REPORT NO. FRA/ORD-79/36

TDOP

TRUCK DESIGN OPTIMIZATION PROJECT PHASE II

ANALYTICAL TOOL ASSESSMENT REPORT

WYLE LABORATORIES
SCIENTIFIC SERVICES & SYSTEMS GROUP

Colorado Springs Division
4620 Edison Avenue
Colorado Springs, Colorado 80915

AUGUST 1979

Document is available to the U.S. public through
the National Technical Information Service
Springfield, Virginia 22161

Prepared for
U.S. DEPARTMENT OF TRANSPORTATION
Federal Railroad Administration
Office of Research and Development
Washington, D.C. 20590

NOTICE

This document is disseminated under the sponsorship of the U.S. Department of Transportation in the interest of information exchange. The United States Government assumes no liability for the contents or use thereof.

NOTICE

The United States Government does not endorse products or manufacturers. Trade or manufacturer's names appear herein solely because they are considered essential to the object of this report.

Technical Report Documentation Page

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
FRA/ORD-79/36		

4. Title and Subtitle	5. Report Date
TRUCK DESIGN OPTIMIZATION PROJECT PHASE II Analytical Tool Assessment Report	August 1979
	6. Performing Organization Code
	8. Performing Organization Report No.
7. Author(s) L. Johnson, A. Gilchrist, M. Healy, C. Bush, G. Sheldon	TDOP Technical Report TR-06
9. Performing Organization Name and Address Wyle Laboratories Scientific Services & Systems Group 4620 Edison Avenue Colorado Springs, CO 80915	10. Work Unit No. (TRAIS)
	11. Contract or Grant No. DOT-FR-742-4277
	13. Type of Report and Period Covered
12. Sponsoring Agency Name and Address Department of Transportation Federal Railroad Administration (FRA) Office of Research and Development Washington, DC 20590	Technical Report October 1978 - May 1979
	14. Sponsoring Agency Code FRA/RRD-12

15. Supplementary Notes

The TDOP Phase II Analysis Plan (FRA/ORD-78/34) is a companion document to this report.

16. Abstract

One of the objectives of the Truck Design Optimization Project (TDOP) Phase II is to define the performance of newer Type II freight car truck designs versus the standard, three-piece Type I truck. To accomplish this dynamic performance evaluation, TDOP Phase II will utilize field test data and analytical tools. The analytical tools, consisting primarily of freight car truck simulation models and their supporting computer programs, will be used to extend and interpret the field test results.

The purpose of this report is to document the selection of candidate analytical tools from existing models and computer programs for validation and for use in TDOP Phase II. This report establishes the assessment criteria, surveys 59 existing analytical tools, evaluates in detail 16 of the more promising, and from these, selects a set of tools for validation and subsequent use on TDOP Phase II.

17. Key Words	18. Distribution Statement
Freight Car Truck Performance Freight Car/Truck Dynamics, Linear and Nonlinear Modeling, TDOP Phase II, Rail Car Model Assessment	Document is available to the public through the National Technical Information Service, Springfield, VA 22161

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price
Unclassified	Unclassified	57	A04 A01 micro

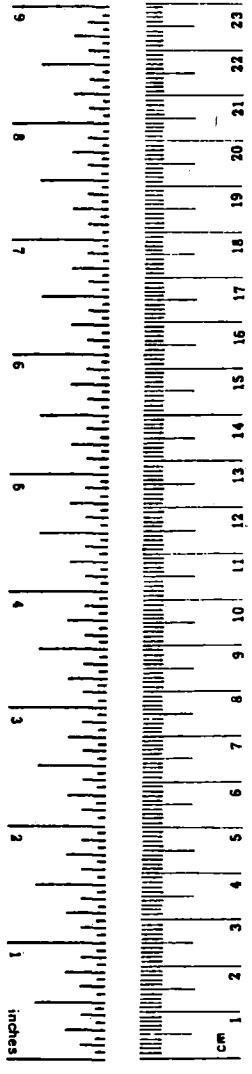
Form DOT F 1700.7 (8-72) Reproduction of completed page authorized

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

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

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



Approximate Conversions from Metric Measures

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

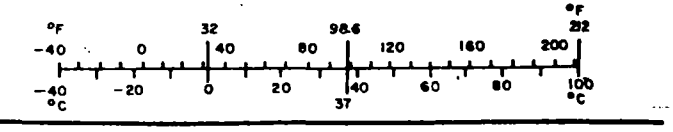


TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

SECTION 1 - INTRODUCTION

One of the objectives of the Truck Design Optimization Project (TDOP) Phase II is to define the performance of the standard three-piece Type I freight car truck vs. the newer Type II truck designs. In addition, TDOP Phase II will examine the incremental cost benefits to be derived from adoption of the newer Type II designs.

To perform this cost benefit analysis and dynamic performance evaluation, TDOP Phase II will acquire data from field tests and from analytical tools. Considerable test data exist on Type I trucks from TDOP Phase I. However, additional field tests will be conducted during TDOP Phase II to supplement Phase I data in Type I truck curve negotiation. In addition, TDOP Phase II will conduct an extensive series of field tests on Type II trucks. Analytical tools will then be applied to extend and interpret the results of these field test programs. The purpose of this report is to assess the existing body of analytical tools and select several candidates for validation and subsequent application.

In this report the term "analytical tool" refers to any analytical method employed to predict and understand the car/truck dynamics. The set of analytical tools includes, among other things, models which are considered here to be the set of equations describing the car/truck dynamics and the computer program implementing these equations.

Models can range from simple engineering models to a complex set of simultaneous, nonlinear, partial-differential equations used to describe the dynamic motion of a rail vehicle. Engineering models provide insight and qualitative analysis with a minimum of calculation and time expended (and are often the most efficient analytical tool in terms of the accuracy and usefulness of results with respect to the engineering time and effort required). On the other hand, complete, nonlinear, time domain models seek to simulate all of the pertinent nonlinear dynamics and responses from actual freight cars.

The analytical tools of most interest to TDOP Phase II are those models and computer programs which have been used in other car/truck modeling research and development projects. Some of these tools can be directly applied to TDOP Phase II with a minimum of effort and cost.

Although new tools will be used when no other existing model or program provides the required capability, the emphasis in TDOP Phase II is on the acquisition, assessment, and validation of existing tools, not the development of new analytical tools.

In brief, the objectives of this report are to:

- Establish the assessment criteria (see Section 2)
- Survey existing analytical tools (see Section 3)
- Assess and evaluate the most promising analytical tools (see Section 4)
- Select a set of analytical tools for validation (see Section 5)

SECTION 2 - ASSESSMENT CRITERIA

2.1 PERFORMANCE REGIME RELEVANCE

In the survey of analytical tools, a fundamental assessment criterion was the applicability of a tool to one or more of the performance regimes that will be used to determine freight car truck performance. TDOP Phase II will use these four performance regimes:

- Lateral Stability, which refers to the tendency of a truck to oscillate (hunt) with severe lateral and yawing motions while operating at high speeds on tangent track.
- Curve Negotiation, which measures the ability of a truck to negotiate a curve with a minimum of flange contact and wear on the rail and the wheels.
- Ride Quality, which is defined as the normal vibration environment that both the lading and the truck components are exposed to during non-extreme in-service operation.
- Trackability, which is the ability to maintain equal loads on all wheels during all extremes of in-service operation. Subsets of this regime include harmonic roll and bounce, curve entry and exit, and track twist load equalization.

Within each performance regime, performance indices have been defined to measure a truck's performance; (1) these indices are summarized in Table 2-1.

Table 2-1. Performance Regimes and Indices

Performance Regime	Performance Index
Lateral Stability	• Critical Speed • Magnitude of Lateral Acceleration
Trackability	• Wheel Unloading Index • Max. Roll Amplitude • Rate of Energy Dissipation • Derailment Potential (as measured by lateral to vertical force ratio)
Curve Negotiation	• Lateral force on leading outer wheel per 1000 pounds axle load per degree of curve under, at, and over balance speed • Wear Index • Derailment Potential
Ride Quality	• Transmissibility - Sway - Bounce - Yaw - Pitch - Roll

(1) Cappel, K.L., "TDOP Phase II Introductory Report," FRA Report No. FRA/ORD-78/53, November 1978.

2.2 TDOP PHASE II APPLICATION

After it was determined that an analytical tool was applicable to a performance regime(s), its suitability for TDOP Phase II was evaluated. An analytical tool had to be:

- Capable of performing (or supporting) a dynamic analysis that would meet a TDOP Phase II objective. These objectives are:
  - To define the performance of both standard and premium trucks in quantitative terms, represented by performance indices.
  - To establish a plan for collecting economic data on the cost of acquiring, operating, and maintaining the standard three-piece truck.
  - To establish a quantitative basis for evaluating the economic benefits to be derived from improved freight car trucks.
  - To supply the basis for a performance specification for freight car trucks.
- Compatible with the digital computers available to Wyle Laboratories' Colorado Springs Division.
- Capable of analyzing those truck/carbody configurations under study by TDOP Phase II with relatively minor modification, if any.
- Available within the required time frame.

2.3 ANALYTICAL CAPABILITY PACKAGE

In order to qualify for further consideration, a tool had to complement the other selected tools so that when, taken together, the total body of analytical tools would provide the required set of analytical capabilities. For example, the final package of tools had to be capable of defining the relationships below for each of the following: 1) tangent track, 2) curved track, 3) transition sections, 4) switches and special track work, and 5) variation in operating speed.

- The effect of truck/car design parameters, tolerances, and component wear on cargo vibration/shock.
- The effect of truck/car design parameters, tolerances, and component wear on track forces.
- The effect of truck/car design parameters on component forces, stresses, failure modes, and component life.

2.4 FIDELITY

The fidelity of an analytical tool is determined by its level of validity, accuracy, and precision, as described in the following paragraphs.

2.4.1 Validity

The validity of a tool refers to its ability to predict dynamic responses correctly. Specific questions relative to validity include:

- What is the purpose of the analytical tool? What methods are used to implement the simulation? Validity must be assessed relative to the intended use of the model. For example, a relatively simple tool used to predict qualitative behavior of a system may be acceptable if its predictions are within rough orders of magnitude, while a more detailed tool designed to investigate the effect of a specific system's nonlinearity must predict responses to much closer tolerance levels.
- Has there been any prior verification of the analysis and the equations? Did the assessment process reveal any discrepancies or problems? Has the tool been used by researchers other than the authors? If so, is it reasonable to expect that they reviewed the analysis and equations as well?
- Has there been any prior validation? While we intend to validate all selected tools against test data, what implications do prior validation efforts have for TDOP Phase II?

2.4.2 Accuracy

The accuracy of a tool refers mainly to the degree to which the physical constants of the actual system entered into the tool can be quantitatively defined for the model. This criterion relates primarily to the definition of system nonlinearities. Specific questions considered in the detailed assessment include:

- How were important nonlinear effects handled?
- How do model inputs correspond to measurable quantities of the actual system?
- Did any special problems in defining inputs come to light during the assessment review?

2.4.3 Precision

The precision of a tool refers mainly to its sensitivity to unavoidable round-off and truncation errors. This criterion is particularly difficult to assess because of the dependence on the particular computer system used. Potential problem areas are inversions of large matrices or digital smoothing of measured wheel profile data.

2.5 VERIFIABILITY

The verifiability of a tool refers to its ability to be compared with test results, not its ability to predict the results per se. Specific questions considered in the detailed assessments relative to this criterion include:

- What are the outputs?
- What are the response quantities that will be required for validation?

- How do the main outputs of the tool relate to specific measurements made in Phase I tests or to test measurements planned for Phase II?

- Would the verifiability of the tool be improved by any special test requirements for the validation tests of this model?

2.6 UTILITY

The utility of a tool is a general topic related to projected costs in utilizing and adapting this model to TDOP Phase II applications. Specific questions considered in this detailed assessment include:

- How cost effective is the program?

- Has the tool been used extensively? Were any noteworthy problems experienced?

- What is the documentation quality? Is the source code available? Can the author of the analytical tool be contacted?

- How difficult will it be to prepare the input or interpret the output?

Table 2-2. Summary of Assessment Criteria

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

SECTION 3 - SURVEY OF ANALYTICAL TOOLS

To identify existing tools, a formal survey of the available literature was conducted by Wyle Laboratories, the TDOP Phase II Contractor. The available literature was found to contain several excellent summaries of the tools. (1), (2), (3) These, plus recent technical papers and descriptions of concurrent research projects, provided information on programs for the preliminary survey.

The survey was limited to either existing tools or those near completion in concurrent research projects. In many cases, methods rather than actual models were discussed in the literature. Since our objective was to assemble existing analytical tools, we did not review analytical methods in detail unless a program source code, which implemented the method under discussion, was indicated. Simplified engineering models (e.g., single degree-of-freedom systems) were also not included in the survey. While engineering models are highly useful in providing insight and will, in fact, be used frequently in TDOP analyses, the purpose of this report is to review and select more comprehensive candidate tools for validation. The results of the preliminary survey are summarized in Table 3-1.

After the preliminary survey, all but the clearly unsuitable analytical tools were studied in greater detail. Where formal program documentation was available, it was obtained and reviewed. When attempts were made to obtain tools thought to be available, some problems were encountered, such as: a number of programs were tied to a particular set of equipment, such as hybrid computers; other programs were proprietary; still others were programs from foreign sources for which no contact was readily available.

(1) Perlman, A.B., "An Evaluation of Computer Programs for the Analysis, Prediction and Simulation of Rail Vehicle Dynamics," August 1974, Transportation Systems Center (TSC) Task Memorandum RR-515, Rail System Dynamics Program.

(2) Cooperrider, N.K. and Law, E.H., "A Survey of Railway Vehicle Dynamics Research," Journal of Dynamic Systems Measurement and Control, Trans. ASME, Special Issue on Ground Transportation, June 1974.

(3) Breese, R.G., ed., "Track-Train Dynamics Bibliography," AAR, Railway Progress Institute, Chicago, Illinois, 1973.

Table 3-1. Survey of Existing Tools

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
1	AAR	Steady State	2, 3, 4 Axle Rigid Truck Curve Negotiation Model	In-house on tape. No plans to use.	Program designed for rigid locomotive truck analysis. Not suitable for the more flexible freight car trucks, especially Type II. Does not model wheel conicity or gravitational effects.
2	AAR	Dynamic Time Domain	Nonlinear Time Domain Curving Program	In-house on tape.	Final documentation not complete. Modified version of Smith's Illinois Institute of Technology Masters Thesis Program.
3	Battelle	Steady State	SSCUR2-2 Axle Steady State Curve Negotiation	Requires funding for modification and documentation. No plans to obtain.	Similar to Law and Cooperrider steady state program. Models Metroliner; as such, not directly applicable for freight car trucks.
4	Battelle	Steady State	SSCUR3-3 Axle Locomotive Steady State Curve Negotiation	No plans to obtain.	Not directly applicable to freight car truck curving analysis.
5	Battelle	Dynamic Time Domain	Nonlinear Curve Entry for 9 dof Half Car Model	Requires funding for modification and documentation. No plans to obtain.	Similar to a Law and Cooperrider program. Models Metroliner; as such, not suitable for freight car trucks.
6	Battelle	Dynamic Time Domain	Full Car Curving Model	Requires funding for modification and documentation. No plans to obtain.	Same as Law and Cooperrider program. Models Metroliner; as such, not suitable for freight car trucks.
7	Law/ Cooperrider	Steady State	Nonlinear Steady State Curving of a 9 dof Rail Vehicle	Program source tape in-house. No documentation available to date. Program installed on Interdata.	Suitable for Type I freight car trucks. Considered very desirable for use on TDOP Phase II.
8	Law/ Cooperrider	Steady State	Nonlinear Steady State Curving of a 17 dof Rail Vehicle	Program installed on Interdata.	Suitable for Type I and some Type II freight car trucks. Considered very desirable for use on TDOP. Similar to the 9 dof with the addition of primary suspension elements.
9	Law/ Cooperrider	Dynamic Time Domain	Nonlinear Curve Entry for 11 dof Half Car Model	Available without documentation. No plans to obtain.	Same as 6. Models Metroliner. Not suitable for freight car trucks.
10	Law/ Cooperrider	Dynamic Time Domain	Nonlinear Curve Entry for 9 dof Full Car Model	Metroliner version available without documentation. Freight car model available in future.	Same as 7. Presently models Metroliner. Is being modified at Clemson to model freight car, but completion may not be timely for TDOP use.

Table 3-1. Survey of Existing Tools (continued)

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
11	Law/ Cooperrider	Dynamic Time Domain	CURVELOCO - 27 dof Nonlinear 6 Axle Locomotive on Tangent, Spiral, Curved Track	Available from TASC. No plans to obtain. Formal documentation in progress.	Not suitable for freight car trucks.
12	AAR	Dynamic Time Domain	Dynamic Curving Model of 6 Axle Locomotive	No plans to obtain.	Not suitable for freight car trucks.
13	Japanese Rail	Steady State	Side Thrust of Curving Wheels	Foreign source, no local contact known. No plans to obtain.	Similar capabilities expected in other programs.
14	British Rail	Steady State	Steady State Curving, Flexible Trucks	Foreign source, no local contact known. No plans to obtain.	Similar capabilities expected in other programs.
15	Nichio-Japan	Steady State	Steady State Curving	Foreign source, no local contract known. No plans to obtain.	Similar capabilties exist in other programs.
16	AAR-TTD	Eigenvalue	Freight Car Hunting Model	User's manual in-house. Source deck in-house on tape.	Adaptable to Type II trucks. Linear analysis only, assumes spherical wheel profile. Current plans are to use in parallel with a similar program by Law and Cooperrider.
17	AAR-TTD	Nonlinear Time Domain	Lateral-Vertical Model	Source code in-house on magnetic tape.	Detailed modeling of truck masses, wheel and rail profiles defined mathematically, 2 dof reserved for carbody. No plans to use on TDOP. Capability available in other programs.
18	AAR-TTD	Time Domain Solution, Numerical Integration	Nonlinear Hunting Model	Operational, available when documentation is completed. (Requested from AAR 4/15/78; still not available as of 1/19/79, and hence not selected for detailed assessment.)	Similar to lateral-vertical model with more complexity and degrees of freedom in the math model.
19	AAR-TTD	Force Balance at Equilibrium	Quasi-Static Lateral Train Stability	Operational, user's manual exists. Requested from AAR 4/25/78 but not received to date.	Cannot be used to directly evaluate truck performance. Ignores all internal forces. Does not fit into required analysis areas, but can be obtained if need arises.
20	Arizona State	Subroutine to Support Time Domain Lateral Stability Program	WHRAIL, a Wheel/Rail Contact Geometric Constraint Subroutine	Available, source deck and documentation, in-house. Asymmetric version of the subroutine also available for use on TDOP.	Utilized in HUNTCT. One of the best available subroutines forcalculating wheel/rail interaction effects.

Table 3-1. Survey of Existing Tools (continued)

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
21	SPTCo. TDOP Phase I	Frequency Domain, Time Domain Optional	Graphical Output Oriented Computer Model (Frequency Domain Model)	Source code received as GFP from TDOP Phase I. A MITRE review has pointed out several problem areas.	Documentation indicates some unconventional trucks cannot be modeled. Limited to linear analysis with describing function techniques used to handle Coulomb friction. Problems with the conversion techniques identified. No plans to use.
22	Clemson U.	Time Domain Solution, Numerical Integration	Nonlinear Wheelset Dynamic Response to Random Lateral Rail Irregularities	Availability assumed.	Good for studying the nonlinear dynamics of a single wheelset. However, does not fit into required analysis areas as total truck not considered. Can be obtained if need arises.
23	TSC	Frequency Domain	LATERAL	Program installed. Descriptive manual in-house.	Includes creep effects, but no detailed description of wheel/rail interaction. Designed for lateral, roll, and yaw only, no vertical. No prior validation.
24	Wyle	Time Domain Solution	HUNTCT	Program installed. Little formal documentation available at the present time.	Truck hunting program which includes detailed carbody/lading modeling. Many nonlinear capabilities. Easily adaptable to Type II trucks. Some validation with Phase I data performed by comparing calculated and observed kinematic frequency.
25	AAR	Time Domain Solution, Numerical Integration	Detailed Lateral Stability Model for a Consist	Available. User's documentation being prepared, descriptive manual exists. Since definite need for this model has not been determined, installation will await definition of a specific application.	Overall train models cannot be used directly to evaluate truck performance. Does not fit into required analysis areas, but can be obtained if need arises.
26	Law/ Cooperrider	Eigenvalue	Linear 9 dof Freight Car	Fortran source code available for Univac or IBM. The 17 dof version is more applicable to the goals of TDOP and will be used instead.	Linear 9 dof (lateral, yaw, and warp of each truck; and lateral, yaw, and roll of car) spin and lateral spin creep effects and gyroscopic effects. Allows wheelset and suspension asymmetries.
27	Law/ Cooperrider	Eigenvalue	Linear 17 dof Rail Car	Source code in-house on tape. Program installed and test cases run on the Interdata.	Lateral and yaw of each wheelset; lateral, warp, and yaw of each truck, and lateral, yaw, and roll of body. Provides for modeling radial trucks. Spin creep and gyroscopic terms included. Allows for wheelset and suspension asymmetries. Plans are to utilize this program for lateral stability analysis.
28	Law/ Cooperrider	Eigenvalue	Linear 19 dof Rail Car	Fortran source code for IBM available. Report of references in preparation. Documentation in preparation. No plans to install. Author recommended #27 as more applicable to TDOP goals.	Modification of 17 dof model with two additional degrees of freedom representing body bending and torsion.

Table 3-1. Survey of Existing Tools (continued)

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
29	Law/ Cooperrider	Eigenvalue	Linear 23 dof Freight Car	Card deck for IBM-370. Status of documentation unknown. No plans to install.	Modification of 19 dof model with four additional degrees of freedom representing torsional flexibility of each wheelset.
30	Law/ Cooperrider	Describing Function Analysis with Iterative Search for Limit Cycle Conditions	Quasi-Linear 9 dof Freight Car	Operational on ASU Univac 1110 but requires further development for general use.	Model of linear 9 dof freight car model with nonlinear wheel/rail gemoetry and Coulomb friction at wear plate, center plate, and bearing adapters.
31	J.H. Wiggins	Eigenvalue Time Domain Response to Periodic Input	DYNALIST II	Readily available. Installed on the CDC Cybernet System. Good documentation.	General linear systems modeling capability. Allows up to 50 degrees of freedom and 25 system components. Response to sinusoidal or stationary random rail irregularities. Limited to linear system analysis. Readily adaptable to Type II trucks.
32	MELPAR	Time Domain Numerical Integration	Dynamic Rail Car Simulation Program	Good documentation and user's manual. Copies of deck available from FRA. No plans to install due to high cost and availability of alternative programs.	Variable degrees of freedom, nonlinear analysis. High run costs and great complexity makes use and validation impractical. While not completely unsuitable for TDOP needs, it was not selected for detailed assessment because of the ready availability of more cost effective alternatives.
33	IIT	Time Domain Solution Includes Nonlinear Effects	Dynamics of a Freight Element in a Railroad Freight Car	Descriptive manual and source listing in-house.	Other models which operate with similar capability are available. Adaptability to Type II trucks is difficult due to Lagrangian derivation. No plans to use this model on TDOP Phase II.
34	MITRE	Time Domain Simulation	FRATE	Installed on the CDC CYBERNET system. Documentation complete.	Program is based on FRATE 11 with improved input-output capabilities. Currently set up for modeling the 89-foot flat car, but can be used for other vehicles as well by changing input parameters.

Table 3-1. Survey of Existing Tools (continued)

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
35	AAR-TTD	Time Domain Simulation	Flexible Body Railroad Freight Car Model	Operational on Interdata. Documentation complete.	20 dof. Not easily modified to simulate Type II trucks. Some validation in terms of wheel lift-off test data.
36	MIT	Combination of Numerical Integration and Force Balance at Equilibrium	Response to Track Cross Level Variations	Availability unknown. Documentation exists in the form of a descriptive manual.	Nonlinear capabilities. Adaptability to Type II trucks unknown. Alternative models for the same purpose available.
37	MIT	Time Domain Solution Numerical Integration	General Vehicle Dynamic Model	Availability unknown. User's manual in-house. Program may not have been used since 1966 since no mention has been found after the manual was written. Model considered unavailable for this reason.	Not recommended due to high computer cost factor. Similar capabilities appear to be available in more cost-effective programs.
38	Battelle	Frequency Domain Solution	TRKVEH	Requires funding for modification and documentation. No plans to obtain.	Limited to linear analysis. Lateral model has only partial representation of wheel rail kinematics. No evidence of prior validation. Unknown adaptability to Type II trucks.
39	Battelle	Frequency Domain Solution	TRKVPSD	Requires funding for modification and complete documentation. Source deck in-house.	Limited to linear analysis. Lateral model has only partial representa tion of wheel/rail kinematics. No evidence of prior validation. Appears to differ from TRKVEH in that output is in form of power spectral density. 7 dof model. Adaptability to Type II trucks unknown.
40	Wyle	Time Domain Solution	FRATE 11	Available, program installed on the Interdata. Documentation complete.	Nonlinear 11 dof. Easily adaptable to Type II trucks. Evidence of prior validation exists.
41	Wyle	Time Domain Solution	FRATE 17	Available, documentation complete.	Nonlinear 17 dof. Easily adaptable to Type II trucks. Evidence of prior validation exists.
42	Battelle	Eigenvalue	CARHNT	Requires funding for modification. No plans to obtain.	Calculates eigenvalues and eigenvectors of the characteristic equations in lateral stability regime.
43	Battelle	Eigenvalue	TRKHNT	Requires funding for modification and documentation. No plans to obtain.	Similar to #6 except emphasizes truck as opposed to entire vehicle.

Table 3-1. Survey of Existing Tools (continued)

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
44	TSC	Frequency Domain	FULL	Descriptive manual in-house, program installed.	Linear model for vehicle pitch and vertical responses.
45	TSC	Frequency Domain	HALF	Descriptive manual in-house, program installed.	Linear model for rock and roll responses. Includes compliant track structure.
46	TSC	Frequency Domain	FLEX	Descriptive manual in-house, program installed.	Linear model for rock and roll responses. Includes one mode for car flexibility.
47	Japanese Railways	Unknown	Vehicle on a Bridge	Foreign source, no local contact known. No plans to obtain.	Little is known of this program beyond a brief mention in a TSC review.
48	British Rail	Numerical Integration	Wheel-Rail Force	Foreign source, no local contact known. No plans to obtain.	Investigates interaction between wheel and rail in vertical plane in detail.
49	Japanese Rail	Unknown	Variation of Wheel Load	Foreign source, no local contack known. No plans to obtain.	Investigates wheel/rail forces at rail discontinuities.
50	British Rail	Unknown	Dynamic Loading of Rail Joints	Foreign source, no local contact known. No plans to obtain.	Investigates rail forces at rail discontinuities.
51	Battelle	Solves Beam Equation	Rail on Elastic Foundation	Requires funding for modification and documentation. No plans to obtain.	Investigates rail foundation (ballast) forces.
52	AAR/TTD	Eigenvalue	Locomotive Hunting Model	No plans to obtain because of orientation to locomotive trucks.	Generates critical speeds of locomotives.
53	Chang, Garg	Time Domain	6 Axle Locomotive Response	No plans to obtain because of orientation to locomotive trucks.	Written specifically for 6-axle locomotive.
54	AAR-TTD	Numerical Integration Time Domain	Detailed Vertical Train Stability Model	No current need identified for TDOP. Can be obtained if a need arises.	Emphasis on car interactions, does not separate truck modeling.

Table 3-1. Survey of Existing Tools (continued)

EXISTING ANALYTICAL TOOLS					
NO.	SOURCE	TYPE	NAME/DESCRIPTION	AVAILABILITY/STATUS	COMMENTS
55	TRW	Frequency Domain	Rail Vehicle Roadbed Study	No plans to obtain.	Developed for high speed, mass transit application. Apparently has not been used for some time.
56	MITRE	Eigenvalue	MITRE Random Process	No plans to obtain because of inappropriate application.	Perlman[4] calls program "unsuitable for dynamic stability analysis." Primary applications elsewhere.
57	Battelle	Time Domain Numerical Integration	Nonlinear Freight Car Model	Requires funding for modification and documentation. No plans to obtain.	Emphasis on rail foundation stresses, rail discontinuities, wheel/rail forces.
58	United Aircraft	Critical Speed	UAC-4	No plans to obtain because of inappropriate application.	Written specifically for the single turbotrain application.
59	United Aircraft	Critical Speed	UAC-6	No plans to obtain because of apparent lack of documentation.	Perlman[4] notes "not documented in any detail."

[4]Perlman, A.B., "An Evaluation of Computer Programs for the Analysis, Prediction, and Simulation of Rail Vehicle Dynamics," August 1974, TSC Task Memorandum RR-515, Rail System Dynamics Program.

SECTION 4
DETAILED ASSESSMENT OF CANDIDATE TOOLS

After completing the preliminary review, the most promising analytical tools were selected for further assessment. This preliminary choice was based on the projected requirements within each dynamic regime. Table 4-1 lists the 17 analytical tools and two subroutines selected for more detailed assessment. The remainder of this section provides a complete assessment of each of these tools and subroutines.

Table 4-1. Tools Selected for Detailed Assessment

TOOL	PRIMARY PERFORMANCE REGIME
Freight Car Hunting (AAR)	Lateral Stability
Lateral/Vertical Model (AAR)	Lateral Stability
17 dof Eigenvalue (Law & Cooperrider)	Lateral Stability
HUNTCT (Wyle)	Lateral Stability
Freight Car Curving Model (AAR)	Curve Negotiation
9 dof Steady State Curving Model (Law & Cooperrider)	Curve Negotiation
17 dof Steady State Curving Model (Law and Cooperrider)	Curve Negotiation
DYNALIST II (TSC)	Ride Quality
FULL (TSC)	Ride Quality
HALF (TSC)	Ride Quality
FLEX (TSC)	Ride Quality
LATERAL (TSC)	Ride Quality
TDOP Phase I Model (SPTCo)	Ride Quality
Flexible Body Railroad Freight Car Model (AAR)	Trackability
FRATE (MITRE)	Trackability
FRATE 11 (Wyle)	Trackability
FRATE 17 (Wyle)	Trackability
SUPPORTING SUBROUTINES	
WHRAIL, Symmetric Wheel/Rail Geometric Constraint Routine (Law & Cooperrider)	Lateral Stability Subroutine
WHRAIL, Asymmetric Wheel/Rail Geometric Constraint Routine (Law & Cooperrider)	Lateral Stability Subroutine

4.1 FREIGHT CAR HUNTING MODEL (AAR) - CHEUNG, GARG, AND MARTIN

Introduction

The Freight Car Hunting Model was developed by the Association of American Railroads (AAR) Track/Train Dynamics group to investigate lateral stability in freight car trucks. The model is a 25 degree-of-freedom (dof) linear representation. Using matrix methods, the system equations are solved to produce eigenvalues and eigenvectors (natural frequencies and mode shapes) from which critical speeds for truck hunting are obtained.

Model Application Areas

The Freight Car Hunting Model provides the capability for analyzing the lateral stability of freight car trucks. In terms of TDOP requirements, this model should provide an excellent means of gaining insight into the hunting behavior of complex trucks. Predictions of critical speed are obtained from the model. It is thus directly capable of being related to that performance index.

Model Description

The configuration for this lumped mass model is shown in Figure 4-1. (Note: All of the illustrations that appear in Section 4 have been extracted from the appropriate source documents referenced in the footnotes.) The model represents a freight car with Type I trucks having 25 degrees of freedom (see Table 4-2). Each truck model consists of a pair of side frames which are connected by linear spring and damping elements to the bolster and wheelsets. The carbody is a single rigid mass element. The wheelsets are modeled assuming symmetric wheels and are characterized by a single effective value of conicity. The track is assumed to be completely rigid and thus does not enter into the formulation.

The linear representation assumes small amplitude displacements. The equations of motion for the system are derived using Newtonian methods. It is assumed that the model could be modified to reflect Type II trucks, however, this may require additional documentation from AAR on the specific means by which the system equations are implemented in the program.

Program Implementation

The program is coded in FORTRAN and can be run on either the CDC or Interdata system. The program employs matrix inversion techniques to obtain the solution for natural frequencies and mode shapes.

Program input consists of the values of the various lumped masses, moments of inertia, effective spring and damping rates, and geometrical data. Output is in tabulated form giving a verification of the input and the following calculated results for a given speed: frequency and damping for each normal mode and the normalized mode shape.

Documentation exists in the form of a User's Manual, (1) which provides an excellent description of the general theory forming the basis of the program, as well as sample input and program results. Additionally, the appendices of the manual provide a helpful discussion of the gravitational stiffness and creep relations. However, the documentation does lack an explanation of the specific equations implemented in the program. Such information would facilitate the task of modifying the input and system equations in the program if necessary to address Type II trucks.

Assessment

The Freight Car Hunting Model will be of use in TDOP Phase II as a means of gaining insight into the hunting phenomenon and establishing general relations between car and truck parameters and critical hunting speed. Results must be carefully scrutinized and interpreted, however, with regard to the underlying assumptions of small amplitude displacements and linearity which are used.

(1) Cheung, T.H.W., Garg, V.K., and Martin, G.C., "User's Manual, Freight Car Hunting Model," AAR Document R-251, February 1977.

No evidence of previous attempts to validate the program has been obtained. Accuracy levels will not be as high as with a model accounting for suspension and wheel/rail nonlinearities because of the approximations needed to linearize the system equations. The impact of round-off error will be moderate as a result of the matrix inversion solution technique. The matrix involved is sufficiently small to preclude severe precision problems. Verification of this model will be performed by comparing critical speeds and mode shapes obtained from the model versus actual test data. In that this linear model is primarily intended to provide insight into truck behavior and establish general relationships between truck parameters and lateral stability rather than hard quantitative results, a fair amount of tolerance can be used in comparing model results with test data. Although the quantitative results may not match test data exactly, the ease with which this linear model can be used and the efficiency of the solution technique would be sufficient to justify its use in TDOP.

Conclusions

The Freight Car Hunting Model shows promise as a useful tool for establishing general relationships between truck and car body parameters and lateral stability. Quantitative results may be somewhat in error due to the linear approximations used in the solution. The model uses matrix manipulations to obtain natural frequencies and mode shapes, and hence critical speeds, for a 25 dof truck/body representation. The computer program is straightforward and generally well documented, although additional clarification on the implementation of system equations in the programming would be useful.

Table 4-2. Degrees of Freedom - Freight Car Hunting Model

NUMBER	LOCATION	MOTION
1	Axle 1	Lateral
2	Axle 2	Lateral
3	Axle 3	Lateral
4	Axle 4	Lateral
5	Bolster 1	Lateral
6	Bolster 2	Lateral
7	Body	Lateral
8	Side Frame 1	Lateral
9	Side Frame 2	Lateral
10	Side Frame 3	Lateral
11	Side Frame 4	Lateral
12	Side Frame 1	Longitudinal
13	Side Frame 2	Longitudinal
14	Side Frame 3	Longitudinal
15	Side Frame 4	Longitudinal
16	Axle 1	Yaw
17	Axle 2	Yaw
18	Axle 3	Yaw
19	Axle 4	Yaw
20	Bolster 1	Yaw
21	Bolster 2	Yaw
22	Bolster 1	Roll
23	Bolster 2	Roll
24	Body	Yaw
25	Body	Roll

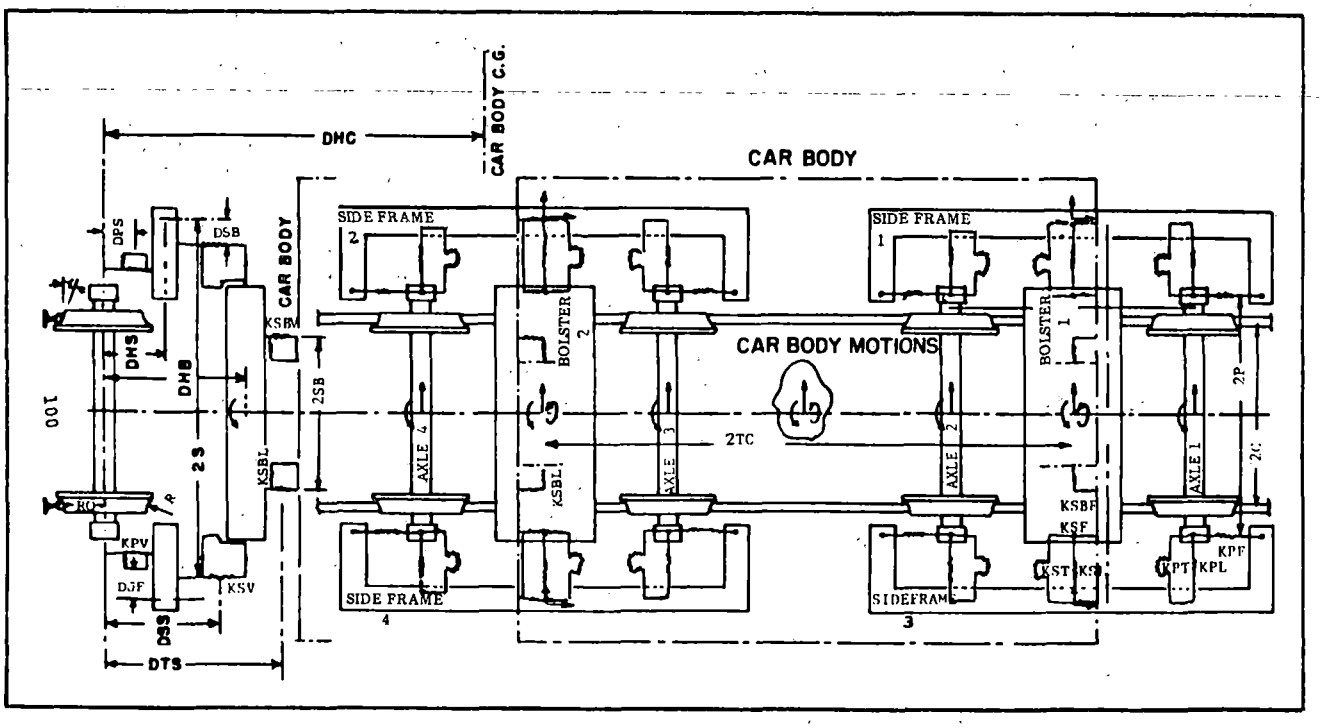


Figure 4-1. Freight Car Hunting Model Configuration

4.2 LATERAL/VERTICAL MODEL (AAR) - TSE AND MARTIN

Introduction

The Lateral/Vertical (L/V) Model was developed at AAR by the Track/Train Dynamics group as a tool to investigate lateral stability of freight cars. In particular, the model can be used to make a determination of the approximate ratio of lateral to vertical forces at the wheel/rail interface, thereby giving an indication of the potential for wheel climb and derailment. The model involves 14 degrees of freedom. Only a single truck and half a carbody are represented. The representation of the truck allows for nonlinearities such as center plate Coulomb damping. The solution technique is by time integration.

Model Application Areas

The Lateral/Vertical Model can be used to address the lateral stability problem. As a time domain model, its principal contribution would be in the investigation of wheel climb and derailment potential rather than the identification of hunting modes and critical speeds which can be investigated more efficiently with the various frequency domain models that have been assessed.

Model Description

The model is shown in Figure 4-2. The model includes one truck supporting a half carbody including bolster. The two wheelsets are linked by inertia-less side frames. The truck stiffness with respect to warping is modeled by diagonal spring elements connecting the side frames. Spring elements also support the half carbody on the side frames. The suspension model also provides for Coulomb damping. The half carbody accounts for vertical and roll motions but neglects carbody yaw, pitch, and lateral displacements. A total of 14 degrees of freedom is included in the model (see Table 4-3).

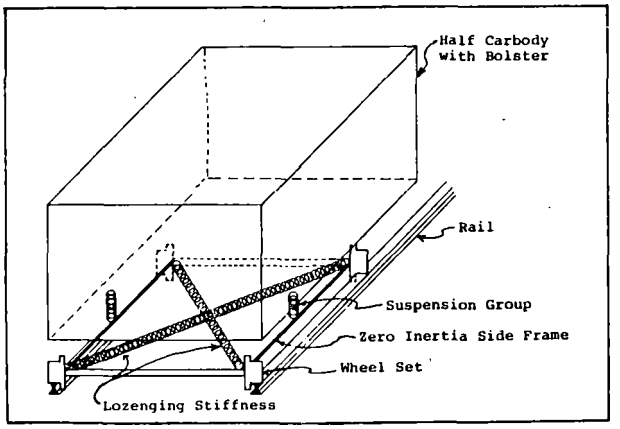


Figure 4-2. Lateral/Vertical Model Configuration

Table 4-3. Degrees of Freedom
Lateral/Vertical Model

1.	Vertical displacement (bounce) of the car body
2.	Roll of the car body
3.	Vertical displacement of the front axle
4.	Roll of the front axle
5.	Yaw of the front axle
6.	Pitch (spin) of the front axle
7.	Longitudinal displacement of the front axle
8.	Lateral displacement of the front axle
9.	Vertical displacement of the rear axle
10.	Roll of the rear axle
11.	Yaw of the rear axle
12.	Pitch of the rear axle
13.	Longitudinal displacement of the rear axle
14.	Lateral displacement of the rear axle

The model focuses on the geometrical relationship between wheel and rail. The program requires the user to define the wheel and rail profiles in terms of a series of fourth order polynomial segments. The technique is similar to that employed in the Law/Cooperrider wheel/rail constraint subroutines, but is more coarse and requires greater user effort to precalculate polynomial coefficients. Other than the wheel/rail interface description, no additional aspects of the track are included in this model. The model is based on small amplitude displacements but does involve nonlinear suspension elements. The documentation indicates that Newtonian methods were used in deriving the system equations. The form of the model appears to offer sufficient flexibility to be adapted to Type II truck configurations with perhaps extra attention required to treat the nonlinear functional relationships.

Program Implementation

The Lateral/Vertical Model is coded in FORTRAN and can be run either on the CDC or Interdata systems. Simple first order time integration (Euler's method) is used to produce a time domain solution to the equations of motion with a given set of initial conditions.

Up to 20 cards of input data with five or six items of data per card are required to run the program. The input data include the necessary geometrical information to define the truck and the wheel/rail profiles, the stiffness and damping elements, the inertias, initial conditions, the run time, and time step.

Program output is printed at a specified user defined interval during the simulation. Printed at each interval are the current time, an indication of the wheel/rail contact regime, creepage and contact forces including the ratio of lateral to vertical forces, carbody displacements and velocities, and finally, axle displacements and velocities. The output can be plotted with Wyle in-house routines.

The program is described and documented in a User's Manual. (2) The model is described in the User's Manual but no detailed development of the system equations is included. A detailed explanation of the required program input and its format is included along with sample input and output.

Assessment

This model was selected for assessment because of its fairly detailed representation of the wheel/rail interaction. Although it could be used to investigate truck hunting, this time domain model would not be as efficient as frequency domain models in the identification of critical speeds, for example. The use of this model, therefore, should be restricted to the wheel climb and derailment aspects of lateral stability.

The Contractor has had no previous experience with the Lateral/Vertical Model prior to TDOP Phase II. No evidence of any validation of the model with test data is contained in the available documentation. The level of detail in the truck and wheel/rail representation are such that a high level of accuracy can be expected. Similarly, the form of equations and solution techniques should present no significant machine precision problems.

Conclusions

The Lateral/Vertical Model is best suited for the investigation of wheel/rail forces, in particular the interactions leading to wheel climb and derailment. The strength of the model is in the detail with which the wheel/rail profiles are defined; however, other tools selected for Phase II assessment treat the profiles in a similar manner, but one which is easier to use. Therefore, the Lateral/Vertical Model has not been selected for validation.

4.3 17 DOF EIGENVALUE MODEL - LAW AND COOPERRIDER

Introduction

The 17 Degree of Freedom Eigenvalue Model is one of several linear freight car models which have been developed by Law and Cooperrider. The other models have various numbers of degrees of freedom but the solution techniques employed are similar. Eigenvalue/eigenvector stability analyses are performed from which natural frequencies and mode shapes are obtained.

Model Application Areas

The 17 dof Eigenvalue Model can be applied in the lateral stability investigation. The natural frequencies and mode shapes obtained from the eigenvalue analysis can be related to critical hunting speeds and damping factors.

Model Description

The model consists of a single rigid mass representing the carbody. The carbody is supported on two trucks, each having two wheelsets. The 17 degrees of freedom include:

Lateral displacement of each wheelset	4
Yaw of each wheelset	4
Warp of each truck	2
Yaw of each truck	2
Lateral displacement of each truck	2
Carbody lateral displacement	1
Carbody roll	1
Carbody yaw	1
Total:	17

Gravitational stiffness, spin creep, and gyroscopic terms are included in addition to the more significant terms. The model can accomodate asymmetrical loading front to rear and nonidentical front and rear wheelsets and suspension parameters. Also included in the model is a provision for modeling bending and shear connections between wheelsets, such as those implemented in a radial axle truck.

The equations of motion are linear and homogeneous. However, no detailed documentation is available explaining their derivation. A general discussion can be found in Report No. FRA-OR&D-77-36 by Law, Hadden, and Cooperrider. The formulation is considered sufficiently flexible to allow the representation of Type II trucks.

Program Implementation

The 17 dof Eigenvalue Model is coded in FORTRAN and has been previously implemented on IBM and Univac machines.

The program makes use of the QR Transform technique for obtaining the eigenvalues of real symmetrical or asymmetrical square matrices after first recasting the system equations in upper diagonal matrix form.

The program output includes a recapitulation of input parameters, the coefficients of the mass damping and stiffness matrices, the eigenvalues, and the eigenvectors in magnitude/phase form.

At the present time, no formal documentation of this program is known to exist.

(2) Tse, Y.H. and Martin, G.C., "User's Manual, Lateral/-Vertical Model," AAR Document R-237, February 1977.

Assessment

This program is useful in TDOP Phase II to estimate the critical speed of Type I and Type II trucks and to estimate the stability margins as a function of speed within the context of the lateral stability regime analysis task.

The TDOP Phase II Contractor has had some limited experience with this program. Currently, the program is available on the in-house Interdata and on a time share CDC 7600.

The accuracy of this model is expected to be limited as a result of the linear approximations used in the formulation for components which are distinctly nonlinear, such as center plate damping, flange contact, snubber damping, etc. Other than the linear approximations, there does not appear to be other factors in the model development which would compromise its validity. The size of the matrices for this 17 dof model is sufficiently small so that no computational precision problems should arise.

No report of prior validation of this model is known to exist. There may be difficulties in attempts to validate this model since the nonlinearities, as well as track disturbance which the model does not account for, are known to influence critical speed. Also, determining frequency and damping of the least damped mode from test data may be difficult due to the influence of significant nonlinearities as identified above. However, recent work in reducing rail vehicle test data by Law and Cooperrider has shown promise in being able to estimate these parameters.

From the standpoint of utility, the formulation is sufficiently flexible so that both Type I and Type II trucks may be addressed. The model is linear so that if modifications are necessary, they should be relatively simple. The linear frequency domain model should be efficient with respect to run costs. Generating input and the interpretation of output is straightforward.

Conclusions

The 17 dof Eigenvalue Model is a linear frequency domain program for performing lateral stability analyses of freight cars with Type I and most Type II trucks. Because it is linear, it is not expected to provide close quantitative agreement with test data. However, its ease of use and low cost make it attractive for doing preliminary analyses.

4.4 HUNTCT (WYLE) - HEALY

Introduction

HUNTCT is a nonlinear time domain computer simulation of a complete rail vehicle system including trucks, and wheelsets, along with track structure. It can be used for investigating lateral stability, lading environment, wheel/rail forces, and rail foundation stresses. The program, an expanded version of an earlier Wyle model FRATE 11, uses the Law and Cooperrider Wheel/Rail Geometric Constraint Subroutine to provide nonlinear force displacements at the wheel/rail interface.

Model Application Areas

As a rather comprehensive freight car model, the program can be used to investigate a number of performance areas. Specifically, the model is best suited for examining the detailed motions resulting from truck hunting and the forces produced in curve entry and exit. The level of detail is such that the model has the potential to relate virtually any car or truck parameter to a given performance index. Since it is a time domain solution, it is well suited to the investigations of detailed motions and forces as well as the identification of critical speeds and mode shapes.

Model Description

A description of the model appears as Figure 4-3. The model provides the option of representing the carbody as a rigid mass or as a flexible body. With the basic rigid

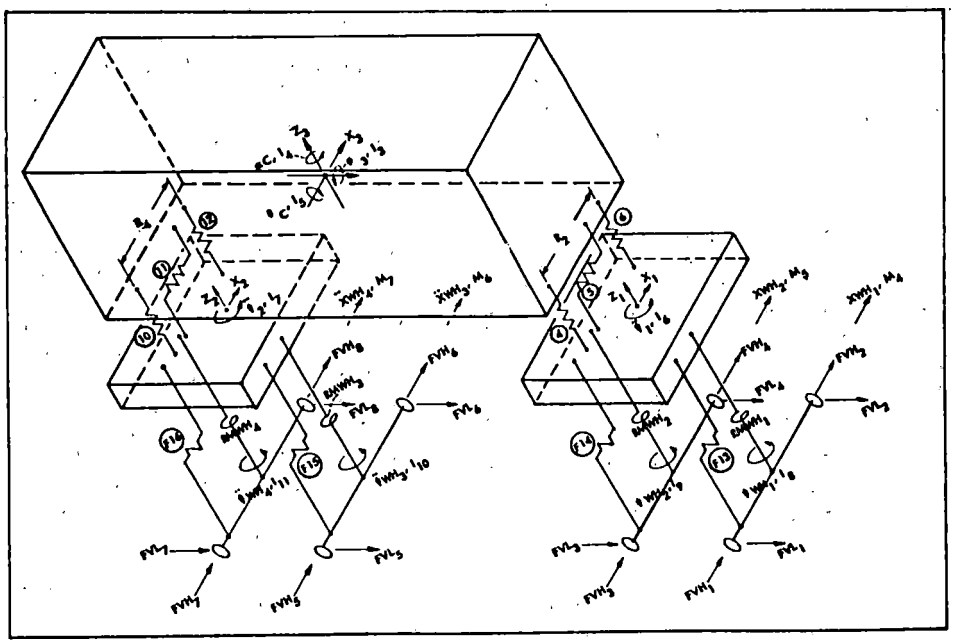


Figure 4-3. HUNTCT Model

carbody representation, the model has 21 degrees of freedom. When the flexible carbody option is used, an additional degree of freedom is included for each natural mode of carbody flexure considered. The truck is currently modeled as a single mass with vertical, lateral, yaw, and roll degrees of freedom. The truck model also provides for coupling between wheelsets in the yaw sense (lozenging stiffness). A lumped mass having lateral and yaw degrees of freedom is included for each wheelset. Vertical and roll motions of the wheelset are constrained by the wheel/rail geometry with the assumption of no wheel lift off. Detailed calculations of the wheel/rail interface are carried out for each wheelset. The effective track mass, stiffness, and damping in the vertical sense are lumped with the truck.

The non-linear equations of motion are developed using Newtonian methods. The equations are formulated so as to correspond to physical components of the actual rail cars and trucks so that the model can be adapted to include more or less detail quite easily.

Program Implementation

The program is coded in FORTRAN and has been operated on CDC and Interdata computers. The program employs a standard fourth order Runge-Kutta integration technique to produce the time domain solution to the system equations.

Some pre-processing to obtain input data is necessary. For example, the carbody bending mode shapes must be obtained to be supplied as program input if the flexible carbody option is to be used. The program produces time history data of the motions and forces of the model components in both plotted and printed form.

In-house documentation on HUNTCT exists as a compilation of notes, sample cases, listings, and internal memoranda.

Assessment

The potential use of the HUNTCT Model in TDOP Phase II will be in performing detailed analysis of the motions and forces produced in hunting and curve negotiation. HUNTCT provides a complementary capability to the linear frequency domain models selected for investigating the lateral stability and curve negotiation regimes. The model was developed at Wyle and hence its use is well understood by the Contractor. It has been used in comparison with TDOP Phase I data and also to predict responses at the RDL test facility.

Some experimental validation has also been carried out. In a preliminary comparison with TDOP Phase I data, the program has been shown to demonstrate the fundamental hunting mechanisms and has predicted kinematic wavelength and critical speed information. Phase I test data are limited in the amount of information suitable for investigating hunting behavior and validating the forces and amplitude responses predicted by the program. The effort to validate HUNTCT is referenced in two internal memos. (3) The verifiability of the model has been demonstrated by the facility with which model output has been compared with Phase I test data.

(3) Wyle Internal Memo, "Validation of the Wyle Hunting Program with TDOP Data," No. W-071778.4, M. Healy, July 1978, and "Comparisons, HUNTCT and Phase I Test Data," No. W-071079.3, M. Healy, July 1979.

The usage cost for HUNTCT is proportional to the amount of time simulated. In general, the cost will be higher than for a frequency domain program involving a complete analysis. However, it is a relatively efficient time domain model. Typical applications will incur costs of from $1.00 to $2.00 per second of simulated time.

Conclusions

The HUNTCT Model provides a potentially useful means of examining the details of the track/train dynamic interactions in the time domain. The program will have its primary benefit in its application to the lateral stability and curve entry and exit studies. The model accounts for many nonlinearities in representing the truck. The basic number of degrees of freedom is 21 with additional degrees added for each mode of carbody flexibility represented. The model has been developed and used extensively by the TDOP Phase II Contractor. Some preliminary validation has been carried out but additional validation with Phase II data is required.

4.5 FREIGHT CAR CURVING MODEL (AAR)

Introduction

The Freight Car Curving Model is a nonlinear analysis program which uses time integration techniques to simulate the dynamic curving behavior of railroad freight cars. This model allows 43 degrees of freedom and features particularly detailed representation of the trucks. Nonlinearities include spring bottoming, clearances, and Coulomb damping.

Model Application Areas

The Freight Car Curving Model has been specifically developed to investigate curve negotiation including entry and exit, and could be applied in that area as part of the TDOP Phase II analytical task.

Model Description

The program employs 43 degrees of freedom to represent the dynamics of a freight car. Each truck is represented by five masses. These are the bolster, the two side frames, and the two wheelsets. Each of those masses has four degrees of freedom consisting of lateral, yaw, vertical, and roll motions. Thus, there are 20 degrees of freedom associated with each truck. The remaining three degrees of freedom are associated with the rigid carbody (which is free to translate laterally and vertically) and the roll. The masses in the model are connected by nonlinear springs and dampers. Spring characteristics, for instance, may be defined to represent bottoming and free clearance.

The rails are characterized by effective lateral and vertical stiffness and damping elements. The track can be divided into tangent, spiral, or constant curvature segments.

The equations of motion for the system are determined by applying Lagrange's equation for a nonconservative holonomic system. The result is 43 second order differential equations. The solution of these equations gives the time response of the system to the given input of track curvature. Because of the Lagrangian derivation of the equations, the modification of the program to represent Type II trucks may be difficult.

Program Implementation

The Freight Car Curving Model is implemented in FORTRAN and has been installed on an IBM 370/158. The equations of motion are integrated using a Wilson-Theta or Runge-Kutta numerical integration technique to obtain the time response.

The required input data include vehicle physical data, geometrics, mass and inertias, spring and damping coefficients, clearances, track curvature, initial conditions, extreme forces, time step, total simulation time, options, and printing required.

Printed program output consists of the input data and the mass, stiffness, and damping matrices (optional) and at each print interval during the simulation the displacement for each degree of freedom, center plate and side bearing deflections and loads, suspension deflections and forces, wheel/rail friction and flange forces, vertical wheel loads, the L/V ratio, damping forces, and the track curvature forcing functions.

Plotted output is also available. Up to 15 of the following variables may be plotted: lateral displacement of axle 1, flange forces for each wheel of axle 1, L/V ratio for the outer wheel of axle 1, four center plate loads, four sidebearing loads, and roll angle of the carbody.

No formal documentation of this program is known to exist. Information on the program has been obtained in the form of two technical notes by Willis and Smith (4) and Garg and Singh. (5)

Assessment

This program provides the capability of simulating the dynamic response of a freight car during curve entry and curve negotiation. Specifically, the program would permit the calculation of dynamic wheel/rail forces, L/V ratios, wheelset lateral displacements, and angle of attack of wheelsets.

The truck and carbody models are sufficiently detailed so that there is a clear correspondence between model elements and the actual physical components. This should permit an accurate determination of model parameters. Although the model is somewhat complex, comparable simulations have been used by the Contractor without precision problems. The detail of the model offers the potential for close agreement with test results, however, no previous validation work has been reported.

There are two drawbacks to the flexibility of the program. First, the track input consists only of idealized curvature data. No allowance is made for actual irregularities which may be present in actual test results. Second, the Lagrangian methods used in the derivation of equations make modifications of the model more difficult.

The model will be relatively expensive to use, both from the standpoint of computer costs (approximately six minutes execution time on the IBM 370/158 per second of simulated time) and the user preparation time, since the amount of input data is extensive.

Conclusions

The Freight Car Curving Model provides the capability of detailed simulation of curve negotiation dynamics.

The model is a nonlinear, 43 degree-of-freedom representation. Time integration is used to solve the equations of motion. The level of detail in the simulation offers the possibility of good validation with test results, however, no validation has been carried out to date. Because of the complexity of the model, lack of documentation, and relatively high cost, this model was not selected for validation.

4.6 9 DOF STEADY STATE CURVING MODEL - LAW AND COOPERRIDER

Introduction

The 9 dof Steady State Curving Model is a nonlinear model which can be used to study the curving behavior of a freight car in terms of forces and displacements developed in a constant radius turn. The model represents a standard three-piece roller bearing truck.

Model Application

The model is specifically oriented towards the curve negotiation performance regime within the TDOP analytical framework. Its main usefulness is in calculating estimates of the slip and flange contact boundaries for nonlinear vehicles.

Model Description

The nine degrees of freedom considered in this model are lateral, yaw, and warp motion for each of two trucks and lateral, yaw, and roll motion for the carbody. The carbody is assumed to be rigid. Nonlinearities which are considered include wheel/rail geometric constraint functions and suspension elements. The wheel/rail geometric constraints are handled by either the symmetric wheel/rail constraint subroutine or the asymmetric wheel/rail constraint subroutine which are assessed elsewhere in this section. The creep force versus displacement relationship is considered to be linear and expressible in terms of Kalker's creep coefficients. The track is assumed to be rigid.

(4) Willis, T. and Smith, K.R., "A Mathematical Simulation of the Curve Entry and Curve Negotiation Dynamics of Flexible Two Axle Railway Trucks," ASME Paper No.-76-WA/RT-14.

(5) Garg, V.K. and Singh, S.P., "Dynamic Curve Negotiation Behavior of a Freight Car," Session 34, Paper E.3, Heavy Haul Railways Conference, Perth, Australia, September 1978.

The steady state equations for the model are of the form:

$$\left[A(\underline{\lambda},\underline{K})\right]\{x\} = \left[B(\underline{\lambda},\underline{K})\right]\begin{Bmatrix}1/R\\ \theta_d\end{Bmatrix} + [E]\begin{Bmatrix}H_o\\ M_o\\ N_o\end{Bmatrix}$$

where $[A(\underline{\lambda},\underline{K})]$ is the coefficient matrix having elements that are functions of $\underline{\lambda}$ and $\underline{K}$; $\{x\}$ is the n element vector of vehicle components displacements, where n is the number of degrees of freedom for the particular model; λ is the vector of wheel/rail geometric coefficients (composed of λ, Δ, a, and δ_o for each wheelset) and $\underline{K}$ is the vector of effective slopes or "spring" constants for the nonlinear suspension elements. The right hand side consists of the curve input term, $[B(\underline{\lambda},\underline{K})]\ [1/R_{\theta d}]\ T$, and the vector, $[E][H_o\ M_o\ N_o]\ T$, where H_o, M_o, and N_o are the external resultants of coupler forces and moments from other cars and $[E]$ is the nx3 matrix that allocates $[H_o M_o N_o]\ T$ to the correct equations; $[B(\underline{\lambda},\underline{K})]$ is an nx2 matrix composed of vehicle geometry, inertial, and suspension terms.

Program Implementation

The program is coded in FORTRAN and has been run on an IBM 370 and Interdata machine.

The solution of the steady state equilibrium equations is accomplished by iteration. At each successive step, a tentative solution vector, x is produced. If that solution vector is different by more than some small amount from the previous solution vector, an additional step is performed to obtain an improved solution vector, etc., until convergence is achieved. At each step, the spring rates and geometric constraints are updated so as to be consistent with the current solution vector.

The program input consists of the nonlinear wheel/rail constraint functions (generated by the auxiliary routines as pointed out earlier), force-deflection curves for the suspension elements, external forces and moments, vehicle geometry, masses, track curvature, and cant deficiencies to be analyzed.

The iterative solution procedure is applied once for each value of cant deficiency specified as input up to the point of flange contact. After flange contact is determined, the maximum effective friction coefficients are compared to the adhesion level. The maximum radius, if any, at which the maximum effective friction coefficients are sufficiently close to the adhesion limit is designated as that for wheel slip. The next value of cant deficiency is then used and the entire procedure repeated.

Program output consists of a summary of the slip and flange contact results in both printed and plotted form.

No formal documentation is known to exist for this program.

Assessment

This program provides the capability to determine the steady state curving performance for Type I trucks. It also permits the investigation of the effects of parameter variations on steady state curving performance. The program does not allow any axle freedom and thus cannot be adapted to Type II trucks.

The program has been installed and run on the in-house Interdata computer. The significant nonlinearities associated with curve negotiation are accounted for in the model, and the model elements are sufficiently well defined for Type I trucks so that model parameters can be accurately determined. The number of equilibrium equations is sufficiently small and convergence has been found to be fairly rapid so that computational precision is expected to be good.

The validation of the model is currently under study by Law and Cooperrider. It is expected that validation will be successful for curving in which flange contact does not occur. The TDOP validation of this model will depend on the accurate measurement of wheel/rail forces during Phase II testing.

The use of the program has been found to be fairly easy. The output is readily interpreted and should be easily relatable to test data. Also, the program is relatively efficient.

Conclusions

The 9 dof Steady State Curving Model is a nonlinear model which is useful for predicting the steady state curving behavior of freight cars with Type I trucks. Its main usefulness is in calculating estimates of the slip and flange contact boundaries for nonlinear vehicles. It is expected to be computationally efficient and relatively inexpensive to use. The shortcomings of the model which may limit its usefulness to TDOP Phase II are its inability to predict steady state performance during flange contact and its inability to represent axle freedom of Type II trucks.

4.7 17 DOF STEADY STATE CURVING MODEL - LAW AND COOPERRIDER

Introduction

The 17 dof Steady State Curving Model is a nonlinear model which can be used to study the curving behavior of a freight car in terms of forces and displacements developed in a constant/radius turn. The model is similar to the 9 dof Steady State Curving Model by the same researchers but allows extra degrees of freedom which permit the representation of some Type II trucks.

Model Application

Like the 9 dof model, this model is specifically oriented towards the curve negotiation performance regime within the TDOP analytical framework. Its main usefulness is in calculating estimates of the slip and flange contact boundaries for nonlinear vehicles.

Model Description

In addition to the nine degrees of freedom in the earlier model, the 17 dof Steady State Curving Model includes lateral and yaw freedom for each of the four wheelsets. The stiffness and damping characteristics associated with those additional degrees of freedom, like those of the original 9 dof, may be defined to be nonlinear. This program likewise uses the Law and Cooperrider wheel/rail geometric constraint routines. The creep force vs. creepage relationship is considered to be linear and expressible in terms of Kalker's creep coefficients. The track is assumed to be rigid.

The form of the system equations and their derivation follows that of the 9 dof model (see 9 dof Steady State Curving Model Assessment).

Program Implementation

The program is coded in FORTRAN and has been run on an IBM 370 machine.

The solution of the steady state equilibrium equations is accomplished by iteration. At each successive step, a tentative solution vector, x, is produced. If that solution vector is different by more than some small amount from the previous solution vector, an additional step is performed to obtain an improved solution vector, etc., until convergence is achieved. At each step, the spring rates and geometric constraints are updated to be consistent with the current solution vector.

The program input consists of the nonlinear wheel/rail constraint functions (generated by the auxiliary routines as pointed out in the Model Description Section), force-deflection curves for the suspension elements, external forces and moments, vehicle geometry, masses, track curvature, and cant deficiencies to be analyzed.

The iterative solution procedure is applied once for each value of cant deficiency specified as inputs up to the point of flange contact. After flange contact is determined, the maximum effective friction coefficients are compared to the adhesion level. The maximum radius, if any, at which the maximum effective friction coefficients are sufficiently close to the adhesion limit is designated as that for wheel slip. The next value of cant deficiency is then used and the entire procedure repeated.

Program output consists of a summary of the slip and flange contact results in both printed and plotted form.

No formal documentation is known to exist for this program.

Assessment

This program provides the capability to determine the steady state curving performance for Type I trucks and some Type II trucks since the additional axle freedoms are accounted for. It also permits the investigation of the effects of parameter variations on steady state curving performance.

The significant suspension and wheel/rail interface nonlinearities which affect curve negotiation are accounted for in the model and the model elements are sufficiently well defined for Type I and some Type II trucks so that model parameters can be accurately determined. Even with 17 dof, the number of equilibrium equations is still sufficiently small and convergence has been found to be fairly rapid so that computation precision is expected to be good. No previous attempts to validate this program are known to be reported. The prospects for validation appear to be good for curving in which flange contact does not occur. The TDOP validation of this model will depend on the accurate measurement of wheel/rail forces during Phase II testing. Verifiability will be satisfactory as with the 9 dof Curving Model.

Conclusions

The 17 dof Steady State Curving Model is a nonlinear model which has been selected for validation for predicting the steady state curving behavior of freight cars with Type I and some Type II trucks. Its main usefulness is in calculating estimates of the slip and flange contact boundaries including the effects of nonlinear vehicle suspension elements. It is expected to be computationally efficient and relatively inexpensive to use. The shortcomings of the model which may limit its usefulness to TDOP are its inability to predict steady state performance during flange contact.

This program has not been implemented by Wyle personnel, therefore, no assessment of the ease of use or overall utility of the program is as yet possible. It is expected that this program will be somewhat more complex to use due to the additional input required and that cost will increase by at least $(17/9)^2$ over the 9 dof version due to the additional computation required.

4.8 DYNALIST II (TSC) - HASSELMAN AND BRONOWICKI

Introduction

The DYNALIST program is a general purpose computer program which solves systems of linear second order differential equations. Dynamic models of freight cars with up to 50 degrees of freedom can be analyzed both in the time and frequency domains. DYNALIST can be applied to a number of performance regimes including lateral stability, trackability (harmonic roll and bounce subset), and ride quality. DYNALIST was developed under the auspices of the DOT's Transportation Systems Center (TSC) by J. H. Wiggins Company.

Model Application Areas

Because of its generality and its ability to perform both time and frequency response analysis, this program can be used in a variety of different applications. DYNALIST, like the linear frequency domain models, can be used to produce estimates of critical hunting speeds. The limitation in this regard is in the ability to represent inherently nonlinear trucks with a linear model. In the linear time domain DYNALIST can be used to estimate harmonic roll behavior.

Although linear modeling may be overly simplified for obtaining quantitatively accurate results for hunting and harmonic roll phenomena, linear models are likely to be sufficient for ride quality analysis. Hence, DYNALIST can readily be applied to this performance area.

The flexibility of DYNALIST should prove particularly useful in comparing the effects of Type I versus Type II trucks.

DYNALIST can readily compute performance indices which are linear combinations of displacements, velocities, accelerations, and forces.

Model Description

The DYNALIST II program has no particular model structure but rather the program allows the user to define the structure by means of the input. The structure may be composed of rigid bodies, wheelsets with lateral degrees of freedom, model mass elements, springs, and dampers. Flexible bodies can also be included by using an appropriate modal representation. The program determines the equations of motion for the system defined by the user in the general form.

$$[M]\ \ddot{x} + [C]\ \dot{x} + [K]\ x = F(t)$$

Note that the mass, damping, and stiffness matricies [M] [C], and [K] may be asymmetric. The model is limited to a total of 50 degrees of freedom. The forcing function F(t) can be harmonic, periodic, or random in character. The capability of simulating arbitrary periodic inputs allows the representation of transient responses.

Program Implementation

The FORTRAN source code for DYNALIST was obtained from TSC and the program was successfully run on CDC machines.

The solution to the equations is by matrix manipulation which first transforms the system matrices in the arbitrary user defined coordinate system to the generalized coordinate system which decouples the equations. Eigenvalues and eigenvectors for the system are then solved for and the forced response in the user defined coordinate system is obtained by superposition of the appropriate normal modes.

The required program input consists of data which define the type of components making up the model, geometrical data which orient the components relative to one another, physical parameters (masses, spring and damping rates) and data defining the forcing function.

Program output can be plotted as well as printed. An extensive graphics capability is built in to the DYNALIST software. Response quantities which can be obtained from DYNALIST include acceleration, velocity, and displacements at node points or at intermediate points, i.e., linear combinations of node point responses. Internal spring and damper forces can also be included in the output. Documentation of DYNALIST is available in a four-volume FRA Report. (6) These reports include a Theoretical Manual and a User's Manual with sample cases. The manuals are complete and rather detailed.

Assessment

This program provides a general means of creating a linear model for analyzing freight car dynamics. In terms of TDOP needs, DYNALIST can best be exploited in performing ride quality and harmonic roll investigations, the TDOP areas in which linear analysis can be applied with the least idealization.

DYNALIST has been used previously for freight car modeling and has also been validated by comparison of its results with "text book" sample cases. Accuracy is judged to be good within the limits of linearized modeling. The verifiability is also considered good because of the ability of DYNALIST to provide a wide range of output quantities (accelerations, velocities, displacements of nodes, and linear combinations thereof).

A background in dynamic modeling is needed for using DYNALIST to define model structures. Otherwise, the documentation is such that the program has proven easy to use by the unfamiliar. Also, it should be noted that J. H. Wiggins Company, developers of DYNALIST, are thoroughly familiar with its capabilities and limitations and are participating as subcontractors to Wyle on TDOP.

The cost for running a 17 dof benchmark case on the CDC system was less than $10 per run.

Conclusions

DYNALIST provides a flexible dynamic modeling capability oriented toward rail car applications. Linear models with up to 50 degrees of freedom can be defined. TDOP applications for DYNALIST are likely to be in the ride quality, harmonic roll, and possibly lateral stability areas. The model can be used to obtain both frequency and time domain responses. DYNALIST has been used in the past by a variety of organizations and is well documented. The potential for validation of linear DYNALIST models for ride quality and harmonic roll areas is judged to be good. The program is relatively efficient and inexpensive to run.

4.9 FULL, HALF, FLEX, AND LATERAL (TSC) - PERLMAN AND DIMASI

These four models constitute a suite of programs which are intended to provide a comprehensive modeling capability for freight car dynamic behavior. Each of the models is discussed in detail in the following subsections. Although these models are linear and sufficiently simple so that they could be duplicated by appropriate DYNALIST modeling, the group has been included as validation candidates because they are fully developed, available, and fairly well documented.

4.9.1 FULL

Introduction

FULL is a linear frequency domain model of the vertical dynamics of a rail vehicle. It is a relatively simple model with six degrees of freedom represented.

Model Application Areas

This model could be used in TDOP Phase II to evaluate the first order effects of vertical suspension parameters on ride quality. As a simple, linear model, its main utility is in providing qualitative results which allow the analyst to gain insight into the physical interactions which are represented and to identify trends.

The model is sufficiently general so as to include either Type I or Type II trucks.

(6) Bronowicki, A. and Hasselman, T.K., "DYNALIST II, A Computer Program for Stability and Dynamic Response Analysis of Rail Vehicle Systems," Report No. FRA-OR&D-75-22, July 1976.

Model Description

The FULL Model configuration is depicted in Figure 4-4. The six degrees of freedom represented are bounce and pitch of each of the two rigid body trucks and bounce and pitch of the rigid carbody. Damping and stiffness elements are all considered to be linear.

The wheel/rail interface is idealized as a fixed sinusoidal wave shape. The wheelset and truck spacing and the forward speed of the vehicle determine the phase difference between the sinusoidal input at the individual wheelsets. Optionally, the track may be represented in terms of a power spectral density, however, the subroutine RAILPL, which makes this feature possible, is not included in the documentation.

The equations of motion for the model are derived using Newtonian methods. Besides linearity of the springs and dampers, small amplitude displacements are also assumed.

The model can be applied to Type I and Type II trucks.

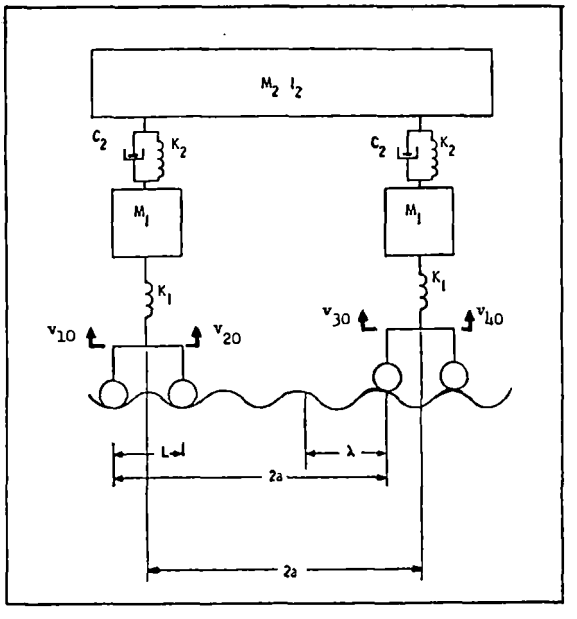


Figure 4-4. FULL Model Moving Over a Sinusoidal Rail Irregularity

Program Implementation

FULL is coded in FORTRAN. It was developed by TSC for a Digital Equipment Corporation (DEC) PDP-10 machine. A listing of the program is included in the available documentation; from this listing, cards were punched to implement FULL on the CDC Cybernet system accessible by the Contractor.

The equations of motion for this simple model are solved directly to obtain frequency domain transfer functions.

Program input consists of values for the masses, spring and damper coefficients, and geometrical data, as well as forward speed and program option control data.

Program output consists of transfer function data (transmissibility versus frequency) and various displacement and acceleration response data versus frequency for the given input. The TSC version of the program calls up TSC in-house plotting routines. Equivalent routines from the Contractor's program library can be used.

The program documentation for FULL is combined with that for three other frequency domain analysis programs generated by TSC: HALF, FLEX, and LATERAL. (7)

The documentation is contained in two volumes available from NTIS. Volume I is the Technical Report and Volume II contains the Appendices. The Technical Report contains discussions on application of the model, description of the model, equations of motion, solution procedure and program flow, description of input and output, and examples of plotted output data. The Appendices contain the input data format and program listing.

In general, the program documentation is good. A nomenclature section providing a clear definition of all program variables would be helpful. More complete discussion of control inputs and input parameter ranges is needed and a complete sample case would be very helpful.

Assessment

FULL is a model which can be used for first order analysis of ride quality behavior. Its main advantage is its simplicity and ease of use.

Although the program has been used by TSC for rail vehicle dynamic analyses, no other users are known. The experience of the Contractor with FULL is limited to that gained in implementing the program on the CDC system and verifying its operation.

The documentation does not discuss any analytical or experimental validation of the program. It is recommended that an analytical validation be carried out on the program as implemented by the Contractor. That task should be relatively simple considering the nature of the model. FULL is not expected to yield close quantitative agreement with test data. Rather, the model will be considered validated if test data trends and qualitative results, particularly in the vertical ride quality regime, are predicted. Carbody accelerations over the trucks, as well as at the center, will be the principal response quantities to be monitored.

Cost for running this simple model are low; the sample case provided by TSC was run for under $3 on the CDC Cybernet System.

Conclusions

FULL is a six dof linear frequency domain model of the vertical dynamics of a rail vehicle. The model has TDOP Phase II applicablity for the ride quality regime only. It is a very simple, easy to use program which may be used for both Type I and Type II truck evaluations.

(7) Perlman, A.B. and DiMasi, F.P., "Frequency Domain Computer Programs for Prediction and Analysis of Rail Vehicle Dynamics," NTIS Report No. FRA-OR&D-76-135.II, Vol. I & II, December 1975.

4.9.2 HALF

Introduction

HALF is a linear frequency domain model of the vertical dynamics of half a rail vehicle and the track structure. Because of its extremely simplified representation of the vehicle, this model is not appropriate in the frequency regime where carbody pitch is excited. Two degrees of freedom are used in representing the half vehicle and truck.

Model Application Areas

This model has very little application to TDOP Phase II. The structure of the model can be reproduced very simply by DYNALIST, for instance, which has the advantage of allowing both time and frequency domain analysis. The HALF Model purports to offer a unique representation of track deflections and wheel loads (see the Assessment paragraph for an explanation of how simpler techniques could be used).

Model Description

The HALF Model configuration is shown in Figures 4-5 and 4-6. Figure 4-5 shows the 2 dof half vehicle/truck model while Figure 4-6 shows the representation of the track. The model is used to obtain solutions for the response of the system due to a sinusoidal variation in the unloaded vertical position of the track at a given forward speed.

The equations describing the motion of the half vehicle/truck are developed from Newtonian methods. A closed form solution for the track deflections is also used.

The model has no restrictions as to Type I or Type II trucks.

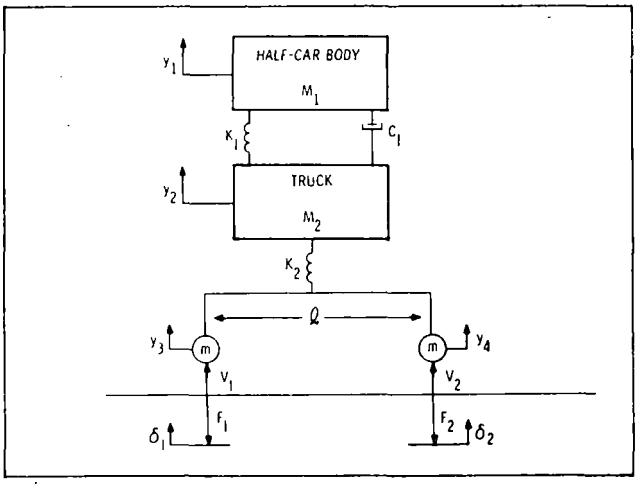


Figure 4-5. HALF Vehicle Model

Program Implementation

The program is coded in FORTRAN. It was developed by TSC for a DEC PDP-10 machine. The documentation includes a program listing from which cards were punched to implement HALF on the CDC Cybernet system accessible by the Contractor.

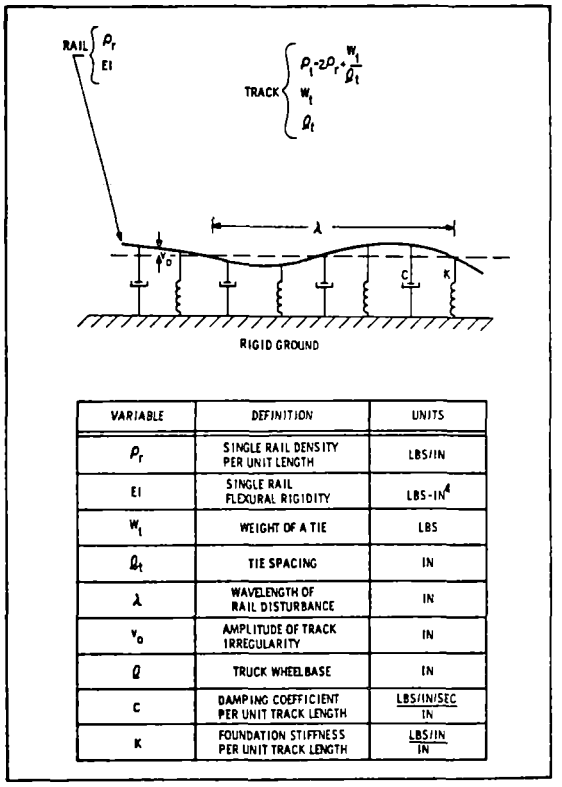
As with FULL, transfer function techniques are used to obtain solutions.

Program input consists of values for the masses, spring and damper coefficients, wheel base, track characteristics, and forward speed.

Program output includes transfer function data versus frequency. The program produces printed and plotted output using equivalent plotting routines which were substituted for those in the TSC version.

The program documentation for HALF is combined with that for three other frequency domain analysis programs generated by TSC: FULL, FLEX, and LATERAL. (8)

The documentation is contained in two volumes available from NTIS. Volume I is the Technical Report and Volume II contains the Appendices. The Technical Report contains discussions on application of the model, description of the model, equations of motion, solution procedure and program flow, description of input and output, and examples of plotted output data. The Appendices contain the input data format and program listing.



VARIABLE	DEFINITION	UNITS
ρ_r	SINGLE RAIL DENSITY PER UNIT LENGTH	LBS/IN
EI	SINGLE RAIL FLEXURAL RIGIDITY	LBS-IN^4
W_t	WEIGHT OF A TIE	LBS
ℓ_t	TIE SPACING	IN
λ	WAVELENGTH OF RAIL DISTURBANCE	IN
y_o	AMPLITUDE OF TRACK IRREGULARITY	IN
ℓ	TRUCK WHEELBASE	IN
C	DAMPING COEFFICIENT PER UNIT TRACK LENGTH	$\frac{\text{LBS/IN/SEC}}{\text{IN}}$
K	FOUNDATION STIFFNESS PER UNIT TRACK LENGTH	$\frac{\text{LBS/IN}}{\text{IN}}$

Figure 4-6. HALF Track Model

(8) Ibid.

In general, the program documentation is good. A nomenclature section providing a clear definition of all program variables would be helpful. More complete discussion of control inputs and input parameter ranges is needed and a complete sample case would be very helpful.

Assessment

The HALF Model does not appear to offer any unique capabilities with regard to TDOP needs. On the one hand, the simplification of the vehicle/truck model can only be justified under a limited set of circumstances and, on the other, the representation of the track is overly complicated for the following reason. The track is represented as being continuous resting on a continuous damped, elastic foundation. In steady state with such a configuration, there can be no vertical dynamic contribution due to the track and foundation itself. Only the track irregularities in the at-rest condition produce any dynamic contribution. Because the entire system is linear, the problem could have been separated into two parts and the steady state solutions superimposed (see Figure 4-7).

Solutions to the nondynamic steady state solution are well documented. (9) The remaining half of the solution could easily be done closed form also.

The extent of TSC use or validation attempts with this model have not been reported. Because of its limited applicability, it is recommended that validation of this model be given a very low priority.

In terms of utility, the program is rather simple and inexpensive to use.

Conclusions

HALF is a very simplified model of limited application to the TDOP analysis of vertical freight car dynamics. The model purports to provide the capability of predicting wheel/rail vertical forces, however no evidence of validation is known to exist. The representation of the track is overly elaborate considering the simplification of the vehicle portion of the model. It is recommended that validation attempts with this model be given a very low priority.

(9) Kenney, J.T., "Steady State Vibrations of Beam on Elastic Foundation for Moving Load," Journal of Applied Mechanics, Trans. ASME, Vol. 21, No. 4, December 1954, pp. 359-364.

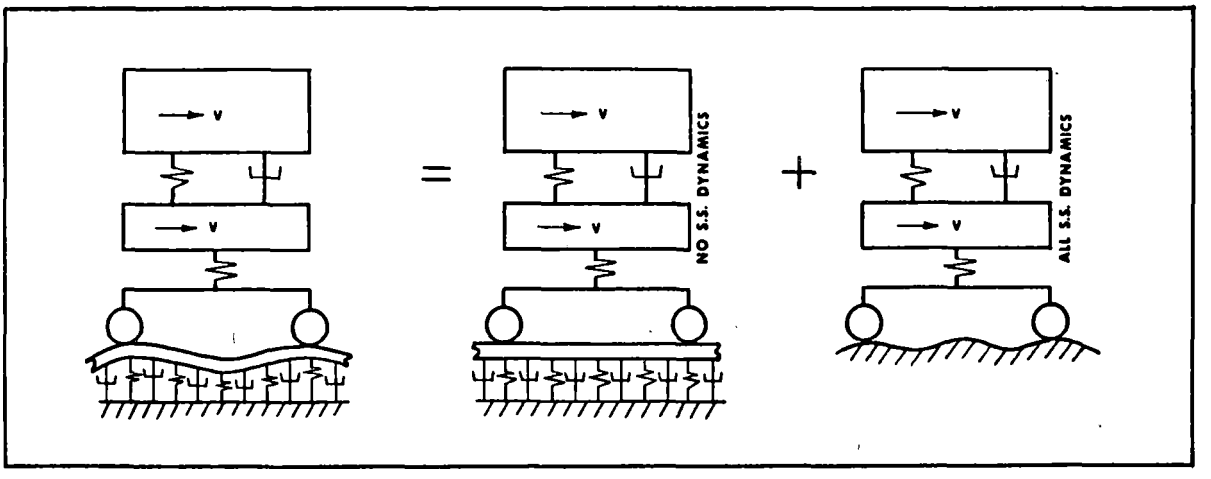


Figure 4-7. Superposition of Configuration Equivalent to HALF Model

4.9.3 FLEX

Introduction

The FLEX Model is a linear frequency domain program for analyzing the vertical dynamics of rail vehicles. The model provides four basic degrees of freedom in the vehicle and trucks and adds another degree of freedom to represent first mode bending of the carbody. Another degree of freedom is also added for a lumped mass representing lading suspended from, or supported by, the vehicle carbody. There are thus a total of six degrees of freedom in this representation.

Model Applications Areas

This model could be applied like the other TSC Model, FULL, to investigate first order effects of suspension parameters on ride quality. The addition of the carbody bending is intended to make the program useful for evaluating the response of vehicles having first mode bending natural frequencies likely to be excited by the content of the track input. The addition of the extra lumped mass for lading provides a means for evaluating interactions of the lading with vertical motions of the carbody. This model, like the other TSC models, can be duplicated by a suitable DYNALIST application. Its capabilities therefore are not unique. The main advantage to the program is that it is immediately available, requiring little user preparation other than the input deck.

Model Description

The configuration for the FLEX Model is depicted in Figure 4-8. As can be seen, the only differences between the FLEX and FULL models are the addition of the bending mode and the lading mass.

Four options are available for modeling the carbody flexibility. In the first option, the carbody is represented by a uniform unconstrained beam. The mode shape is approximated by that for a free-free beam. The second option represents the carbody as a distributed mass, and the bending mode shape is specified in tabular form. In the third option, the carbody is modeled as a uniform beam having a constant area moment of inertia and elastic modulus. The distributed mass and bending mode shapes are specified in tabular form. The fourth option is the most general with the area moment of inertia, the carbody mass distribution, and the bending mode shape input in tabular form.

The wheel/rail interface is idealized as a sinusoidal wave shape. The wheelset and truck spacing and the forward speed of the vehicle determine the phase difference between the sinusoidal inputs at individual wheelset locations.

The equations of motion for the system represented by the FLEX Model were developed by Lagrangian methods.

The representation of FLEX is suitable, provided a particular Type II truck can be represented by a single rigid body in the vertical sense.

Program Implementation

The program is written in FORTRAN and was implemented by TSC on a DEC PDP-10 computer. The implementation by the Contractor was achieved by keypunching cards as contained in the coding listed in the program documentation from TSC. This was done on the CDC Cybernet system.

The solution involves manipulation of the complex matrices describing the system in the frequency domain to obtain complex transfer functions.

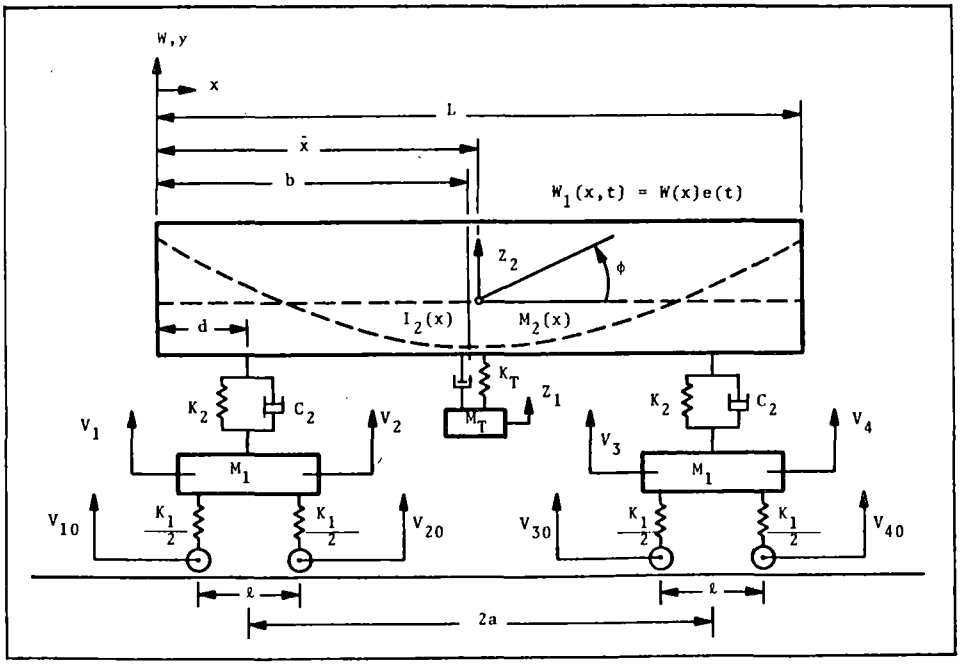


Figure 4-8. Full Car Dynamic Model with Flexible Carbody

Program input consists of values for the masses, spring and damper coefficients, geometrical data, and forward speed. Depending on the program options used for the flexible carbody, additional data may be required to characterize the bending mode shape.

All input values are automatically printed. If intermediate results are requested, the program prints the mass, stiffness, and damping matrices, and the normalized solution vector at each frequency point. The displacement responses as a function of frequency are always printed for three locations: carbody center of gravity, carbody over truck center, and hanging mass center of gravity. Displacement responses are plotted if requested. Acceleration responses and acceleration spectra are printed and plotted if requested. Equivalent plotting routines have been substituted for the routines referenced in the original TSC version of the program.

The program documentation for FLEX is combined with that for three other frequency domain analysis programs generated by TSC: HALF, FULL, and LATERAL. (10)

The documentation is contained in two volumes available from NTIS. Volume I is the Technical Report and Volume II contains the Appendices. The Technical Report contains discussions on application of the model, description of the model, equations of motion, solution procedure and program flow, description of input and output, and examples of plotted output data. The Appendices contain the input data format and program listing.

In general, the program documentation is good. A nomenclature section providing a clear definition of all program variables would be helpful. More complete discussion of control inputs and input parameter ranges is needed and a complete sample case would be very helpful.

Assessment

FLEX can be used in the TDOP application area of ride quality evaluation, however the linear approximations and other idealizations are not likely to yield better than qualitative results. Use of FLEX must also be considered in light of the duplicate capabilities of the more general DYNALIST program.

The Contractor's familiarity with FLEX is limited to that which has been gained in the exercise of implementing the program on the CDC Cybernet system and verifying its operation with the sample case provided in the documentation.

The documentation does not discuss any analytical or experimental validation of the program. For experimental validation of the lading motions, no TDOP tests are applicable. It is believed that some Trailer-on-Flatcar (TOFC) data collected at the Transportation Test Center may be used for that purpose. Otherwise, validation requirements for FLEX are similar to FULL. No accuracy or precision problems are foreseen with this model.

The sample case was run on the CDC Cybernet System for less than $10.00.

(10) Perlman and DiMasi, op.cit.

Conclusions

FLEX is a six dof linear frequency domain model. Its unique features are its provision for modeling vertical simple bending of the carbody and for modeling the interactions of carbody with suspended or supported lading. FLEX can be applied in the ride quality area for establishing qualitative relationships and trends in the effect of suspension parameters on vertical dynamics. Validation of the lading interaction portion of the program would have to be done with non-TDOP data since no suitable instrumentation of lading is envisioned in TDOP. As a fairly simple linear model it can easily be understood and used, and it is inexpensive to run.

4.9.4 LATERAL

Introduction

LATERAL is a 15 degree-of-freedom representation of a rail vehicle developed at TSC for evaluating lateral dynamics. LATERAL is a linear frequency domain program.

Model Application Areas

This model can be used in TDOP Phase II to provide qualitative analysis of lateral dynamics particularly relating to ride quality. Because of the linear idealization, close quantitative predictions of vehicle motions are not expected to be possible. Program output can, on the other hand, be related to ride quality performance indices.

Model Description

The configuration of the LATERAL Model is shown in Figure 4-9. The 15 degrees of freedom represented are:

Lateral wheelset displacement	4
Wheelset yaw	4
Lateral truck displacement	2
Truck yaw	2
Lateral carbody displacement	1
Carbody yaw	1
Carbody roll	1
TOTAL	15

The car and truck models involve only rigid body motions. Futhermore, it is assumed that vertical translation and pitch rotational motions are decoupled from lateral motions.

The wheel/rail interface accounts for gravitational stiffness as well as for the effects of creep forces. Gyroscopic effects can also be included as a user option. The track is represented as being rigid with one of two irregularities. The track is assumed to have a sinusoidal variation in track centerline alignment or crosslevel alignment. The two options are mutually exclusive within the program. The equations of motion for the system represented in LATERAL are derived using Lagrangian techniques. The current representation in LATERAL is suitable only for Type I trucks. Modification of the equations to adapt the model to some Type II trucks is somewhat difficult due to the Lagrangian derivation.

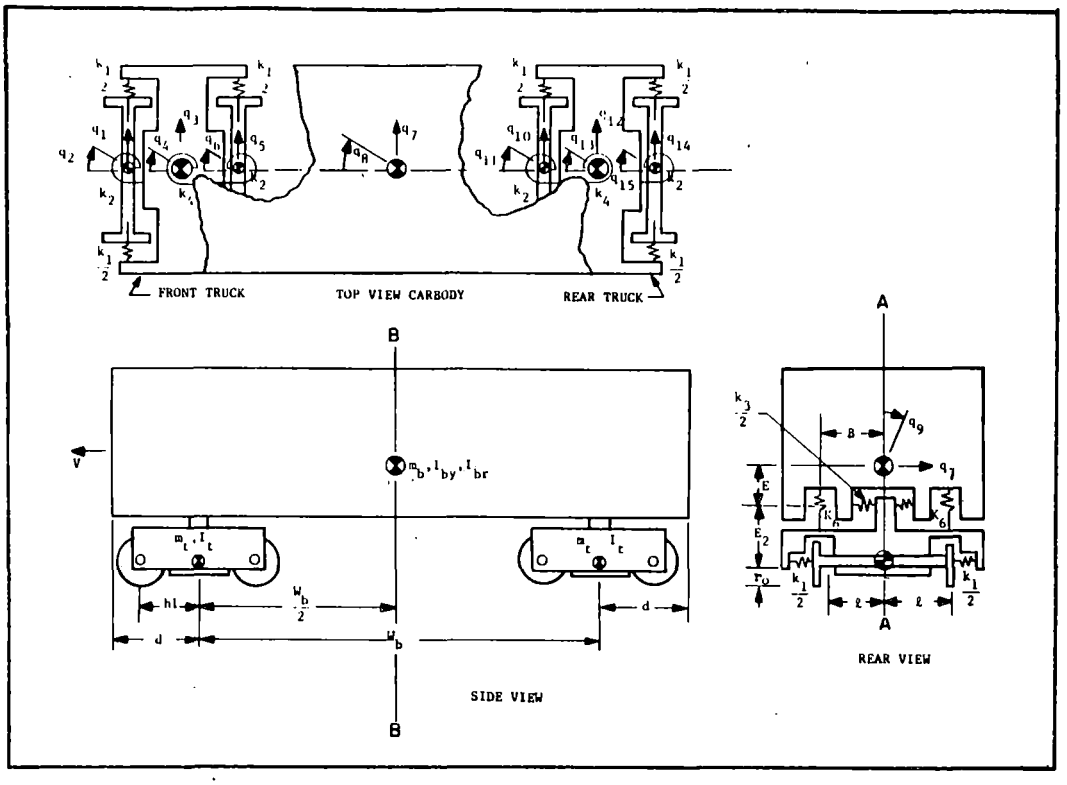


Figure 4-9. Lateral Rail Vehicle Model

Program Implementation

LATERAL has been programmed in FORTRAN. A listing of the source code is included in the program documentation. The program was implemented by keypunching directly from the source code found in the documentation. The solution to the equations of motion are obtained separately for the two different sinusoidal inputs: alignment and crosslevel variation. The solution vector for motions in the 15 degrees of freedom is determined at each frequency of interest by first computing the forcing function coefficients and then premultiplying by the inverse of the complex coefficient matrix obtained from the Lagrangian derived equations of motion.

Program input is composed of the physical data for the components and control logic. The physical data include values for masses, springs, dampers, geometric relationships, and a description of the forcing function. The control logic determines the forcing function option alignment or crosslevel variation and the number of frequencies and form of output to be processed.

Output consists of a printing of the coefficients of the mass, damping, and stiffness matrices describing the system. For each frequency processed the elements of the normalized solution are printed and plotted. Both displacement and acceleration responses are possible.

The program documentation for LATERAL is combined with that for three other frequency domain analysis programs generated by TSC: FULL, HALF, and FLEX. (11)

The documentation is contained in two volumes available from NTIS. Volume I is the Technical Report and Volume II contains the Appendices. The Technical Report contains discussions on application of the model, description of the model, equations of motion, solution procedure and program flow, description of input and output, and examples of plotted output data. The Appendices contain the input data format and program listing.

In general, the program documentation is good. A nomenclature section providing a clear definition of all program variables would be helpful. More complete discussion of control inputs and input parameter ranges is needed and a complete sample case would be very helpful.

Assessment

The TDOP analysis area to which LATERAL may best be applied is in determining the first order effects of suspension parameter changes on ride quality. As a linear model its main utility is in providing qualitative results allowing the analyst to gain insight and predict trends.

Familiarity of the Contractor with this program is limited to that gained in implementing the program on the CDC Cybernet System and running the sample case provided in the documentation.

(11) Perlman and Di Masi, op. cit.

So far as is known, no previous analytical or experimental validation attempts of LATERAL have been reported. It is recommended that the program be verified with more test cases in addition to validation attempts with experimental results. Test results should be easily related to model outputs. The model fidelity should be considered satisfactory if it proves capable of producing sound qualitative results. The accuracy of the model representation is judged to be satisfactory according to the intent of the program. The extent of the system equations is not such as to cause computational precision problems. In validating the program, no special test requirements are needed. The validation process for LATERAL is facilitated by the plotting capability in the program.

The program is considered to be rather easy and inexpensive to use. The sample case run on the CDC system cost less than $7.00.

Conclusions

LATERAL is a 15 degree-of-freedom linear model of the lateral dynamics of a rail vehicle. The model can be applied in the ride quality area for obtaining primarily qualitative results due to the linear idealization. The program is not well suited nor easily modified to represent Type II trucks. It is fairly easy and cost efficient to use.

4.10 TDOP PHASE I MODEL (SPTCo.)

Introduction

The TDOP Phase I Model is a frequency domain program which represents a freight car by 13 degrees of freedom. The objective of the program is to calculate lateral stability characteristics and to a lesser extent predict vehicle motions in regimes such as the harmonic roll excitation.

The model was developed by Southern Pacific Transportation Company (SPTCo.) in conjunction with the testing carried out in TDOP Phase I. Although the frequency domain model requires linearity in the representation, an attempt was made to characterize some nonlinear suspension elements as linear on a rational basis.

Unfortunately, comparison of Phase I test results with predictions from this model have not been good. The reasons will be brought out later in this assessment.

Model Application Areas

The model primarily addresses lateral stability but in theory should also predict general truck and carbody motions including a first order representation of vertical car bending.

Model Description

The 13 degrees of freedom of this model are illustrated by the series of Figures 4-10 through 4-15. The first three of those figures depict the allowable truck motions which are:

Yaw of the side frame pairs	2
Yaw of the bolsters	2
Lateral displacement of the side frames	2

Figures 4-13 through 4-15 show the remaining carbody motions represented by the model. These are:

Roll of the fore half carbody	1
Roll of the rear half carbody	1
Relative pitch of the fore and aft carbodies	1
Vertical displacement of the fore half carbody	1
Vertical displacement of the aft half carbody	1
Yaw of the combined half carbodies	1
Lateral displacement of the combined half carbodies	1

The representation of the carbody in the vertical sense by two lumped masses gives a first order approximation to the first vertical bending mode.

Note in Figure 4-14 that provision is made for including drawbar forces in the model. However, Phase I testing did not include instrumentation of the drawbar forces. The other forces and moments accounted for in this model, in addition to those due to the usual springs and dampers, are gyroscopic moments, moments due to center of gravity offset of the carbodies, gravity restoring forces from the wheel/rail interactions, and creep forces generated at the wheel/rail interface.

The wheel/rail interface is idealized by assuming that the wheel and rail have constant but different radii of transverse curvature at the contact point. The representation results in a single value of effective conicity which can be used in the linear representation.

The representation of the track input is made by decomposing actual track measurements of the Phase I test sites into a 200-term Fourier series at frequencies from 0.1 Hz to 20 Hz at 0.1 Hz increments.

In the development of the TDOP Phase I Model, an attempt was made to characterize the nonlinear Coulomb damping elements by equivalent viscous damping elements in a rational manner. The coefficient for an equivalent viscous damper was determined by equating the energy dissipated per cycle of oscillation at a given frequency and peak amplitude. Reference (12) pertaining to this technique indicates that it may be applied if:

a. The forcing motion is sinusoidal.

b. Motions are continuous (i.e., no mechanical stops).

c. The Coulomb damping does not unduly change the harmonic waveform.

The feasibility of the application of this technique to the TDOP Phase I Model is discussed under the Assessment paragraph.

The equations of motion were developed using Newtonian methods. The derivation of equations (not their implementation in the program) was verified independently by MITRE as part of their TDOP review function.

(12) Jacobsen, L.S. and Ayre, R.S., Engineering Vibrations, McGraw-Hill Book Company, Inc., 1958.

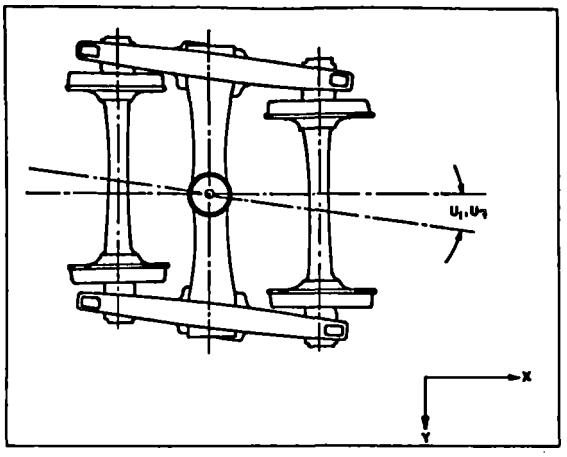


Figure 4-10. TDOP Phase I Model - Yaw of the Side Frame Pairs

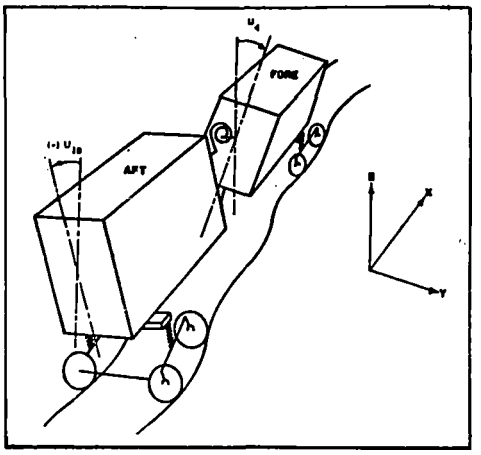


Figure 4-13. TDOP Phase I Model - Roll of the Half Carbody

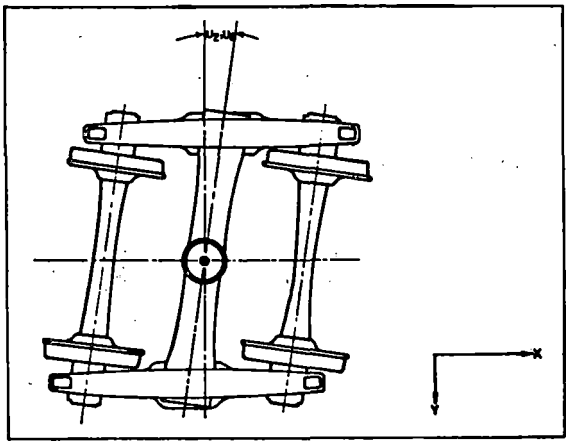


Figure 4-11. TDOP Phase I Model - Yaw of the Bolsters

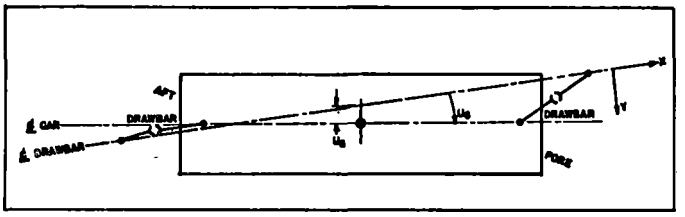


Figure 4-14. TDOP Phase I Model - Lateral Displacement and Yaw of the Combined Half Carbodies

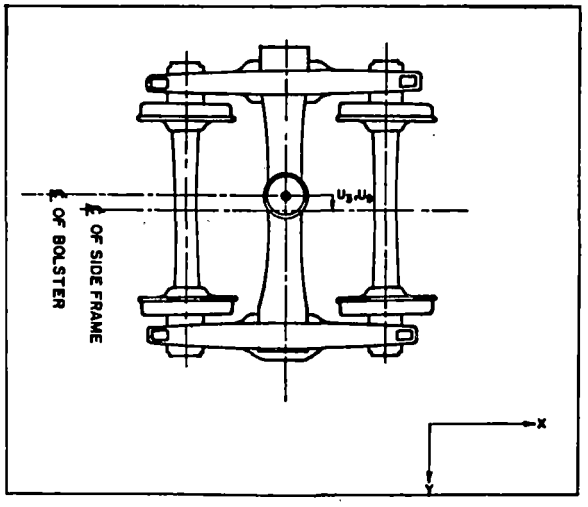


Figure 4-12. TDOP Phase I Model - Lateral Displacement of the Side Frames

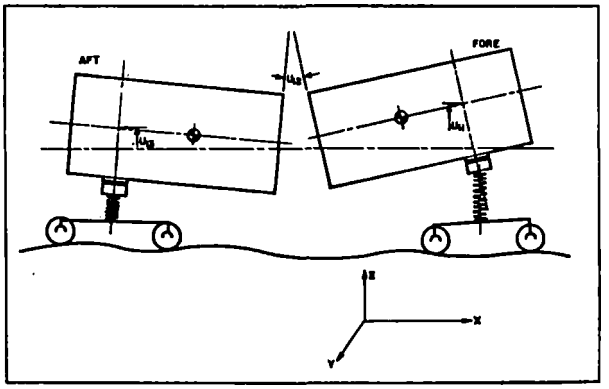


Figure 4-15. TDOP Phase I Model - Relative Pitch and Vertical Displacement of Half Carbodies

Implementation

The computer program for the TDOP Phase I Model is in FORTRAN. It was furnished by the FRA at the outset of Phase II, and has been run successfully on the in-house Interdata system.

The system equations are solved in the frequency domain by inverting the matrix of complex coefficients associated with the 13 coordinates corresponding to the degrees of freedom in the model. The inverted coefficient matrix is multiplied by the vector of forcing functions, which are also in complex frequency form, to obtain the solution of the motions of the 13 general coordinates.

Program input is composed of the following quantities:

a. Speed, drawbar forces, track section

b. Car and coupler length, truck centers

c. Weights of carbody sections

d. Center of gravity location of each section

e. Roll and pitch moments of inertia of each section

f. Moments of inertia and rotational stiffness of entire carbody

g. Truck dimensions

h. Bolster and side frame weights and moments of inertia

i. Wheelset weights and moments of inertia

j. Wheel type, wheel and rail radii, contact angle

k. Spring constants

l. Bolster to side frame rotational resistance

m. Center plate rotational resistance

The program produces both printed and plotted output. Subroutines which drive CALCOMP plotters are used so the program is machine specific in that regard. Twenty-one different computed variables are calculated by the program after solving the simultaneous complex equations for each frequency. These outputs are:

- Bounce, lateral, pitch, and roll acceleration (carbody)
- Yaw acceleration
- Fore truck swivel and tram (parallelogramming)
- Axles 1, 2, and 3 lateral acceleration
- Axle 1 left vertical acceleration (at adapter)
- Axle 1 vertical acceleration
- Axle 1 vertical force
- Fore bounce acceleration
- Car bending acceleration (about lateral (y) axis)
- Fore and aft lateral acceleration
- Car twist acceleration (about longitudinal (x) axis)
- Fore car axle relative roll angle
- Fore bolster-side frame lateral and vertical displacement

These quantities attempt to match the model results as closely as possible to Phase I testing instrumentation for ease of comparison.

These types of output plots are possible:

a. Nyquist diagrams

b. Power spectral density plots

c. Time domain plots (by superposition of individual frequency responses)

Documentation of the TDOP Phase I Model (13) is fairly extensive. It includes explanations of the derivation of equations, the input and output, and provides sample cases. The User's Manual, however, does not point out many limitations of the program which have been uncovered.

Assessment

The TDOP Phase I Model has received considerably more scrutiny than most of the other models which have been assessed as part of this Phase II task. Also it is one of the few models assessed which has been compared extensively with actual test data. It has been found that there is considerable discrepancy between model and test results. Mr. N. Sussman of the MITRE Corporation has previously assessed the shortcomings of the TDOP Phase I Model, also referred to as the Frequency Domain Model or FDM.

The reasons for the inadequacy of the model are summarized from Mr. Sussman's critique (14) below. The Contractor concurs with these criticisms.

- Equivalent Viscous Damping - The technique of using equivalent viscous damping based on equating the cyclic energy dissipation is a valid representation for systems exhibiting sinusoidal or near sinusoidal steady-state behavior. The TDOP Phase I Model attempts to use such a technique without proper regard to the actual motions involved and is hence a substantial source of error.

(13) Southern Pacific Transportation Company, "Math Modeling Report, Vol. I, TDOP Phase I," Interim Report, July 1976.

(14) Sussman, N.E., "Critique of Frequency Domain Model-Solution Techniques," Report No. FRA/ORD-78/-12.IV, February 1978.

- Superposition of Responses - The TDOP Phase I Model employs superposition of steady state responses at 200 discrete frequencies of the track frequency domain representation. While valid for linear systems, the inclusion of Coulomb damping even by means of a viscous equivalent may render superposition of responses invalid, and hence may be an additional source of error.

- Iterative Technique for Obtaining Responses - The TDOP Phase I Model obtains the equivalent Coulomb damping terms by iteration. Mr. Sussman points out that even for one dof systems under a wide range of conditions such an iterative scheme will produce substantial error in the steady state motion amplitudes.

- Power Spectral Density Calculations - The power spectral densities calculated by the TDOP Phase I Model were shown to have little statistical confidence based on the sampling techniques used.

- Other Sources of Errors - These include limitations arising from the track geometry measurement system used and uncorrected programming errors.

The correspondence of model elements with actual freight car elements is generally good, making the definition of model coefficients relatively easy other than the characterization of the nonlinearities previously discussed. The extent of the model is such that computational precision difficulties are not significant.

To the credit of the model is the extent of its output capability and the selection of output quantities to correspond with test measurements. The exercise of validating the model was facilitated because of the good correspondence. Because of the iterative solution technique for each frequency processed, the cost to run the TDOP Phase I Model is considerably greater than an ordinary single pass, fixed coefficient frequency domain model of comparable size. The extent of the entire plotting capability is about a third of the total cost for a run.

Conclusions

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

4.11 FLEXIBLE BODY RAILROAD FREIGHT CAR MODEL (AAR) - TSE AND MARTIN

Introduction

The Flexible Body Railroad Freight Car Model is a nonlinear time domain model, primarily of the vertical and roll dynamics of a rail vehicle. There are 20 degrees of freedom in the representation. The model was developed as part of the AAR Track/Train Dynamics program.

Model Application Areas

The main TDOP Phase II application of this model is in the area of trackability; specifically, of harmonic roll and bounce and ride quality. With modifications to generalize track input, the model could be applied to the track twist load equalization subset of the trackability regime.

Model Description

The model is illustrated in Figure 4-16. Note that the carbody flexibility is approximated to a first order by using two lumped masses with a compliant connection between them. The two wheelsets of a truck are lumped together with the side frame pairs. The track inputs to such a side frame/wheelset combination are the average of the vertical displacements of the front and rear wheelsets of a truck on each side, left and right.

The degrees of freedom are identified as:

Vertical displacement of each half carbody	2
Lateral displacement of each half carbody	2
Roll of each half carbody	2
Pitch of each half carbody	2
Yaw of each half carbody	2
Vertical displacement of each bolster	2
Roll of each bolster	2
Vertical displacement of front and rear side frame/wheelset combinations	2
Roll of each side frame/wheelset combination	2
TOTAL	20

The connection of the carbody to the bolster is modeled using nonlinear springs with a force going to zero as the carbody lifts off the center plate. Nonlinear springs are also used to represent the action of the side bearings. They do not exert any force on the carbody until the roll is such that the clearances are taken up.

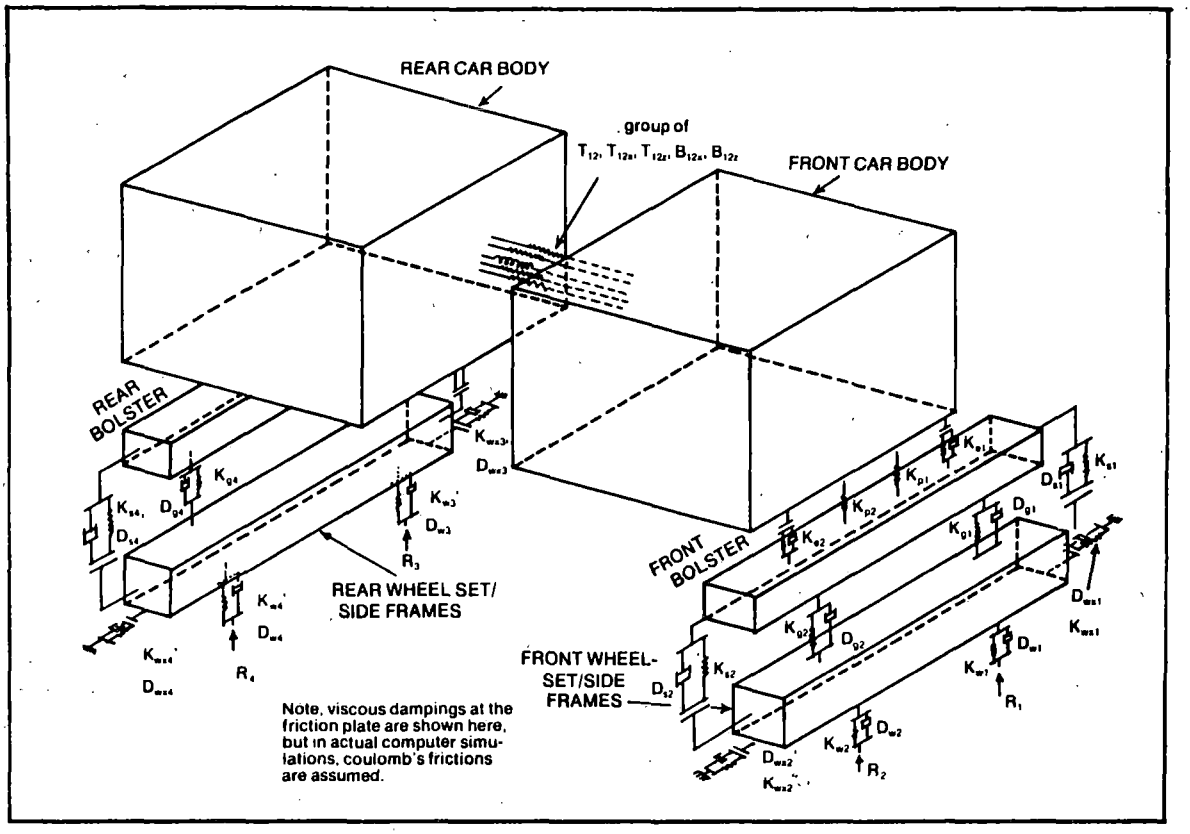


Figure 4-16. Flexible Carbody

The bolster is connected to the side frame/wheelset combination by vertical and lateral springs and Coulomb friction elements. Coulomb damping elements also act when gib clearance is taken up. The track stiffness and damping is included on a lumped element basis. Two basic forcing functions representing the static vertical rail displacement are possible. The first assumes half-staggered rail represented by a rectified sine wave which excites the harmonic roll oscillations. The second assumes sinusoidal track variation with left and right rails in phase which excites bounce oscillations. An updated version of the program allows for other forcing functions which include the effects of curvature and ramps as well as user-defined track input.

The equations of motion are derived using Lagrangian methods. In the derivation, it is assumed that displacements are small and that yawing and pitching of the bolsters and side frame/wheelset combinations is small and can be neglected. Because of the Lagrangian derivation of this rather complex model, the adaptation of the model to represent Type II trucks would be quite difficult.

Program Implementation

FORTRAN is the language used for this program. It was developed for use on AAR's IBM 370/158 computer. Since being acquired by the Contractor it has been successfully run on the in-house Interdata computer.

The equations of motion are cast in normal second order differential form in terms of the Lagrangian generalized coordinates. The equations are first recast as a system of first order differential equations. They are uncoupled by a Gaussian elimination process and finally integrated for the time domain solution using a Runge-Kutta technique. Both the Gaussian elimination and Runge-Kutta routines are standard computer software packages.

The input for the program is relatively extensive as there are a large number of elements to define, many of which require not just a single coefficient but a functional relationship defined.

In addition, the geometry must be defined. The program allows various options in the selection of viscous or Coulomb damping for various damping elements. Also, the program provides a feature whereby the time domain solution may be restarted from the point of termination of a previous run. Care must be taken in the selection of the time step of integration. The user must verify that the chosen time step is small enough so that the integration accuracy is not affected.

Output consists of a partial recapitulation of the input, displacements, velocities, and acceleration components for each degree of freedom, loadings on center plates, side bearings, bolsters, springs, dampers, and track. Spring and rail deflections can also be printed. No plotting capability is offered.

The Flexible Body Railroad Freight Car Model is documented in three volumes available from AAR. (15) The volumes consist of Technical Documentation, a User's Manual, and a Programming Manual. The documentation is comprehensive and quite detailed. It includes two sample cases in the User's Manual.

Assessment

The Flexible Body Railroad Freight Car Model can be used to investigate the area of harmonic roll and bounce behavior of the trackability regime. It also provides a means of studying the effect on ride quality of nonlinear Coulomb damping in terms of vehicle vibration levels.

The level of detail in the Flexible Body Railroad Freight Car Model and its representativeness of Type I trucks are such that it is expected to produce results which will compare favorably with Type I truck test data on a quantitative basis. Without difficult modifications to generalize the model to Type II trucks, its validity for those is questionable.

The carbody representation can be accurately related to measurable freight car parameters. The truck model is representative of Type I trucks in the vertical sense and it is expected that model parameters for such trucks can be accurately defined. The truck model may not represent Type II trucks as well and there may be difficulty in defining model parameters for certain of those trucks. Track and foundation compliance and damping can be accurately defined for this model.

No computational precision problems are expected with this program.

The Technical Documentation volume includes a report on the comparisons which were made of results produced by the Flexible Body Railroad Freight Car Model with a similar independently developed hybrid computer model of the A. Stucki Company. The results from the two models were in close agreement. The exercise served to analytically validate the model. In a validation against an actual test, AAR obtained good agreement between the Flexible Car Body Model and test data from a standard loaded gondola car. Test and model results of the same car with suspension variations were not in as close agreement, however.

The extent of output is sufficient to facilitate the validation task, although the task would be eased further by the provision of a plotting capability.

The program is made difficult to use by the extent of input which must be supplied. The documentation is well organized to ease the task. The experience of AAR indicates approximately two minutes of IBM 370/158 CPU time per second of simulation. The Contractor has found with the sample case that about three minutes of Interdata CPU time is used per second of simulated time.

(15) Tse, Y.H., and Martin, G.C., "Flexible Body Railroad Freight Car," Vol. I Technical Documentation, AAR Document R-190; Vol. II User's Manual, AAR Document R-200; Vol. III Programming Manual, AAR Document R-260.

Conclusions

The Flexible Body Railroad Freight Car Model is a complex, nonlinear time domain program. It can be applied in the harmonic roll and bounce analyses as part of the trackability investigation and also may be used to study ride quality. With a more general representation of track input, the capability of the model would be extended to cover the load equalization area. The model is best suited to representing Type I trucks. Its ability to represent some Type II trucks is questionable and modifications may be difficult due to the complex Lagrangian derivation of the equations of motion. The program has received independent analytical scrutiny; validation efforts with test data have been made although results as yet have not been reported. The program is well documented. It has been selected for validation because it complements the capabilities of DYNALIST and other linear models with similar application.

4.12 FRATE (MITRE) - KACHADOURIAN, SUSSMAN, AND ANDERES

Introduction

The FRATE Model is a nonlinear time domain representation of a freight car with provision for modeling lading, specifically Trailer-On-Flatcar (TOFC) configurations. A total of 27 basic degrees of freedom are involved, 11 of which are used to represent the rigid carbody and truck dynamics. The model provides for the modeling of carbody flexibility in terms of normal modes and nonlinear spring and damping characteristics. Each mode included increases the total degrees of freedom by one. The model is an extension of models developed by M.J. Healy of Wyle Laboratories.

Model Application Areas

FRATE emphasizes the vertical and roll phenomenon of vehicles and lading. It is thus suitable for the areas of harmonic roll and bounce and for ride quality. The truck equations are easily related to the actual truck configuration and can readily be modified if required to reflect a Type II assembly.

Model Description

Figures 4-17 and 4-18 describe the full FRATE/TOFC configuration. The carbody/truck degrees of freedom are:

Lateral displacement of each truck	2
Vertical displacement of each truck	2
Roll of each truck	2
Lateral displacement of carbody	1
Vertical displacement of carbody	1
Yaw of carbody	1
Pitch of carbody	1
Roll of carbody	1
TOTAL	11

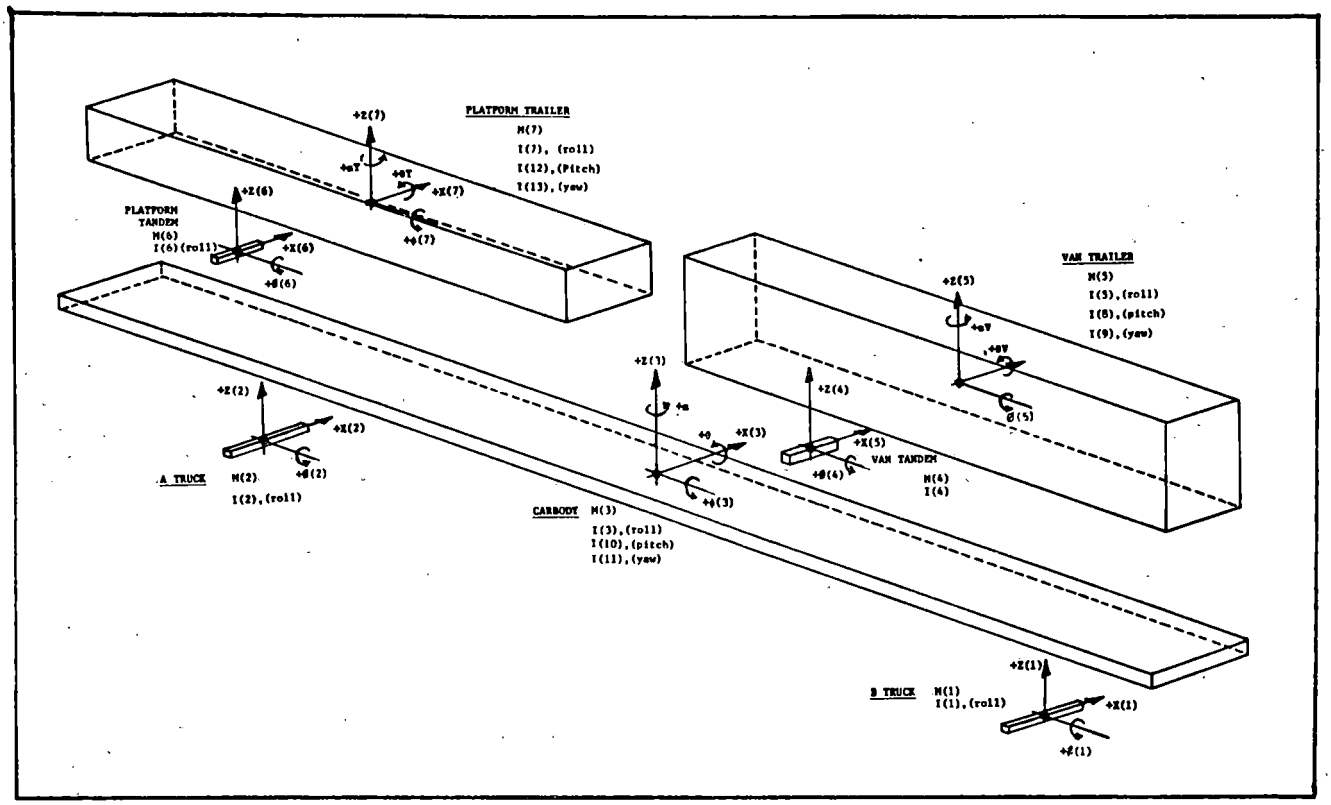


Figure 4-17. FRATE/MITRE Model - TOFC,
Inertia, and Degree of Freedom Notation

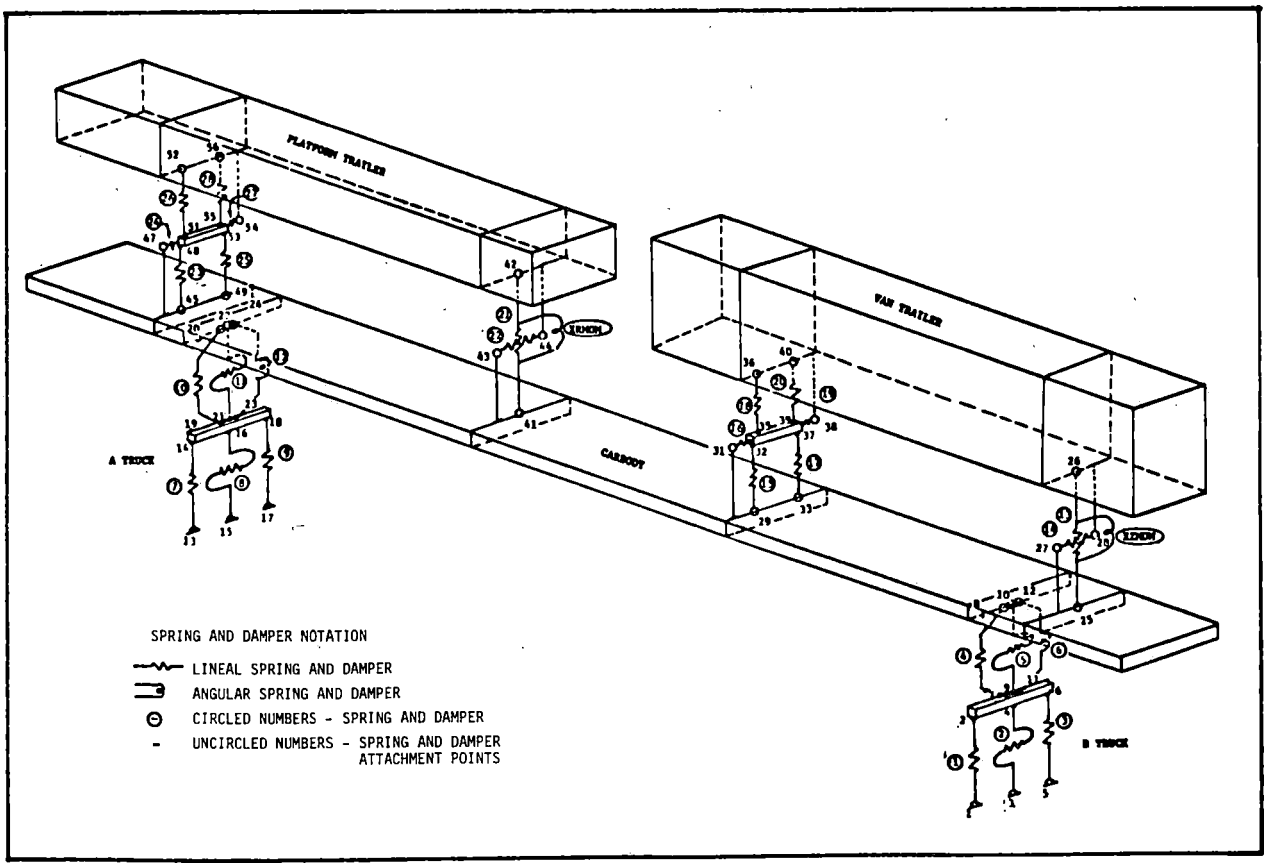


Figure 4-18. FRATE/MITRE Model - TOFC,
Spring Damper Notation

Each TOFC is represented by two rigid masses (trailer body and wheelset) having eight degrees of freedom. By various program options, one or both of the TOFCs can be eliminated, reducing the extent of the model to represent an ordinary freight car. Similarly, the inclusion of carbody flexibility by specifying its normal modes is a user option and can be omitted when a rigid carbody representation suffices. Track compliance and damping characteristics are lumped with other truck suspension elements in FRATE.

The vertical excitation is generated by sinusoidal vertical displacements of the left and right sides of each truck. Thus, inputs from front and rear wheelsets of each truck are averaged. Alternately, the trucks can be excited laterally with sinusoidal displacement variations. In either case, the input frequency may be fixed or may be increased or decreased with time to simulate frequency response sweep testing. Also, the user may specify that the sinusoidal input terminate after so many cycles to obtain a decay response.

The equations of motion are derived as a set of second order differential equations using Newtonian methods. The model includes second order angular motion terms and is thus valid for angular deflections of up to ten degrees. The model can quite readily be modified to account for the differences of particular Type II trucks.

Program Implementation

FRATE has been coded in FORTRAN and implemented on the CDC Cybernet system. Due to the extent of its core requirements, the program cannot be implemented on the Contractor's Interdata system.

The system of second order differential equations describing the motion of the system in terms of the 27+ degrees of freedom are recast as an equivalent set of first order differential equations. That equation set is solved by numerical integration using a Runge-Kutta routine.

Program input is divided into four groups each of which is handled by the convenient FORTRAN NAMELIST convention. The four groups are:

a. Program control variables

b. Excitation variables

c. Vehicle parameters (masses, geometry, spring constants, etc.).

d. Mode shapes data

Program output can be of three types:

a. Time histories

b. Envelopes (maxima and minima of each oscillation are plotted versus time)

c. Debug

Standard sets of variables are printed and/or plotted versus time depending on whether vertical or lateral excitation is specified. The debug option produces printed output only, giving the values of all input and state variables at a particular time step. It is a useful built-in diagnostic feature. There is excellent documentation of the program contained in two volumes. (16) Volume I is a User's Manual while Volume II is a Technical Manual. The User's Manual includes sample input and output.

Assessment

FRATE has capabilities which can be applied to the TDOP Phase II task of analyzing trackability (harmonic roll subset) and ride quality regimes. Its unique features are the inclusion of lumped masses and suspension elements to represent lading such as "piggyback trailers." Also, it allows the option of frequency sweep excitation in both the vertical and lateral sense.

The basic model has been scrutinized both in its original development and by MITRE in its extension of the model. FRATE has also been validated by comparison of its results against test data obtained at the FRA's Rail Dynamics Laboratory (RDL), Pueblo, Colorado. The model proved capable of predicting resonances and mode shapes and of predicting nonlinear phenomena such as the difference in resonance produced in a frequency up sweep versus a down sweep. The verifiability of the model has been demonstrated.

The elements of the model are readily relatable to components of an actual freight car facilitating the accurate definition of model parameters. With due care in the choice of step size to avoid numerical instability in the Runge-Kutta integration process, there should not be any computational imprecision.

As has been shown in the previously described validation efforts, the model is quite capable of producing results which can be compared directly with test data including time history and frequency response plots.

Because of the relation of this model to the original FRATE model, this version can be readily understood and used by the Contractor. Since FRATE is a time domain model, costs for establishing frequency responses will be considerably greater than linear frequency domain models. The utility of FRATE is in its ability to predict effects of nonlinearities on a quantitative basis. Typical costs for a run according to the User's Manual are approximately $20 for a response to single cycle excitation and $90 for a complete frequency sweep.

(16) Kachadourian, G., Sussman, N.E., and Anderes, J.R., FRATE Vol. I User's Manual; Vol. II Technical Manual (pre-publication copy), FRA/ORD-78/59, September 1978.

Conclusions

FRATE is a nonlinear time domain simulation program with application to harmonic roll and ride quality analysis. The model uses 11 degrees of freedom to characterize the trucks and rigid body motions of the car and eight degrees freedom are used for each of two assemblies representing sprung lading (e.g., TOFC). In addition to those 27 degrees of freedom, the model provides for flexibility of the carbody by including its normal modes as additional degrees of freedom.

The model has been validated against RDL tests. The model was selected for further validation in TDOP Phase II on the basis of those favorable results and its flexibility in terms of ease of modification to reflect Type II trucks.

4.13 FRATE 11 (WYLE) - HEALY

Introduction

FRATE 11 is a nonlinear time domain model using 11 degrees of freedom for the basic carbody and trucks. Additional degrees of freedom for body flexibility may also be included using a normal mode representation. The model focuses on vertical and roll dynamics. Nonlinearities treated include Coulomb suspension damping, spring clearances, stops, etc.

Model Application Areas

FRATE 11 can be applied to studies of car and truck vertical and roll motions. Its primary use is for nonlinear analysis of harmonic roll and bounce. It can be used also for predicting lading environment motions, wheel/rail forces, and foundation stresses.

Model Description

The FRATE 11 model is the earliest version of the three FRATE models (FRATE 17 and FRATE/MITRE being the other two). As such, the representation is simpler than either of the other two. The model is depicted in Figure 4-19. Note that the representation of the trucks and carbody is the same as for FRATE/MITRE. FRATE 11, however, makes no provision for lading elements not integral with the carbody. The 11 basic degrees of freedom are:

Lateral displacement of each truck	2
Vertical displacement of each truck	2
Roll of each truck	2
Lateral carbody displacement	1
Vertical carbody displacement	1
Carbody roll	1
Carbody gain	1
Carbody pitch	1
TOTAL	11

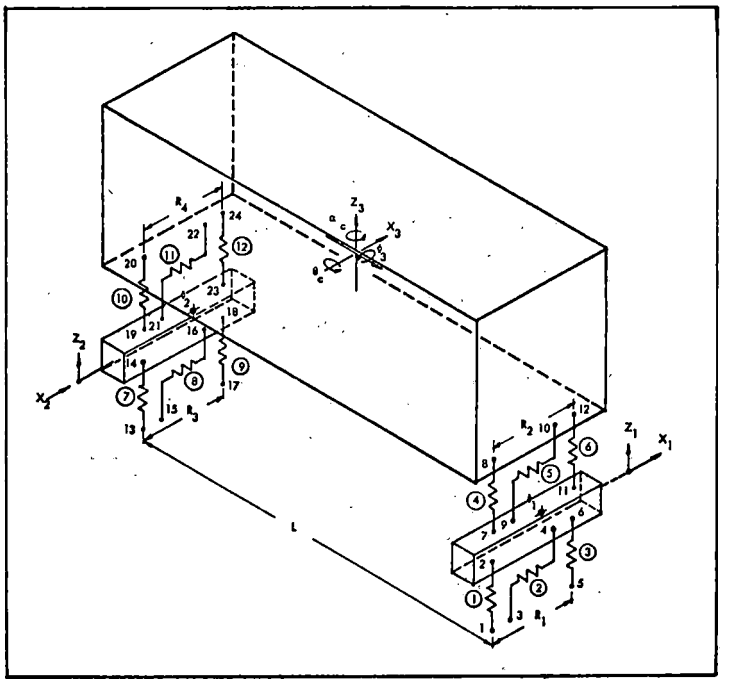


Figure 4-19. FRATE 11 Model

In addition, one degree of freedom is added for each normal mode of vehicle flexibility included in the carbody representation. Mass, damping, and stiffness characteristics of the track are included by lumping them with corresponding elements representing the truck.

The model accepts vertical excitation of the trucks in the form of sinusoids or optionally tabulated functions of time (or distance for a given forward speed).

The equations of motions for this system are developed using Newtonian methods. The formulation can be thus easily modified, if necessary, to account for peculiarities of particular Type II trucks.

Program Implementation

The program implementation is in FORTRAN and has been successfully run on both the CDC Cybernet system and an in-house Interdata computer. The set of second order differential equations of motion describing the 11 dof system plus normal carbody modes are recast as a set of first order differential equations and integrated in time using a standard Runge-Kutta package to obtain the time domain solution. Input for the program consists of physical parameters including stiffness and damping nonlinearities, truck and car geometry, the carbody modal shapes, excitation description, and integration control variables. Printed program output consists of the forces, accelerations, velocities, and displacements versus time, along with the current excitation. The utility of the program could be improved by better output formatting. Plotted results on the CDC system are also possible.

Documentation exists in the form of a User's Manual (17) and supporting theoretical material. An ASME paper (18) also provides information on the development and validation of the program. The User's Manual contains sample input and output.

Assessment

Like the other FRATE models, FRATE 11 is principally oriented toward analysis of roll and vertical dynamic phenomena including ride quality. Its advantages over FRATE/MITRE are less input required since fewer elements are included in the representation, and the Contractor's familiarity with this program. Otherwise, FRATE/MITRE provides the same capabilities as FRATE 11 with improved output readability.

The FRATE 11 model has been validated against test data obtained from actual vehicles operating on shimmed track and also against test data collected at the Rail Dynamics Laboratory. The two validation exercises involved two different types of railcars. Model parameters can, in general, readily be related to measurable physical quantities. Refinements in choice of damping coefficients were made to obtain the best comparison of model and test results. Machine precision is not a cause for concern with this model.

The verifiability of the model has been demonstrated by the validation exercises which have been carried out. (19) No special instrumentation was needed in the tests for the validation exercise. Some of the test data used for validation were collected before the model was developed.

The utility of the model in the TDOP Phase II analytical task stems from its prior use by the Contractor, its prior validation, and its relatively efficient operation. The User's Manual estimates costs of $.50 to $1.00 per second of simulated time for typical applications.

Conclusions

The FRATE 11 Model is a nonlinear time domain simulation model. It uses 11 degrees of freedom in the basic car and truck representation. Additional degrees of freedom are used for carbody flexibility in terms of normal modes. The model is the antecedent of both FRATE/MITRE and FRATE 17. FRATE 11 can be applied to the analysis of harmonic roll and bounce and also ride quality of the lading environment. The model has been validated against actual test data. The FRATE 11 model developed by the Contractor has been used extensively by the company's TDOP Phase II analysts.

(17) Healy, M.J., "A Nonlinear Computer Program for the Analysis of Dynamic Motions of a Flexible Rail Vehicle," Report No. WR-76-001-C, Wyle Laboratories, Colorado Springs Facility, March 1976.

(18) Healy, M.J., "A Computer Method for Calculating Dynamic Responses of Nonlinear Flexible Rail Vehicles," ASME Paper 76-RT-5, April 1976.

(19) Ibid.

4.14 FRATE 17 (WYLE) - HEALY

Introduction

FRATE 17 is a nonlinear time domain model using 17 degrees of freedom for the trucks and body. Additional degrees of freedom for carbody flexibility may also be included using a normal mode representation. The model focuses on vertical and roll dynamics. The extra degrees of freedom in this model compared to FRATE 11 are used to further define truck dynamics. Nonlinearities treated include Coulomb damping, spring clearances, stops, etc.

Model Application Areas

FRATE 17 is intended for use in studying vertical and roll car/truck motions. Its primary use is for nonlinear analysis of harmonic roll and bounce with the increased detail of the model allowing greater insight into force levels at the truck.

Model Description

FRATE 17 was developed directly from FRATE 11. The additional six basic degrees of freedom are due to the inclusion of an additional mass for each truck which allows side frames and wheelsets to be treated separately from the bolster. The additional masses each have vertical, lateral, and roll freedom of motion (see Figure 4-20).

As with the other FRATE models, carbody flexibility is represented by the addition of degrees of freedom for the number of carbody normal modes included. Mass, damping, and stiffness characteristics of the track are included by lumping them with the corresponding elements representing the wheelsets and side frames.

The treatment of excitation is similar to FRATE 11 also. Track input may be in the form of sinusoids or precalculated displacement versus location (time) functions. The formulation is thus easily modified, if necessary, to account for peculiarities of particular Type II trucks.

Program Implementation

FRATE 17 is coded in FORTRAN and is operational on the CDC Cybernet system. Standard Runge-Kutta integration similar to FRATE 11 is used to solve the differential equations of the system in the time domain.

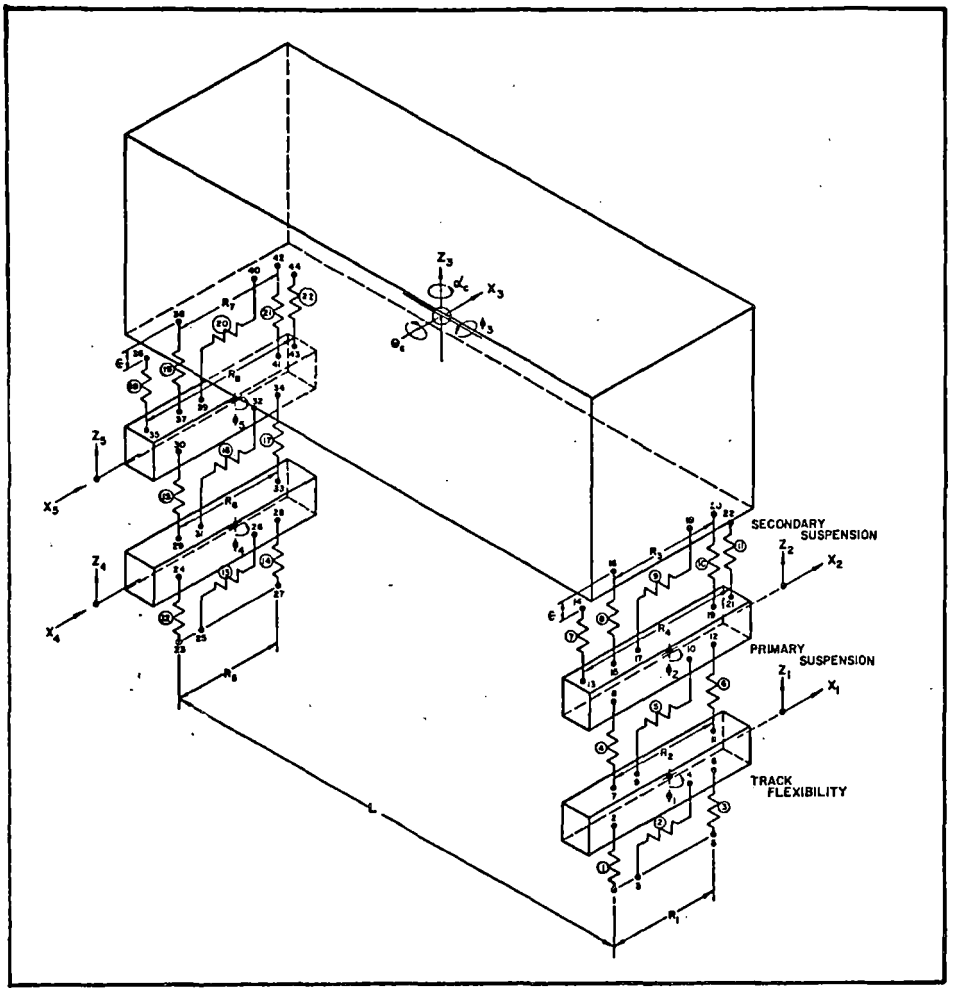


Figure 4-20. FRATE 17 Model

Input for the program consists of physical parameters, including the stiffness and damping nonlinearities, truck and car geometry, the carbody mode shapes, excitation description, and integration control variables. Printed program output consists of forces, accelerations, velocities, and displacements versus time, along with the current excitation. The utility of the program would be improved with a revised output format for printed results. Plotted output is possible. Documentation exists in the form of a User's Manual, (20) and a Theoretical Manual (21). An ASME paper (22) compares the 11 dof and 17 dof models.

Assessment

Like the other FRATE models, FRATE 17 is principally oriented toward analyzing roll and vertical dynamics. The model can be used in that regard to characterize the lading environment ride quality. Its unique advantage over FRATE/MITRE and FRATE 11 is the level of detail in the truck model allowing a more comprehensive analysis of forces and motions associated with the truck, e.g., center plate/bolster/side bearing interactions.

FRATE 17 has been validated in the same manner as FRATE 11. (23) It was found to produce equally good comparisons with test data for roll motions as FRATE 11. The documentation suggests that FRATE 11 is, therefore, appropriate for studies primarily interested in carbody motions for relatively heavy carbodies where the carbody inertia effects are more significant than truck phenomenon, and FRATE 17 is applicable if detailed truck motions and forces are desired or if light carbodies are simulated.

The verifiability of the model has been demonstrated by the validation exercises which have been carried out. No special instrumentation is required to obtain results compatible with the model.

The utility of the model in the TDOP analytical task stems from its prior use by the Contractor, its prior validation, and its relatively efficient operation. The documentation indicates costs of less than $1.00 per second of simulated time for typical applications.

(20) Healy, March 1976, op. cit.

(21) Healy, M.J., "Assumptions in the Dynamic Response Computer Program," Wyle Report WR-76-002-C, April 1976.

(22) Healy, April 1976, ASME Paper, 76 RT 5, op. cit.

(23) Ibid.

Conclusions

FRATE 17 is a nonlinear time domain simulation model. It uses 17 degrees of freedom to model trucks and carbody excluding the degrees of freedom used to represent the flexibility of the carbody in terms of normal modes. The model is an extension of FRATE 11. The six additional degrees of freedom in the basic representation give a more detailed breakdown of the truck. FRATE 17 is suitable for analysis of harmonic roll and bounce and also ride quality of the lading environment with particular emphasis on the effects of the carbody/truck interface. The program has been validated against actual test data. It has been used extensively by the Contractor.

4.15 WHRAIL, SYMMETRIC WHEEL/RAIL GEOMETRIC CONSTRAINT ROUTINE - LAW & COOPERRIDER

Introduction

The Symmetric Wheel/Rail Geometric Constraint Routine is the first of two routines developed by Cooperrider, Law, et al. to obtain the geometric relationships between wheelset and rails for a given combination of wheel profile, rail profile, wheel gage, rail gage, and cant angles versus the lateral displacement of the wheelset with respect to the rails. This first routine assumes symmetry between left and right wheel profiles and left and right rail profiles. A separate assessment report deals with a subsequently developed procedure, the Asymmetric Wheel/Rail Geometric Constraint Routine, in which no left/right wheel and rail symmetry assumptions are made.

The routine is not a dynamic model but rather an auxiliary program which precalculates such wheel/rail constraint relationships as effective conicity, gravitational stiffness, difference in rolling radii, and wheelset roll angle for all practical values of lateral wheelset displacement. The tabulation of these constraint functions serves as input to a number of dynamic models requiring this information.

Model Application Areas

The Wheel/Rail Geometric Constraint Routines are presently capable of generating data for three of the 17 models selected for detailed assessment in TDOP Phase II. These are the Contractor's lateral stability model, HUNTCT, and Law and Cooperrider's own 9 and 17 dof curving models. With modifications, there is no doubt that other models could make use of these routines, especially the curving and lateral stability models.

Model Description

Figures 4-21 and 4-22 show the basic configuration treated by both the Symmetric and Asymmetric Wheel/-Rail Geometric Constraint Routines. Note, however, that in the symmetric case, left and right cant angles as well as wheel and rail profiles are identical. The wheelset and rails are assummed to be rigid. The wheel and rail profiles are each defined by a series of 15 fourth order polynomial curve segments.

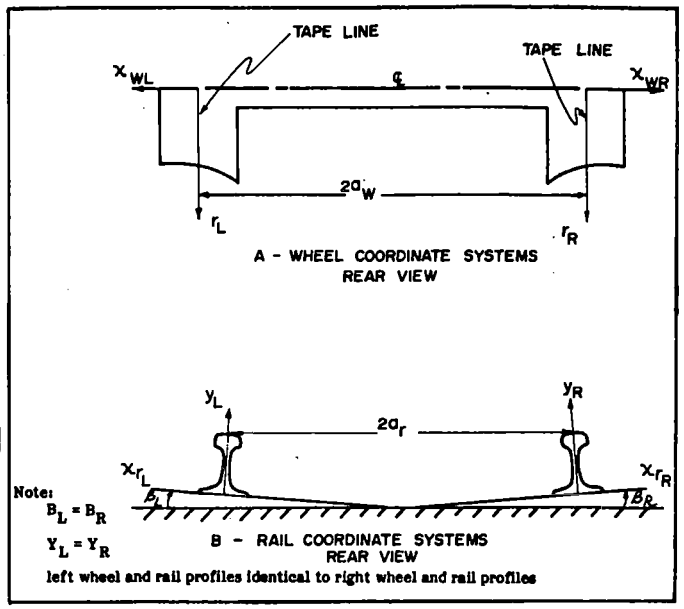


Figure 4-21. WHRAIL Routine (Symmetric)

Program Implementation

The routine has been coded in FORTRAN. A more efficient version of the program has been developed by the Contractor and used extensively in conjunction with HUNTCT. It can be used on either the CDC Cybernet system or the in-house Interdata computer.

The solution first involves defining the polynomial curve segments from the point-by-point profile definition. Then, for a given wheelset lateral displacement, a trial value of roll angle is chosen. Based on the tentative roll angle, the profile geometries are used to determine contact points, resulting in a new roll angle. The two values of roll angles are checked and, if not sufficiently in agreement, another iteration is performed. When agreement is reached, the roll angle, contact angles, rolling radii, vertical height of contact points, etc., are saved. The procedure continues for the values of lateral wheelset displacement required.

Program input consists of data defining the rail and wheel profiles, the cant angle, and wheel and rail gage.

Output consists of a recapitulation of input and specification of the coefficients of the polynomials used for each fitted segment and each of the following versus lateral wheelset displacement:

- Contact location on wheels
- Contact location on rails
- Rolling radii
- Height of point of contact on rails
- Contact angles
- Roll angle of wheelset
- Vertical displacement of wheelset
- Radii of curvature

The development of the routine is well documented. The documentation (24) contains sections on the theory behind the analytical approach to the experimental validation of the routine and user's guidelines for the routine. Sample input and output are included.

Assessment

The Symmetric Wheel/Rail Constraint Routine provides an efficient means of treating the details of wheel/rail geometrical relationships which are of fundamental importance in the formulation of curving and lateral stability models. The geometrical constraint data produced by this routine can readily be adapted to the needs of other dynamic models. The routine is useful for the clarity of its documentation and simplicity.

The Contractor has become familiar with the use of this routine in conjunction with the lateral stability model, HUNTCT.

The techniques used in the routine have been successfully validated against laboratory wheel/rail mock-ups. The representation is easily related to physically measurable quantities and the solution is not subject to computational imprecision.

The verifiability of the routine by means of the static laboratory mock-ups has been demonstrated. Verifying the routine against in situ measurements would require sophisticated profile measuring devices.

The clear and ample documentation makes this routine quite easy to use. Since the routine need be run only once for a particular combination of wheelset and rails to provide the data for many runs of the dynamic model, it is particularly inexpensive to use.

Conclusions

The Symmetric Wheel/Rail Constraint Routine is an auxiliary routine to support dynamic curving and lateral stability models. From data defining the profile of a wheel and a rail, the routine calculates rolling radii, contact angles, wheelset roll angle, and other important geometrical information important to the dynamic models. This routine assumes symmetry between left and right rails and wheels. The routine has been validated successfully against static laboratory mock-ups of rails and wheelset. It is an efficient, well documented, and useful analytical tool.

4.16 WHRAILA, ASYMMETRIC WHEEL/RAIL GEOMETRIC CONSTRAINT ROUTINE - LAW & COOPERRIDER

Introduction

The Asymmetric Wheel/Rail Geometric Constraint Routine is the second of two routines developed to obtain the geometric relationships between wheelset and rails for a given combination of wheel profile, rail profile, wheel gage, rail gage and cant angles versus the lateral displacement of the wheelset with respect to the rails. This second routine assumes no symmetry between left and right wheel profiles and left and right rail profiles. A separate assessment report deals with an earlier developed procedure, the Symmetric Wheel/Rail Geometric Constant Routine, in which left/right wheel and rail symmetry assumptions are made.

The routine is not a dynamic model but rather an auxiliary program which precalculates such wheel/rail constraint relationships as effective conicity, gravitational stiffness, difference in rolling radii, and wheelset roll angle for all practical values of lateral wheelset displacement. The tabulation of these constraint functions serves as input to a number of dynamic models requiring this information.

Model Application Areas

The Wheel/Rail Geometric Constraint Routines are presently capable of generating data for three of the 17 models selected for detailed assessment in TDOP Phase II. These are the Contractor's lateral stability model, HUNTCT, and Law and Cooperrider's own 9 and 17 dof curving models. With modifications there is no doubt that other models could make use of these routines, especially the curving and lateral stability programs.

Model Description

Figures 4-21 and 4-22 show the basic configuration treated by both the Symmetric and Asymmetric Wheel/-Rail Geometric Constraint Routines. Note, however, that in the symmetric case, left and right cant angles as well as wheel and rail profiles are identical. The wheelset and rails are assummed to be rigid. The individual wheel and rail profiles are each defined by a series of 15 fourth order polynomial curve segments.

Program Implementation

The routine has been coded in FORTRAN. A more efficient version of the program has been developed by the Contractor and used extensively in conjunction with HUNTCT. It can be used on either the CDC Cybernet system or the in-house Interdata computer. The asymmetry results in a more elaborate solution technique but is, in general, similar to the simple symmetric case.

The solution first involves defining the polynomial curve segments from the point-by-point profile definition. Then, for a given wheelset lateral displacement, a trial value of roll angle is chosen. Based on the tentative roll angle, the profile geometries are used to determine contact points using a Fibonacci search technique, resulting in a new roll angle. The two values of roll angles are checked and, if not sufficiently in agreement, another iteration is performed. When agreement is reached, the roll angle, contact angles, rolling radii, vertical height of contact points, etc., are saved. The procedure continues for the values of lateral wheelset displacement required.

Program input consists of data defining the rail and wheel profiles, the cant angle, and wheel and rail gages.

Output consists of a recapitulation of input and specification of the coefficients of the polynomials used for each fitted segment and each of the following versus lateral wheelset displacement:

- Contact location on wheels
- Contact location on rails

(24) Cooperrider, N.K., Law, E.H., et al., "Analytical and Experimental Determination of Nonlinear Wheel/Rail Geometric Constraints," Report No. FRA-OR&D-76-244, December, 1975.

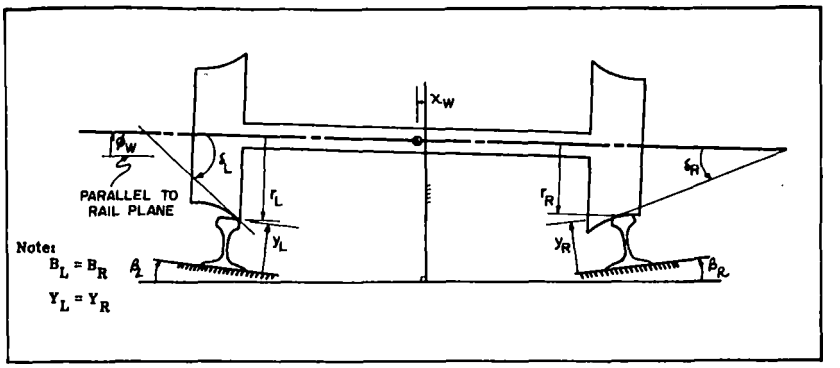


Figure 4-22. WHRAILA Routine (Asymmetric)

- Rolling radii
- Height of point of contact on rails
- Contact angles
- Roll angle of wheelset
- Vertical displacement of wheelset
- Radii of curvature

The development of the routine is well documented. The documentation (25) contains sections on the theory behind the analytical approach and user's guidelines for the routine. Sample input and output are included.

Assessment

The Asymmetric Wheel/Rail Constraint Routine provides an efficient means of treating the details of wheelset/rail geometrical relationships which are of fundamental importance in the formulation of curving and lateral stability models. The geometrical constraint data produced by this routine can readily be adapted to the needs of other dynamic models. This asymmetric routine complements the earlier symmetric routine.

The Contractor has become familiar with the use of this routine in conjunction with the lateral stability model, HUNTCT.

The documentation does not give evidence of an experimental validation effort similar to that given for the symmetric wheel/rail constraint routine. The representation is easily related to physically measurable quantities and the solution is not subject to computational inprecision.

The verifiability of the routine by means of the static laboratory mock-ups is considered demonstrated on the basis of the validation exercise undertaken for the symmetric case. Verifying the routine against in situ measurements would require sophisticated profile measuring devices.

(25) Cooperrider, N.K. and Heller, R., "User's Manual for Asymmetric Wheel/Rail Contact Characterization," Report No. FRA/ORD-78/05, December 1977.

The documentation makes this routine quite easy to use. Since the routine need be run only once for a particular combination of wheelset and rails to provide the data for many runs of the dynamic model, it is particularly inexpensive to use.

Conclusions

The Asymmetric Wheel/Rail Constraint Routine is an auxiliary routine to support the dynamic curving and lateral stability models. From data defining the profile of individual wheels and rails, the routine calculates rolling radii, contact angles, wheelset roll angle, and other important geometrical information important to the dynamic models. This routine assumes no symmetry between left and right rails and wheels. It is an efficient, well documented, and useful analytical tool that complements the earlier routine which assumes left/right wheel and rail symmetry.

SECTION 5 - CANDIDATES FOR VALIDATION

Upon completion of the detailed assessments, 14 analytical tools were selected for validation and application (see Table 5-1). Areas of possible application within each performance regime for the 14 analytical tools and the two supporting subroutines are shown in Tables 5-2 through 5-5. Three tools (the Lateral/Vertical, Freight Car Curving, and the TDOP Phase I models) were not selected for validation because either the same capability was provided by another tool, or they were overly complex and too costly. The specific reasons for their elimination are described in subsections 4.2, 4.5, and 4.10, respectively.

During the assessment, a number of weaknesses with existing tools were encountered. For example, within the curve negotiation regime, no satisfactory tool was available to predict the wheel/rail lateral force during flange contact. These forces are one of the primary requirements for evaluating curving performance and estimating wear during curving. Several models are currently being developed which do account for the force between the wheel and rail during flange contact, but they are not yet available for assessment. These models may be utilized at a future date. Meanwhile, the current plan is to rely more heavily on test data from the TDOP Phase II test program in evaluating curving performance.

Within the lateral stability regime, the dynamics observed during hunting are not adequately modeled or predicted by existing simulation programs. For example, field test data taken during TDOP Phase I show the front and rear trucks hunting at speeds differing by as much as 20 mph in certain cases. In addition, the data show that trucks alternately burst into and out of hunting during high speed runs. Existing lateral stability simulations do not predict such behavior. It is thought that this behavior is due to asymmetries in the wheel profiles. Work is currently underway to allow the modeling of such responses. For example, the Asymmetric Wheel/-Rail Geometric Constraint Routine, used in conjunction with tools such as HUNTCT, can be used to incorporate the dynamics dependent on wheel asymmetries. Presently available lateral stability models, while not predicting this observed behavior, do allow trends and parameter sensitivity to be studied.

Within the harmonic roll and bounce subset of the trackability regime, more accurate modeling of the friction snubber mechanism is required. To fulfill this need, the FRA has sponsored a Friction Snubber Force Measurement System field test as part of TDOP Phase II effort. The results from this test will contribute to the development of more accurate modeling. The ride quality regime will also benefit from an improved friction snubber model.

The remainder of this section describes in brief the 14 analytical tools and the two subroutines; why they were selected; and how Wyle expects to apply them to the analysis objectives of TDOP Phase II.

Table 5-1. Candidates For Validation

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

Table 5-2. Lateral Stability Regime Tools

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

* Law & Cooperrider

Table 5-3. Curve Negotiation Regime Tools

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

Table 5-4. Trackability Regime Tools

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

Table 5-5. Ride Quality Regime Tools

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

5.1 DYNALIST II (TSC)

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

DYNALIST was selected for TDOP Phase II validation because of its flexibility, the extent of its prior use, its excellent graphics capability, and good documentation. It is particularly useful because of its capability of performing analysis in both the time and frequency domains. The disadvantage of DYNALIST is its restriction to linear analysis.

It is expected that DYNALIST will be applied in the lateral stability, trackability, and ride quality analysis regimes of TDOP. DYNALIST will be used to obtain qualitative results in these regimes. Due to its versatility, it is expected to be extremely useful and efficient in predicting the effects of minor changes in model configuration or comparisons of Type I and Type II trucks.

5.2 HUNTCT (WYLE)

HUNTCT is a nonlinear, time-domain model. The model has 21 degrees of freedom in the basic rigid body representation. Optionally, carbody flexibility may be included by adding an additional degree of freedom per carbody mode of flexure.

HUNTCT was selected for validation on the basis of its capability to perform detailed analysis of lateral stability phenomena, including the representation of significant nonlinearities in the truck model and the wheel/rail interface. Also, HUNTCT has the advantage of having been given some analytical and experimental scrutiny already. Having been developed at Wyle, it is well understood by the Contractor's TDOP analysts. It is easily modified to incorporate specific aspects of a Type II truck if necessary.

It is expected that HUNTCT will be used in the lateral stability regime. HUNTCT could also be used if required in curve entry and exit analysis to supplement steady state curve negotiation models.

5.3 FRATE (MITRE, WYLE)

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

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

5.4 FREIGHT CAR HUNTING (AAR)

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

The Freight Car Hunting Model was selected for TDOP validation on the basis of the insight it can provide in investigating lateral stability. In particular, it provides somewhat greater detail in the representation of Type I trucks. If necessary, the program could be modified to account for individual differences of Type II trucks. The program is sufficiently well documented that it can readily be used. Another advantage of the Freight Car Hunting Model is the efficiency of the frequency domain analysis which it uses. No previous validation work is cited by AAR.

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

5.5 17 DOF EIGENVALUE (LAW & COOPERRIDER)

The Law and Cooperrider 17 dof Eigenvalue Model is a linear representation developed for analyzing hunting behavior. The program provides natural frequencies and mode shapes for the configuration described by the 17 degrees of freedom. Although it is a linearized model, the level of detail is sufficient to allow investigation of the effects of many truck components.

The 17 dof Eigenvalue Model was selected for TDOP validation as a complementary program to the AAR Freight Car Hunting Model, which is also a linear frequency domain model. In particular, the Law and Cooperrider program is well suited for addressing investigations of Type II truck behavior and the effects of vehicle front/rear asymmetry.

It is expected that the results obtained with the 17 dof Eigenvalue Model will, like the AAR hunting model, identify trends and establish relationships which will then be examined in greater detail by nonlinear models.

5.6 9 & 17 DOF STEADY STATE CURVING (LAW & COOPERRIDER)

These two models are nonlinear representations of a freight car in steady curvings. The program iterates until convergence to an equilibrium solution is achieved. An extra eight degrees of freedom in the 17 dof Steady State Curving Model are used to account for lateral and yaw motions of individual wheelsets.

These two models were selected for their ability to relate truck and wheelset parameters to curving behavior. Both models make use of the Law and Cooperrider Wheel/Rail Constraint Routines which have been validated experimentally. These programs lack formal documentation but some background material on them has been received. The equilibrium solution techniques are expected to be more efficient than repeated time domain solutions.

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

5.7 FLEXIBLE BODY RAILROAD FREIGHT CAR (AAR)

The Flexible Body Railroad Freight Car Model is a nonlinear, time domain program which represents freight car vertical and roll dynamics. The model uses 20 degrees of freedom. The carbody is divided into two lumped masses allowing a first order representation of carbody bending and torsional flexibility.

The model was selected to complement the FRATE models and other linear models in the analysis of the trackability (harmonic roll and bounce) and ride quality areas. In particular, the model is unique in its representation of body twist. The model has been analytically verified previously and is well documented. The Flexible Body Railroad Freight Car Model is expected to produce detailed results relating carbody roll, bounce, and twist motions with trackability and ride quality performance indices.

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

These four models are intended to be complementary frequency domain models. In relation to other models selected for TDOP validation, these would be classified as simple, linear analytical tools. HALF, FULL, and FLEX deal strictly with vertical motions. LATERAL computes transmissibilities for the carbody suspension with respect to sinusoidal track alignment variations.

These models were selected to provide an efficient means of obtaining qualitative results in the ride quality area. The programs are fairly well documented and have been implemented on the Contractor's Interdata system.

Results from these simple, linear models will be scrutinized with respect to results from more sophisticated nonlinear models such as FRATE and the AAR Flexible Body Railroad Freight Car Model. Validation of HALF will receive a low priority because the representation of the track is overly elaborate considering the simplification of the vehicle portion of the model. Half was not altogether eliminated, however, because of its close relationship to the other three TSC models which are expected to be validated. It was felt that this suite of programs should be treated as a unit.

5.9 WHRAIL, AND WHRAILA, SYMMETRIC AND ASYMMETRIC WHEEL/RAIL GEOMETRIC CONSTRAINTS (LAW & COOPERRIDER)

These two routines have been selected for use in TDOP Phase II as the most sophisticated wheel/rail representation available. These two routines are to be used as auxiliary programs for such models as HUNTCT and the 9 and 17 dof curving models which require detailed simulation of wheel/rail geometrical relationships. It may also be possible to adapt other models with less sophisticated wheel/rail geometrical representations for one or both of these routines. The Lateral/Vertical Model would be one such candidate.

The Symmetric Wheel/Rail Geometric Constraint program will be used predominantly, except where significant left/right asymmetry in either wheelsets or rails is identified by test data. In such cases the asymmetric version will be used.

Both routines have been validated with laboratory "mock-ups." In TDOP Phase II, they will be implicitly validated when the models which use them are validated.

APPENDIX A - SIMPLE ENGINEERING MODELS

Section 5 points out that simple models will be used in the various performance regimes to complement the use of more sophisticated models being validated and applied. A set of such models which has thus far been identified as being useful is here documented. These models, however, are not necessarily a complete set. As the program continues, additional simple models may be required as new insights and behavior quirks are found in exercising and validating the sophisticated models, and in interpreting test data. The simple models include kinematic models, steady state models, and simple spring/mass models.

KINEMATIC MODELS

The simplest kinematic model is that of a free wheelset. Numerous references giving an explanation of the motion of a free wheelset may be found in the literature, the earliest being Klingel (1) and Carter (2). For such motion it has been established that the wavelength of the laterally oscillating wheelset is

$$S = 2\pi\sqrt{\frac{ar}{\lambda}}$$

and the frequency is

$$f = \frac{V}{2\pi}\sqrt{\frac{\lambda}{ar}}$$

Where:

- a - half-rail gauge
- r - nominal rolling radius
- λ - conicity or wheel taper ratio

These fundamental kinematic wavelength and frequency relationships provide a means of checking the more sophisticated lateral stability models especially at low speed where inertial effects are not significant.

The simple kinematic model can be extended to include the effects of the primary suspension stiffness, wheel/rail geometry, creep, and inertia in a 2 dof model in which coupled wheelset lateral and yaw displacements are represented (3). With this representation it is found that there is a critical speed above which any wheelset disturbance will grow with time. The critical speed is given by

$$V_c^2 = \frac{2\left[k_{py} d_1^2 \bar{F} + a^2 (k_{px} + k_g)\right]}{a^2 m_w + \bar{F} I_{wy}} \cdot \frac{\lambda}{ar}$$

Where:

- k_{px}, k_{py} - lateral and longitudinal primary stiffnesses per bearing respectively
- k_g - gravitational stiffness determined by the axle loading and wheel/rail geometry
- d_1 - distance from truck centerline to the axle bearing
- m_w - mass of a wheelset
- I_{wy} - yaw moment of inertia of a wheelset
- $\bar{F}$ - ratio of lateral to longitudinal creep forces

The kinematic wave length is unchanged, hence the frequency at the critical speed is

$$f_c = \frac{V_c}{2\pi}\sqrt{\frac{\lambda}{ar}}$$

A second 2 dof representation is also used by several researchers (4,5) to find the critical speed for a rigid two-axle truck. For that representation the secondary suspension is taken into account along with truck inertial effects. Assuming nearly equal lateral and longitudinal creep the critical speed for a symmetric rigid truck is given by Law and Cooperrider (6) as:

$$V_c^2 = \frac{\left[k_r + 2\ell^2 k_{g} + 2(\ell^2 + a^2)(k_{sx} + k_g)\right]}{I_{ty} + m_t(\ell^2 + a^2)} \cdot \frac{ra(\ell^2 + a^2)}{\frac{ra}{\lambda}(\ell^2 + a^2)}$$

Where:

- k_r - rotational centerplate stiffness per truck
- k_{sx} - lateral secondary stiffness per side frame
- ℓ - the truck wheelbase
- I_{ty} - total truck yaw moment of inertia
- m_t - total truck mass

The kinematic frequency for a rigid truck is

$$f = \frac{V}{2\pi}\sqrt{\frac{\lambda}{ar}\left(\frac{a^2}{\ell^2 + a^2}\right)}$$

(1) Klingel, "On the Performance of Railway Vehicles on a Straight Track," Organ Fortschr. Eisenb. Wes., Vol. 38, 1883, pp. 113-123.

(2) Carter, F.W., "On the Stability of Running of Locomotives," Proceedings of the Royal Society, Series A, Vol. 121, 1928, pp. 585-611.

(3) Wickens, A.H., "The Dynamic Stability of Railway Vehicle Wheelsets and Bogies Having Profiled Wheels," International Journal of Solids and Structures, Vol. I, No. 3, 1965, pp. 385-406.

(4) Cooperrider, N.K., "The Lateral Stability of Conventional Railway Passenger Trucks," Proceedings of the First International Conference on Vehicle Mechanics, Wayne State University, Detroit, Michigan, July 1968.

(5) Tuten, J.M., Law, E.H., and Cooperrider, N.K., "Lateral Stability of Freight Cars with Axles Having Different Wheel Profiles and Asymmetric Loading," ASME Paper No. 78-RT-3, April 1978.

(6) Ibid.

BIBLIOGRAPHY

Breese, R. G., ed., "Track-Train Dynamics Bibliography," Association of American Railroads, Chicago, Illinois, 1973.

Bronowicki, A. and Hasselman, T.K., "DYNALIST II, A Computer Progam for Stability and Dynamic Response Analysis of Rail Vehicle Systems," Report No. FRA-OR&D-75-22, July 1976.

Cappel, K.L., "TDOP Phase II Introductory Report," FRA Report No. FRA/ORD-78/53, November 1978.

Carter, F.W., "On the Stability of Running of Locomotives," Proceedings of the Royal Society, Series A, Vol. 121, 1928, pp. 585-611.

Cheung, T.H.W., Garg, V.K., and Martin, G.C., "User's Manual, Freight Car Hunting Model," AAR Document R-251, February 1977.

Cooperrider, N.K., "The Lateral Stability of Conventional Railway Passenger Trucks," Proceedings of the First International Conference on Vehicle Mechanics, Wayne State University, Detroit, Michigan, July 1968.

Cooperrider, N.K. and Heller, R., "User's Manual for Asymmetric Wheel/Rail Contact Characterization," Report No. FRA/ORD-78/05, December 1977.

Cooperrider, N.K. and Law, E.H., "A Survey of Railway Vehicle Dynamics Research," Journal of Dynamic Systems, Measurement and Control, Trans. ASME, Special Issue on Ground Transportation, June 1974.

Cooperrider, N.K., Law, E.H., et al. "Analytical and Experimental Determination of Nonlinear Wheel/Rail Geometric Constraints," Report No. FRA-OR&D-76-244, December 1975.

Garg, V.K. and Singh, S.P., "Dynamic Curve Negotiation Behavior of a Freight Car," Session 34, Paper E.3, Heavy Haul Railways Conference, Perth, Australia, September 1978.

Healy, M.J., "A Computer Method for Calculating Dynamic Responses of Nonlinear Flexible Rail Vehicles," ASME Paper 76-RT-5, April 1976.

Healy, M.J., "A Nonlinear Computer Program for the Analysis of Dynamic Motions of a Flexible Rail Vehicle," Report No. WR-76-001-C, Wyle Laboratories, Colorado Springs Facility, March 1976.

Healy, M.J., "Assumptions in the Dynamic Response Computer Program," Wyle Report WR-76-002-C, April 1976.

Jacobsen, L.S. and Ayre, R.S., Engineering Vibrations, McGraw-Hill Book Company, Inc., 1958.

Kachadourian, G., Sussman, N.E., and Anderes, J.R., FRATE Vol. I User's Manual; Vol. II Technical Manual, (prepublication copy) FRA/ORD-78/59, September 1978.

Kenney, J.T., "Steady State Vibrations of Beam on Elastic Foundation for Moving Load," Journal of Applied Mechanics, Trans. ASME, Vol. 21, No. 4, December 1954, pp. 359-364.

Klingel, "On the Performance of Railway Vehicles on a Straight Track," Organ Fortschr. Eisenb. Wes., Vol. 38, 1883, pp. 113-123.

Law, E.H., Hadden, J.A., and Cooperrider, N.K., "General Models for Lateral Stability Analyses of Railway Freight Vehicles," Report No. FRA-OR&D-77-36, June 1977.

Meisenholder, S.G., "Dynamic Interaction Aspects of Cable-Stayed Guideways for High Speed Ground Transportation," Journal of Dynamic Systems, Measurement, and Control, Trans. ASME, Special Issue on Ground Transportation, June 1974, pp. 180-192.

Perlman, A.B., "An Evaluation of Computer Programs for the Analysis, Prediction and Simulation of Rail Vehicle Dynamics," August 1974, Transportation Systems Center (TSC) Task Memorandum RR-515, Rail System Dynamics Program.

Perlman, A.B. and DiMasi, F.P., "Frequency Domain Computer Programs for Prediction and Analysis of Rail Vehicle Dynamics," NTIS Report No. FRA-OR&D-76-135.II, Vol. I & II, December 1975.

Southern Pacific Transportation Company, "Math Modeling Report, Vol. I, TDOP Phase I," Interim Report, July 1976.

Sussman, N.E., "Critique of Frequency Domain Model-Solution Techniques," Report No. FRA/ORD-78/12.IV, February 1978.

Tse, Y.H., and Martin, G.C., "Flexible Body Railroad Freight Car," Vol. I Technical Documentation, AAR Document R-190; Vol. II User's Manual, AAR Document R-200; Vol. III Programming Manual, AAR Document R-260.

Tse, Y.H. and Martin, G.C., "User's Manual, Lateral/Vertical Model," AAR Document R-237, February 1977.

Tuten, J.M., Law, E.H., and Cooperrider, N.K., "Lateral Stability of Freight Cars with Axles Having Different Wheel Profiles and Asymmetric Loading," ASME Paper No. 78-RT-3, April 1978.

Wickens, A.H., "The Dynamic Stability of Railway Vehicle Wheelsets and Bogies Having Profiled Wheels," International Journal of Solids and Structures, Vol. I, No. 3, 1965, pp. 385-406.

Willis, T. and Smith, K.R., "A Mathematical Simulation of the Curve Entry and Curve Negotiation Dynamics of Flexible Two Axle Railway Trucks, "ASME Paper No. 76-WA/RT-14.

Wyle Internal Memo, "Validation of the Wyle Hunting Program with TDOP Data," No. W-071778.4, M. Healy, July 1978, and "Comparisons, HUNTCT and Phase I Test Data," No. W-071079.3, M. Healy, July 1979.

Thus the effect of the truck wheelbase is to reduce the kinematic frequency of a truck below that of a simple wheelset. As in the case of a sprung wheelset the critical frequency for a rigid truck is governed by the kinematic frequency relationship:

$$f_c = \frac{V_c}{2\pi}\sqrt{\frac{\lambda}{ar}\left(\frac{a^2}{\ell^2+a^2}\right)}$$

TRUCK/WHEELSET MOTIONS

Certain of the more sophisticated models to be assessed in the TDOP Phase II effort include degrees of freedom for which simple models can be used to explain high frequency behavior.

For example, it has been found that a 1 dof spring/mass model can explain the high frequency lateral oscillations of the two wheelsets with respect to the side frames. The model consists of one lumped mass for the two side frames, one lumped mass for the two wheelsets, and a spring representing the lateral primary suspension. The natural frequency for such a system is given by

$$f = \frac{1}{2\pi}\sqrt{\frac{2(m_w + m_s)k_{px}}{m_w m_s}}$$

Where:

m_s - mass of side frame

The simple model is accurate provided the primary suspension stiffness k_{px} is large relative to the gravitational stiffness and the secondary lateral stiffness. Another 1 dof model explains high frequency wheelset/truck warp motions. The model consists of two relational inertias, one representing that of the wheelsets yawing in phase, the second representing the rotational inertia of the side frames and bolster as the truck warps. The two rotational inertias are connected by the primary suspension sitffnesses. For this system the natural frequency is

$$f = \sqrt{\frac{(k_w + 4k_{py}d_1^2)(2I_{wy} + 2m_s d_1^2 + I_{by})}{(2m_s d_1^2 + I_{by})(2I_{wy})}}$$

Where:

k_w - warp stiffness per truck (typically small compared to $4k_{py}d_1^2$)

I_{by} - yaw moment of inertia of the bolster

This model is accurate for large values of primary longitudinal stiffness k_{py}.

A third 1 dof model can be used to predict another wheel set/truck natural frequency in which the wheelsets move laterally 180 degrees out of phase and the truck warps in phase with one of the two wheelsets, 180 degrees out of phase with the other. For small motions the wheelsets can be considered to be rotating about the center of the truck. The 1 dof model is then one rotational inertia representing the wheelset "rotation," a rotational inertia for the truck, and the lumped stiffness of the primary suspension. The frequency is given by

$$f = \frac{1}{2\pi}\sqrt{\frac{2k_{px}\ell^2(m_w\ell^2 + I_{sy})}{(m_w\ell^2)(I_{sy})}}$$

Where:

I_s - yaw moment of inertia of the side frame

Again the above analysis applies when k_{px} is large.

CARBODY NATURAL FREQUENCIES

A great deal of insight can be obtained by comparing test data and model results with calculated natural frequencies for the fundamental carbody motions. These include:

- sway (pure lateral)
- bounce
- roll
- pitch
- yaw

Bounce, pitch, and yaw can generally be treated to a first order degree of accuracy as not being coupled to other motions. In the case of sway and roll, however, there is often significant coupling giving rise to the so called lower center roll and upper center roll motions. For those motions a 2 dof model with coupled carbody lateral and roll motions is required. The frequencies for the uncoupled motions are given below.

$$f_{sway} = \frac{1}{2\pi}\sqrt{\frac{4k_{sx}}{m_c}}$$

Where:

m_c - carbody mass

$$f_{bounce} = \frac{1}{2\pi}\sqrt{\frac{4k_{sz}}{m_c}}$$

Where:

k_{sz} - secondary suspension vertical stiffness per side frame

$$f_{roll} = \frac{1}{2\pi}\sqrt{\frac{4k_{sz}d_1^2}{I_{cr}}}$$

Where:

I_{cr} - carbody roll moment of inertia about the center of gravity

---

$$f_{pitch} = \frac{1}{2\pi}\sqrt{\frac{2k_{sz}(\ell_f^2 + \ell_r^2)}{I_{cp}}}$$

Where:

ℓ_f - distance from front truck center to the carbody center of gravity

ℓ_r - distance from rear truck center to the carbody center of gravity

I_{cp} - carbody pitch moment of inertia about the center of gravity

---

$$f_{yaw} = \frac{1}{2\pi}\sqrt{\frac{2k_{sx}(\ell_f^2 + \ell_r^2)}{I_{cy}}}$$

Where:

I_{cy} - carbody yaw moment of inertia about the center of gravity

For the coupled roll and lateral motions the lower and upper center roll natural frequencies are given by:

$$f_{l.c.\ roll} = \frac{1}{2\pi}\sqrt{A - \sqrt{A^2 - B}}$$

$$f_{u.c.\ roll} = \frac{1}{2\pi}\sqrt{A + \sqrt{A^2 - B}}$$

Where:

$$A = \frac{2k_{sx}(I_{cr} + m_c h^2) + 2m_c k_{sz} d_1^2}{m_c I_{cr}}$$

$$B = \frac{16k_{sx}k_{sz}d_1^2}{m_c I_{cr}}$$

h = distance from carbody center of gravity to center plate

Note that when h = 0 the lower and upper center roll frequencies will be identical to the uncoupled frequencies.

STEADY STATE VERTICAL DYNAMICS

A simple model can be used to represent the steady state dynamics of a load moving over a continuous beam supported by an elastic foundation. While the analysis is more complex than can be treated here, the solution is well documented by Kenney (7) and Meisenholder. (8) In the latter, nondimensional beam deflection and stress are plotted as a function of nondimensional speed with various other parameter variations including foundation damping and load distribution.

OTHER SIMPLE MODELS

Simple spring mass models are also likely to be used to obtain insight and check more sophisticated models in the steady state curving, trackability, and ride quality areas. These models are not sufficiently defined at the present time to document in this report.

(7) Kenney, J.T., "Steady State Vibrations of Beam on Elastic Foundation for Moving Load," Journal of Applied Mechanics, Trans. ASME, Vol. 21, No. 4, December 1954, pp. 359-364.

(8) Meisenholder, S.G., "Dynamic Interaction Aspects of Cable-Stayed Guideways for High Speed Ground Transportation," Journal of Dynamic Systems, Measurement, and Control, Trans. ASME, Special Issue on Ground Transportation, June 1974, pp. 180-192.