In this linear theory, the deflection z in response to the applied point load P is used to define the track stiffness K_z directly under the load ($x = 0$). Track modulus U is derived from the measured stiffness K_z and is defined as a stiffness per unit length along the rail. It should not be confused with the "modulus" of soil mechanics (of which subgrade modulus is an example), which is the load applied to a standard plate size (area) to obtain unit deflection.

While the beam-on-elastic-foundation (BOEF) model of railroad track provides a good first approximation of track stiffness or static modulus, the basically nonlinear behavior of track, particularly wood tie, cut spike track construction, becomes apparent in field measurements. For example, rail vertical deflections calculated from the BOEF theory are compared in Figure H-12 with deflections measured under the wheels of both empty and loaded 100-ton-

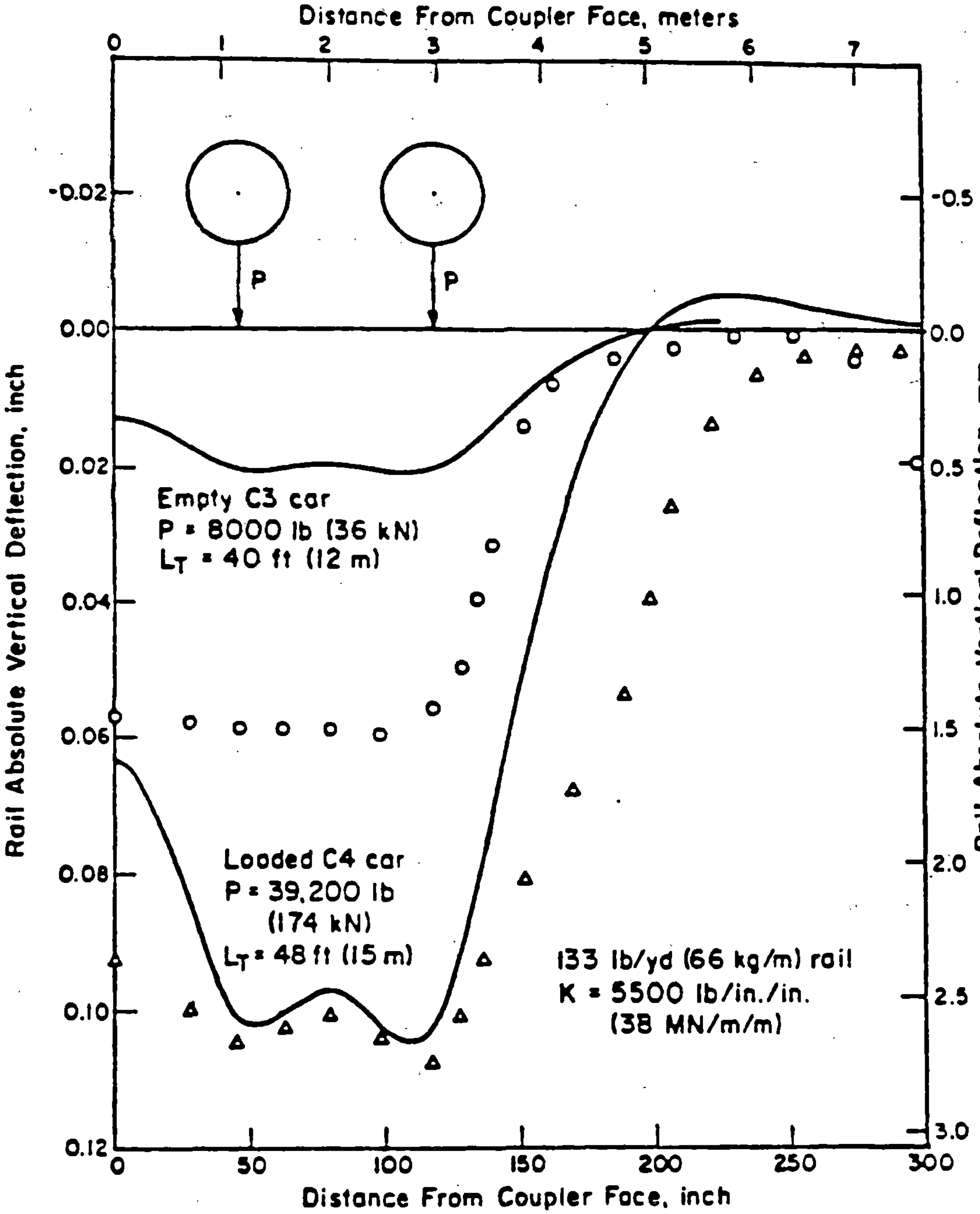


FIGURE H-12. COMPARISON OF IDEALIZED (LINEAR BEAM ON ELASTIC FOUNDATION) AND MEASURED RAIL DEFLECTION SHAPES UNDER WHEEL LOADS OF COUPLED CARS capacity hopper cars. While the agreement between predicted and measured deflections is excellent for the loaded cars, substantially more deflection of the track is measured under the empty-car vertical wheel loads than is predicted. The nonlinear rail/tie and tie/ballast stiffnesses typically include considerable free-play until all components are seated under load.

The track, whether cross ties on ballast or compliant fasteners on beams or slab, is actually a series of discrete springs supporting the rail beam. These springs are then in series with springs representing the ties, ballast and subgrade, forming the overall "elastic foundation". For the crosstie track, beam bending rigidity is explicitly provided by the rail; while on slab track the concrete slab or "invert" provides additional bending rigidity in a two-layer beam/elastic foundation system.

H.4.1.1 Rail/Tie Vertical Stiffness

Typical values of the vertical stiffness associated with the rail/tie portion of the track structure (through the rail seat pad and fastener, for example) are given in Table H-2. Rail/tie force/deflection experiments have been conducted, including field measurements on wood ties [10] and laboratory measurements on concrete ties [11]. An example of quasistatic force deflection curves on wood ties in good and poor condition is given in Figure H-13. The two curves on each plot represent two consecutive cycles from an 800-lb preload, with some semi-permanent deformation or hyteresis. These curves can be adequately defined by a bilinear approximation:

$$V_z = K_{p1}\Delta z \qquad \text{for } \Delta z < \delta \tag{5a}$$

$$V_z = K_{p2}(\Delta z - \delta) \qquad \text{for } \Delta z > \delta \tag{5b}$$

where

Δz = relative vertical deflection, rail to tie

δ = value of Δz at which stiffness K_{p2} is applied.

TABLE H-2. VERTICAL STIFFNESS OF RAIL/TIE FASTENER SYSTEMS

Fastener/Pad Type	$K_{z_{tie}}$ (lb/in)
Wood tie (softwood) [Birmann, DB]	$2.8(10)^5$ to $8.4(10)^5$
Wood tie (hardwood) "	$1.7(10)^6$ to $2.8(10)^6$
Wood tie, 14" tie plate [BCL/Chessie]	
at V_z = 12,800 lb	$6.1(10)^5$
at V_z = 17,000 lb	$1.2(10)^6$
Concrete tie [JNR Tokaido Shinkansen]	$3.4(10)^5$ to $5.4(10)^5$
Concrete tie [NEC fasteners]	$4.0(10)^6$ to $8.1(10)^6$
at V_z = 8,000 lb	$4.0(10)^6$ to $5.5(10)^6$
at V_z = 15,000 lb	$5.6(10)^6$ to $8.1(10)^6$
RN rail fastener and chevron sole pad	$2.2(10)^5$ in extension
[French, per R. Sonneville]	$4.5(10)^6$ in compression
BARTD	$4.4(10)^5$ to $5.7(10)^5$

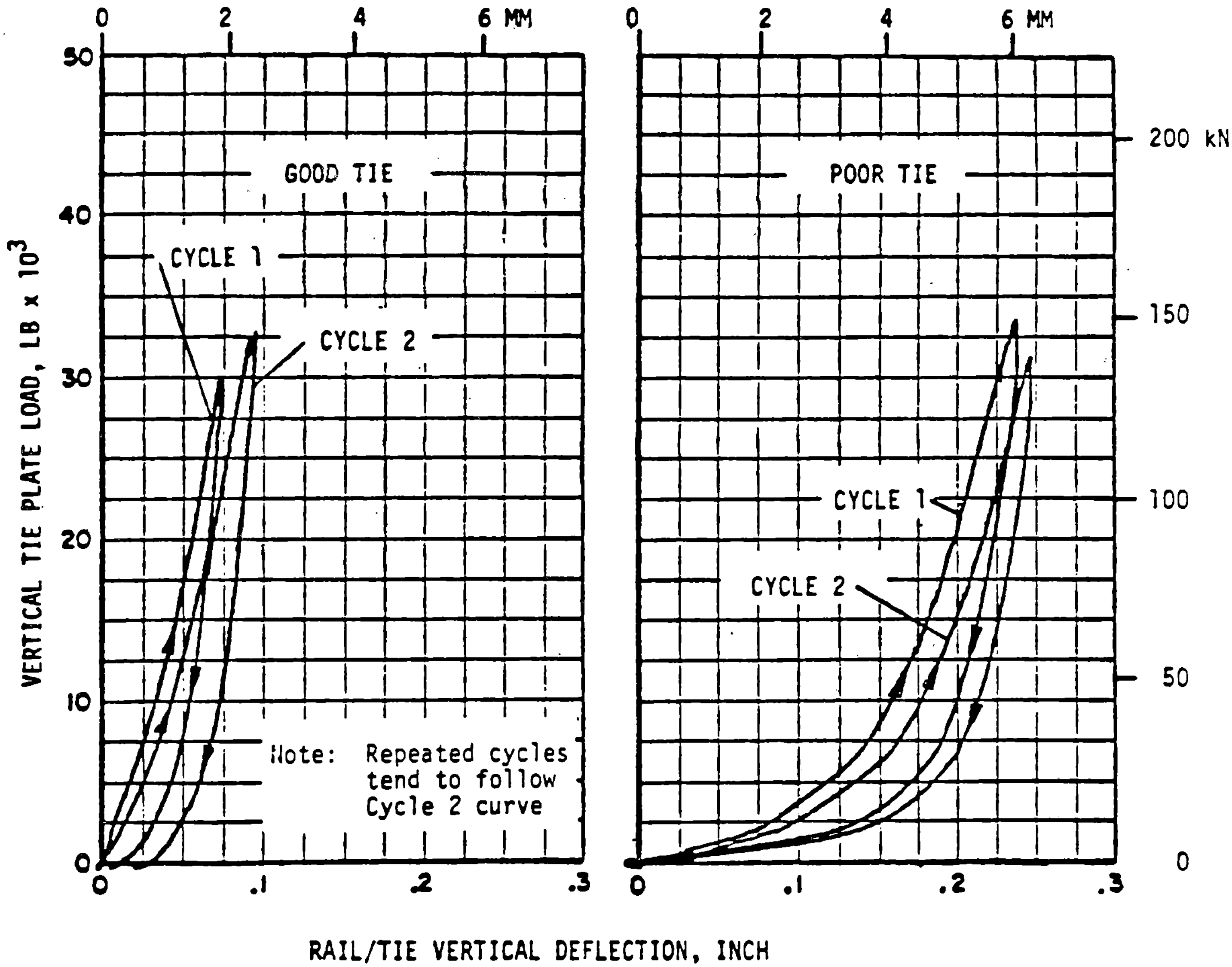


FIGURE H-13. COMPARISON OF RAIL-TO-TIE FORCE/DEFLECTION CHARACTERISTICS FOR WOOD TIES IN GOOD AND POOR CONDITIONS

For the two examples in Figure H-13, the following values can be used:

Condition	K_{p1}, lb/in.	δ_z, in.	K_{p2}, lb/in.
Good	170,000	0.03	440,00
Poor	36,000	0.18	390,000

Results from laboratory tests of specimens of the stiff concrete tie rail seat pads typical of the Northeast Corridor track are shown in Figure H-14, along with secant spring rates (the straight-line slope between two load levels). The tangent spring rates (defined as the slope tangent to the static load level) range from 4.0 $(10)^6$ lb/in. under a typical passenger car wheel, and up to 8.1 $(10)^6$ lb/in. under a locomotive or heavy freight carwheel. These pads are an order of magnitude stiffer than those used by the Japanese National Railways on the Tokaido Shinkansen Line.

H.4.1.2 Track Overall Stiffness

A range of typical track vertical stiffness values for both wood tie and concrete tie track structures is given in Table H-3. Some of these values are calculated from point-load force/deflection measurements, taken by observing the rail vertical deflection through an off-track surveyor's transit during vertical wheel/rail load circuit calibrations. These are, therefore, midcrib stiffness values. Other values were derived from force/deflection measurement under locomotive or freight car wheels, and therefore included some additional deflection due to adjacent wheel loads. This effect is addressed in a later section.

Measurements on good wood-tie track with 133 lb/yd rail, under a single vertical load [12] showed vertical stiffness values of 200,000 $\pm$75,000 lb/in. for bolted-joint rail (BJR) track, and 290,000 $\pm$120,000 lb/in. for continuous-welded rail (CWR) track, for seven measurement sites in each section. The force/deflection curves from these experiments can be adequately represented by:

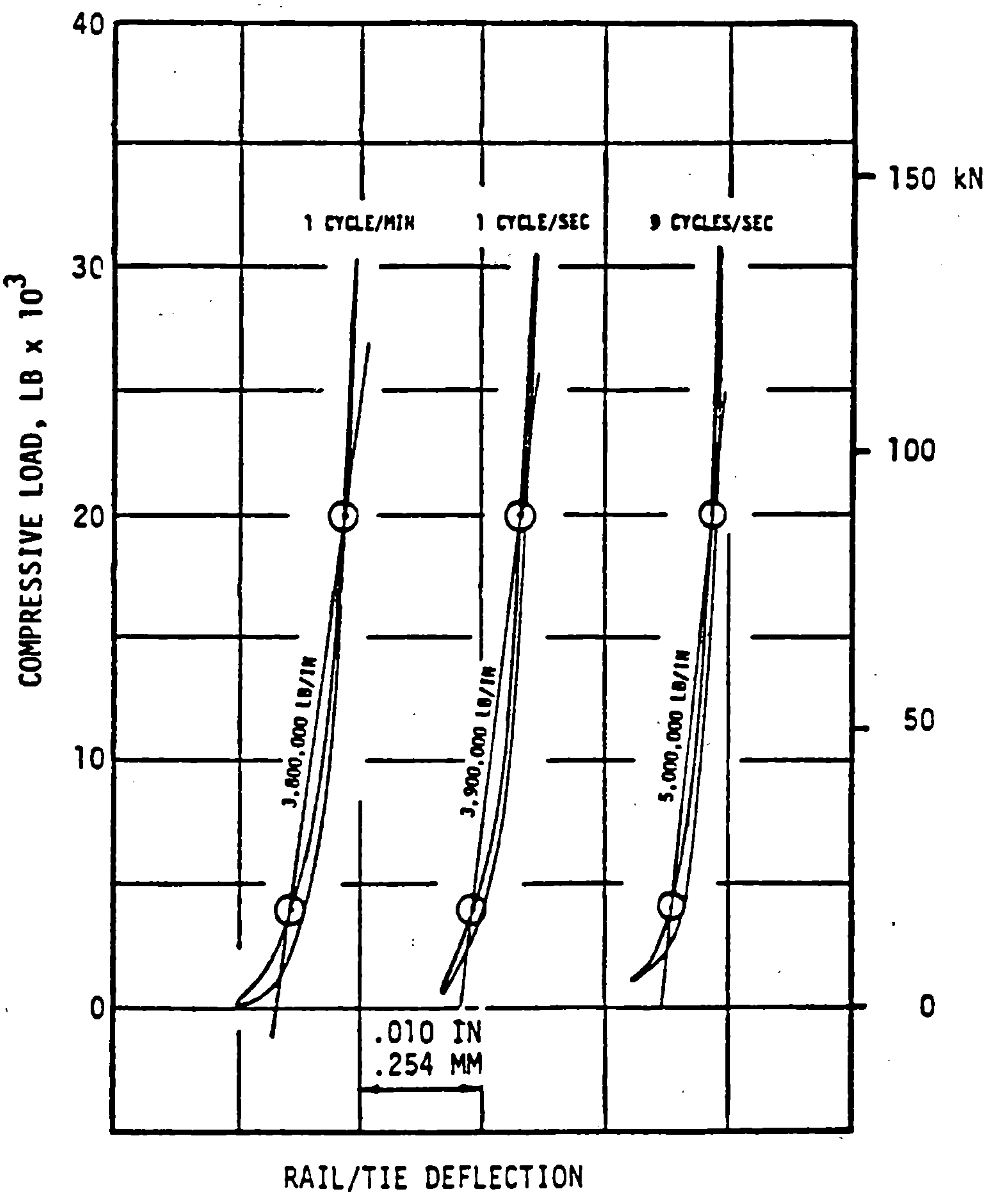


FIGURE H-14. RAIL/TIE FORCE/DEFLECTION CHARACTERISTICS FROM LABORATORY TESTS OF NEC CONCRETE TIE FASTENER AND RAIL SEAT PAD SYSTEM

TABLE H-3. MEASURED VALUES OF TRACK OVERALL STATIC VERTICAL STIFFNESS

Specific Track Conditions	K_z (lb/in)
	(static, per rail)
Wood Tie Track:	
Branch line, 100 lb/yd bolted-joint rail (BJR), poor to fair ties, cinder ballast (Note 1)	123,000 to 191,000*
Main line freight, 136 lb/yd continuous-welded rail (CWR), limestone ballast (Note 2)	267,000 to 298,000*
Main line freight, 133 lb/yd BJR (away from joint region), slag ballast (Note 3)	190,000 to 420,000*
Main line freight, 133 lb/yd CWR, slag ballast (Note 3)	260,000 to 520,000*
Main line freight, 133 lb/yd CWR, slag ballast, rock/sand subgrade (Note 4) -- summer	520,000 to 660,000
-- winter (frozen ballast)	650,000 to 920,000
NEC main line passenger/freight, 140 lb/yd CWR, traprock ballast (Note 1)	351,000 to 492,000*
FAST track (TTC), 136 lb/yd CWR, granite ballast, new track	249,000 to 307,000
FAST track (TTC), 136 lb/yd CWR, slag ballast, existing track	197,000 to 282,000
Concrete Tie Track:	
NEC main line passenger/freight (Aberdeen), 140 lb/yd CWR, traprock ballast (Note 2)	496,000*
Main line freight (Richmond), 132 lb/yd CWR, granite ballast (Note 2)	500,000 to 606,000*
FAST track (TTC), 136 lb/yd CWR, granite ballast, new track -- Zone I	506,000 to 550,000
Main line freight (Florida), 132 lb/yd CWR, rock ballast on top of old track, sand (Note 4)	853,000 to 3.1×10^6
JNR (Tokaido Shinkansen), 60 tonne/cm pad, 200 mm ballast on 25 mm "Ballastmat	511,000

Note 1 -- based on deflections between 15,000 and 30,000 lb vertical load
2 -- based on deflections between 18,000 and 32,000 lb vertical load
3 -- based on deflections between 11,000 and 17,000 lb vertical load
4 -- based on tangent stiffness at a 32,000 lb vertical load
* -- measurements under point load

$$V_z = K z^n \tag{6a}$$

$$V_z = n K [V_z/K]^{(n-1)/n} \tag{6b}$$

where

K and n are empirically-derived constants, and

K_z is the linear, tangent stiffness at the given wheel load, V_z.

From these two different track sections, both representing high-quality, well maintained wood-tie track with 133-lb/yd rail and slag ballast in southwestern desert country, the following average values were found:

Track Type	V_z (lb)	n	K	K_z (lb/in.)	U (lb/in.in.)
BJR*	8,000	1.6	$6.5(10)^5$	200,000	2,160
BJR*	32,000	1.6	$6.5(10)^5$	336,000	4,300
CWR	8,000	1.9	$1.4(10)^6$	230,000	2,600
CWR	32,000	1.9	$1.4(10)^6$	444,000	6,240

(Tangent static values at load V_z, per rail)

*Away from joint region.

From past experience, it appears sufficiently accurate to estimate the track vertical stiffness at a given wheel load by the relationship

$$K_z = K_{zo}[V_z/V_{zo}]^{0.5} \tag{7}$$

where

K_{zo} = the tangent stiffness determined at load V_{zo}.

To illustrate the use of Equation (7), the approximate secant stiffness on mainline concrete-tie track is given in Table H-3 as 496,000 lb/in. over the load range of 18,000 to 32,000 lb. Assuming F_{zo} is the average of these two load extremes, 25,000 lb, the track tangent stiffness under light and heavy wheel loads can be calculated:

$$K_z = 496{,}000 \ (8000/25000)^{0.5} = 281{,}000 \text{ lb/in., light wheel;}$$
$$K_z = 496{,}000 \ (32000/25000)^{0.5} = 561{,}000 \text{ lb/in., heavy wheel.}$$

H.4.1.3 Effects of Rail Joints

In the vertical plane, a rail joint produces a "soft spot" due to the reduction in rail bending rigidity at the joint. An evaluation of rail joints by Prause [4] showed that the ratio of stiffness at the joint to vertical stiffness well away from the joint ranges from 0.25 for a completely loose joint to 0.5 for the joint considered as a pinned connection. Tight joint bars (fishplates) add some additional bending rigidity to the joint, and values ranging from 0.50 to 0.75 have been used in the past to simulate joint effects.

The deflection shape of the rail under static wheel load at the joint can be approximated by a cosine function. This simple relationship has been used by British Rail in their analytical investigations of joint impact loads [13]. The "span" of the joint dip, in terms of the BOEF relationships (see Equations 1-4), is $5/\beta$, typically 6 to 9 ft on either side of the joint.

Even with a perfect rail surface geometry, the reduced stiffness at a joint results in a periodic geometry error input due to the greater vertical deflection under load in the vicinity of the joint.

H.4.1.4 Vertical Track Stiffness Variations due to Resurfacing

A series of physical measurements were taken on revenue track sites [33]. Vertical track stiffness measurements were made both before and after surfacing.

Vertical track stiffness was measured by loading both rails at one tie with a hydraulic jack system suspended beneath a loaded hopper car. Resulting deflections were measured with a theodolite, sighted on a machinist's scale attached to the rail directly beneath the applied load. Figure H-15 is a summary of the mean track vertical deflection to the point load application as determined by averaging the responses of 7 to 10 locations at each test site. Note that the consolidated stiffness of the four concrete tie test sections all fall in a small range of values to the left of the plot. The stiffness of the freshly-surfaced concrete tie track at Streator as well as the wood tie section at Streator both show more "slack action" at lower load levels resulting in overall lower secant stiffness; about one-half of that of the consolidated concrete tie sections. Two observations can be made from evaluating the results of the sites. (1) The tangent stiffness at maximum nominal wheel loads for a given track would not be substantially altered by surfacing which indicates that this stiffness is controlled by the portion of the track structure remaining consolidated. However, the secant stiffness which is influenced strongly by the amount of free play directly beneath the tie will strongly be affected by the loss of consolidation generated by surfacing, and (2) as shown in Figure H-16, the variability of track stiffness is strongly influenced by surfacing. That is, thoroughly consolidated track will generally show a very low variation of stiffness from one location to another. Whereas freshly surfaced track will show a large variation in support from one tie to the next. It can be assumed that the variability and support will contribute to an immediate deterioration in the initial surface established by the surfacing operation as the track begins to settle.

H.4.1.5 Influence of Adjacent Wheel Loads

There is some additional deflection under a given wheel load due to loading at adjacent wheels. This effect varies in magnitude according to the bending stiffness of the rail, the rail/tie local stiffness, and the tie/ballast/subgrade stiffness. These influences from adjacent wheel loads are illustrated in Figures H-17 and H-18 for both heavy freight car and locomotive axles, and for empty freight car axles. Tie plate vertical load is seen to peak directly under a wheel, and is reduced only 10 percent to 25 percent from this peak midway between pairs of axles. Vertical deflections show little variation from the maximum value as pairs of adjacent trucks pass over, particulary under lightly loaded or empty cars. The track in this example was 133 lb/yd CWR, good wood ties with heavy slag ballast, with a modulus of 8300

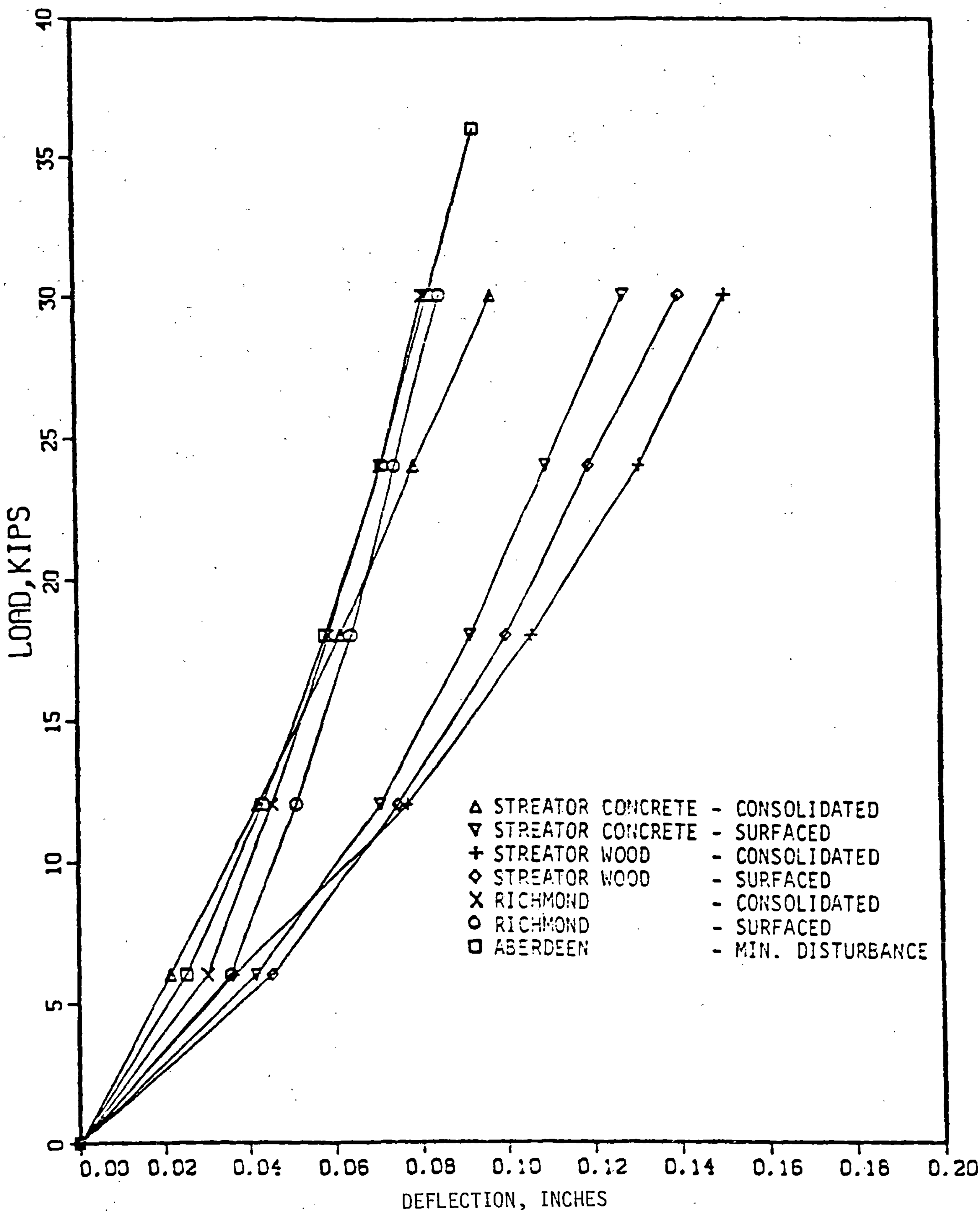
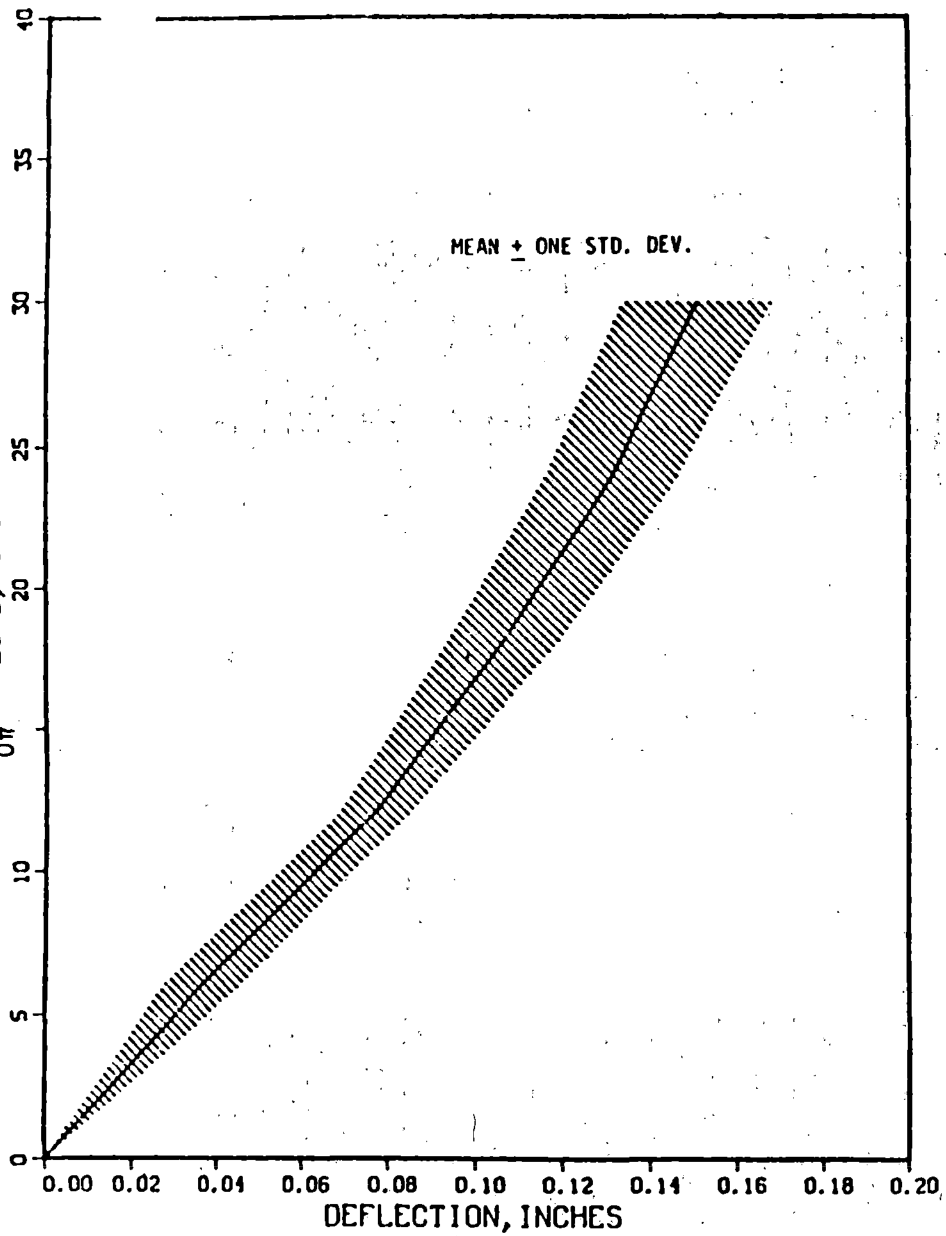
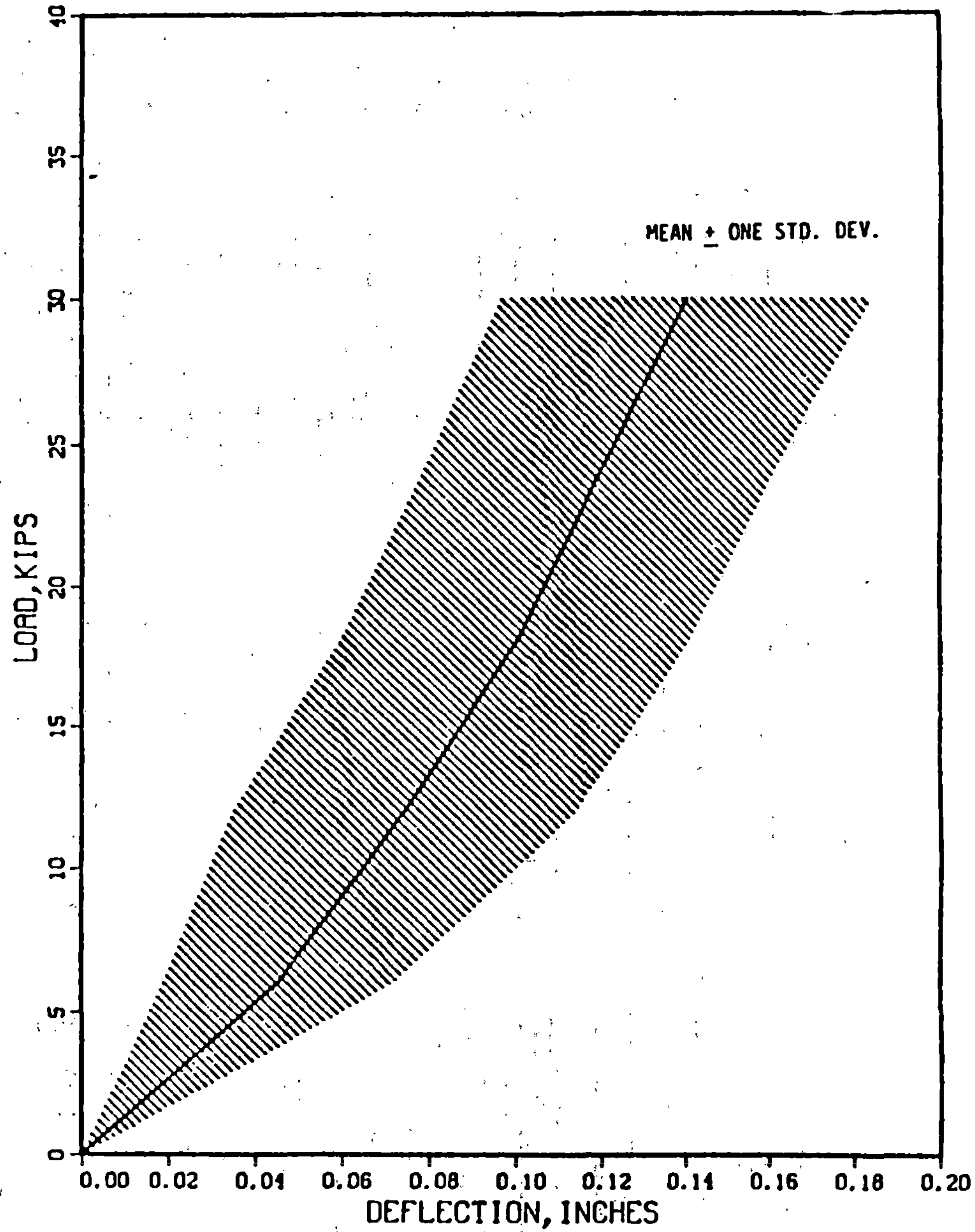


FIGURE H-15. MEAN TRACK VERTICAL RESPONSE TO POINT LOAD APPLICATION



TRACK VERTICAL RESPONSE TO POINT LOAD APPLICATION AT STREATOR-WOOD (CONSOLIDATED TRACK)



TRACK VERTICAL RESPONSE TO A POINT LOAD APPLICATION AT STREATOR-WOOD (AFTER SURFACING)

FIGURE H-16

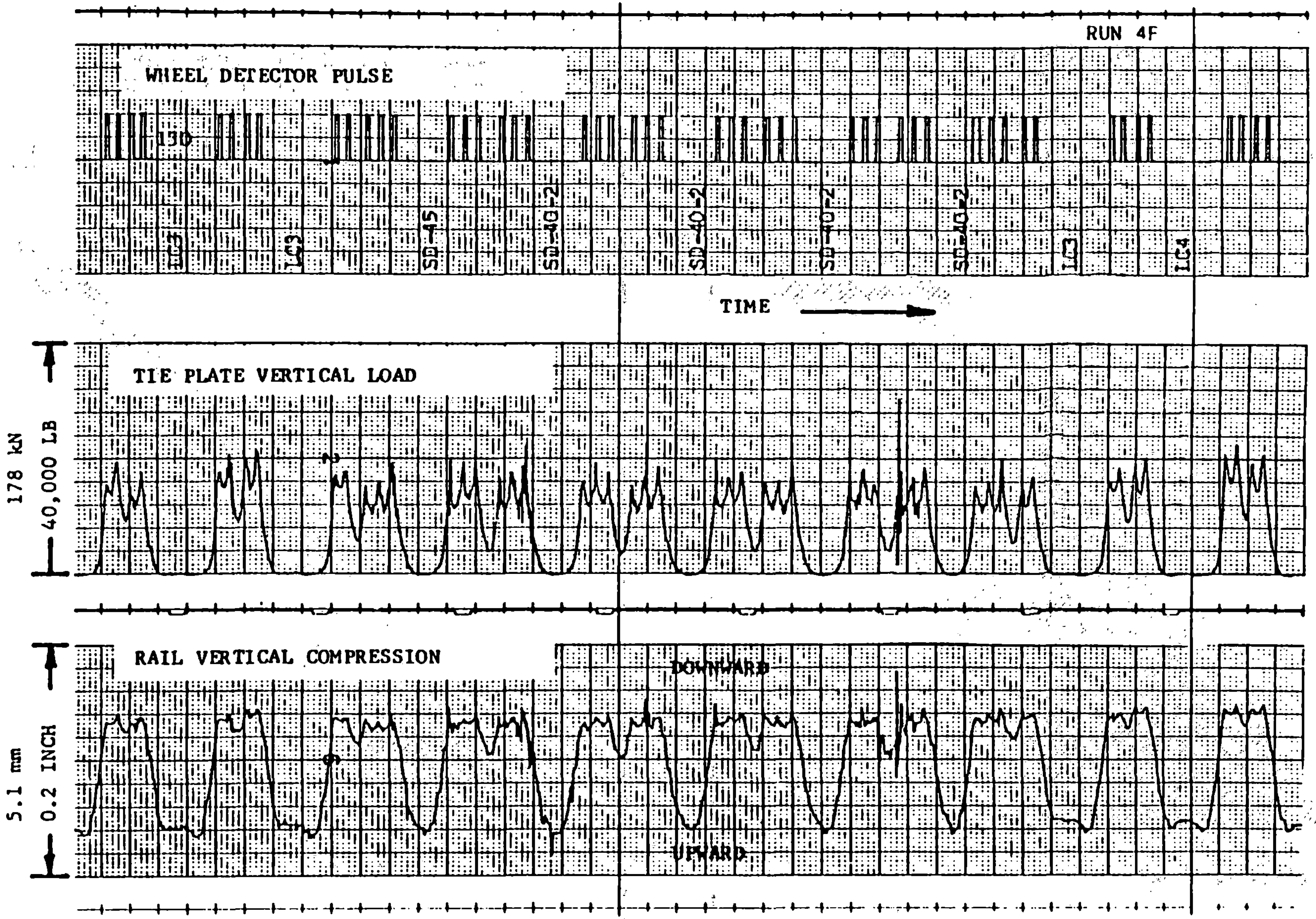


FIGURE H-17. TIE PLATE VERTICAL LOAD AND RAIL VERTICAL DEFLECTION UNDER HEAVY FREIGHT CARS AND LOCOMOTIVES OF UNIT TRAIN (TRAIN SPEED 50 MPH, 80 KPH)

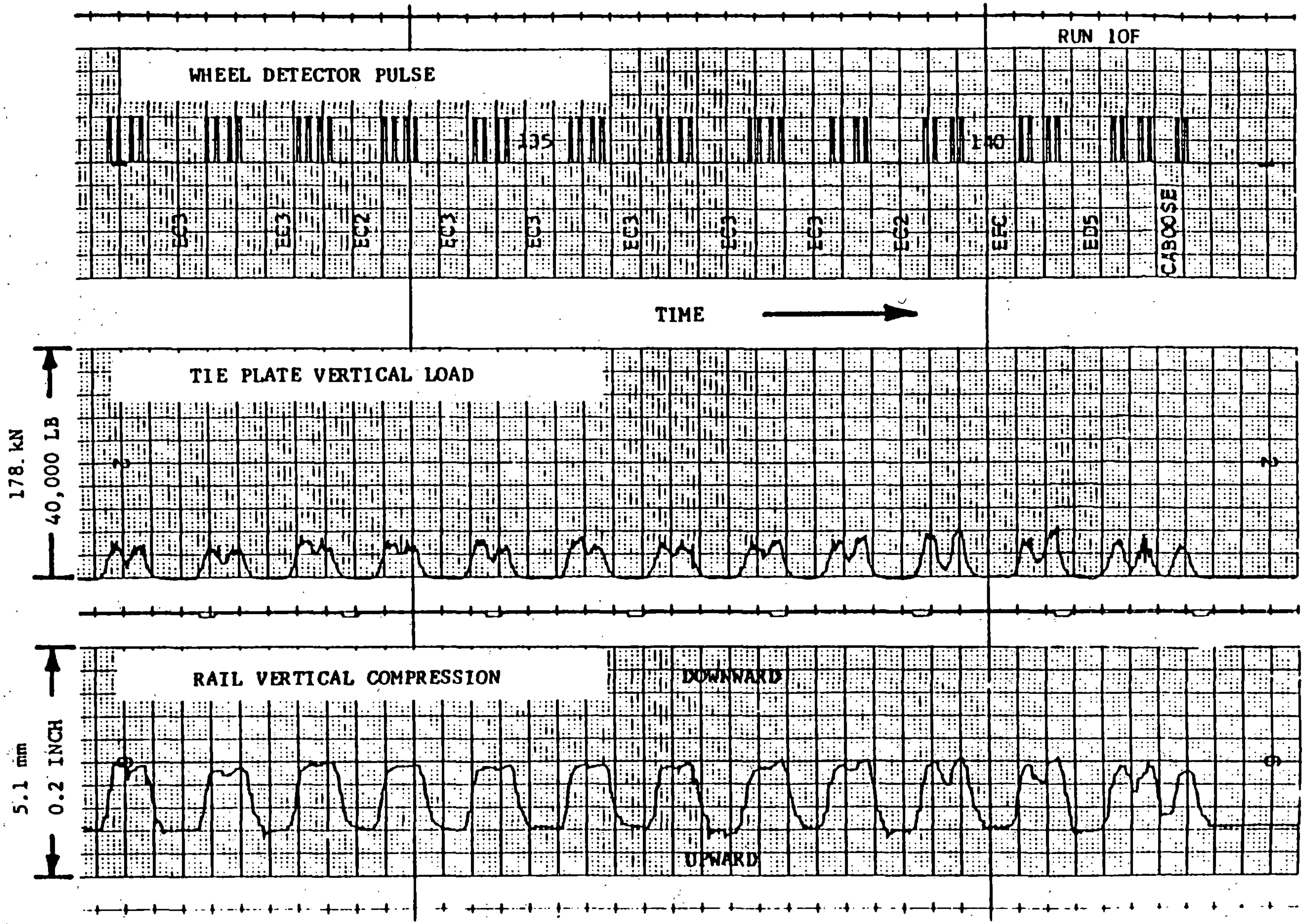


FIGURE H-18. TIE PLATE VERTICAL LOAD AND RAIL VERTICAL DEFLECTION UNDER EMPTY AND LIGHTLY-LOADED FREIGHT CARS (TRAIN SPEED 60 MPH, 97 KPH) lb/in./in., (K_z = 550,000 lb/in.) under a heavy wheel load, 3400 lb/in./in. (K_z = 280,000 lb/in.) under a light wheel load.

The vertical deflection under one wheel due to the load at an adjacent wheel can be estimated from linear BOEF relationships (see Equation 2). Added deflection (as a percentage of point load deflection) is plotted in Figure H-19 for three values of track modulus as a function of distance to the adjacent axle. Linear superposition is assumed. With the wider spacing of axles on passenger cars and two-axle locomotives, and with stiffer track, some negative additional deflection (uplift) can occur. The plots in Figure H-19 do not account for the nonlinear characteristics, particularly with lighter wheel loads, but can be used to estimate adjacent-axle effects under heavier wheel loads.

H.4.1.6 Track Vertical Dynamic Characteristics

Measurements of the dynamic compliance of railroad track were made as part of an investigation of methods for measuring track dynamic characteristics [14]. Different types of force excitation, such as sinusoidal, random, and pulse inputs, were used at preload levels from 2500 to 15,000 lb on the rail. This track consisted of 85 lb/yd rails on old wood ties in good condition, with and without tie plates, and on a cinder ballast. The tangent stiffness on the nonlinear force/deflection curve ranged from 158,000 to 258,000 lb/in. (U = 2070 to 3970) under a 15,000-lb preload. Typical results for vertical dynamic characteristics under a 15,000 lb preload showed:

- Resonant frequency: 30 to 45 Hz
- Effective mass: 2500 to 5500 lb mass per rail
- Damping: 15 to 45 percent of critical.

Dynamic stiffness values calculated from sinusoidal and random excitations were usually found to be much higher than stiffness values from pulse excitation. The pulse, however, is more representative of a passing wheel load, and these results are shown in Tables H-4 and H-5. From these test results, the track may be represented vertically by a single degree-of-freedom spring-mass-damper system over a bandwidth of 0 to 80 Hz with reasonable accuracy.

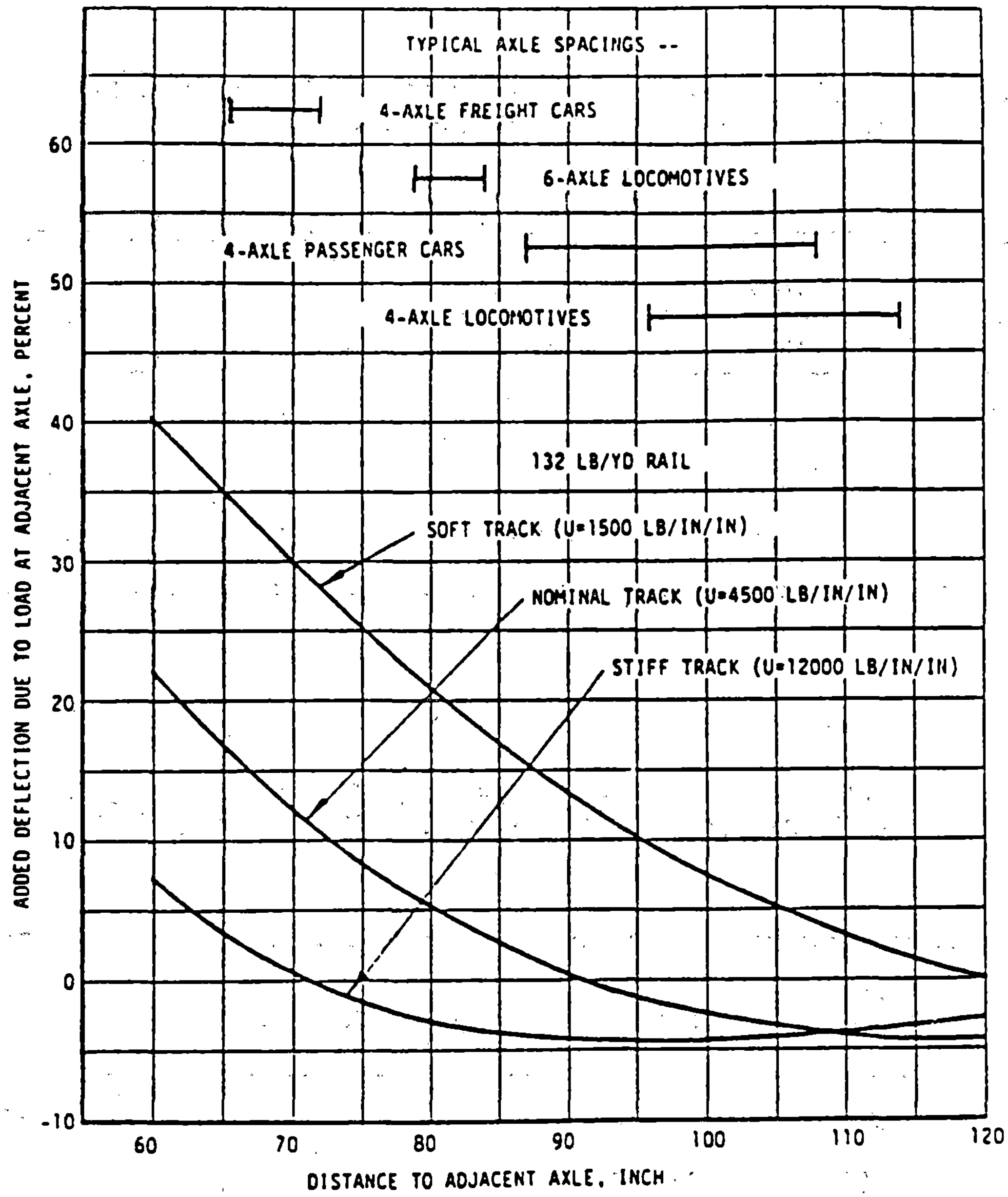


FIGURE H-19. VERTICAL DEFLECTION UNDER A WHEEL DUE TO VERTICAL LOAD UNDER ADJACENT WHEEL (SAME RAIL) -- LINEAR BEAM-ON-ELASTIC-FOUNDATION RELATIONSHIPS

TABLE H-4. TRACK VERTICAL DYNAMIC CHARACTERISTICS AS A FUNCTION OF VERTICAL PRELOAD LEVEL (REFERENCE 14)

Vertical Preload, lb	Vertical Stiffness K_z, lb/in.	Natural Frequency f_n, Hz	Damping, %	Mass M_z, lb
2500	19,000	18.0	18	573
7500	55,000	18.5	11	1571
15,000	213,000	21.0	31	4722

TABLE H-5. TRACK VERTICAL DYNAMIC CHARACTERISTICS AS A FUNCTION OF PULSE DURATION, 15,000-LB (67 kN) PRELOAD (REFERENCE 14)

Pulse Duration, ms	Peak Amplitude, lb	Dynamic Stiffness K_z, lb/in.	Natural Frequency f_n, Hz	Damping, %	Mass M_z, lb
10	3900	398,000	47	31	1760
15	5100	327,000	34	16	2765
20	6000	306,000	28	16	3816
25	6300	220,000	20	15	5378

Frequency analysis of wheel/rail loads measured under revenue freight traffic on mainline track [12] showed strong vertical response in the 41 to 46 Hz range on CWR, 32 to 35 Hz on BJR, with additional lower-amplitude spectral peaks at 49-57 Hz, 72-82 Hz, 110 Hz, 140 Hz, and 370 Hz. Because the track is a complex, distributed mass dynamic system, these other higher frequency modes of vibration occur. On concrete tie track, higher beam bending modes in the tie can be important. The rail itself will exhibit vibrational modes at 700-800 Hz and above.

H.4.2 Track Lateral Modulus

The track lateral support characteristics are considerably more complex than those in the vertical direction. The rail presents a non-symmetric beam in lateral bending, with both lateral and torsional stiffness characteristics. Also, two distinct loading modes are involved: the individual wheel (gauge spreading) loads, and the wheelset net lateral (panel shifting) loads.

The lateral stiffness of the track which is of primary importance for vehicle response is first defined at the wheel/rail lateral contact point. Up to the point of flange contact, the wheelset is weakly coupled to rail through the wheel/rail creep forces, which typically consist of a small spring rate (a few hundred lb/in. at most) and a relatively large speed-dependent damping term. Once the wheelset has flanged, however, the rail lateral stiffness and damping dominate. It is these terms that constitute the rail lateral support characteristics.

Track lateral force/deflection data have shown that the lateral stiffness is a highly nonlinear function of both the lateral and vertical load components on the rail, as well as the type and condition of the rail, ties, fasteners, and tie/ballast interactions. The track lateral stiffness consists of two different springs that act under different load paths as shown in Figure H-20: first, the gauge spreading stiffness (K_r), and second, the tie-to-ground stiffness (K_b). Both stiffnesses exhibit nonlinear rates dependent on the vertical load, plus hysteresis, and permanent deformation above certain load levels. The tie-to-ground stiffness (K_b) acts in series with the rail lateral/torsional stiffness (K_r), and results in additional deflection of the

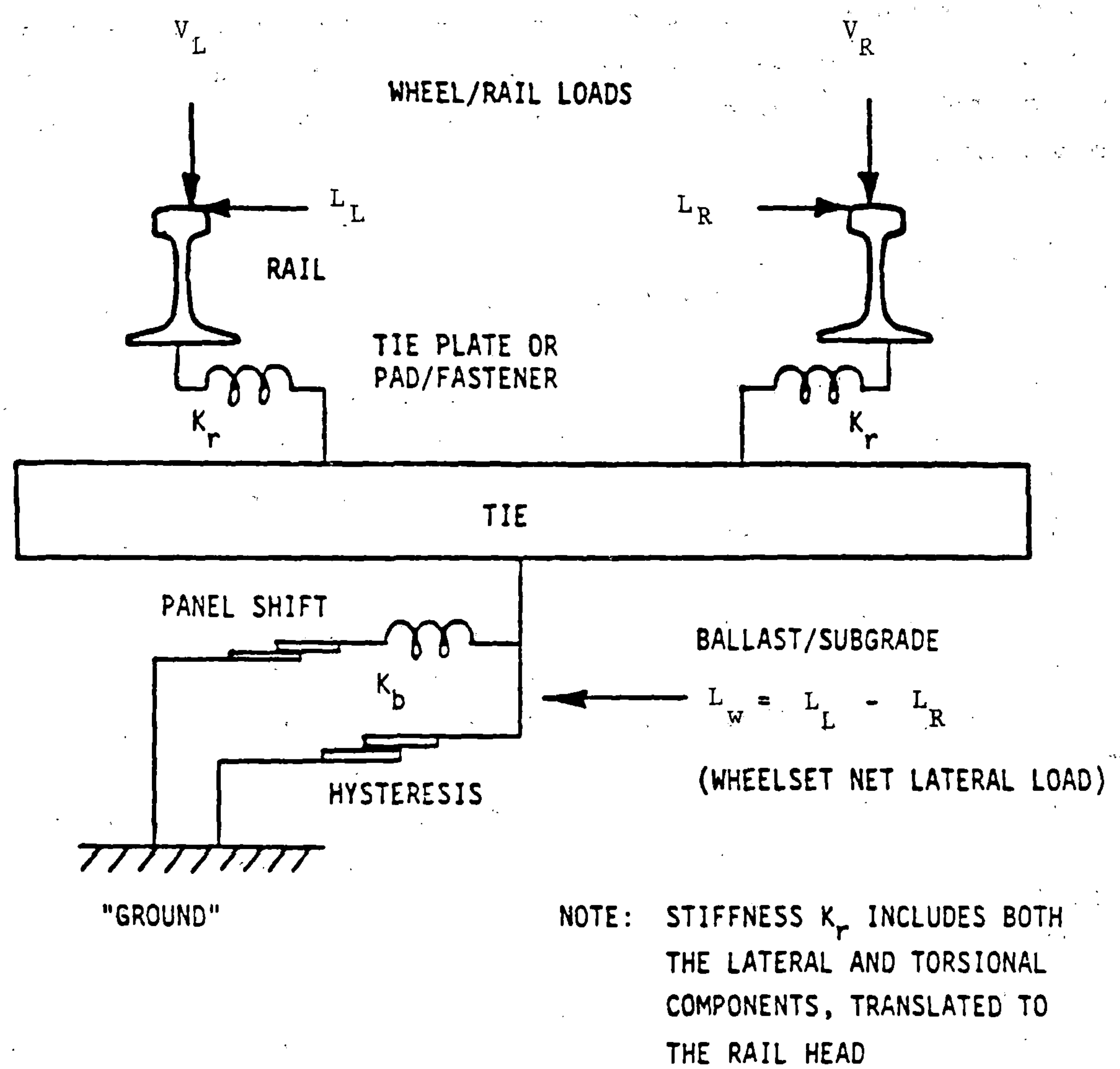


FIGURE H-20. SCHEMATIC OF TRACK LATERAL STIFFNESS CHARACTERISTICS track in response to the wheelset net lateral load (L_W); whereas the individual rail stiffnesses (K_r) respond to each wheel load and result in rail head deflections relative to the tie. Most available force/deflection data refer to the gauge-spreading (K_r) stiffnesses. Some methods of testing, such as the French (SNCF) "derailing car", will provide the series combination of K_r and K_b stiffnesses. Other tests have concentrated on the effective tie/ballast stiffness, K_b [15].

H.4.2.1 Rail/Tie Lateral Stiffness

Lateral stiffness characteristics of the rail head in response to a lateral gauge-spreading load applied between the rail heads depend strongly on the simultaneously applied vertical load. Typical force/deflection curves on good wood-tie track under different vertical load levels are shown in Figure H-21. The rail is seen to be laterally stiff up to an L/V ratio of about 0.33, but greater deflection occurs above this L/V ratio. At higher deflections, greater than about 0.4 inch, the rail again begins to stiffen. From tests on this high-quality wood-tie, cut-spike track [16], empirical equations were developed to describe the lateral force/deflection characteristics as a function of the vertical and lateral gauge-spreading forces:

$$L_G = (42{,}000 - 0.37V_z)y + 0.46V_z(1 - e^{-17.6y}) \qquad (8a)$$

$$K_r = 42{,}000 - 0.37V_z + 8.10V_z e^{-17.6y} \qquad (8b)$$

where

L_G and V_z are in pounds, y in inches, for a newly constructed track (spikes tightened).

A second section of this track was purposely weakened by removing the spikes from every other tie plate. Force/deflection measurements were made under different vertical load levels, and the following empirical equations were developed:

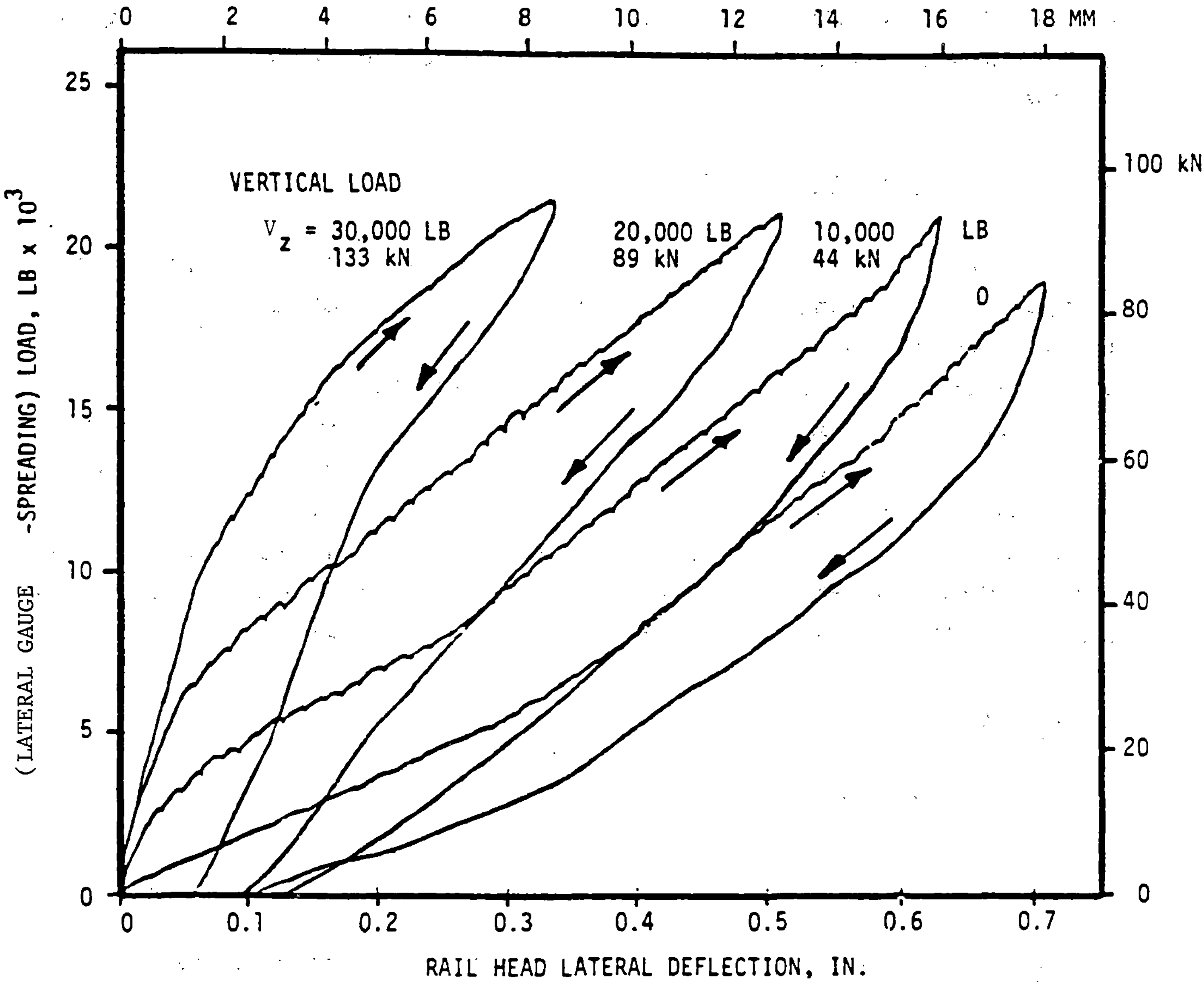


FIGURE H-21. TYPICAL LATERAL FORCE/DEFLECTION CHARACTERISTICS FOR DIFFERENT VERTICAL LOADS -- GOOD WOOD TIE, CUT SPIKE TRACK WITH 136 LB/YD (67 KG/M) RAIL

$$L_G = (24{,}000 + 0.10V_z)y + 0.40V_z(1 - e^{-15.4y}) \tag{9a}$$

$$K = 24{,}000 + 0.10V_z + 6.16V_z\, e^{-15.4y} \tag{9b}$$

Simple, bilinear representations of track lateral stiffness were developed to define the expected range of track structural characteristics [1]. These are shown in Figure H-22 for a 30,000-lb vertical wheel load, along with plots of the empirical representations given above. These linear curves are in the form:

$$y = L_G/K_{r1} \text{ for } L_G/V_z < (L/V)_b \tag{10a}$$

$$y_b = V_z(L/V)_b/K_{r1} \text{ at } L_G/V_z = (L/V)_b \tag{10b}$$

$$y = [L_G - V_z(L/V)_b]/K_{r2} + y_b \text{ for } L_G/V_z > (L/V)_b \tag{10c}$$

where

L_G = lateral wheel/rail force,
V_z = vertical wheel/rail force,
y = rail head lateral deflection.

Based on the review of available data, the following track stiffness characteristics were chosen to represent the nominal and extreme track conditions:

Track Condition	K_{r1}(lb/in.)	(L/V)b	K_{r2}(lb/in.)
Stiffest realistic track construction	$100{,}000 + 15V_z$	0.75	100,000
Nominal wood-tie/cut-spike track	$40{,}000 + 5V_z$	0.33	40,000
Minimally weak track	$15{,}000 + 2V_z$	0.33	15,000

H.4.2.2 Tie/Ballast Lateral Stiffness

Track (rail head) lateral stiffness characteristics from tests where a gauge-spreading load is applied do not include the tie-to-ballast compliance.

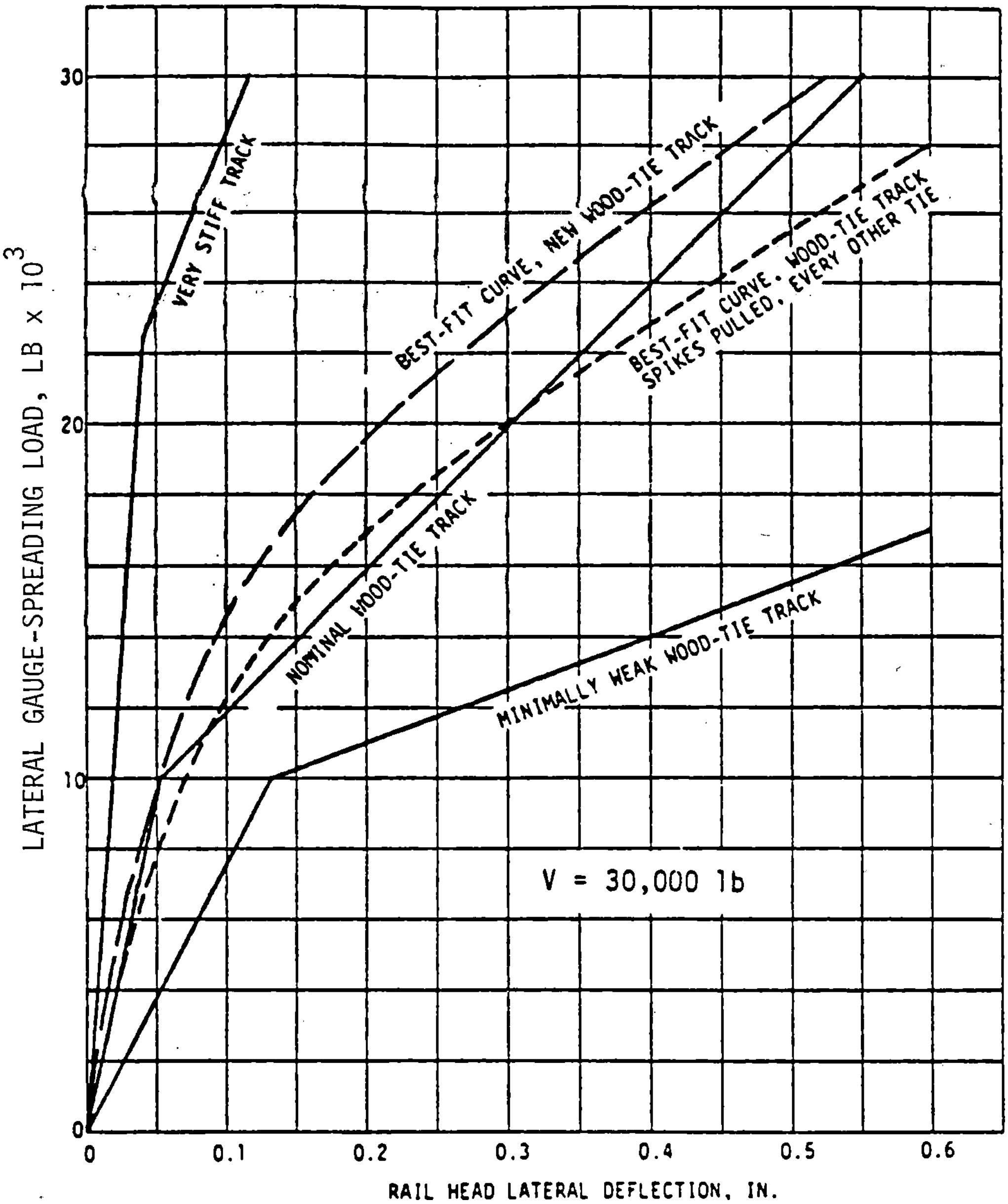


FIGURE H-22. BILINEAR REPRESENTATION OF TRACK LATERAL FORCE VERSUS DEFLECTION CHARACTERISTICS COMPARED WITH BEST-FIT CURVES OF NOMINAL WOOD-TIE TRACK

In curving, both the leading outer and inner wheels develop high outward (gauge-spreading) loads, with a relatively small net lateral load on the wheelset. Under dynamic conditions such as the vehicle response to a line error in tangent track, or during truck hunting, loads may occur predominantly on one wheel, and a high net lateral load is developed that passes through the tie to the ballast.

Tests of track lateral stability by the SNCF [15] using their "Derailing Wagon" car show typical nonlinear force/deflection curves. For concrete sleepers, 60 Kg/m (132 lb/yd) rail and well-consolidated ballast, deflections at the tie of 0.5 mm for L_W = 15,000 lb, 1.0 mm for L_W = 19,000 to 22,000 lb under a vertical load V = 21,500 lb were measured. These deflections indicate a track lateral tie/ballast stiffness (K_b in Figure H-20) of 480,000 to 760,000 lb/in. for the track panel. Similar tests in the Netherlands with light rail (46kg/m) and hardwood ties on gravel ballast showed a lateral stiffness of 430,000 lb/in. with a standard deviation of 100,000 lb/in. for 20 samples, under a vertical load of 24,000 lb per rail [17].

H.4.2.3 Effects of Adjacent Wheel Loads

The apparent lateral stiffness of the rail is strongly influenced by both the vertical and lateral loads at adjacent wheels. These effects have been explored by the Canadian National [18] in tests that applied two vertical loads at a 66-inch spacing, and a single lateral load at one of the two vertical load points. Results from the Vehicle/Track Interaction Tests [19] showed clearly the effects of adjacent vertical loads on apparent track stiffness. In Figure H-23, stiffness curves are plotted by linear regression analysis on load versus deflection data for leading wheels, first from the locomotive with the nearest adjacent wheel 108 inches away, and second from the 100-ton hopper cars, with an adjacent wheel 70 to 72 inches away, and (for half the axles) a second adjacent wheel approximately 80 inches away. Lateral forces at these adjacent wheels are negligibly small, so that effects on apparent stiffness were due primarily to the vertical loads. An apparent stiffness 33 percent higher is calculated under the hopper car wheels than under the more widely-separated locomotive wheels. Results from different measurement

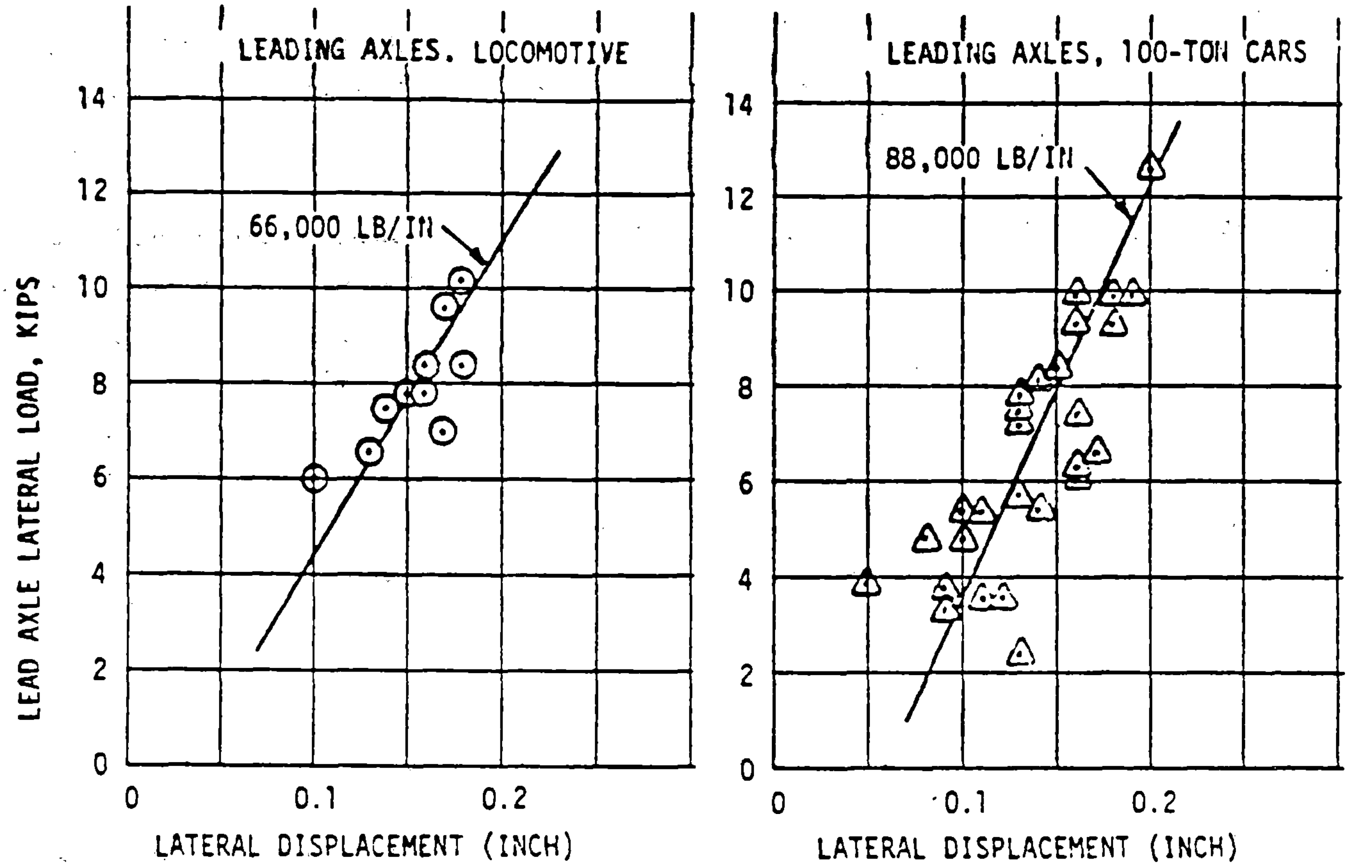


FIGURE H-23. EXAMPLE OF EFFECT OF ADJACENT VERTICAL WHEEL LOADS ON APPARENT LATERAL STIFFNESS (SOFT* TRACK SECTION, V/T INTERACTION TESTS [19])

* Note: "soft" section in as-found condition, many ties in poor condition, spikes loose or missing

locations are given in Table H-6 for this branch line track, having a vertical modulus ranging from 1350 to 2420 lb/in./in.

The effects of adjacent lateral wheel loads are less clearly defined from the available field test data. From the Vehicle/Track Interaction Tests [19], lateral deflections under trailing locomotive wheelsets were typically 40 to 70 percent of the deflection under the leading wheelset, even though the lateral loads were small, or even inwardly-directed. Under trailing hopper car wheelsets, lateral deflections ranged from 70 to 90 percent of the deflection under the leading wheelset. Lateral loads were for the most part negligibly small under the trailing wheelset.

Track lateral deflections due to a two-point (5 ft apart) lateral loading are plotted in Figure H-24 from FAST track experiments [20]. On concrete tie track, the deflection due to a single load is approximately 50 percent at 30 inches from the load, and nearly zero at 60 inches from a single load: the influence zone is substantially longer.

The BOEF formulae cannot be used even to approximate the deflections under one wheel due to loads at another, because of the complex lateral and torsional bending effects. More sophisticated means, such as finite element modeling, must be used; or else estimates from field experiment data must be made.

H.4.2.4 Effects of Rail Joints

Variations in lateral stiffness also occur at a rail joint. Unlike the vertical stiffness, however, the rail lateral stiffness is substantially increased by the higher lateral bending rigidity at the joint due to the joint bars. Tests were conducted on two rail joints during the Perturbed Track Tests [16]. Rail lateral stiffness at the joint, (the track as built was 136 lb/yd rail, wood ties on 19.5-inch centers, and 4 spikes per tie plate), was approximately 2 times stiffer than the mid-rail location under the different vertical loads, up to an L/V ratio of roughly 0.40. Above this L/V ratio, the lateral stiffness was comparable for both the joint and the mid-rail locations. In Section 3.5, where the two hold-down spikes per plate were removed, and all spikes on every other tie were removed to simulate weak track, the

TABLE H-6 EFFECTS OF ADJACENT AXLES AND AXLE SPACING ON APPARENT RAIL LATERAL STIFFNESS (VEHICLE/TRACK INTERACTION TESTS) HIGH RAIL OF 5-DEGREE CURVE

	Runs of May 28 (no shims) Axles	Site 4H V	Site 4H L	Site 4H Slope*	Site 4H Corr	Site 7H V	Site 7H L	Site 7H Slope*	Site 7H Corr	Site 10H V	Site 10H L	Site 10H Slope*	Site 10H Corr
Section 1.3 ("Soft")	Locomotive, Axles 1 & 3 (Lead Axles)	32.8 ±1.8	8.0 ±1.3	52.2	0.80	35.5 ±2.8	9.0 ±2.7	72.6	0.83	33.1 ±2.0	4.1 ±0.1	40.6	0.66
	Hopper Car, Axles 11, 15, 19 (Lead Axles, 1 adj.)	33.0 ±4.7	6.2 ±2.5	70.6	0.87	33.3 ±3.9	7.9 ±2.8	88.5	0.74	32.8 ±4.4	5.9 ±2.0	100.5	0.84
	Hopper Car, Axles 13, 17 (Lead Axles, 2 adjacent)	33.9 ±3.2	7.4 ±2.8	130.1	0.78	33.1 ±3.2	7.7 ±2.3	92.4	0.59	31.0 ±4.3	4.6 ±1.2	75.6	0.90
	Hopper Car, Axle 13 (AAR 1:20 profile)	34.7 ±2.7	9.5 ±2.0	69.6	0.90	34.0 ±3.3	8.9 ±2.0	63.3	0.69	32.0 ±4.8	4.7 ±0.9	76.3	0.90
	Hopper Car, Axle 17 (CN worn profile)	33.0 ±3.8	5.2 ±1.4	65.4	0.52	32.2 ±3.1	6.4 ±2.0	115.3	0.43	29.9 ±4.0	4.5 ±1.6	75.0	0.91
Section 2.4 ("Stiff")	Locomotive, Axles 1 & 3 (Lead Axles)	32.7 ±4.3	8.7 ±1.5	172.0	0.63	35.0 ±6.0	7.3 ±1.9	59.1	0.89	35.2 ±4.3	8.7 ±1.3	150.0	0.82
	Hopper Car, Axles 11, 15 19 (Lead Axles, 1 adj.)	34.1 ±5.0	8.3 ±2.8	147.5	0.88	32.8 ±5.4	8.5 ±1.7	64.1	0.92	36.3 ±6.3	12.6 ±2.8	151.0	0.79
	Hopper Car, Axles 13, 17 (Lead Axles, 2 adjacent)	34.3 ±5.5	10.0 ±2.5	165.5	0.75	31.2 ±6.0	7.4 ±1.6	123.9	0.33	33.7 ±7.5	6.8 ±2.6	299.5	0.32
	Hopper Car, Axle 13 (AAR 1:20 profile)	33.7 ±4.5	11.6 ±2.5	147.5	0.98	33.8 ±3.7	8.3 ±1.4	58.8	0.94	34.4 ±6.7	9.0 ±1.6	155.7	0.48
	Hopper Car, Axle 17 (CN worn profile	30.6 ±6.1	8.4 ±1.3	80.9	0.84	36.1 ±8.1	6.4 ±1.3	94.6	0.36	33.0 ±8.9	4.6 ±0.6	*****	****
		kips	kips	kips/in		kips	kips	kips/in		kips	kips	kips/in	

* Average of slopes, linear regression analysis with L as "x", then L as "y". ***** = neg. values
Locomotive axle spacing = 108"; hopper car 1 adj. = 70", 2nd adj. = 80".

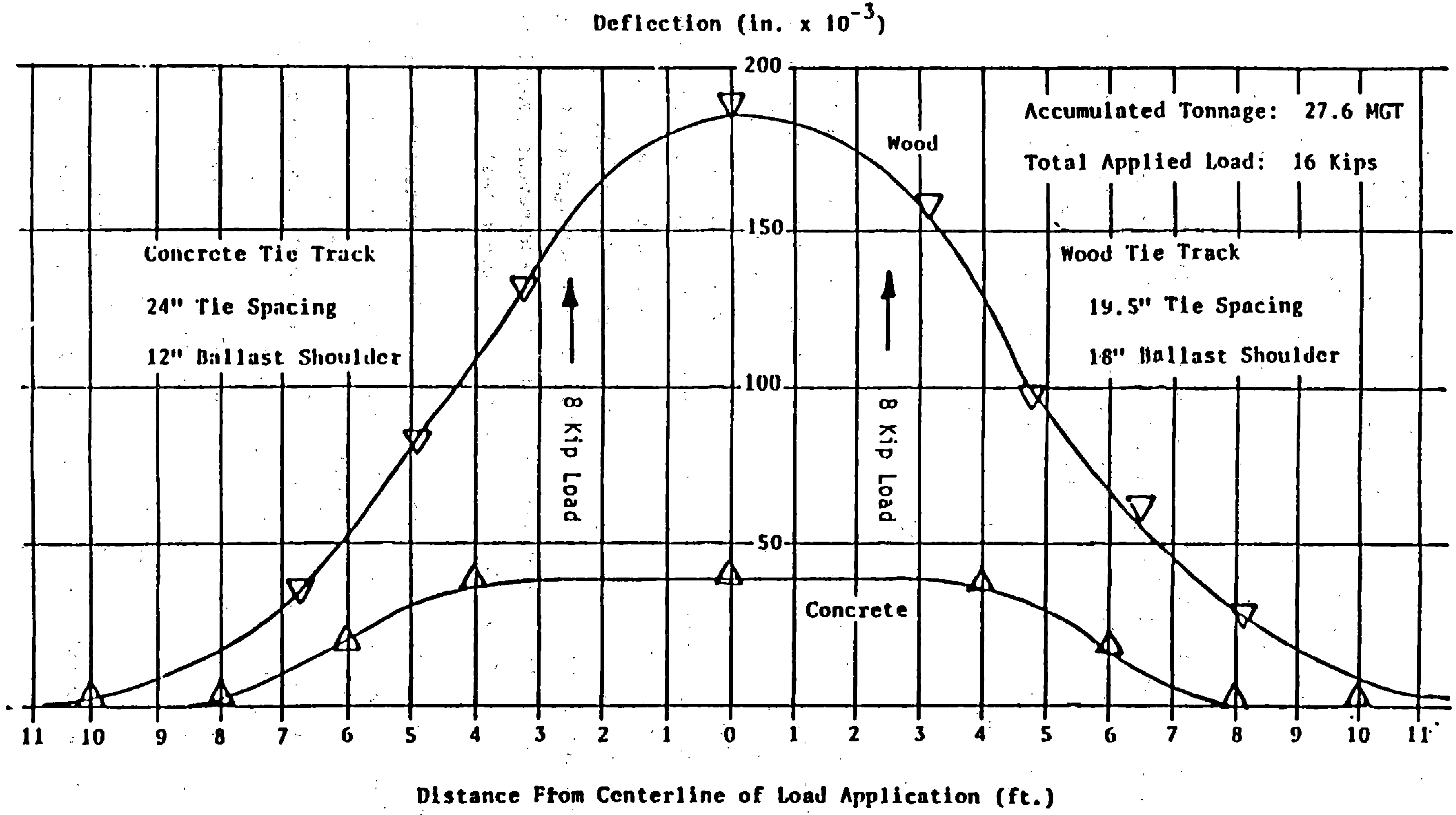


FIGURE H-24. TRACK LATERAL DEFLECTION DUE TO TWO-POINT LATERAL LOAD APPLIED TO BASE OF RAIL (NO VERTICAL LOAD) [20] rail lateral stiffness at the joint was approximately 1.5 times the mid-rail stiffness up to an L/V ratio of roughly 0.45, and almost 2 times for higher L/V ratios. While this sample is most certainly limited in number, the results indicate the higher stiffness at the joint and the possibility of significant variation along the track due to rail spiking and tie conditions.

Since the vehicle wheelset is weakly coupled laterally to the rail except when in hard flange contact, the change in stiffness in the joint region would not induce a significant force input to the wheelset. On curves where the lead outer wheel flange is in contact, the increase in stiffness at the joint would cause a periodic variation in lateral force, but this is predicted to be minor relative to the basic curving forces.

H.4.2.5 Effects of Load Contact Position

An additional factor affecting the rail apparent lateral stiffness is the lateral position on the rail running surface of the wheel/rail contact patch. This directly affects the moment into the rail due to the vertical load vector. It can also affect the moment due to lateral load: up to flange contact, the lateral load is introduced at the running surface height; but after flange contact, the effective height of the lateral load vector is indeterminant, part passing through the tread contact patch, part through the flange. Although load position effects have not been thoroughly investigated, they are thought to contribute to the wide scatter in deflection measurement (see Figure H-23, for example, which is a small population of vehicle axles).

A preliminary investigation of these effects has been conducted by W.T. So as part of this program, using a finite-element model of the Northeast Corridor concrete-tie track. Results from the model are shown in Figure H-25. By moving the vertical load contact point from the rail centerline to a point 0.5 inch toward the gauge side, the apparent lateral stiffness of the rail at the same L/V ratio is nearly doubled. From the work of Cooperrider and Law [21], the lateral position of the wheel/rail contact "point" can vary in excess of ± 1 inch, as shown in Figure H-26. This can easily be confirmed by checking the rail running surface on tangent track after passage of a long freight train. For any particular wheel and rail profile, however, the posi-

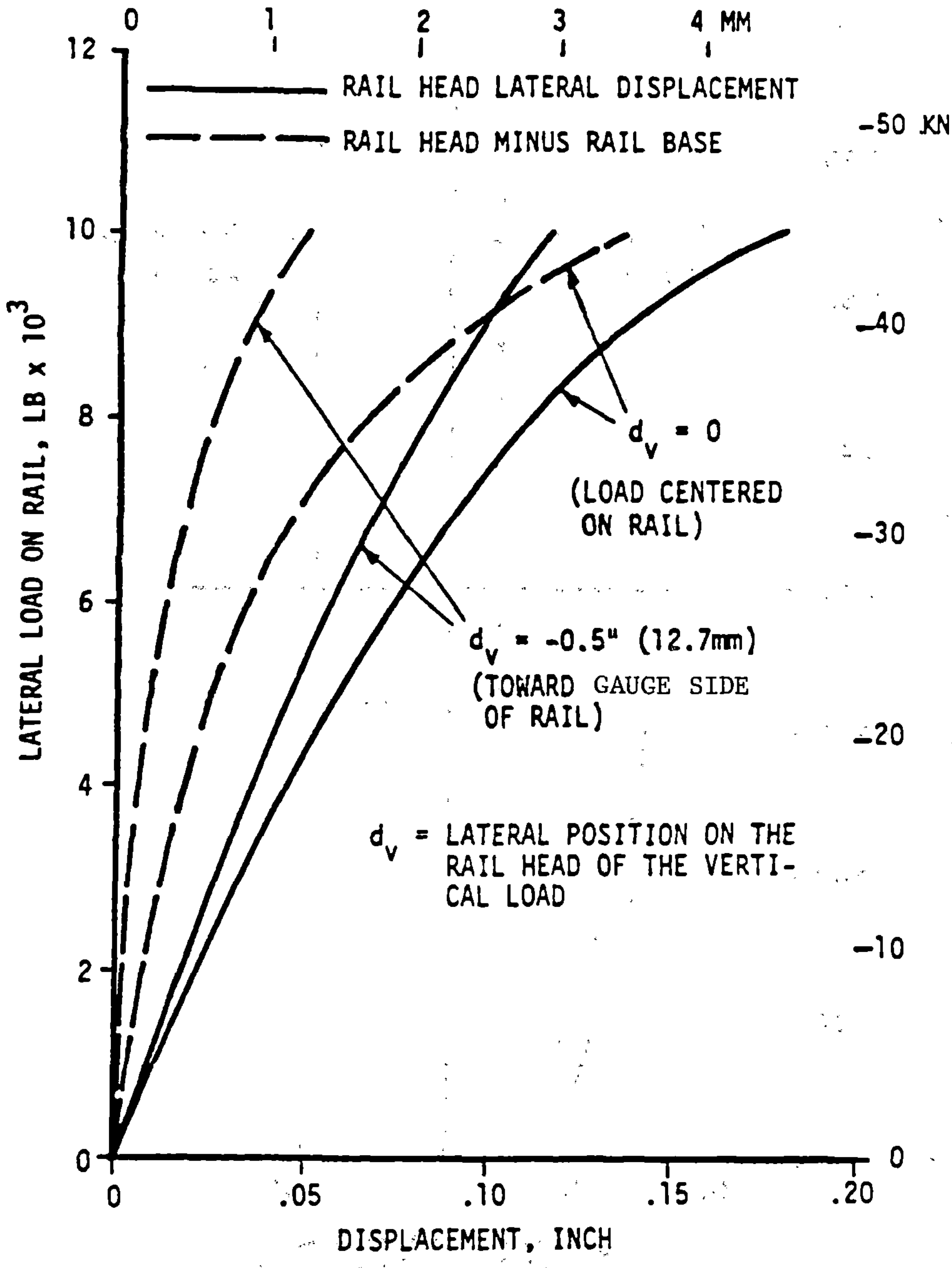


FIGURE H-25. RAIL LATERAL DISPLACEMENTS PREDICTED FOR NORTHEAST CORRIDOR CONCRETE TIE TRACK UNDER 30,000 LB (133 KN) VERTICAL LOAD AND VARIED LATERAL LOAD -- FINITE ELEMENT MODEL

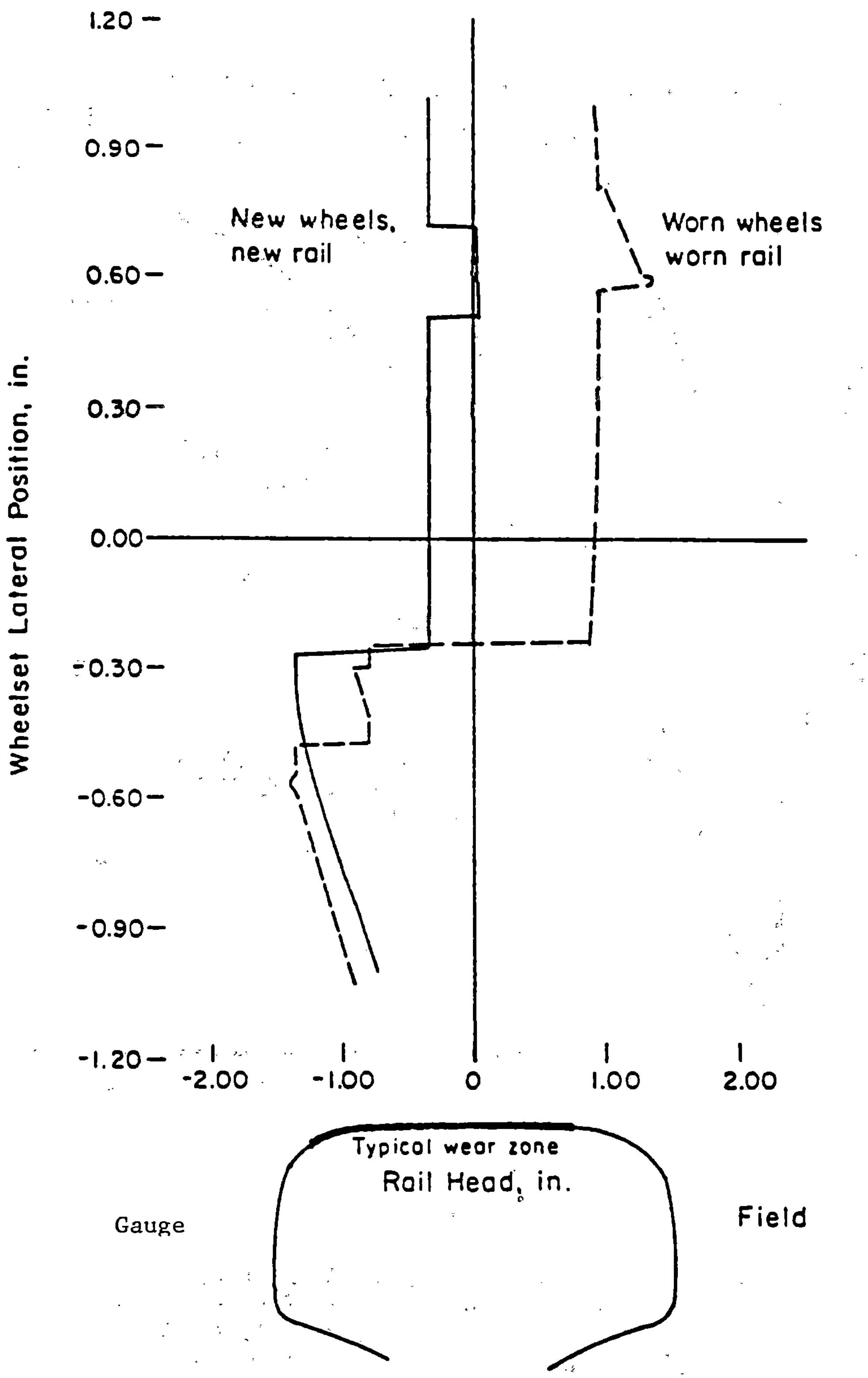


FIGURE H-26. VARIATION IN WHEEL/RAIL CONTACT POSITION ON RAIL RUNNING SURFACE AS A FUNCTION OF WHEELSET LATERAL POSITION AND WHEEL/RAIL PROFILE tion remains fairly constant over most of the wheelset lateral displacement, relative to the rail head, and a constant stiffness characteristic can be assumed for modeling a given wheelset.

H.4.2.6 Lateral Dynamic Characteristics

Track lateral dynamic compliance has also been measured [14] using sinusoidal, random and pulse force excitation through servo-controlled hydraulic actuators and several levels of vertical preload. Results are given in Table H-7 for this particular track, a little-used branch line with 85 lb/yd (42 kg/m) rail on old wood ties in good condition, single-shoulder tie plates and cinder ballast. Both random and pulse force inputs are compared. As expected, the effective stiffness in the lateral direction increased dramatically with higher vertical preload levels. The effective mass in the lateral direction was substantially lower than the vertical mass, ranging from 300 to 600 lb (136 to 272 kg) for this branchline wood tie track structure. Concrete tie track will, of course, have a much higher effective mass in the lateral direction due to the much greater tie weight. On wood-tie track, however, the wheelset mass will dominate over the track mass, and the track can be adequately simulated in the lateral direction by a nonlinear stiffness and damping characteristic.

TABLE H-7. COMPARISON OF LATERAL STIFFNESS PARAMETERS OBTAINED USING RANDOM AND PULSE FORCE EXCITATION

	Vertical Preload, lb	Lateral Stiffness, lb/in.	Natural Frequency, Hz	Damping, percent
Random Excitation	2500	13,000	17.5	45.0
	5000	73,000	50.0	37.0
	15,000	250,000	90.0	31.0
Pulse Excitation	5000	38,000	30.0	37.5
	15,000	200,000	72.0	11.0

H.4.2.7 Variations in Lateral Dynamic Stiffness due to Respiking Track

A specially-prepared track section with alignment and cross level errors was utilized during AEM-7 locomotive evaluation tests on the Northeast Corridor to provide a controlled transient geometry disturbance [27]. Both wayside wheel/rail load and deflection transducers and instrumented wheelsets were used during these tests to evaluate track loads and L/V ratios. Tests were conducted at speeds up to 110 mi/h to investigate operation over track geometry errors.

Standard vertical and lateral load-measuring strain gauge patterns were used at the wayside site. Two pairs of lateral displacement transducers were used to measure rail head and rail base deflections.

Installation of the track perturbation caused some disturbance of normal track strength due to respiking the rail. The rail exhibited about 0.030 to 0.060 inch lateral motion under vertical load alone; and under lateral load, the rail head deflection was approximately twice the rail base deflection. The vertically shimmed rail showed little lateral deflection under vertical load, but rail head and base lateral deflections were roughly equal under lateral load to about 0.1 inch. Minor respiking was performed before each day's testing, and there was evidence in loads and deflections of the track loosening up under the higher-speed test runs and high lateral loads. Extensive realignment and respiking was done before the last day of testing, and results of the data analysis show strong evidence of the ties being spike-killed.

The variation in rail head lateral deflection due to respiking as a function of both vertical and lateral loads is shown in Figure H-27.

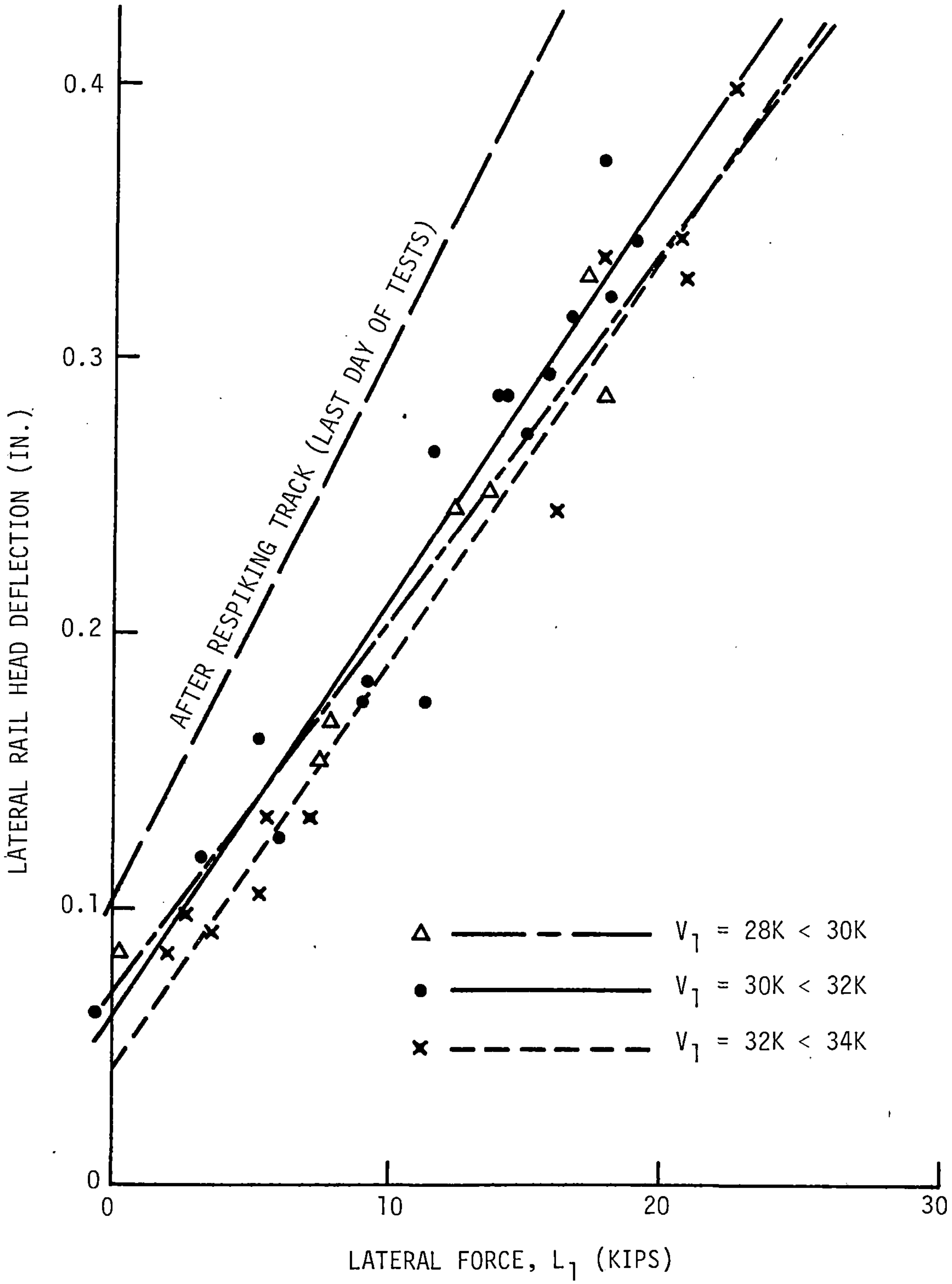


FIGURE H-27. RAIL HEAD LATERAL DEFLECTIONS VERSUS LATERAL FORCE FOR THE LEAD AXLES OF AEM-7 LOCOMOTIVE OVER VARYING SPEED RANGES WITH TRACK GEOMETRY ERRORS

H.5 COMPUTER SIMULATION OF VERTICAL/LATERAL TRACK STIFFNESS

Both the lateral and vertical track modulus may be simulated in a rail vehicle model to provide more realistic response predictions. Different levels of complexity of track modulus simulation may be employed, depending on the purpose of the model. For example, a vehicle model aimed at ride comfort studies may include track modulus as a simple spring/damper, while a model aimed at the study of impact loads would require several dynamic degrees of freedom in the track modulus portion, or even a finite-element or Fourier series representation of the track, to provide meaningful results. Different vehicle/track models are used in the following subsections to illustrate the modeling of track vertical and lateral stiffness in specific cases as they were handled for this study.

H.5.1 Simulation of Vertical Modulus

The vertical modulus of the track can be modeled at several levels of complexity, depending on the purpose of the vehicle/track model and the desired accuracy and bandwidth required for this purpose. For example, a vehicle model aimed primarily at ride comfort studies can be restricted to a bandwidth of 50 Hz, and the track overall vertical stiffness can be used, with the track effective mass lumped with the wheelset unsprung mass. A model to study impact loads due to rail joints or wheel flats, on the other hand, will require a bandwidth in excess of 1000 Hz, and individual track structural elements must be modeled with their appropriate masses and interconnecting stiffnesses.

H.5.1.1 Linear Random-Vibrations Vehicle-Track Model

A linear freqency-domain model, Program TRKVPSD, has been used in several forms to explore rail vehicle ride quality and wheel/rail forces in response to random track geometry inputs [22]. This program incorporates separate vertical/pitch and roll/yaw/lateral models to calculate response to average surface or average alignment and cross level geometry errors. Power spectral density response is calculated by the generalized relationship:

$$P_o(\omega) = |H^2| \; P_i(\omega) \qquad (11)$$

where

P_i = input (track geometry) power spectrum,

P_o = vehicle response power spectrum,

H = system transfer function,

ω = frequency, rad/sec.

In early versions of the program, each truck was represented as a two mass system: a sprung mass (the truck frame), and an unsprung mass (the wheelset mass and the truck effective mass for two or three wheelsets per truck). In the later versions, individual wheelset unsprung masses were included, along with the pitch torsional effects. By proper manipulation of the equations of motion, the track geometry input is applied at the true wheel/rail interface, as shown below and in Figure H-28. Assuming the wheel/rail load is at this interface, and that the model frequency bandwidth is much less than the wheel/rail resonant frequency:

$$f << \frac{1}{2\pi} \sqrt{K_H/M_{tr}} \quad ,$$

where

K_H = Hertzian contact stiffness.

$$Z_a = Z_r + e_z, \; Z_a^* = Z_r^* + e_z^* \qquad \text{(see Figure H-28)} \qquad (12)$$

$$M_{tr}\ddot{Z}_r + K_{zr}Z_r + V_{wz} - \lambda_v K_{zr} Z_r^* = 0 \qquad (13)$$

$$M_{tr}\ddot{Z}_r^* + K_{zr}Z_r^* + V_{wz}^* - \lambda_v K_{zr} Z_r = 0 \qquad (14)$$

where

$$\lambda_v = e^{-\beta L_a} [\mathrm{Cos}\beta L_a + \mathrm{Sin}\beta L_a] \qquad \text{(see Equations 1-4)}$$

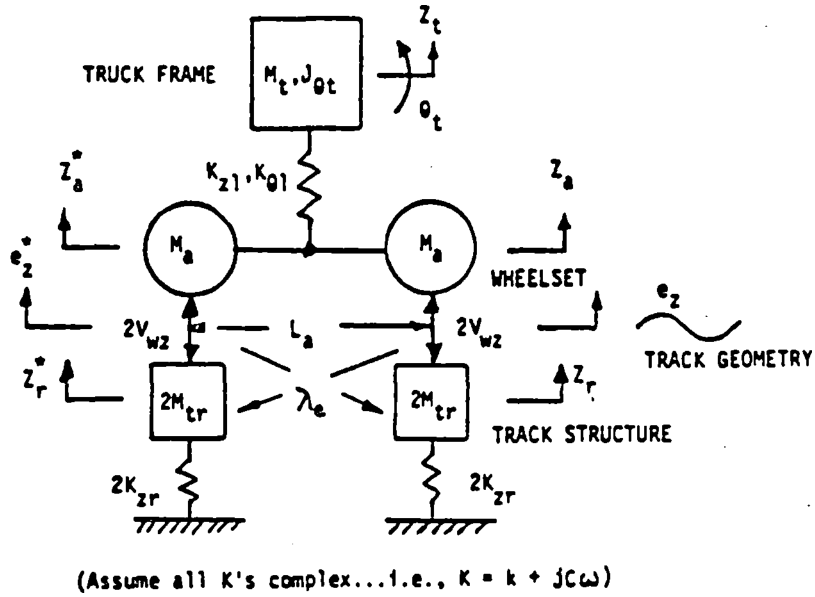


FIGURE H-28 SKETCH OF VEHICLE TRUCK/TRACK STRUCTURE MODEL FOR VERTICAL/PITCH DYNAMIC RESPONSE

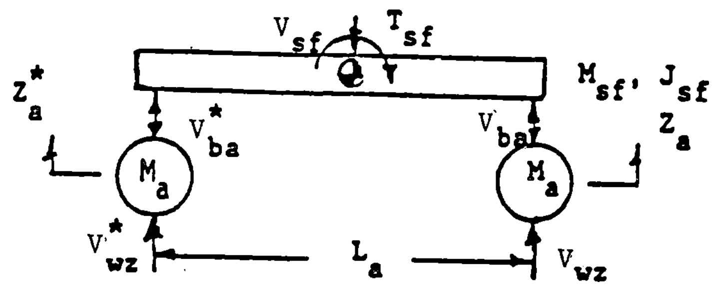


FIGURE H-29. SIDE FRAME/EQUALIZER NOMENCLATURE

The side frame/Equalizer beam effective mass and mass moment of inertia in pitch can also be included at this point, see Figure H-29:

$$M_z \ddot{Z}_a + 2V_{ba} - 2V_{wz} = 0, \quad M_z \ddot{Z}_a^* + 2V_{ba}^* - 2V_{wz}^* = 0 \tag{15}$$

$$M_{sf}(\ddot{Z}_a + \ddot{Z}_a^*)/2 = V_{ba} + V_{ba}^* - V_{sf} \tag{16}$$

$$J_{sf}\,(\ddot{Z}_a - \ddot{Z}_a^*)/L_a = L_a(V_{ba} - V_{ba}^*)/2 - T_{sf} \tag{17}$$

$$V_{sf} = K_{z1}\,(Z_a + Z_a^*)/2 - Z_t \tag{18}$$

$$T_{sf} = K_{\theta 1}\,(Z_a - Z_a^*)/L_a - \theta_t \tag{19}$$

By combining Equations 16 through 19 . . .

$$V_{ba} = \overline{M}_{sf}\ddot{Z}_a + \overline{M}'_{sf}\ddot{Z}_a^* + (K_{z1}/4 + K_{\theta 1}/L_a^2)Z_a + (K_{z1}/4 - K_{\theta 1}/L_a^2)Z_a^* - (K_{z1}/2)Z_t - (K_{\theta 1}/L_a)\theta_t \tag{20}$$

$$V_{ba}^* = \overline{M}_{sf}\ddot{Z}_a^* + \overline{M}'_{sf}\ddot{Z}_a + (K_{z1}/4 + K_{\theta 1}/L_a^2)Z_a^* + (K_{z1}/4 - K_{\theta 1}/L_a^2)Z_a - (K_{z1}/2)Z_t + (K_{\theta 1}/L_a)\theta_t \tag{21}$$

where . . . $\overline{M}_{sf} = M_{sf}/4 + J_{sf}/L_a^2$ and $\overline{M}'_{sf} = M_{sf}/4 - J_{sf}/L_a^2$.

Then combining Equations 12 through 15 with Equations 20 and 21 . . .

$$(M_a + 2M_{tr} + 2\overline{M}'_{sf})\ddot{Z}_a + (2\overline{M}'_{sf})\ddot{Z}_a^* + 2(K_{z1}/4 + K_{\theta 1}/L_a^2 + K_{zr})Z_a$$

$$+ 2(K_{z1}/4 - K_{\theta 1}/L_a^2 - \lambda_v K_{zr})Z_a^* = (K_{z1}/2)Z_t - (K_{\theta 1}/L_a)\theta_t \tag{22}$$

$$= 2M_{tr}\ddot{e}_z + 2K_{zr}e_z = 2\lambda_v K_{zr}e_z^*$$

$$(M_a + 2M_{tr} + 2\overline{M}_{sf})\ddot{Z}_a^* + (2\overline{M}'_{sf})\ddot{Z}_a + 2(K_{z1}/4 + K_{\theta 1}/L_a^2 + K_{zr})\ddot{Z}_a^*$$

$$+ 2(K_{z1}/4 - K_{\theta 1}/L_a^2 - \lambda_v K_{zr})Z_a - (K_{z1}/2)Z_t + (K_{\theta 1}/L_a)\theta_t \tag{23}$$

$$= 2M_{tr}\ddot{e}_z^* + 2K_{zr}e_z^* - 2\lambda_v K_{zr}e_z$$

The final set of equations for the wheelsets, then, includes the track geometry inputs both at the particular wheelset and at the adjacent wheelset through the "influence coefficient." Note that both first and second derivatives of the track geometry inputs are required in this formulation. For the power spectral density input, this poses no problem, since a sinusoidal input at the given frequency, f, with a root-mean-squared amplitude, E, is implied [23]"

$$e_z = E \sin 2\pi ft = \sqrt{p_i(f)} \quad \sin 2\pi ft \tag{24}$$

where

$P_i(f)$ = track geometry spectrum (in^2/Hz)
$= P_i'(\lambda)/V$ [$P_i'(\lambda)$ = in^2/cycle/ft]

V = vehicle speed, ft/sec

f = frequency, Hz

t = time, sec.

λ = wavelength, ft.

Track geometry components at the front and rear wheelsets are given in Table H-8. Components preceded by the imaginary operator, j [$j = \sqrt{-1}$] are entered into the imaginary portion of the complex, $P_i(\omega)$. Solutions are then generated in the computer program by sweeping through the frequency band of interest, generally in 1/10th or 1/12th octave steps; and model response is calculated at each frequency by matrix inversion and multiplication techniques.

TABLE H-8. TRACK GEOMETRY INPUT COMPONENTS AT FRONT AND REAR WHEELSETS FOR RANDOM-VIBRATIONS (PSD) VEHICLE/TRACK MODEL

Phase-shifted track geometry inputs

$$e_z = E \sin\omega t = E$$

$$\dot{e}_z = \omega E \cos\omega t = j\omega E$$

$$\ddot{e}_z = -\omega^2 E \sin\omega t = -\omega^2 E$$

$$e_z^* = E \sin(\omega t - \tau)$$

$$= E (\sin\omega t \cos\tau - \cos\omega t \sin\tau)$$

$$= E \cos\tau - jE \sin\tau$$

where $\tau = 2\pi L_a/\lambda$

λ = geometry wavelength

$$\dot{e}_z^* = \omega E \cos(\omega t - \tau)$$

$$= j\omega E \cos\tau + \omega E \sin\tau$$

$$\ddot{e}_z^* = -\omega^2 E \sin(\omega t - \tau)$$

$$= -\omega^2 E \cos\tau + j\omega^2 E \sin\tau$$

Since this model is linear by nature, linear values of track vertical stiffness must be used. These are generally assumed to be the tangent stiffness calculated at the particular vertical static wheel load as an "operating point". Recommended values of the track modulus parameters are given in Table H-9 for a single degree-of-freedom track model under each wheel of the vehicle model. These values correspond to the extremes and nominal values of track cited in Section H.4.2.1.

TABLE H-9. RECOMMENDED VERTICAL MODULUS VALUES FOR VEHICLE/TRACK SIMULATION

Track Description	Wheel Load (lb)	Stiffness* (Tangent) K_{zr}(lb/in)	Track Effective Mass $\frac{*}{2}$ M_{tr}(lb sec^2/in)	Adjacent Wheel Influence Coefficient, λv**	Track Natural Frequency, f_n (Hz)	Damping Ratio C/Ccr
Soft	8000	65,000	1.83	.25	30	.30
	32,000	130,000	3.66	.25	30	.30
Good (Nominal)	8000	175,000	2.77	.18	40	.25
	32,000	350,000	5.54	.18	40	.25
Stiff	8000	438,000	4.44	.13	50	.20
	32,000	875,000	8.88	.13	50	.20

* Per-rail values

** Estimated from field test data, 2-axle locomotive (L_a = 108")

H.5.1.2 Nonlinear Wheel/Rail Impact Load Model

While the above equations can form the basis for a nonlinear, time-domain computer model, a more complex representation of the wheel/rail interface and the track structure are needed to study impact loads. A simplified discrete-mass model of the wheel, rail, and track structure, similar to that described by Bjork [24], was used to investigate loads generated by rail joints and wheel flats [22]. A nonlinear Hertzian contact stiffness, K_H, was used directly to generate the wheel/rail load:

$$V_{wz} = (2K_H \Delta Z_w/3)^{1.5} + C_H \Delta\dot{Z}_w \geq 0 \tag{25}$$

$$\Delta Z_w = Z_r - Z_a + e_z + \Delta Z_H \tag{26}$$

$$\Delta \dot{Z}_w = \dot{Z}_r - \dot{Z}_a + \dot{e}_z \tag{27}$$

$$\Delta Z_H + 1.5(V_{w,static})^{2/3}/K_H \tag{28}$$

Note that K_H is related to the "Hertzian flexibility constant", G, used by British Rail [23] as follows:

$K_H = 3/2G = 3.03 \times 10^5 \ (\text{lb})^{2/3}/\text{in.}$ for nominal wheel and rail profiles, and

$a = GV_{wz}^{2/3}$, the wheel/rail "indentation" [25].

A linear stiffness, K_{HH}, about the static wheel load can also be calculated:

$K_{HH} = K_H V_{wz}^{1/3} = 9.7 \times 10^6 \text{lb/in.}$ for $V_{wz} = 32{,}000$ lb.

Other elements in the track structure are modeled in a simplified, but nonlinear way and are shown in Figure H-29. The equations for forces on the mass elements in Figure H-30 are as follows:

$$V_{zr} = (0.5K_r \Delta Z_{tr})^2 + C_r \Delta \dot{Z}_{tr} \tag{29}$$

$$\Delta Z_{tr} = Z_t - Z_r + \Delta Z_r \tag{30}$$

$$\Delta \dot{Z}_{tr} = \dot{Z}_t - \dot{Z}_r \tag{31}$$

$$\Delta Z_r = 2(V_{w,static})^{0.5}/K_{zr} \tag{32}$$

$$V_{zt} = K_t(Z_b - Z_t) + C_t(\dot{Z}_b - \dot{Z}_t) + V_{w,static} \tag{33}$$

$$V_{zb} = K_b Z_b + C_b \dot{Z}_b + V_{w,static} \tag{34}$$

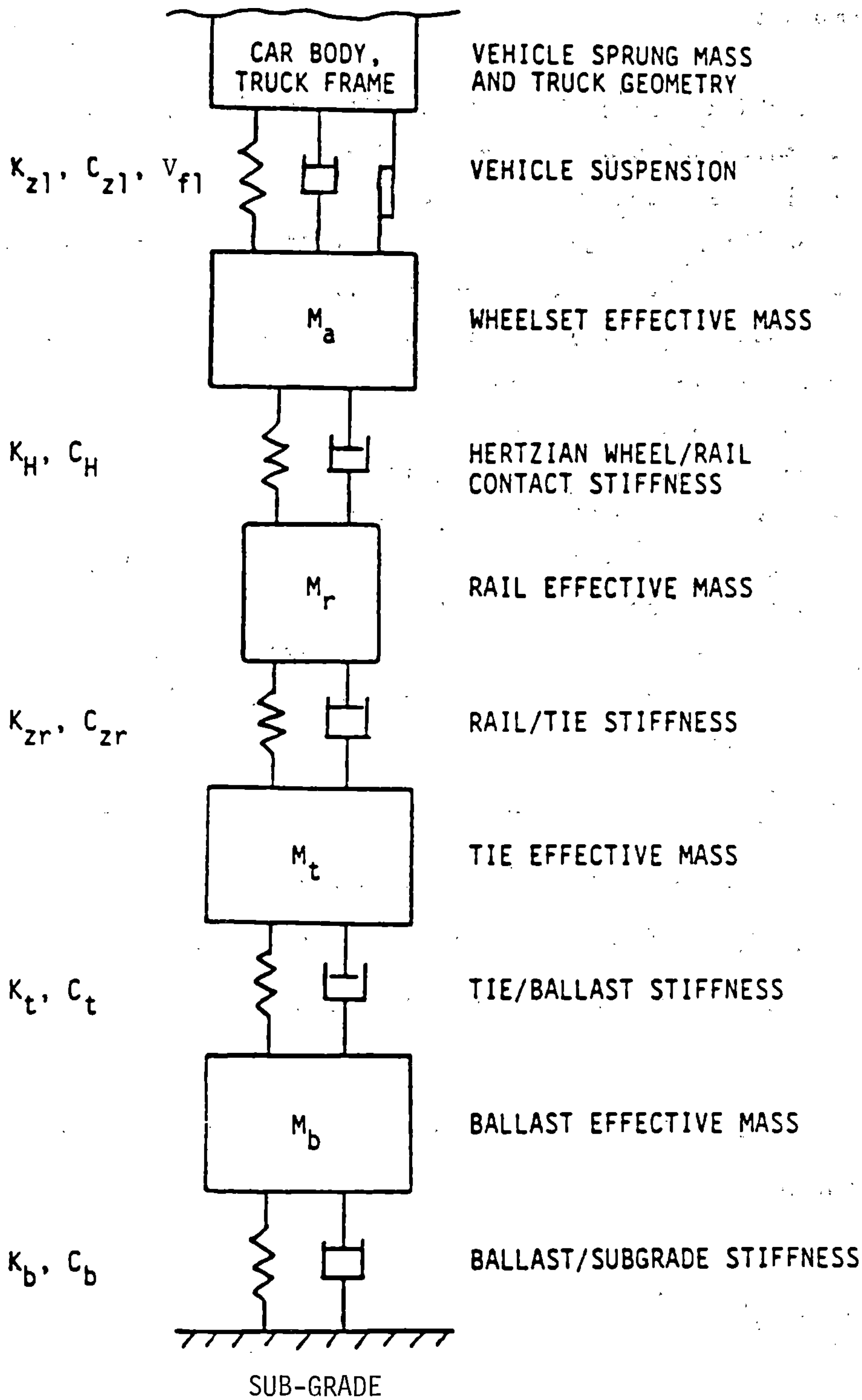


FIGURE H-30. SKETCH OF WHEEL/RAIL DISCRETE-MASS IMPACT LOAD MODEL

Track structure component values are determined primarily from the linear BOEF formulae:

K_{zr} = $2[4EI(k_p/L_t)^3]^{0.25}$, the rail/tie stiffness, lb/in,
k_p = individual pad stiffness, lb/in,
L_t = tie (pad) spacing along rail, in,
M_r = $3w_r[EI/K_r]^{1/3}/386$, the rail effective mass, lb-sec^2/in,
w_r = rail weight per unit length, lb/in,
K_t = $2[4EI(u_b A_T/L_t)^3]^{0.25}$, track structure stiffness, lb/in,
a = tamped length, tie end, in,
b = tamped width of tie, in,
A_T = ab, tamped area, tie end, in^2,
C = 1.04[1 + a/28.5b], a shape factor,
u_b = $CE_b/[(1 - \nu^2)A_T^{0.5}]$, the "soil modulus", lb/in^3,
E_b = ballast modulus (use 40,000 lb/in^2),
M_t = $3w_t[EI/K_t]^{1/3}/386$, tie effective mass, lb-sec^2/in,
w_t = (Tie weight)/$2L_t$ = tie weight per length along track,
K_{TR} = $2[4EIU^3]^{0.25}$, track overall stiffness,
K_b = $[K_{TR}^{-1} - K_{zr}^{-1} - K_t^{-1}]$ the ballast effective stiffness
ν = Poisson's ratio (use 0.5 for ballast)
M_b = $K_{TR}/(2\pi f_n)^2 - M_r - M_t$, the ballast effective mass,
f_n = track fundamental natural frequency, Hz.

All damping factors are handled as viscous linear damping by the relationship . . .

$$C_i = 2\zeta[K_i M_i]^{0.5}$$

where ζ = ratio of damping to critical damping.

The vertical rail geometry error (or wheel running surface geometry error), e_z and $\dot{e}_z$, is applied as a discrete time input function. The classical dipped-joint function used by British Rail [13] is in the form . . .

$$e_z(x) = -d(1 - \cos\pi x/S), \quad 0 < x < S/2, \tag{35a}$$

$$e_z(x) = -d[1 - \cos(\pi x/S + \pi)], \quad S/2 < x < S, \tag{35b}$$

where

d = joint dip depth, S = dip span,

x = distance along rail.

This function was also used to modulate the rail effective stiffness, K_r, reducing the stiffness directly at the joint to a value from 0.5 to 0.75 times the initial stiffness value.

A versine function suggested by Lyon [13] has proven to be representative of a service-worn slid wheel flat:

$$e_z = -0.5D_f[1 - \cos(2\pi Vt/L_f)] \tag{36a}$$

$$e_z = -(D_f\pi V/L_f)\sin(2\pi Vt/L_f) \tag{36b}$$

where

$D_f = L_f^2/16R_w$, the flat length

L_f = the flat length, R_w the wheel radius.

This "rounded" flat depth is roughly one-half the depth of the freshly slid flat that has not been battered by rotation.

H.5.1.3 Model Results of Freight Vehicle Responses for Varying Vertical Track Modulus

A study using an existing computer program "HALF," developed at the Transportation Systems Center, was used by the Analytic Sciences Corporation. The program is designed for the study of continuous track movement under a linear half vehicle with equalized trucks on symmetric track. It has been adapted to provide evidence of the severity that variations of vertical track modulus have upon vehicle response [28].

Two vehicles and a range of track conditions were used to provide information on the consequences of varying vertical track compliance on the test results. The cars were chosen to include troublesome derailment characteristics and a broad range of suspension frequencies and loads.

Particular modes of vibration were chosen for investigation representing a range of potential track induced excitation. The wavelengths are chosen to provide a fundamental resonance within the running range of the vehicle. They are 19.5 ft in bounce, 39.0 ft in roll and, a truck wheel base of 5.5 ft for body twist. Since the equivalent absolute values of amplitude in vehicle roll and twist are not the normally measured variables, results are only provided for percentage change in vehicle response calculated at each frequency (or speed).

A wide range of vertical track modulus values were used in the study. The nominal value of 3500 lb/in. per in. was used to typify track under relatively "hard" conditions. Variation of modulus to ± 60 percent of the nominal value were used for the results. Further runs were carried out at values of nominal modulus of 800 to 1000 lb/in. per in. of track regarded as a better description of "soft" track.

Typical outputs of the computer program HALF for the vehicles are given in Figures H-31 to H-34.

H.5.2 Simulation of Lateral Track Modulus

Up to the point of flange contact, the wheelset is weakly coupled laterally to the other rail through creep forces, described by the basic equations in the following yaw/lateral matrix [22,26]:

	Y_a	Ψ_a
y_a	$2f_{22}/V$	$-2f_{23}/V$
$\dot{y}_a$	$2V_a(\zeta\delta_o + \varepsilon)/g$	$f_{11}g\lambda_c/r_o$
$\dot{\psi}_a$	$2f_{23}/V$	$f_{11}g^2/2V$
ψ_a	$-2f_{22}$	$2f_{23} = V_a g\delta_o/2$

where

V = axle forward velocity, in./sec

V_a = static vertical axle load, lb

δ_o = angle between contact plane and horizontal, rad

ε = rate of change of contact plane slope with lateral displacement of wheelset

ζ = rate of change of distance between wheelset centerline and contact points with lateral displacement of wheelset

λ_c = effective conicity (rate of change of rolling radius with lateral displacement of wheelset), rad

g = effective gauge (lateral distance between contact points with wheelset centered), in.

r_o = wheel tread radius, in.

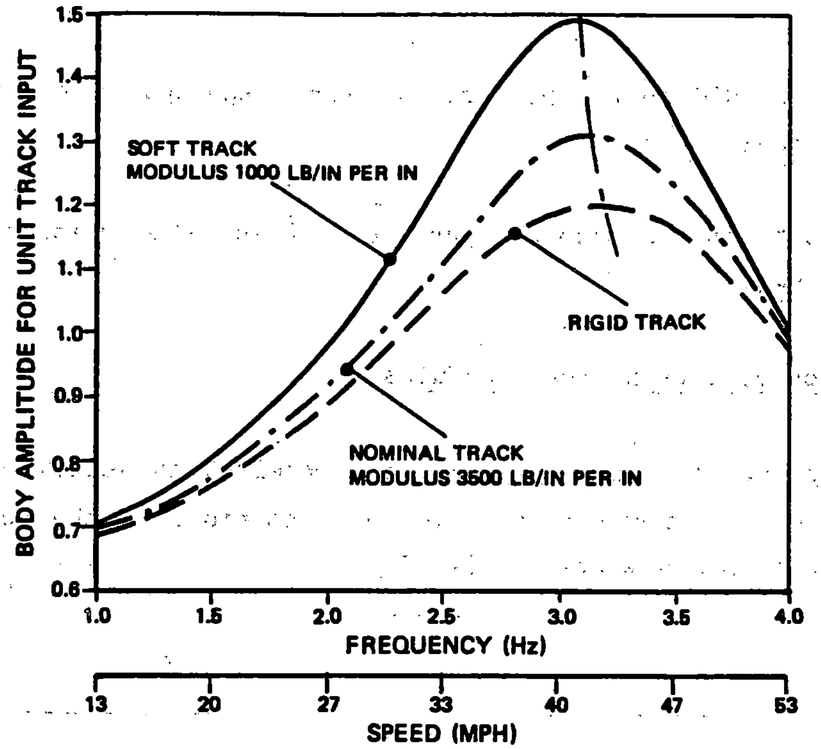


Figure H-31 Body Bounce - 70 Ton Flatcar Empty

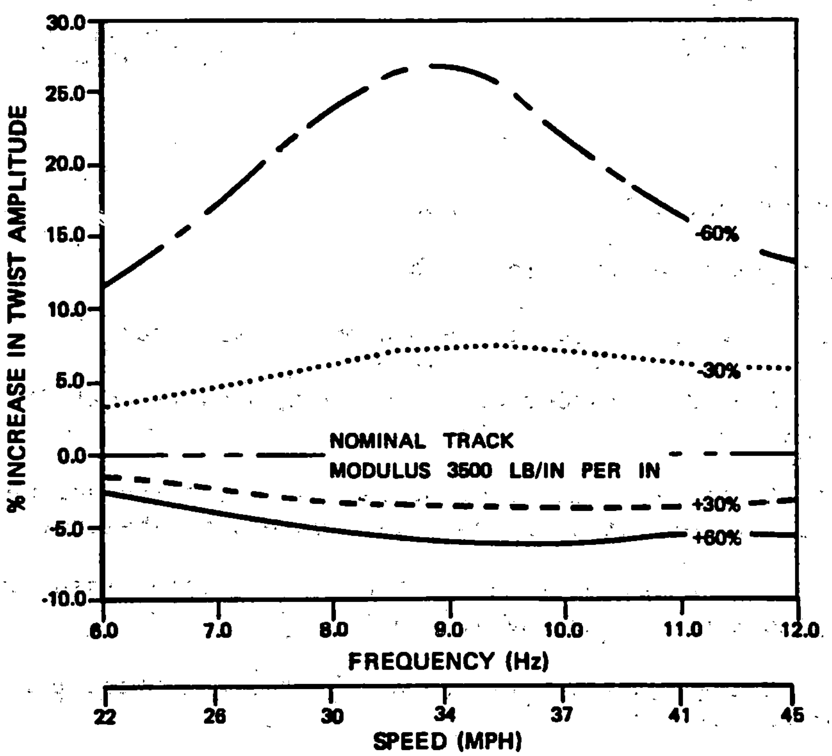


Figure H-32 Body Twist - 70 Ton Flatcar Empty

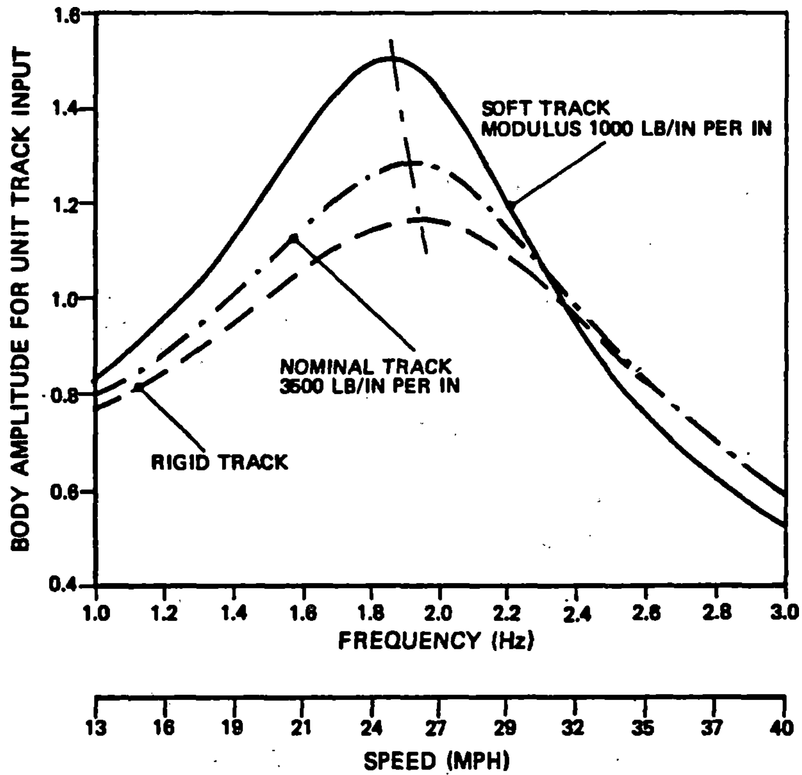


Figure H-33 Body Bounce - 70 Ton Boxcar Fully Loaded

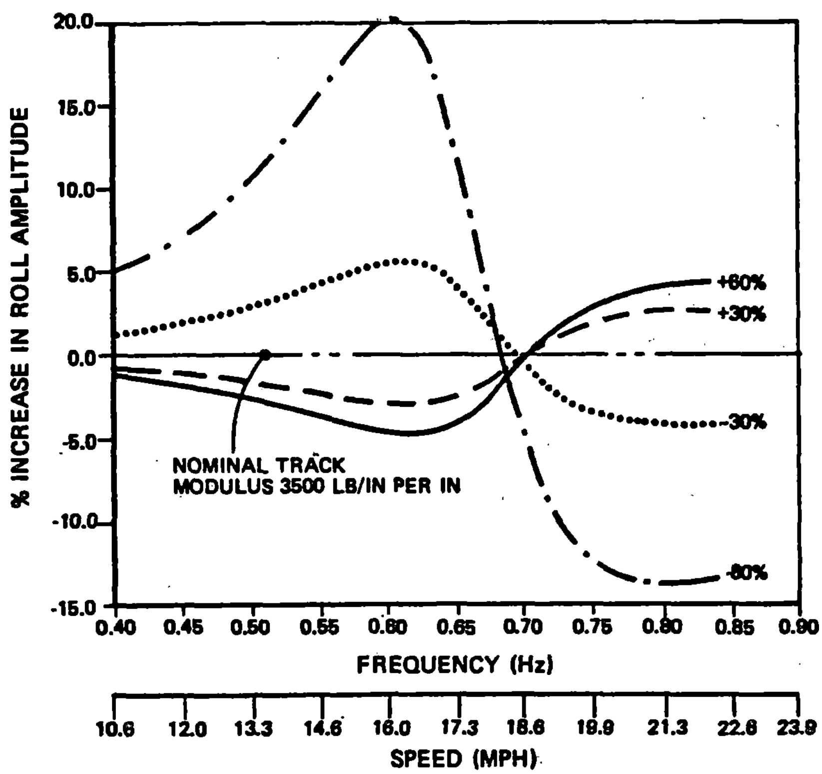


Figure H-34 Lower Center Roll - 70 Ton Boxcar Fully Loaded

f_{11} = longitudinal creep coefficient, lb

f_{22} = lateral creep coefficient, lb

f_{23} = lateral/spin creep coefficient, lb-in.

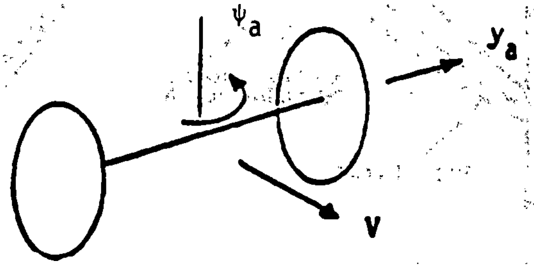


FIGURE H-35. POLARITY OF MOTIONS NOTATION

Notation and polarity of motions vary from author to author, see Figure H-35, and the above configuration ignores gyroscopic effects, the spin creep effects, and nonlinearities that might exist in the wheel/rail geometry. However, it is representative of currently used models.

Up to the point of flange contact, then, the lateral wheel/rail forces are functions of these creep-related terms:

$$L_w = f[(y_a - y_r - y_t - e_y), (\dot{y}_a - \dot{y}_r - \dot{y}_t), \psi_a, \dot{\psi}_a] \tag{37}$$

where

e_y = rail alignment geometry error
y_r = lateral deflection of the rail
y_t = lateral deflection of tie (see Figure H-20).

Once the nominal flange clearance, Δy_r, is exceeded, the lateral flanging force can be defined as:

$$L_{fy} = K_r[y_a - y_r - y_t - e_y - \Delta y_r] + C_r[\dot{y}_a - \dot{y}_r - \dot{y}_t] \tag{38}$$

where $y_r' = L_w/K_y$, the rail deflection due to creep forces,

$$y_r = y_r' + L_{fy}/K_y \tag{39}$$

These relationships, for illustration, have been defined in terms of a linear rail lateral stiffness, K_y, but not by any means restricted by linearity. The forces and deflections are shown in Figure H-36.

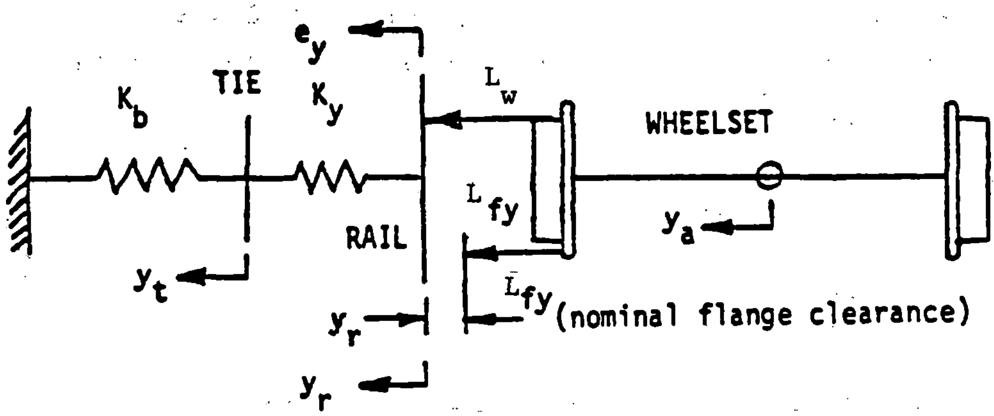


FIGURE H-36. SCHEMATIC OF WHEEL/RAIL CREEP AND FLANGING FORCES

As shown in Figure H-20, the tie lateral motion is a function of <u>net</u> lateral wheel load, and therefore we have to define left and right rail forces:

$$y_t = (L_L - L_R + L_{fL} - L_{fR})/K_b \tag{40}$$

Note that in these definitions the rail and the tie are not explicitly given as separate degrees of freedom in a dynamic sense; deflections at the given locations can be defined simply from force balances. However, by adding masses at these points, the model may be expanded dynamically.

Two other effects are desired to complete the model: first, the added deflection under one wheel due to lateral load at an adjacent wheel; and second, the lateral stiffening due to a vertical load at the adjacent wheel. These both can be handled in a straightforward manner:

$$y'_{r1} = L_{y1}/K_{y1} + \lambda_L [L_{y2} + L_{fy2})K_{y2} = L_{y1}/K_{y1} + \lambda_L y_{r2} \quad (41a)$$

$$y'_{r2} = L_{y2}/K_{y2} + \lambda_L y_{r1} \quad (41b)$$

where λ_L = the lateral deflection "influence coefficient".

Subscripts 1 and 2 refer to Wheelsets #1 and #2, adjacent wheels on the same rail. Note that the rail lateral stiffness need not be the same under the two wheels, and may be influenced by the vertical wheel loads. For example, for the bilinear representation of Equation 12, the lateral stiffness can be represented by:

$$K_{y1} = \alpha + \beta (V_{z1} + \lambda_v V_{z2}) \quad (42a)$$

$$K_{y2} = \alpha + \beta (V_{z2} + \lambda_v V_{z1}) \quad (42b)$$

where λ_v = the vertical wheel load "influence coefficient".

Recommended values for simulating the track lateral stiffness characteristics are presented in Table H-10.

TABLE H-10. RECOMMENDED LATERAL TRACK STIFFNESS VALUES FOR VEHICLE/TRACK SIMULATION

Track Description	Rail Lateral Stiffness Parameters: α (lb/in)	β	$(L/V)_b$	K_{rz}* (lb/in)	Tie/Ballast Lateral Stiffness K_b (lb/in)	Adjacent Wheel Load Influence Coefficient λ_L **
Soft	15,000	2	0.33	15,000	113,000	0.50
Nominal	40,000	5	0.33	40,000	300,000	0.36
Stiff	100,000	15	0.75	100,000	750,000	0.27

* See Figure H-22

** Estimated from field test data, 2-axle locomotive (L_a = 108")

H.5.2.1 Nonlinear Time-Domain Vehicle-Track Model

To illustrate the application of lateral track modulus to vehicle/track simulation, a nonlinear time-domain model currently used by Battelle (program PERTRK) will be described. This is a 17 degree-of-freedom model that includes the yaw and lateral motions of four wheelsets, and the yaw, lateral, and roll motions of the truck and car body sprung masses. Lateral mass effects of the track are not included, so that only the stiffness and damping effects of the track lateral modulus are considered. No vertical degrees of freedom are included, but vertical wheel loads are estimated from force and torque balances on the individual wheelsets. Provision for zero vertical force (wheel lift) is included in the model equations. While neither the wheelset mass moment of inertia in roll nor the track vertical mass effects are considered as degrees of freedom, the track stiffness and damping are incorporated as a torsional impedance in series with the primary roll suspension of the vehicle.

A number of basic nonlinearities are included in the model. Both the primary and secondary lateral and roll suspensions are modeled as bilinear spring elements to simulate suspension stops; and Coulomb friction is included with the viscous damping in each. The centerplate central bearing (the truck frame/car body interface) includes a friction breakaway torque along with a viscous damping torque. The track lateral modulus is modeled as a nonlinear spring with flange clearance.

The main program of PERTRK acts in a supervisory role, handling input and output chores, including printouts, summary tables of maximum values, and (if desired) graphical plots through the Calcomp plotter and DISSPLA plot routines. The actual dynamic modeling is done in three subroutines: 1) RK24, a fourth-order Runge-Kutta integration routine for second-order differential equations; 2) GEOPRT, a subroutine for calculating alignment, gauge, and cross-level geometry errors as a function of distance (speed and time), in this particular case a versine function [2]; and 3) UPDATE, a subroutine for calculating the instantaneous acceleration for each degree of freedom.

Specific steps in Subroutine UPDATE are given in Figure H-37. The program performs three tasks in the following sequence for each integration time step:

Subroutine UPDATE

Step 1. Set interim variables to zero or initial condition at T = 0.

2. Call Subroutine GEOPRT (calculate track geometry for time, T)

3. Calculate wheel/rail lateral differential positions and velocities.

4. Calculate wheel/rail creep forces and torques.

5. Check adhesion limits and set maximum forces if exceeded.

6. Calculate flanging forces if differential position exceeds nominal flange clearance.

7. Calculate primary and secondary suspension lateral forces and roll torques.

8. Calculate vertical wheel/rail loads from force and torque balance.

9. Calculate total rail deflections (print variables).

10. Calculate values for next time step (not used in differential equations for the rest of this time step):

    a. Rail lateral stiffness values

    b. Rail deflections under creep forces only

    c. Tie deflections under net lateral loads

    d. Nonlinear rail stiffness parameters

11. Calculate force balance divided by mass (acceleration) for each of 17 degrees of freedom.

12. Return to Program PERTRK or to Subroutine RK24.

FIGURE H-37. COMPUTATIONAL STEPS IN PROGRAM PERTRK, SUBROUTINE UPDATE

- o Calculate forces based on displacements and velocities of the degrees of freedom and track geometry errors (inputs) existing at time T,

- o Calculate new displacements and rail stiffness values for the next time step (T = T + DT) based on these forces,

- o Calculate accelerations for the degrees of freedom based on these forces, displacements, and velocities at time T.

The integration time interval, DT, is a fixed parameter and must be determined pragmatically for a particular vehicle model by making exploratory uns at several stip sizes, then checking program outputs for any changes in results. Calculation accuracy depends on the model bandwith (highest natural frequencies of degrees of freedom) and degree of nonlinearity. For example, "well-behaved" vehicle models like the GP40-2 and AEM-7 locomotives run with no significant loss in accuracy with a time step DT=2 milliseconds. The highly nonlinear freight car, however, requires very small time steps (less than 0.2 milliseconds) to provide even a stable solution.

An example of time-history plots of wheel forces from the program is shown in Figure H-38. In this simulation, the response of a GP40-2 locomotive to a severe geometry error (1.5-inch line error, 0.5-inch crosslevel error) was explored to determine the effects of track lateral modulus on vehicle dynamic response. For the track perturbation to the left, the lead wheel first, then the trailing wheel of the lead truck impacts the right rail, followed by a secondary impact of greater amplitude on the left rail. A third, minor impact load occurs in the right rail beyond the perturbation. This response was typical of the AEM-7 locomotive tests conducted with this type of perturbation on the Northeast Corridor track during evaluation tests [27].

The effects of the three values of track lateral stiffness plotted in Figure H-22 were explored with the GP40-2 locomotive model. Variation in maximum wheel lateral load with speed are shown in Figure H-39, while in Figure H-40 the peak rail lateral deflections under these loads are plotted. Note in Figure H-38 that the peak leading and trailing wheel loads do not occur at the same time, but rather at nearly the same location in the track.

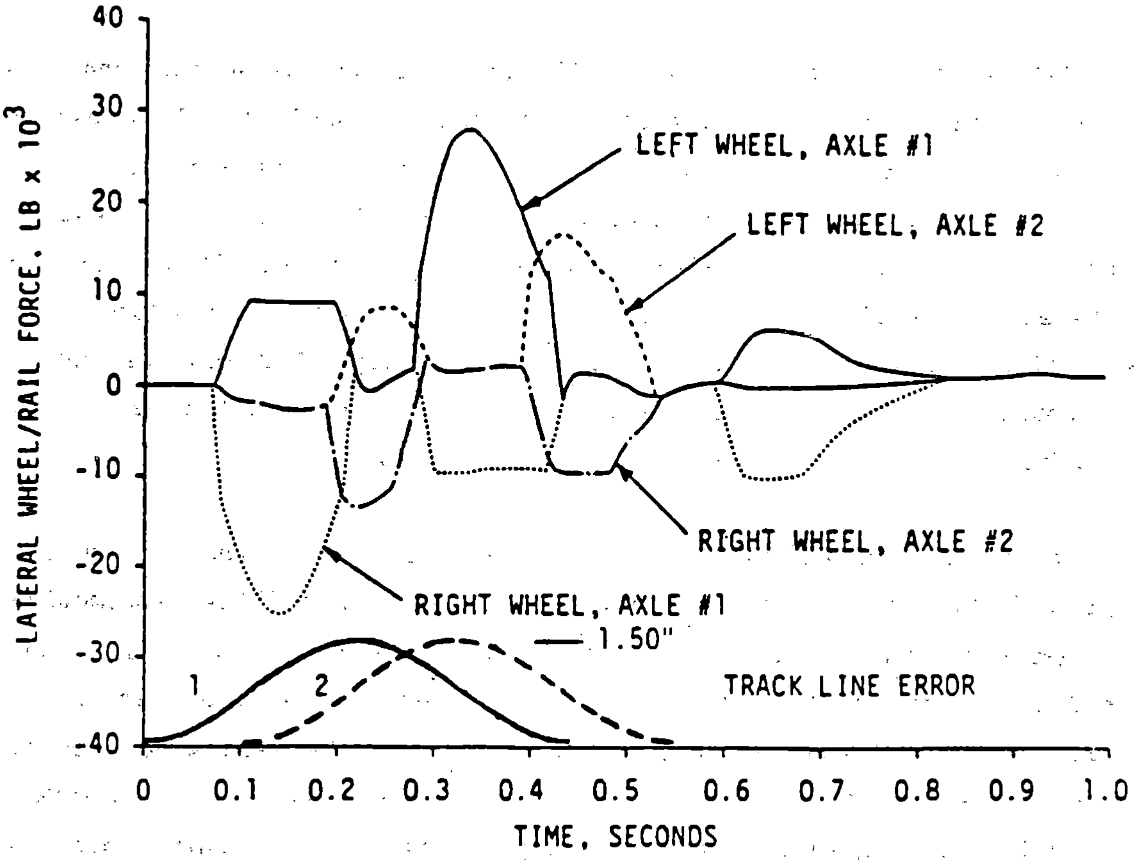


FIGURE H-38. TIME HISTORY OF PREDICTED WHEEL/RAIL LATERAL FORCES DUE TO TRACK GEOMETRY ERROR -- NOMINAL TRACK STIFFNESS, SIMULATED GP40-2 LOCOMOTIVE AT 60 MI/H

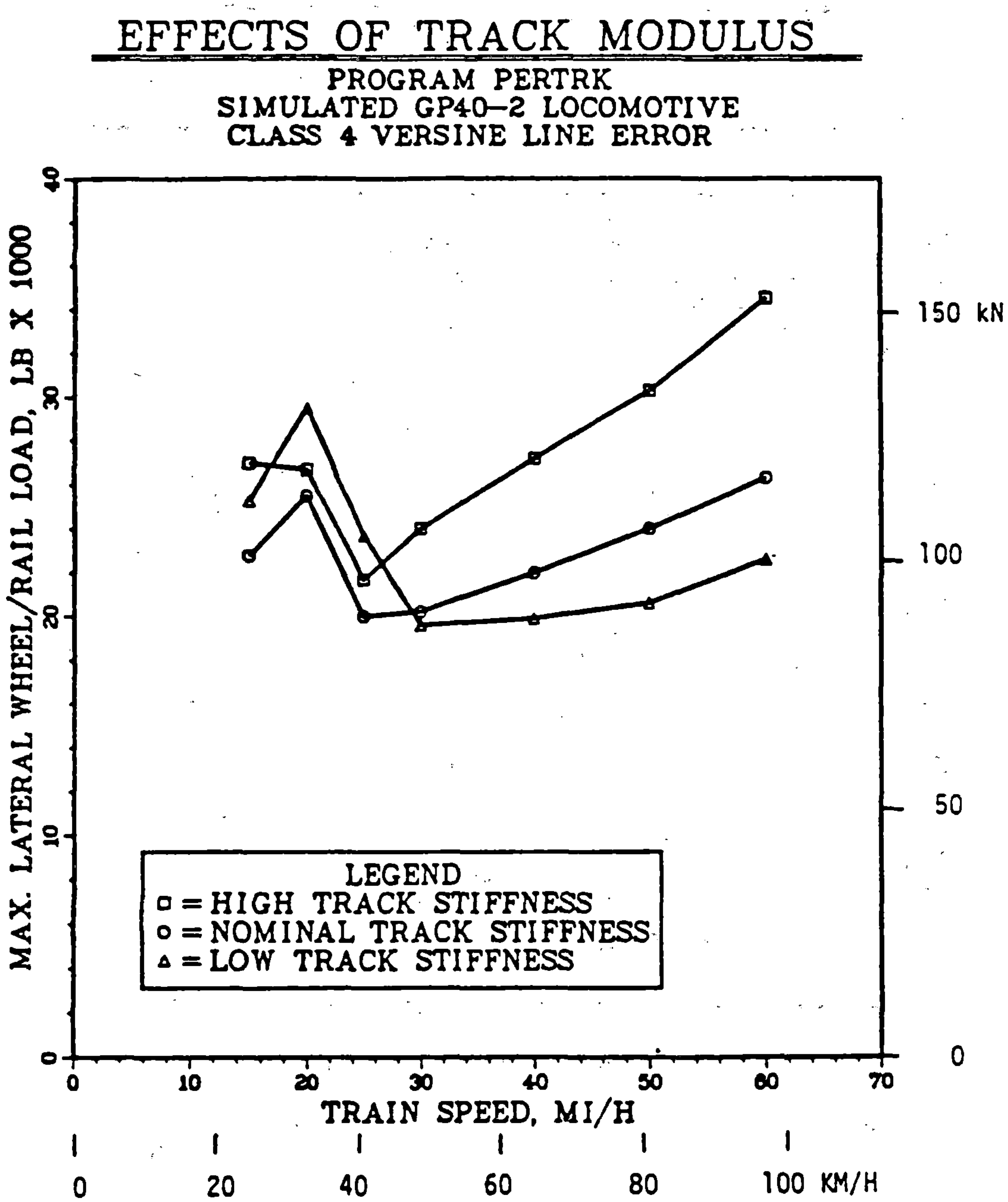


FIGURE H-39. EFFECT OF TRACK LATERAL STIFFNESS ON MAXIMUM LEADING-AXLE LATERAL WHEEL/RAIL FORCES

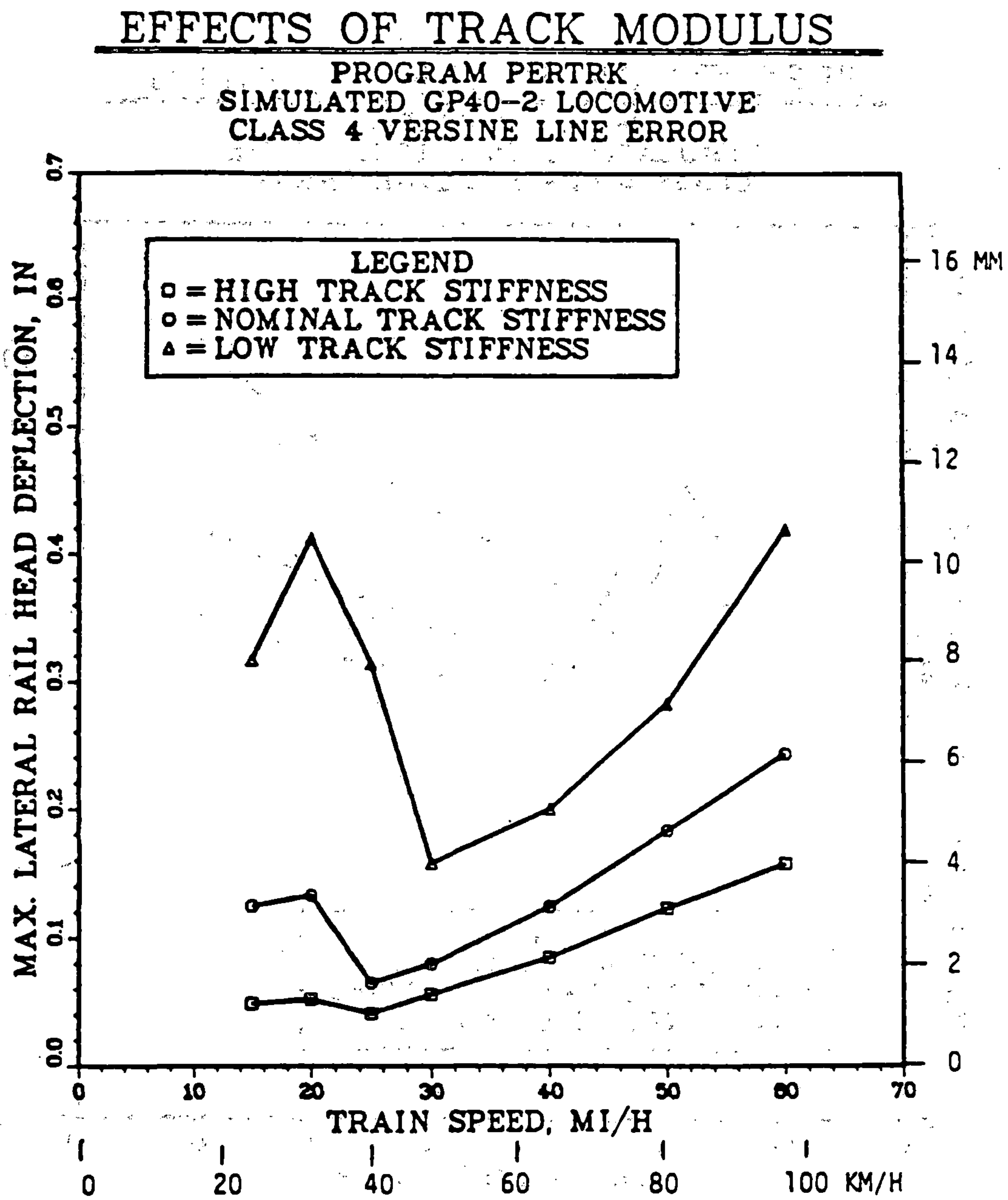


FIGURE H-40. EFFECT OF TRACK LATERAL STIFFNESS ON MAXIMUM RAIL LATERAL DEFLECTION UNDER LEADING AXLE

A 100-ton freight car truck, more easily swiveled and with a shorter wheelbase, would in some cases produce higher simultaneous lateral loads (higher total-truck loads on one rail), and greater rail lateral deflections will occur on softer track.

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SECTION I
PERFORMANCE INDICES

I-1 INTRODUCTION

A major thrust of current research in rail vehicle dynamics involves the assessment of safety on the basis of analysis and measurement. An issue central to this work is the requirement to measure nearness to derailment objectively and reliably. The detailed phenomena of derailment are so exceedingly complex that most existing dynamic models fall far short of capturing them. For this reason, a small number of indirect measures (notably the L/V ratio) have been used over the years to correlate derailment tendency with response characteristics which are relatively easy to measure or simulate. Such indirect measures are called performance indices.

The objective of this report is to suggest performance indices which can be used to predict derailment of locomotives and cars. To achieve this goal, we first identify the various ways derailments can occur. Then we determine the processes or dynamic system responses that can lead to derailment in each of these modes. We briefly discuss the role of analytical methods to investigate and predict dynamic behavior, and the concept of model hierarchy is introduced to relate the physical system to mathematical models of varying complexity. Key variables and parameters which influence the derailment processes are identified; these factors are combined to synthesize candidate performance indices using both physical reasoning and experimental results. Critical threshold values of the candidate indices are suggested when possible. Procedures to select and validate the most promising performance indices are outlined.

I-2. DERAILMENT MODES

Based on an examination of the mechanics of the vehicle/track system, five fundamental derailment modes have been identified: wheel climb, rail deflection, wheel lift, car-truck separation, and sudden component failure. Each of these derailment modes describes how the vehicle permanently leaves the track, and is characterized by a unique derailment indicator - a measure which determines that the derailment mode threshold has been exceeded. Table I-2.1 summarizes the modes and their associated indicators.

TABLE I-2.1
DERAILMENT MODES AND INDICATORS

MODE	DESCRIPTION	DERAILMENT INDICATOR
Wheel Climb	Flange moves laterally beyond the rail	$y_i \geq y_{\ell im}$, any wheel
Wheel Lift	Flange moves vertically above the rail	$z_i \geq z_{\ell im}$, any wheel
Rail Deflection	Rail moves outward so that gauge exceeds wheelset width	$b_i \geq \ell_w$, any axle
Car-Truck Separation	Carbody centerplate separtes from truck centerplate	$h_i \geq h_{\ell im}$, either truck
Sudden Component Failure	Force or stress in a critical vehicle or track component exceeds an allowable level	$\left.\begin{array}{l}\sigma_i \geq \sigma_{\ell im_i} \\ \text{or} \\ F_i \geq F_{\ell im_i}\end{array}\right\}$ any location

I-2.1 WHEEL CLIMB

Wheel Climb is defined as the mode of derailment which is produced whenever the position of the wheel flange passes laterally over the rail centerline as shown in Fig. I-2-1. The derailment indicator for this situation is simply expressed as the lateral displacement of any wheel flange beyond a threshold value, i.e., $Y_i \geq Y_{\ell im}$, i = 1 to N, where N is the number of axles.

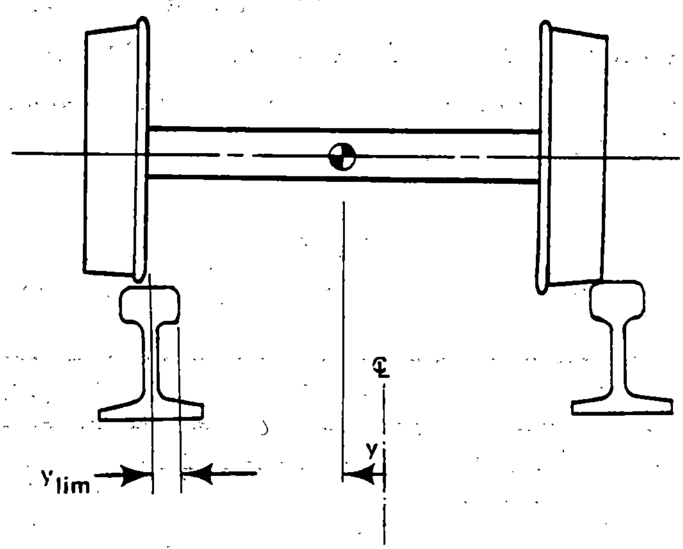

Figure I-2-1 Wheel Climb Derailment Mode

I-2.2 WHEEL LIFT

A Wheel Lift derailment is said to occur whenever the lower edge of the flange lifts above the railhead; thus both bounce and tipping lead to wheel lift. An indicator for impending derailment is that the vertical displacement for any wheel exceeds a critical threshold value, i.e., $Z_i \geq Z_{\ell im}$, i = 1 to N. This situation can result from either a vertical translation or rotation of the axle in the transverse plane as illustrated in Fig. I-2-2.

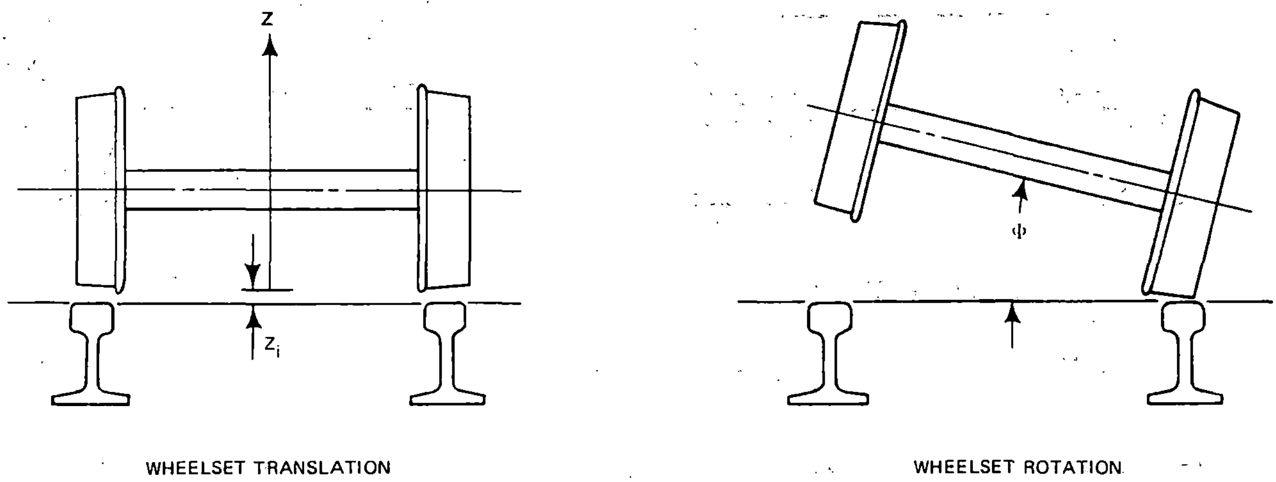

Figure I-2-2 Wheel Lift Derailment Mode: Bouncing or Tipping

I-2.3 RAIL DEFLECTION

Rail Deflection is defined as a derailment mode in which the rail gauge exceeds the wheelset width. This mode of derailment can result from either a lateral rectilinear (rail spread) or rotational (rail rollover) displacement of the rail heads as shown in Fig. I-2-3. In either case the derailment indicator is given by the condition that the track gauge b exceeds the wheelset width ℓ_w; i.e., $b_i \geq \ell_w$, i = 1 to N/2.

I-2.4 CAR-TRUCK SEPARATION

Car-Truck Separation is defined as the derailment mode that exists when the carbody centerplate disengages from the centerplate of the truck. This is expressed simply by the

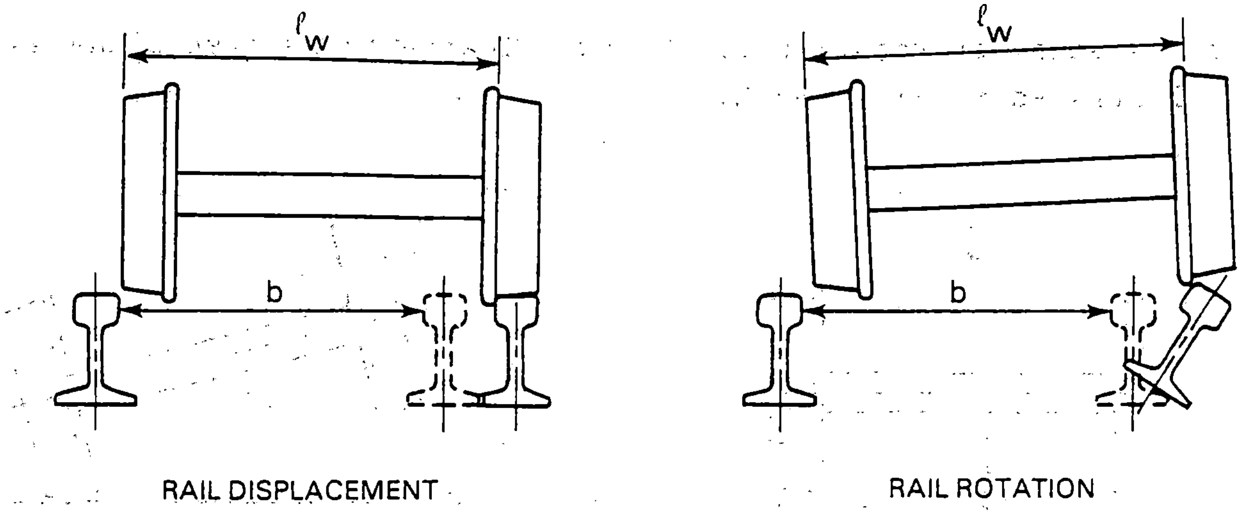

Figure I-2.3 Rail Deflection Mode: Spread and Rollover

derailment indicator $h_i \geq h_{\ell im}$, i = 1,2. Centerplate separation is illustrated in Fig. I-2.4. Car-Truck Separation can result from either excessive bounce or roll of the carbody.

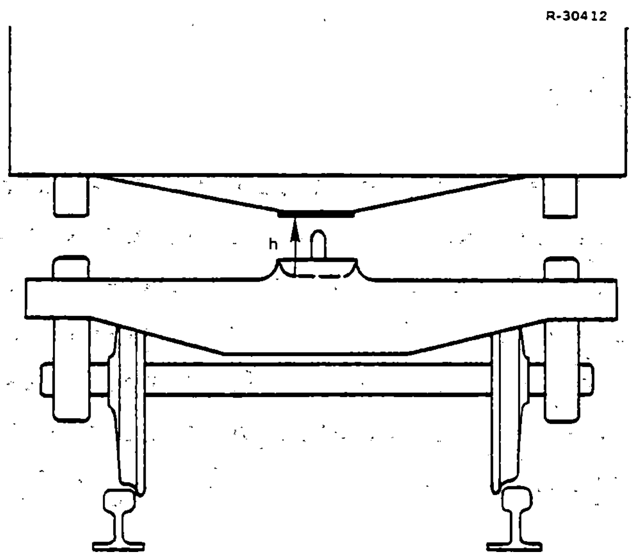

Figure I-2.4 Car-Truck Separation Derailment Mode

I-2.5 <u>SUDDEN COMPONENT FAILURE</u>

Sudden Component Failure resulting in derailment can arise in a number of ways. For the purposes of this study, this derailment mode will be said to occur whenever a force or stress in a critical vehicle or track component exceeds a limiting value. Axle breakage, for example, is a type of sudden component failure. Derailment indicators for stress or force exceedences are of the form $\sigma_i \geq \sigma_{\ell im_i}$ or $F_i \geq F_{\ell im_i}$, $i = 1$ to M, where M is the number of components.

The place of fatigue in sudden component failure deserves mention at this point. It is well known that vehicles in service undergo a gradual weakening (due to wear and fatigue) as they age, and that such deterioration is a significant consideration in design. When we focus on derailment, however, we are concerned with conditions as they are at a fixed moment in time. The parameters $F_{\ell im}$ and $\sigma_{\ell im}$ describe the current condition of the vehicle, not its history; if the strength is low, the likelihood of derailment is enhanced irrespective of the cause of the weakness. Thus from a derailment point of view, the result of fatigue, not the fatigue process itself, is of interest. Methods to relate rate of deterioration to dynamic response could be developed in ways analogous to those presented here, but do not fall within the scope of a derailment study.

The fact that a derailment indicator exceeds its threshold value does not guarantee that a derailment will occur. It does, however, mean that an unacceptably perilous condition exists, which, for the purposes of this analysis, is tantamount to derailment. An example is provided by the car-truck separation mode. It is conceivable that the carbody could rise clear of the center pin and fall back into place again, with no ill effects beyond a violent shock. This event would nevertheless be counted as a derailment in our simulation studies, and properly so: it is not feasible to precisely model the complex of factors (such as wind gusts, rolling resistance, local impact geometry, etc.) which determine the exact trajectory of the separated truck and carbody. Similar arguments apply to the choice of thresholds for other modes. As a result of using derailment indicators, one should expect to err slightly in the direction of <u>overestimating</u> derailment probability.

I-3. DERAILMENT PROCESSES

Associated with each of the five derailment modes is a discrete set of derailment processes. A process is a characteristic pattern of dynamic response which can lead to derailment. Important derailment processes include:

- Hunting
- Lateral/Vertical Response
- Steady Curving
- Spiral Negotiation
- Dynamic Curving
- Train Buckling
- Stringlining
- Pitch and Bounce
- Twist and Roll
- Response to Discontinuities (frogs, switches, rail mismatch, etc.).

These nine processes are identified with corresponding derailment modes in Table I-3-1. Each row shows the processes commonly associated with a given mode, while the columns show modes of derailment which can be expected to result from each process.

Table I-3-1 reveals that derailment modes fall naturally into two classes: those in which large forces are present at the time of derailment (wheel climb and rail deflection), and those in which large displacements are sufficient (wheel lift and car-truck separation). In either case,

TABLE I-3-1

DERAILMENT MODES AND PROCESSES

MODE \ PROCESS	HUNTING	LATERAL/YAW RESPONSE	STEADY CURVING	SPIRAL NEGOTIATION	DYNAMIC CURVING	TRAIN BUCKLING	STRINGLINING	PITCH AND BOUNCE	TWIST AND ROLL	DISCONTINUITY RESPONSE
WHEEL CLIMB	X	X	X	X	X	X	X			X
WHEEL LIFT		X		X	X			X	X	X
RAIL DEFLECTION	X	X	X	X	X	X	X			X
CAR-TRUCK SEPARATION		X		X	X			X	X	X
SUDDEN COMPONENT FAILURE	X	X	X	X	X	X	X	X	X	X

excessive forces or stresses may be expected during some portion of the response (not necessarily at the time of derailment), so sudden component failure may be caused by any of the processes identified.

Classification of derailments by mode and process provides a framework for modeling and analysis. If we wish to investigate a particular derailment mode, for example, we must assure that we have dynamic models capable of representing all the associated processes.

The following paragraphs define each process and describe how they lead to derailment.*

I-3.1 HUNTING

Hunting is defined as the unforced lateral response of the rail vehicle due to the tapered wheel profile; it can occur on tangent or curved track. Although the term is often used loosely to encompass stable but lightly damped motion, we construe hunting strictly as oscillations which are either sustained (i.e., limit cycle) or unstable. Both primary (body, low-speed) and secondary (truck, high-speed) hunting are included, the criterion being that hunting can exist on geometrically perfect track.

Hunting leads to derailment because the oscillatory motion is normally limited by impact of the flange against the rail. The resulting high forces may cause wheel climb or rail deflection.

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*Track Shift, meaning a gradual shifting of the track centerline as a result of traffic, is not itself a direct cause of derailment; it causes a deterioration of geometry over time, which in turn may contribute to derailment by any of the processes. Track buckling as a result of longitudinal stresses in the rail (from thermal expansion, traction, or braking) can indeed derail a train, but such a derailment would be more triggered by the passage of the train than caused by its dynamic response. (Because the locomotive is both the first vehicle to pass over the track and also one source of large lateral and longitudinal forces, it is likely to be the catalyst of buckling. This phenomenon is thus an important one to consider in systematic locomotive/track safety studies.) Neither shift nor buckling, therefore, can be considered vehicle derailment processes, although they are important modes of track failure.

I-3.2 LATERAL/VERTICAL RESPONSE

Forced response in the lateral/vertical plane due to track irregularities may take on a wide variety of characteristics including violent flanging and excessive carbody roll. Like hunting, but unlike twist-and-roll and pitch-and-bounce (below), this process is closely tied to the creep/conicity wheelset guidance mechanism.

I-3.3 STEADY CURVING

Negotiation of a constant radius curve at constant speed can lead to high wheel/rail forces (due to either centripetal or kinematic constraints) and thence to derailment. In principle, extremely high unbalance could cause a vehicle to tip over in wheel lift or car-truck separation, but the requisite conditions are well outside realistic limits.

I-3.4 SPIRAL NEGOTIATION

Spiral negotiation refers to passage through spirals and reverse curves at constant speed. The changing radius of curvature acts as a forcing function which sets up transient motion of the vehicle. Curve entry and exit are widely recognized as derailment trouble spots.

I-3.5 DYNAMIC CURVING

Dynamic curving refers to passage through curves in which high lateral forces may be generated between wheel and rail due to geometric irregularities. It is associated with variation in wheel load and many other vehicle factors not yet clearly identified.

I-3.6 TRAIN BUCKLING

When a train is under compressive or buff load, it tends to assume a buckled configuration within the constraints of the rails. The resulting lateral forces transmitted through the couplers may be quite large. This pair of forces acting on each car gives rise to a yaw moment, which must be resisted by lateral wheel/rail forces in addition to those due to other causes.

I-3.7 STRINGLINING

A train under tension or draft force tends to straighten out "like a stringline". This poses no problem on tangent track, provided that the coupler forces are not excessive; but in a curve, the cars may be pulled against the inner rail with considerable force. Stringlining can also occur as the result of large transient jerk actions.

I-3.8 PITCH AND BOUNCE

When rail irregularities (surface, gauge) are symmetric they lead to response in the pitch and bounce direction only. Lateral forces are low, but wheel lifts and car-truck separation can occur. The wheelset guidance mechanism does not play a part so models for pitch and bounce tend to be especially simple (unless the details of track dynamics are included).

I-3.9 TWIST AND ROLL

Large periodic roll displacements of the carbody, and attendant wheel lift, can occur when antisymmetric rail irregularities (crosslevel, alignment) come at a frequency near a suspension resonance. The American practice of staggered rail joints results in a strong antisymmetric input on bolted-joint rail, and twist and roll is therefore a serious problem at low speeds. Although creep guidance does play a role in twist and roll, it is subordinate to the periodic input and is customarily deleted from models intended to portray this phenomenon.

I-3.10 DISCONTINUITY RESPONSE

Movement over rail discontinuities produces transient loads on the wheelsets. Negotiation of turnouts, special trackwork, rail end mismatches, and other localized irregularities may cause immediate derailment by means of high forces, or may act to initiate other dynamic responses.

I-4. MODEL HIERARCHY

With an understanding of the relationships among derailment modes and processes, we seek to develop the ability to identify critical derailment conditions using mathematical descriptions of the physical system. These mathematical models are used to describe system performance for a wide variety of vehicles on different quality track and in several modes of operation.

The need to introduce a hierarchical structure of mathematical models is evident when one attempts to use existing models which have been developed to perform detailed simulations of rail systems. Most are complex and require a considerable amount of computer time. Thus, one looks to find ways of simplifying such detailed models by reducing their order and complexity.

An attendant disadvantage of this model reduction is that some of the derailment indicators described in Section 2 may not be captured by simplified models. Recall that the derailment indicators, although of simple form themselves, imply models capable of following the system through a very severe (hence complicated) time history. The conflict between efficiency and accuracy of models is resolved by introducing a <u>three-level model hierarchy</u>, (Fig. I-4-1), together with procedures for its application.

The most faithful reproduction of the physical system is "reality" itself. Thus the highest level in the hierarchy is assumed by the reality model, or instrumented field test.

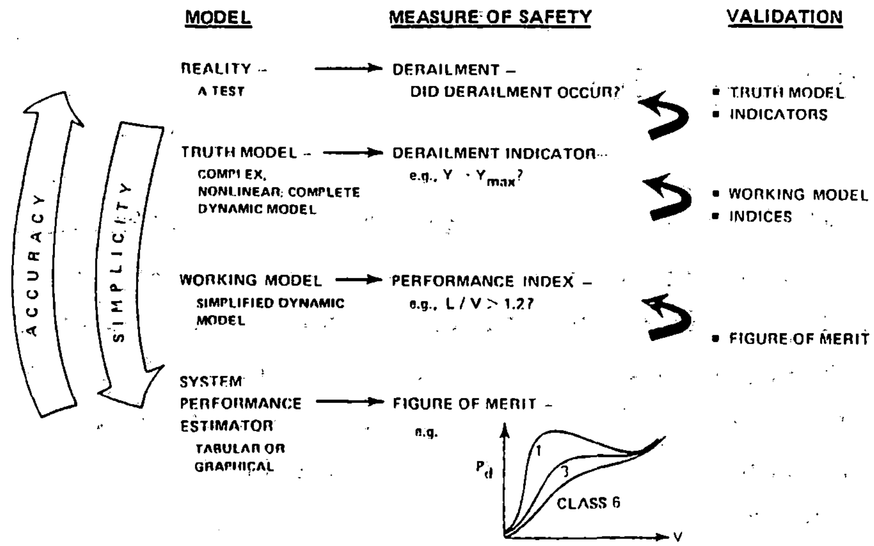

Figure I-4-1   Model Hierarchy

Such an experimental approach to studying the derailment processes is very costly to implement. Moreover, full-scale testing can lead to misleading conclusions unless it is well understood and controlled. Reality is nonetheless the basis against which subordinate models must be validated; a derailment in a test is an unquestionable fact, not merely an indication or a probability.

<u>Truth models</u> are defined as those mathematical models which represent our best efforts to characterize the dynamic system. By definition, they produce as outputs <u>derailment indicators</u>, which give a direct indication of when derailment would occur. In practice, these truth models (when they exist at all) usually suffer from long execution times especially undesirable when one is conducting parametric studies. Therefore, we are compelled to simplify the complex truth models to the point where cost-effective derailment parametric studies become feasible.

These simplifications result in <u>working models</u>, so named because they are suited to frequent use. Working models produce <u>performance indices</u>, which constitute <u>indirect</u> measures of derailment potential. A good performance index measures the magnitude or severity of one or more derailment processes, and is well correlated with the derailment indicators produced by truth models. Oftentimes we must accept performance indices as providing our only insight into the derailment cause-and-effect relationship. There are cases where a working model can produce an output which is nominally the same as derailment indicators (e.g., wheelset lateral displacement), but which is nevertheless only an approximation to the derailment indicator as computed using the truth model. If the approximation is good, then such an output is a reasonable choice for a performance index; but <u>it is still a performance index (not a derailment indicator) by virtue of its use in a working model</u>.

The hierarchical structure and supporting terminology may be illustrated by considering how the models will be used. "Reality" is assumed to represent the physical system. For example, should we want to consider the wheel climb derailment process, an appropriately instrumented test vehicle and track would represent reality. Our "truth" model, representing the best available mathematical model of this derailment process, may be a 27 degree-of-freedom nonlinear model with detailed wheel-rail geometry and creep, capable of following the flange up to the crown of the rail. A simplified version is our "working" model, which might be a 14 degree-of-freedom linearization of the equations of motion (although nonlinear working models are also perfectly acceptable). The truth model is established by detailed physical analysis, supported by measurements and tests; the working model is a simplified version which is confirmed by comparison with the truth model.

In a field test, the conditions for derailment can be discovered by carrying the test to the point of actual or impending derailment. The result of this expensive and potentially dangerous experiment would be conclusive, but could not be extrapolated to other vehicles or other track conditions. The truth model, by contrast, could be used to simulate any desired test. One of its outputs would be lateral wheel displacements, accurately simulated, which constitute a derailment indicator. A simple inspection of the displacement would suffice to determine whether a simulated derailment has occurred.

The need for a performance index now becomes clear when we observe that the linearized working model does not produce accurate values of lateral wheelset displacements beyond flange contact. However, some manipulations of the model output can yield an approximate value for flanging force. an appropriate performance index might be the ratio of lateral and vertical forces acting on the wheel. Most existing "derailment criteria", such as the L/V ratio, are really performance indices. A most important, but difficult, step is to identify or define the performance index which best characterizes the derailment potential for each of the derailment modes. Both experiment and analysis are required to correlate a candidate performance index with a derailment indicator. This issue is discussed further in Section I-6.

I-5. PERFORMANCE INDICES AND KEY FACTORS IN DERAILMENT PREDICTION

Developing and refining performance indices demands a thorough knowledge of potential derailment modes and processes, and of the dynamic interaction of track, truck, and carbody components. The block diagram shown in Fig. I-5-1 graphically illustrates the interrelationships of these important elements. The dynamic system includes the rail vehicle and the track over which it travels. The system is described by a set of differential equations: customarily, one set of equations describes the vehicle dynamics and another set describes the track dynamics. The dynamics of the vehicle and the track are closely coupled in both directions, as shown by the interaction blocks in the figure. The entire set of coupled dynamic equations constitutes the working model.

The set of inputs to the working model includes:

- Vehicle Parameters
- Operating Conditions
- Track Structure Characteristics
- Nominal Track Geometry
- Track Geometry Variations.

Vehicle parameters include masses, stiffnesses, damping and friction coefficients, geometry, etc. The track structure is described by its compliance (stiffness and damping) and maximum load capacity. Grade, curvature, gauge, and superelevation make up the set of nominal track geometry characteristics;

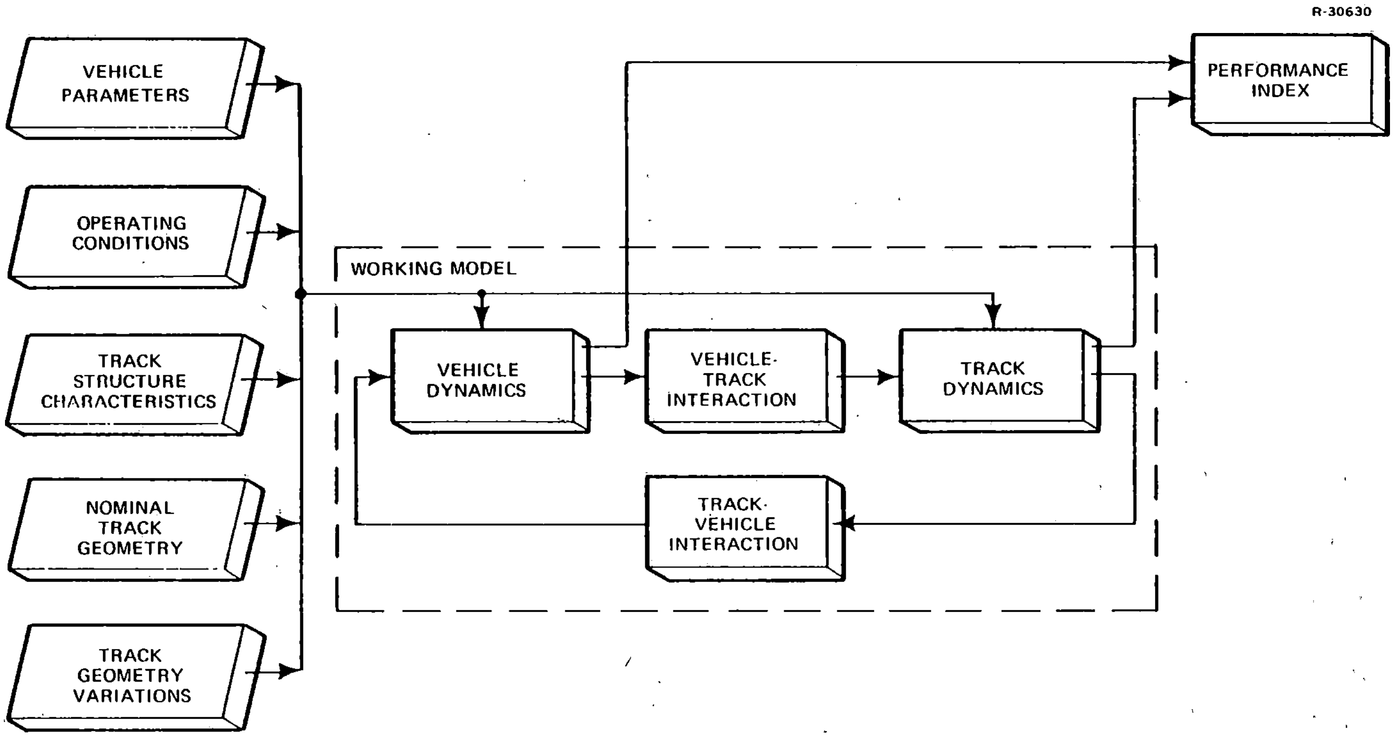

Figure I-5-1 Elements of Rail Vehicle Dynamics Analysis surface, alignment, gauge variation, and crosslevel describe variations from the nominal. Operating conditions include vehicle speed and inter-car forces describing either buff or draft conditions. The output of the working model includes all important system variables that may be needed to compute performance indices.

A major objective of this study is to develop expressions for performance indices which include, in a single functional relationship, those factors which best correlate with derailment for each derailment process. Attributes of a good performance index include simplicity, completeness, and statistical efficiency. Table I-5-1 is a tabulation of the candidate performance indices which have been identified to date from the literature, suggestions from the railroad community, and independent analysis; Table I-5-2 defines the variables. Appendix I-A contains further information on these indices, including provenance, applications, and definitions. Some of these indices can be expected to apply to one or more derailment modes as a whole; others, to only some of the processes within a mode. Associated with each index is a candidate set of <u>critical threshold values</u> which, if exceeded, imply derailment. These values,too, may differ among modes and processes, since different phenomena are at work. It should be emphasized that the performance indices of Table I-5-1 are candidates only, the validity of which must be established and compared using dynamic analyses and tests as described in Section I-6.

Table I-5-3 is a summary of the relationships among performance indices, modes, and processes. The following sections discuss selected performance indices in more detail.

TABLE I-5-1
CANDIDATE PERFORMANCE INDICES

PERFORMANCE INDEX NUMBER	FUNCTIONAL FORM
1	F_L/F_V
2	$F_L/F_V - S$
3	$v_f^2 - (2\lvert h_f - z_f\rvert F_V/M_W)\,([F_L\{-u_e - \mathrm{ctn}\,\alpha_f\}/F_V] + [1 - u_e\,\mathrm{ctn}\,\alpha_f])$
4	$(F_L/F_V)t_i - \pi S(i_w/G_w)\sqrt{M_W h_f/F_V}$
5	$V_f^2 - 2gh_f$
6	$\dfrac{2M_T\delta_F V^2\alpha_e}{gr_o b(M_T + \frac{M_B}{2})} - S$
7	$\dfrac{2F_{CD}}{g(MT + \frac{M_B}{2})}\left(\dfrac{\ell_c/R}{1 + (\ell_c/R)^2}\right)\dfrac{1}{\sqrt{1 - (\ell_T/R)^2}} - S$
8	$\dfrac{F_{CB}\sin\psi_C}{g(M_T + \frac{M_B}{2})}\left(\dfrac{\ell_C}{\ell_T}\right) - S$
9	$\left(\dfrac{z_w - z_r}{h_f}\right)$
10	$-F_{zw}$
11	$\lvert\phi_B\rvert$
12	$\dfrac{1}{2h_f}\left[\dfrac{k_{rz}z^2 + w}{g(M_T + \frac{M_B}{2})} - \dfrac{g(M_T + \frac{M_B}{2})}{k_{rz}}\right]$
13	$F_L - F_{L,\ell im}$
14	$\dfrac{F_L}{F_V} - \dfrac{\ell_V}{\ell_L}$
15	$\left(F_V\dfrac{\ell_L}{\ell_S}\right)\left(\dfrac{F_L}{F_V} - \dfrac{\ell_V}{\ell_L}\left[1 + \beta\dfrac{\ell_S}{\ell_V}\{1 - \dfrac{\beta}{1 + 2\beta}\}\right]\right)\left(1 + 2\beta\left[1 - \dfrac{\beta}{1 + 2\beta}\right]\right)^{-1} - F_{S,\ell im}$
16	$z_{cb} - z_{ct}$
17	$z_{cb} - z_{ct} + \ell_{cp}\lvert\phi_b - \phi_t\rvert$
18	$-F_{cp}$
19	$\dfrac{1}{2h_{cp}}\left[\dfrac{k_{cpz}(\Delta z_+)^2}{gM_B} - \dfrac{gM_B}{k_{cpz}}\right]$
20	$\sigma_{max} - \sigma_{\ell im}$
21	$F_{max} - F_{\ell im}$
22	$\dfrac{F_L}{F_V} - G(t_1)$
23	$a_{rms} - a_{rms,\,\ell im}$
24	$F_L - F_{L,\,\ell im}(F_{ax}, F_V)$

TABLE I-5-2

DEFINITION OF TERMS

a_{rms}	RMS acceleration, any critical point
$a_{rms,\ \ell im}$	permissible limit, any RMS acceleration
b	half track gauge
F_L	lateral force (per wheel, axle, truck, truck side, or car)
F_V	vertical force (per wheel, axle, truck, truck side, or car)
F_{CB}	coupler buff force
F_{CD}	coupler draft force
F_{CP}	upward force on carbody from truck bolster
F_{max}	maximum value, any critical load
$F_{\ell im}$	permissible limit, any critical load
$F_{L,\ \ell im}$	maximum allowable lateral force on one rail
F_{ax}	axial force in rail string
F_{zw}	upward force on wheel from rail
$F_{s,\ \ell im}$	maximum allowable spike pulling force
g	acceleration due to gravity
G_w	gauge
$G(t_i)$	graphical safety limit on L/V as function of impact time
h_f	flange height
h_{cp}	bolster centerplate depth
i_w	wheelset radius of gyration about contact point
k_{rz}	vehicle linearized rail stiffness

TABLE I-5-2

DEFINITION OF TERMS (Continued)

k_{cpz}	vertical linearized centerplate stiffness
K_t	rail torsional stiffness between adjacent ties, base clamped
K_w	rail section pure rotational stiffness, base clamped, per tie
ℓ_c	distance from car center to coupler pin
ℓ_T	distance from car center to truck centerplate
ℓ_L	vertical distance from gauge point to base of rail
ℓ_V	horizontal distance from gauge point to outer base of rail
ℓ_s	width of rail base
M_w	mass, wheelset
M_T	mass, truck
M_B	mass, carbody
R	curve radius
S	Nadal's limit, defined $(\tan \alpha_f - \mu_e)/(1 + \mu_e \tan \alpha_f)$
V	forward speed
V_f	impact velocity, lateral, flange against rail
z_f	impact point distance up flange
z_w	wheel vertical displacement
z_r	rail vertical displacement
z_{+w}	positive vertical wheelset displacement from equilibrium
z_{cb}	carbody centerplate vertical displacement

TABLE I-5-2

DEFINITION OF TERMS (Continued)

z_{ct}	truck centerplate vertical displacement
Δ_{z+}	positive vertical wheelset displacement from equilibrium
z_{cb}	carbody centerplate vertical displacement
Δ_{z+}	positive vertical car-truck relative displacement
α_e	wheelset effective conicity
α_f	flange conicity
β	defined K_t/K_w
δ_F	flange clearance (one side)
μ_e	effective flanging friction coefficient (signed)
σ_{max}	maximum value, any critical stress
$\sigma_{\Pi im}$	permissible limit, any critical stress
ϕ_B	carbody roll angle
ϕ_T	truck bolster roll angle
ψ_C	maximum one-sided coupler angle

I-5.1 THE L/V RATIO

The oldest and most firmly established performance index for derailment is the "L/V" ratio (PI#1), given by the ratio, F_L/F_V, lateral to vertical force applied to the rail by one or more wheels. A large value of F_L/F_V means that destabilizing (climbing or overturning) lateral forces dominate the stabilizing vertical force. This ratio has been commonly used

TABLE I-5-3
SUMMARY OF CANDIDATE PERFORMANCE INDICES AND THEIR APPLICATION

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MODE \ PROCESS	HUNTING	LATERAL/YAW RESPONSE	STEADY CURVING	DYNAMIC CURVING	TRAIN BUCKLING	STRINGLINING	PITCH AND BOUNCE	TWIST AND ROLL	DISCONTINUITY RESPONSE
WHEEL CLIMB	1, 2, 3, 4, 5, 6, 22	1, 2, 3, 4, 5, 22,25,26	1, 2	1, 2, 3, 4, 5, 22,25,26	1, 2, 8	1, 2, 7			1, 2, 3, 4, 5, 22,24,25,26
WHEEL LIFT		1, 9, 10, 11, 12		1, 9, 10, 11, 12			1, 9, 10, 12	1, 9, 10, 11, 12	1, 9, 10, 11, 12
RAIL DEFLECTION	1, 13, 14, 15	1, 13, 14, 15 26	1, 13, 14, 15	1, 13, 14, 15 26	1, 13, 14, 15	1, 13, 14, 15			1, 13, 14, 15 26
CAR-TRUCK SEPARATION		11, 16, 17, 18, 19		11, 16, 17, 18, 19			16, 18, 19	11, 16, 17, 18, 19	11, 16, 17, 18, 19
SUDDEN COMPONENT FAILURE	20, 21, 23	20, 21, 23	20, 21, 23	20, 21, 23	20, 21, 23	20, 21, 23	20, 21, 23	20, 21, 23	20, 21, 23

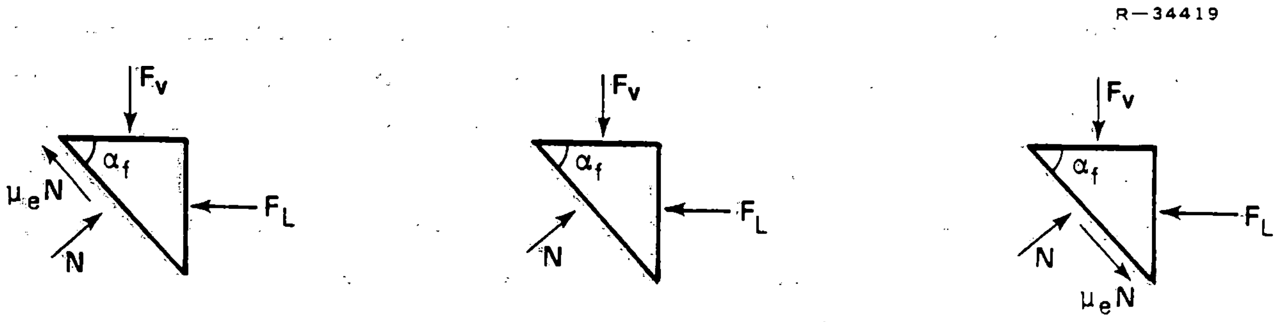
by itself in a "rule of thumb" fashion, with a value above 0.6 being generally accepted as dangerous (Ref. 3,4). Precisely which forces and thresholds are used in computing the index depends on the derailment mode: the forces are per wheel for wheel climb (typical threshold 0.9) or per truck side for rail deflection (e.g., rail rollover -- typical threshold 0.6). Track shift is a special case. Because of the large wavelength characteristic of buckled track (20-60 ft), the net forces per car are more appropriate in this case.

More important than its role as a self-contained performance index, however, is the prominent place of the L/V ratio as a component of other indices.

A well-known performance index for wheel climb is Nadal's formula (PI#2), which compares L/V to the expression

$$S = \frac{\tan \alpha_f - \mu_e}{1 + \mu_e \tan \alpha_f} \qquad \text{(I-5-1)}$$

where α_f is the cone angle of the flange, and μ_e is the effective coefficient of friction acting between flange and rail. In Nadal's original analysis (Ref. 5), μ_e was taken as μ (the ordinary sliding friction coefficient) for positive angle of attack (i.e., the wheelset yawed so as to roll into the rail); $-\mu$ for negative angle of attack; and zero for zero angle of attack (i.e., glancing). Later work (Ref. 6, 7) has resulted in more complicated expressions for μ_e, which nonetheless approach Nadal's limits for large angles of attack. The convergence is fairly rapid, and Nadal's formula with $\mu_e = +\mu$ has been suggested as a close, conservative approximation (Ref. 8, 9). Nadal's formula may be derived very simply by considering the requirements for equilibrium on an inclined plane, remembering that because of the rolling motion of the wheelset, the direction of the friction force may be either up or down the rail depending upon the angle of attack (Fig. I-5-2).



(a) POSITIVE ANGLE OF ATTACK

(b) ZERO ANGLE OF ATTACK

(c) NEGATIVE ANGLE OF ATTACK

Figure I-5-2   Force Diagrams for Nadal's Formula

Another performance index involving the L/V ratio (PI#14) is applicable to rail deflection (specifically roll-over):

$$\frac{F_L}{F_V} - \frac{\ell_V}{\ell_L} \qquad \text{(I-5-2)}$$

where ℓ_V and ℓ_L are respectively the vertical and lateral moment arms between the point of application (presumably near the gauge side of the rail head) and the center of rotation (presumably at the field side of the base). Figure I-5-3 illustrates that when this index exceeds zero, an unrestrained rail will tip over. For many rail sections, ℓ_V/ℓ_L is on the order of 0.6 in flanging conditions. F_L and F_V here are on a per-truck-side basis. Use of this threshold value is probably too restrictive, since actual rail is torsionally stiff and restrained by fasterners all along its length. Other estimates of the critical threshold levels for track shift and

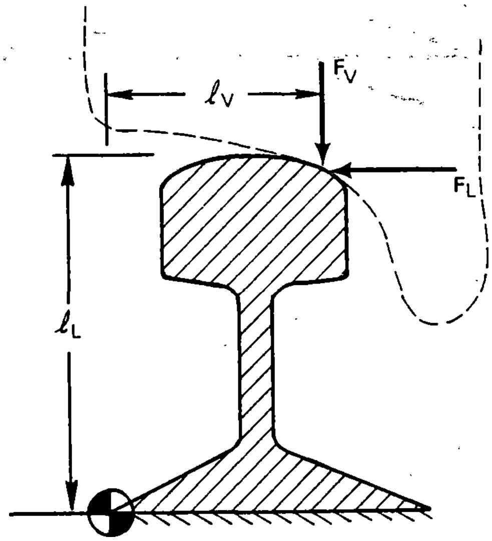


Figure I-5-3 Geometry of Rail Rollover

rail rollover expressed as F_L and F_V are given in Ref. 10. Ref. 10 also breifly summarizes previous results that have been obtained using scale model testing for critical steady L/V ratios as a function of wheelset angle of attack and friction coefficients.

PI#24 is an index for track shift (buckling) derailments. It is an empirical form relating the lateral force necessary to cause catastrophic buckling to the vertical (stabilizing) and axial (destabilizing) loads, per car. Preliminary steps toward such an empirical relation have been taken elsewhere (Ref. 1, 2); it can be expected to resemble Fig. I-5-4 qualitatively.

In summary, the L/V ratio is a useful performance index itself, but it is most valuable as a component of more complex indices. The next sections illustrate some of these.

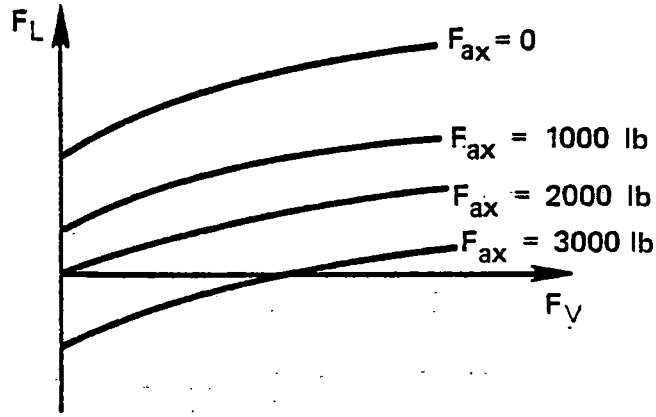


Figure I-5-4 Empirical Relation for Track Buckling (Hypothetical)

I-5.2 WHEEL CLIMB DURING IMPACT *

Nadal's formula is essentially a criterion for static sliding instability. In many situations, such as hunting and response to discontinuous track, the flange impacts the rail with considerable momentum. This momentum tends to drive the flange further up the rail than the static analysis would indicate, so there have been attempts to include some dynamic effects in wheel climb derailment indices.

One such index (PI#3) is derived using an energy conservation approach and the geometry of Fig. I-5-2. The wheelset is assumed to have an effective mass of M_w, and the requirement to absorb its energy before the flange passes the top of the rail sets a limit on the flanging velocity. PI#4, due to Matsudaira of the Japanese National Railways (Ref. 11), is momentum based. It requires an estimate of the impact duration, t_i, which is reported to be near 0.01 sec. Empirically based indices, such as PI#22, can also be very useful.

* Before developing any performance indices for wheel climb, the most resent efforts of L.M. Sweet's scaled model experiments and analysis should be reviewed. See reference number 44 in Section C.

This index compares the L/V ratio to an experimental curve such as the one shown in Fig. I-5-5 from JNR tests (Ref. 11). In this case, the curve shows safe L/V ratio as a function of impact duration. PI#5 is a very simple index for impact, dealing with the total energy available and neglecting friction and other external forces.

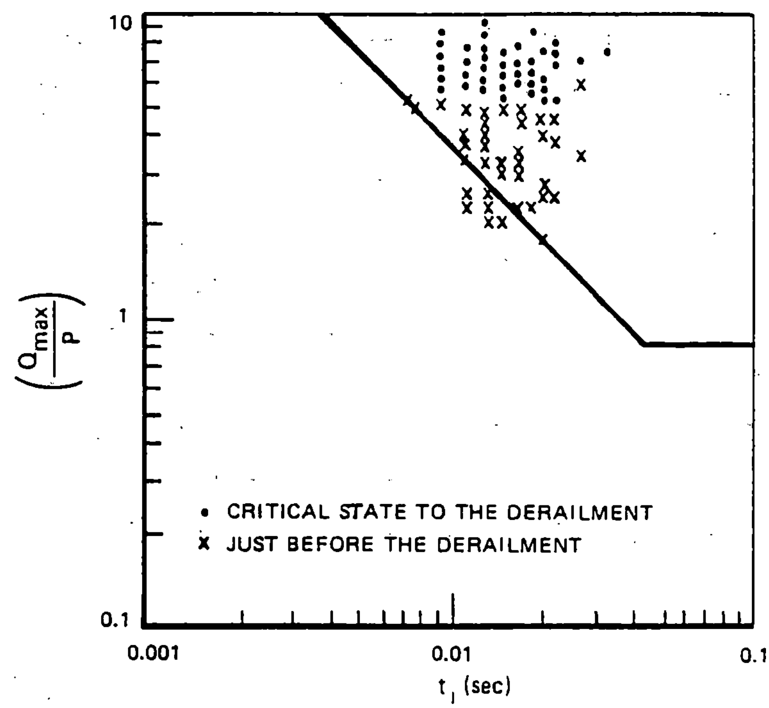


Figure I-5-5 Experimental Safety Curve (JNR, Ref. 11)

Notice that the L/V ratio appears explicitly in each of these indices. Observe also that they are applicable to all wheel climb processes except those which are inherently quasi-static (steady curving, buckling, and stringlining).

I-5.3 WHEEL CLIMB FOR PARTICULAR PROCESSES

The performance indices presented up to this point apply to any wheel climb derailment process, the only distinction being between static and dynamic behavior. It is unimportant whether a given value of the L/V ratio, for example, arises because of hunting or response to a discontinuity. On the other hand, some processes are well enough defined to permit estimates of forces and displacements to be made without detailed simulation. When these quantities are substituted in the general performance indices, we obtain new, special-purpose indices which are relatively easy to use but restricted in applicability.

PI#6, for example, is a special case of Nadal's formula (PI#2) for hunting. It is assumed that hunting is sinusoidal at the single-wheelset kinematic wavelength, that the mass of the oscillating body is half the vehicle mass, and that the amplitude is equal to the flange clearance. Based on these assumptions, it is easy to estimate the peak lateral force. Obviously the effects of violent flanging are not captured.

The next two indices (I#7 and #8) are for wheel climb due to buff and draft, again using Nadal's formula. Derailment is assumed to be due only to coupler forces in both cases. In PI#7, the car is assumed to be in a plane curve at low speed (so that centripetal effects can be neglected), coupled to other identical cars in a stringlining situation. In PI#8, the car is in a buckled condition on tangent track with the couplers transmitting axial force at their extreme positions. In each case, the force per truck side is calculated and combined with PI#2. Figure I-5-6 illustrates both cases; the actual situation is usually more complicated and may be either more or less severe.

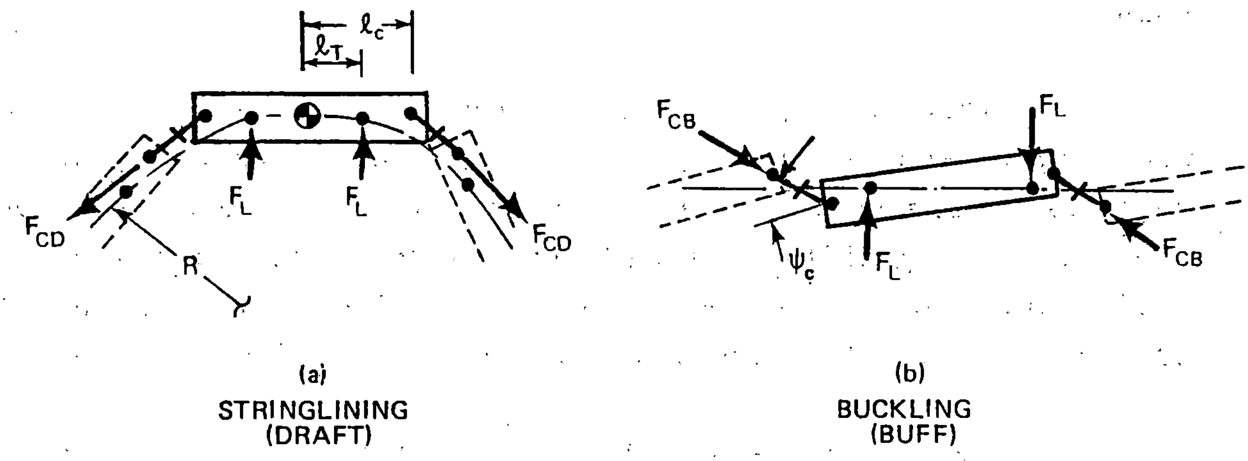


Figure I-5-6 Geometry for Buff and Draft Calculations

I-5.4 WHEEL LIFT INDICES

Wheel lift is defined by the wheel flange passing above the rail. It is a low-force mode of derailment (in contrast to wheel climb, which requires high lateral forces), and the applicable performance indices tend to be relatively simple in form due to there being no need to include friction and impact effects. This is not to say that the dynamic models used to evaluate the indices are necessarily simple, but only that the algebra performed on the model output is minimal.

One possible performance index (PI#9) is formally identical to the derailment indicator in that it measures the distance of the wheel above the rail. It is a performance index by virtue of its being computed from the output of a working model which permits wheel-rail separation -- it is thus an approximation of the derailment indicator. The same concept applies in principle for other modes as well, but it is difficult to imagine a working model which could approximate the derailment indicators of the high-force modes (wheel climb and rail deflection) in the vicinity of their critical values.

PI#10 is applicable to models which do not allow wheel-rail separation (e.g., linearized models which treat the rail as a spring). If the contact force becomes negative, wheel lift is possible. PI#12 extends this idea, using energy equivalence to relate bounce height obtained with a push-pull rail spring to the corresponding height assuming separation.

A major cause of wheel lift derailment is excessive roll of the carbody, resulting in unloading one side of the truck. A possible performance index applicable to processes involving roll is simply the carbody roll angle (PI#11). As a rule of thumb, 9° roll to one side is frequently said to be an unsafe condition.

I-5.5 RAIL DEFLECTION INDICES

The rail deflection derailment mode has been defined as deformation of the track to the extent that a wheel or wheelset falls between the rails. As explained in Section 2, rail spread and rail rollover are two manifestations of this mode. Rail deflection usually involves large values of lateral force, and PI#13 is a simple comparison of F_L to a threshold. Because vertical force does not enter this index, it is likely to be most applicable to shearing failures of the rail or its support.

PI#14, an index for tipping of an unrestrained rail, has been discussed above. In an effort to model the true situation more accurately, PI#15 was derived using a discrete, five-segment model of the rail and spikes. It includes (in the parameter β) torsion of the rail along its length; the effect of missing spikes may be approximated by reducing β. When the peak roll-over force exceeds the capacity of the spike, derailment is declared. Estimates of threshold values of rail rollover are summarized in Ref. 10 for a number of rail restraint conditions and analysis assumptions.

Among the effects which are difficult to include in manageable performance indices are: combined loading from multiple axles; state of stress in the rail, including axial load; rail and track dynamics; and joints. These might be accomodated by the use of empirical correction factors.

I-5.6 CAR-TRUCK SEPARATION INDICES

Car-truck separation is a low-force mode with many similarities to wheel lift. A derailment is declared whenever the carbody and truck bolster centerplates disengage. PI#16 is the vertical separation between centerplates, formally identical to the derailment indicator. The critical threshold value of 1.25 inches represents the depth of the bowl; the centerpin would remain engaged for several more inches, but would be unable to sustain significant shear. PI#17 is a more conservative index which measures whether the centerplates have disengaged on one side only. It, along with PI#11 (carbody roll angle) is applicable to processes which are not purely vertical. Two other performance indices (PI#18 and #19) are precisely analogous to PI#10 and #12 for wheel lift.

I-5.7 SUDDEN COMPONENT FAILURE INDICES

Physical failure of key vehicle or track components, other than those classified as rail deflection, constitute sudden component failure derailments. The derailment indicators are comparisons of force or stress levels with allowable limits at each critical location. These locations and the critical conditions for each must be determined by analysis of the specific design in question, so it is futile to particularize here. Threshold failure levels should be set low enough to allow for fatigue aging.

Two indices for component failure (PI#20 and #21) take the form of derailment indicators. A third (PI#23) uses RMS acceleration levels -- possibly weighted in some way -- to assess the severity of the vibration environment (Ref. 12).

I-6. SELECTION OF PERFORMANCE INDICES AND THRESHOLD VALUES

The performance indices discussed in the last section are only candidates for evaluation. Although they are all postulated on physical grounds, further analysis is required to

- Set appropriate threshold values for each candidate index, and
- Select the most predictive index for each mode and process.

This section describes the general approach to both facets of the problem. The hierarchy of dynamic models is invoked as a basis for selection.

I-6.1 DETERMINATION OF THRESHOLDS

Associated with each candidate performance index is a critical threshold value: when the index exceeds this threshold, a derailment is declared. Many of the indices in Table I-5-1 are arranged so that the threshold value is, by definition, unity or zero. In these cases there is another factor in the index (a limiting force or displacement, an effective coefficient, etc.) whose value is uncertain and which is thus analogous to an unkown threshold. In any event, we have for each candidate performance index an adjustable parameter which we are free to choose so that the best possible correlation between the index and the derailment indicator is achieved.

Define D and P as the Boolean variables which are true when derailment is declared by a derailment indicator and by a particular performance index, respectively. Then we seek the threshold which minimizes the probability of error, i.e.,

$$\min P_e = \min \Pr (P\times\bar{D}) + (\bar{P}\times D) \qquad \text{(I-6-1)}$$

where overbars denote complements, and the operators are Boolean. This criterion places equal weight on "misses" ($\bar{P}\times D$) and "false alarms" ($P\times\bar{D}$).

The probabilities required for the minimization of Eq. I-6-1 are developed from a set of validation runs using the truth model. Recall that the truth model is capable of producing the derailment indicators directly, as well as any necessary performance indices; usually, in fact, several performance indices and derailment indicators are output simultaneously by a truth model. By exercising the truth model for a variety of cases (parameters, inputs, and initial conditions), it is possible to build up a pair of probability density functions for the value of the performance index. One distribution, designated $p(PI|\bar{P})$, is for all times when the derailment indicator is not true; the other, $p(PI|D)$, for times when derailment is declared. Figure I-6-1 shows hypothetical distributions. A test value of V_o for the performance index threshold value is illustrated. Since areas under the probability density function represent probabilities, it is easily shown that the sum of the two shaded areas in the figure is the probability of error. All that remains is a simple one-dimensional optimization problem to find the V_o which minimizes P_e. P_e is the complementary level of confidence associated with the index. This is a simple special case of parameter identification (Ref. 13).

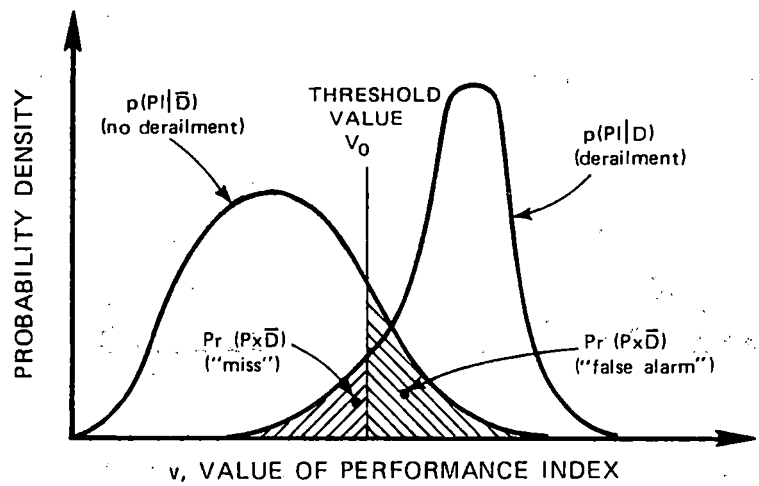


Figure I-6-1 Hypothetical Performance Index Probability Densities

I-6.2 SELECTION OF BEST INDEX

Typically we have several candidate performance indices for each mode-process pair. The critical threshold value for each can be independently optimized as above to minimize P_e. Now, to choose among the candidates, we have only to select the performance index with minimum probability of error. As illustrated in Fig. I-6-2, a good performance index is one for which the two conditional probability density functions are maximally distinct.

This procedure must in general be carried out for each process and mode (33 pairs, from Table I-3-1). Because there are typically substantially fewer truth models than such pairs, however, the amount of computation is somewhat less than might be anticipated.

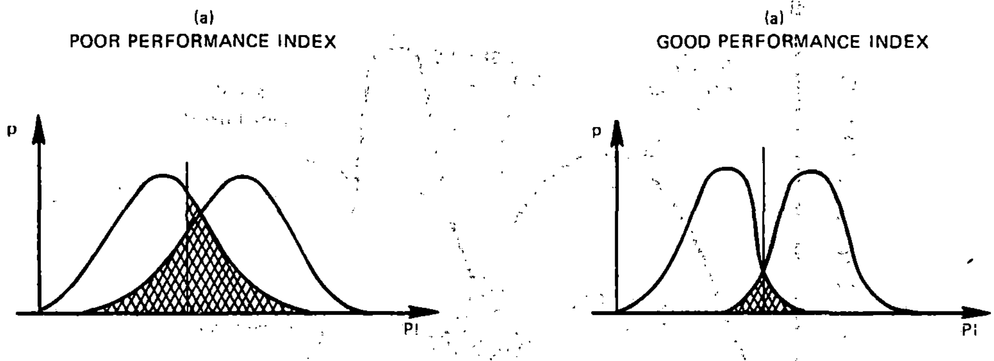


Figure I-6-2 Selection of Performance Index

I-6.3 APPLICATION OF MODEL HIERARCHY

The preceding sections have demonstrated how the truth model, with its capability to generate both derailment indicators and performance indices, can be used to validate, calibrate, and select performance indices. The truth model's complexity is justified by the fact that it is only used a relatively few times. Working models can then be used for the parametric studies themselves, with the assurance that the selected performance inices are good predictors or estimates of derailment indicators.

This is but one aspect of a larger validation problem, however. The derailment indicators must themselves be checked against full-scale tests ("reality") to confirm that they correctly predict derailment. Furthermore, the dynamic models at each level must be compared against those at the next higher level to assure their adequacy. Figure I-6-3 is a schematic representation of the validation process. Three important points are:

- Validation is strictly hierarchical, each level being the arbiter of the next.
- Final authority rests with the highest level present: field tests if they exist, truth models otherwise, and so on down the hierarchy.
- Both models and measures of safety require validation.

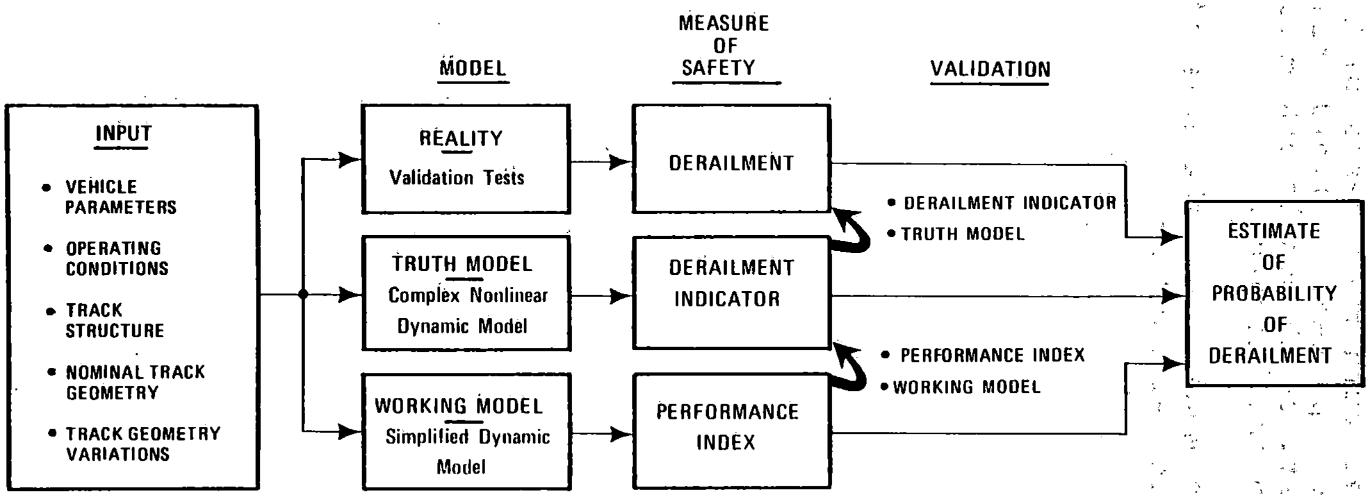


Figure I-6-3   Model Hierarchy and Validation

I-7. CONCLUSIONS

This report has set forth a number of candidate performance indices which can be assessed for their ability to predict derailment of locomotives and cars. It has been shown that performance indices cannot be considered in a vacuum, but must be related to their intended use of considerations of physical phenomena, dynamic models, and validation procedures. Each of these elements has been discussed within the framework of model hierarchy.

Derailment can be said to occur by one of the following five fundamental modes:

- Wheel Climb
- Wheel Lift
- Rail Deflection
- Car-Truck Separation
- Sudden Component Failure

These modes describe the state of affairs at the instant the derailment occurs. By contrast, a derailment process refers to a more general pattern of response leading up to derailment. The following processes have been identified:

- Hunting
- Lateral/Vertical Response
- Steady Curving

- Dynamic Curving
- Train Buckling
- Stringlining
- Pitch and Bounce
- Twist and Roll
- Response to Discontinuities

Derailment, by any mode or process, is an extremely complicated phenomenon. The dynamic behavior of a vehicle changes rapidly and dramatically as the derailment proceeds, and models of the kind commonly used to investigate more benign response regimes are incapable of following this behavior. Nevertheless, it is possible to develop truth models which can approximate the actual system quite well up to the very point at which derailment becomes inevitable. Because of their sophistication, truth models can yield relatively simple derailment indicators which make it apparent whether a derailment would occur in a given situation. Working models are simplifications of truth models, introduced because truth models tend to be too complex for frequent use. Because they cannot produce reliable estimates of derailment indicators, it is necessary to process working model output to form performance indices. Performance indices are thus indirect measures of safety; they may assume a wide variety of forms, of which one is the familiar L/V ratio. Table I-5-1 and Appendix A contain other examples.

The topic validation has been discussed in Section I-6. We showed there that for this study, two parallel components of validation are important:

- <u>Models</u> must be compared against more complex models (or experiments) to demonstrate that their responses agree within acceptable error

- <u>Performance indices</u> and other indirect measures of derailment must be confirmed in relationship to more direct measures.

There has been relatively little work to date in the latter area of validation. It is nevertheless equally as important as model validation when the objective is a determination of safety.

Appendix I-A is a list of candidate performance indices with data on their sources and areas of proposed application. No attempt has been made here to draw distinctions as to the various indices' validity, for three reasons: first, there has been very little systematic theoretical or experimental study to establish performance indices, so that most are conjectural at this time; second, the index of choice may vary according to mode and third, there exists a tradeoff among model simplicity, index simplicity, and accuracy which will vary according to the application. This list of candidate performance indices is, of course, open-ended. The proposed performance indices have <u>not</u> been validated through full scale testing programs. They are presented to demonstrate the interactions of the various factors affecting them and how they could be developed.

APPENDIX I-A

TABULATION OF CANDIDATE PERFORMANCE INDICES

This appendix contains a detailed summary of the candidate performance indices (PI's) described in Section I-5. A standard form, shown in Fig. I-A-1, has been adopted to allow a compact presentation of the following data:

- Functional form of the index
- Numerical reference code
- Brief description of the physical basis of the index
- Source, if not trivial
- Applicable modes and processes
- Critical threshold values
- Necessary definitions.

Where a single threshold value applies to an entire derailment mode, it is entered beside that mode and extended with an arrow; otherwise, threshold values appear beside the appropriate processes. Processes to which a PI does not apply are excluded by an "X".

A blank form is provided for the reader's use in proposing additional indices.

REFERENCE NUMBER

FUNCTIONAL FORM

PHYSICAL BASIS

SOURCE IN LITERATURE

PI #	
4	$(F_L/F_V)t_i - \pi S\,(i_w/G_w)\,\sqrt{m_w h_f/F_v}$
BASIS	IMPACT OF ROTATIONAL BODY
SOURCE	JNR (Ref. 11,9)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR-TRUCK SEP.		WHEEL CLIMB	0	RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
				TRACK SHIFT				STRINGLINING	
								DISCONT. RESP.	

THRESHOLD VALUE FOR ENTIRE MODE

INAPPLICABLE PROCESS

Definitions: t_i = duration of impact

i_w = radius of gyration about contact, wheelset

G_w = gauge

DEFINITIONS NEW TO THIS PI

Figure I-A-1   Form for Performance Index Tabulation

TABLE I-A-1

<u>PERFORMANCE INDEX TABULATION *</u>

* The following proposed performance indices have <u>not</u> been validated through full scale testing programs. They are presented to demonstrate the interactions of the various factors affecting them and how they could be developed.

PI #	F_L/F_V
1	
BASIS	Rail: Static Moment Balance Climb: Static Force Balance
SOURCE	Various (Ref. 3,9)

CRITICAL THRESHOLD VALUE

WHEEL LIFT	.8	CAR · TRUCK SEP.		WHEEL CLIMB	.9	RAIL DEFLECTION	.6	SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS F_L = lateral force } per wheel for lift and climb,
F_V = vertical force } per truck side for deflection

PI #	F_L/F_V
2	
BASIS	Static Force Balance
SOURCE	Nadal (Ref. 5)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR · TRUCK SEP.		WHEEL CLIMB	0	RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS $S = (\tan \alpha_f - \mu_e \tan \alpha_f$

α_f = flange angle

μ_e = effective coefficient of friction; function of angle of attack

PI #	
3	$V_f^2 - (2[h_f - z_f]F_V/M_W)([F_L\{-\mu_e - ctn\alpha_f\}/F_V] + [1-\mu_e ctn\alpha_f])$
BASIS	Energy to decelerate mass following impact.
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR · TRUCK SEP.		WHEEL CLIMB	0	RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT. RESP.		LAT/VERT RESP.		LAT/VERT RESP.	↓	LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING	↓	HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING	x	STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING	↓	DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING	x	BUCKLING		STEADY CURVING	
				STRINGLINING	x	STRINGLINING		DYN. CURVING	
				DISCONT. RESP.	↓	DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS V_f = lateral velocity of wheelset at flange impact

h_f = height of flange

z_f = height of flange of impact

M_W = wheelset effective mass

PI #	
4	$(F_L/F_V)t_i - \pi S(i_W/G_W)\sqrt{M_W h_f/F_V}$
BASIS	Momentum of wheelset in impact.
SOURCE	JNR (Ref. 11,9)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR · TRUCK SEP.		WHEEL CLIMB	0	RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.	↓	LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING	↓	HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING	x	STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING	↓	DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING	x	BUCKLING		STEADY CURVING	
				STRINGLINING	x	STRINGLINING		DYN. CURVING	
				DISCONT. RESP.	↓	DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS t_i = duration of impact

i_W = radius of gyration about contact, wheelset

G_W = gauge

PI #	$V_f^2 - 2gh_f$
5	
BASIS	Energy conservation, impact, no external forces
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB	0	RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING	↓	HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING	x	STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING	↓	DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING	x	BUCKLING		STEADY CURVING	
				STRINGLINING	x	STRINGLINING		DYN. CURVING	
				DISCONT. RESP.	↓	DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

PI #	$\dfrac{2M_T\delta_F V^2\alpha_e}{g r_o b(M_T+\frac{M_B}{2})} - S$
6	
BASIS	Nadal's formula (#2) for peak force in hunting, per wheel
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING	0	HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS
m_T = truck mass with wheelsets
m_B = carbody mass
δ_F = single-sided flange clearance
α_e = effective conicity
r_o = wheel radius
b = half gauge

PI #	$\frac{2F_{CD}}{g(M_T + \frac{M_B}{2})} \left(\frac{\ell_c/R}{1 + (\ell_c/R)^2} \right) \frac{1}{\sqrt{1 - (\ell_T/R)^2}} - S$
7	
BASIS	Nadal's formula (#2) for high draft force curving, per truck side
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING	0	STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS F_{CD} = coupler draft force R = radius of curvature.
ℓ_c = distance car center to coupler pin
ℓ_T = distance car center to centerplate

PI #	$\frac{F_{CB}\sin\psi_C}{g(M_T + \frac{M_B}{2})} \left(\frac{\ell_C}{\ell_T} \right) - S$
8	
BASIS	Nadal's formula (#2) for high buff force curving, per truck side
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING	0	BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS F_{CB} = coupler buff force
ψ_C = maximum one-sided coupler angle

PI #	$\left(\frac{z_w - z_r}{h_f}\right)$
9	
BASIS	Geometry, wheel above rail by flange height. Like indicator.
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT	1	CAR-TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.	↓	LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE	↓	PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING	↓	TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.	↓	DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL	↓	DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS z_w = wheel vertical displacement

z_r = rail vertical displacement

PI #	$- F_{zw}$
10	
BASIS	Vertical wheel/rail force should be positive
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT	0	CAR-TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.	↓	LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE	↓	PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING	↓	TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.	↓	DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL	↓	DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS F_{zw} = upward force on wheel from rail

PI #	$\lvert \phi_B \rvert$
11	
BASIS	Carbody roll causing unloading
SOURCE	Field experience, various sources (Ref. 3,9)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.	9^{o}	LAT/VERT RESP.	15^{o}	LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING	9^{o}	TWIST/ROLL	15^{o}	STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.	9^{o}	DYN. CURVING	15^{o}	DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL	9^{o}	DISCONT. RESP.	15^{o}	BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS ϕ_B = carbody roll angle

PI #	$\frac{1}{2h_f}\left[\frac{k_{rz} z_w^2}{g(M_T + \frac{M_B}{2})} - \frac{g(M_T + \frac{M_B}{2})}{k_{rz}}\right]$
12	
BASIS	Energy equivalence: convert linear ail spring results to bounce height
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT	1	CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT. RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS k_{rz} = vertical linearized rail stiffness

z_{+w} = positive vertical wheelset displacement from equilibrium

PI #	$F_L - F_{L,\ell im}$
13	
BASIS	Shear mode failure of rail, fasteners, tie, ballast, per truck side
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION	0	SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS $F_{L,\ell im}$ = maximum allowable lateral force on one rail

PI #	$\frac{F_L}{F_V} - \frac{\ell_V}{\ell_L}$
14	
BASIS	Moment balance, tipping, unsupported rail section, per truck side
SOURCE	Various (Ref. 3,9)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION	0	SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS ℓ_V = horizontal distance from gauge point to outer base of rail
ℓ_L = vertical distance from gauge point to base of rail

PI #	
15	$\left(F_V \frac{\ell_L}{\ell_S}\right)\left(\frac{F_L}{F_V} - \frac{\ell_V}{\ell_L}\left[1 + \beta \frac{\ell_S}{\ell_V}\left(1 - \frac{\beta}{1+2\beta}\right)\right]\right)\left(1 + 2\beta\left[- \frac{\beta}{1+2\beta}\right]\right)^{-1} - F_{s,\ell im}$
BASIS	Discrete model of spike pull, symmetric, five ties
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION	0	SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS ℓ_s = width of rail base

$\beta = K_t/K_w$

K_t = rail torsional stiffness between adjacent ties, base clamped

K_w = rail section stiffness per tie, base clamped

$F_{s,\ell im}$ = maximum allowable spike pulling force

PI #	
16	$Z_{cb} - Z_{ct}$
BASIS	Geometry: centerplates not mating, unable to support shear; like indicator
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.	125"	WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS z_{cb} = vertical displacement of car centerplate

z_{ct} = vertical displacement of truck centerplate

PI #	
17	$z_{cb} - z_{ct} + \ell_{cp} \lvert \phi_b - \phi_t \rvert$
BASIS	Geometry: centerplates not mating at one edge only
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR-TRUCK SEP.	.125"	WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE	x	HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS ℓ_{cp} = centerplate diameter

ϕ_b = carbody roll

ϕ_t = truck bolster roll

PI #	
18	$-F_{cp}$
BASIS	Vertical car-truck force should be positive
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR-TRUCK SEP.	0	WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS F_{cp} = upward force on carbody from truck bolster

PI #	$\frac{1}{2h_{cp}}\left[\frac{k_{cpz}(\Delta z_+)^2}{gM_B} - \frac{gM_B}{k_{cpz}}\right]$
19	
BASIS	Energy equivalence: convert linear spring results to bounce height
SOURCE	TASC

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.	1	WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS h_{cp} = bolster centerplate depth (approx. 1.25 in)

h_{cpz} = vertical linearized centerplate stiffness

Δz_+ = positive vertical car-truck relative displacement

PI #	$\sigma_{max} - \sigma_{\ell im}$
20	
BASIS	Failure; like indicator
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	0
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS σ_{max} = maximum value, any critical stress

$\sigma_{\ell im}$ = permissible limit on stress

PI #	
21	$F_{max} - F_{\ell im}$
BASIS	Failure; like indicator
SOURCE	-

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR-TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	0
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	↓
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	↓
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	↓
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	↓
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	↓
				STRINGLINING		STRINGLINING		DYN. CURVING	↓
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	↓
								STRINGLINING	↓
								DISCONT. RESP.	↓

<u>DEFINITIONS</u> F_{max} = maximum value, any critical load

$F_{\ell im}$ = permissible limit on load

PI #	
22	$\frac{F_L}{F_V} - G(t_i)$
BASIS	Impact Experiments and PI#4
SOURCE	JNR (Ref. 6,7)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR-TRUCK SEP.		WHEEL CLIMB	0	RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.	↓	LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING	↓	HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING	x	STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING	↓	DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING	x	BUCKLING		STEADY CURVING	
				STRINGLINING	x	STRINGLINING		DYN. CURVING	
				DISCONT. RESP.	↓	DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

<u>DEFINITIONS</u> $G(t_i)$ = graphical safety limit as function of impact time.

PI #	$a_{rms} - a_{rms,\ell im}$
23	
BASIS	Vibration environment
SOURCE	ACORN (Ref. 12)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR · TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	0
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	↓

DEFINITIONS a_{rms} = acceleration (weighted) Rms value, specified point

$a_{rms,\ell im}$ = permissible limit on a_{rms}

PI #	$F_L - F_{L,\ell im}(F_{ax},F_v)$
24	
BASIS	Track buckling induced by lateral force (per car)
SOURCE	Chessie, Princeton, ACORN (Refs. 1,2,12)

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR · TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION	0	SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.	↓	BUCKLING	
								STRINGLINING	
								DISCONT. RESP.	

DEFINITIONS $F_{L,\ell im}(\ )$ = empirical function for onset of buckling

F_{ax} = axial force in rail

PI #	
BASIS	
SOURCE	

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
				TRACK SHIFT				STRINGLINING	
								DISCONT. RESP.	

PI #	
BASIS	
SOURCE	

CRITICAL THRESHOLD VALUE

WHEEL LIFT		CAR - TRUCK SEP.		WHEEL CLIMB		RAIL DEFLECTION		SUDDEN COMPT. FAILURE	
LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT RESP.		LAT/VERT. RESP.	
PITCH/BOUNCE		PITCH/BOUNCE		HUNTING		HUNTING		PITCH/BOUNCE	
DYN. CURVING		TWIST/ROLL		STEADY CURVING		STEADY CURVING		HUNTING	
DISCONT. RESP.		DYN. CURVING		DYN. CURVING		DYN. CURVING		TWIST/ROLL	
TWIST/ROLL		DISCONT. RESP.		BUCKLING		BUCKLING		STEADY CURVING	
				STRINGLINING		STRINGLINING		DYN. CURVING	
				DISCONT. RESP.		DISCONT. RESP.		BUCKLING	
				TRACK SHIFT				STRINGLINING	
								DISCONT. RESP.	

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