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# OVERVIEW OF FREIGHT SYSTEMS R&D

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## Overview of FRA's Freight Systems R&D

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### ABSTRACT

This paper presents an overview of the freight systems research and development activities in the Office of Research and Development within the Federal Railroad Administration. It describes the mission, goals and objectives of the Office of Freight Systems and briefly summarizes the research, development, test and evaluation activities of the past year. Also outlined is work currently in progress. The purpose of the paper is to inform the railroad technical community, the public, and other government organizations of the federally sponsored research conducted in connection with the FRA Improved Rail Freight Service Program.

### INTRODUCTION

The purpose of this paper is to convey to the railroad technical community, the public, and other government organizations an understanding of the organizational mission, objectives, and current activities of the Office of Freight Systems.

The Office of Freight Systems was established in May, 1975 as part of an overall reorganization of the Federal Railroad Administration (FRA) research and development effort. The realignment was instituted to provide for an organizational structure more in tune with the emerging awareness that the nation's rail system needs required Federal assistance if it was to continue to be economically viable and a significant contributor to an efficient, safe, well-balanced, and environmentally sound national transportation systems.

The Office of Freight Systems within the Office of Research and Development (see Figure 1) serves the rail community and the public through the conduct of research, development, test and evaluation (RDT&E) activities designed to stimulate the advancement of rail freight technologies. In this connection, the office mission statement reads, "to plan, implement, sponsor, and evaluate freight railroad research, development, and demonstration programs designed to improve freight systems and to serve as the principal point of contact for such programs." This mission statement is further expanded in

the missions assigned to the two divisions within the office.

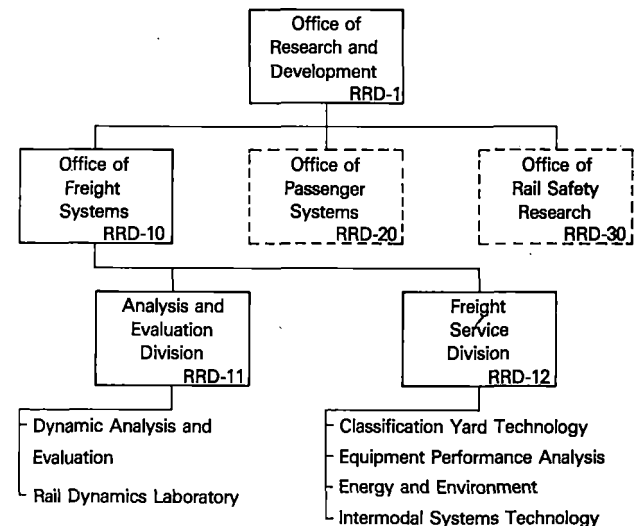


Figure 1  
Office of Freight Systems-Organization

The Analysis and Evaluation Division mission reads, "to plan, implement and sponsor the development of specialized research facilities and conduct analyses and evaluation programs pertaining to rail freight systems and subsystems". The Freight Service Division mission is "to conceive, plan, promote and implement research and development projects designed to function as a catalyst and to stimulate commercial exploitation

of system improvements for national rail freight service".

Within these assigned organizational missions, the Office of Freight Systems, in its administration of the Improved Rail Freight Service Program, strives for the implementation of research and development that provides outputs which contribute to two of the FRA goals, namely: (1) to promote and assist development of the railroad industry as an efficient, economically sound, and privately owned national railroad network that can attract that share of the market for intercity freight movement which is commensurate with its inherent economic advantages; (2) to otherwise facilitate rail transportation's contribution to the Nation's goals, including those relating to national security, social needs, energy conservation, and environmental protection.

In carrying out its mission, the Office participates in joint Government/industry cost sharing projects with the Association of American Railroads, the Railway Progress Institute, and individual railroads and suppliers; it also participates in international technology information exchanges in those areas for which it is responsible. In addition, it performs project management functions in connection with contracted research. To extend its technical capabilities the Office annually establishes an agreement to have certain portions of its research either conducted or monitored by personnel of the Transportation Systems Center (TSC) in Cambridge, Mass. or the Transportation Test Center (TTC) in Pueblo, CO. Inter-agency agreements are also established with other Federal agencies for the performance of research in support of the program.

A brief description of the activities within each division follows.

#### ANALYSIS AND EVALUATION DIVISION

The Analysis and Evaluation Division is comprised of two major subprogram areas; specifically, (1) Analysis & Evaluation and (2) Rail Dynamics Laboratory.

The goals of the Analysis and Evaluation subprogram are:

- Develop analytical models to describe the dynamic effects of (1) the wheel/rail interface, (2) variable truck characteristics on the performance of rail freight cars.
- Provide and operate a Facility for Accelerated Service Testing (FAST) to produce results aimed at reduction of procurement, operating, and maintenance costs of rolling stock and track.
- Evaluate available Maintenance of Way (MOW) equipment in a cooperative industry/FRA project to provide data for the more effective allocation of

MOW resources.

Accomplishments in this area in the past year have included the following.

- The initiation of a development project with Rutgers University, under contract DOT-FR-767-4323, to dynamically characterize three densities of lading in shippers comprised of three types of corrugated board. Also, to be investigated will be a single 3 ply and 5 ply bag. Lastly, a pallet that is comprised of an unconstrained single high density lading in a corrugated shipper will be characterized. Typical damage to be prevented is shown in Figure 2.
- The other significant effort in this subprogram is the "Freight Car Vehicle Dynamics" research project being conducted by Clemson and Arizona State University. A number of interim reports describing the progress of this program have been issued and are available through NTIS. They include:
  - (1) Analytical and Experiments Determination of Nonlinear Wheel/Rail Geometric Constraints, December, 1975, PB 252290
  - (2) General Models for Lateral Stability Analysis of Railway Freight Vehicles (to be published)
  - (3) An Investigation of Rail Car Model Validation (to be published)
  - (4) Linear Analysis Model for Railway Freight Cars (to be published)
  - (5) Nonlinear Analysis Models for Railway Freight Cars (to be published)
  - (6) Field Test and Validation of Railway Freight Car Models. (to be published)
  - (7) Users Manual for Asymmetric Wheel/Rail Contact Characterization Program (to be published)
  - (8) Users Manual for Kalkers Simplified Theory of Rolling Contact (to be published)

The current status of this project will be presented in the accompanying paper entitled "Research in Freight Car Dynamics."

- Within the Analysis and Evaluation program only the rolling stock portion of the FAST program is addressed; the track portion is covered in the Improved Track Structures program which is the responsibility of another office within OR&D. To date the track structure has accumulated over 100 MGT and the rolling stock has seen an average mileage of over 60,000 miles. The recent status of this project

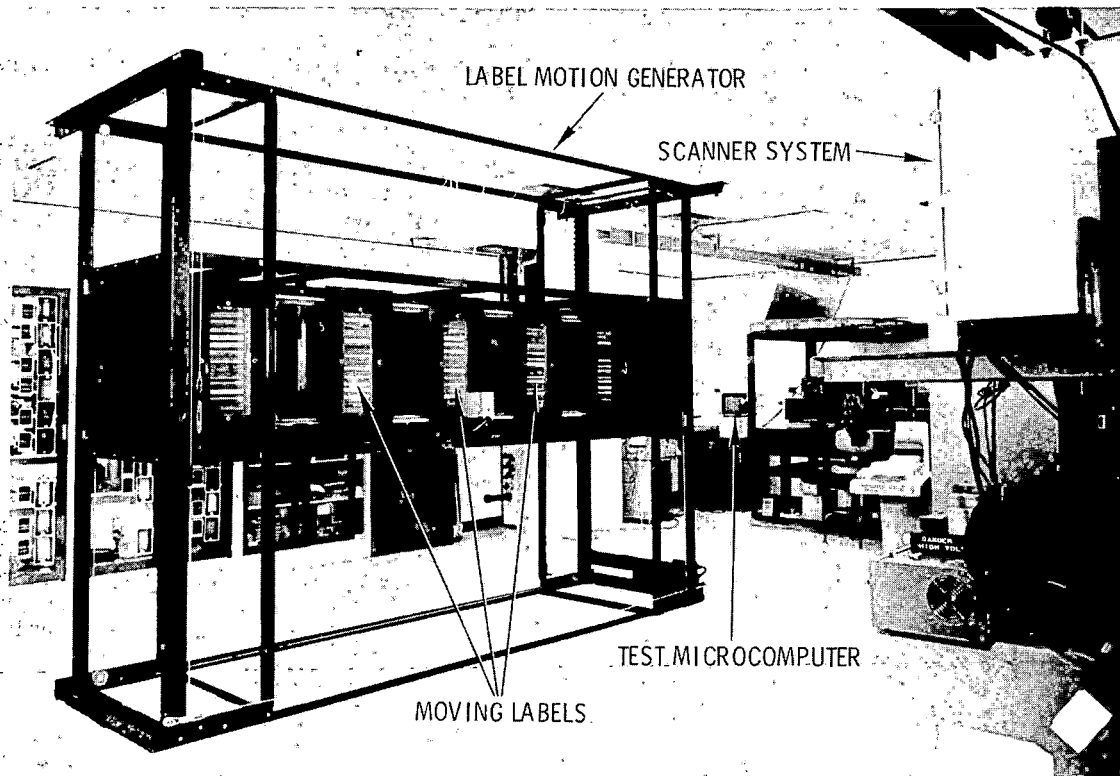


Figure 2  
Example of Lading Problem Being Investigated

will be presented in the accompanying paper entitled, "FAST Mechanical Equipment Test Results to Date Future Plans."

- At the time of this writing, a request for proposal has been issued. The proposals received are undergoing review to select a contractor who will:
  - (1) Develop a structure (e.g., indices of merit, equipment performance, etc.) for the purposes of evaluating and analyzing existing or prototype MOW equipment not currently used by American railroads.
  - (2) Develop a procedure whereby items selected in step (1) can be evaluated.
  - (3) Finally, develop a procedure that outlines the necessary planning steps, schedules, documents and representative agreements or contracts that will allow testing of MOW equipment in an operational railroad environment.

The goal of the Rail Dynamics Laboratory (RDL) subprogram is to:

- Provide a facility of perform tests of full scale railroad and transit vehicles under controlled laboratory conditions.

The RDL is planned to be operational at the beginning of 1978 and will be capable of investigating problems associated with:

- Suspension Characteristics
- Rock and Roll
- Component stress
- Component and vehicle natural frequencies
- Analytical model validation
- Adhesion
- Ride comfort
- Acceleration
- braking
- lading response.
- hunting

The current status of this program will be presented in the accompanying paper entitled "RDL Test Planning, Scheduling, and Budgeting."

FREIGHT SERVICE DIVISION

In the Freight Service Division there are four subprogram areas of RDT&E activity, namely:

(1) Classification Yard Technology, (2) Equipment Performance Analysis, (3) Energy/Environment, and (4) Intermodal Systems Technology. A brief description of each subprogram area, its recent activity, and current undertakings follows.

In the Classification Yard Technology area the major program objectives are to:

- Develop technologies that will substantially reduce car delays in yards,
- Quantify areas where yard improvements are feasible and desirable,
- Evaluate components and systems that will improve efficiency in yards; in cooperation with the railroads and suppliers, and
- Improve the effectiveness of railroad communication and control systems.

Accomplishments in this area in the past year have included the following:

- Technical reports relating to optical automatic car identification (ACI) were provided to the industry in May, 1976 and June, 1977. These reports provided the technical data needed to assess the current industry standard ACI system. The latest reports describe options to increase performance of the optical scanner system to 97% from the current level of 80% and indicate a potential label life of 17 years.
- A report entitled, "Railroad Classification Yard Technology; A Survey and Assessment" became available to the industry and public in January, 1977. The report identified technical areas that will provide the highest payoffs for upgraded or new yards. It also provided an inventory of the yard population in the U.S. and indicated that some 200 new or major reworked classification yard projects can be projected in the next 25 years. The report is available from the National Technical Information Service (NTIS) Springfield, VA 22161 under Accession Number PB 264 051.
- A "Research Plan for EMC [Electromagnetic Compatibility] Study of the Communication and Control Systems in a Railroad Classification Yard" was provided to interested parties in the industry in July 1977. The plan includes provisions for the participation of three railroads (Santa Fe, Southern, RF&P) in field tests and for coordination of the project through the Inductive Interference Committee, Communication and Signal Section, Association of American Railroads. This work is being performed for the FRA by the Department of Defense Electromagnetic

Compatibility Analysis Center, at Annapolis, Maryland.

Representative of RTD&E in this program area is the accompanying paper, "Optical ACI -- A New Look". The new and continuing work underway at this time includes:

- In addition to the yard EMC study previously mentioned, this work area has, at the request of industry, been expanded to include the EMC/EMI impact of railroad electrification. The research plan for this expanded effort is now in the formative stage. This research is expected to result in a report of the research findings in spring 1978. The report will contain a characterization of the yard's EM environment, aid in the understanding of existing equipment compatibility, and provide recommendations for improvements. In connection with electrification, the report will indicate a recommended testing methodology for determining the potential impact of electrification on C&S equipment and the surrounding environment.
- A feasibility study of alternatives in car speed control in classification yards has been initiated. This competitive procurement will produce a technical report identifying and recommending the most promising concepts for further development.
- An assessment of approaches to car presence detection in connection with yard information and control systems has been initiated. This work should lead to a better understanding of the detection requirements and the development of the optimum solution. A report of this work will be made available to industry.
- After concluding the laboratory verification (see Figure 3) of Optical ACI Scanner system improvements, three final technical reports will be made available to the industry. This research was carried out in cooperation with the Research and Test Department, AAR and the Rolling Stock Committee of the Railway Progress Institute.
- As a result of the previously mentioned yard survey and assessment, a contract has been awarded to the Stanford Research Institute to develop a yard design methodology. This multi-year, multi-phased project, begun in April 1977, will result in interim technical reports on each phase of development and ultimately in a design methodology handbook for use by the industry. This work is being coordinated with the Yard and Terminals Committee of the American Railway Engineering Association (AREA) which has established a special liaison subcommittee for this purpose.





Figure 3  
TSC laboratory setup for testing OACI scanner system improvements.

In general, all RDT&E activities associated with the electrical engineering discipline as it relates to rail freight service are handled in this subprogram area.

In the Equipment Performance Analysis area the major program objectives are to:

- Increase railroad profitability through the reduction of lading damage which can be attributed to dynamic phenomena associated with rolling stock suspension systems.
- Develop those technologies that will help to reduce costs occurring from the wear and maintenance of rail vehicle components.
- Quantify the information necessary to provide economic-based performance data and specifications upon which sound investment decisions may be reached.
- Improve train handling and make-up times where system components are the limiting factor.

Accomplishments in this area in the past year have included the following:

- Ten reports on test philosophy, measurements and economic analysis pertaining to Phase I of the Truck Design Optimization Project (TDOP) were made available

to industry. In addition, some 209 magnetic data tapes containing actual field test measurements were made available through NTIS.

- Two assessments of opportunities for technical innovation, one in braking and one in coupling, were completed. The reports of this work are being finalized and will be made available to the industry and public this Fall. The coupling work was coordinated with the AAR Advanced Coupler Concepts Program while the braking effort was coordinated with the AAR Brake Equipment Committee.
- Completion of the development of a friction snubber force measurement system. This is the subject of an accompanying paper, "Truck Performance -- Friction Snubber Force Measurement System" and is representative of our RDT&E efforts in this program area.

With respect to new and continuing work in the area of equipment performance the following is characteristic.

- The most significant activity will include the continuance of TDOP with the award of the Phase II contract to Wyle Laboratories. This phase of TDOP concerns itself primarily with the Type II, Special Purpose truck but will also encompass some Type I, General Purpose

truck testing as indicated below. As in Phase I, a railroad, in this case the Union Pacific, will provide, as a sub-contractor to Wyle, the necessary facilities for the conduct of field testing. Also, as was done in Phase I, industry consultants representing various viewpoints and areas of expertise will be employed, and close coordination of project developments will be effected between TDOP and the Track-Train Dynamics Program.

- In light of the recommendations stemming from the previously mentioned braking and coupling technology assessments, investigatory research in the areas of electro-pneumatic brakes, friction material versus wheel wear, automatic coupling concepts, load sensing devices, disc brakes, and wheel thermal capacity will be initiated. These studies will examine both technical and economic aspects. Reports will be made available to industry as they become available.
- Field testing of the Friction Snubber Force Measurement System (FSFMS) will be incorporated into Phase II of the TDOP. The Barber S2 and ASF Ride Control trucks will be tested loaded and unloaded, with new and worn friction shoes, on curved and tangent track using the newly instrumented side frames (see Figure 4). Findings from this work will be incorporated into the Phase II reports to be published during the coming year.

- Drawing upon our previous work done in connection with TOFC/COFC aerodynamic drag studies, additional testing in the area of aerodynamic drag on different freight car configurations will take place this year along with the validation of mathematical models developed from several wind tunnel tests. The full-scale validation of TOFC/COFC wind tunnel data is now underway at the Transportation Test Center. Equipment for these tests has been made available by the Trailer Train Company and Trailmobile Division of Pullman, Inc. Ensco Inc., supported by Brewer Engineering Laboratories, Inc. are the contractors for this work. A report on the second series of wind tunnel tests should be made available approximately one year from now.

In general, those subjects dealing with mechanics, dynamics and the mechanical engineering discipline as related to rail freight service reside in this subprogram area.

In the Energy and Environment area the major program objectives are to:

- Exploit and improve the inherent energy and environmental advantages of the rail mode.
- Assist railroad industry with practical guidelines and test procedures contributing to overall noise abatement and the reduction of noxious emissions.

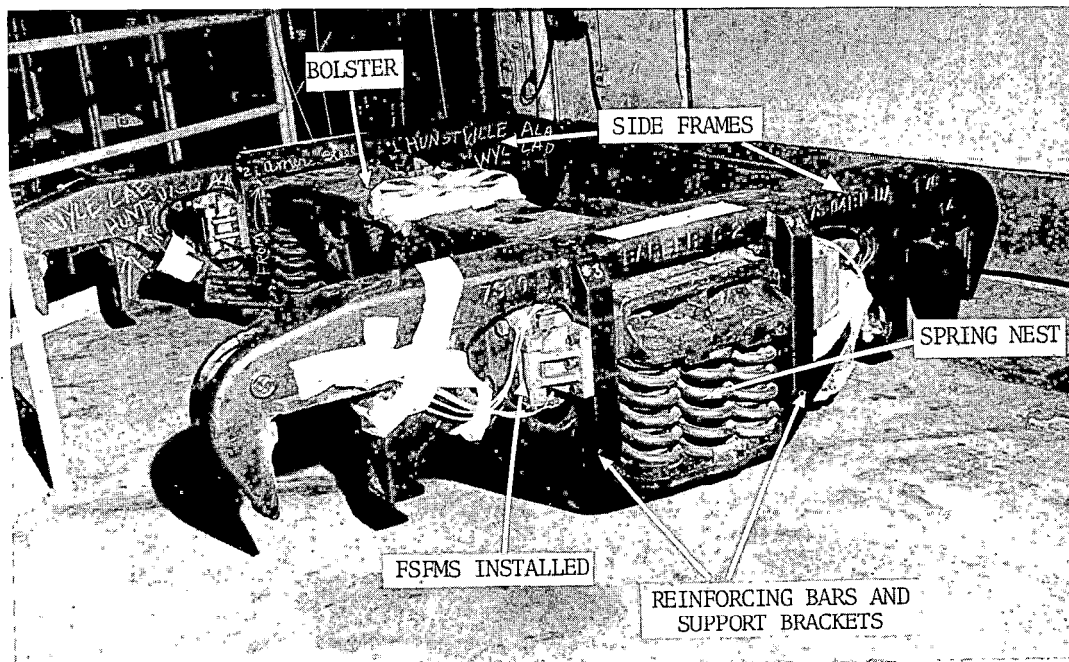


Figure 4  
Standard Barber S-2 truck equipped with new Friction Snubber Force Measurement System (FSFMS).

- Develop and demonstrate energy conservation techniques having economic as well as technical payoffs.
- Quantify areas where locomotive improvements will produce further efficiencies.

Accomplishments in the past year in this program area have included the following:

- An interim technical report, Fuel Efficiency Improvement in Rail Freight Transportation: Multiple Unit Throttle Control to Conserve Fuel, December 1976 (NTIS Accession No. PB 262 470) describing preliminary test results (see Figure 5) of the railroad developed "Fuel Saver" device was made available. A final report incorporating additional field test data will become available in November 1977. Representative of work accomplished in this program area and related to this subject is the accompanying paper, "Energy Conservation Multiple Unit Locomotive Throttle Control."

- In conjunction with the Intermodal Systems Technology subprogram a wind tunnel aerodynamic drag study was completed on TOFC/COFC equipment. The quantification of fuel consumption in various rail freight operations, including TOFC/COFC was also concluded. A technical report entitled "Aerodynamic Forces on Freight Trains, Volume I-Wind Tunnel Tests of Containers and Trailers on Flatcars," December 1976, was made available to the industry and public (NTIS Accession No. PB 264 304). An interim technical report entitled, "Railroads and the Environment-Estimation of Fuel Consumption in Rail Transportation, Volume II-Freight Service Measurements" September 1977, (Report No. FRA/ORD-75/74. II), is also being made available to the industry and public through NTIS at the present time.



Figure 5

FRA research engineer logging operational data during revenue operations in the evaluation of a multiple unit locomotive throttle control device.

- A preliminary assessment of the potential for the reduction of aerodynamic and mechanical train resistance including a determination of the economic factors involved. A technical report will be made available to the industry and public in December 1977. Recommendations for further research and suggested alternatives will be incorporated in the report.

New and continuing research underway at this time includes:

- Award of a contract to the Garrett Corporation to conduct a systems analysis and bench test feasibility study to determine the potential for the application of flywheel energy storage technology to a yard locomotive. The results of this analysis will become available in November 1978.

- Award of a contract, again to the Garrett Corporation, to conduct a feasibility study on the potential for applying wayside energy storage technology to line-haul locomotives engaged in grade operations; where there is potential for producing large amounts of recyclable energy through dynamic or regenerative braking.
- Establishment of a technical assistance agreement with Lawrence Berkely Laboratories, a Department of Energy National Laboratory administered by the University of California, to provide back-up support to the overall energy and environment program area by contributing expertise in E&E related matters.
- The research necessary to develop a locomotive data acquisition package (LDAP) has been initiated. This work is intended to produce a research tool that can withstand the harsh environment found in over-the-road locomotive operations while collecting scientific data that will provide an insight into the potential for improvements in locomotive performance under varying operating conditions.
- In connection with LDAP, work has been initiated to definitize, in a magnetic tape file format, three selected test route profiles representative of normal rail freight transport operations. This data base, augmented with LDAP gathered information, should begin to provide the necessary knowledge base to augment analytical analysis related to energy and economic optimization of propulsive power.

This program area deals with subjects requiring multi-disciplinary skills related to the energy and environmental aspects of rail freight service.

In the Intermodal Systems Technology area the major program objectives are to:

- Support the rail industry's efforts to increase its market share in freight transportation, especially in high revenue traffic.
- Promote realization of superior levels of service to be achieved through an optimized truck/rail/truck intermodal system.
- Assist railroads in attainment of the maximum level of efficiency for door-to-door movement of intercity freight through full exploitation of the rail mode's fundamental advantages.
- Support achievement of major improvements in safety for the public, yard personnel, train crews and goods in transit.

During the past year activities in this program area have included:

- As noted previously under the energy and environment program area, completion of a wind tunnel TOFC/COFC equipment aerodynamic drag characterization study and field measurements of the fuel consumption characteristics associated with dedicated TOFC/COFC trains.
- With the cooperation of the Atchison, Topeka and Santa Fe Railway, Pullman-Standard, Trailer Train, American Steel Foundries, and the National Castings Division of Midland-Ross, a functional test and mode shape characterization was completed on two existing lightweight intermodal cars, namely: the "skeleton car" one of which had been modified to carry trailers (see Figure 6) vis-a-vis its former container only configuration. Performance of these cars was compared to that of a conventional all-purpose TTX car which served as the base line. To determine the influence of wear on the parameters measured the cars have thus far been subjected to approximately 50,000 miles of revenue service and a re-run of the instrumented data collection tests conducted at the outset of the project. Further revenue service exposure will continue to 125,000 miles, at which time another instrumented test will be conducted. A technical report on this activity will become available in June 1978. An accompanying paper, "Intermodal Cars--New Developments," will briefly describe the test procedures, data acquisition methodology and developments in software associated with over-the road ride vibration testing. This paper is considered representative of the RDT&E in the intermodal system technology area.

New work underway includes:

- In August 1977, the award of two parallel contracts for the first phase of a comprehensive intermodal systems engineering study. The two Phase I contractors, namely: Peat, Marwick, Mitchell & Company and A. T. Kearney, Inc. will be expected to bring to bear slightly different philosophical approaches to the broadscope initial assessment study. This will lead to a Phase II effort to the selected contractor for performance of a more indepth analysis based on the selected options from the Phase I performance. Both contractors have assembled impressive teams to accomplish the task at hand. PMM&Co. has teamed with R. L. Banks & Associates, Battelle Columbus Laboratories, Boeing Computer Services, and Thomas K. Dyer, while A. T. Kearney has teamed with Herbert O. Whitten & Associates and the GM Transportation Systems Division. The contractors will interface with an ad hoc Intermodal Technical Committee which

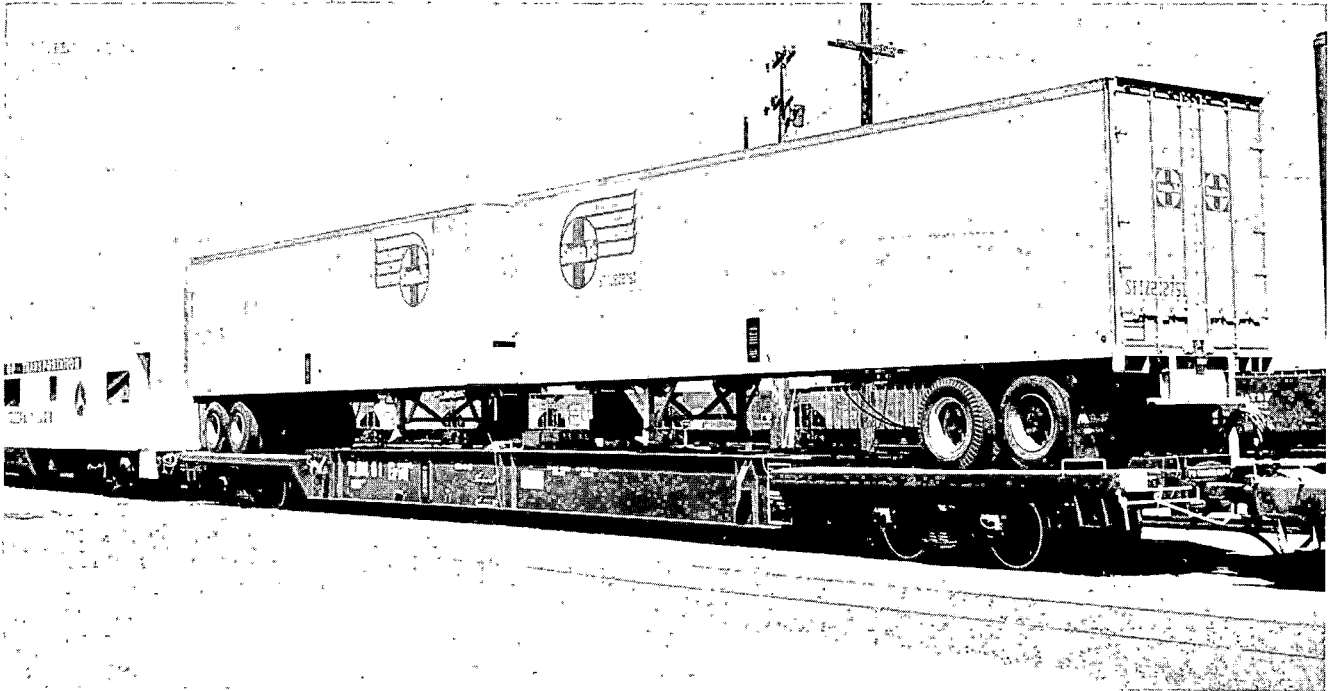


Figure 6  
Instrumented trailers and "skeleton car" along with FRA Data Acquisition Car, T-5,  
used in the Lightweight Flat Car Evaluation Project.

has been established by the AAR, Operating-Transportation General Committee for the purpose of providing recommendations pertaining to technical issues related to intermodal systems. Results of the six-month Phase I study are expected to become available in February 1978. The Phase II effort should be completed in May 1979 and result in a prospectus or Intermodal System Development Plan for consideration and use by the industry.

- o Initiation this month of an investigation into the feasibility of developing a lightweight, low-profile TOFC car to relieve the problem of clearance restrictions for this type of traffic, principally in the northeast. Should such a development prove feasible, the design, fabrication and test of three prototype cars will be undertaken.

The major area of interest in this program area concerns itself with industrial engineering as related to rail freight service.

#### SUMMARY

As can be seen from the foregoing the Office of Freight Systems manages a diverse RDT&E program whose principal purpose is to promote the continued viability of the Nation's railroads. Much of the program structure can be seen to be related to recent publications that attempt to redefinitize the research needs of the industry. For obvious reasons, like the availability of re-

sources, not everything that needs to be done can be done; hence, priorities must be established. In this regard, the Office of Freight Systems attempts to address principally near-term (in the normal R&D connotation) solutions to persistent problems, while at the same time keeping an eye to the future, especially where front-end development of the knowledge base is required for sound and timely commitment to innovation.

Recognizing its own limitations the Office of Freight Systems encourages and solicits critique of its program endeavors. In an effort to provide a forum through which this can occur, the Office seeks a tie-in with the appropriate segments of the rail transportation industry that can relate to its various projects. With this in mind, it is hoped that the examples given in this overview report reflect this philosophy.

#### ACKNOWLEDGMENTS

The authors particularly wish to thank the many (too numerous to mention by name) individuals in the industry that have displayed an interest in the development of the program. Their counsel and advice, cooperation and assistance, support and confidence are greatly appreciated.

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# Research in Freight Car Dynamics

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## ABSTRACT

This paper describes the Freight Car Dynamics project conducted by Clemson and Arizona State Universities under sponsorship of the Federal Railroad Administration. A series of models and associated solution techniques for predicting freight car stability, forced response and curving behavior were developed in this effort. Validation tests were planned and carried out by the Association of American Railroads and the Union Pacific Railroad. The test data have been analyzed by several techniques including spectral analysis and the random decrement method. The validation process, involving comparison of predicted and experimental transient response, power spectral densities and steady state values, is in progress.

## INTRODUCTION

This research project, under the direction of Professors E. Harry Law of Clemson University and Neil K. Cooperrider of Arizona State University, has the objective of providing tools and techniques to analyze the dynamic behavior of railroad freight cars. The effort entails development and correlation of theoretical techniques for predicting freight car dynamic behavior, and use of the techniques to investigate the behavior of present and proposed freight car designs. The project is sponsored by the Federal Railroad Administration with support and cooperation from the Association of American Railroads and the Union Pacific Railroad.

Derailments, damaged freight, distorted track and worn or broken vehicle components are problems that result from undesirable freight car dynamic behavior. The models and analytical techniques developed in this effort can be used to determine causes of present dynamic problems, and to design components and vehicles that alleviate such problems. To study the full range of railroad operating conditions pertinent to lateral vehicle dynamics, models and analyses have been developed for rail freight vehicle stability, forced response and curving.

Until recently, experimental and analytical

evaluations of rail car dynamics have gone down separate paths. This lack of interaction can be attributed to the fact that the experimental and analytical work has been done by different groups. Not surprisingly, those organizations, such as the British Rail and Japanese National Railway research groups, that first undertook the rail vehicle dynamics analyses, have also been the first to use integrated analytical-experimental evaluation methods for rail vehicle design. This project has built on this background to develop an approach for dealing with the dynamic problems of freight vehicles of the type in use in North America. In the course of this work several innovations have been introduced into rail vehicle dynamics analysis and testing including development of numerical techniques to deal with arbitrary wheel and rail head profiles, the use of quasi-linearization techniques to handle nonlinear characteristics, the use of an hydraulic excitation system during the vehicle tests, and the application of the random decrement technique in analyzing the test data.

Research of this type involves modeling, correlation or validation of models, and analysis of the model behavior. In this program, quite

a number of models have been developed for freight car behavior on tangent track, during curve entry, and in curve negotiation. Because this is an exploratory study into the theory of freight car behavior, these models differ widely in complexity. One of the outputs of the project will be recommendations for the appropriate use of each model and analysis approach.

Validation of these models is underway. Experimental data for the validation effort has been provided by tests carried out by the Association of American Railroads and the Union Pacific Railroad. After validation, the models will be utilized to examine current vehicle and track maintenance procedures and to suggest amendments to the procedures. The models will be supplied to the railroad industry for use in evaluating supplemental devices, in studying possible modifications for current freight car trucks, and in exploring new design concepts.

This paper is intended as an overview of this effort. The theoretical developments achieved in this effort are discussed in the next section, followed by descriptions of the validation techniques and field tests. The current status and future work are summarized in the final section.

## THEORETICAL DEVELOPMENTS

### Approach

Our approach in this project has been to develop and to investigate the conditions of applicability of a number of modeling and analysis approaches. For example, computer solutions for the stability behavior of rail vehicles that use linearized models and eigenvalue/eigenvector solution techniques are several orders of magnitude less expensive than solutions obtained by direct numerical integration. Consequently, eigenvalue/eigenvector stability analyses offer considerable promise for use in the design process where a large number of candidate designs are to be evaluated.

After analysis of the test results, we will compare the results from the various models and analyses with test results to evaluate the conditions and range of applicability of each model and analysis approach. For many purposes, analyses may be used that are inexpensive computationally as compared with other analyses. It is important to identify the purposes for which such analyses can be used if these models and analyses are to be of maximum benefit to the railroad industry.

In each of the three areas pertinent to lateral rail vehicle dynamics (hunting or lateral stability, forced response, and curving behavior), we have followed this approach. For investigating hunting stability we have developed six models (classified in terms of the numbers of degrees of freedom) and three analyses, or techniques to solve the model equations of motion,

for forced response one model and three analyses, and for curving behavior two models and three analysis approaches. These models and analyses are briefly described in the following sections.

Before most of these models and analyses could be developed, it was necessary to develop methods for determining the nonlinear wheel/rail geometric constraint functions for arbitrary wheel and rail transverse profiles. Methods for calculating both linear and nonlinear creep force/creepage relationships were also needed. The work involved in pursuing these objectives is also described in the following sections.

### Wheel/Rail Geometric Constraints

The transverse or lateral profiles of the wheels and rails have a strong influence on all aspects of the lateral dynamic behavior of rail vehicles. Wheel/rail geometric constraint functions such as the differences of left and right wheel rolling radii and left and right wheel/rail contact angles and wheelset roll angle appear explicitly in the equations of motion. These functions are usually highly nonlinear and depend, to the first order, on the wheelset lateral displacement relative to the rails (and to the second order on the wheelset yaw angle). The forms of these nonlinear functions are governed by the lateral wheel and rail head profiles, the wheel and rail gauge, and the rail cant angles. Examples of these constraint functions are shown in Figures 1 and 2 for new and CN profile wheels on new rails.

Linear dynamic and steady state curving analyses have generally represented the influence of the wheel/rail geometry by constant values of effective conicity (obtained by linearization of the curve of the difference in rolling radii) and lateral gravitational stiffness (obtained by linearization of the curve for the difference of contact angles). The linearization technique depends on the type of analysis to be performed. Examples of methods used for linearization are linearization about the assumed equilibrium point, least squares fit of a straight line characteristic over a specified range, and quasi-linearization [1-3].

Perhaps the only wheel/rail profile combinations that are easy to linearize are those of new wheels on new and worn rails. For these combinations, the important wheel/rail geometric constraints are essentially linear up to the value of lateral displacement where flange contact occurs. For wheels having other than a straight taper, the constraint functions are nonlinear in the lateral wheelset displacement over the entire range of lateral displacement. If other than straight tapered wheels are to be considered, it is essential, for prediction of lateral dynamic behavior, to know these wheel/rail geometric constraint functions.

In this program, two analyses and accompanying computer programs with Users' Manuals have been developed to calculate these wheel/rail geometric constraint functions. The first of these [4]

addresses the case where the left and right wheels and the left and right rails are, respectively, mirror images of each other. The second [5] relaxes this condition and permits the consideration of wheels and/or rails that are not mirror images of each other. The latter case is of course the more realistic one. Additionally, a technique utilizing cubic splines for calculating the curvature of the wheel and the rail is incorporated in the analysis and program reported in [5]. This information is essential for calculating creep coefficients and nonlinear creep force/creepage relationships.

A technique was also developed under this program [6] for fast and efficient digitization of graphical wheel and rail profile data. These digitized data are needed as input to the wheel/rail geometric constraint programs of [4] and [5].

#### Creep Force/Creepage

The shear stresses acting between wheel and rail in the contact region give rise to creep forces and moments. Estimating the level of creep forces and moments that prevails for a given vehicle is perhaps the most difficult aspect of estimating the parameters necessary for a theoretical analysis of rail vehicle dynamics.<sup>(1)</sup>

Kalker's nonlinear theory on creep [8, 9] is regarded by most to be the most complete theory available. Kalker's theoretical predictions of the nonlinear creep force characteristics have been substantiated by laboratory experiments [7]. In addition to predicting the longitudinal and lateral forces due to the relative longitudinal and lateral velocities or creepages between wheel and rail, Kalker's linear and nonlinear theories (of all those available) are the only ones that predict the lateral creep force due to a relative angular velocity component normal to the contact area between wheel and rail. This force, called the lateral/spin creep force, is usually less than the lateral creep force due to lateral creepage while wheel/rail contact remains in the tread region of the wheel. However, as the contact point moves towards the flange and the wheel/rail contact angle increases (as happens during curving, hunting, and incipient wheel climb), the lateral/spin creep force becomes much larger than the lateral force due to lateral creepage. Consequently, the importance of Kalker's theories in providing a basis for understanding the creep force mechanism and the effects of creep on rail vehicle dynamics should not be underestimated.

A computer program for calculating the linear creep forces and moments utilizing Kalker's linear theory has been developed during this project, and made available to those investigators requesting it. Additionally, the conversion from ALGØL to FØRTRAN of Kalker's program embodying his simplified nonlinear creep theory [8] has been completed recently. This theory is considerably more economical

of computation time than Kalker's complete nonlinear theory [9] and, as shown in Figure 3, agrees well with experimental results published in [11]. A version of this program was developed in subroutine form so that it might be incorporated easily in FØRTRAN computer programs for rail vehicle dynamics analysis.

#### Hunting Stability

One of the most severe problems facing the railroad industry today is that of ensuring that the various rail vehicles in service have an adequate margin of safety with regard to hunting stability. Unfortunately, the practical solution of the hunting problem for the complete fleet of rail vehicles is a long way off. However, the analytical modeling and analysis techniques developed in this effort should enable designers of rail vehicles to develop new vehicle designs as well as corrective measures for existing designs that offer sufficient safety against hunting.

Several different models and analysis techniques have been developed on this project. As the validation efforts continue, we will determine the conditions, range of validity, and appropriate application of each model and technique. As discussed previously, the intent has been also to develop the simplest credible model consistent with the ultimate use to which it will be put.

#### Linear Analyses

A series of models for the lateral dynamics of a single railway vehicle have been developed. These are shown in Table 1 and are classified by the numbers of degrees of freedom. Eigenvalue/eigenvector stability analyses have been developed and programmed for each of these models after linearization. These analyses predict the frequency and damping of oscillatory modes and the time constants for overdamped modes as well as the shape of each mode of the vehicle. This information permits the estimation of stability margins from the variation of the damping with speed of the least damped mode. An example of such theoretical results is shown in Figure 4.

It has been shown by comparison of results for the various models that the truck model used in the 9 degree-of-freedom (DOF) model is adequate for use in stability analyses of vehicles with roller bearing trucks. The critical speeds predicted by the 9, 17, and 19 DOF models for a vehicle having roller bearing trucks are very similar. However, the shape of the least damped mode predicted by the 19 DOF model differs from that predicted by the 9 or 17 DOF not only in the car body mode shapes due to car body flexibility but in the phasing of the motions of the front and rear trucks. Based on these results, it may be concluded that in the initial design stages of vehicles equipped with roller bearing trucks the 9 DOF model may be used to investigate stability.

<sup>(1)</sup>Hobbs [7] has reviewed both theoretical and experimental work in the area of creep.



Later in the design process, the effects of car body flexibility should be considered.

The selection of input data for use in these eigenvalue/eigenvector analyses is not a simple matter even when component test data are available. For vehicles like conventional North American freight cars, the lateral suspension characteristics are dominated by dry friction and other nonlinearities such as deadband and saturation. Thus, choosing effective or equivalent linear suspension characteristics requires considerable expertise and judgement as does interpretation of the results.

These computer programs have been used to conduct parameter studies encompassing the various configurations tested by the Association of American Railroads on the Union Pacific Railroad during the recent field tests.

To illustrate the application of these models in a design study, the 11 and 23 DOF models were used to examine the effects on hunting stability of various primary suspension elements and car body flexibility [12, 13]. In addition, a generic model of truck with interconnected wheelsets was formulated and a range of values for the interconnection suspension elements was examined. Typical results from this study are shown in Figure 5 where the critical speed for hunting instability is plotted versus interconnection shear stiffness for a vehicle having interconnected wheelsets.

These models and analyses have also been used to examine a potential maintenance problem. As a freight car accumulates service mileage, the wheels on a given truck develop different transverse profiles. A brief study was conducted to examine the effects on stability of using a different wheel profile for each of the two axles of a truck [14]. Various combinations of wheel profiles were examined. A typical result is shown in Figure 6 where critical speed for hunting is shown for a nominally empty 80 ton hopper car with various wheel profile configurations. The axles labeled "N" are those with the standard AAR new profile while those labeled "P" are those with profiled wheels having an effective conicity of about 0.31 together with a substantially increased value of gravitational stiffness. It can be seen that trucks with different wheel profiles on the leading and trailing axles exhibit critical speeds that depend strongly on the direction of travel (see Configuration 8 of Figure 6). Thus it would appear that one maintenance objective should be to maintain all wheel profiles of a truck to a common profile.

The operational practice of loading freight cars asymmetrically fore and aft was also examined briefly for its effects on stability [14]. It was found that stability was increased slightly when the vehicle was loaded in the rear as opposed to the front.

However, this difference usually was not as great as the difference in stability between empty and fully loaded cars.

As can be seen from this brief discussion, these models and analyses have been used and may be used in the future to answer various questions concerned with maintenance and operational practices as well as those concerned with vehicle design. The primary advantages of these linearized analyses are that they are very economical with regard to computer costs and offer a great deal of insight into the effects of various parameters on the vehicle dynamics.

#### Nonlinear Analyses of Hunting

The suspensions of North American freight cars are dominated by nonlinearities such as dry friction, deadband, and limiting or saturation. In addition, the wheel/rail interaction process is characterized by nonlinear wheel/rail geometric constraints and nonlinear creep force/creepage relationships. These nonlinearities strongly affect the lateral dynamic response of rail vehicles.

As discussed previously, there are many uses for linearized stability analyses of rail vehicles. These should be used with considerable care and judgement for vehicles with strongly nonlinear characteristics such as the freight car. For those cases where a detailed examination of the effects of these nonlinearities is desired and warranted, nonlinear analyses must be used. These should be used with discretion as the computation costs are usually at least an order of magnitude greater than those of linear analyses.

We are using two types of nonlinear analysis for examining the hunting stability of freight cars. One, called quasi-linear analysis, seeks to utilize linear analysis techniques in a special way to effect considerable computation cost savings over the standard approach to analyzing complex nonlinear systems, direct integration of the equations of motion [1, 2, 3]. We have also taken two approaches to integrating directly the equations of motion. We are using both numerical integration methods on digital computers and analog integration on a state-of-the-art hybrid computer. Although fewer companies have direct access to hybrid computers than to digital computers, hybrid computation can offer significant cost savings as compared with digital integration. These questions are discussed more completely in [15].

Quasi-linear analysis of hunting may be used to compute the existence and stability characteristics of limit cycles. The work reported<sup>(2)</sup> in [1, 2, 3] represents the first efforts in this area and results obtained for a

<sup>(2)</sup>This development of quasi-linear techniques for rail vehicle dynamic analysis was primarily supported by FRA through Transportation Systems Center Contract No. DOT-TSC-902.

simplified model agree very well with those obtained by integration of the equations of motion [2]. Thus, although only recently developed as an analysis technique for problems in rail vehicle dynamics, the future for the application of quasi-linear analysis techniques to rail vehicle dynamics problems appears bright.

Results obtained via quasi-linear analysis for the limit cycle amplitude versus speed are shown in Figure 7 for the 9 DOF freight car model. Unstable limit cycles may be thought of as stability boundaries while stable limit cycles represent the amplitude finally attained after hunting has started. These analyses may also be used to estimate the levels of the forces between wheels and rails and between vehicle components (depending on the particular model used) during hunting. We have applied the quasi-linear analysis technique to the 9 DOF freight car model described in Table 1 as well as to simpler models. However, it may be used in conjunction with almost any model.

The computation costs in developing curves such as those shown in Figure 7 via quasi-linear analysis are much less than the costs would be using hybrid computation and at least several orders of magnitude less than the costs associated with numerical integration of the equations via digital computation.

We are analyzing two nonlinear models of rail vehicles via direct integration. The first model is the 5 DOF half-car model described in Table 1. Nonlinearities considered are suspension friction and wheel/rail geometric constraints. This model is implemented on the Clemson University Engineering Computer Laboratory hybrid computer<sup>(3)</sup>. Because of machine capacity limitations, we cannot implement rail vehicle models with more degrees of freedom or nonlinear creep force/creepage relationships or consider large wheel/rail contact angles in the creepages or geometric constraints. Nevertheless, we feel that we have demonstrated the feasibility of hybrid computation for rail vehicle problems and have achieved cost-effective results with the half car model. We also have developed a numerical integration program on the digital computer for the 5 DOF model.

The second model we are analyzing via direct integration is the 9 DOF model described in Table 1<sup>(4)</sup>. This is being performed by numerical integration on digital computers<sup>(5)</sup>. As the cost of computation with numerical integration increases geometrically with the order of the system, our approach has been to develop the simplest credible model (in terms of the number of degrees of freedom) for a freight car equipped with roller bearing trucks. This implies neglecting degrees of freedom such as bolster pitch and sideframe rock that in all probability have only a minor influence (if any) on vehicle stability. However, recognizing the importance of effects such as nonlinear wheel/rail geometric constraints,

suspension friction, and nonlinear creep force/creepage relationships, we have attempted to model these effects with a high degree of fidelity. Large wheel/rail contact angles are considered in both the creepage expressions and the geometric constraint functions. A heuristic nonlinear creep force/creepage relationship is used based on Kalker's theory that includes the effects of spin creep on the lateral force. The feasibility of actually using Kalker's simplified nonlinear creep theory [8, 10] is being investigated<sup>(6)</sup>. As wheel/rail normal forces and contact geometry change dynamically, these effects are included in the program as they affect the creep forces and gravitational stiffness. Suspension friction is also included<sup>(7)</sup>. The computer program embodying this analysis is in the development stage.

In summary, we have developed six different models of freight vehicles (Table 1) and are using four different analysis techniques (linear eigenvalue/eigenvector analysis, quasi-linear analysis, numerical integration via digital computer, and analog integration using a hybrid computer) for evaluating the lateral stability of railway freight cars. We anticipate establishing the utility of each of these models and analysis approaches and have used them in a preliminary fashion to address questions concerned with vehicle design, maintenance, and operation.

#### Forced Response

Methods are needed that will enable the designer to estimate the acceleration levels and the forces between vehicle components. We have developed analyses that answer this need by predicting these quantities as the rail vehicle traverses irregular, rough track. To date our work has focused on developing cost effective analysis techniques for the forced response of the 9 DOF model described in Table 1.

<sup>(3)</sup> This system is described in [15] and comprises an EAI-680 analog computer linked to a PDP-15 digital computer through an EAI-693 interface.

<sup>(4)</sup> Actually, the model being used in this instance has 13 DOF. The additional four degrees of freedom are rigid body angular rotation of each wheelset about the axle centerline. This is necessary to describe accurately the dynamics of vehicles traversing curves at a constant forward speed.

<sup>(5)</sup> We are using two models of digital computers: the UNIVAC 1110 at Arizona State University, and the IBM 370/3165-II at Clemson University.

<sup>(6)</sup> There is no doubt as to the technical feasibility. However, computation time and costs are expected to increase and may not be acceptable unless additional efficiencies are effected elsewhere in the program.

<sup>(7)</sup> The choice of a suitable digital algorithm for analyzing dry friction is by no means straightforward. Our efforts in this area are described in [16].

The simplest and least expensive analysis approach uses standard linear frequency analysis techniques long used by vibrations and automatic control engineers. These techniques yield results in either of two forms. In the first, the amplitudes of the vehicle response variables (displacements, accelerations, forces across suspension elements and between wheel and rail) are obtained as a function of the frequency of the track alignment or cross level irregularity for given magnitudes of these irregularities. In the second, the power spectral densities (PSD's) of the same vehicle response variables are obtained in response to the PSD for either track alignment or cross level. As for the linearized analyses of hunting stability, the models must be linearized and effective or equivalent linear values chosen for suspension and wheel/rail characteristics.

To more accurately consider the effects of the suspension and wheel/rail nonlinearities, quasi-linear analysis was used [1, 2, 3]. The output characteristics of the various non-linear suspension elements may depend on either or both the magnitude and frequency of the input. The quasi-linear approach preserves this dependency and enables the designer to calculate the various vehicle response variables of interest. As in the case of the linear forced-response analysis, the track alignment irregularities are considered to be either harmonic at a fixed amplitude or representative of a Gaussian random process. When the input is considered to be random, results are obtained in the form of PSD's for the various vehicle response variables. However, in this case, these PSD results depend nonlinearly on the root mean square and shape of the alignment input spectrum.

As in the case of the stability analyses using the quasi-linear approach, the application of these techniques to the forced response of rail vehicles appears to be a very cost effective approach. Computation costs are on the order of several orders of magnitude less than those associated with the commonly used technique of digital integration of the equations of motion. The calendar and engineer time required to generate and interpret results of a quasi-linear analysis as compared with a digital integration program reflect these same ratios. However, it should be stated that there are approximations and assumptions necessary in a quasi-linear analysis that are not necessary when using a direct integration approach. Whether these pose difficulties in a given situation depend on the information desired. These topics are more fully addressed in [1].

The last analysis technique we are using to examine the response of the 9 DOF freight car to lateral alignment irregularities is that of digital integration of the equations of motion. The same analysis and computer program that is being developed to examine hunting stability of the 9 DOF model will be used here with prescribed lateral alignment irregularities. These may be of almost any form ranging from "bumps" to sinusoids to random signals. While

we anticipate that it may be necessary to use this last analysis approach in a limited number of cases, we anticipate that the computation costs will be very high requiring at a minimum 15 sec of computer time to simulate 1 sec of real time. Results of the digital integration analysis for the forced response of the 9 DOF freight car will be in the form of time histories of the response variables.

Summarizing, work in this project on the lateral forced response of freight cars has focused on developing linear, quasi-linear, and numerical integration analyses of the 9 DOF freight car model. The cost, facility of use, and results obtainable with these techniques vary. One of the results of the research will be to establish the most cost-effective approach for given types of problems faced by vehicle designers.

### Curving Behavior

The insurance of good curving performance should be of primary concern to the vehicle designer. Good curving performance is characterized by lack of contact between the wheel flanges and the rails. When flange contact does occur, it leads to increased rates of wheel and rail wear, higher levels of vibration in the vehicle, and an increased propensity for derailment. These factors have long been recognized. However, only with the relatively recent work of Newland [17] and Boocock [18] has the groundwork been laid for the development of curving analyses appropriate for designing for good curving performance.

The analysis techniques developed by Newland and Boocock are linear approaches that predict flange contact and slip boundaries. That is, for a given vehicle design, the combination of track curvature and cant deficiency necessary for wheel slip and flange contact may be calculated. If these values are plotted, slip and flange contact boundaries may be constructed on a graph of cant deficiency vs. inverse curve radius for the particular vehicle. Design changes that promote flange-free traversal of curves of smaller radius may be identified easily.

Three general approaches to the question of estimating curving performance have been used in this project. In the first, we have developed steady state analyses of the linearized equations of a freight car. These are solved to develop the slip and flange contact boundaries. This is the type of analysis that is inexpensive to do and may be used to evaluate a host of design possibilities.

The second approach entails a refinement of the first that enables the designer to evaluate the effects on the slip and flange contact boundaries of profiled or worn wheels. Actual nonlinear wheel/rail geometric constraint functions are introduced into the equations of motion which are then solved iteratively. Two models may be used in these analyses. The first

is a 9 DOF model of a freight car with conventional roller bearing trucks. The second is a 17 DOF model of a rail car with two generalized truck models. In this latter model, the car body has lateral, yaw, and roll degrees of freedom; each truck frame has lateral, yaw, and warp degrees of freedom; and, each wheelset has lateral and yaw degrees of freedom. Interconnected wheelsets and primary suspension elements may be easily considered with this model.

The third approach utilizes the complete nonlinear equations of motion for the 9 DOF freight car with roller bearing trucks. This analysis and program is the same one described previously for forced response and stability investigations. It employs digital integration of the equations of motion and consequently is much more expensive computationally than either of the first two approaches. However, as large contact angles and a nonlinear creep force/creepage relationship are considered, this analysis may be used to predict wheel/rail forces and L/V ratios under conditions of hard flanging. Consequently, this program may be of greatest use in studies of wheel and rail wear and derailment. The results of this program are in the form of time histories of the vehicle response variables such as displacement, accelerations, and forces between vehicle components and between wheel and rail. Curve entry and exit as well as curve negotiation may be investigated. It should be noted that while this latter model and analysis is quite detailed and entails considerable modeling fidelity, some factors that may be quite important in addressing questions such as derailment are not included. The most important of these are track flexibility and simultaneous two point wheel/rail contact for a given wheel.

In summary, we have developed (or are in the latter stages of developing) a range of models and analyses for investigating curving behavior. The first two of these are inexpensive computationally and should be of particular use to the vehicle designer. The third model and analysis is expensive computationally and is probably of most use in the latter steps of vehicle design or in studies of such phenomena as derailment and wheel/rail wear.

## VALIDATION TECHNIQUE

### Approach

Theoretical model validation may be undertaken at many different levels. At the lowest level, a qualitative correlation is obtained between theoretically predicted trends and experimentally observed behavior. For example, almost all linear stability analyses of rail car lateral dynamics predict that vehicles whose wheelsets have high "effective" conicities will hunt at lower speeds than those with low "effective"

conicity wheelsets. This trend agrees with observations of rail vehicle operation.

A great deal of qualitative validation has been done to strengthen confidence in the analytical tools. In stability analyses, the effects of changes in primary yaw and warp stiffness on critical hunting speeds have been qualitatively correlated. In curving analyses, the effects of yaw stiffness and wheelset conicity on lateral to vertical force ratios have also been qualitatively correlated.

The value of qualitatively validated analyses should not be underestimated. Such models are invaluable in making design changes and in devising successful experiments, because they provide information about the sensitivity of the vehicle behavior to parameter changes and also provide a framework for interpreting and understanding the test results.

A second level of validation entails correlation of a single, usually critical, value from the analysis with experimental results. For example, the analytical predictions for the critical speed when hunting begins or the resonant speed for rock and roll behavior would be compared with experimental measurements of the same variable.

Too much validation of this sort is done. This can be quite dangerous because it may lend false confidence to an analytical model. This is particularly true when those model parameters that cannot be measured are varied to obtain agreement between analysis and experiment. This second level of validation is generally of questionable value. It is far better to proceed directly to a full validation of the type described below.

The highest level of validation entails a fairly complete quantitative correlation of analytical and experimental results. A frequently used approach is the direct comparison of experimental and analytical time histories of variables such as acceleration, displacement or force level. Another possibility is the comparison of power spectral density curves. A third possibility for validating stability analyses by comparing the variation of system damping ratios with speed is discussed in detail below.

Because any mathematical model is only valid for a limited range of conditions, the validation comparison need only cover the range of model validity. The most common limitation is a bound on the frequency range of the mathematical results. For example, most lateral rail vehicle dynamic analyses are not valid beyond 20 Hz, and many are not valid above 10 Hz. Other limitations may concern the amplitudes of the motions (to avoid suspension or wheel tread nonlinearities), the type of car body, the type of wheel tread, or the nature of the track irregularities.

This highest level of validation is necessary before one can rely on the quantitative results

of a mathematical analysis. Our objective, in this project, is to achieve a quantitative validation of theoretical analyses for hunting stability, forced response, and curving behavior. To our knowledge, a quantitative validation of this type for the lateral dynamics of a rail vehicle has not been successfully completed. Our approach to each of these areas is briefly discussed below. A more detailed discussion of these matters is found in [19, 20, 21].

### Stability

Sustained hunting oscillations are one of the most important problems associated with freight car dynamic behavior. A major objective of this project is the development of mathematical models that will predict the speed at which sustained hunting oscillations occur, the influence of design changes on this speed, and the stability margin available at lower speeds.

The rail freight car behavior at any speed can be loosely<sup>(8)</sup> described as the sum of motions in several different modes. Each mode is characterized by a particular frequency, damping ratio and mode shape, where the mode shape is a particular amplitude and phase relationship between the motions of the various system components. Terms such as upper center roll, lower center roll, nosing, and fish tailing are often used to describe such mode shapes.

The stability of the freight car dynamic response is determined by the mode that has the least amount of damping. For a linear system, the response of any variable is mathematically expressed as,

$$x_i = \sum_{j=1}^n a_{ij} e^{-\zeta_j \omega_{nj} t} \cos(\omega_j t + \phi_j)$$

where:

- n - number of state variables
- $\zeta_j$  - damping ratio for mode j
- $\omega_j$  - frequency of mode j
- $\omega_{nj}$  - undamped frequency of mode j
- $\phi_j$  - phase angle for mode j

Thus, a stable system will have  $\zeta_j > 0$  for all modes and an unstable system will have  $\zeta_j < 0$ . The transient response of a motion for several different damping ratios is depicted in Figure 8.

Validation of the mathematical analyses can be achieved by comparison of predicted and experimentally determined damping ratios, frequencies and mode shapes at several speeds. As shown in the analytical results for freight car behavior depicted in Figure 4, the damping ratio of the least damped mode varies with speed. The intersection of the damping ratio

curve with the horizontal axis occurs at the predicted critical speed of hunting.

We attempted to obtain this system damping information from the field tests in two different ways. During certain test maneuvers, a hydraulic truck forcer system applied a torque between the truck and the car body. This system caused an initial translation and angular displacement of the truck components. The objective in using the forcer was to obtain transient response data that would provide damping ratio information of the type shown in Figure 4.

This same system damping versus speed information is being extracted from the random response tests on the tangent track using the random decrement technique. The random decrement technique, originally developed for aircraft flutter test analysis, [22], also provides a transient "signature." Figure 9 illustrates a random decrement signature obtained for the test vehicle at 15 mph. As one can see, the damping ratio and frequency can be determined directly from these signatures to produce an experimental curve of damping ratio versus speed such as that shown in Figure 10 for the test vehicle with CN Profile A wheels and constant contact side bearing. A least squares curve fit was used to match a damped sinusoidal response to the random decrement signature. Note that the 20 Hz noise caused by the motor-generator set on the instrumentation car may also be seen in the random decrement signature. To our knowledge, this is the first application of the random decrement technique for rail vehicle stability analysis.

The validation process entails comparing analytically determined values for the damping ratio and speed such as those shown in Figure 4, with experimental results for the comparable configuration. We are now doing this.

### Forced Response

The mechanics of validating the forced random response of a freight vehicle are simpler than stability or curving analysis validation. Previous studies have shown that response to specific track irregularities, such as low joints, [23] and response to vertical track irregularities [20], can be predicted fairly well by analytical means. For example, power spectral densities from experimental vertical acceleration measurement made in the TDOP tests [20] were compared with analytical computed PSD's. As seen in Figure 11, quite good agreement was obtained, despite the nonlinear friction present in the system.

Previous attempts to validate analyses for lateral response to random rail irregularities have not been very successful. The British Rail Research Center effort [24] attributed

<sup>(8)</sup> This description is mathematically correct for linear systems, and can be used with caution for nonlinear systems such as the rail freight car.

their difficulties to three factors: 1) the use of profiled wheels whose "effective" conicity depended on the amplitude of the motion, 2) the unknown level of the actual creep coefficients, and 3) the fact that the actual lateral input spectrum was not known because they did not measure the rolling line offset. The British researchers assumed the rolling line offset was equal in magnitude to the centerline alignment irregularity and adjusted the input spectrum accordingly. Effort was made in our project to avoid these difficulties by testing with conical as well as profiled wheels, measuring rail head profiles to assess rolling line offset, and devising tests to determine the actual creep coefficients.

The shape of the least damped mode, and in some cases those of other modes can also be obtained from the forced response data. Cross-spectral densities used in conjunction with PSD's provide transfer functions between variables that can be used to find the relative amplitudes and phases between the component motions. This provides additional information to strengthen confidence in the validity of the mathematical analysis.

Computer programs to carry out the data processing and reduction described here have been developed and Users' manuals [25] prepared. These will be turned over to the National Technical Information Service in the near future.

In this project, an attempt will be made to validate the curving analysis described earlier. During the field tests, relative wheel/rail displacements were measured during curving for all test configurations. In addition, instrumented wheelsets were employed to obtain lateral and vertical wheel/rail forces in the two configurations with standard AAR new wheel profiles. The test vehicle was operated through two curves of different curvature at three different operating speeds. The validation will involve a comparison of predicted and measured wheelset lateral and yaw displacements for the various vehicle configurations and test conditions. Comparison with the steady state wheel/rail force data will be made, where possible.

#### Validation Discussion

Our estimate of the enormity of the task we set for ourselves has grown steadily over the course of the project. Perhaps the greatest problem faced in any validation study involves obtaining accurate data for the system parameters needed in the mathematical analysis. In this study, uncertainty over the creep coefficients and creep force laws governing the relationships between the wheelset motions and the contact forces causes the greatest difficulty. Our analytical studies demonstrate a strong dependence of vehicle stability on the creep coefficients. For example, in the unloaded, new wheel configuration, we found a variation from 44 mph to 80 mph in critical speed as the creep coefficients vary from 50% to 100% of the

values predicted by Kalker's theory. This sensitivity to creep coefficients was borne out by test experience. The new wheel configuration tested had instrumented wheels with new AAR profiles but a few thousand miles of running fitted in one truck, while the other truck was equipped with new wheels of the same profile with the casting scale intact. The truck with "run in" wheels began hunting around 55 mph while the other truck did not hunt until nearly 80 mph.

A test maneuver employing the hydraulic forcer system was designed to extract independently the needed creep coefficient information. Unfortunately, shortcomings in the experimental procedures used in these creep tests rendered the data unusable. Thus the creep coefficients must be extracted indirectly from the test results, a procedure that undermines our confidence in the validation results.

Similar uncertainties exist in some of the vehicle parameters. Martin-Marietta [26] tested one of the two trucks actually used on the test vehicle, but some characteristics, such as the friction levels in the suspension, may differ between trucks and vary over time as surfaces wear and atmospheric conditions vary. An uncertainty also exists in the value of the extremely sensitive centerplate friction torque. This was not measured for these trucks, and consequently the actual friction values are not known. An estimate based on tests conducted by ASF [27] was used in the analytical work.

Lack of precise knowledge of the track data also poses difficulties for the validation effort. As explained in the next section, the track geometry data available to us is limited because of two accidents with the track measuring vehicles. In addition, we were able only to measure rail head profiles at a limited number of stations and hence can not construct a continuous estimate of rolling line offset.

Perhaps the most difficult problem is caused by the nonlinearities in wheel/rail geometry and suspension. These nonlinearities due to curved wheel profiles, dry friction and suspension stops cause the vehicle dynamic response to depend on the amplitude of the motion. Because the amplitude depends on the imprecisely known track input, direct comparison of experimental and analytical results is difficult. Our validation effort attempts to avoid this as much as possible by comparing indirect attributes such as frequency, damping, etc.

As in all activities, the validation process is better understood in hindsight than foresight. In particular, many aspects of the test conduct should have been done differently to produce better experimental data for the validation effort. As a result of shortcomings in the testing procedure certain vehicle parameters are not well known, the creep coefficients were not found, fewer non-hunting data points than desired were obtained, and the

transient response data from the hydraulic forcer exercises is limited. Thus, more estimation than we would like is involved in the validation process. However, we have redundancy in our procedures and consequently have sufficient data to complete the validation process. Our experience should be invaluable in insuring that the next generation of validation tests will avoid these problems and provide a higher level of confidence in the validated models.

## FREIGHT CAR FIELD TESTS

### Test Description

The field tests to provide data for the validation effort were planned in cooperation with the Association of American Railroads and conducted by the Association of American Railroads and the Union Pacific Railroad. The tests were conducted during late fall and early winter of 1976-77 on the Union Pacific mainline west of Las Vegas. The test objectives, test philosophy and test requirements for these tests are discussed in detail in the program planning document [28].

The test vehicle, shown on the test site in Figure 12, was a Louisville and Nashville Railroad Company, 80 ton, open hopper equipped with 70 ton (6 X 11 journal) A-3 Ride Control trucks. The component characteristics of one of the trucks were determined in tests conducted by Martin-Marietta [26]. The field tests were conducted with the vehicle in the eight different configurations of wheel profile, load, side bearing pressure, track warp stiffness and centerplate condition shown in Table 2. As can be seen, the tests were conducted with the car in a range of conditions similar to those encountered in revenue service. Light and loaded vehicles, lubricated and dry centerplate, new AAR profile and Canadian National "Profile A"<sup>(9)</sup> wheels were variations to obtain data to determine whether the theoretical analyses predict the effects of such changes on vehicle dynamic performance. The truck warp stiffener and the constant contact side bearings with variable load were tested to investigate concepts that may improve vehicle dynamic behavior.

The test vehicle was equipped with the hydraulic truck forcing system, mentioned earlier, that exerted a torque between truck and car body. This system caused an initial translation and angular displacement of the truck components to provide a controllable initial condition and subsequent transient response.

The L & N Hopper car was instrumented by the AAR Research Center to measure 22 acceleration values, 49 displacement values, wheel/rail forces for one truck set of the new wheels, and train speed. This instrumentation included 14 displacement transducers to measure the relative lateral and angular position of the wheel relative to the rail. This latter instru-

mentation was developed by Reaction Instruments for the AAR.

The signals from the transducers were conditioned, digitized and recorded by the instrumentation system on board the AAR 100 Instrumentation car. The data was sampled at 100 samples/second and recorded on 1600 BPI magnetic tape. Header and trailer records on the tapes provide calibration and test conduct information.

The tests were conducted on both tangent and curved track sections on the Union Pacific Railroad. The AAR provided the test car, instrumentation car, test manager and test crew, while the motive power, caboose and train crew were provided by the Union Pacific Railroad.

The tangent test site was a 12,000 foot section of continuous welded rail on the Union Pacific mainline in the Mojave Desert between Yermo, California and Las Vegas, Nevada. Tangent tests for each configuration were conducted at four different speeds. In turn, both "unforced" and "forced" runs were made over the test zone at each speed. In the "forced" runs, the hydraulic forcer system was repetitively activated to cause an initial displacement and released to allow a transient motion.

The curving tests were conducted on Union Pacific track between Sloan and Arden, Nevada. The tests were run with each configuration at three speeds through a one degree and six degree curve.

The curved and tangent track geometry was measured by the Union Pacific track geometry car in the fall of 1974. Because of the long delay before carrying out the tests, attempts were made during and immediately after the testing period to resurvey the track with the FRA Track Measurement cars and the Track Survey Device. Both these attempts ended disastrously. The FRA cars were severely damaged in a derailment on the curve test site and the Track Survey Device was involved in a collision with a revenue train. The tangent test zone was finally remeasured by the Track Survey Device this fall. However, the curve test zone was reworked in the interim and consequently the information about track geometry during the tests was lost.

During the test period, the rail head geometry was measured at about 200 locations throughout the test zones. This information was processed to determine variation in wheel/rail geometry along the track with both new AAR and CN Profile A wheels. The processed data for the variation in conicity, contact angle, etc., also provides an estimate of the rolling line offset and hence will permit the estimate of its influence on the vehicle dynamics.

<sup>(9)</sup>The CN Profile A wheels used in these tests were developed by the Canadian National Railroad with the goal of achieving long tread life and good curving performance.

## Data Processing and Reduction

The test data collected during the field tests nearly filled eleven reels of 1600 BPI magnetic tape. Table 3 summarizes the test conditions under which data was gathered. The first processing step entails reading the raw data tapes, converting to engineering units, combining channels to compute the desired model variables, computing desired statistics such as mean values, standard deviations, and histograms, and plotting the time histories of selected model variables. At present this process has been completed for the tangent tests with configurations 2, 3, 4, 6, 7 and 8.

A typical response of one truck to the hydraulic forcers is shown in Figure 13. Note that in some cases, insufficient time was left between turning the forcer off and turning it on to observe several cycles of the damped sinusoidal response. This problem was purely operational. The technique proved capable of providing the desired transient response information.

Figure 14 shows the Configuration 6 (CN Profile A Wheels) A-truck lateral displacement during the unforced tangent tests of 35 mph. Note here that hunting starts and stops during the test run at constant speed. This behavior, which has been observed by others, is probably due to either the amplitude dependence of stability for the nonlinear system or due to changes in the rail head or surface condition along the track. We are using our nonlinear analyses and rail head profile data to investigate these two possibilities.

The next step in data processing is to use the random decrement and spectral analysis programs to analyze the random data. Examples of the random decrement results have been discussed earlier. We had planned to extract damping ratio and frequency directly from the forcer test data. Unfortunately, usually too little time was left between forcer applications during testing so only a very few runs provided useful data for estimating damping ratio.

The power spectral densities have proven very useful. For example, Figure 15 shows a PSD for the A-Truck lateral displacement. Note that the least damped mode, at 1.4 Hz is easily identifiable as well as a lesser peak at the wheel revolution frequency, 8 Hz. We have found that damping ratio estimates obtained from these PSD's agree remarkably well with values obtained by the random decrement technique.

Additional processing to compute cross-spectral densities and transfer functions between variables is the last data processing step. This processing yields mode shapes, i.e., amplitude and phase relationships between component motions.

## CURRENT STATUS AND SUMMARY

### Completed Models

Briefly, we have completed the development

of all the models concerned with lateral stability, lateral forced response, and curving performance. These models are described in Table 1.

### Completed Analyses and Computer Programs

We have completed development of all computer programs dealing with the linear and quasi-linear analysis of freight car stability as well as the hybrid computer program and digital computer program for the stability analysis of the 5 DOF model. A digital integration routine has been programmed for the response of the linear 9 DOF model to initial conditions. The programs for the linear and quasi-linear forced response of the 9 DOF model are also operational. The programs for the linear and nonlinear steady state curving performance of the 9 and 17 DOF models are complete. These calculate performance at given values of cant deficiency and track curvature. A search algorithm and accompanying subroutine for the calculation of slip and flange contact boundaries has been developed.

Supporting programs have been developed for use in calculating creep force/creepage relationships. Two programs have been developed to find: a) the linear creep coefficients using Kalker's linear theory, b) the nonlinear creep force/creepage relationships using Kalker's simplified theory [10].

Programs have also been developed to provide data concerning the wheel/rail geometric constraints. A hybrid computation procedure has been created to digitize accurately and quickly graphical wheel and rail profile data [6]. Two programs have been written to find the wheel/rail contact characteristics and resulting geometrical constraint relationships. One of these treats the case where both wheels and rails are mirror images of each other [4] while the other treats the general case where left and right wheel and/or rail profiles are arbitrary [5].

### Efforts Underway

Work is underway on integrating the subroutine for calculating slip and flange contact boundaries into the programs for calculating both the linear and nonlinear steady state curving performance of the 9 and 17 DOF models. The digital integration program for calculating the stability, forced response, and curve entry and negotiation of the 9 DOF model is being developed.

Documentation on these various theoretical efforts is only partially complete. We plan to issue a report on the modeling and analysis for the linear studies of the stability and forced response of the 9 DOF model. The linear studies for steady state curving performance of the 9 and 17 DOF models will be included in this report. A second report will be issued describing the nonlinear studies of the stability, forced response, and curving behavior for the various models.



## Validation Effort

The validation field tests have been completed and the raw test data, in the form of 22 data files on 11 magnetic digital tapes, has been received from the AAR. The AAR plans to prepare a report documenting the instrumentation and conduct of these field tests.

An array of computer programs has been prepared to process the test data. These programs include the following:

READAAR100 - reads the raw data tapes furnished by the AAR, converts the recorded signals from binary to engineering units, combines signals to form the model variables of the 9 and 19 degree of freedom models, prints header and trailer records, and writes processed data in disc files and on magnetic tape.

DATAPLOT - prepares CALCOMP plots of any data channel or model variable.

PSD - computes and plots any or all of the following functions: power spectral density, cross spectral density, auto correlation, cross-correlation, probability density, cumulative distribution, mean, standard deviation, transfer function and coherence.

RANDEC2 - finds a random decrement signature by prewhitening and averaging a series of time-domain data records, finds the least squares fit of a decaying sinusoid to the random decrement signature, prints the damping ratio and speed, and plots the random decrement signature and fitted curve.

DCOR - corrects the initial damping ratio estimate from RANDEC2 to compensate for speed variations during the tests.

All of the tangent test data, 11 of the 22 files, has been processed by the AAR100 program. Random decrement signatures and PSD's have been computed for most of these cases. Data plots have been prepared for selected variables in all runs. Thus, the tangent test data analysis is nearly complete.

The curving and creep test data has not been processed yet because position references for the wheel/rail displacements, needed to determine the actual wheel-to-rail positions, are not known. Due to an oversight, the measurements made to fix these references during the tests were insufficient to completely determine the reference values for all the displacement transducers. We will attempt to reconstruct this missing reference information by statistical means. If successful, data plots and statistics for the component displacements during the creep and curving tests will be prepared.

Two informal reports on the test plans and validation procedures have been prepared [19, 28], and one interim report on validation techniques written [21]. Validation of the stability and forced response models is now underway. A report on the results of this

effort is planned for next spring. The validation of the curving model depends on the success of the attempt to fix the displacement transducer references.

## Railroad Applications

To achieve maximum utility for application by the railroad industry, we are developing a variety of models, analyses, and computer programs for the analysis of the lateral dynamics of freight cars. This is necessary because there are many different kinds of questions asked by the industry that can be answered by such analyses. For example, to examine trends in performance with various design parameters, linear analyses are particularly attractive as much information can be obtained for modest computation costs. If a detailed examination of the effects of parameters such as suspension friction, curved wheel profiles, or adhesion level is needed, nonlinear analyses should be used. The need for these in a particular instance should be established carefully as computation costs for nonlinear analyses can be high. As we have discussed, it is especially critical to use models with a) the smallest number of degrees of freedom necessary to describe adequately the behavior being investigated, and b) high modeling and analysis fidelity for phenomena that are expected to influence strongly the behavior. (Typical of these are wheel/rail interaction effects and suspension friction.) The first condition is necessary because computation costs increase extremely rapidly with the number of degrees of freedom.

After analysis of the field test results is complete together with the validation of the various models, we will establish the range of applicability of each of the models. This is necessary because of the widely differing applications anticipated for such models and the wide range of associated computation costs. It would be a disservice to the industry to circulate only the most complex models and analyses because of the associated high computation costs and the high risk of "overkill" in using such models and analyses in many instances.

We have described briefly above initial studies that have been conducted to demonstrate the utility of the various models and analyses. These studies have treated questions associated with vehicle design as well as maintenance and operation. We anticipate that these models and analyses can be used extensively by the railroad industry in a similar manner and that the types of questions addressed will be limited only by the imagination of the user.

As formal documentation of much of the modeling and analysis work has yet to be completed, there has been to date only limited distribution of the analyses and computer programs. Nevertheless, we have distributed informally the programs for the work on wheel/rail geometry, creep force/creepage characteristics, linear stability analyses, and quasi-linear analysis to researchers in the U.S.A., Canada,

and Europe. We have also made every attempt to communicate the results of our work to the rail engineering community through distribution of informal reports and presentations, formal presentations through the technical societies, and tutorials at the MIT Rail Vehicle Dynamics short courses in the summers of 1975 and 1976. In doing so, we have hoped to achieve the exchange of ideas and information (as well as obtain constructive criticism) between researchers and potential users that is so necessary to the eventual success of this work.

### Recommendations

This effort has provided a great deal of knowledge that should be useful in future research projects. We believe that the simpler of the models developed in this effort will prove adequate for most studies of stability and forced response. However, the presence of nonlinearities, particular dry friction, will probably necessitate use of quasi-linearization or direct integration solution techniques in one form or another. Additional work is needed to improve the computational efficiency of these techniques so that they can be widely used by the railroad industry. It appears that more detail is appropriate in modeling the regime of curving under hard flange contact and derailment. Rail flexibility, three dimensional geometry, and a nonlinear creep theory that accurately includes spin effects should be investigated for inclusion in such a model. Efficient solution techniques, to make solution of the nonlinear curving problems feasible for design work should also receive additional attention.

Several suggestions for future field testing are evident from our experience. First, the test operations team should include at all times someone familiar with the theory and intended use of the test data. This person should have immediate access to test data to enable him to judge the quality of the test data and make immediate suggestions for improving test procedures. If we had done this, most of our problems with insufficient data would have been eliminated. Second, the importance of vehicle and environmental characteristics should be emphasized. Too often this is overlooked in the pressure to get "test" data. Procedures to determine regularly friction and wheel condition are needed. A procedure similar to that attempted in our tests to determine creep force conditions by employing the hydraulic forcers should be developed further.

Several suggestions concerning test equipment emerged. Our experience has demonstrated that the forcer system can provide useful data, and that stability information can also be extracted from random response data by the random decrement technique. The wheel/rail displacement transducers provided very good data, and should be used whenever possible. However, the high degree of redundancy in the instrumentation on our test vehicle should

not be necessary in future tests. The wide differences in behavior between the two trucks do suggest the need to include instrumentation on both trucks.

New ground in model validation has been broken during this project. The random decrement technique has proven highly successful for assessing stability. This process could be incorporated into an onboard mini-computer to allow on-line assessment of stability margins. The forcer system proved capable of introducing controlled inputs, and would be extremely useful in testing to determine creep conditions. An assessment of the importance of rolling line offset will be made from the processed wheel and rail head data. We expect that careful track measurements, including wheel and rail head profile measurements are appropriate for future tests.

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TABLE 1 - VEHICLE MODELS

Number of Degrees of Freedom	Description of Degrees of Freedom
5*	Half car model; one roller bearing truck with warp, yaw, and lateral DOF; half car body with lateral and roll DOF.
9	Full car model; two roller bearing trucks with warp, yaw, and lateral DOF; car body with lateral, yaw and roll DOF.
11	Half car model; one generalized truck with lateral, yaw, and torsional DOF of each of two wheelsets as well as lateral, warp, and yaw DOF of the truck frame; half car body with lateral and roll DOF.
17	Full car model; two generalized trucks with lateral and yaw DOF of each of two wheelsets as well as lateral, warp, and yaw DOF of the truck frame; car body with lateral, yaw, and roll DOF.
19	Full car model; two generalized trucks with lateral and yaw DOF of each of two wheelsets as well as lateral, warp, and yaw DOF of the truck frame; car body with rigid body lateral, roll, and yaw DOF. The use of a two mass approximation to the car body permits a first approximation to flexible car body torsion and lateral bending thus giving the car body a total of 5 DOF.
23	Full car model; this model is identical to the 19 DOF model discussed above with the addition of an axle torsional degree of freedom for each of the four axles. The effects of independently rotating wheels or axle torsional flexibility may be examined with this model.

\*This model was developed in the early stages of the research when it was thought there was a possibility of performing tests with a similar physical configuration on the Japanese National Railways (JNR) roller rig.

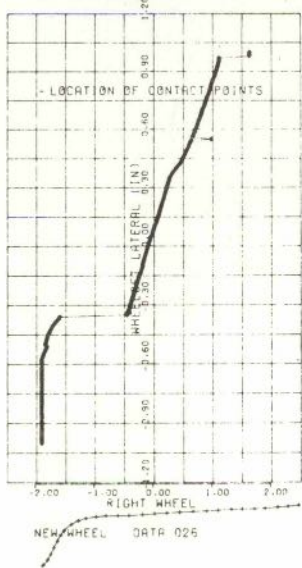
TABLE 2 - TEST VEHICLE CONFIGURATIONS

<u>Configuration</u>	<u>Wheels</u>	<u>Load</u>	<u>Side-Bearings</u>	<u>Truck Stiffner</u>	<u>Centerplate</u>
1	CN Profile A	Empty	0	None	Dry
2	CN Profile A	Empty	0	None	Lubricated
3	CN Profile A	Empty	2000 PSI	None	Lubricated
4	CN Profile A	Empty	4000 PSI	None	Lubricated
5	CN Profile A	Empty	0	ON	Lubricated
6	New	Empty	0	None	Lubricated
7	New	Loaded	0	None	Lubricated
8	CN	Loaded	0	None	Lubricated

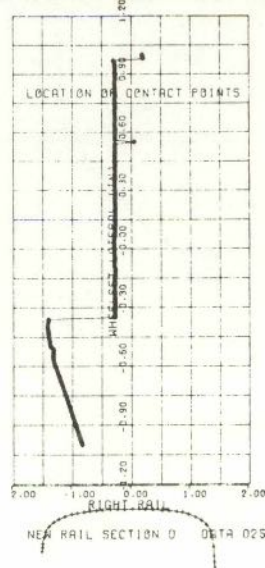
TABLE 3 - FIELD TESTS

<u>Configuration</u>	<u>Critical Speed</u>	<u>Speeds Run</u>	<u>Notes</u>
1. CN Wheels Empty Dry C.P.	40 mph	15, 25, 30, 35, 40 mph 15, 30, 35 mph 10, 30, 35 mph 20, 30, 35 mph	Unforced Forced 1° Curve 6° Curve
2. CN Wheels Empty Lubed C.P.	35-45 mph	15, 25, 35, 40 mph 20, 30, 35, 40 mph 10, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6° Curve
3. CN Wheels Empty 2000 PSI Airbags	50-60 mph	25, 35, 45, 50 mph 20, 30, 35, 40 mph 10, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6° Curve
4. CN Wheels Empty 6000 PSI Airbags	70-80 mph	40, 50, 60, 65, 70, 75 mph 50, 60, 65, 70 mph 10, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6° Curve
5. CN Wheels Empty Warp Stiffner	55-60 mph	35, 45, 50, 55 mph 35, 40, 45, 50 mph 10, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6° Curve
6. New Wheels Empty	"A" Truck 60 mph "B" Truck 80-88 mph	25, 35, 45, 55 mph 25, 35, 45, 55 mph 20, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6° Curve
7. New Wheels Loaded	No Hunting No Hunting	40, 50, 60, 70, 80 mph 40, 50, 60, 70, 80 mph 10, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6° Curve
8. CN Wheels Loaded	45-58 mph	20, 30, 40, 50 mph 50 mph 10, 30, 40 mph 20, 30, 40 mph	Unforced Forced 1° Curve 6°

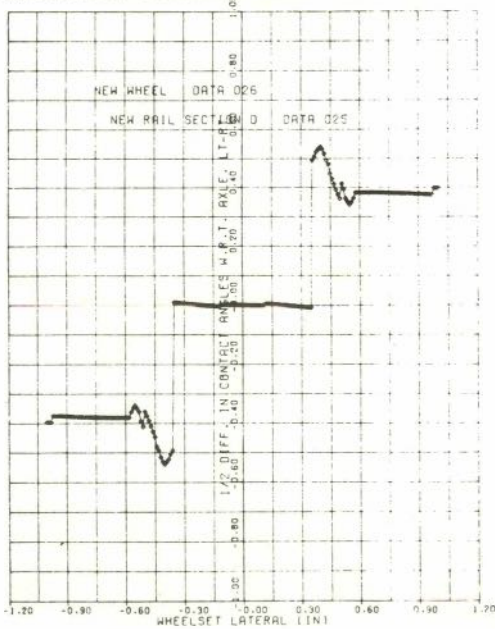
a. WHEEL CONTACT POSITION



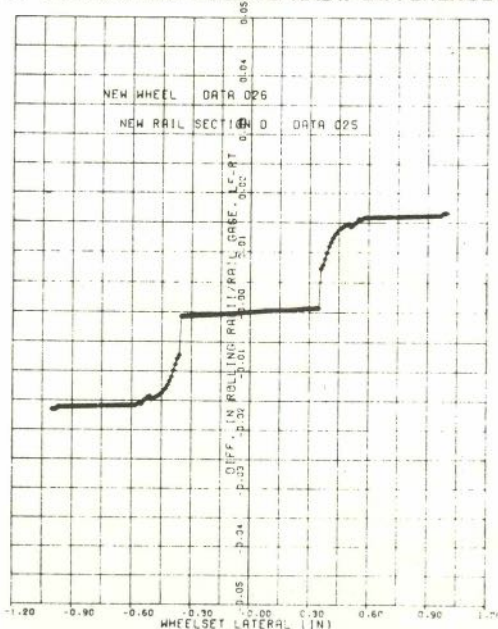
b. RAIL CONTACT POSITION



d. ONE HALF CONTACT ANGLE DIFFERENCE



c. NORMALIZED ROLLING RADII DIFFERENCE



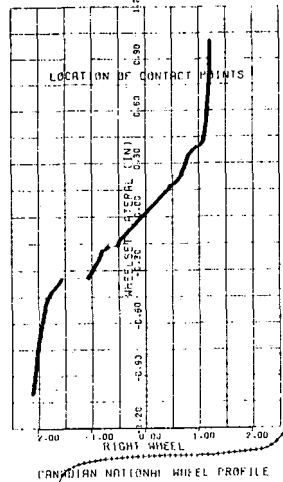
Wheel Gage 53 in

Rail Gage 56.5 in

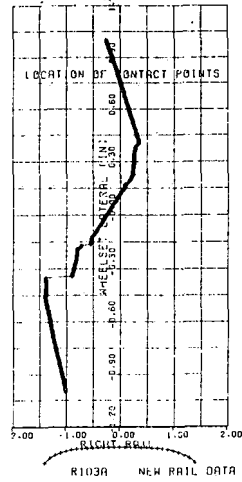
Rail Cant 0.025

Figure 1. Wheel/Rail Contact Characteristics and Geometric Constraint. Functions for New Wheels/New Rails

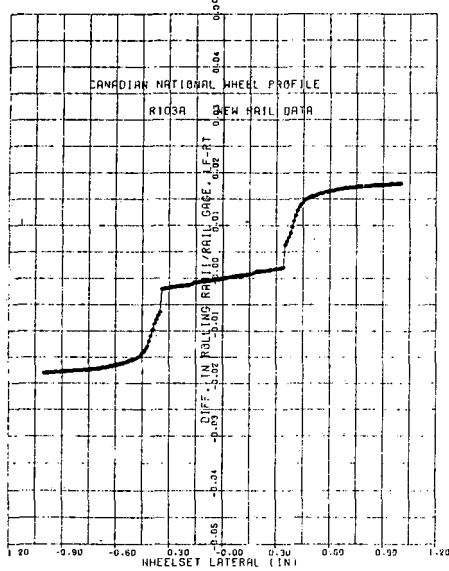
a. WHEEL CONTACT POSITION



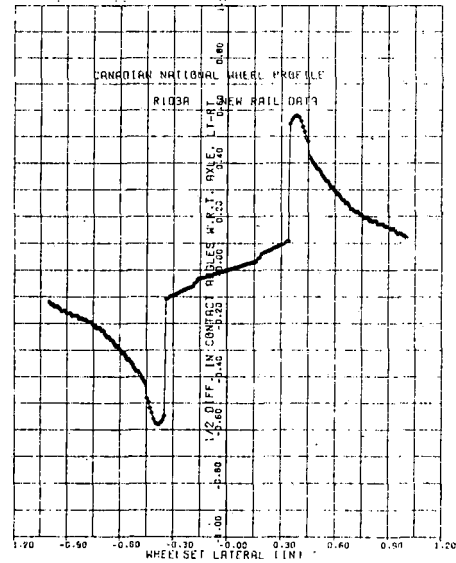
b. RAIL CONTACT POSITION



c. NORMALIZED ROLLING RADII DIFFERENCE



d. ONE HALF CONTACT ANGLE DIFFERENCE



Wheel Gage 53 in.  
Rail Gage 56.5 in.

Rail Cont 0.025

Figure 2. Wheel/Rail Contact Characteristics and Geometric Constraint Functions for CN "Profile A" Wheels/ New Rails

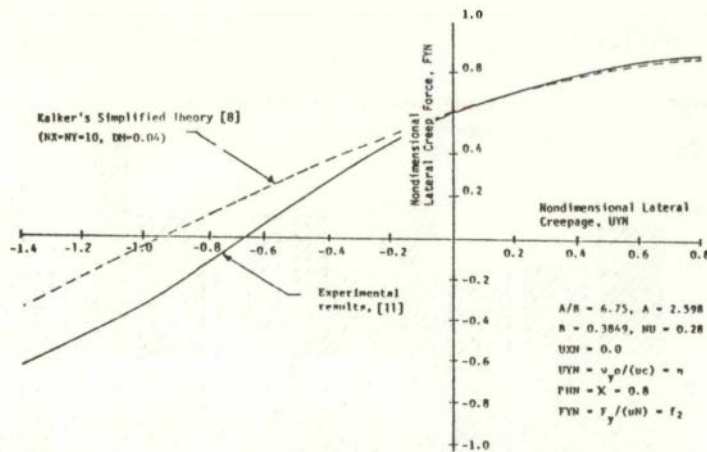


Figure 3. Comparison of Kalker's Simplified Theory with the Experimental Results of Figure 7, Reference [11].

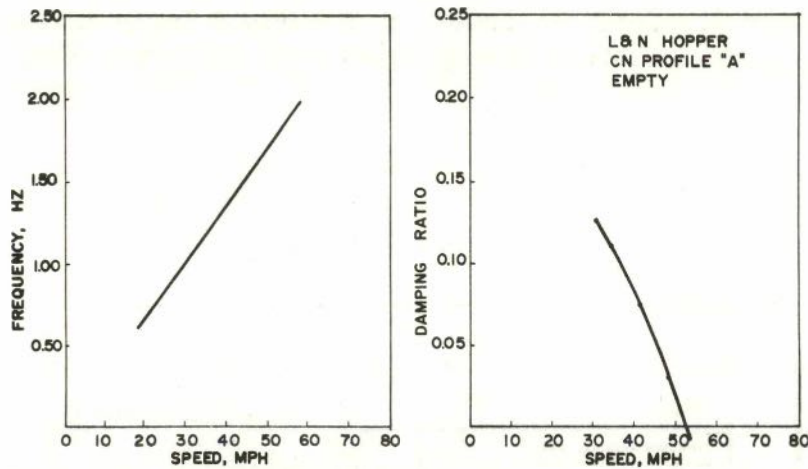


Figure 4. Theoretical System Damping Ratio and Frequency vs. Speed Relationships

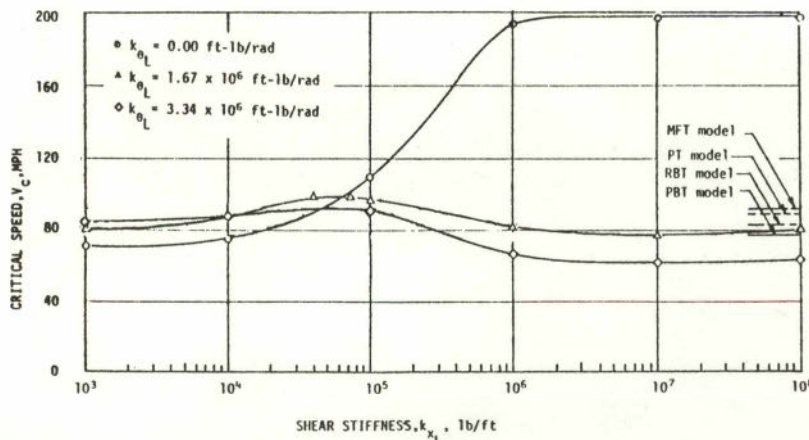


Figure 5. Effect of Shear Stiffness on Critical Speed of 11 DOF Model with Interconnected Wheel-sets for Different Values of Bending Stiffness

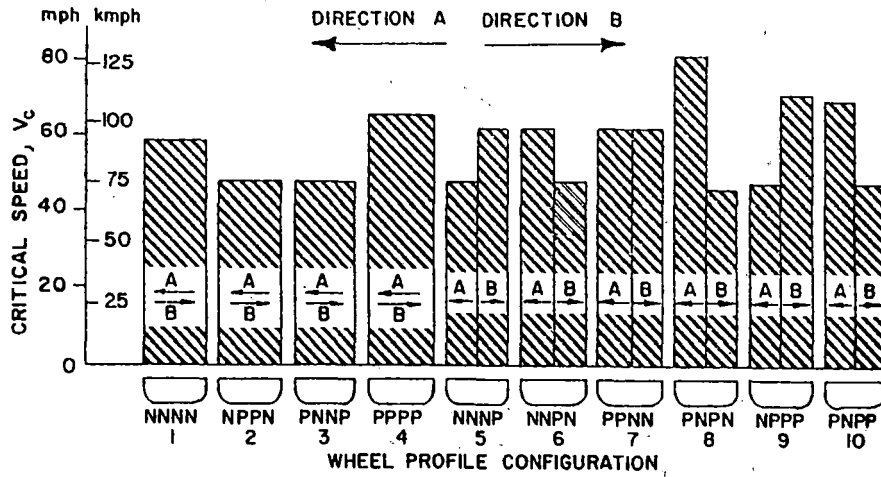


Figure 6. Critical Speeds of a Nominal Empty Freight Vehicle Having Axles with Different Wheel Profiles

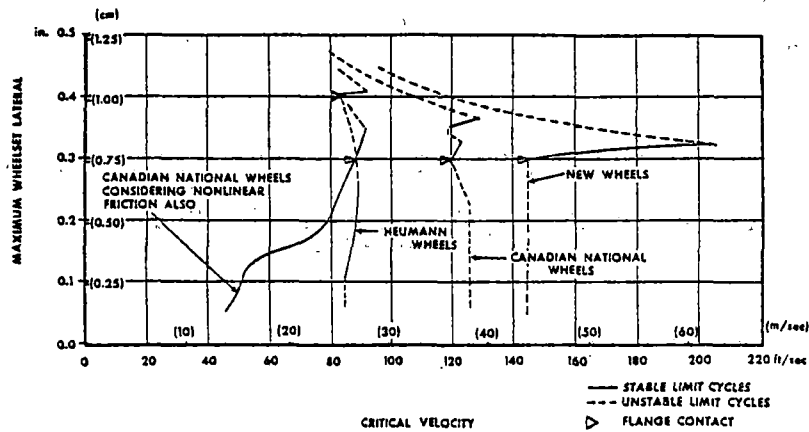


Figure 7. Limit Cycle Amplitude (Wheelset Lateral Displacement) vs. Speed for 9 DOF Empty Freight Car



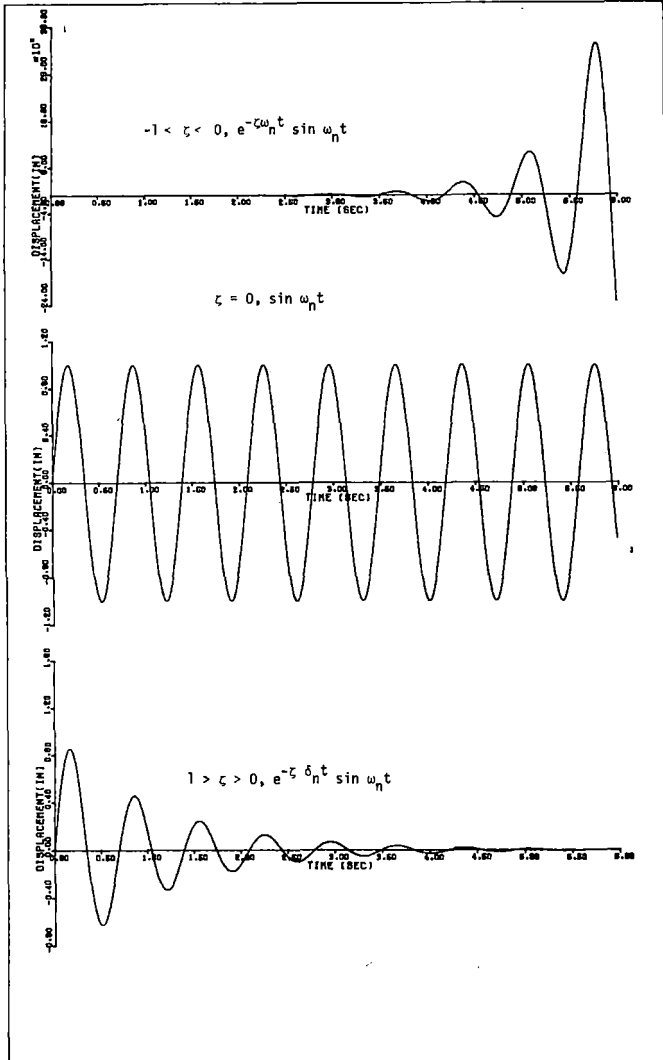


Figure 8. Typical System Dynamic Response

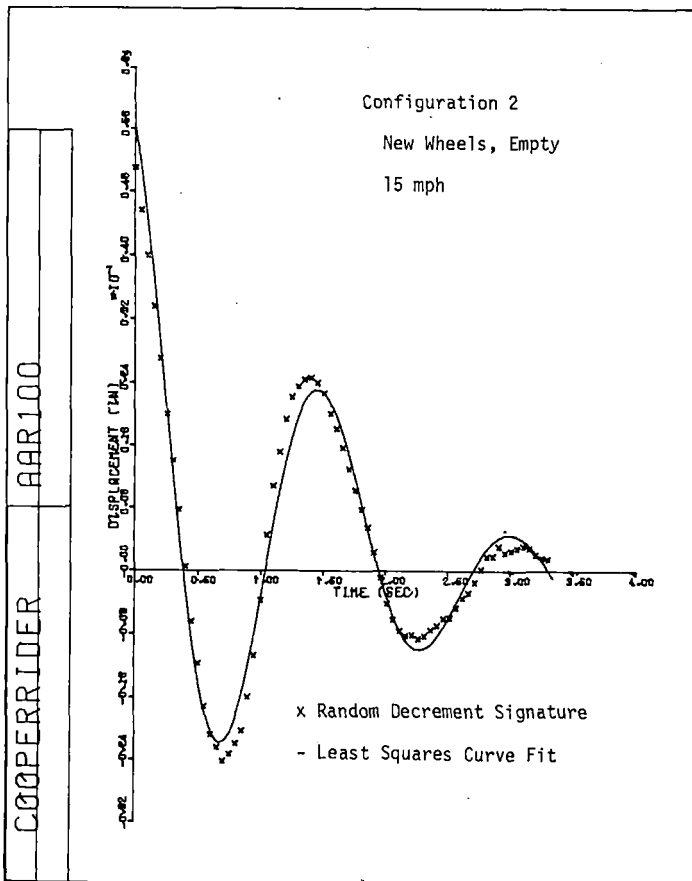


Figure 9. Random Decrement Signature

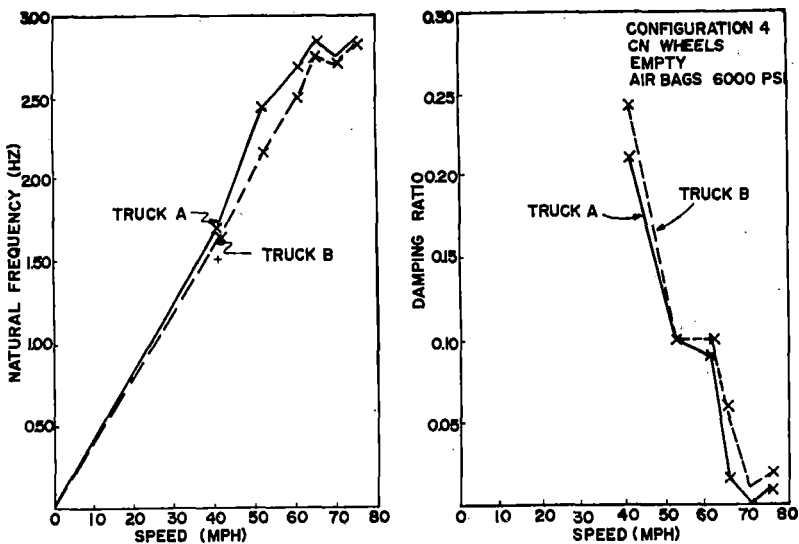


Figure 10. Experimental Damping and Frequency vs. Speed Relationships

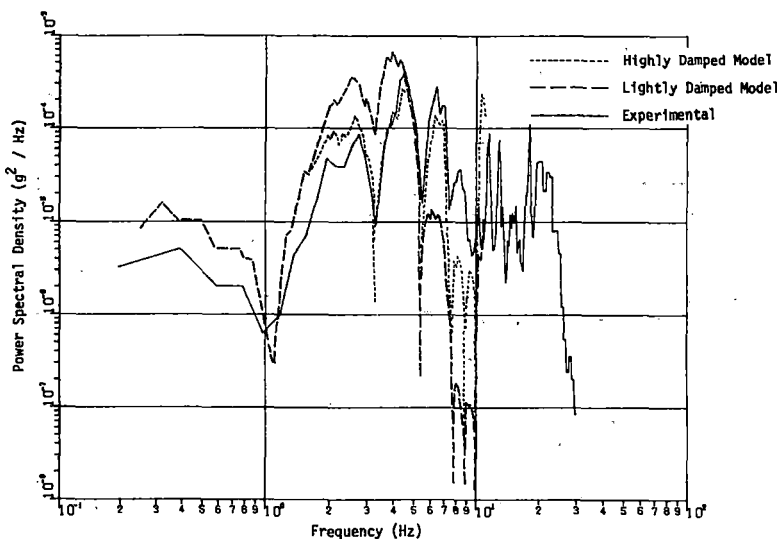


Figure 11. Freight Car Body Vertical Acceleration Spectra for Empty Vehicle at 100 ft/sec on CWR [20]

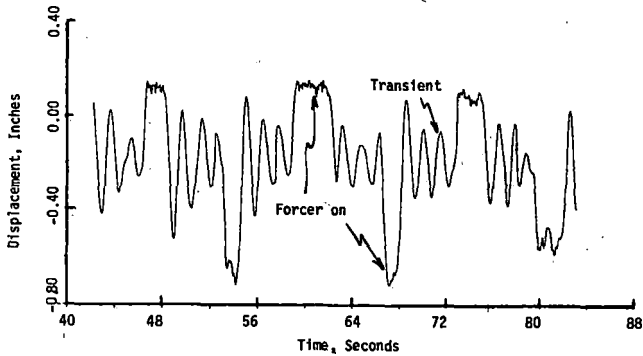
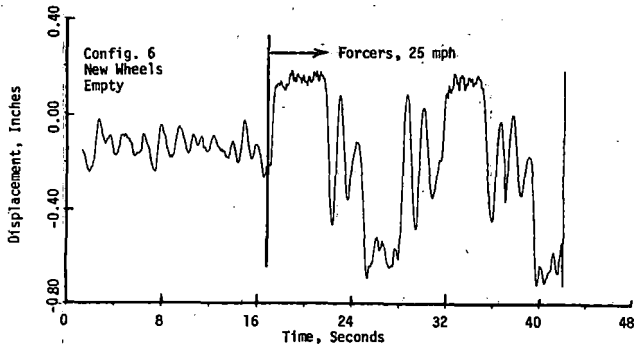


Figure 13. Truck Lateral Displacement Using Hydraulic Forcers

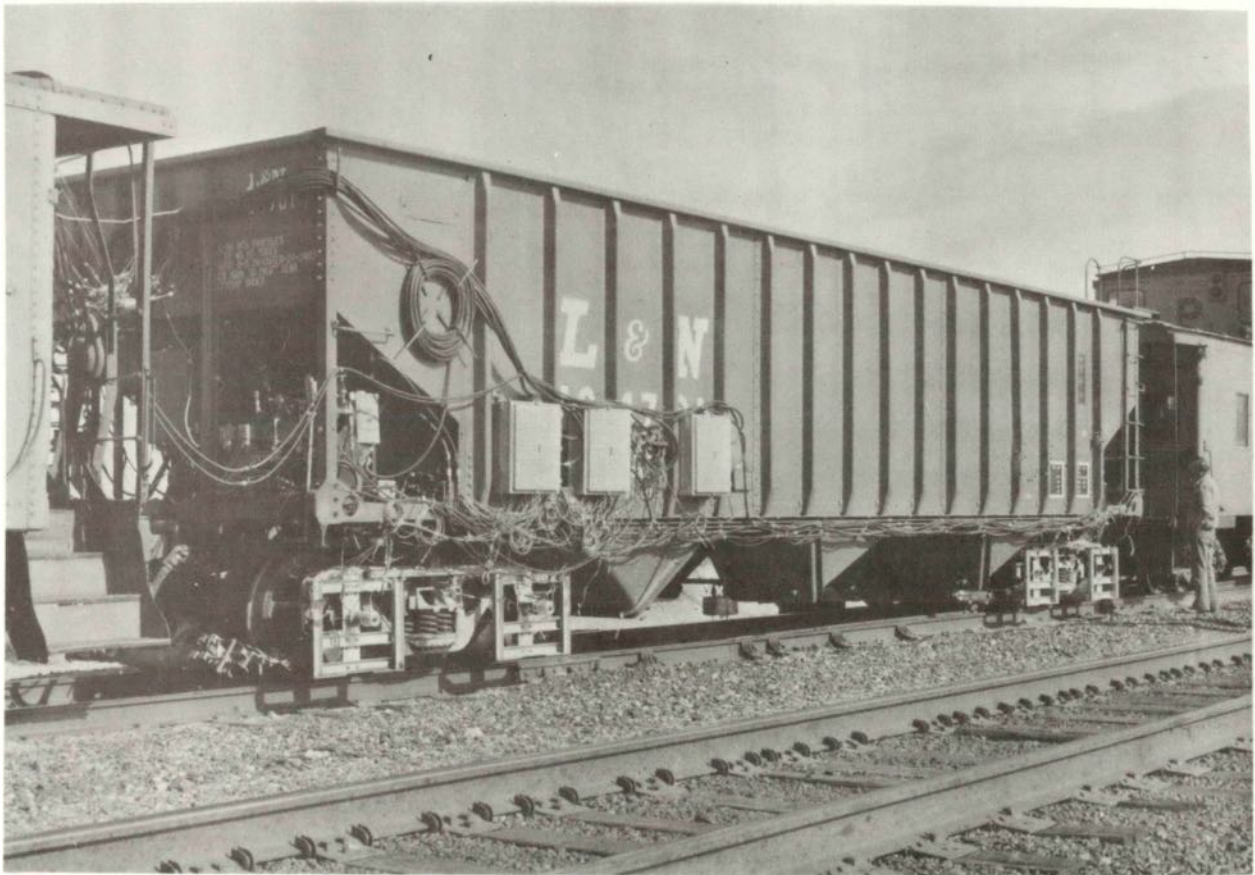


Figure 12. L & N Hopper Test Car

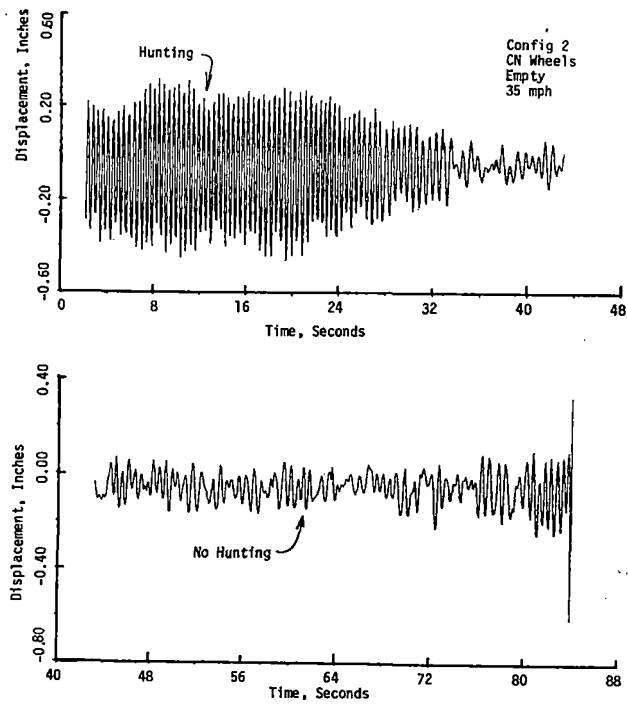


Figure 14. Truck Lateral Response to Random Rail Irregularities

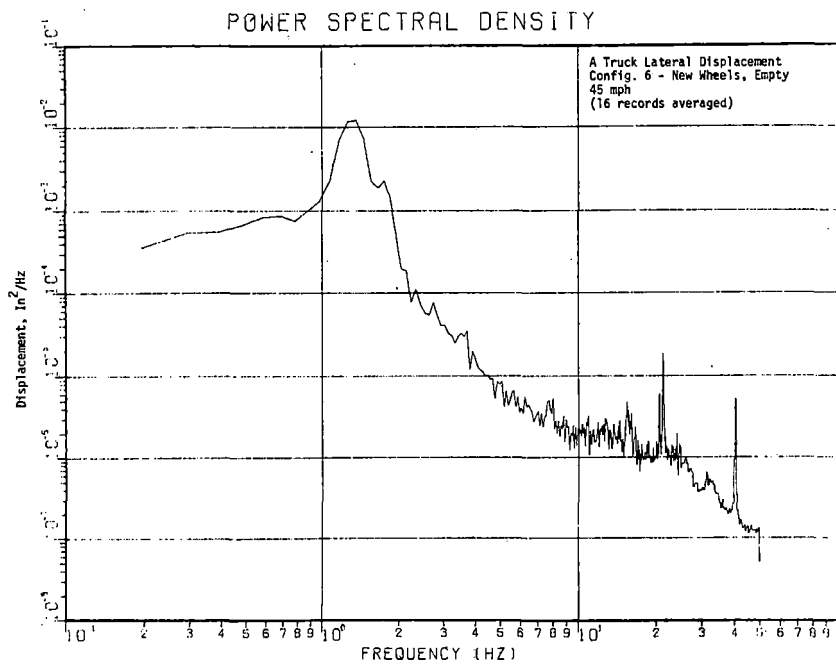


Figure 15. Typical Truck Lateral Displacement PSD

OVERVIEW OF FREIGHT SYSTEMS  
R&D REPORT NO FRA/ORD-77/58  
OCTOBER 1977

ERRATA

"Rail Dynamics Laboratory Requirements and Hardware Configurations"

Page 90 first sentence under Fig. 6, Vibration Test Unit should read as follows:

"The vertical excitation modules (each under independent servo control) are designed around a 60,000 lb (27,216 kg) hydraulic actuator, equipped with a 200 gpm ( $.0126 \text{ m}^3/\text{s}$ ) high performance servo-valve."

Page 90 first sentence of second major paragraph from bottom starting "The hydraulic flow demands ..." should be changed to read as follows:

"The hydraulic flow demands of the various excitation modules and hydrostatic bearing elements at peak excitation levels can be as high as 1000 gpm ( $.0631 \text{ m}^3/\text{s}$ ) @ 3,000 psi (20,684,271  $\text{N}/\text{m}^2$ ). This has been provided for via three 360 gpm ( $.0227 \text{ m}^3/\text{s}$ ) variable volume pumping systems each capable of delivering the rated flow at 3,000 psi (20,684,271  $\text{N}/\text{m}^2$ )."

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